# Piping and Pipeline Engineering

## Design, Construction, Maintenance, Integrity, and Repair

George A. Antaki

# Piping and Pipeline Engineering

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# Piping and Pipeline Engineering

**Design, Construction, Maintenance, Integrity, and Repair**

***George A.Antaki***
*Aiken, South Carolina, U.S.A.*

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# Preface

This book is written to present in sequence, and in a way that balances practice and theory, the fundamental principles in materials, design, fabrication, inspection, testing, operation, maintenance and integrity of plant piping systems and pipelines.

The book is intended for designers, construction engineers and inspectors, project engineers, system and maintenance engineers. It applies to plant piping systems and pipelines in power, utilities, waterworks, and chemical process industries, as well as oil and gas pipelines.

I wrote this book with two objectives in mind: first, to convey the big picture, the fundamental qualitative steps to any successful piping and pipeline activity, whether it is a routine maintenance work package, or a new multi-million dollar project. The second objective is to explain the quantitative details, calculations and techniques essential in supporting competent decisions.

Over the years, each industry has developed expertise and technology to resolve its unique challenges. Yet, the fundamental engineering concepts (materials, design, construction and integrity) are the same, and much is to be gained by understanding how different industries approach and resolve similar problems. That is why the book covers both piping systems and pipelines.

The first chapter explains the many codes, standards and regulations, essential in the work of industry. This first chapter, as does the rest of the book, focuses on the practice in the United States, in particular the American Society of Mechanical Engineers' B31 Code. But this practice is similar in many ways to other codes, standards and practices applied around the world.

Chapter 2 highlights the seven fundamental areas of competent piping and pipeline engineering. Presented in the format of a checklist, this chapter is a road map to successful piping and pipeline projects and operations.

Chapter 3 describes pipe and fitting material characteristics and properties that constitute the foundation of the design rules, construction methods, inspection and maintenance practices, and integrity analysis.

Chapters 4 to 12 address mechanical design and integrity, starting with common operating conditions (pressure, weight, temperature), progressing to occasional operating conditions (vibration, water-hammer, pressure transients), and concluding with extreme loading (high winds, earthquake, and explosions). Chapters 13 and 14 present the unique design aspects of sub-sea and underground pipelines.

Logically following design is shop fabrication and field erection. These are addressed in chapters 15 to 19, and include welding, mechanical joining, non-destructive examination, pressure and leak testing of fittings, components and whole systems.

The piping system or pipeline having been designed and constructed is now placed into service. While performing its function, the system starts to age and degrade. Chapter 20 covers the complex question of corrosion and degradation mechanisms in a practical manner to help field engineers understand, classify and diagnose the causes and effects of

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corrosion and degradation in service. Having recognized the inevitable degradation mechanisms at play during operation, comes what is perhaps the most critical decision of operations personnel, maintenance inspectors and field engineers: to determine the fitness for continued service of a degraded component or system. The knowledge gained in the previous chapters (codes, materials, design, fabrication, inspection, testing, corrosion) is used in chapter 21 to make fitness-for-service and run-or-repair decisions.

Chapter 22 covers maintenance and in-service inspection practices and techniques, including an introduction to failure analysis. Whether maintenance practices are regulated (for example in the nuclear power industry or oil and gas pipeline industries) or left to the discretion of operating companies, several fundamental maintenance strategies described in chapter 22 will apply. Chapter 23 describes a broad range of pipe and pipeline repair techniques, each with its advantages and shortcomings.

Chapter 24 covers the unique aspects of plastic pipe and fittings, and Chapter 25 is an introduction to valve selection and sizing.

George Antaki

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# 1
# Codes, Standards and Practices

## 1.1 A BRIEF HISTORY OF PIPING TECHNOLOGY

The art of design and construction of piping systems and pipelines dates back to the earliest civilizations. Its progress reflects the steady evolution of cultures around the world: the needs of developing agricultures, the growth of cities, the industrial revolution and the use of steam power, the discovery and use of oil, the improvements in steel making and welding technology, the discovery and use of plastics, the fast growth of the chemical and power industries, and the increasing need for reliable water, oil and gas pipelines.

### Mesopotamia

In the valley formed by the Tigris and Euphrates (present day Iraq), between 3000 BC and 2000 BC, rose the first city-states of Ur, Uruk and Babylon. In this land, which the Greeks called Mesopotamia ("between two rivers"), man established irrigated agriculture on a grander scale than ever seen before. Networks of irrigation channels were fed by river water. At the same time, aqueducts carried potable water from springs through miles of desert. To reduce losses by evaporation, the aqueducts were partly covered or run underground. Within cities, water was distributed in cylindrical pipes made of baked clay.

### China

At about the same time, and half a world away, the Chinese supplied water to their villages in bamboo pipes and used wooden plug valves to control flow. Bamboo wrapped with wax was also used to carry natural gas, while large water pipe conduits were made of hollow wood logs.

### Indus Valley

As early as 2500 BC, the sophistication of indoor plumbing and wastewater drainage was characteristic of the Indus Valley cities (present day Pakistan and north western India). Houses in Harrapa and Mohenjo-Darro made use of short earthenware pipes placed back-to-back to channel water. Interestingly, these short pipes appear to have been produced in standard sizes: approximately 1 ft long and 4" in diameter. Drainage ran in street trenches covered with flat rectangular stone slabs.

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## Egypt

In ancient Egypt, 3000 BC, canals were used to divert the Nile waters and irrigate fields. Drinking water was obtained directly from wells or by boiling river water. There are few reports of the use of pipes. In one instance, approximately 400 yards of copper pipes were found in the temple of Sahuri, assembled from 16" long sections made by hammering 1/16" thick sheets of copper into cylinders.

## Crete

On the island of Crete, between 2000 BC and 1500 BC, the Minoans had installed a clever water supply to the palace of Knossos (famous for the legend of the Minotaur, part man part bull, who haunted its labyrinths). Earthenware pipes carried water from nearby mountains to the palace. The pipes were slightly conical in shape, the narrow end of one pipe section fitting into the large end of the next section.

## Greece

The Greeks, 1600 BC to 300 BC, used earthenware, stone, bronze and lead pipes. In many cases one end of the pipe section was tapered, while the opposite end was expanded, the tapered end of one pipe fit into the expanded end of the next section, much like today's bell and spigot joints.

Greek blacksmiths "welded" pieces of iron by hammering red-hot ends together. There is however no evidence that this type of welding was used to fabricate pipe. Whatever the fabrication technique, the pipe joints must have been reliable since the hydraulic profile of one pipeline implies that static pressure due to differences in elevation must heave reached up to 300 psi at low points.

## Rome

The Romans deserve special mention in the field of piping engineering. Some of their achievements in water works remained unmatched until modern times. The Roman imperial period between 400 BC and 150 AD saw the building of over 200 stone aqueducts to carry waters to three separate outlets: public baths, city fountains and a few private homes. The fountains played the role of surge tanks in case of water hammer due to sudden changes in flow. The water supply of Rome itself is reported to have been around 300 gallons per person, a high figure, even by today's standards.

The control of Rome's water supply was entrusted to a commissioner, helped by technical consultants and an administrative staff. Countless slaves acted as masons, repairmen, and even quality inspectors. The Romans were proud of their waterworks. The Roman water commissioner Frontinus noted "With such an array of indispensable structures carrying so many waters, compare if you will the idle pyramids or the useless, though famous, works of Greeks".

A variety of pipe materials were used: lead, wood with iron collars at joints, earthenwear, bronze, and, in the more prestigious villas, silver. Lead pipes were fabricated by folding flat strips into conduits of circular, oblong or even triangular cross sections. The longitudinal seams were then soldered. The Romans perfected mixtures of cement or

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mortar to line the inside of pipelines. Another sealing technique consisted in throwing wood ash into the water to clog cracks and stop leaks.

The size of pipes was designated by the width of the initial strip, measured in "fingers". Pipes and inlet orifices to control flow were carefully inspected… and stamped. Rome's water regulations were clear: "none but stamped pipes must be set in place". For example, a section of lead pipe clearly shows the letters "therma triani" stamped in relief.

## Middle Ages

In Western civilization, the fall of Rome reversed the advances achieved in the science and art of piping and waterworks. Except for works by the Moors, waterworks were largely ignored in middle age Europe. Towns reverted back to wells, springs and rivers for water. As for wastewater, it was simply disposed into the streets. The exceptions appeared to have been certain abbeys that had well maintained metallic water and earthenware sewer networks. An example of color-coded flow diagram, a predecessor to modern day P&ID's (piping and instrumentation diagrams), has survived to our days. Hollowed trees were used to convey water; they were made watertight by a variety of means such as the use of sealant made of mutton fat mixed with crushed bricks.

## Renaissance

Interestingly, with the invention of the printing press, one of the first books printed in the fifteenth century was Frontinus' Roman treatise on waterworks. During that period of renewal, several aqueducts were repaired and placed back in service. At the same time, metallurgy had reached a point where cast iron pipe could be produced.

## The Age of Enlightenment

The waterworks of 17$^{th}$ and 18$^{th}$ century Europe are marked by advancements in pumping technology and the expanded use of cast iron pipe. Jealous of his minister's palace, the French "Sun King" Louis XIV ordered the building of 1400 fountains for his palace at Versailles. But the palace was situated on high grounds and the water had to be pumped uphill. The king entrusted the famous scientist Mariotte (1620–1684) to solve this problem. With a limitless budget, but on a tight schedule, Mariotte experimented with a number of pipe materials, including glass, before selecting cast iron and, in the process, perfecting the theory of strength of beams in bending. In England of the mid-18$^{th}$ century, the London Bridge Waterworks Company reported over 54,000 yards of wooden pipe and 1,800 yards of cast iron.

## The Industrial Revolution

In the 19$^{th}$ century, piping technology would develop at an accelerated pace. The catalysts of this growth were the emerging oil industry, the distribution of natural gas and the increasing need for steam and water. Wood was still in use, but lap-jointed wrought iron,

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riveted or flanged, was taking hold. The pipe flange was perfected by S.R.Dresser in the 1880's.

Gas lighting was introduced in London in 1807, with pipelines made from musket barrels available in great numbers at the end of the Napoleonic wars. In the U.S., the first gas transmission line was installed in Baltimore in 1816.

In 1825, the Englishman Cornelius Whitehouse developed a method for fabricating pipe in one furnace pass from hot strips formed through a die or bell. In the United States, the first pipe furnace was built in 1830, in Philadelphia. Between 1850 and 1860, the Bessemer process made quality steel available in large quantities, and triggered the production of pipe by cold bending of sheet metal and riveting the seams.

When, in Pennsylvania, E.L.Drake discovered oil in 1859, it was transported by wagons. In 1865, S.Van Syckel successfully piped oil over 6 miles from oil field to loading station. His pipeline consisted of 2" diameter, 15 ft long wrought iron lap welded pipe sections. This breakthrough was understandably opposed by the railway companies who prohibited pipelines from crossing their tracks. "Pipeline walkers" were hired by the oil companies to guard against sabotage and give early warning of leaks, an early version of today's air patrols and "in-service inspection" programs.

Towards the end of the 19$^{th}$ century seamless pipe made its appearance, having evolved from the manufacture of tubular bicycle frames, an industry fast growing at the time. In the second half of the 19$^{th}$ century, the use of steam was growing in transport (locomotives and steam boats), in city heating (through underground steam pipelines), and in industry. An 1883 "Note Relating to Water-Hammer in Steam Pipes" (reproduced in part in Chapter 9) shows how well engineers understood the flow of steam in pipes.

At the threshold of the 20$^{th}$ century, piping technology was poised for unprecedented growth due to improvements in welding, in materials and in pumping. At the same time, standardization of materials and designs became a financial and safety necessity, and industries came to rely more on codes and standards, while national engineering societies and industry institutes became an important source of innovation and improvements.

### Time Line

Key milestones in the development of piping and pipeline technologies are listed in Table 1-1.

**Table 1-1** Time Line of Piping Technology

| | |
|---|---|
| 3000 BC | Mesopotamia: Baked clay pipe used for water distribution. |
| 3000 BC | China: Bamboo pipes carry water or gas. |
| 3000 BC | Egypt: copper sheets hammered into cylinders used as water pipes. |
| 2500 BC | Indus valley: earthenware pipe of standard size for indoor plumbing. |
| 2000 BC | Crete: Tapered pipes made of earth, bronze and lead. |
| 1000 BC | Greece: Blacksmiths "weld" by hammering red hot metals together. |
| 1000 BC | Greece: Hydraulic profile points to pipes carrying 300 psi. |
| 400 BC | Rome: Lead, wood with iron collars, earthenware used to carry water. |

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400 BC Rome: Cylindrical, oblong and triangular pipe cross-sections used.
400 BC Rome: Only stamped pipes used in waterworks.
400 BC Rome: Pipe sizes standardized and labeled by width of initial strip.
400 BC Romans favorably compare their waterworks to “idle” pyramids.
500 The Middle Ages…
1601 Porta (Italy) designs a steam drum mounted atop a furnace.
1650 Mariotte designs piping system for 1400 fountains at Versailles.
1652 First U.S. water works (Boston).
1707 Papin (France) designs a steam engine counterweight relief device.
1738 Bernoulli publishes “Hydrodynamica”.
1774 James Watt (England) operates a steam engine, 18” in diameter.
1808 First steam boat, New York to Albany, 150 psi steam, 4 mph.
1812 Welding of firearm barrels (UK).
1815 Coal gas used to light London streets.
1815 Discarded musket barrels used as gas distribution pipe (UK).
1817 Philadelphia city council recommends safety valves on ship boilers.
1824 Patent for longitudinally welded pipe (UK).
1825 Fabrication of seamless tube (UK).
1830 Franklin Institute investigates steam boiler explosions.
1833 Steamboat 6-month inspections put into US law.
1836 First US wrought iron pipe mill (Philadelphia).
1850 Wöhler studies the endurance limit of metals.
1852 Steamboat act rules design and construction of boilers.
1854 Hartford steam boiler explosion. Jury calls for boiler regulation.
1859 First commercial oil well produces 20 barrels/day, Pennsylvania.
1862 First oil pipeline, 1000-ft long operates by gravity, Pennsylvania.
1862 Standard pipe thread dimensions.
1863 Second oil pipeline, 2” dia. cast iron, 2.5 miles long, pumped flow.
1864 Connecticut appoints steam boiler inspectors.
1865 Steamship Sultana explodes, killing 1500 returning prisoners of war.
1865 Oil transport pipeline 6 miles, 2” lap-welded iron pipe, tested 900 psi.
1866 Oil well gathering line, 2” pipe 4 miles.

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1867 First insurance policy for boilers.
1869 Development of celluloid plastic.
1877 Forge welding of iron boiler
1879 Oil pipeline, 109 miles, 6” diameter, Pennsylvania.
1880 Formation of the American Society of Mechanical Engineers.
1881 Formation of the American Water Works Institute.
1884 Standard Methods for Steam Boiler Trials.
1885 Henry Clay Mine disaster, 27 boilers explode and kill hundreds.
1885 Bauschinger measures small strains with mirror extensometer.
1886 Patent for Mannesman seamless pipe mill (Germany).
1886 Standard pipe and thread sizes recommended by ASME.
1886 Wood (1200 barrels) and wrought iron (15,000 b.) oil storage tanks.
1887 First patent for arc welding (England).
1887 Steel pipe, butt and lap welded (Wheeling, W.Va).
1889 Formation of American Steam Boilers Manufacturers Association.
1892 Arc welding used in locomotive factories.
1894 ASME adopts a standard flange template.
1895 Oil steel line pipe becomes available.
1896 NFPA founded.
1898 Burst tests of cast iron cylinders.
1901 A manufacturers’ standard is issued for flanges to 250 psi.
1901 Pipeline for batch refined oil products, Pennsylvania.
1903 Metallographic analysis of stages of fatigue failure.
1905 Steam explosion in a Brockton, Massachusetts, shoe factory, 58 dead.
1905 Charpy test developed to assess notch effects on toughness.
1906 Massachusetts forms a five-men Board of Boiler Rules.
1906 472 miles, 8” pipeline, threaded, Oklahoma to Texas.
1906 Beneficial effects of heat treatment discovered in Germany.
1908 Massachusetts enacts first boiler construction law.
1908 AWWA “Standard Specification” for cast iron pipe.
1908 First discovery of Middle Eastern oil (Persia).
1910 Manufacturers’ committee formed to design a line of flanged fittings.

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1911 Ohio adopts Massachusetts' law.
1911 Ten states and nineteen cities have boiler laws.
1911 First ASME committee for boilers and vessels specifications.
1911 Oxyacetylene welding replaces threads on gas pipeline.
1912 Lincoln Electric Institute introduces the welding machine to the U.S.
1912 Pipe screwing machine replaces "hand-tong gangs".
1913 Standard Oil begins thermal cracking oil to get gasoline.
1914 ASME publishes Standard for Pipe Flanges, Fittings and Bolting.
1915 ASME I Rules for the Construction of Stationary Boilers, 114 pages.
1917 Pump manufacturers form the Hydraulic Institute.
1919 AWS American Welding Society formed.
1920 Oxyacetylene torch welding replaces threaded connections.
1920 Welded seam pipe starts to replace riveted seam pipe.
1921 Publication of ASME III Code for Boilers for Locomotives.
1921 Union Carbide hydrocarbon cracking plant.
1921 Committee B16 organized.
1923 Publication of ASME IV Heating Boilers.
1924 Issue of API standards.
1924 Publication of ASME II Materials.
1925 Commercial fabrication of arc welded pressure vessels.
1925 Publication of ASME VIII Pressure Vessels.
1926 Publication of ASME VII Care of Power Boilers.
1926 First meeting of ASME "Project B31" Sectional Committee.
1926 Geckeler (Germany) publishes vessel head design formulas.
1928 First edition of API 5L specification for pipelines.
1928 Publication of first American Standards Association B16 Standard.
1928 Work begins under B16 to standardize dimensions of valves.
1928 Electric arc welding of 40-ft sections of seamless oil line pipe.
1929 Sokolow (Russia) applies ultrasonic waves to measure wall thickness.
1930 Electric arc welding.
1930 Development of expanded line pipe, with increased yield.
1931 Fusion welding permitted as joining practice in the ASME Code.
1931 X-ray radiography introduced in the ASME Code, 4" thickness limit.

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1931 ASME introduces weld porosity charts.
1931 Production of PVC pipe in Germany.
1932 Timoshenko publishes external pressure formulas.
1932 Discovery of oil in Bahrain.
1933 Imperial Chemical Industries develops polyethylene.
1934 Joint API-ASME Committee Unfired Pressure Vessels.
1935 Roark publishes stresses in cylinders under concentrated radial load.
1935 ASME B31 “Power, Gas and Air, Oil, District Heating”.
1935 Iron pipe sizes modified for steel, lower wall thickness, same weight.
1936 First publication of ANSI B36.10 carbon steel pipe sizes.
1937 Work begun to standardize welded fittings, today’s B16.9.
1938 Discovery of oil in Saudi Arabia.
1938 Dupont develops Teflon.
1939 Construction of 96-mile 24/26 in. Pto. La Cruz pipeline, Venezuela.
1940 Scale model tests used to design steam lines for flexibility.
1940 Submerged arc welding developed in shipyards.
1941 Welding and brazing qualification.
1941 First offshore oil well, Texas.
1942 ASME B31 “American Standard Code for Power Piping”.
1942 Molybdenum added to prevent graphitization of steam steel pipe.
1943 TD Williamson launches first steel pig to remove paraffin deposits.
1944 Vessel design safety factor changed from 5 to 4.
1945 Miner publishes “Cumulative Damage in Fatigue”.
1946 ASA standard for socket welded fittings, today’s B16.11.
1946 National Board Inspection Code.
1946 Vessel design safety factor returned to 5 at end of war.
1947 Angle beam ultrasonic waves used to inspect welds.
1947 First offshore platform out of sight of land.
1947 Products batching pipeline, Texas to Colorado.
1949 B36.19 standard sizes of stainless steel pipe, down to schedule 10S.
1950 Trans-Arabian pipeline 30/31 in. Saudi Arabia to Syria.
1951 First publication of standard gasket dimensions B16.21.

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1951 Vessel design safety factor permanently returned to 4.
1952 B31.1.8 “Gas Transmission and Distribution Piping Systems”.
1952 Glass reinforced plastic pipe comes into production.
1952 15 ft-lb adopted as an acceptable lower bound of impact toughness.
1952 Introduction of schedule 5S for stainless steel pipe in B36.19.
1953 Drop-weight test used as a measure of nil ductility transition.
1953 First edition of API 1104 for pipeline weld inspections.
1955 ASME B31 code splits into separate books.
1955 Markl’s thermal expansion formula introduced in B31.1.
1955 ASTM organizes group to write plastic pipe standards.
1956 Closed form solution for ship piping under dynamic load.
1956 Kellog publishes “Design of Piping Systems”.
1958 Advisory Committee on Nuclear Plant Piping.
1959 Publication of B31.3 “Petroleum Refinery Piping”.
1959 Publication of B31.4 “Oil Transportation Piping Systems”.
1961 Publication of ASME X Fiber Reinforced Plastic Vessels.
1961 Langer publishes design fatigue curves for vessels.
1962 Post-weld heat treatment introduced in the ASME code.
1962 Publication of B31.5 Refrigeration Piping.
1962 Publication of first ASME Code Case N-1 for Nuclear Piping.
1962 First commercial reeled-pipe vessel for laying subsea pipe.
1965 ASME III Locomotives code replaced by ASME III Nuclear Vessels.
1966 Publication of ASME B31.7 Nuclear Piping.
1967 ASA becomes US American Standards Institute USAS.
1967 Occasional loads appear in B31.1 with a 1.28 allowable.
1967 Fracture mechanics introduced in vessel design and failure analysis.
1968 Publication of 49CFR192 federal safety rules for pipelines.
1969 USAS becomes American National Standards Institute ANSI.
1969 Publication of B31.7 Code for Nuclear Piping.
1970 B31 Case 70 “Normal, Upset, Emergency and Faulted” conditions.
1970 Publication of ASME XI In-service Inspection Nuclear Components.
1970 Publication of ASME III Nuclear Components.

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1970 Investigation of the strength of corroded pipe (later B31.G).

1971 B31.7 moved to ASME III.

1971 Publication of ASME V Non-Destructive Examination.

1971 Publication of ASME VI Care and Operation of Heating Boilers.

1973 Publication of rules to evaluate the strength of corroded pipelines.

1973 ASME Code Case 1606 introduces 2.4S allowable.

1974 B31.6 Chemical Plant Piping (not issued) to B31.3 (Code Case 49).

1977 Initial service of 48" North Slope oil pipeline, Alaska.

1982 ANS Committee B16 becomes ASME Committee.

1982 Publication of ASME B31.9 Building Services Piping.

1984 Creation of the Edison Welding Institute, Ohio.

1986 Publication of ASME B31.11 Slurry Transportation Piping.

1990 US interstate pipelines: $274{,}000^{+}$ miles gas, $168{,}000^{+}$ miles liquid.

1993 First use of API 5L X80 line pipe (Germany).

1995 NBIC expands scope to cover "pressure retaining items".

1996 B31.3 "Chemical and Refinery" becomes "Process Piping".

1996 Accountable pipeline safety act.

1999 Publication of ASME XII Transport Tanks.

2000 Publication of API 579 Fitness-for-Service.

2000 Pipelines integrity management plan introduced in 49CFR

2000 4,400 companies have ASME accreditation, 74% in U.S.-Canada.

## 1.2 NATIONAL CODES, STANDARDS AND GUIDES

In the United States, there are many organizations that develop and publish standards, guides and rules of engineering practice. These organizations can be grouped into four main categories [Leight].

(1) Professional societies, such as the American Society of Mechanical Engineers (ASME) or the American Society of Civil Engineers (ASCE), publish design, construction and maintenance standards and guides that reflect the state-of-the-art in their profession. These standards may be imposed by federal, state or local law, in which case they become codes. This is the case for example for Section I, Power Boilers, of the ASME Boiler and Pressure Vessel Code, which is imposed by state law in most states in the U.S. Other professional societies include the American Institute of Chemical Engineers (AIChE), the American Institute of Steel Construction (AISC), the American

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Concrete Institute (ACI), ASM International (formerly American Society for Metals), and the Materials Technology Institute of the Chemical Process Industries (MTI).

(2) Trade associations that write standards to promote, perfect and explain the use of products developed by their members, for example the Nickel Development Institute (NiDI), the American Iron and Steel Institute (AISI), the American Petroleum Institute (API), and the American Water Works Association (AWWA).

(3) Testing and certification organizations such as Underwriters Laboratories (UL), Factory Mutual (FM) and the International Conference of Building Officials' Evaluation Services (ICBO ES), that independently test and certify equipment, components and items.

(4) Standards developing organizations such as ASTM International (formerly the American Society for Testing and Materials), whose primary purpose is the writing and issue of standards to improve reliability, promote public health and commerce.

Following is a list of professional societies, trade associations, testing and certification organizations, research institutes, regulatory bodies, and standards developing organizations whose work relates to the design, fabrication, operation, maintenance, repair and safety of pressure equipment, piping systems and their support structures.

| | |
|---|---|
| AA | —Aluminum Association, Washington, DC. |
| AASHTO | —American Association of State Highway and Transp. Off., DC. |
| ABMA | —American Boiler Manufacturers Association, Arlington, VA. |
| ACS | —American Chemical Society, Washington, DC. |
| ACI | —American Concrete Institute, Detroit, MI. |
| ACPA | —American Concrete Pipe Association, Irving, TX. |
| AGA | —American Gas Association, Arlington, VA. |
| AIChE | —American Institute of Chemical Engineers, New York. |
| AIPE | —American Institute of Plant Engineers, Cincinnati, OH. |
| AISC | —American Institute of Steel Construction, Chicago, IL. |
| AISI | —American Iron and Steel Institute, Washington, DC. |
| ANSI | —American National Standards Institute, New York, NY. |
| ANS | —American Nuclear Society, La Grange Park, IL. |
| API | —American Petroleum Institute, Washington, DC. |
| APFA | —American Pipe Fittings Association, Springfield, VA. |
| AREA | —American Railway Engineering Association, Washington, DC. |
| ASCE | —American Society of Civil Engineers, Reston, VA. |
| ASHRAE | —American Society of Heating, Refrig. and Air Cond. Engrs, Atlanta. |
| ASME | —American Society of Mechanical Engineers, New York, NY. |

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|---|---|
| ASNT | —American Society for Non-Destructive Testing, Columbus, OH. |
| ASPE | —American Society of Plumbing Engineers, Westlake, CA. |
| ASQC | —American Society for Quality Control, Milwaukee, WI. |
| ASTM International | —West Conshohocken, PA. |
| AWS | —American Welding Society, Miami, FL. |
| AWWA | —American Water Works Association, Denver, CO. |
| | Batelle Memorial Institute, Columbus, OH. |
| BOCA | —Building Officials & Code Admin. International, Country Club Hills, IL. |
| CABO | —Council of American Building Officials, Falls Church, VA. |
| CMA | —Chemical Manufacturers Association, Washington, DC. |
| CDA | —Copper Development Association, Greenwich, CT. |
| CAGI | —Compressed Air and Gas Institute, Cleveland, OH. |
| CGA | —Compressed Gas Association, Arlington, VA. |
| CISPI | —Cast Iron Soil Pipe Institute, Chattanooga, TN. |
| | Cryogenic Society of America, Oak Park, IL. |
| CSA | —Construction Specifications Institute, Alexandria, VA. |
| DIRA | —Ductile Iron Research Association, Birmingham, AL. |
| EEI | —Edison Electric Institute, Washington, DC. |
| EJMA | —Expansion Joint Manufacturers Association, Tarrytown, NY. |
| EMC | —Equipment Maintenance Council, Lewisville, TX. |
| EPRI | —Electric Power Research Institute, Palo Alto, CA. |
| EWI | —Edison Welding Institute, Columbus, OH. |
| FIA | —Forging Industry Association, Cleveland, OH. |
| FM | —Factory Mutual, Norwood, MA. |
| HI | —Hydraulic Institute, Parsippany, NJ. |
| IAMPO | —International Assoc. of Mech. and Plumbing Off., South Walnut, CA. |
| ICBO | —International Conference of Building Officials, Whittier, CA. |
| ICRA | —International Compressors Remanufacturers Assoc., Kansas City, MO. |
| IEEE | —Institute of Electrical and Electronic Engineers, New York, NY. |
| | Institute of Industrial Engineers, Atlanta, GA. |
| ISA | —Instrument Society of America, Research Triangle, NC. |

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|---|---|
| MCA | —Manufacturing Chemical Association, Washington, DC. |
| MSS | —Manufacturers Stand. Society of Valves and Fittings Industry, Vienna, VA. |
| NACE | —National Association of Corrosion Engineers, Houston, TX. |
| | National Board of Boiler and Pressure Vessel Inspectors, Columbus, OH. |
| | National Certified Pipe Welding Bureau, Bethesda, MD. |
| | National Corrugated Steel Pipe Association, Washington, DC. |
| NCPI | —National Clay Pipe Institute, Lake Geneva, WI. |
| NEMA | —National Electrical Manufacturers Association, Washington, DC. |
| NFPA | —National Fire Protection Association, Quincy, MA. |
| NFSA | —National Fire Sprinklers Association, Patterson, NY. |
| NiDI | —Nickel Development Institute, Toronto, Canada. |
| NIST | —National Institute of Standards and Technology, Gaithersburg, MD. |
| NRC | —Nuclear Regulatory Commission, Washington, DC. |
| NTIAC | —Nondestructive Testing Information Analysis Center, Austin, TX. |
| OSHA | —Occupational Safety and Health Administration, Washington, DC. |
| PEI | —Petroleum Equipment Institute, Tulsa, OK. |
| PFI | —Pipe Fabricators Institute, Springdale, PA. |
| PLCA | —Pipe Line Contractors Association, Dallas, TX. |
| PPFA | —Plastic Pipe and Fittings Association, Glen Ellyn, IL. |
| PMI | —Plumbing Manufacturers Institute, Glen Ellyn, IL. |
| PPI | —Plastics Pipe Institute, Washington, DC. |
| RETA | —Refrigeration Engineers and Technicians Association, Chicago, IL. |
| RRF | —Refrigeration Research Foundation, North Bethesda, MD. |
| SBCCI | —Southern Building Code Congress International, Birmingham, AL. |
| SES | —Standards Engineering Society, Dayton, OH. |
| SFPE | —Society of Fire Protection Engineers, Boston, MA. |
| SME | —Society of Manufacturing Engineers, Dearborn, MI. |
| SPE | —Society of Petroleum Engineers, Richardson, TX. |
| SPE | —Society of Plastics Engineers, Fairfield, CT. |

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| SSFI | —Scaffolding, Shoring and Forming Institute, Cleveland, OH. |
| SSPC | —Steel Structures Painting Council, Pittsburgh, PA. |
| SMACNA | —Sheet Metal and Air Cond'g. Contr. National Assoc., Merrifield, VA. |
| STI | —Steel Tank Institute, Northbrook, IL. |
| SWRI | —Southwest Research Institute, San Antonio, TX. |
| TEMA | —Tubular Exchanger Manufacturers Association, Tarrytown, NY. |
| TIMA | —Thermal Insulation Manufacturers Association, Mt. Kisco, NY. |
| TWI | —The Welding Institute, Cambridge, UK. |
| UL | —Underwriters Laboratories, Northbrook, IL. |
| UNI | —Uni-Bell PVC Pipe Association, Dallas, TX. |
| VMAA | —Valve Manufacturers Association of America, Washington, DC. |
| | Vibration Institute, Willowbrook, IL. |
| | Zinc Institute, New York, NY. |

The American National Standards Institute (ANSI) is a federation of standards writing bodies, government agencies, companies and consumers that coordinates the activities of standard writing organizations, and offers accreditation to standards writing organizations and product certifiers, including regular audits. As part of the accreditation process, ANSI requires standards writing organizations to follow a consensus process by which new standards or revisions are reviewed and approved by majority of the technical standards writing body (some standards committees have adopted a 2/3 rather than a 51% majority rule), a supervisory board (such as the ASME Boards listed in section 1.7), the public, and a final review by the ANSI Board of Standards Review. The standards writing rules provide for an appeals process at various levels, including appeal to ANSI itself. American national standards are normally reaffirmed or revised every five years. ANSI may administratively withdraw a standard that has not been reaffirmed or revised within ten years. ANSI is also the U.S. representative on the International Standards Organization (ISO).

At times, government agencies also write their own standards. However, starting in the 1990's, there has been a concerted effort by U.S. federal departments and agencies to use national consensus standards where they exist. This effort was formalized in the National Technology Transfer and Advancement Act of 1995, Public Law 104–113, section 12.

## 1.3 PIPING AND PIPELINE CODES

In the United States, the "family" of documents that govern the design and construction of pressure piping is the ASME B31 pressure piping code. The term "pressure piping" refers to piping systems or pipelines operating at or above 15 psig, one atmosphere above

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the atmospheric pressure. Piping systems operating below atmospheric pressure, all the way down to vacuum, are also included in the scope of several ASME B31 sections. The ASME B31 code consists of several "sections", each covered in a separate "book". The individual code sections are numbered ASME B31.X, and each separate book is sometimes referred to as a "code". "B31" is simply a sequence number assigned to the project kicked-off in 1927 to develop pipe design rules. And the number ".1", ".3", etc. that follows "B31" reflected initially the original chapter numbers of ASME B31, which have now evolved into separate code books. These are:

ASME B31.1 Power Piping: fossil fueled power plant, nuclear powered plant with a construction permit pre-dating 1969 (B31.7 for 1969–1971, and ASME III post-1971).

ASME B31.2 Fuel Gas Piping (obsolete).

ASME B31.3 Process Piping: hydrocarbons and others. Hydrocarbons includes refining and petrochemicals. Others includes chemical process, making of chemical products, pulp and paper, pharmaceuticals, dye and colorings, food processing, laboratories, offshore platform separation of oil and gas, etc.

ASME B31.4 Liquid Petroleum Transportation Piping: upstream liquid gathering lines and tank farms, downstream transport and distribution of hazardous liquids (refined products, liquid fuels, carbon dioxide).

ASME B31.5 Refrigeration Piping: heating ventilation an air conditioning in industrial applications.

ASME B31.6 Chemical Plant Piping (transferred to B31.3)

ASME B31.7 Nuclear Power Plant Piping (transferred to ASME III)

ASME B31.8 Gas Transmission and Distribution Piping: upstream gathering lines, onshore and offshore, downstream transport pipelines and distribution piping.

ASME B31.9 Building Services Piping: low pressure steam and water distribution.

ASME B31.10 Cryogenic Piping (transferred to B31.3)

ASME B31.11 Slurry Transportation Piping: mining, slurries, suspended solids transport, etc.

There are also two separate ASME B31 publications: ASME B31G Manual for Determining the Remaining Strength of Corroded Pipe, and ASME B31.8S Managing System Integrity of Gas Pipelines

The code for design and construction of nuclear power plant piping systems is the ASME Boiler & Pressure Vessel Code, Section III, while their maintenance, in-service inspection and repair is covered in Section XI.

Waterworks codes cover transport, treatment and distribution of fresh water, and collection, treatment and effluent of used water. They include AWWA C151 (ductile iron), AWWA C200 series and M11 (steel), AWWA C300 series and M9 (concrete), AWWA C900 series and M23 (plastics), AWWA M45 (fiberglass), etc.

Fire protection codes cover transport and distribution of water for fire fighting, and sprinkler systems (National Fire Protection codes).

Building plumbing codes apply to commercial and private distribution and use of water and effluents (International Building Code).

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## 1.4 SCOPE OF ASME B31 CODES

Each ASME B31 Code section is published as a separate book. Some code sections apply to a specific industry, for example in its current scope ASME B31.1 applies to power plants or steam producing plants fired by fossil fuels (non-nuclear). ASME B31.4 applies to liquid hydrocarbon transportation pipelines, associated tank farms and terminals. ASME B31.8 applies to gas and two phase gathering lines, separators, transmission pipelines and associated compressors, and gas distribution piping. ASME B31.9 applies to building services, typically air and steam. On the other hand, ASME B31.3 is a code of very broad application, including chemical, petrochemical, pharmaceutical, utilities in process plants, support systems in pipeline terminals and pumping stations, process of radioactive or toxic materials, food and drug industry, paper mills, etc. Under certain conditions, an ASME B31 code may permit the owner to exclude some systems from code scope. In some cases such exclusions may however not be permitted under federal, state or local regulations.

The ASME B31 codes provide minimum requirements. They do not replace competence and experience. The owner, or the contractor, is expected to apply his or her knowledge to supplement the code requirements for a particular application. For example, when systems operate at temperatures that are atypically low or high, the owner or the designer may need to impose additional design and fabrication requirements. This is the case, for example, for sections of gas or oil pipelines at temperatures below −20°F or above 250°F.

## 1.5 BOILER AND PRESSURE VESSEL CODE

In the United States, the family of ASME Boiler and Pressure Vessel codes, ASME B&PV, governs the design and construction of pressure vessels. The term pressure vessel refers to vessels operating at or above 15 psig, one atmosphere above the atmospheric pressure, or subject to external pressure. In addition to design and construction, the ASME B&PV codes also address material specifications and properties (ASME B&PV II), examination and leak testing techniques (ASME B&PV V), and maintenance and repair (ASME B&PV VI, VII, XI). Components designed and fabricated according to the ASME B&PV Code are stamped to indicate compliance. Following is a partial description of scope of the ASME Boiler & Pressure Vessel Code sections.

The ASME B&PV Code, Section I “Power Boilers”, applies to boilers in which steam or other vapor is generated at a pressure of more than 15 psig; high-temperature water boilers intended for operation at pressures exceeding 160 psig and/or temperatures exceeding 250°F. Components that comply with ASME B&PV Section I are stamped S=boiler, PP=pressure piping, E=electric boilers, M=miniature boilers, V=boiler safety valve.

The ASME B&PV Code, Section II “Materials” compiles the material specifications and material properties for materials used in the construction of ASME components. If a material is listed in ASME Section II, its ASTM specification number is preceded by the letter “S”. For example the designation SA106 applies to an ASTM A106 pipe material

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"listed" in ASME Section II, permitted for use in the construction of ASME boilers and pressure vessels.

The ASME B&PV Code, Section III Division 1 applies to safety related components of nuclear power plants: vessels, piping, tanks, pumps and valves. The applicable stamps are: N for vessels, NP for piping, and NPT for components. The non-nuclear piping, or "balance of plant piping" is typically designed and fabricated to ASME B31.1. Piping systems in earlier nuclear power plants, licensed before 1971, are designed and constructed to ASME B31.1 or B31.7. ASME III Division 2 applies to the containment building of a nuclear power plant, and Division 3 applies to shipping containers for nuclear materials.

The ASME B&PV Code, Section IV Heating Boilers applies to hot water supply boilers, with the following services: steam boilers for operation at pressures not exceeding 15 psi; hot water heating boilers and hot water supply boilers for operating at pressures not exceeding 160 psi or temperatures not exceeding 250°F. Water heaters are exempted when their heat input is less than 200,000 But/hr, and their water temperature is less than 210°F, and their water capacity is less than 120 gal.

The ASME B&PV Code, Section V addresses the various techniques for non-destructive examinations (NDE) and testing (NDT), such as visual examinations, liquid penetrant testing, magnetic particles testing, radiography, ultrasonic inspections, pressure testing (hydrostatic or pneumatic), and leak testing.

The ASME B&PV Code, Section VI contains the "Recommended Rules for the Care and Operation of Heating Boilers", while Section VII contains the "Recommended Guidelines for the Care of Power Boilers".

The ASME B&PV Code, Section VIII "Pressure Vessels" addresses the design and fabrication of "unfired" pressure vessels (as opposed to "fired" boilers). These vessels are stamped "U" to signify "unfired". The following classes of vessels are exempted from the scope of Section VIII Division 1: those within the scope of other sections (for example a Section X fiberglass vessel); fired process tubular heaters; pressure containers which are part of components of rotating or reciprocating mechanical devices (for example pump or compressor casings); piping systems, pipelines, and their components (for example a valve body). Also excluded from the scope of Section VIII are vessels for containing water under pressure, up to 300 psi, 210°F, and 200,000 Btu/hr, or 120 gal; vessels having an internal or external operating pressure not exceeding 15 psi, with no limitation on size; vessels having an inside diameter, width, height, or cross section diagonal not exceeding 6 in., with no limitation on length of vessel or pressure; and pressure vessels for human occupancy.

Division 2 of ASME VIII addresses the design and construction of unfired pressure vessels, but it relies on more detailed analyses and more fabrication constraints than Division 1, while allowing a lower safety factor. Division 3 of ASME VIII addresses thick vessels for high-pressure service. The applicable stamps for ASME B&PV Code Section VIII are: U=Div.1 pressure vessel, U2=Div.2 pressure vessel, U3=Div.3, UM=miniature vessel and UV=safety valves

The ASME B&PV Code, Section IX addresses "Welding and Brazing Qualification", including welder and weld procedure qualification.

The ASME B&PV Code, Section X addresses the design and fabrication of fiber reinforced pressure vessels for general service. It sets minimum requirements for the

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materials of fabrication; test procedures for mechanical properties of laminates, and design rules.

The ASME B&PV Code, Section XI "Rules for In-service Inspection of Nuclear Power Plants" applies to periodic inspections of nuclear power plant components as well as to the evaluation of degraded conditions, and their repair.

The ASME B&PV Code, Section XII is a recent document that covers the design and fabrication of transport pressure vessels.

An ASME Post Construction Code is under development that will include rules and guidance for inspection planning of pressure equipment, methods for flaw assessment, and techniques for repair and testing of pressure equipment.

## 1.6 FEDERAL AND STATE LAWS

In the United States, in most cases, a national standard is imposed as a code by federal, state or local laws. Federal laws that address pressure vessels and piping systems include:

10 CFR Energy, Part 50 Domestic Licensing of Production and Utilization Facilities (regulatory requirements for nuclear power plants structures, systems and components, applicability of the ASME Boiler and Pressure Vessel code).

29 CFR Labor, Part 1910 Occupational Safety and Health Standards (mechanical integrity, inspection and testing, management of change, certification of coded vessels, ASME compliance for air receivers, lockout and tagout of energy sources, hot tap).

40 CFR Protection of Environment, Part 264 Standards for Owners and Operators of Hazardous Waste Treatment, Storage, and Disposal Facilities (tank systems, leak tightness, overpressure protection, double isolation).

49 CFR Transportation, Part 192 Transportation of Natural Gas and Other Gas by Pipeline: Minimum Federal Safety (gas pipelines, ASME B31.8). Part 193 Liquefied Natural Gas Facilities: Federal Safety Standards. Part 194 Response Plans for Onshore Oil Pipelines. Part 195 Transportation of Hazardous Liquids Pipelines (liquid pipelines, ASME B31.4).

State laws addressing the application of the ASME Boiler & Pressure Vessel Code are summarized in Table 1-2, which inevitably oversimplifies complex state laws and regulations. Note that, at the time of this writing, all but one of the fifty states had boiler laws (ASME B&PV Section I), and several states had pressure vessel laws (ASME B&PV VIII). Generally, these laws do not apply to federal facilities, where the responsible federal department imposes its own vessels and pressure safety requirements. For example, the U.S. Department of Energy requires compliance with the ASME Codes (B31 and B&PV) through a U.S. Department of Energy Order.

State laws for boilers and pressure vessels are quite detailed. For simplicity, Table 1-2 lists the exceptions to code compliance permitted by state laws [API 910]. For example, if "Vessels" is listed, this means that, in that state, pressure vessels do not need to comply with Section VIII of the ASME B&PV code, while "<5 $ft^3$" means that vessels smaller than 5 $ft^3$ do not have to comply with the ASME code. The actual state law should be consulted for a complete and updated understanding of its scope.

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## **Table 1-2** Simplified Summary of State Exclusions of ASME I and ASME VIII

| | |
|---|---|
| Alabama | Law under consideration, 1999. |
| Alaska | <15 psi not in place of public assembly<br><5 $ft^3$ |
| Arizona | Indian reservations<br>Vessels |
| Arkansas | Water heater <200,000 BTU/hr<br>Air <12 gal or 150 psig<br><15 psig and 5 $ft^3$ and 6" ID |
| California | Air tanks <150 psi and 1.5 $ft^3$ (must have relief valve) |
| Colorado | Waiver of rules for PV on remote sites (variance request) |
| Connecticut | Vessels |
| Delaware | No exception |
| Florida | No exception |
| Georgia | <5 $ft^3$ and <250 psig<br><1.5 $ft^3$ and 600 psig<br>water <300 psi or <210°F<br>water <200,000 BTU/hr or <210°F or <120 gal; with relief<br>"Most" research vessels<br>Vessel used in generation of electricity or in public utilities<br>Vessel used to generate steam; with owner-user program |
| Hawaii | Liquids <120 gal<br><5 $ft^3$ and <250 psig<br><1.5 $ft^3$ and 600 psig<br>cold water storage |
| Idaho | Water <120 gal<br><5 $ft^3$ and <250 psig<br><1.5 $ft^3$ and 600 psig<br><120 gal+200F+200,000 BTU/hr and 160 psi |
| Illinois | Cities >500,000 pop.<br>Water <180F<br>Farms for agricultural purposes<br><5 $ft^3$ and 250 psig<br><1.5 $ft^3$<br><15 $ft^3$ and <250 psig not in place of public assembly |
| Indiana | Water <180F <5 $ft^3$ (prior 1971)<br><15 $ft^3$ not in place of public assembly (prior 1971)<br><5 $ft^3$ with 250 psi RV (post 1971)<br><15 $ft^3$ with 300 psi RV not in public place (post 1971)<br><1.5 $ft^3$ |

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| | |
|---|---|
| Iowa | Vessels, except steam |
| Kansas | Vessels |
| Kentucky | Refineries |
| Louisiana | Vessels |
| Maine | No exception |
| Maryland | No exception |
| Massachusetts | Vessels other than air tanks, reinf. plastic, refrigerant. |
| Michigan | Vessels |
| Minnesota | <5 $ft^3$ with 100 psig RV<br>water <120 gal<br>Farms<br>Refineries<br><5 $ft^3$ steam laundry pressing<br>Non-hazardous liquids <140F or 200 psi |
| Mississippi | <5 $ft^3$ and <250 psig<br><1.5 $ft^3$ and <600 psig<br><120 gal<br>Well head site |
| Missouri | <15 $ft^3$ and <250 psi not in place of public assembly<br><5 $ft^3$ and <250 psi in public place<br><1.5 $ft^3$<br>Water <120 gal<br>Water <120F and <150 psig and non-hazardous<br>Non-explosive<br>Farms<br>Steam coil vapor cleaners <6 gal and <350F and RV |
| Montana | Vessels |
| Nebraska | Vessels |
| Nevada | <120 gal<br><5 $ft^3$ and <250 psig<br><1.5 $ft^3$ and <600 psig |
| New Hampshire | <5 $ft^3$ and <250 psig<br><1.5 $ft^3$ and <3000 psig<br>Water <125 psig<br>Domestic water |
| New Jersey | No exception |
| New Mexico | Vessels |
| New York | No exception |
| North Carolina | Drilling gas and other products<br>Agricultural use<br><5 $ft^3$ and <250 psig |

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| | |
|---|---|
| | <1.5 $ft^3$ and <600 psig<br>Water <120 gal, at ambient temperature<br>Water <110F<br>Construction req'ts do not apply to certain PV pre'81 |
| North Dakota | Water <200,000 BTU/hr and <160 psi and <250F<br>Portable steam cleaner |
| Ohio | No exception<br>Ohio "Pressure Piping Systems Code" |
| Oklahoma | Water <120 gal<br>Remote gas or oil production<br>Research vessels<br>Hot water supply heaters |
| Oregon | <5 $ft^3$ with 150 psi RV<br>Water <120 gal |
| Pennsylvania | No exception |
| Puerto Rico | No exception |
| Rhode Island | Remote oil or gas production |
| South Carolina | Law under consideration. |
| South Dakota | Vessels |
| Tennessee | <5 $ft^3$<br>Water <200,000 BTU/hr and 210°F |
| Texas | Vessels |
| Utah | No exception |
| Vermont | No exception |
| Virginia | No exception |
| Washington | <5 $ft^3$ |
| West Virginia | Pr. Vessels |
| Wisconsin | No exception |
| Wyoming | "Certain exceptions" (refers to regulations) |

## 1.7 ASME COUNCIL ON CODES AND STANDARDS

The activities leading to the development of the ASME codes and standards take place within the framework of the ASME Council on Codes and Standards. The council is comprised of several boards:

The Board on Pressure Technology

B16 Standardization of Valves, Flanges, Fittings and Gaskets.

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B31 Code for Pressure Piping.

Boilers and Pressure Vessels.

Post Construction.

The Board on Nuclear Codes and Standards

Boilers and Pressure Vessels (Nuclear).

Nuclear Air and Gas Treatment Equipment.

Cranes for Nuclear Facilities.

Nuclear Quality Assurance.

Operation and Maintenance of Nuclear Power Plants.

Qualification of Mechanical Equipment.

The Board on Safety Codes and Standards

Elevators and Escalators.

Manlifts.

Compressor Systems.

Cranes, Derricks, Hoists.

Automotive Lifting Devices.

The Board on Standardization

Plumbing Materials and Equipment.

Screw Threads.

Standardization of Bolts, Nuts, Rivets and Screws.

Surface Qualities.

Practice for Preparation of Graphs and Charts.

The Board on Accreditation, Registration, and Certification

Authorized Inspection Agencies.

Boilers and Pressure Vessels.

ISO 9000 Registration.

Pressure Relief Device Laboratories.

Qualification of Elevator Inspectors.

The Board on Performance Test Codes

Steam Turbines.

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Pumps.

Fans.

Pressure and Flow Measurement.

Steam Traps.

## 1.8 ASME B16 STANDARDS

Sectional Committee B16 was created in 1921 to unify national standards for pipe flanges and fittings, and to facilitate the procurement, assembly and replacement of plumbing and industrial pipefittings. This scope later expanded to cover valves and gaskets, and currently includes the following standards:

B16.1 Cast iron pipe flanges and flanged fittings

B16.3 Malleable iron threaded fittings

B16.4 Cast iron threaded fittings

B16.5 Pipe flanges and flanged fittings, NPS ½ through NPS 24

B16.9 Factory-made wrought steel butt welding fittings

B16.10 Face-to-face and end-to-end dimensions of valves

B16.11 Socket-welding and threaded forged steel fittings

B16.12 Cast iron threaded drainage fittings

B16.14 Ferrous pipe plugs, bushings and locknuts with pipe threads

B16.15 Cast bronze threaded fittings classes 125 and 250

B16.20 Metallic gaskets for pipe flanges—ring joint, spiral-wound, and jacketed

B16.21 Nonmetallic flat gaskets for pipe flanges

B16.22 Wrought copper and copper alloy solder joint pressure fittings

B16.23 Cast copper alloy solder joint drainage fittings

B16.24 Cast copper alloy pipe flanges and flanged fittings: Class 150, 300, 400, 600, 900, and 2500

B16.25 Buttwelding ends

B16.26 Cast copper alloy fittings for flared copper tubes

B16.28 Wrought steel buttwelding short radius elbows and returns

B16.29 Wrought copper and wrought copper alloy solder joint drainage fittings—DWW

B16.32 Cast copper alloy joint fittings for solvent drainage systems

B16.33 Manually operated metallic gas valves for use in gas piping systems up to 125 psig, sizes ½ through 2

B16.34 Valves—flanged, threaded and welding end

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B16.36 Orifice flanges

B16.38 Large metallic valves for gas distribution, manually operated, NSP-2 ½ to 12, 125 psig maximum

B16.39 Malleable iron threaded pipe unions, classes 150, 250, and 300

B16.40 Manually operated thermoplastic gas shutoffs and valves in gas distribution system

B16.41 Functional qualification requirements for power operated active valve assemblies for nuclear power plants

B16.42 Ductile iron pipe flanges and flanged fittings classes 150 and 300

B16.43 Wrought copper and copper alloy solder joint fittings for solvent drainage systems

B16.44 Manually operated metallic gas valves for use in house piping systems

B16.45 Cast iron fittings for solvent drainage systems

B16.47 Large diameter steel flanges NPS 26–60

B16.48 Steel line blanks

## 1.9 API STANDARDS AND RECOMMENDED PRACTICES

The American Petroleum Institute publishes codes, standards, studies and recommended practices related to piping, tanks, vessels, equipment and systems used in the petroleum industry. In addition to design and fabrication rules, the API documents address maintenance, in-service inspections, evaluation of component degradation, also referred to as “fitness-for-service” or “fitness-for-purpose”, and repairs. Even though these documents are developed for use in the petroleum industry, there are many API standards relevant to the chemical process industry in general. API refining documents related to pressure equipment include:

510 Pressure Vessel Inspection Code: Maintenance, Inspection, Rating, Repair, and Alteration.

530 Fired Heater Tubes.

570 Piping Inspection Code: Inspection, Repair, Alterations, and Rerating of In-Service Piping Systems.

571 Conditions Causing Failure.

572 Inspection of Pressure Vessels.

573 Inspection of Fired Boilers and Heaters.

574 Inspection of Piping, Tubing, Valves, and Fittings.

575 Inspection of Atmospheric and Low Pressure Storage Tanks.

576 Inspection of Pressure Relieving Devices.

577 Welding Inspections.

578 Material Verification Program for New and Existing Alloy Piping Systems.

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579 Fitness-for-Service.

581 Base Resource Document—Risk Based Inspection.

582 Recommended Practice and Supplementary Welding Guidelines for the Chemical, Oil, and Gas Industries.

598 Valve Inspection and Test.

610 Centrifugal Pumps for Petroleum, Heavy Duty Chemical and Gas Industry Services.

611 General Purpose Steam Turbines for Petroleum, Chemical, and Gas Industry Services.

612 Special Purpose Steam Turbines for Petroleum, Chemical, and Gas Industry Services.

613 Special Purpose Gear Units for Petroleum, Chemical and Gas Industry Services.

617 Centrifugal Compressors for Petroleum and Gas Industry.

618 Reciprocating Compressors for Petroleum, Chemical and Gas Industry Services.

619 Rotary-Type Positive Displacement Compressors for Petroleum, Chemical, and Gas Industry Services.

620 Design and Construction of Large, Welded, Low Pressure Storage Tanks.

650 Welded Steel Tanks for Oil Storage.

651 Cathodic Protection of Above Ground Storage Tanks.

652 Lining of Above Ground Petroleum Storage Tanks.

653 Tank Inspection, Repair, Alteration, and Reconstruction Code.

674 Positive Displacement Pumps—Reciprocating.

675 Positive Displacement Pumps—Controlled Volume.

676 Positive Displacement Pumps—Rotary.

686 Machinery Installation and Installation Design

751 HF Acid.

850 API Standards 620, 650 and 653 Interpretations.

920 Prevention of Brittle Fracture of Pressure Vessels.

937 Evaluating Design Criteria for Storage Tanks with Frangible Roof Joints.

941 Steels for Hydrogen Service at Elevated Temperatures and Pressures in Petroleum Refineries and Petrochemical Plants.

945 Avoiding Environmental Cracking in Amine Units.

1107 Pipe Line Maintenance Welding.

2510 Design and Construction of LPG Installations.

API valve publications include:

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589 Fire Test for Evaluation of Valve Stem Packing.

591 User Acceptance of Refinery Valves.

594 Wafer Check Valves.

595 Cast Iron Gate Valves.

598 Valve Inspection and Testing.

599 Metal Plug Valves—Flanged and Welding Ends.

600 Steel Gate Valves, Flanged and Butt Welding Ends.

602 Compact Steel Gate Valves.

603 Class 150 Cast Corrosion Resistant Flanged End gate Valves.

604 Ductile Iron Gate Valves, Flanged Ends.

606 Compact Steel Gate Valve, Extended Body.

607 Fire Test for Soft Seat Quarter Turn Valves.

608 Metal Ball Valves, Flanged and Butt Welding Ends.

609 Butterfly Valves Lug Type and Wafer Type.

API pipeline transportation publications include:

5L Specification for Line Pipe.

5L1 Railroad Transportation of Line Pipe.

6D Specification for Pipeline Valves.

10E Application of Cement Lining to Steel Tubular Goods, Handling, Installation, and Joining.

15HR Specification for High Pressure Fiberglass Line Pipe.

15LE Specification for Polyethylene Line Pipe.

15LP Specification for Thermoplastic Line Pipe.

15LR Specification for Low Pressure Fiberglass Line Pipe.

15L4 RP Care and Use of Reinforced Thermosetting Resin Line Pipe.

1102 Steel Pipelines Crossing Railroads and Highways.

1104 Welding of Pipelines and Related Facilities.

1109 Marking Liquid Petroleum Pipeline Facilities.

1110 Pressure Testing of Liquid Petroleum Pipelines.

1111 Design, Construction, Operation, and Maintenance of Offshore Hydrocarbon Pipeline and Risers.

1113 Developing a Pipeline Supervisory Control Center.

1114 Design of Solution-Minded Underground Storage Facilities.

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1115 Operation of Solution-Minded Underground Facilities.

1117 Movement of In-Service Pipelines.

1123 Development of Public Awareness Programs by Hazardous Liquid Pipeline Operators.

1130 Computational Pipeline Monitoring.

1132 Effects of Oxygenated Fuels and Reformulated Diesel Fuels on Elastomers and Polymers in Pipeline/Terminal Component.

1149 Pipeline Variable Uncertainties and Their Effects on Leak Detectability.

1155 Evaluation Methodology of Software Based Leak Detection Systems.

1156 Effects of Smooth and Rock Dents on Liquid Petroleum Pipelines.

1157 Hydrostatic Test Water Treatment and Disposal Operations for Liquid Pipeline Systems.

1158 Analysis of DOT Reportable Incidents for Hazardous Liquid Pipelines, 1986 Through 1996.

1160 Managing System Integrity for Hazardous Liquid Pipelines.

1161 Guidance Document for the Qualification of Liquid Pipeline Per

sonnel.

2200 Repairing Crude Oil, Liquefied Petroleum Gas and Product Pipelines.

API overpressure protection publications include:

11V7 Repair, Testing, and Setting Gas Lift Valves.

520 Sizing, Selection, and Installation of Pressure Relieving Devices in Refineries.

521 Guide for Pressure Relieving and Depressurizing Systems.

526 Flanged Steel Safety Relief Valves.

527 Seat Tightness of Pressure Relief Valves with Metal-to-Metal Seats.

2000 Venting Atmospheric and Low Pressure Storage Tanks. Non-Refrigerated and Refrigerated.

## 1.10 MANUFACTURERS STANDARDIZATION SOCIETY

The Manufacturer's Standardization Society (MSS) of the Valve and Fittings Industry has developed several standards related to the fabrication of pipe, fittings, valves and pipe supports. These include:

SP-6 Standard finishes of contact faces of pipe flanges and connecting end flanges of valves and fittings.

SP-9 Spot facing for bronze, iron, and steel flanges.

SP-25 Standard marking system for valves, fittings, flanges, and unions.

SP- Class 150 corrosion resistance gate, globe, angle, and check valves with flanged and butt-

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| | |
|---|---|
| 42 | weld ends. |
| SP-43 | Wrought stainless steel butt-welding fittings. |
| SP-44 | Steel pipeline flanges. |
| SP-45 | By-pass and drain connections. |
| SP-51 | Class 150LW corrosion resistant cast flanges and flanged fittings. |
| SP-53 | Quality standards for steel casting and forgings for valves, flanges, and fittings and other piping components, magnetic particle examination method. |
| SP-54 | Quality standards for steel casting for valves, flanges, and fittings and other piping components, radiographic examination method. |
| SP-55 | Quality standards for steel casting for valves, flanges, and fittings and other piping components, visual method for evaluation of surface irregularities. |
| SP-58 | Pipe hangers and supports—materials, design, and manufacture. |
| SP-60 | Connecting flange joint between tapping sleeves and tapping valves. |
| SP-61 | Pressure testing of steel valves. |
| SP-65 | High-pressure chemical industry flanges and threaded stubs for use with lens gaskets. |
| SP-67 | Butterfly valves. |
| SP-68 | High-pressure butterfly valves with offset design. |
| SP-69 | Pipe hangers and supports selection and application. |
| SP-70 | Cast iron gate valves, flanged, and threaded ends. |
| SP-71 | Gray iron swing check valves, flanged and threaded ends. |
| SP-72 | Ball valves with flanged or buttwelding ends for general service. |
| SP-73 | Brazing joints for wrought and cast copper alloy solder joint pressure fittings. |
| SP-75 | Specification for high test wrought buttwelding fitting. |

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SP-77 Guidelines for pipe support contractual relationships.

SP-78 Cast iron plug valves, flanged and threaded ends.

SP-79 Socket welding reducer inserts.

SP-80 Bronze gate, globe, angle and check valves.

SP-81 Stainless steel, bonnetless flanged knife gate valves.

SP-82 Valve pressure testing methods.

SP-83 Class 300 steel pipe unions socket welding and threaded.

SP-85 Cast iron globe and angle valves flanged and threaded ends.

SP-86 Guidelines for metric data in standards for valves, flanges, fittings, and actuators.

SP-87 Factory made buttwelding fittings for class 1 nuclear piping applications.

SP-88 Diaphragm type valves.

SP-89 Pipe hangers and supports—fabrication and installation practices.

SP-90 Guidelines on terminology for pipe hangers and supports.

SP-91 Guidelines for manual operation of valves.

SP-92 MSS valve user guide.

SP-93 Quality standard for steel castings and forgings for valves, flanges, and fittings and other piping components liquid penetrant examination method.

SP-94 Quality standard for ferritic and martensitic steel castings for valves, flanges, and fittings and other piping components—ultrasonic examination method.

SP-95 Swaged nipples and bull plugs.

SP-96 Guidelines on terminology for valves and fittings.

SP-97 Integrally reinforced forged carbon steel branch outlet fittings—socket welding, threaded, and buttwelding ends.

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SP-98 Protective coatings for the interior of valves, hydrants and fittings.

SP-99 Instrument Valves

SP-100 Qualification requirements for elastomer diaphragms for nuclear service diaphragm type valves.

SP-101 Part-turn valve actuator attachment—flange and driving compo

nent dimensions and performance characteristics.

SP-102 Multi-turn valve actuator attachment—flange and driving component dimensions and performance characteristics.

SP-103 Wrought copper and copper alloy insert fittings for polybutylene systems.

SP-104 Wrought copper solder joint pressure fittings.

SP-105 Instrument valves for code applications.

SP-106 Cast copper alloy flanges and flanged fittings, class 125, 150 and 300.

SP-107 Transition union fittings for joining metal and plastic products.

SP-108 Resilient-seated cast iron eccentric plug valves.

SP-109 Welded fabricated copper solder joint pressure fittings.

SP-110 Ball valves threaded, socket welding, solder joint, grooved and flared ends.

SP-111 Gray-iron and ductile-iron tapping sleeves.

SP-112 Quality standards for evaluation of cast surface finishes—Visual and tactile method.

SP-113 Connecting joint between tapping machines and tapping valves.

SP-114 Corrosion resistant pipefittings threaded and socket welding, class 150 and 1000.

SP-115 Excess flow valves for natural gas service.

SP-116 Service line valves and fittings for drinking water systems.

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SP-117 Bellows seals for globe and gate valves.

SP-118 Compact steel globe and check valves—flanged, flangeless, threaded and welding ends.

SP-119 Belled end socket welding fittings, SS and copper nickel.

SP-120 Flexible graphite packing system for rising stem steel valves.

SP-121 Qualification testing methods for stem packing for rising stem valves.

SP-122 Plastic industrial ball valves.

SP-123 Non-threaded and solder joint unions for use with copper water tube.

SP-124 Fabricated tapping sleeves.

SP-125 Gray iron and ductile iron in-line spring loaded center guided check valves.

SP-126 Steel in-line spring assisted center guided check valves.

SP-127 Bracing for piping systems seismic-wind-dynamic design, selection, and application.

## 1.11 PIPE FABRICATION INSTITUTE STANDARDS

The Pipe Fabrication Institute's Engineering Standards provide guidance in several areas related to the fabrication, dimensioning, marking, cleaning and testing of new pipe:

ES-1 Internal machining and solid machined backing rings for circumferential butt weld.

ES-2 Method of dimensioning piping assemblies.

ES-3 Fabricating tolerances.

ES-4 Hydrostatic testing of fabricated piping.

ES-5 Cleaning of fabricated piping.

ES-7 Minimum length and spacing for welded nozzles.

ES-11 Permanent marking on piping materials.

ES-16 Access holes, bosses, and plugs for radiographic inspection of pipe welds.

ES- Wall thickness measurement by ultrasonic examination.

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| | |
|---|---|
| 20 | |
| ES-21 | Internal machining and fit-up of GTAW root pass circumferential butt weld. |
| ES-22 | Recommended practice for color-coding of piping materials. |
| ES-24 | Pipe bending methods, tolerances process and material requirements. |
| ES-25 | Random radiography of pressure retaining girth butt welds. |
| ES-26 | Welded load bearing attachments to pressure retaining piping materials. |
| ES-27 | Visual examination—The purpose, meaning and limitation of the term. |
| ES-29 | Abrasive blast cleaning of ferritic piping materials. |
| ES-30 | Random ultrasonic examination of butt welds. |
| ES-31 | Standard for protection of ends of fabricated piping assemblies. |
| ES-32 | Tool calibration. |
| ES-33 | Circumferential buttwelds in the arc of pipe bends. |
| ES-34 | Painting of fabricated piping. |
| ES-35 | Nonsymmetrical bevels and joint configurations for butt welds. |
| ES-36 | Branch reinforcement work sheets. |
| TB 1 | Pressure temperature ratings of seamless pipe used in power plant piping systems. |
| TB 3 | Guidelines clarifying relationships and design engineering responsibilities between purchasers' engineers and pipe fabricator erector. |

## 1.12 AMERICAN INSTITUTE OF STEEL CONSTRUCTION

The Manual of Steel Construction of the American Institute of Steel Construction (AISC) is typically followed for the design of pipe and equipment support structures. It includes dimensions and properties of structural steel members, rules for beam and girder design, column design, connections and weld joints, and specifications.

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## 1.13 AMERICAN CONCRETE INSTITUTE

The standards of the American Concrete Institute are usually followed to size pipe support concrete anchor bolts. The applicable publications are:

ACI-318 Building Code Requirements for Reinforced Concrete.

ACI-349 Requirements for Nuclear Safety Related Concrete Structures.

ACI-355 State of the Art Report on Anchorage to Concrete.

## 1.14 NACE

The National Association of Corrosion Engineers (NACE) publishes recommended practices and standards for the selection, testing and corrosion protection of pipe materials. These include:

RP0169 Control of external corrosion on underground or submerged metallic piping systems;

RP0170 Protection of austenitic steels and other austenitic alloys from polythionic acid stress corrosion cracking during shutdown of refinery equipment;

RP0175 Control of internal corrosion in steel pipelines and piping systems;

RP0178 Fabrication details, surface finish requirements, and proper design considerations for tanks and vessels to be lined for immersion service;

RP0181 Liquid-applied internal protective coatings for oilfield production equipment;

RP0184 Repair of lining systems;

RP0185 Extruded Polyolefin resin coating systems for underground-submerged pipe;

RP0187 Design considerations for corrosion control of reinforcing steel in concrete;

RP0188 Discontinuity (holiday) testing of protective coatings;

RP0189 On-line monitoring of cooling waters;

RP0190 External protective coatings for joints, fittings, and valves on metallic underground or submerged pipelines and piping systems;

RP0192 Monitoring corrosion in oil and gas production with iron counts;

RP0285 Corrosion control of underground storage tank systems by cathodic protection;

RP0286 The electrical isolation of cathodically protected pipelines;

RP0288 Inspection of linings on steel and concrete;

RP0291 Care, handling, and installation of internally plastic-coated oilfield tubular goods and accessories;

RP0296 Wet H2S cracking.

RP0375 Wax coating systems for underground piping systems;

RP0387 Metallurgical and inspection requirements for cast sacrificial anodes for offshore

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applications;

RP0388 Impressed current cathodic protection of internal submerged surfaces of steel water storage tanks;

RP0472 Methods and controls to prevent in-service environmental cracking of carbon steel weldments in corrosive petroleum refining environments;

RP0475 Selection of metallic materials to be used in all phases of water handling for injection into oil-bearing formations;

RP0490 Holiday detection of fusion-bonded epoxy external pipeline coatings of 250 to 760 μm (10 to 30 mils);

RP0572 Design, installation, operation and maintenance of impressed current deep groundbeds;

RP0575 Internal cathodic protection systems in oil-treating vessels; RP0590 Deaerator cracking;

RP0675 Control of external corrosion on offshore steel pipelines;

RP0775 Preparation and installation of corrosion coupons and interpretation of test data in oilfield operations;

Item No. 54276 Cathodic protection monitoring on buried pipelines;

Item No. 54277 Specialized surveys of buried pipelines.

## 1.15 MATERIALS INSTITUTES

The Materials Technology Institute (MTI) and the Materials Properties Council (MPC) sponsor research in materials technology, corrosion and performance. ASM International (formerly the American Society for Metals) publishes data and textbooks on materials, material properties, corrosion, failure analysis, etc., including the classic ASM Handbook, which provides a comprehensive and practical treatment of the science and engineering of ferrous and non-ferrous metals. The Handbook consists of the following volumes:

Volume 1: Properties and Selection, Irons, Steels, and High-Performance Alloys

Volume 2: Properties and Selection, Nonferrous Alloys and Special-Purpose Materials

Volume 3: Alloy Phase Diagrams

Volume 4: Heat Treating

Volume 5: Surface Engineering

Volume 6: Welding, Brazing, and Soldering

Volume 7: Powder Metal Technologies and Applications

Volume 8: Mechanical Testing and Evaluation

Volume 9: Metallography and Microstructures

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| Volume 10: | Materials Characterization |
| Volume 11: | Failure Analysis and Prevention |
| Volume 12: | Fractography |
| Volume 13: | Corrosion |
| Volume 14: | Forming and Forging |
| Volume 15: | Casting |
| Volume 16: | Machining |
| Volume 17: | Nondestructive Evaluation and Quality Control |
| Volume 18: | Friction, Lubrication, and Wear Technology |
| Volume 19: | Fatigue and Fracture |
| Volume 20: | Materials Selection and Design |
| Volume 21: | Composites |

## 1.16 NATIONAL BOARD

The National Board of Boiler and Pressure Vessels Inspectors is an organization that assembles chief boiler and vessel inspectors from the United States, Canada and Mexico, aimed at achieving and maintaining the safety of pressure retaining components. The National Board publishes the National Board Inspection Code [NBIC]. The NBIC code provides rules for inspection, repair and alterations of pressure components such as boilers, pressure vessels and piping systems, and pressure relief devices.

Organizations can seek a National Board “R” or “VR” certificate of authorization for the repair of vessels or relief devices respectively, under the oversight of an Authorized Inspection Agency, and under a formal quality program.

## 1.17 FLOW CONTROL INSTITUTE STANDARDS

| | |
|---|---|
| FCI 68 | Recommended Procedure in Rating Flow and Pressure Characteristics of Solenoid Valves for Gas Service |
| FCI 69 | Pressure Rating Standard for Steam Traps |
| FCI 70–1 | Standard Terminology and Definition for Filled Thermal Systems for Remote Sensing Temperature Regulators |
| FCI 70–2 | Control Valve Seat Leakage |
| FCI 73–1 | Pressure Rating Standard for ‘Y’ Type Strainers |
| FCI 74–1 | Spring Loaded Lift Disc Check Valve Standard |

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| FCI 78–1 | Pressure Rating Standards for Pipeline Strainers Other Than 'Y' |
| FCI 79–1 | Proof of Pressure Ratings for Pressure Regulators |
| FCI 81–1 | Proof of Pressure Ratings for Temperature Regulators |
| FCI 82–1 | Recommended Methods for Testing and Classifying the Water Hammer Characteristics of Electrically Operated Valves |
| FCI 84–1 | Metric Definition of the Valve Flow Coefficient Cv |
| FCI 85–1 | Production Testing for Steam Traps |

## 1.18 HYDRAULIC INSTITUTE PUMP STANDARDS

| | |
|---|---|
| HI 1.1–1.5 | Centrifugal Pumps |
| HI 1.3.4 | Centrifugal Pumps for Horizontal Baseplate Design |
| HI 1.6 | Centrifugal Pump Tests |
| HI 2.1–2.5 | Vertical Pumps |
| HI 2.6 | Vertical Pump Tests |
| HI 3.1–3.5 | Rotary Pumps |
| HI 3.6 | Rotary Pump Tests |
| HI 4.1–4.6 | Sealless Pump Tests |
| HI 6.1 -6.5 | Reciprocating Pump Tests |
| HI 6.6 | Reciprocating Pump Tests |
| HI 7.1–7.5 | Controlled Volume Pumps |
| HI 8.1–8.5 | Direct Acting (Steam) Pumps |
| HI 9.1–9.5 | Pumps—General Guidelines |
| HI 9.3.3 | Pumps—Polymer Material Selection |
| HI 9.6.1 | Centrifugal and Vertical Pumps for NPSH Margin |
| HI 9.6.3 | Centrifugal/Vertical Pumps Allowable Operating Region |
| HI 9.8 | Pump Intake Design |

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## 1.19 REFERENCES

API 910, Digest of State Boiler, Pressure Vessel, Piping, and Above Ground Storage Tanks Rules and Regulations, American Petroleum Institute, Washington, D.C.

Leight, W. and Collins, B., Setting the Standards, Mechanical Engineering Magazine, February 2000.

NBIC, National Board Inspection Code, ANSI/NB-23, The National Board of Boiler and Pressure Vessel Inspectors, Columbus, OH.

Uniform Boiler and Pressure Vessel Laws Society, Synopsis of Boiler and Pressure Vessel Laws, Rules and Regulations, Louisville, KY.

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# 2
# Fundamentals

## 2.1 COMPETENCE

Whether the task at hand is a multi-million dollar design and construction project, a repair, a procurement, or a small maintenance work package, the difference between success and failure is technical competence: the right knowledge at the right time. This obvious fact applies across all disciplines: civil-structural, process-mechanical, electrical, instrumentation and controls, maintenance and operations. The best flow charts, quality plans, execution plans, project management logics, tracking and trending, or cost-schedule analyses can not replace the fundamental need for technical competence at the engineering level and at the corporate level.

## 2.2 AT THE ENGINEERING LEVEL

The seven fundamental areas of competence in the mechanical engineering discipline are illustrated in the exploded view of Figure 2-1, they are: (1) materials, (2) design, (3) construction, (4) inspection, (5) testing, (6) maintenance, and (7) operations. Note the close interrelationship between these seven areas.

The sign of a good operation is the tight fit and continuous feedback between the boxes of Figure 2-1, embodied in the competence of its people, the streamline of its organization and the completeness, elegance and clarity of its procedures.

Operations (production) envelop all other activities; indeed, profitable and responsible operations are the source and final objective of all other activities. Operations, the outer enveloping box, benefits from the good decisions and is hindered by the poor decisions made in each of the other six boxes.

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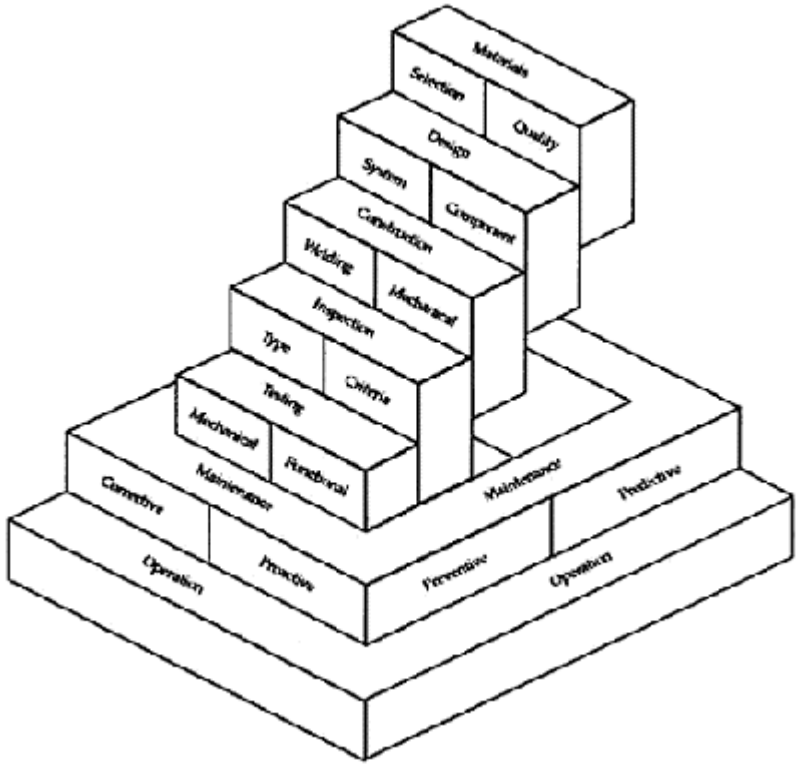


**Figure 2-1** The Seven Essentials of Successful Mechanical Engineering

In each of the seven fundamental areas, the responsible engineer must make a series of decisions to achieve the optimum and most cost-effective operation. The following checklists summarize the key decision points related to piping and pipeline activities.

### 2.2.1 Materials

Choice of material specifications.
- Code listed or unlisted materials.
- Consistency of material form with code.
- Compatibility of materials with fluid and temperature range.
- Compatibility of materials with each other.
- Minimum temperature and toughness.
- Maximum temperature and strength.
- Weldability, welding electrodes.
- Supplementary requirements (carbon equivalent, corrosion testing, etc.).
- Lining, coating, paint, cathodic protection, galvanizing.
- Need for material test reports, positive material identification, and traceability.

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Oversight and hold points.
Permitted repairs to materials, and post-repair inspections.
Experience with similar materials and service.
Degradation mechanisms and service life.
Corrosion allowance.
Design limitations (flow, temperature, trace elements, cleaning, etc.).
Material records, filing and retrieval logic.

### 2.2.2 Design

#### *2.2.2.1 System Design*

System function.
Technology of the process: fundamentals, experience and development.
Process cost-benefit.
Operating and safety logic and viability.
Process and instrumentation diagrams (P&ID).
System descriptions.
Redundancies and separation.
Layout drawings.
Interfaces with other systems.
Pressures, temperatures and flow rates (throughput).
Heat transfer and heat loss.
Design life, future throughput.
Sizing pumps, valves, compressors, blowers.
Power supplies.
Air, water, utilities supplies.
Normal operation.
Credible accident conditions.
Overpressure protection (relief valves, rupture disks, interlocks, etc.).
Safety and environmental analyses.
Operating envelope and technical specifications.
System design records filing and retrieval logic.

#### *2.2.2.2 Component Design*

Codes and standards.
Regulatory and contractual requirements.
Design Specifications (input) and Design Reports (output).
Equipment interfaces and code boundaries.
Layout logic and key dimensions.
Hazard classification (lethal, safety class, etc.).
Design maximum and minimum pressures (vacuum, internal or external pressure).
Design maximum and minimum temperatures.
Normal operating loads (weight, pressure, temperature, valve thrust, lifting, etc.).
Local and contact bearing stresses.

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Fatigue loads and cycles.
Extreme loads (wind, seismic, explosion, water-hammer, etc.).
Extreme loads qualification.
Load combinations.
Corrosion allowance.
Strength and stability design.
Fracture prevention and toughness design.
Design of openings and reinforcements.
Overpressure protection.
Equipment nozzle loads.
Component load-displacement limits (mechanical fittings, expansion joints, etc.).
Unlisted components (rating of non-B16 fittings, proof testing, etc.).
Design of support structures and attachments.
Provisions for maintenance access.
Experience with similar designs and service.
Piping specifications.
Equipment specifications (pumps, valves, compressors, tanks, vessels, etc.).
Mechanical flow sheets with piping and equipment specifications.
Detailed design drawings, isometrics and orthographics.
Design quality assurance: software, calculations, drawings, and specifications.
Professional Engineer certifications.
Mechanical design records filing and retrieval logic.

### 2.2.3 Construction

Procurement and “make or buy” decisions.
Fabrication and procurement specifications.
Bid criteria and suppliers selection.
Fabricator experience, schedule, warranties, compensation.
Drawings, critical dimensions and tolerances.
Planning sequence: procurement, shop fabrication and field erection.
Personnel, craft experience, training, proficiency.
Personnel and process certifications.
Welding procedures.
Procedures for bending, flange joining, and threading.
Procedure for mechanical joining, compression fittings and assembly.
Pre-heat and post-weld heat treatment.
Leak testing (hydro, pneumatic, sensitive leak, etc.).
Control and disposition of non-conformances.
Permitted repairs of fabrication flaws.
Construction records filing and retrieval logic.
Hold points and inspection interfaces.
As-built reconciliation of design.

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### 2.2.4 Quality Control Inspections

Examinations (quality control) and inspections (quality assurance).
- Examination methods and percentages.
- Personnel and process certifications.
- Calibration of examination tools.
- Surface or volumetric examinations.
- Acceptance criteria.
- Hold points and independent inspections by owner representative.
- Control and disposition of non-conformances.
- Documentation of inspections.
- Inspection records filing and retrieval logic.

### 2.2.5 Preoperational Testing

#### *2.2.5.1 Mechanical Testing*

Leak testing technique (hydro, pneumatic, sensitive leak, etc.).
- Test boundaries.
- Test fluid and pressure.
- Testing cautions.
- Test personnel.
- Hold points.
- Start-up testing.
- Vibration startup monitoring.
- Thermal expansion startup monitoring for hot lines.
- Test acceptance criteria.
- Control and disposition of non-conformances.
- Mechanical test records filing and retrieval logic.

#### *2.2.5.2 Operational Testing*

Component operability tests.
- System flow tests.
- Instrumentation and controls tests.
- Commissioning.
- Measurements of key process variables and acceptance criteria.
- Turnover process from engineering and maintenance to operations.
- Operational test records filing and retrieval logic.

### 2.2.6 Maintenance

Corrective: run-to-failure
- Preventive: adjust or replace at fixed intervals.
- Predictive: inspect and decide based on fitness-for-service analysis.

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Active vs. reactive (corrective, run-to-failure) by system and by component.
Active: planned (fixed interval) vs. predictive (trended) maintenance.
Analysis and trending of maintenance history.
Reliability analysis.
Maintenance feedback and lessons learned process.
On-stream (in-service) or outage (shutdown) maintenance.
Sampling and corrosion monitoring.
Visual (external or internal, direct or remote).
Surface (liquid penetrant, magnetic particles, etc.).
Volumetric (radiography, ultrasonic, etc.)
Periodic hydrotest.
Fitness-for-service (run-or-repair) analysis: API-579, ASME XI, B31.G, etc.
Qualified welding or mechanical repairs
Corrosion repairs, lining and coating.
National Board certification of repair program.
Repair stamp (R, VR).
End of system or component life
Shutdown and disposition plan.
Maintenance records filing (electronic database) and retrieval logic.

### 2.2.7 Operation

Operator experience, proficiency and certification.
Operating procedures.
Operating envelopes and limits.
Emergency procedures.
Understanding operating processes, access to system engineer.
Role of shift engineer.
Alarms, facility drills and emergency response.
Standing orders and shift orders.
Shift turnover process.
Authorities to operate equipment.
System descriptions and up-to-date system diagrams.
Equipment labels.
Normal operating loads in service: weight, temperature, pressure, flow.
Accident loads: large vibration, water-hammer, winds, seismic, explosion.
Log and analysis of operating and accident conditions.
Understanding degradation in service and abnormal response.
Analysis of abnormal events.
Lessons learned, plant, company and industry feedback.
Engineering reviews and hold points.
Post-maintenance turnover.
Operating records filing and retrieval logic.
Audits, critiques, self-assessment and continuous improvement.
Prioritization process for upgrades and projects.
Interface operations-maintenance-engineering.

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Temporary modification process.
Change control process.
Production objectives, measurement, trending, and improvement.

## 2.3 AT THE CORPORATE LEVEL

Competence at the corporate level is a balance between four competing priorities, Figure 2-2: maximum production, low cost, smart safety and continuous development. Maximum production and low cost are self-evident priorities, and the pull is naturally strong in these two directions. If this pull overwhelms the other two priorities, smart safety and continuous development, profits could be increased in the short term but the operation is headed for serious troubles. The costs of poor safety and neglected technology and personnel development are unforgiving.

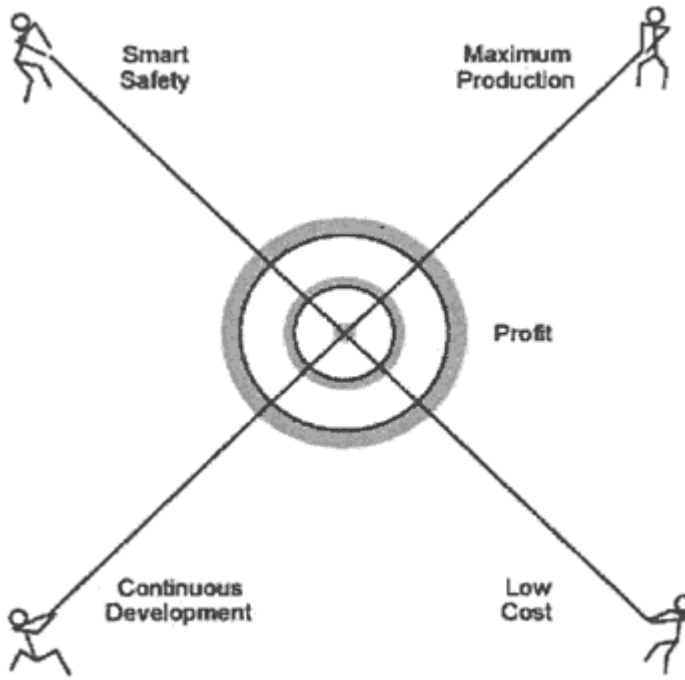


**Figure 2-2** Profit as a Balance of Four Competing Priorities

Safety is driven first by ethics (sense of responsibility towards the public, the worker, and the environment) and second by profit. In the case of piping and pipelines, lax safety will manifest itself as failures and accidents, in some cases with overwhelming consequences: cost of lost production, cost of failure analysis, cost of repairs, cost of

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cleanup, cost of legal process and liabilities, insurance costs, cost of regulatory hearings, investigations and fines, cost of retrofits on other similar systems, and loss of corporate reputation with its workers, the public, the press, its clients, its competitors, regulators, and in financial markets.

Note the use of the term “Smart Safety” in Figure 2-2. Smart safety means that time, effort and money allocated to safety and safety upgrades must be wisely distributed. An overkill in one safety improvement will be accomplished at the expense of another, more pressing safety need that goes unanswered.

Continuous development is innovation: invention, development, application and improvement. It is the never-ending search for better ways of doing better things. It is achieved through vision, leadership, drive, improvement in company staff knowledge and behavior, leading to developments in technology and processes. It does require investment in training, research and development. It is the engine for the other three priorities in Figure 2-2. In today’s piping and pipeline industries, this includes improvements in materials (for example corrosion resistant alloys and high strength line pipe), processes, components (valve operators with solid state controls, improved compressor and pump technology), design software (for example stress analysis, finite element and computational fluid dynamics software that are so much better than 20 years ago), and risk-based criteria (for example risk-based inspections in the process and nuclear industries, reliability based design in subsea pipelines), construction and welding technologies, non-destructive examinations and in-service inspections (“smart” pipeline pigs with improved sensitivity, pulsed eddy current inspection through insulation, acoustic emission testing), accurate and rapid leak detection (sensitive leak test techniques, real-time line balance and pipeline hydraulic modeling for quick leak detection), maintenance (reliability centered maintenance, industries forums and publications) and operations. In the United States, much of this work is achieved through industry groups, research institutes, consulting and engineering companies, national laboratories, and the operating companies themselves. These activities are essential for long-term profitability.

Personnel development, through planned on-the-job coaching and outside technical training is one of the top priorities of a good manager. The Department head of a large Engineering firm was asked “Don’t you worry that you will spend all this money on training your engineers and then see them leave?”, to which he replied “No, I only worry that they will not be trained…and stay”.

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# 3
# Materials

## INTRODUCTION: A GOOD PASTRY

Making a quality component is not unlike baking a good pastry. To the base element (iron in one case; flour in the other), we add key ingredients (alloys in one case; eggs, sugar, salt, water, etc. in the other). Each alloying element, like each baking ingredient, brings a unique and valuable characteristic to the mix. In the process of combining the key ingredients, we take care to avoid the introduction of impurities (sulfur in one case, a hair in the other). The ingredients are mixed and poured (casting or forging in one case; pouring in pots and pans in the other).

The mixture is then placed in the oven (heat treatment in one case; baking in the other). The difference between annealing, stress relieving, tempering, etc. can be seen as the difference between broiling, baking, grilling, etc. each with its heating rate, holding time and temperature, and cooling rate. The item has now reached its desired texture (mechanical and physical properties in one case; texture and moisture in the other). Moisture desirable for a cake can be seen as ductility desirable for a pipe. Crisp desirable in a crust can be seen as hardness desirable in a bearing surface.

The item is then given its final shape (machining in one case; carving in the other), its finish (lining, coating, painting, pickling, cleaning in one case; icing and trimming in the other), and finally its marking (specification required markings in one case; a "Happy Birthday" message in the other). Care must be exercised in both cases not to cause damage during transportation and storage.

In the final count, what will matter is the performance of the product: its compatibility with the fluid, the environment and the service in one case; its compatibility with the host's palate and digestive system in the other.

## 3.1 FERROUS PIPE

As illustrated in Figure 3-1, materials used in piping systems can be classified in two large categories: metallic and non-metallic. Metallic pipe and fitting materials can in turn be classified as ferrous (iron based) or non-ferrous (such as copper, nickel or aluminum based). Finally, within the category of ferrous materials, we can differentiate between two large groupings: wrought or cast irons, and steels.

### 3.1.1 Wrought Iron

In the Middle Ages, crushed iron was melted in stone furnaces, with the hot charcoal serving as a source of carbon. Crushed limestone was added to facilitate the removal of

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slag and impurities. The nearly pure iron formed that way, called wrought iron, was widely used for tools and weapons. Pure iron is soft and gains strength by alloying with carbon to form cast irons or steels.

Today, the term wrought generally applies to a product form achieved by working, which in practice is any product form other than a casting. Wrought materials include forging, billets (less than 6" deep) or blooms (more than 6" deep) made into bars, tubes or structural shapes, and slabs made into sheets, skelp (plate for making seam welded pipe) or plates.

### 3.1.2 Cast Iron

The term cast iron describes a series of iron and carbon alloys with a carbon in excess of 1.7% (percents refer to weight), which corresponds to the amount retained in solid solution at the eutectic temperature [ASTM A 644]. Other definitions of cast iron place the limit at 2% carbon and 1% silicon [ASM], or 3% carbon and 1.5% silicon [ASTM A 48].

Cast irons have good flow properties when melted, and are therefore well suited to pour into castings. They can be alloyed with Si, Ni or Cr to improve their abrasion or corrosion resistance. Cast irons can be classified by their mechanical properties [ASTM A 48] or by the condition of the carbon contained in the metal, in which case cast irons are classified as gray cast iron, ductile iron, white iron and malleable iron.

Gray cast iron is a cast iron produced by slow cooling of the iron from the melt, with a large proportion of graphite in the form of flakes in a matrix of ferrite and pearlite. The graphite flakes give the metal its gray color and gray fracture appearance. The silicon content makes gray cast iron easy to machine. It is commonly used for machinery pedestals and engine blocks (ASTM A 48, ASTM A 278, ASTM A 319, ASTM A 436) [Van Droffelaar].

Ductile iron is a rapidly cooled cast iron with a large proportion of its graphite in spherical nodules (ASTM A 395, ASTM A 439, ASTM A 536, ASTM A 571). It is strong and ductile, better suited for shock applications than gray cast iron.

White iron is cast iron with carbon in the combined form of cementite $Fe_3C$ (ASTM A 360, ASTM A 532). It is a hard material but fractures more easily with a silver white colored fracture surface.

Malleable iron is a white cast iron that has been annealed, with a large proportion of its graphite evolved from cementite to elongated clusters that are ductile but still maintain a good hardness (ASTM A 47).

Cast iron pipe (ASTM A 74) comes in 5 ft and 10 ft lengths, extra heavy (marked XH) or service (marked SV) wall thickness. The pipe's standard chemistry requirements are 0.75% maximum phosphorous and 0.15% maximum sulfur. Mechanical properties are not specified by the ASTM standard, and mechanical tests will not be conducted on cast iron pipe, unless the buyer requires minimum tensile or bending properties. Ductile iron culvert pipe (ASTM A 716) is manufactured in 18 to 20 ft lengths, in 14" to 54" diameter, with a minimum yield of 42 ksi and a minimum tensile strength of 60 ksi, an elongation at rupture of 10% or more, and a minimum Charpy V-notch toughness of 7 ft-lb at 70°F. Ductile iron gravity sewer pipe (ASTM A 746) is manufactured in 18 to 20 ft

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lengths, comes in thickness classes 50, 51 and 52. It has the same mechanical properties as ductile iron culvert pipe (ASTM A 716).

### 3.1.3 Steel Pipe and Fittings

#### *3.1.3.1 Essential Characteristics*

Engineering materials have four essential characteristics that are closely interrelated:

1—Chemistry: the primary element (iron in the case of ferrous metals), alloying elements (nickel, chromium, etc. with ferrous metals), incidental elements (small amount of unintended elements), and impurities (sulfur, phosphorous, etc.).

2—Physical properties: density, modulus of elasticity, coefficient of thermal expansion, electrical and heat conduction, etc.

3—Microstructure: atomic structure, metallurgical phase, type and size of grains.

4—Mechanical Properties: strength (yield, ultimate, elongation at rupture) and toughness (Charpy, nil ductility transition temperature, fracture toughness, ductile vs. brittle appearance of fracture surface).

When a mill buys raw material to be used in forging, extruding or casting to fabricate fittings and components, it primarily buys a good chemistry (homogeneous chemistry with no impurities or flaws). Then, as the material is forged, extruded, cast and heat-treated, its microstructure is altered and its mechanical properties are set. In the case of carbon steel skelp used to fabricate seam-welded pipe, the initial plate chemistry, microstructure and mechanical properties are all essential since the skelp will not be significantly altered during the fabrication process.

#### *3.1.3.2 Carbon Steels*

Steels are first classified according to their chemistry. Steel pipe and fittings are alloys of iron (Fe) and carbon, containing less than 1.7% carbon. They can be classified in three groups: carbon steels, low alloy steels and high alloy steels. Common steel pipe material specifications are listed in Table 3-1.

Carbon steels consist of iron, less than 1.7% carbon, less than 1.65% manganese, incidental amounts of silicon (Si), aluminum (Al), and limits on impurities such as sulfur (Su), oxygen (O), nitrogen (N), and no specified minimum for elements such as Al, Cr, Co, Ni, Mo, Ni [ASM, ASTM A 941].

Carbon steel is the most common pipe material in the power, chemical, process, hydrocarbon and pipeline industries. Carbon steel pipe specifications commonly used in steam, water or air service include ASTM A106 and ASTM A53. A common steel for pipelines is API 5L. Carbon steels can in turn be classified as “mild”, “medium” and “high” carbon. Mild steel is a carbon steel with less than 0.30% carbon. Medium carbon steel has 0.30% to 0.60% carbon. High carbon steel has over 0.6% carbon.

#### *3.1.3.3 Alloy Steels*

Alloy steels are steels containing deliberate amounts of alloying elements, such as 0.3% chromium (Cr), 0.3% nickel (Ni), 0.08% molybdenum (Mo), etc [ASTM A 941]. Low

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alloy steels are alloy steels that contain less than the minimum percentages of alloys that define an “alloy steel”. In other definitions, low alloy steels are steels with less than 5% total alloys [ASM].

Alloy steels are common in high temperature service, such as high-pressure steam lines in power plants, heat exchanger and furnace tubes, and chemical reactor vessels. Examples of low alloy steels include 0.5Cr-0.5Mo (ASTM A 335 P2), 1Cr-0.5Mo (ASTM A 335 P12), 1.5Cr-0.5Mo (ASTM A 335 P11), 2Cr-1Mo (ASTM A 335 P3b), 2.25Cr-1Mo (ASTM A 335 P22), 3Cr-1Mo (ASTM A 335 P21). Intermediate alloy steels contain between 3% and 10% Cr, such as 4 to 9Cr −0.5 to 1 Mo (ASTM A 335 P5 to P9).

Each alloying element plays a unique role in improving the material’s properties:

Carbon (C) increases strength (yield and ultimate) and hardness, at the cost of reduced ductility (elongation at rupture) and notch toughness (Charpy, nil ductility transition temperature NDT).

Manganese (Mn) deoxidizes and desulfurizes steel. It traps sulfur impurities, avoiding brittle iron-sulfides, improves hot-workability and refines grain. If Mn/C >3, the manganese improves impact toughness. Above 0.8% manganese tends to harden steel.

Silicon (Si) is a deoxidizer that captures dissolved oxygen and avoids porosities. Improves castability.

Chromium (Cr) increases resistance to abrasion and wear. Above 11.5% Cr forms a stable oxide protective layer. Cr also improves resistance to high temperature hydrogen attack and graphitization.

Molybdenum (Mo) is a grain refiner. It enhances creep resistance and high temperature strength. It improves resistance to pitting corrosion in many environments.

Nickel (Ni) causes a significant improvement in fracture toughness and fatigue resistance. Above 7% it causes the atomic structure to be austenitic at room temperature.

Aluminum (Al) improves the deoxidization achieved with Silicon.

Copper (Cu) scavenges sulfur and improves atmospheric corrosion resistance.

Vanadium (V) refines steel grains, improving its mechanical properties. It improves a steel’s resistance to hydrogen attack at high temperature.

The most common impurities in steel are sulfur and phosphorous. Sulfur (S) is an impurity that forms brittle, crack-prone iron-sulfide.

Phosphorus (P) increases the ultimate strength of steel; otherwise, it is mostly considered an impurity that forms brittle, crack prone iron-phosphide, particularly during heat treatment or high temperature service.

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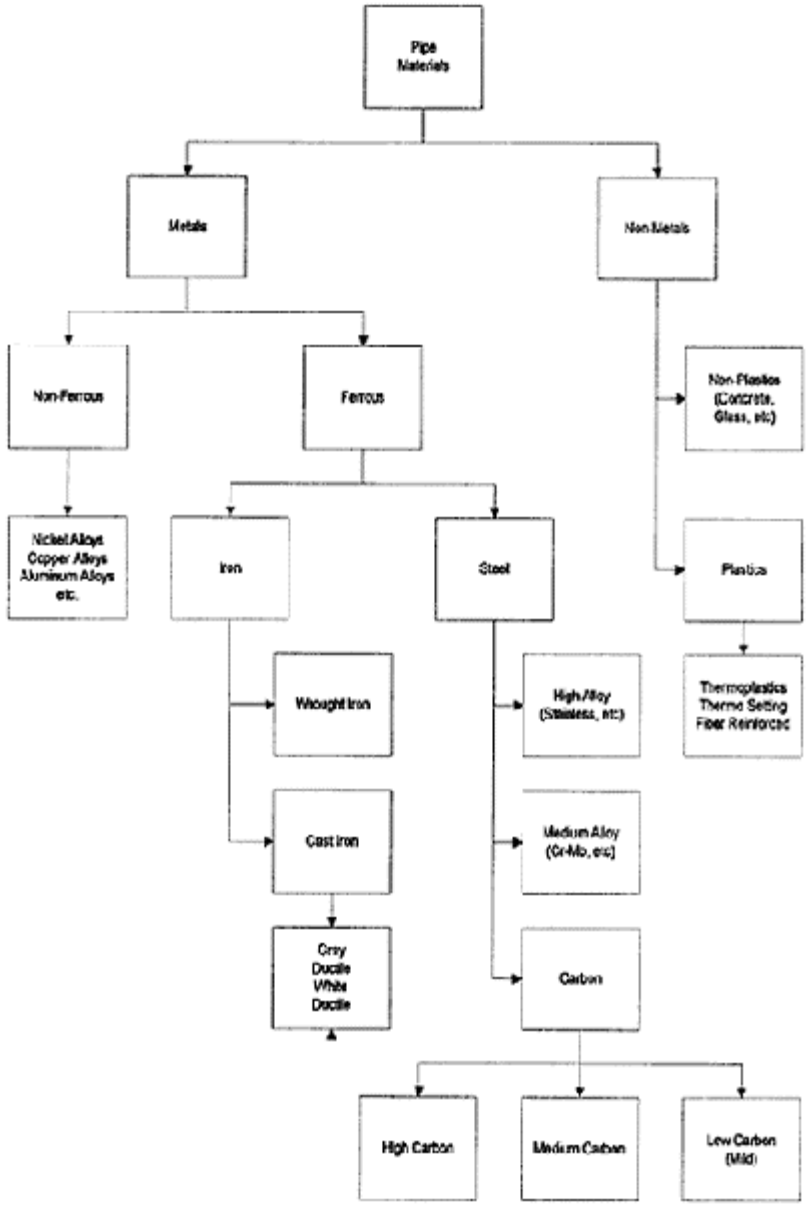


**Figure 3-1** Pipe and Fitting Materials

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## **Table 3-1** Common ASTM Steel Pipe Specifications

| | |
|---|---|
| A 53 | Pipe, steel, black and hot dipped, zinc coated welded and seamless. |
| A 106 | Seamless carbon steel pipe for high temperature service. |
| A 134 | Pipe, steel, electric fusion arc welded (sizes NPS 16 and over). |
| A 135 | Electric resistance welded steel pipe. |
| A 139 | Electric fusion arc welded steel pipes (NPS 4 and over). |
| A 211 | Spiral welded steel or iron pipe. |
| A 252 | Welded and seamless steel pipe piles. |
| A 312 | Seamless and welded austenitic stainless steel pipe. |
| A 333 | Seamless and welded steel pipe for low temperature service. |
| A 335 | Seamless ferritic alloy steel pipe for high temperature service. |
| A 358 | Electric fusion welded austenitic chromium nickel alloy steel pipe for high temperature service. |
| A 369 | Carbon and ferritic alloy steel forged and bored pipe for high temperature service. |
| A 376 | Seamless austenitic steel pipe for high-temperature central station service. |
| A 381 | Metal arc welded steel pipe for use with high-pressure transmission systems. |
| A 405 | Seamless ferritic alloy steel pipe specially heat treated for high temperature service. |
| A 409 | Welded large diameter austenitic steel pipe for corrosive or high temperature service. |
| A 426 | Centrifugally cast ferritic alloy steel pipe for high temperature service. |
| A 430 | Austenitic steel forged and bored pipe for high temperature service. |
| A 451 | Centrifugally cast austenitic steel pipe for high temperature service. |

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| | |
|---|---|
| A 452 | Centrifugally cast austenitic steel cold-wrought pipe for high temperature service. |
| A 523 | Plain ended seamless and ERW steel pipe for high pressure pipe type cable circuits. |
| A 524 | Seamless carbon steel pipe for atmospheric and lower temperatures. |
| A 530 | General requirements for specialized carbon and alloy steel pipe. |
| A 587 | Electric welded low carbon steel pipe for the chemical industry. |
| A 589 | Seamless and welded carbon steel water-well pipe. |
| A 660 | Centrifugally cast carbon steel pipe for high temperature service. |
| A 671 | Electric fusion welded steel pipe for atmospheric and lower temperatures. |
| A 672 | Electric fusion welded steel pipe for high-pressure service at moderate temperatures. |
| A 691 | Carbon and alloy steel pipe electric fusion welded for high-pressure service at high temperature. |
| A 696 | Steel bars, carbon, hot wrought or cold finished, special quality for pressure piping components. |
| A 731 | Seamless and welded ferritic and martensitic stainless steel pipe. |
| A 790 | Seamless and welded ferritic/austenitic stainless steel pipe. |
| A 795 | Black and hot dipped zinc coated (galvanized) welded and seamless pipe for fire protection use. |
| A 714 | High strength low alloy welded and seamless pipe. |
| A 733 | Welded and seamless carbon steel and austenitic stainless steel pipe nipples. |
| A 778 | Welded unannealed austenitic stainless steel tubular products. |
| A 813 | Single or double welded austenitic stainless steel pipe. |
| A 814 | Cold worked welded austenitic stainless steel pipe. |
| A 865 | Threaded couplings, steel black or zinc coated galvanized welded or seamless for use in steel pipe joints. |

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| | |
|---|---|
| A 872 | Centrifugally cast ferritic/austenitic stainless steel pipe for corrosive environments. |

#### *3.1.3.4 High Alloy Steels*

High alloy steels contain over 10% Cr. A common high alloy steel is stainless steel, with a Cr content in the order of 18% [ASTM A 941]. Stainless steels are fabricated either as martensitic steels (for example ASTM A 217 castings), ferritic steels (for example ASTM A 268 Tubing Types 405, 430), or austenitic stainless steels (for example ASTM A 312 or A 376 piping). Type 304 stainless steel (C max 0.08%) is commonly used because it has good corrosion and oxidation resistance, excellent strength and ductility, is easily welded, formed, even cold, and machined. Type 316 is similar to 304 but with more molybdenum, which makes it generally more resistant to sea water, chlorides, and sulfurs; and it exhibits better high temperature properties. The low carbon types (304L and 316L) have a maximum carbon of 0.03%, which is useful in decreasing the precipitation of intergranular carbides and the risk of intergranular corrosion (Chapter 20). Types 321 and 347 are comparable to 304, and generally more resistant to intergranular corrosion.

Steel pipe fittings can be made from forging, bars, plates or pipe. Their chemical and mechanical properties conform to ASTM standards, such as ASTM A 234 for carbon and alloy steels, or ASTM A 403 for austenitic stainless steel, while dimensional and proof test requirements are specified in ASME B16.9 and B16.28 for fittings with butt welded ends, and B16.11 and MSS-SP-79 for fittings with threaded or socket welded ends. Forgings are commonly made of ASTM A 105 or ASTM A 181 carbon steel, and ASTM A 182 alloy steels. Common plate materials used in piping systems include ASTM A 570 for carbon steel and ASTM A 240 for stainless steel.

### 3.1.4 Steel Line Pipe

In the U.S., carbon steel line pipe used in gas and oil pipelines is generally fabricated in accordance with the American Petroleum Institute specification API 5L. In its 42$^{nd}$ edition, the specification introduced two product specification levels: PSL1 is a standard material, which follows the earlier API 5L specification, while PSL2 introduces improvements in weldability (carbon equivalent) and mandatory fracture toughness (minimum Charpy V-notch test and maximum value of yield and tensile strength).

## 3.2 NON-FERROUS PIPE

Non-ferrous pipe and fitting materials are metallic materials with a non-iron matrix; for example, aluminum and its alloys, nickel and its alloys, and copper and its alloys.

### 3.2.1 Aluminum Alloys

Aluminum is obtained by mining and processing aluminum ore (bauxite), which contains aluminum oxide, iron, silicon and impurities. Aluminum is about one third the density of

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steel; it is easily machined or formed and readily welded. It is reactive with oxygen and forms a tough protective oxide layer. However, useful mechanical properties are typically limited to no more than 300°F. Wrought aluminum and alloys are identified by a four-digit number. The 1000 series corresponds to pure aluminum, 2000 series corresponds to Al-Cu alloys, 3000 series corresponds to Al-Mn alloys, 4000 series corresponds to Al-Si alloys, 5000 series corresponds to Al-Mg alloys, 6000 series corresponds to Al-C Mg-Si alloys (with 6061 being a common aluminum pipe material), and 7000 series corresponds to Al-Zn alloys. The four-digit number of Aluminum alloys is usually followed by a letter that identifies the type of heat treatment applied to the material. For example F is as-fabricated, H is strain hardened, W is solution heat-treated, T corresponds to other heat treatment.

### 3.2.2 Nickel Alloys

Nickel is a ductile metal, with high strength, and good corrosion resistance, it is helpful in stabilizing the austenitic structure of stainless steel, which is why close to half the production of nickel is used as stainless steel alloy. Nickel based alloys are valuable in corrosive or high temperature applications, they include Hastelloy$^{R}$ (40% to 60% Ni and 15% to 25% Cr) ASTM B 282, ASTM C 276 (Ni-Mo-Cr); Monel$^{R}$ (66% Ni and 32% Cu) ASTM B 165 (seamless pipe), ASTM B 164 (flange), and ASTM B 366 (fittings); Inconel$^{R}$ (50% to 75% Ni and 10% to 25% Cr) ASTM B 167 (seamless Pipe), ASTM B166 (flange), and ASTM B 366 (fittings). Information on the use of nickel alloys can be obtained from the Nickel Development Institute (NiDI), Toronto.

### 3.2.3 Copper Alloys

Copper (and alloys containing over 90% Cu), bronze (Sn and Cu alloys) and brass (Cu with 20% to 40% Zn) have been used to make pipes as early as 3000 BC because they are soft, easy to form, and corrosion resistant in water service. A common copper tube material is ASTM B 88, available in three tubing sizes: K, L and M, Table 3-2.

**Table 3-2** Some Sizes of Copper Tubing

| Size | OD | Type K wall | Type L wall | Type M wall |
|---|---|---|---|---|
| 1/4 | 0.375” | 0.035” | 0.030” | - |
| 1/2 | 0.625” | 0.049 | 0.040 | 0.028 |
| 3/4 | 0.875 | 0.065 | 0.045 | 0.032 |
| 1 | 1.125 | 0.065 | 0.050 | 0.035 |
| 2 | 2.125 | 0.083 | 0.070 | 0.058 |

## 3.3 FABRICATION OF STEEL PIPE

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### 3.3.1 Pipe Size

Commercial steel pipe is fabricated either by piercing and extruding a hot billet (seamless pipe) or by bending then welding steel plates or skelp (longitudinal or spiral seam welded pipe). In either case, the fabricator produces a pipe with dimensions (diameter and thickness) that comply with a standard, such as ASME B36.10 for carbon steel pipe, ASME B36.19 for stainless steel pipe, API 5L for line pipe. Pipe mills also produce custom sizes, typically in the very large diameters. A standard schedule pipe up to 12" has an inner diameter close to its nominal pipe size (NPS). Pipe 14" and larger has an outer diameter equal to its NPS. Commercial pipe sizes are listed in Appendix A.

Pipes are specified by their nominal size and schedule. Unlike pipes, tubes (or tubing) can have round or square cross section. Cylindrical tubing generally has an outer diameter equal to its nominal size, but not in all cases, as illustrated in Table 3-2. Pipe schedules were introduced in the 1930's in an effort to standardize and replace the designations of Standard (STD), Extra Strong (XS), and Double Extra Strong (XXS), in use since the late 1800's. The schedule number of stainless steel pipe (ASME B36.19) is followed by the letter S, and includes lower schedules with thinner walls than carbon steel pipe (such as schedule 5S and 10S) for low-pressure corrosive service. What is the origin of these schedule numbers? According to the 1955 edition of ASME B36.10, the relationship between pipe size and schedule was originally based on the following formula, a formula that unfortunately works well for certain sizes and schedules but not for others. The question of the origin of the schedule numbers (20, 40, 80, etc.) remains unanswered for now. The 1955 formula is

$$t = \frac{sch}{1000}\frac{D}{1.75} + 0.1"$$

t=pipe wall thickness, in
sch=pipe schedule (ASME B36.10 or B36.19)
D=pipe outside diameter, in

### 3.3.2 Seamless Pipe

Seamless pipe is fabricated by piercing a hot cylindrical billet and forming a seamless tube. The "seamless" fabrication process follows several steps, depending on the applicable material specification. These steps typically include the operations shown in Figure 3-2: (a) a forging is heated to white metal temperature, (b) the white-hot ingot is forged and elongated into cylindrical bars, (c) the white-hot bars are pierced and sized to the right diameter and thickness, (d) the pipe is hydrotested, (e) mechanical properties are verified against the material and procurement specifications, (f) the ends are beveled or threaded, and (g) the pipe is cleaned, marked and readied for shipment. Alloys such as stainless steel would also be passivated (pickling, descaling) by immersion into a warm acid bath, followed by water rinse and drying in air (passivation), the pipe is then measured, weighed and marked.

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### 3.3.3 Seam Welded Pipe

Seam welded pipe is made from skelp (the name given to plate used in pipe fabrication) deformed in an O-shape or spiral shape, then welded along a longitudinal or spiral seam. There are several types of seam welded steel pipe: electric resistance welded pipe, furnace butt welded pipe, arc welded pipe, electric flash welded pipe, and double submerged arc welded pipe. Electric resistance welded pipe (ERW) is made from plate, longitudinally butt welded by heat from electric current, without filler metal. Furnace butt-welded pipe is made from a heated plate drawn through a welding bell and butt welded by compression of the plate edges in the hot furnace. Arc welded pipe is made from plate butt welded by manual or automatic arc, with single or multiple passes on the outside diameter (OD) and inside diameter (ID), with or without filler. Electric flash welded pipe is made from plate, longitudinally butt welded by heat from electric resistance. Double submerged arc welded pipe is made by the submerged arc welding process, typically with passes on the ID and OD.

The fabrication process of seam welded pipe follows the steps illustrated in Figure 3-3: (a) steel plates (skelp) are welded end-to-end and rolled into coils, the outer edges are beveled, (b) plates are bent progressively on a pipe mill into a U then O shape, (c) the seam is welded and heat treated (typically by induction or gas furnace annealing), (d) the weld is inspected (depending on the material specification, seam inspection may be as little as visual or as much as 100% radiography), (e) the pipe is hydrotested, (f) mechanical properties are verified for weld and base material, including tensile properties (yield stress, ultimate strength and elongation at rupture) and ductility by ring crush, (g) the ends are beveled or threaded, and (h) the pipe is cleaned (pickling for stainless steel), measured, weighed and marked. Note that for 6” and larger pipe, a seamwelded pipe with 100% radiography can be a cost-effective alternative to a seamless pipe.

From the 1920’s to as late as the 1980’s in some pipe mills, line pipe (API 5L) was seam welded using low frequency ac current (360 Hz) or dc current. Under these conditions, there must be a very close contact between the electrode and the skelp to achieve continuous fusion. That is why some pipes fabricated during that period exhibit lack of fusion along the seam, referred to as cold welds or stitched welds [Kiefner]. This condition is practically inexistent in modern seam welded pipe using high frequency ac current (in the order of 450 kHz).

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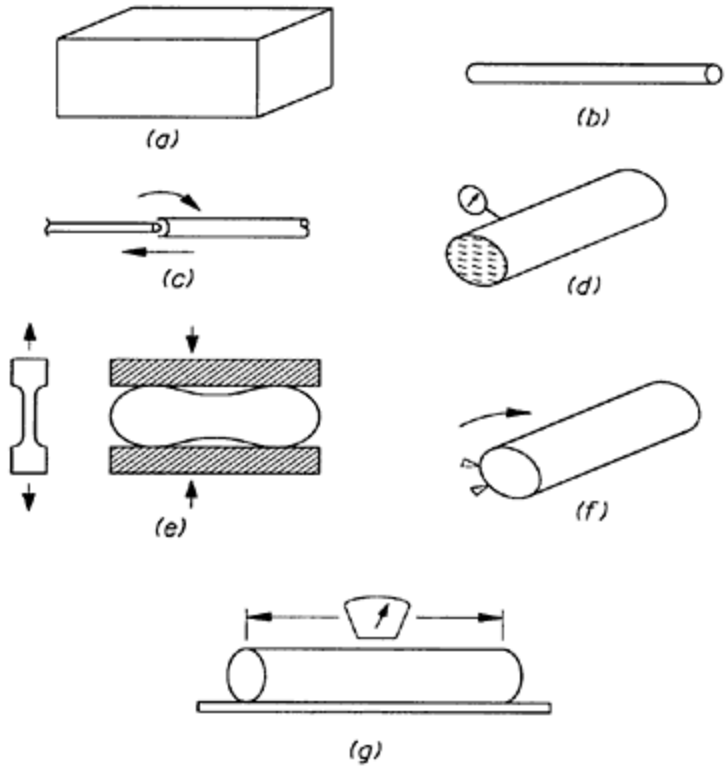


**Figure 3-2** Overview of Seamless Pipe Fabrication

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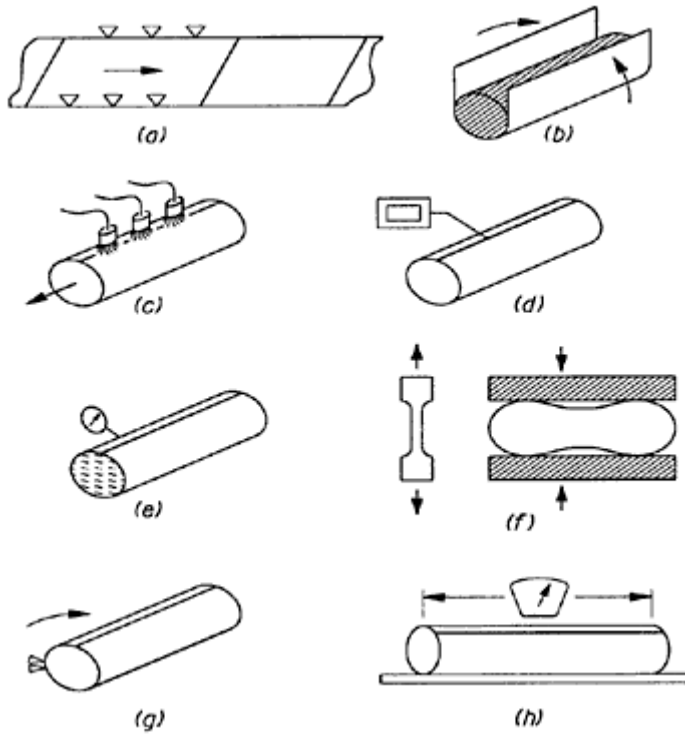


**Figure 3-3** Overview of Seam Welded Pipe Fabrication

### 3.3.4 Documentation

Coming out of the pipe or fitting mill, the finished product is typically traceable to a heat number (steel mill identification) and lot number (pipe mill identification). A heat typically identifies material produced in the steel mill from a single melt. Chemical analysis is usually reported on a heat basis. Parts of a same heat have the same chemistry. A lot generally, but not always, refers to parts subjected to the same fabrication process and finishing treatment (such as same machining, welding, heat treatment). Parts of the same lot therefore have similar metallurgy, chemistry and mechanical properties. The heat number is assigned in the steel mill, while the lot number is assigned in the pipe mill. A lot can be made from one or several heats of material.

A material test report (MTR) documents the chemistry and mechanical properties of the pipe or fitting. It is common practice for the owner to obtain a copy of the MTR for alloy pipe (such as stainless steel or nickel alloys) and carbon steel line pipe (API 5L) and for all materials in critical service. A certificate of conformance (C-of-C) is a statement by the pipe or fitting manufacturer that the product conforms to a certain specification, it does not record the actual chemistry or mechanical properties measured during fabrication. A C-of-C may be sufficient for carbon steel pipe in non-critical applications.

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A positive material identification (PMI) is a check of alloy composition of a metal, possibly using a hand-held alloy analyzer [API 578].

Quality control (the hands-on check of the fabrication process) and quality assurance (clear, streamlined yet complete documentation) are essential. In critical applications (toxic or flammable service, costly or essential components, long lead times, special alloys) nothing replaces knowing first-hand the steel and component suppliers, their experience, expertise, efficiency, pride in their work, and the capability of their pipe and fitting mill. For a facility engineer, the knowledge gained from a one-day visit at a pipe mill is priceless.

### 3.3.5 Microstructure

The fabrication process must deliver a pipe that exhibits three characteristics: (1) chemistry within a specified range, (2) mechanical properties typically above some minimum values, and (3) a sound microstructure.

The microstructure of a metal is the structure of its crystals and grains, which is determined by microscopic examination of a sample of metal. To understand a material's microstructure, consider first what takes place as steel cools down from a molten, liquid state. The liquid metal starts to solidify at a number of points distributed throughout its volume, first at the surface (which is colder) and then towards the center of the ingot or piece, as illustrated in Figure 3-4. Around these scattered nuclei of solid metal, the atoms of iron and alloying elements take their place in a well-structured matrix as they solidify.

As the temperature continues to drop and more metal solidifies, these well structured atomic lattices grow into crystals and grains, Figure 3-4, until all the metal has solidified and the grains have grown to the point where they touch each other, constituting grain boundaries. The atomic structure within a grain and the grain size will depend on several factors, including the chemical composition of the material and its heat treatment.

The equilibrium phase diagram for carbon steel is shown in Figure 3-5. To represent, for example, an ASTM A 106 Grade B pipe material with 0.2% carbon, we place a point on the bottom horizontal line (which corresponds to the ambient temperature) at 0.2% carbon (a point to the extreme left of the %C axis in Figure 3-5). If the pipe is now heated to the melting point, for example during welding, we move vertically up on the phase diagram at 0.2% carbon up to the liquid zone, at approximately 2800°F. As the pipe cools down it will solidify to white metal, which is represented on the phase diagram by moving vertically down from the liquid zone, down the same vertical line at 0.2% carbon. As we reach about 2600°F, we have entered the zone noted "austenite". At this temperature, the hot white metal is solid and atoms of iron in each grain have placed themselves in a face centered cubic arrangement (fcc), as illustrated in the bottom sketch of Figure 3-6, with an atom at each corner of a cube (A) and one at the center of each face (C). The carbon atoms locate themselves between the iron atoms. As the temperature continues to drop, we continue to slide vertically down on the phase diagram at 0.2% carbon.

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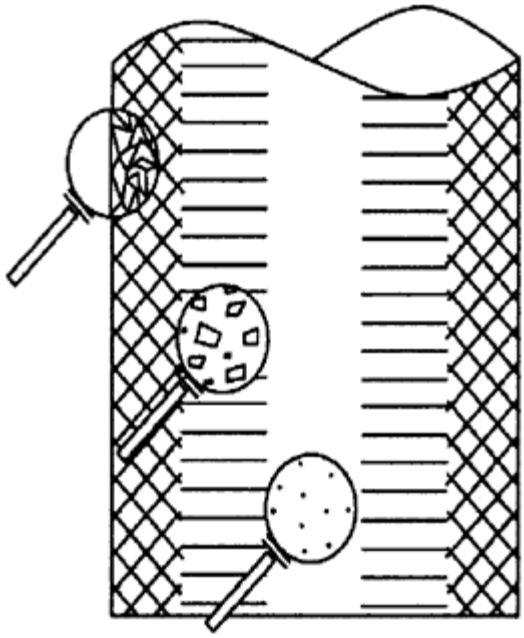

**Figure 3-4** Growth of Atomic Lattice into Grains

Below approximately 1600°F, part of the austenite atomic structure (fcc) evolves into ferrite which is body-centered-cubic (bcc), shown as top sketch of Figure 3-6, with an atom at each corner of a cube (A) and one at the center of each cube (B). The space between iron atoms is now smaller and some carbon atoms are no longer accommodated in the crystalline matrix. They combine with iron to form iron carbide (cementite $Fe_3C$). Steel at room temperature is therefore made of ferrite grains and a mixture of ferrite and cementite called pearlite.

Below 1333°F, and if the cooling process is sufficiently slow (cooling in still air or in furnace) all the austenite has been converted to ferrite (fcc) and cementite. If this cooling process is too rapid, the orderly change of atomic structure will not have time to take place, and a distorted atomic structure, martensite, that is neither bcc nor fcc, will form. Martensite is hard (in the order of Rockwell C 55 and ultimate strength as high as 300 ksi) but it is also brittle, prone to cracking. When welding in-service, the fluid flowing in the line tends to accelerate the cooling process in the weld bead and heat affected zone, forming martensite, which is prone to brittle cracking, particularly in the presence of hydrogen.

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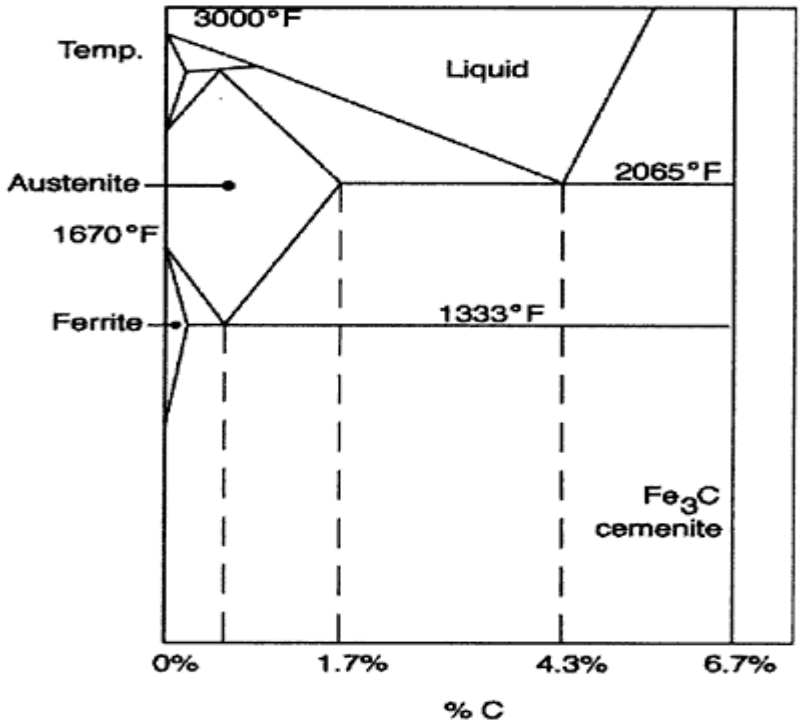


**Figure 3-5** Simplified Phase Diagram of Carbon Steel

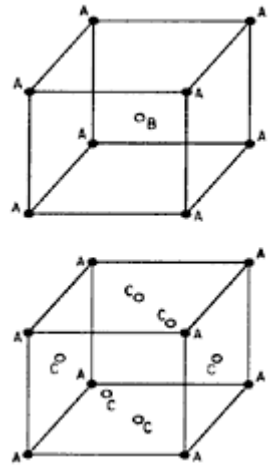

**Figure 3-6** Atomic Structure of Carbon Steel

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We can see that the temperature at which the metal is heated and the speed at which it is cooled down (form very slow if cooled in furnace, to very quick if dropped in water) will affect its atomic structure and grain size and, as a result, its weldability, and its mechanical, metallurgical and corrosion resistant properties. A small grain size results in a more ductile material, with better toughness. Another way to affect grain size is by addition of grain refining elements such as aluminum, columbium (niobium), titanium or vanadium [ASTM A 941]. This steel making practice is called “fine grain practice”. Grain size is measured and assigned a grain size number in accordance with ASTM E 112.

The study of the metal’s microstructure, metallography is performed by optical or electron microscopy. Metallography unveils the metal’s microstructure, its grain morphology as well as its flaws, such as cracks, voids or inclusions. Grain size can be viewed at magnifications of around 100X and classified according to reference comparison standards [ASTM E 112] or by computerized imaging techniques.

## 3.4 FABRICATION OF PIPE FITTINGS AND COMPONENTS

### 3.4.1 Forging

Many pipe fittings and almost all flanges are made by forging. The forging process starts with the fitting fabricator buying bar stock or billets from a steel mill or from a supplier’s warehouse. The chemistry of the initial bars or billets is important since it will not be altered during the fabrication process. The chemistry of the base metal, its alloys and carbon content, is verified by the fitting manufacturer, and sometimes the metallurgy will also be checked. When specified in the order, special chemistry, such as equivalent carbon content, is also verified. If specified by the buyer, corrosion testing in acids or other environment, such as testing of stainless steel for susceptibility to intergranular corrosion [ASTM A 262] usually takes place at the mill’s laboratory.

Mechanical properties (such as yield stress, ultimate strength, elongation at rupture, fracture toughness) will be affected by the forging process, and for that reason they are verified at the end of the forming process. The mechanical properties of a forging can be verified on a section of the actual component, or on a “test bar” that has undergone the same forging and heat treatment process as the part. Soundness of the base metal, the lack of defects or inclusions, is important, particularly at the center of large billets where flaws will not be removed by machining of the surface. In the fabrication sequence, the bar or billet is first cut into disks or smaller bars, as determined by the forging designer, and illustrated in Figure 3-7 (a).

The cut material is then placed in a furnace, Figure 3-7(b), and heated to white metal, then placed either in an open die or a closed die. In open die forging, the piece is pressed while manually rotated and shaped into a round bar or other simple solid shape, or pressed against a die to give it shape. In closed die forging, the piece of hot metal is hammered into shape between a top and bottom contoured die, Figure 3-7(c), then trimmed. The forged shapes are heat treated in a furnace, Figure 3-7(d), and then cooled either rapidly by water quenching, or more slowly in air, or even slower if left to cool in the furnace. Mechanical properties are measured by destructive testing of a sample that

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has undergone the full fabrication process, including heat treatment, Figure 3-7(e). The parts are then machined in mills or lays to the required dimensions and surface finish, Figure 3-7(f). They are cleaned, stamped with standard markings and heat number, coupled to their material test report and then readied for shipment.

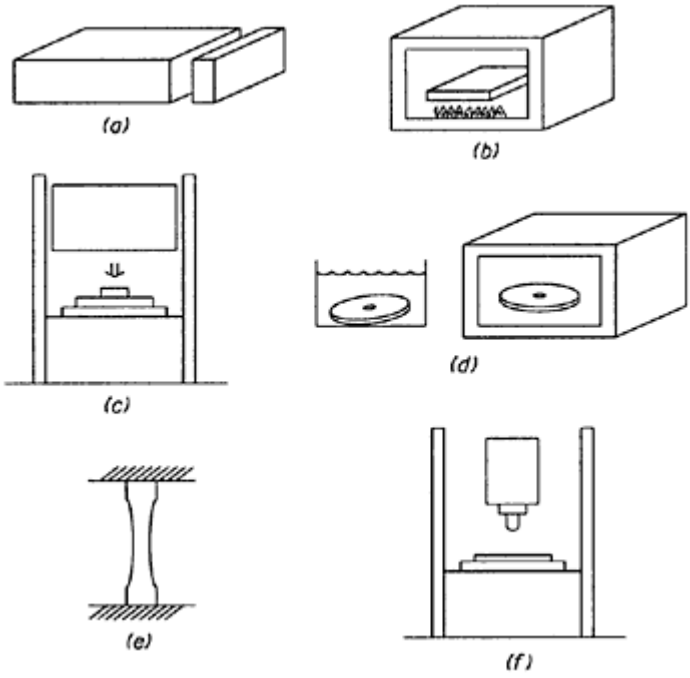


**Figure 3-7** Simplified Diagram of the Forging Process

### 3.4.2 Casting

A casting is made by pouring a molten metal into a mold. The metal then solidifies into the desired shape. Steel castings can be welded [ASTM A 488]. The mechanical properties of castings are usually verified on a tension specimen from each heat. They may be either cut from an appendage to the actual casting or fabricated as a test specimen undergoing the same heat treatments as the casting. Castings for pressure retaining parts are hydrotested in accordance with ASME B16.5 for the applicable pressure rating [ASTM A 703]. Table 3-3 lists some common material specifications for steel castings. The designation of cast stainless steels is based on the Alloy Casting Institute's recommendation [ASTM A 703] and has a form such as CF8M, in which the first letter (here "C") is a service classification letter (such as "C" for corrosion resistant, or "H" for heat resistant above 1200°F). The second letter (here "F") is a letter that indicates the % Cr and % Ni [ASTM A 703], with the first letters of the alphabet A, B, etc. having close to 0% nickel, and the last letters W, X, Y and Z having nickel in the range of 60% to

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80%. In our example “F” indicates 20% Cr with 10% Ni. The following number (here “8”) is the carbon content in units of 0.01%. For example, “8” indicates 0.08% carbon. The last letters (here “M”) are special element letters, such as M (molybdenum), C (colombium), Cu (copper), A (controlled ferrite), or W (tungsten).

**Table 3-3** Specifications for Steel Castings

| | | |
|---|---|---|
| A 27 | - | Steel castings, carbon general application |
| A 128 | - | Steel castings, austenitic manganese |
| A 148 | - | Steel castings, high strength |
| A 216 | - | Steel casting for welding, high temperature (Gr. WCA, WCB and WCC) |
| A 217 | - | Martensitic steel pressure retaining, high temperature (Gr. WC1 to WC11, C5, C12, CA15) |
| A 297 | - | Steel castings Fe-Cr and Fe-Cr-Ni (Gr. HF to HX) |
| A 351 | - | Austenitic and duplex steel castings (Gr. CF3 to CF10, CH8 to 20, CK20, HK30, etc.) |
| A 352 | - | Steel castings ferritic or martensitic low temperature (Gr. LCA to LCC, LC1 to LC9) |
| A 389 | - | Steel castings pressure containing, high temperature (Gr. C23 and C24). |
| A 447 | - | Austenitic steel casting high temperature (Type I and II) |
| A 487 | - | Steel castings for pressure service (Gr. 1 to 16, Classes A to C, Gr. CA15 to 6NM) |

### 3.4.3 Sheet Metal

Sheets are made from hot or cold rolled metal. The cold-formed sheets have a smoother and less oxidized finish than hot-formed sheets. Steel “sheets” are manufactured and specified by their thickness, typically below ¼”, expressed in gage. In contrast, “plates” are specified by their actual thickness in inches. There are several definitions of gage thickness, Table 3-4 is one example.

**Table 3–4** Examples of Sheet Gage Metal Thickness

| Gage | 16 | 14 | 12 | 11 | 10 |
|---|---|---|---|---|---|
| SMACNA (in) | .0538 | .0667 | .0966 | .1116 | .1265 |
| AWS rolled (in) | .0678 | .0747 | .1046 | .1196 | .1345 |

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| | | | | | |
|---|---|---|---|---|---|
| AWS galvanized (in) | .0710 | .0785 | .1084 | .1233 | .1382 |

### 3.4.4 Pipe Specifications

A piping system is made from a number of pipes, fittings and components. Having selected the pipe material and grade, the selection of the matching fitting or component, forging or casting specification and grade is not self evident. Because of the multitude of pipe materials, engineering firms or projects will develop pipe specifications ("pipe specs" or "pipe codes") for different types of services. For example, the pipe specification for a stainless steel system may list the pipe as ASTM A 312 TP304; small bore fittings (2" and smaller) as ASTM A182 F304; large bore fittings (larger than 2") as ASTM A403 WP-304; and flanges as ASTM A 182 F304. An example of pipe specification is given in Chapter 4.

Pipe fittings are fabricated to conform to the dimensional requirements of standards ASME B16 (Chapter 1). Each piping and pipeline engineer should take the time to carefully read at least ASME B16.5 (flanges), ASME B16.9 (butt welding fittings), and ASME B 16.11 (threaded and socket welding fittings). Often times, engineers and mechanics are surprised to find out that these standards do not control every dimension of the fitting; they do specify end-to-end lengths and end bevels, but they will not necessarily specify maximum wall thickness, and much of the fitting quality is left to the competence of the fabricator, with no enforcement authority to verify that a fitting marked ASME B16 does indeed comply with the standard's materials, fabrication, heat treatment and dimensional requirements. The same applies to proof testing. Fittings are qualified by proof testing of a production unit. A good fabricator will repeat this test when changing fabrication methods or parameters. The owner must therefore exercise good judgment in selecting a reliable supplier. Best value (quality/price) is more important than lowest price.

### 3.4.5 Machining and Finishing

Pipe and fittings can be formed, machined or finished in many ways: cutting by hand tools (for example cutting soft tubing, or preparing pipe bevels for welding), drilling using a rotating drill bit (such as drilling of bolt holes in a flange forging), bending (cold bending of tubing or small diameter pipe or hot induction bending of large diameter line pipe). Pipe bending may be accomplished with a mandrel (a solid mandrel with articulated end disks inserted inside the pipe and rotate as the pipe is bent) or with packed sand, or without mandrel or packed sand. Finishing the pipe surface is achieved by grinding, sandblasting, brushing or polishing. The surface finish is typically defined in average surface roughness (arithmetic average of deviations of the actual surface from the mean line) or as the root mean square "rms" (root mean square of deviations of the actual surface from the mean line) [ASME B46.1]. The surface finish is typically expressed in microinches (μin). For example, the typical average surface roughness of a flange face for a metal ring is very smooth at 60 μin or less, while it is serrated at 500 μin for a soft sheet gasket.

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### 3.4.6 Base Metal Imperfections

It is clear from the wide range of fabrication techniques that pipes, fittings (such as tees or elbows) and components (such as valves or strainers) can contain imperfections introduced in the foundry or during the fabrication process. If these imperfections are large they would constitute flaws or defects and the pipe or component would be rejected or repaired if permitted in the material specification.

Flaws, if they do exist, may consist of shrinkage porosities or cracks, inclusions, laminations, laps, seams, holes, or hot tears. The welds of welded pipes or fittings may also include weld flaws such as cracks, lack of fusion or lack of penetration, undercuts, inclusions, cavities, protrusions or misalignments (Chapter 16). But with today's technologies and quality assurance, significant mill flaws that will cause failures or leaks during hydrotest or in-service are very rare.

Non-metallic inclusions (for example oxides, sulfur or manganese) are an example of base material imperfections. They appear as stringers in the skelp. During the making of the plate, these stringers are forced along the grain direction, parallel to the skelp's faces. They become circumferential imperfections when the skelp is bent to form the pipe. If the stringer is at the edge of a plate, it can be pushed radially inward or outward as the two ends are welded together to form the pipe. In this case, the stringer acts as a crack, in the form of a hook, part radial and part circumferential. The skelp edge defect has now become a weld defect and the radial portion of the crack can be the source of failure by crack propagation under large hoop stress or a source of accelerated corrosion called "grooving corrosion" [Kiefner].

While significant flaws (the kind of flaws that must be repaired or could cause failure in service) area a rare occurrence in reputable pipe mills, shallow surface scratches and marring as a result of machining, handling and transport are inevitable. These shallow surface marks are generally acceptable, provided (a) they are not a source of corrosion (for example scratching a stainless steel surface with a carbon steel tool or brush would leave a carbon steel residue on the surface that would form rust), (b) the remaining wall is above the minimum required by specification, (c) the pipe is not intended for high pressure service (hoop stress larger than 50% yield), and (d) they do not affect the flow or process. A gouge (knife-like cut on the pipe surface) can cause rupture in large hoop stress service, as discussed in Chapter 21, and therefore deserves special attention.

## 3.5 MECHANICAL PROPERTIES

Mechanical properties of pipe and pipe component materials consist of strength, hardness, toughness and fatigue strength.

### 3.5.1 Strength

Yield stress, ultimate strength and elongation at rupture are fundamental mechanical properties of pipe and fitting materials. They reflect the ability of the material to be fabricated and to resist applied loads in service. All three properties are essential for piping systems.

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Minimum strength properties are typically required in standard material specifications. For ASTM pipe, a standard size tensile test specimen of pipe material is cut out from the pipe wall or, for small pipe and tubing, made out of a section of pipe or tube [ASTM A 370]. The specimen is placed in a tensile test machine and a steadily increasing tensile force is applied to the specimen, at a rate between 10,000 psi/min and 100,000 psi/min. Passed yield, the maximum strain rate is 0.5/min [ASTM E 8].

For carbon steel, a plot of the engineering stress (force applied to the specimen divided by the initial specimen cross sectional area) versus the corresponding strain (elongation of the specimen divided by its initial length) will have the general shape shown in Figure 3-8. Compilations of stress-strain curves for a wide range of materials are available through ASM International [ASM Atlas]. Such a curve is referred to as the engineering stress-strain curve, to differentiate it from the true stress-strain curve in which the true stress is the applied force divided by the concurrent necked down area of the specimen, and the true strain is the elongation divided by the concurrent length. In the case of Figure 3-8, between 0 psi and approximately 37,000 psi the stress-strain relationship is linear and steep. At around 37,000 psi the material's stress-strain relationship exhibits a marked departure from linearity. The stress at which this occurs is the yield stress, denoted $S_Y$.

For steel, the yield stress is typically determined as the stress at which removing the applied load would result in a permanent elongation of the specimen of 0.2%. For line pipe, yield is defined on the basis of 0.5% permanent elongation [API 5L]. For a uni-axial tensile test, in the elastic region the relationship between engineering stress and engineering strain is

$$\sigma_e = E\,\varepsilon_e$$

$\sigma_e$=engineering stress, psi

$\varepsilon_e$=engineering strain

E=Young's modulus, psi

Young's modulus, named after British doctor, physicist and Egyptologist Thomas Young (1773–1829), is a measure of the elasticity of a material. It varies with temperature, the higher the temperature the softer the material and the lower its Young's modulus, as shown in Table 3-5.

**Table 3-5** Young's Modulus E ($10^6$ psi) for Various Metals at Temperature

| Material | 70 F | 200 F | 300 F | 400 F |
|---|---|---|---|---|
| Cast Iron | 13.4 | 13.2 | 12.9 | 12.6 |
| Carbon Steel C<0.3% | 29.5 | 28.8 | 28.3 | 27.7 |
| Carbon Steel C>3% | 29.3 | 28.6 | 28.1 | 27.5 |
| ½%–2% Cr—Mo | 29.7 | 29.0 | 28.5 | 27.9 |
| 2–1/4%–3% Cr—Mo | 30.6 | 29.8 | 29.4 | 28.8 |
| 5%–9% Cr—Mo | 30.9 | 30.1 | 29.7 | 29.0 |

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| | | | | |
|---|---|---|---|---|
| Austenitic Steels | 28.3 | 27.6 | 27.0 | 26.5 |
| Aluminum 1060 | 10.0 | 9.6 | 9.2 | 8.7 |

Passed the yield point, in the plastic zone, the strength continues to increase before reaching a maximum value, in the case of Figure 3-8 the maximum stress achieved is around 62,000 psi. This maximum is the ultimate strength of the metal, denoted SU. The increase in strength passed $S_Y$ is called strain hardening or work hardening. An increase in strength can also be achieved by heat treatment, referred to as age hardening. Beyond the ultimate strength, the metal loses its strength, the curve dips downward and the tensile specimen ruptures in two. The elongation at rupture, denoted $e_U$, is the last point to the right of the curve, which in the case of Figure 3-8 is approximately 0.21 or 21%. In the plastic region, the relationship between true stress and true strain is

$$\sigma_t = k\,\varepsilon_t^n$$

$\sigma_t$=true stress, psi
$\varepsilon_t$=true strain
k=strength coefficient, psi
n=strain hardening coefficient

The relationship between stress and strain can also be written in terms of Ramberg—Osgood parameters α and n

$$\varepsilon/\varepsilon_o = \sigma/\sigma_o + \alpha(\sigma/\sigma_o)^n$$

ε=strain
$\varepsilon_o$=reference strain=$\sigma_o/E$
E=Young's modulus, psi
σ=stress, psi
$\sigma_o$=0.2% offset yield stress, psi

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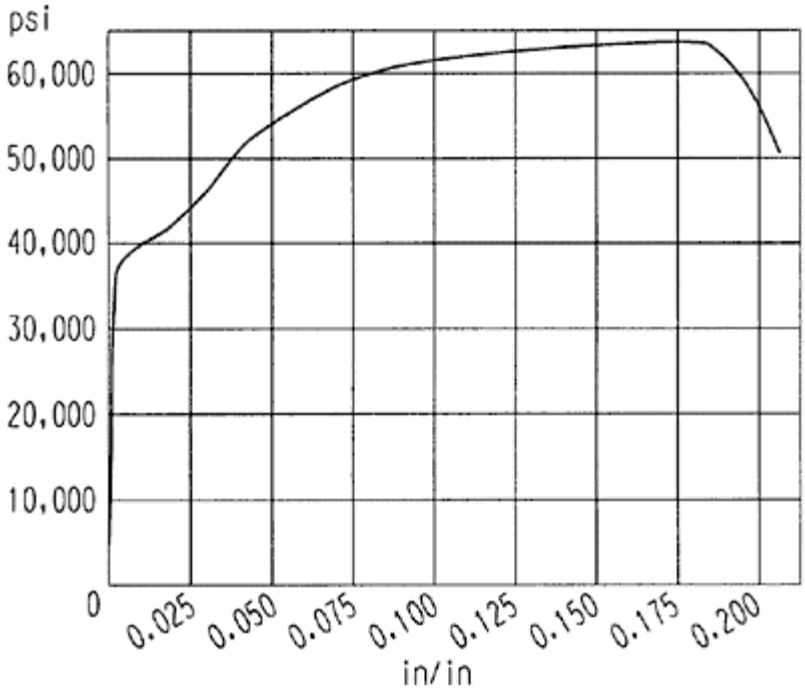


**Figure 3-8** An Engineering Stress-Strain Curve for Carbon Steel

For example, for A 106 Grade B carbon steel pipe at 550°F, $\alpha$=3.80, n= 4.0, $\sigma_o$=37 ksi, and E=27 $10^6$ psi. For type 304 stainless steel at 70°F, $\alpha$=9.16, n=3.20, $\sigma_o$=34.70 ksi, and E=28.3 $10^6$ psi

The yield stress, ultimate strength and elongation at rupture vary statistically, and often range from the specification value as minimum, up to 20% above specification minimum. The yield stress and ultimate strength vary with temperature. The yield stress of carbon steel decreases with increasing temperature, but—surprisingly—its ultimate strength will increase slightly over a range of increasing temperature before it too starts to decrease. Tables 3-6 and 3-7 present the minimum specified yield stress and ultimate strength of several steels [ASME II]. For conservatism in design, the minimum ultimate strength for ASTM A 106 and ASTM A 335 does not reflect hardening with temperature, but is kept constant at 60 ksi between ambient and 400 F.

**Table 3-6** Minimum Yield Stress (ksi) as a Function of Temperature (F)

| ASTM A | 70 F | 200 F | 300 F | 400 F | 1000 |
|---|---|---|---|---|---|
| 106 Gr. B | 35.0 | 31.9 | 31.0 | 30.0 | 19.5 |
| 335 P12 | 32.0 | 28.9 | 27.2 | 26.0 | 20.3 |

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| | | | | | |
|---|---|---|---|---|---|
| 312 Type 304 | 30.0 | 25.0 | 22.5 | 20.7 | 15.6 |

**Table 3-7** Minimum Ultimate Strength (ksi) as a Function of Temperature (F)

| ASTM | 70 F | 200 F | 300 F | 400 F | 1000 |
|---|---|---|---|---|---|
| 106 Gr. B | 60.0 | 60.0 | 60.0 | 60.0 | 34.5 |
| 335 P12 | 60.0 | 60.0 | 60.0 | 60.0 | 48.6 |
| 312 Type 304 | 75.0 | 71.0 | 66.0 | 64.4 | 57.7 |

The mechanical properties of weld material can also be established [AWS]. Typically, a weld is stronger than the base metal because (1) in order to pass the mechanical tests for weld qualification, the weld material is commonly selected to overmatch the base material, (2) the weld material may have additional alloying elements, and (3) in most cases the weld contains residual stresses.

**Table 3-8** Influence of Carbon Content on Mechanical Properties of Steel

| Carbon % | $S_y$ ksi | $S_u$ ksi | $e_U$% | A% |
|---|---|---|---|---|
| 0.06 | 29 | 47 | 46 | 75 |
| 0.21 | 38 | 56 | 42 | 68 |
| 0.38 | 45 | 67 | 35 | 56 |
| 0.60 | 54 | 94 | 20 | 40 |
| 0.89 | 56 | 117 | 13 | 15 |

Although the carbon content in steel is only a fraction of a percent, it has a significant influence on mechanical properties. Table 3-8 illustrates this point. As the carbon content increases, the yield stress ($S_Y$) and ultimate ($S_U$) strength of the steel increase but the elongation at rupture ($e_U$) and the reduction of cross section at rupture (necking, A) decrease. In other words, carbon makes the steel stronger but less ductile.

### 3.5.2 Hardness

Hardness is the resistance of a material to indentation by a hard object of standard shape pushed with a predefined force against the surface. Hardness is an interesting property because it is, indirectly, a reflection of tensile strength and ductility (the harder the material, the higher $S_U$ as shown in Table 3-9, and the lower $e_U$); and, unlike $S_U$, it can be obtained by a surface indentation, without recourse to a destructive test. Hardness is also a good measure of the efficiency of a weld heat treatment process. In certain cases, welding can cause local hard spots (martensitic regions in the case of steel) that are more prone to post-weld or environmental cracking. These hard spots can be eliminated by

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post-weld heat treatment followed by confirmatory hardness testing. With several alloys, the ASME B31.3 code requires heat treatment of weldments to be followed by hardness check to verify that the heat treatment has achieved the objective of softening the weld and heat affected zone, eliminating hard spots. As such, hardness is an indication of the resistance of a weldment to post-weld or environmental cracking.

Hardness can be measured in the laboratory or in the field. It is reported in Brinell, Rockwell or Vickers units. Brinell hardness is obtained with a steel ball, 10 mm in diameter, and an applied load of 500 kg (soft materials) to 3000 kg (hard materials) [ASTM E 10]. Rockwell hardness, Figure 3-9, is obtained with either a steel ball (soft material, Rockwell B) or diamond cone (hard materials, Rockwell C), with a setting load followed by a larger load [ASTM E 18]. Vickers hardness is obtained with a pyramid shaped diamond (laboratory test) [ASTM E 92]. There are also techniques for measuring local micro-hardness.

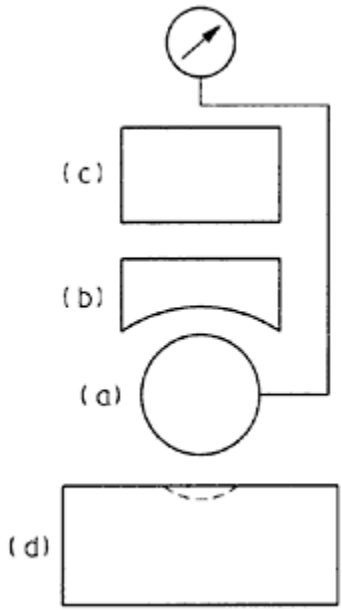


**Figure 3-9** Hardness Test with a Ball (a), Setting Load (b), Large Load (c) Applied to the Specimen (d)

**Table 3-9** Approximate Correlation Hardness—Ultimate Strength [NACE]

| Brinell | Vickers | Rockwell B | Rockwell C | SU ksi |
|---|---|---|---|---|
| 331 | 350 | - | 35.5 | 159 |
| 269 | 284 | - | 27.6 | 131 |

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| | | | | |
|---|---|---|---|---|
| 229 | 241 | 98.2 | 20.5 | 111 |
| 207 | 218 | 94.6 | - | 100 |
| 187 | 196 | 90.9 | - | 90 |
| 167 | 175 | 86.0 | - | 81 |
| 143 | 150 | 78.6 | - | 71 |
| 121 | 127 | 69.8 | - | 60 |
| 111 | 117 | 65.4 | - | 56 |

### 3.5.3 Toughness

Toughness is the ability of a material to absorb impact energy prior to rupture. It is also defined as the material's ability to absorb plastic energy, dynamic or static [ASM Atlas]. It is a function of the material, its temperature and, what somewhat complicates things, its thickness. The thicker the part, the more constrained is the material at its center, and the lower its toughness. The part is too thick and stiff to deform through the thickness, it is in a condition called plane strain. On the contrary, a thinner section of the same material is able to strain outward and the stress is practically constant through-wall, a condition called plane stress. Under internal pressure, a thicker pipe has more strength owing to its wall thickness but a thinner pipe of the same material will exhibit larger plastic deformation before rupture.

This decrease of toughness with wall thickness explains why the ASME code specifies minimum operating temperatures as a function of wall thickness. The thicker the material, the more prone it is to brittle fracture, and the higher its minimum operating temperature. For example, the minimum operating temperature permitted in ASME B31.3 for API 5L X42 is +15°F for t≤0.394" and +70°F for t=1". For ASTM A 106 Grade B it is −20°F for t≤0.5" and +30°F for t=1". For ASTM A 312 type 304 stainless steel it is −425°F regardless of thickness.

#### *3.5.3.1 Charpy V-Notch Toughness*

The failure of materials containing flaws, particularly failure under dynamic load, cannot be understood solely on the basis of the classical strength properties of yield and ultimate strength, even accounting for strain rate effects. To understand these failures, we must understand notch toughness. Notch toughness is the ability of a material to absorb energy in the presence of a sharp notch [Barsom]. It is commonly measured by a dynamic test (Charpy V-notch impact test or drop weight test) or a quasi-static test (fracture toughness test). Charpy V-notch toughness is measured by the strike of a pendulum against a notched specimen [ASTM A 370]. A V-shaped notch is machined in the side of the specimen and the specimen is placed horizontally (Charpy test) or vertically (Izod test) in the test apparatus, Figure 3-10.

The pendulum is swung from an initial position, strikes and ruptures the specimen by a single blow and swings through to a final height. The difference between the initial starting height of the pendulum (initial potential energy) and its maximum swing height

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after impact (final potential energy) is a measure of the impact toughness of the material. It is recorded in units of energy (ft-lb) as a function of the test temperature. Alternatively, the impact toughness of the material can be expressed in percent ductile shear area (fibrous and dull) of the ruptured surface compared to its brittle fracture area (featureless and bright). As can be predicted intuitively, the Charpy V-notch toughness (the energy needed to break a notched specimen by the swing of a Charpy pendulum) decreases with decreasing temperature, as shown in the example of Figure 3-11, with—left to right—a brittle region (lower shelf) at low temperature, a transition region of rising CVN toughness, and a ductile region (upper shelf) at higher temperature. The Nil Ductility Transition Temperature (NDT) is obtained from the inflection point in the curve. Among the many factors that tend to reduce the Charpy V-notch toughness of a material are hydrogen content, high hardness and coarse grain structure.

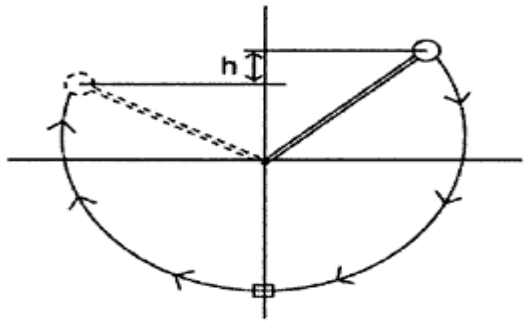


**Figure 3-10** Schematic of a Charpy Test Apparatus

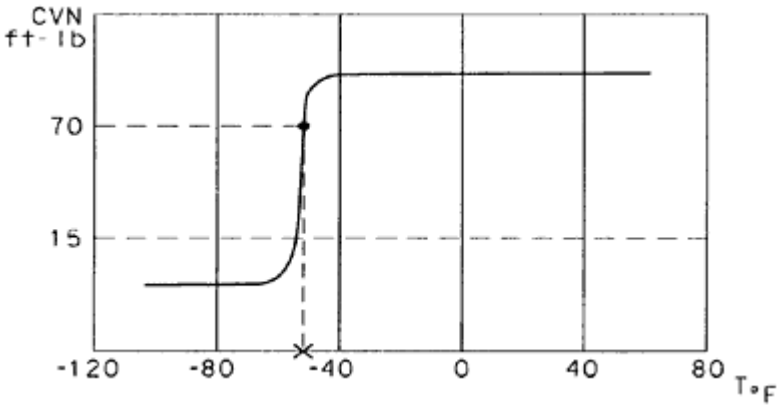


**Figure 3-11** Variation of Charpy Toughness with Temperature—Example

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Material or design codes and standards limit the use of low toughness materials. For example, line pipe may be required to meet API 5L supplementary requirement SR5. In this case, the average of all heats tested for CVN must exhibit a ductile shear fracture appearance over 80% or more of the fractured section at the minimum specified operating temperature (API 5L requirement SR5A). The 80% shear area transition temperature (80% SATT) is the minimum temperature at which a CVN specimen exhibits 80% ductile shear fracture. As an alternative to shear area, the material may be required to exhibit a minimum absorbed energy specified by the buyer, for example 20 ft-lb, at the minimum operating temperature (API 5L requirement SR5B).

A recent specification for a new pipeline project called for a CVN of 140 ft-lb for an API 5L X70 steel gas pipeline to be operated at 1700 psi. The higher the CVN, the better the ability of the pipe to avoid brittle fracture, as will be discussed in Chapter 21. Minimum requirements for pipeline CVN are specified in the gas pipeline code, ASME B31.8. For example, for a methane gas pipeline, the all-heat average of the CVN must meet or exceed a value related to the maximum hoop stress in the pipe, such that

$$CVN > 0.0345\ \sigma^{1.5}\ R^{0.5}$$

CVN=Charpy V-notch toughness at minimum operating temperature, ft-lb

σ=hoop stress in pipe wall due to pressure, psi

R=pipe radius, in

Knowing the minimum operating temperature of a system $T_{min}$, it is important to select a material with a NDT below $T_{min}$; in other words, the lowest operating temperature $T_{min}$ should be in the upper shelf region of the material. For example, cryogenic systems (such as liquid nitrogen) operating at temperatures as low as −350°F will typically be fabricated from stainless steel and not from carbon steel since the NDT of stainless steel is close to −500°F, while it is close to—40°F for carbon steel with standard wall thickness. In low temperature service it is therefore important to verify that pipe and fitting materials are impact tested and exhibit a sufficiently high Charpy toughness. Material specifications that include impact testing are listed in Table 3-10.

**Table 3-10** Specifications for Impact Tested Materials

| Product | ASTM or API |
|---|---|
| Pipe | A333, API 5L |
| Tube | A334 |
| Fitting | A420 |
| Forging | A350 |
| Casting | A352 |
| Bolting | A320 |
| Plate | A20 |

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In practice, it may not be possible to obtain a full size Charpy specimen. In this case full size notch toughness properties must obtained by extrapolation of sub-size specimen test results [Rosenfeld].

### *3.5.3.2 Drop Weight Test*

A second form of dynamic toughness test is the drop weight test. There are primarily three types of ASTM drop weight tests: (1) the nil-ductility drop weight test [ASTM E 208], (2) the drop weight tear test [ASTM E 436], and (3) the dynamic tear test [ASTM E 604]. There is also an API drop weight tear test for line pipe [API 5L3]. The nil-ductility drop weight test is used to establish the ductile-brittle fracture transition temperature, also called nil ductility transition (NDT) temperature. A weld bead is deposited in the center of one face of a 4"× 3.5"×1 or 5"×2"×¾" or 5"×2"×5/8" thick plate of the material to be tested. A 1/16" wide notch is cut in the weld bead to form an initial crack. The plate is simply supported on an anvil, and struck by a guided free falling weight, on the side opposite to the weld bead (the crack in the weld bead is therefore in tension during impact). The test is repeated at 10°F temperature intervals and the material transitions from ductile to brittle. With the drop-weight tear test (DWT) and the dynamic tear test the notch is machined directly into the specimen (there is no weld bead), and the specimen is struck with a falling weight. The material's ductility can then be measured by examination of the fracture surface that will exhibit brittle (cleavage) areas and ductile (shear) areas. For example, line pipe may be required to meet API 5L supplementary requirement SR6, in which case at least 80% of heats tested by DWT testing must exhibit ductile shear fracture over 40% or more of the fractured surface at the minimum specified operating temperature.

### *3.5.3.3 Fracture Toughness*

The fracture toughness of a material, not to be confused with its Charpy V-notch toughness, is a measure of its ability to resist fracture under nearly constant (static) load when the material is cracked. It is an essential property to predict the fracture of a cracked pipe, whether the crack is due to a fabrication flaw or to degradation in service such as fatigue or stress corrosion cracking. A groove is machined in the side of the specimen (Figure 3-12) and a fatigue crack is formed at the crotch of the machined groove. The grooved and cracked specimen is then instrumented and subject to a steadily increasing load until it fractures, as illustrated in Figure 3-12. From this test, an elastic fracture toughness (or "critical crack tip stress intensity") is obtained; it is labeled $K_{IC}$ and is measured in units of ksi(in)$^{0.5}$. The test is performed in an oven, with temperature controls to repeat the test and measure fracture toughness at different temperatures.

In Chapter 21 we will see how fracture toughness is used to evaluate the fitness-for-service of a pipe with a crack.

In addition to the elastic fracture toughness $K_{IC}$, fracture is also measured by crack tip opening displacement CTOD [ASTM E 1290, ASTM E 1820] or J integral for plastic fracture [ASTM E1737 and E1820]. Because there are three modes of crack opening (modes I, II and III, described in Chapter 21) there are three values of the fracture toughness $K_{IC}$, $K_{IIC}$, and $K_{IIIC}$. In pressure equipment, the common failure mode is mode I

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(tensile opening of the crack). Fracture toughness also depends on the rate of applied loading. A dynamic loading will cause the $K_{IC}$ curve to shift to the right, compared to $K_{IC}$ obtained by quasi-static loading, Figure 3-13. In other words, at the same temperature, the metal is more brittle if subject to a dynamic load than a static one.

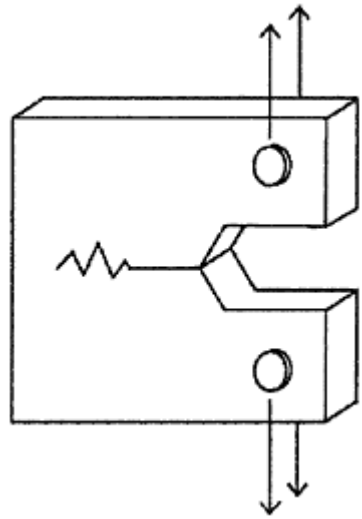

**Figure 3-12** Illustration of a KIC Test Specimen

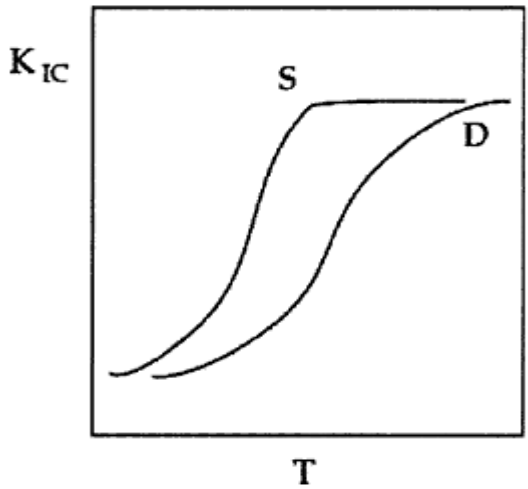


**Figure 3-13** Static (S) and Dynamic (D) Fracture Toughness

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If the material is ductile, fracture will be accompanied by large plasticity, and the elastic-plastic fracture toughness (The J-integral $J_{IC}$) must be considered. In practice, the test procedure to measure $J_{IC}$ [ASTM E 399] is more complex than the $K_{IC}$ test. Under elastic conditions $J_{IC}$ can be obtained from $K_{IC}$ as

$$J_{IC} = \frac{K_{IC}^2}{E/(1-\upsilon^2)}$$

$J_{IC}$=elastic-plastic fracture toughness, ksi.in

$K_{IC}$=linear elastic fracture toughness, ksi(in)$^{0.5}$

E=Young's modulus of the material, ksi

v=Poisson ratio of the material

Because a fracture toughness test requires a sizeable specimen, the machining of a precise notch, the formation of a crack by fatigue, even an elastic fracture toughness test ($K_{IC}$) is relatively difficult to conduct. Unlike the more conventional material properties such as yield stress $S_y$ and ultimate strength $S_u$, it is often difficult to find published values of the fracture toughness $K_{IC}$ for a given material and temperature. At times, it becomes necessary to estimate $K_{IC}$ knowing more conventional material properties such as yield stress $S_y$, ultimate strength $S_u$ or Charpy V-notch impact toughness (CVN). Investigators have developed approximate relationships between fracture toughness and the more common material properties. For example, the Rolfe-Novak-Barsom formula [Barsom, API 579]

$$K_{IC} = S_y\sqrt{\frac{5(CVN)}{S_y} - 0.25}$$

$K_{IC}$=linear elastic fracture toughness, ksi(in)$^{0.5}$

$S_Y$=material yield stress, ksi

CVN=material Charpy V-notch toughness, ft-lb

There are a number of ASTM standards that describe how to conduct fracture toughness tests for Aluminum alloys [ASTM B 645, ASTM B 646, ASTM B 909] and for steels [ASTM E 208, ASTM E 399, ASTM E 812, ASTM E 1221, ASTM E 1290, ASTM E 1304].

### 3.5.4 Fatigue Strength

Fatigue strength is the ability of a material to sustain cyclic stresses without developing a propagating crack. Fatigue strength of base metals and the unique aspects of the fatigue strength of pipefittings are addressed in Chapter 7.

### 3.5.5 Physical Properties

Physical properties include density or specific gravity, Young's modulus and coefficient of thermal expansion.

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The specific gravity is the ratio of a material's mass per unit volume to that of water. Young's modulus is a measure of the material's elasticity, Table 3-5. The coefficient of thermal expansion is a factor, typically labeled α, that relates the thermal expansion ΔL of a material from its original length L, as it is heated an amount ΔT. Some typical values are given in Table 3-11.

The coefficient of thermal expansion α is not a property specified in ASTM material specifications, but it can be obtained for different groups of materials, as a function of temperature from the ASME Boiler & Pressure Vessel Code [ASME II]. The coefficient α is critical in the flexibility analysis of piping systems (Chapter 7).

$$\Delta L=\alpha L \Delta T$$

ΔL=change of length, in
α=coefficient of thermal expansion of the material, 1/°F
L=initial length of the material, in
ΔT=temperature change, °F

**Table 3-11** Coefficient of Thermal Expansion of Some Metals ($10^{-6}$ 1/°F)

| | 100 F | 200 F | 300 F | 400 F |
|---|---|---|---|---|
| CS | 6.50 | 6.67 | 6.87 | 7.07 |
| SS | 8.55 | 8.79 | 9.00 | 9.19 |
| Al | 12.73 | 13.04 | 13.35 | 13.66 |
| Ti | 4.66 | 4.75 | 4.83 | 4.91 |

## 3.6 PROCUREMENT

### 3.6.1 Procurement Specification

Before preparing a procurement specification for pipe, pipe components or fittings, it is advisable to refer to the Pipe Fabrication Institute (PFI) technical bulletin TB3 for responsibilities of the three primary parties: the buyer, the fabricator and the supplier (Chapter 1). The procurement specification for pipe and fittings should address the following points:

Material Specification and Grade: For example, ASTM A 106 Grade B. For valve parts and trims (Chapter 25), or for fittings made of several parts, such as tubing compression fittings (Chapter 18), the material requirements should be explicit for each part or the specification should refer to a vendor catalog model that includes the applicable bill of materials. Soft goods (packing, gaskets, seals, etc.) must also be clearly specified.

Material Specification Supplementary Requirement: Many material specifications have optional supplementary requirements that may be invoked by the buyer. For example, in order to improve weldability, the specification may call out the fifth

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supplementary requirement to ASTM A 106 (S5) which would require the supplier to verify that the chemistry of the supplied pipe has a carbon equivalent CE of 0.5 maximum, where CE is defined as

$$CE = \%C + \frac{\%Mn}{6} + \frac{\%Cr + \%Mo + \%V}{5} + \frac{\%Ni + \%Cr}{15}$$

Owner Supplementary Requirements: The owner may specify additional requirements not part of the standard specification or the standard supplementary requirements. For example, a stainless steel material may be required to be corrosion evaluated in accordance with ASTM A 212 to verify that the particular lot sold is not susceptible to intergranular stress corrosion cracking. Or, the material may be required to have a Charpy V-notch toughness above 15 ft-lb at the minimum temperature defined in the procurement specification.

Material Certificates: The specification should state whether a Certified Material Test Report (CMTR) is to be provided for all materials. The C in CMTR refers to the certifying signature that accompanies the document. Today, many CMTR's are filed and transmitted electronically—without the original signature—and are referred to as MTR's. The MTR is a report of the mechanical properties, chemistry and heat treatment of the material lot supplied, cross-referenced to the heat and lot number of the material. It is particularly useful for alloy steels in corrosive service. On the other hand, MTR's may not be typically available for materials such as copper fittings. An option that may be sufficient in non-critical and non-corrosive applications would be to obtain a Certificate of Conformance, which is simply a formal statement by the supplier that the material meets the specification.

Dimensional Requirements: The purchase order should specify the pipe size (by reference to the schedule or standard sizes of ASME B36.10 for carbon and low alloy steel pipe, ASME B36.19 for stainless steel, API 5L for line pipe, etc.), the fitting rating (for example, the class of an ASME B16.5 flange, the schedule of an ASME B16.9 butt welded fitting, the class of an ASME B16.11 socket or threaded fitting).

Pressure Testing: The first type of pressure test for a pipe is the hydrostatic test conducted at the pipe mill to assure that the component does not rupture or leak through the shell. This is a standard requirement in ASTM or API pipe specifications. For example, ASTM A 106 pipe sections are hydrotested in the mill at a pressure given by

$$P = 2k\frac{S_y t}{D}$$

P=mill hydrostatic test pressure, psi

k=factor equal to 0.6 for carbon steel and 0.5 for stainless steel

$S_y$=minimum material yield stress, psi

t=nominal wall thickness, in

D=pipe outer diameter, in

The second type of test is a leak tightness or seat test for valves, to verify that they do not leak through the seat. When buying a valve the tightness against through-seat leakage must be specified. This is done by choosing one of the leak tightness levels or classes of MSS-SP-61, API 598, or the Flow Control Institute's FCI-70–2(Chapter 25).

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Cleaning: The specification should state that the material must be clean and free of particles, chips, dust, filings, rust, grease, oil, paint, etc. Stainless steel pipe is delivered "pickled" (warm acid descaling, water rinsing and passivated by drying in air). Refer to Pipe Fabrication Institute (PFI) standard ES5 for cleaning.

Surface Finish: This requirement would apply for components such as flange faces where the specification or the applicable standard (in this case ASME B16.5 and MSS-SP-6) specify explicit surface finish requirements.

Coatings and Linings: External coating and internal linings must comply with the material and performance specifications (Chapter 20). They must be accompanied by, or be traceable to, test reports. Coatings and linings are selected by the owner to be compatible with the process fluid, the ambient and the operating parameters (pressure, temperature, flow rate).

Right of Access: The buyer should reserve the right to visit the facilities and observe the operations of the supplier and fabricator.

Markings: There is unfortunately a multitude of standards that specify how a pipe, a fitting or a component, such as a valve, should be marked, and—as a result—it can be quite confusing to decipher the meaning of markings in the field. Examples of markings include:

MSS SP-25 Standard Marking System for Valves, Fittings, Flanges and Unions requires: Manufacturer's name or trademark. Rating designation. Material designation. Melt designation, if required. Valve trim identification. Size. Identification of threaded ends. Ring-joint facing identification.

API 5L Line Pipe requires: Manufacturer. Specification number 5L. Other standard (for API and ASTM dual materials). Size. Weight per foot. Grade class. Manufacture process letter. Heat treatment letters. Test pressure.

ASTM A106 Seamless Carbon Steel Pipe for High-Temperature Service requires: Marking requirements of A530/A530M. Heat Number. Hydrostatic or NDE. "S" for supplementary requirements, if specified. Length of pipe, Schedule. Weight for 4" and larger pipe.

ANSI B16.5 Pipe Flanges and Flanged Fittings requires: Manufacturer's name or trademark. ASTM specification and grade. Rating Class. The mark "B16". The Size.

ASME B16.9 Factory-Made Wrought Steel Buttwelding Fittings requires: Manufacturer's name or trademark. Material and product identification. "WP" in grade symbol. Schedule number or nominal wall thickness. Nominal pipe size.

MSS SP-43 Wrought Stainless Steel Butt-Welding Fittings requires: Manufacturer's name or trademark. "CR" followed by the ASTM or AISI material specification. Schedule number or nominal wall thickness designation. Size.

ASME B16.34 Valves—Flanged, Threaded and Welded End requires: Manufacturer's name or trademark. Valve body material. For cast valves, the heat number and material grade. For forged or fabricated valves the ASTM specification and Grade. Rating. Size.

Packaging and Shipping: The specification should list the materials to be excluded from packages such as lead, mercury, cyanides, tin, etc. For stainless steels a chloride limit would also apply. Also, different materials should be shipped separately. Refer to PFI ES31 for the protection of pipe ends (Chapter 1). Packaged material should be easily traced to lot numbers. Also refer to API 5L1 (Chapter 1) for the rail transport of line pipe.

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Receipt Inspection: Field acceptance should include at least an inspection for damage, cleanliness, identification, markings and traceability to material certificates.

### 3.6.2 Supplier Assessment

It is the responsibility of the owner to carefully select pipe and component suppliers, and verify the quality of delivered goods. Suppliers should be evaluated based on best value (quality/cost ratio) rather than low bid (cost only). The bid evaluation should address the suppliers' experience and provide for a quantitative ranking of their abilities. The ranking system may assign points, for example 0 for poor to 3 for excellent to several factors that affect the quality of fabrication [Billings]. These factors include: experience and reputation, technical competence, written operating and quality control instructions, material controls, welding specifications at work stations, calibrated tools and power supplies, control of welding electrodes, certificates and experience of NDE personnel, independence of quality controls, traceability of materials.

## 3.7 REFERENCES

API 5L, Specification for Line Pipe, American Petroleum Institute, Washington, DC.
API 5L3, Drop Weight Tear Tests on Line Pipe, American Petroleum Institute, Washington, DC.
API 578, Material Verification Program for New and Existing Alloy Piping Systems, American Petroleum Institute, Washington, DC.
ASM, Atlas of Stress-Strain Curves, Boyer, H.E., ed., ASM International, Materials Park, OH.
ASM, Handbook of Corrosion Data, ASM International, Materials Park, OH.
ASM, Metals Reference Book, ASM International, Materials Park, OH.
ASME B36.10, Welded and Seamless Wrought Steel Pipe, American Society of Mechanical Engineers, New York.
ASME B36.19, Stainless Steel Pipe, American Society of Mechanical Engineers, New York.
ASME B46.1, Surface Texture (Surface Roughness, Waviness, and Lay), American Society of Mechanical Engineers, New York.
ASME II, ASME Boiler and Pressure Vessel Code Section II Materials, Part II Properties, American Society of Mechanical Engineers, New York.
ASTM A 48, Standard Specification for Gray Iron Castings, ASTM International, West Conshohocken, PA.
ASTM A 262, Standard Practices for Detecting Susceptibility to Intergranular Attack in Austenitic Stainless Steels, ASTM International, West Conshohocken, PA.
ASTM A 370, Standard Test Methods and Definitions for Mechanical Testing of Steel Products, ASTM International, ASTM International, West Conshohocken, PA.
ASTM A 488, Steel Castings, Welding, Qualification of Procedures and Personnel, ASTM International, West Conshohocken, PA.
ASTM A 644, Standard Terminology for Iron Castings, ASTM International, ASTM International, West Conshohocken, PA.
ASTM A 703, Standard Specification for Steel Castings, General Requirements, for Pressure-Containing Parts, ASTM International, West Conshohocken, PA.
ASTM A 941, Terminology Relating to Steel, Stainless Steel, Related Alloys, and Ferroalloys, ASTM International, West Conshohocken, PA.
ASTM B 645, Standard Practice for Plane-Strain Fracture Toughness Testing of Aluminum Alloys, ASTM International, West Conshohocken, PA.

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ASTM B 646, Standard Practice for Fracture Toughness Testing of Aluminum Alloys, ASTM International, West Conshohocken, PA.
ASTM B 909, Standard Guide for Plane Strain Fracture Toughness Testing of Non-Stress Relieved Aluminum Products, ASTM International, West Conshohocken, PA.
ASTM E 8, Test Methods for Tension Testing of Metallic Materials, ASTM International, West Conshohocken, PA.
ASTM E 10, Standard Test Method for Brinell Hardness of Metallic Materials, ASTM International, West Conshohocken, PA.
ASTM E 18, Standard Test Method for Rockwell Hardness and Rockwell Superficial Hardness of Metallic materials, ASTM International, West Conshohocken, PA.
ASTM E 92, Standard Test Method for Vickers Hardness of Metallic Materials, ASTM International, West Conshohocken, PA.
ASTM E 112, Standard Test Methods for Determining Average Grain Size, ASTM International, West Conshohocken, PA.
ASTM E 208, Standard Test Method for Conducting Drop-Weight Test to Determine Nil-Ductility Transition Temperature of Ferritic Steels, ASTM International, West Consho-hocken, PA.
ASTM E 399, Standard Test Method for Plane-Strain Fracture Toughness of Metallic Materials, ASTM International, West Conshohocken, PA.
ASTM E 436, Standard Test Method for Drop-Weight Tear Tests of Ferritic Steels, ASTM International, West Conshohocken, PA.
ASTM E 604, Dynamic Tear Testing of Metallic Materials, ASTM International, West Conshohocken, PA.
ASTM E 812, Standard Test Method for Crack Strength of Slow-Bend Precracked Charpy Specimens of High-Strength Metallic Materials, ASTM International, West Conshohocken, PA.
ASTM E 1221, Standard Test Method for Determining Plane-Strain Crack-Arrest Fracture Toughness, Kla, of Ferritic Steels, ASTM International, West Conshohocken, PA.
ASTM E 1290, Standard Test Method for Crack-Tip Opening Displacement (CTOD) Fracture Toughness Measurement, ASTM International, West Conshohocken, PA.
ASTM E 1304, Standard Test Method for Plane-Strain (Chevron-Notch) Fracture Toughness of Metallic Material, ASTM International, West Conshohocken, PA.
ASTM E 1737, Standard Test Method for J-Integral Characterization of Fracture Toughness (Discontinued 1998; replaced by E1820), ASTM International, West Conshohocken, PA.
ASTM E 1820, Standard Test Method for Measurement of Fracture Toughness, ASTM International, West Conshohocken, PA.
AWS B4.0 Standard Methods for Mechanical Testing of Welds, American Welding Society, Miami, FL.
AWS D1.3, Structural Welding Code—Sheet Steel, Appendix G Gage Numbers and Equivalent Thickness, American Welding Society, Miami, FL.
Barsom, J.M., Rolfe, S.T., Fracture and Fatigue Control in Structures, ASTM International, West Conshohocken, PA.
Billings, R.M., et. al., World Class Pipe Spool Fabrication, Hydrocarbon Processing, October, 1998.
Kiefner, J.F., Maxey, W.A., Pressure Ratios Key to Effectiveness, Oil & Gas Journal, July 31, 2001.
Kiefner, J.F., Procedure Analyzes Low-Frequency ERW, Flash-Welded Pipe for HCA Integrity Assessments, Oil & Gas Journal, August 5, 2002.
NACE Corrosion Engineer's Reference Book, NACE International, Houston, TX.
Rosenfeld, M.J., Kiefner, J.F., Proposed Fitness-for-Purpose Appendix to the ASME B31 Code for Pressure Piping, October, 1994, the American Society of Mechanical Engineers, New York.

www.TechnicalBooksPDF.com

SMACNA, Rectangular Industrial Duct Construction Standards, Sheet Metal and Air Conditioning Contractors' national Association, Vienna, VA.
Van Droffelaar, H., Atkinson, J.T.N., Corrosion and its Control, An Introduction to the Subject, NACE International, Houston, TX.

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# 4
# Internal Pressure

## 4.1 PRESSURE DESIGN OF PIPING

### 4.1.1 Thin Wall Approximation

Consider a straight section of pipe filled with a pressurized liquid or gas. The internal pressure generates three principal stresses in the pipe wall, as illustrated in Figure 4-1: a hoop stress $\sigma_h$ (also referred to as circumferential or tangential stress), a longitudinal stress $\sigma_l$ (also referred to as axial stress), and a radial stress $\sigma_r$. When the ratio of the pipe diameter to its wall thickness D/t is greater than 20 the pipe may be considered to be thin wall [Cooper, BS 8010]. In this case, the hoop stress is nearly constant through the wall thickness and equal to

$$\sigma_h = \frac{PD}{2t}$$

P=design pressure, psi
D=outside pipe diameter, in
t=pipe wall thickness, in

The longitudinal stress is also constant through the wall and equal to half the hoop stress

$$\sigma_l = \frac{PD}{4t}$$

The radial stress varies through the wall, from P at the inner surface of the pipe to zero on the outer surface.

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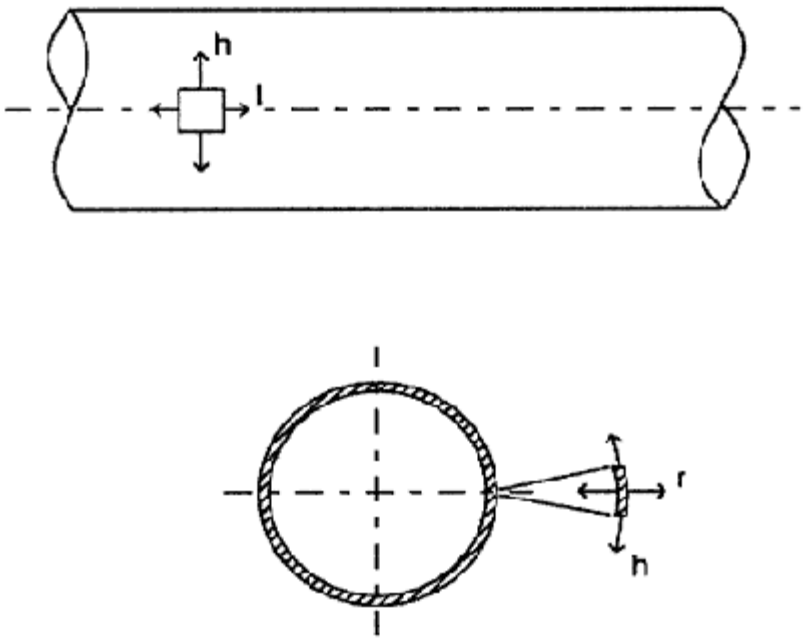


**Figure 4-1** Hoop (h), Longitudinal (l) and Radial (r) Stress Directions

### 4.1.2 Pipeline Design Equation

For oil and gas pipelines, the thickness of the pipe wall is obtained by writing that the hoop stress, which is the largest stress in the pipe, must be limited to a certain allowable stress S. Using the thin wall approximation, this condition corresponds to

$$\frac{PD}{2t} < S$$

P=internal design pressure, psi
D=pipe outer diameter, in
t=pipe wall thickness, in
S=allowable stress, psi
For hazardous liquid pipelines (hydrocarbon, carbon dioxide, etc.), the allowable stress is set at [ASME B31.4]

$S=0.72\ S_Y\ E$

0.72=design factor
E=longitudinal weld joint factor, Table 4-1
$S_y$=specified minimum yield strength, psi, Table 4-2
For gas pipelines, the allowable stress is [ASME B31.8]

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$S=S_Y F E T$

P=design pressure, psi

D=nominal outside diameter, in

$S_y$=specified minimum yield stress, psi, Table 4-2 (commonly referred to as SMYS in the pipeline industry)

F=design factor, Table 4-3

E=weld joint factor, Table 4-1

T=temperature derating factor, Table 4-4

There are two equally plausible versions of the origin of 0.72 $S_y$. The first explanation is that 0.72 $S_Y$ goes back to the early days of fabrication of steel line pipe. In the mill, the pipe was tested to a hydrostatic pressure causing a hoop stress PD/(2t) of 90% $S_y$. In service, the pressure was limited to 80% of the mill hydrotest pressure, or 80%×90% $S_y$=72% $S_Y$. The second explanation is that the 90% $S_Y$ hydrostatic test was reduced by 12.5% for fabrication tolerance on underthickness, then further divided by 1.1 to compensate for the 110% overpressure transient allowance (as was the common practice for water pipelines), which leads to 90% $S_Y$×0.875/1.1=0.72 $S_Y$.

The weld quality or joint efficiency factor E is a factor introduced to account for the quality of the longitudinal or spiral seam in a pipe. It is a function of the reliability and quality of fabrication and the extent of inspections performed in the pipe mill. An electric resistant welded pipe is judged to have a superior seam quality, and its weld joint efficiency factor is assigned the maximum value 1.0. On the other hand, the seam weld of a furnace butt-welded pipe was judged to have a seam weld factor of only 0.6. These values were established decades ago and were based on experience with the various methods of pipe fabrication at the time. The quality of U.S. pipe fabrication and mill inspections have greatly improved since these values were first established, and today the lower E values (which penalize the wall thickness) may well be too restrictive.

### 4.1.3 Yield and Wall Thickness

At ambient temperature, the yield stress used in the minimum wall design equation is the minimum value of the material's yield stress obtained from the pipe material specification.

**Table 4-1** Examples of Longitudinal Weld Joint Factors E [ASME B31.8]

| Material | Pipe Class | E |
|---|---|---|
| ASTM A 53, A106 | Seamless | 1.0 |
| ASTM A 53 | ERW | 1.0 |
| ASTM A53 | Furnace Butt Welded | 0.6 |
| ASTM A 134 | Electric Fusion Arc Welded | 0.8 |
| ASTM A 135 | Electric Resistance Welded (ERW) | 1.0 |
| API 5L | Seamless | 1.0 |

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| | | |
|---|---|---|
| API 5L | Submerged Arc Welded or ERW | 1.0 |
| API 5L | Furnace Butt Welded | 0.6 |

**Table 4-2** Examples of Yield and Ultimate Stress [ASME II Part D]

| Temperature (°F) | A 106 Gr.B $S_v$ [ksi] | A 106 Gr.B $S_u$ [ksi] | A 312 T.304 $S_v$ [ksi] | A 312 12 T. 304 $S_u$ [ksi] |
|---|---|---|---|---|
| 100 | 35.0 | 60.0 | 30.0 | 75.0 |
| 200 | 31.9 | 60.0 | 25.0 | 71.0 |
| 300 | 31.0 | 60.0 | 22.5 | 66.0 |
| 400 | 30.0 | 60.0 | 20.7 | 64.4 |
| 500 | 28.3 | 60.0 | 19.4 | 63.5 |

**Table 4-3** Location Design Factor F [ASME B31.8]

| Location | |
|---|---|
| Class 1 Div. 1: Deserts, farm land, sparsely populated, etc. | 0.8 |
| Class 1 Div.2: Class 1, with line tested to 110% design. | 0.72 |
| Class 2: Industrial areas, town fringes, ranch, etc. | 0.6 |
| Class 3: Suburban housing, shopping centers, etc. | 0.5 |
| Class 4: Multistory buildings, heavy traffic, etc. | 0.4 |

Note: Lower location design factors apply at crossings, compressor stations, etc. The pipeline designer must refer to codes and regulations for the applicable location design factor.

**Table 4-4** Temperature Derating Factor [B31.8]

| Temperature (°F) | T |
|---|---|
| 250 or less | 1.0 |
| 300 | 0.967 |
| 350 | 0.933 |
| 400 | 0.9 |
| 450 | 0.867 |

For older pipelines, where the material specification or the pipe wall thickness is unknown, the Code of Federal Regulations [CFR 49] permits the use of an estimate of yield stress and wall thickness. The yield stress $S_y$ can be based on measured values, in

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which case it is the smallest of 80% of the average measured yield stress, but not more than 52 ksi. The wall thickness t could be the average of four measurements at the pipe end. For example, if we measure 0.333", 0.333", 0.330", 0.330", the average is 0.3315", and the pipe is assigned a schedule 40 with a wall of 0.322". For pipe smaller than 20" nominal pipe size (NPS) the chosen wall (0.322") may not be larger than 1.14 times the smallest measured wall, in our case 1.14×0.330"=0.376". For pipe 20" or more use 1.11 in place of 1.14. Since 0.322"<0.376" the choice of a schedule 40 is valid. The material specification for line pipe [API . permits wall under-thickness ranging from −8% to −12.5% depending on the pipe size and grade. For example, a 20" pipe grade X42, ordered to a nominal wall of 0.5" can have an actual wall 8% smaller or 0.46". The B31.4 and B31.8 pipeline allowable stress of 72%$S_y$ accounts for a 12.5% fabrication under-tolerance.

## 4.2 PRESSURE DESIGN OF PLANT PIPING

### 4.2.1 Lamé's Formula

Without the thin wall approximation, the more general form of the three principal stresses in a closed cylinder subject to internal pressure P is given by Lamé's formula [Den Hartog]

$$\sigma_t = P\frac{r_i^2}{r_o^2 - r_i^2}\left(1 + \frac{r_o^2}{r^2}\right)$$

$$\sigma_r = P\frac{r_i^2}{r_o^2 - r_i^2}\left(1 - \frac{r_o^2}{r^2}\right)$$

$$\sigma_l = P\frac{r_i^2}{r_o^2 - r_i^2}$$

$\sigma_t$=tangential (hoop) stress, psi
$\sigma_r$=radial stress, psi
$\sigma_l$=longitudinal (axial) stress, psi
$r_i$=inner pipe radius, in
$r_o$=outer pipe radius, in
r=radial distance of a point in the pipe wall, in

### 4.2.2 Early Design Equations

Can Lamé's formula be used to design piping systems? This question became of particular interest for the design of boilers in the early 1900's when the use of steam engines was quickly expanding and with it the use of higher steam pressures. At that

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time, several equations were used in the design of boilers, vessels and piping [Parsons]. The wall thickness rule applied by the U.S. Board of Supervising Inspectors of Steam Vessels in the early 1900's was

P=kSt/(3D)

P=working pressure, psi
k=1 for single riveted, 1.2 for double riveted vessels
S=tensile strength, psi
t=wall thickness, in
D=mean diameter, in
Lloyd's rule was
P=C(T−2)B/D

C=safety factor
T=thickness in sixteenths of an inch
B=least percentage of strength of joint
The Board of Trade rule was
P=SB2t/(DC)

British Corporation's rule was
P=C(T−1)B/D

Let's use the U.S. Board of Supervising Inspectors of Steam Vessels as an example. With this rule, a double riveted 60" diameter vessel, 0.5" thick, with an ultimate strength of 50,000 psi would be permitted to operate at a pressure P=1.2 ×50,000×0.5/(3×60)=167 psi. The hoop stress is PD/(2t)=10,020 psi, five times less than the ultimate strength.

The factor of safety of 5 is consistent with the pre-World War II rules of the ASME pressure vessel code. Because of the shortage of steel during World War II, the safety factor was reduced to 4 in 1944, and then increased back to 5 until 1951. It was then permanently reduced to 4, until the late 1990's when it was further reduced to 3.5 in ASME B31.1 and ASME VIII Division 1.

### 4.2.3 Piping Design

In 1951, in an attempt to make the design process more uniform, a task group of the American Society of Mechanical Engineers investigated a number of pressure stress equations and failure criteria. In all, thirty different pressure design formulas and approximations were compiled. Seeking a balance between accuracy and simplicity, the task group proceeded as follows [Burrows]. It was first assumed that as the pipe deforms under pressure, the material maintains a constant volume. This corresponds to a material Poisson ratio v of 0.5, in which case, the Saint Venant, Tresca and Von Mises equivalent stress can be written in a similar form:

Maximum strain energy (Saint Venant):

$$\sigma = \frac{3}{4}(\sigma_t - \sigma_r) = \frac{\sigma_t - \sigma_r}{1.33}$$

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Maximum shear stress (Tresca):

$$\sigma = \sigma_t - \sigma_r = \frac{\sigma_t - \sigma_r}{1.0}$$

Maximum energy (Von Mises):

$$\sigma = \sqrt{\frac{3}{4}}(\sigma_t - \sigma_r) = \frac{\sigma_t - \sigma_r}{1.15}$$

All three stress expressions can be written in the form

$$\sigma = \frac{\sigma_t - \sigma_r}{K}$$

with K=1.0, 1.15 or 1.33. In addition, the average principal stresses through the wall are

$$\sigma_{t,avg} = P\left(\frac{D}{2t} - 1\right)$$

$$\sigma_{r,avg} = -\frac{P}{2}$$

$$\sigma_{l,avg} = P\left(\frac{D}{4t} - 0.75\right)$$

Substituting, we obtain

$$\sigma = \frac{\sigma_{t,avg} - \sigma_{r,avg}}{K} = \frac{P}{K}\left(\frac{D}{2t} - 0.5\right)$$

In 1943, a factor of 0.4 was recommended in place of 0.5 in the stress formula [Boardman]. The task group adopted the recommendation, and selected the maximum shear stress failure criterion, and therefore K=1, which led to

$$\sigma = P\left(\frac{D}{2t} - 0.4\right)$$

At this point, questions were raised regarding the applicability of this equation to high temperature steam service. In light of burst test data available at the time, the task group judged that the choice of the 0.4 factor, while adequate for temperatures up to 900°F, would result in excessively thick pipe for service above 900°F. This would in turn lead to unnecessarily heavy and costly pipe, less flexibility to absorb thermal expansion and larger through-wall thermal gradients. To avoid these difficulties, and based on burst test results, it was decided that between 900°F and 1150°F the 0.4 factor would be gradually

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increased to 0.7. This was the origin of the y factor in today's ASME B31.1 and ASME B31.3 design equations. The values of y are listed in Table 4-5:

$$\sigma = P\left(\frac{D}{2t} - y\right)$$

**Table 4-5** Coefficient y for t<D/6 for Temperatures T (°F) [ASME B31.3]

| Material | T≤900 | 950 | 1000 | 1050 | 1100 | ≥1150 |
|---|---|---|---|---|---|---|
| Ferritic Steel | 0.4 | 0.5 | 0.7 | 0.7 | 0.7 | 0.7 |
| Austenitic Steel | 0.4 | 0.4 | 0.4 | 0.4 | 0.5 | 0.7 |
| Ductile Metals | 0.4 | 0.4 | 0.4 | 0.4 | 0.4 | 0.4 |
| Cast Iron | 0.0 | - | - | - | - | - |

Setting the stress σ equal to its maximum allowable value S multiplied by the weld quality factor E, and rearranging the terms, we obtain the ASME B31.1 and ASME B31.3 design equation

$$t = \frac{PD_o}{2(SE + Py)}$$

t=minimum required wall thickness, excluding manufacturing tolerance and allowances for corrosion (in)
P=internal design pressure, psi
$D_o$=outside diameter of pipe, in
E=joint efficiency factor
y=temperature coefficient (Table 4-5)
S=maximum allowable stress in material, psi

### 4.2.4 Allowable Stress

The allowable stress for pipelines is 72%$S_y$ and does not depend on the material's ultimate strength. The allowable stress for power and process plant piping systems is

$S(T)=\min. \{S_y(T)/SF_y; S_u(T)/SF_u\}$

S(T)=allowable stress at design temperature T, psi
$SF_y$=safety factor applied to yield stress
$SF_u$=safety factor applied to ultimate strength
$S_y(T)$=minimum specified yield stress at design temperature T, psi
$S_u(T)$=minimum specified ultimate strength at design temperature T, psi
For carbon steel pipe in ASME B31.1 applications
$S(T)=\min. \{2\ S_y\ (T)/3; S_u(T)/4\}$

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For carbon steel pipe in ASME B31.3 applications

$S(T)=\min. \{2 S_y (T)/3; S_u (T)/3\}$

For austenitic stainless steel in ASME B31.1 or B31.3 applications

$S(T)=\min. \{90\%S_y(T); Su(T)/3\}$

Where the values of yield stress $S_y$ or ultimate strength $S_u$ at design temperature are larger than at room temperature, the room temperature values are used. Some values of allowable stress are listed in Table 4-6.

**Table 4-6** ASME B31.3 Allowable Stress

| Material | 100°F | 200°F | 300°F | 400°F | 500°F |
|---|---|---|---|---|---|
| A 106 Gr.B | 20.0 | 20.0 | 20.0 | 20.0 | 18.9 |
| API 5L X52 | 22.0 | 22.0 | 22.0 | 22.0 | - |
| A 312 Type 304 | 20.0 | 20.0 | 20.0 | 18.7 | 17.5 |
| B 241 6061 T6 | 12.7 | 12.7 | 10.6 | 5.6 | - |

In the late 1990's, the safety factor of 4 against ultimate strength, used in the design of B31.1 carbon steel pipe, and ASME B&PV Section VIII Division 1 pressure vessels, was reduced to 3.5. A larger safety factor $SF_u$ applies to materials less ductile than steel. For example the allowable stress for cast iron is $S_u/10$, and the allowable stress for malleable iron is $S_u/5$.

### 4.2.5 Wall Thickness Allowance

Having established the minimum required wall thickness, the designer should add a corrosion allowance and, for piping systems but not for pipelines, a fabrication tolerance. The minimum wall thickness required by code plus allowances is then rounded up to obtain the commercial pipe size to be procured and used in construction.

It is up to the designer to select the corrosion allowance, based on experience with similar fluids, pipe materials, temperatures and flow rates (Chapter 20).

The tolerance on wall thickness depends on the pipe material specifications. For example, an ASTM A106 carbon steel pipe may be furnished 12.5% below the specified nominal pipe wall thickness. Therefore, the minimum pipe wall thickness $t_{min}$ calculated by the code design equation needs to be increased by the corrosion allowance C and the fabrication tolerance f (for example with a fabrication tolerance of 12.5% on the pipe wall thickness, f=0.125). With these corrections, we now obtain the commercial pipe size to be procured:

$t=(t_{min}+C)(1+f)$

t=pipe wall thickness, is

$t_{min}$=minimum wall required by Code, in

C=corrosion or threading allowance, in

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f=pipe wall thickness fabrication tolerance

## 4.3 YIELD AND BURST PRESSURE

### 4.3.1 The Von Mises Yield Pressure

Applying the Von Mises criterion, yielding of the pipe wall will take place when the distortion energy reaches a certain limit value X. This can be written as

$(\sigma_h-\sigma_1)^2+(\sigma_1-\sigma_r)^2+(\sigma_r-\sigma_h)^2=X$

We can find the limit value X through a simple tensile test. In this case, $\sigma_h= \sigma_R=0$ and $\sigma_1=F/A$ is the ratio of the applied tensile force F to the metal area A. Yielding will take place when $\sigma_1=S_Y$, in which case the Von Mises criterion can be written as

$(0-S_Y)^2+(SY-0)^2+(0-0)^2=X=2S_Y^2$

By substitution, the internal pressure at which the pipe wall yields is

$$P_y = \frac{S_Y}{\sqrt{\frac{3}{4}\left(\frac{D}{2t}\right)^2 + \frac{3}{2}\left(\frac{D}{2t}\right)+1}}$$

$P_y$=internal pressure at onset of yield, psi

For large diameter to thickness ratio (D/t>>1) we obtain the internal pressure at the onset of yield [Cooper, Sims]

$$P_y = \frac{4tS_Y}{\sqrt{3}D}$$

### 4.3.2 Burst Pressure

As the internal pressure continues to increase beyond the yield pressure $P_y$, the pipe wall will bulge outward and reach a point of instability. In reality, the material is not perfectly uniform and this bulging does not take place exactly uniformly around the circumference but preferentially on one side of the pipe wall. The hoop strain at which instability occurs is [Cooper]

$\varepsilon_i=n/2$

$\varepsilon_i$=strain at onset of instability

n=strain coefficient, from true stress-strain equation (Chapter 3)

Soon after instability (outward bulge in pipe wall), the pipe wall ruptures. The pressure at rupture is the ultimate pressure $P_u$ given by [Cooper]

$P_u=(2\ k\ t/D)e^{-n}\ \{n/[2\ (3/4)^{(1+n)/n}]\}^n$

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$P_u$=ultimate pressure at burst, psi
t=pipe wall thickness, in
k=strength coefficient, psi
D=pipe outer diameter, in

For example, a 6" sch. 40 carbon steel pipe has an actual yield stress of 40 ksi, an ultimate strength of 70 ksi, a strain coefficient n=0.2, a strength coefficient k=100 ksi and an elongation at rupture of 40%. Substituting, we obtain $P_y$= 3.9 ksi and $P_u$=6.2 ksi. Compare these values with the pressure corresponding to a hoop stress equal to the ultimate strength $P=2S_ut/D$=5.9 ksi. The difference between 6.2 ksi and 5.9 ksi is due to the fact that $2S_ut/D$ is an approximation based on an elastic prediction of burst, ignoring plasticity.

## 4.4 PRESSURE DESIGN OF PLASTIC PIPE

The design of plastic pipe is covered in Chapter 24.

## 4.5 PRESSURE DESIGN OF FITTINGS

### 4.5.1 Pressure Rating

The hoop stress distribution in pressurized fittings and components, such as tees, reducers, elbows, nozzles, etc., can not be expressed by a simple equation such as PD/(2t) for the hoop stress in a straight pipe. To eliminate the need for complex design calculations to size fittings, the pressure design of fittings and components relies on a simple approach: first, fittings and components must meet standard dimensions specified in the ASME B16 dimensional standards, and second, fittings and components must be pressure rated, by means of proof tests.

The fittings and components will then be assigned a pressure rating or, for butt-welded fittings (ASME B16.9), they are simply designated by the schedule number of the matching pipe. There are two methods of pressure rating: (1) The method of the ASME Boiler & Pressure Vessel (B&PV) Code and (2) the method of MSS-SP-97 and ASME B16.9. The choice between these two methods is dictated by the applicable piping design code. For example, B31.1 applies the ASME B&PV Section I method for pressure rating, while B31.3 allows the use of either the ASME B&PV Section VIII or the MSS or B16 methods in rating pipefittings and components. The ASME B&PV method for pressure rating is specified in Section I Appendix A-22 for boilers and Section VIII Division 1 Section UG101 for pressure vessels. In the rating process the fitting is subjected to a steadily increasing pressure. At one point, let's say when the pressure has reached a value H, the fitting will start to visibly deform. The "proof yield pressure" is defined as

$P_Y=(H/2)\ (S_{Y,min}/S_{Y,tested})$

$P_Y$=proof yield pressure, psi
H=test pressure at onset of visible yield, psi
$S_{y,min}$=minimum specified yield stress of the material, psi

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$S_{y,tested}$=actual yield stress of the test specimen, psi

As the proof test continues, the pressure will eventually burst the fitting at a pressure we call B. In the pressure vessel rules (ASME VIII) "proof burst pressure" is the burst pressure divided by a safety factor of 5, and corrected for material properties

$P_B=(B/5)\ (S_{u,min}/S_{u,tested})$

$P_B$=proof burst pressure, psi

B=burst pressure of the test specimen, psi

$S_{u,min}$=minimum specified ultimate strength of the material, psi

$S_{u,test}$=actual ultimate strength of the test specimen, psi

When applying the boiler and pressure vessel proof burst formula to piping components, the piping design code safety factor may be used in place of 5. For example, a safety factor of 3 could be used for process piping since the allowable stress is based on $S_u/3$. For cast iron the safety factor would be 10, and for malleable iron it would be 5. The ratio $S_{u,min}/S_{u,test}$ is meant to correct the proof burst pressure to compensate for the fact that the specimen tested is, in most cases, stronger (larger ultimate strength $S_{u,test}$) than the material specification strength which is a minimum $S_{u,min}$. In MSS-SP-97 or B16.9, the proof burst pressure can be based on the burst test of a single specimen, as

$P_B=[B/(3\times1.05)]\ (S_{u,min}/S_{u,tested})]$

$P_B$=proof burst pressure, psi

$S_{u,min}$=minimum ultimate strength of the material specification, psi

$S_{u,test}$=actual ultimate strength of the tested pipe specimen, psi

B=pressure at burst, psi

Figure 4-2 illustrates a test arrangement used to determine the pressure rating of a pipe tee. The specimen will be pressurized with water. The water pressure is then increased until the assembly bursts. If burst occurs in the tee, then the pressure at burst is B and the design pressure or rated pressure is established as a fraction of B. At times, during a burst test, the pipe extensions to which the fitting is attached will burst before the fitting itself. If burst occurs in the pipe extensions, then the tee is stronger than the pipe and is assigned the pressure rating of the matching pipe.

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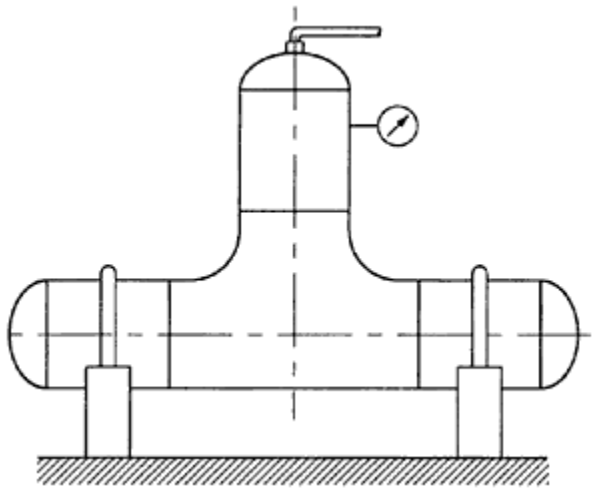

**Figure 4-2** Assembly for Burst Test of an ASME B16.9 Tee

### 4.5.2 Malleable Iron Threaded Fittings

On the basis of proof tests, pressure ratings are developed for a wide range of "standard" fittings. By standard fitting we mean a fitting listed in one of the ASME B16 standards. For example, the pressure-temperature ratings of malleable iron (MI) threaded fittings are shown in Table 4-7.

**Table 4-7** Ratings of Malleable Iron Threaded Fittings [ASME B16.3]

| T (°F) | Class 150 (psi) | Class 300 ¼"–1" (psi) | Class 300 1¼"–2" (psi) | Class 300 2½"–3" (psi) |
|---|---|---|---|---|
| −20 to 150 | 300 | 2000 | 1500 | 1000 |
| 200 | 265 | 1785 | 1350 | 910 |
| 250 | 225 | 1575 | 1200 | 825 |

### 4.5.3 Steel Butt Welded Fittings

The pressure ratings of wrought steel butt-welding fittings (ASME B16.9) will be the "same as the straight seamless pipe of equivalent material". For example, a butt-welding elbow for a 6" schedule 40 piping system will be procured as a schedule 40 ASME B16.9 fitting. The fittings are proof tested at a pressure given by

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$$P_{test} = 105\% \frac{2S_u t}{D}$$

### 4.5.4 Steel Flange Ratings

**Table 4-8** Design Pressure for Class 150 Flange Rating [ASME B16.5]

| T (°F) | ASTM A 105 Carbon Steel (psi) | ASTM A 182 C—½ Mo (psi) | ASTM A 182 F 304 (psi) |
|---|---|---|---|
| −20 to 100 | 285 | 265 | 275 |
| 200 | 260 | 260 | 235 |
| 300 | 230 | 230 | 205 |
| 400 | 200 | 200 | 180 |
| 500 | 170 | 170 | 170 |
| 600 | 140 | 140 | 140 |
| 700 | 110 | 110 | 110 |
| 800 | 80 | 80 | 80 |
| 900 | 50 | 50 | 50 |
| 1000 | 20 | 20 | 20 |

**Table 4–9** Flange Class Pressures (psi) Carbon Steel Group 1.1 [ASME B16.5]

| °F/Class | 150 | 300 | 400 | 600 | 900 | 1500 | 2500 | 4500 |
|---|---|---|---|---|---|---|---|---|
| −20/100 | 285 | 740 | 990 | 1480 | 2220 | 3705 | 6170 | 11110 |
| 200 | 260 | 675 | 900 | 1350 | 2025 | 3375 | 5625 | 10120 |
| 300 | 230 | 655 | 875 | 1315 | 1970 | 3280 | 5470 | 9845 |
| 400 | 200 | 635 | 845 | 1270 | 1900 | 3170 | 5280 | 9505 |
| 500 | 170 | 600 | 800 | 1200 | 1795 | 2995 | 4990 | 8980 |
| 600 | 140 | 550 | 730 | 1095 | 1640 | 2735 | 4560 | 8210 |
| 700 | 110 | 535 | 710 | 1065 | 1600 | 2665 | 4440 | 7990 |
| 800 | 80 | 410 | 550 | 825 | 1235 | 2060 | 3430 | 6170 |
| 900 | 50 | 170 | 230 | 354 | 515 | 860 | 1430 | 2570 |
| 1000 | 20 | 50 | 70 | 105 | 155 | 260 | 430 | 770 |

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Flange class ratings are listed in ASME B16.5 for a range of materials and temperatures. These ratings can be traced back to at least the 1920's. An example of flange ratings for Class 150 is shown in Tables 4-9 and 4.10. Note, as a rule of thumb, that at ambient temperature, the maximum operating pressure of a carbon steel flange of class X is approximately 2.4X. This rule works particularly well for class 300 and larger.

### 4.5.5 Socket Welding and Threaded Fittings

The correlation of fitting class with schedule number for socket-welding and threaded forged steel fittings is given in ASME B16.11, and summarized in Table 4-10. These classes evolved separately and are different from the flange rating classes, which causes confusion in practice. For socket welded fittings, the following dimensions are specified: maximum and minimum inner diameter, mini-mum socket width of 1.09 $t_{pipe}$ and average socket width of 1.25 $t_{pipe}$, end-to-end length, minimum thread and socket length.

**Table 4-10** Class for Threaded and Socket Welded Fittings [ASME B16.11]

| Class | Type | Pipe Schedule | Pipe Wall |
|---|---|---|---|
| 2000 | Threaded | 80 | XS |
| 3000 | Threaded | 160 | - |
| 6000 | Threaded | - | XXS |
| 3000 | Socket Welding | 80 | XS |
| 6000 | Socket Welding | 160 | - |
| 9000 | Socket Welding | - | XXS |

### 4.5.6 Valves

#### *4.5.6.1 Shell Strength*

Steel and alloy, flanged and butt-welding valves (ASME B16.34) are grouped in eight material classes, and are grouped in seven standard classes 150, 300, 400, 600, 900, 1500, or 2500. ASME B16.34 and API 598 valves are given a shell proof test at a gage pressure no less than 1.5 times the 100°F pressure rating PC for 15 minutes. The ASME B16.34 ratings apply to the valve body. Other metallic and non-metallic parts of the valve, such as seals, O-rings, packing, will further limit the upper range of temperatures at which the valve can be used. Conversely, there is a minimum temperature below which a valve should not be used because its body or parts become excessively brittle or may bind. There have been cases where a valve that withstood the shop hydrotest at 1.5 $P_D$ leaked through the packing during the field hydrotest at 1.5 $P_D$ ($S_{70}/S_{hot}$) (Chapter 19).

The minimum wall thickness of an ASME B 16.34 valve is

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$$t = 1.5\frac{P_C d}{2S - 1.2P_C}$$

t=minimum wall thickness, in
$P_C$=pressure rating for the class designation, psi
d=inside diameter, in
S=material allowable stress, psi

*4.5.6.2 Seat Tightness*

Valves designed for shutoff and isolation service, such as stop valves and check valves, are given a seat tightness test, also referred to as closure tightness test. Seat tightness classes are specified in MSS-SP-61, MSS-SP-82, FCI 70–2 and API 598, ASME B16.34, and are reviewed in Chapter 25.

### 4.5.7 Unlisted Components

Here is a rather common situation: a specialty pipefitting or component is sold by a vendor. The fitting or component does not conform to the shape or dimensions of an ASME B16 standard; it is an "unlisted component". The fitting or component is shown in the vendor catalog with a "design" or "maximum permitted" pressure and temperature. How should this design pressure have been established? In establishing the design pressure for an unlisted piping component, three conditions must be satisfied [ASME B31.3]:

(1) The component must conform to a published specification, and

(2) The material must be comparable to listed materials, and

(3) The pressure integrity must be established in two ways:

(3.1) Sizing calculations in accordance with ASME B31, and

(3.2) One of the following four methods

(3.2.1) Extensive successful experience, or

(3.2.2) Experimental stress analysis, or

(3.2.3) Proof test, or

(3.2.4) Detailed stress analysis.

Note that, in ASME B31.3, "extensive successful experience" alone is not sufficient to qualify an unlisted (specialty) component.

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## 4.6 PRESSURE STRESS IN FITTINGS

### 4.6.1 Pipe Elbows and Bends

If an elbow or bend is pressurized, the hoop stress will vary around the circumference, as shown in Figure 4-3. The hoop stress is

$$\sigma_h = \frac{PD}{4t}\left(\frac{2R + r\sin\theta}{R + r\sin\theta}\right)$$

$\sigma_h$=hoop stress in the bend or elbow, psi
P=internal pressure, psi
D=pipe outer diameter, in
t=pipe wall thickness, in
R=radius of curvature of the bend or elbow, in
r=inner radius of the pipe cross section, in
θ=angle around the pipe cross section, counted as 0 at 3 o'clock, grad

Interestingly, the largest hoop stress occurs at the "intrados" of the bend (the inner radius of the bend, 6 o'clock in Figure 4-3), which is also where the bending process naturally results in the thickest wall. Conversely, the smallest hoop stress occurs at the "extrados" (external radius of the bend, 12 o'clock in Figure 4-3), which is where the bending process naturally results in the thinnest wall. On the sides of the elbow or bend (θ=0 or π, 3 and 9 o'clock in Figure 4-3) the hoop stress is the same as in a straight pipe PD/(2t).

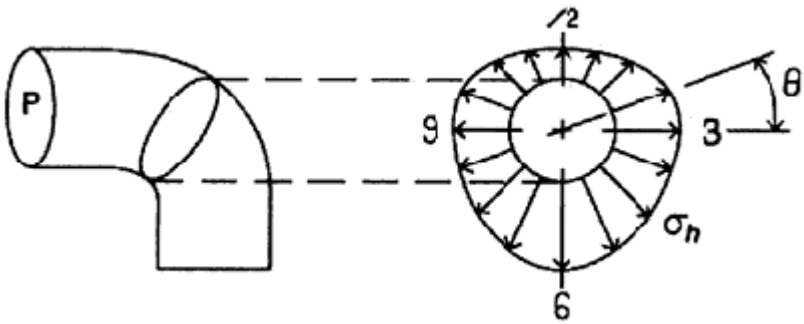


**Figure 4-3** Distribution of Hoop Stress in a Pressurized Elbow

### 4.6.2 Branch Connections and Nozzles

The limit pressure $P_L$ for a 90° branch or vessel nozzle is given by [Rodabaugh]

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$$P_L = P^* P_Y$$

$$P_Y = 2S_Y T / D$$

$$P^* = \frac{(162\left(\frac{t}{T}\right)^2 + 288\frac{t}{T}\frac{d}{D} + 210)\zeta + 155}{128\zeta^2 + (228\left(\frac{d}{D}\right)^2 + 228)\zeta + 152}$$

$$\zeta = \frac{d}{D}\sqrt{\frac{D}{T}}$$

D=header or vessel outer diameter, in
d=branch or nozzle outer diameter, in
T=header or vessel wall thickness, in
t=branch or nozzle wall thickness, in
$S_Y$=material yield stress, psi

For example, a carbon steel vessel with a yield stress $S_y$=48 ksi and an ultimate strength $S_u$=66.7 ksi, with a 90° nozzle, D=23.9", d=12.8", and T=t= 0.24" has a calculated limit pressure $P_L$ of 571 psi [Sang].

The burst pressure for a 90° nozzle intersection, is [Rodabaugh]

$P_b$=1.067 $(P^*)^{0.669}$ (2 $S_U$ T/D)

For the same carbon steel vessel nozzle we calculate the burst pressure $P_b$= 1010 psi. In this case, the measured burst pressure was 1073 psi and occurred by split in the longitudinal plane, as indicated by rupture line R in Figure 4-4 [Sang].

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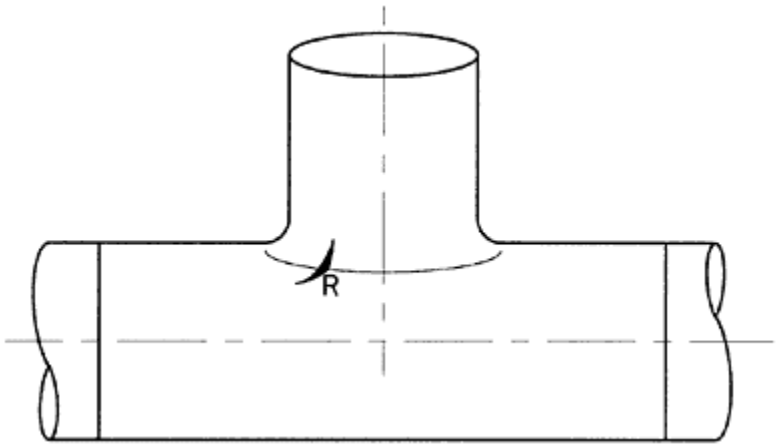


**Figure 4-4** Burst of 90° Intersection at R **[Sang]**

So far, the header-branch intersection was at right angle. The burst pressure of branch intersections at an angle other than 90° has also been studied by the Pressure Vessel Research Council, New York. In a recent series of experiments, two carbon steel vessels, one with a 90° nozzle, which we will call “vessel 1” and the other, “vessel 2”, with a 30° nozzle, were burst tested and analyzed to study their failure mode [Sang]. The rupture of vessel 1 was perpendicular to the nozzle weld, close to the longitudinal plane, Figure 4-4. The rupture of vessel 2 was along the nozzle weld, as indicated by the rupture line “R” in Figure 4-5 [Sang].

The pressure stresses of large branch connections, when the branch diameter d is 1/3 or more of the header diameter D, has also been the subject of several tests and analyses. A correlation was developed for calculating pressure stresses in large d/D branch connections and nozzles for $0.333 \leq d/D \leq 1.0$, and $20 \leq D/T \leq 250$, and $0.333 \leq t/T \leq 3.0$ [Widera].

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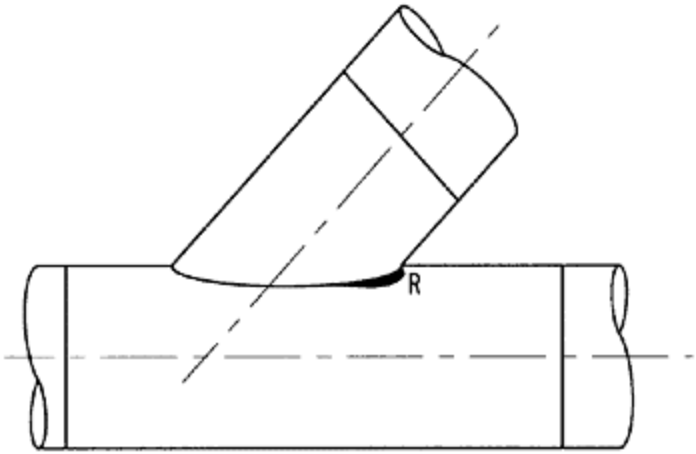


**Figure 4-5** Burst of 30° Intersection at R [Sang]

Then, the stresses at the intersection can be expressed in the form of a parametric equation with coefficients defined in Table 4-11

$$\sigma_i/\sigma_o = X - Y\,(d/D)^a\,(D/T)^b\,(t/T)^c + Z\,(d/D)^d\,(D/T)^e\,(t/T)^f$$

**Table 4-11** Coefficients for Use with $\sigma_i/\sigma_0$ [Widera]

| | $\sigma_i=\sigma_{tv}$ | $\sigma_i=\sigma_{tn}$ | $\sigma_i=S_v$ | $\sigma_i=S_n$ | $\sigma_i=\sigma_{tmv}$ | $\sigma_i=\sigma_{tmn}$ |
|---|---|---|---|---|---|---|
| X | 1.1152 | 1.3530 | −2.1052 | 2.6178 | 1.2498 | 1.2553 |
| Y | −1.5986 | 7.9974 | 3.6824 | −.006508 | −.006554 | −0.03199 |
| Z | 2.2996 | −7.0439 | −0.05724 | 0.4549 | 0.6498 | 0.6738 |
| a | 0.8 | 0.9 | 0.6 | 0.4 | 1.0 | 1.0 |
| b | 0.3 | 0.5 | 0.3 | 0.5 | 0.6 | 0.6 |
| c | 0.3 | 0 | −0.2 | −3.2 | −2.4 | −2.0 |
| d | 0.8 | 0.9 | 1.2 | 0.6 | 1.0 | 1.0 |
| e | 0.4 | 0.5 | 0.3 | 0.6 | 0.5 | 0.5 |
| f | −0.2 | 0.1 | 2.9 | −1.3 | −0.8 | −1.0 |

$\sigma_0$=nominal hoop stress in the header, PD/(2T), psi
$\sigma_{tv}$=membrane plus bending stress in the header (vessel), tangent to opening, psi
$\sigma_{tn}$=membrane plus bending stress in the branch (nozzle), tangent to opening, psi
$\sigma_{tmv}$=local membrane stress in the header (vessel), tangent to opening, psi
$\sigma_{tmn}$=local membrane stress in the branch (nozzle), tangent to opening, psi
$S_v$=surface stress intensity in the header (vessel), psi
$S_n$=surface stress intensity in the branch (nozzle), psi
D=header diameter, in

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d=branch diameter, in
T=header thickness, in
t=branch thickness, in

### 4.6.3 Reinforcement of Branch Connections

Special rules apply to the design of a connection between a header pipe (or "run" pipe) and a branch pipe. Because the stress field at a run-branch intersection is rather complex, the design rules could not be based on a detailed stress analysis of each configuration. Instead, the design of pipe branch connections subject to internal pressure is based on a simple rule called "the area replacement rule". According to the area replacement rule, the disc of metal cut out of the run pipe to create the branch connection must be replaced by metal added around and close to the connection…unless the run pipe or the branch have sufficient wall in excess of the minimum required by code plus future corrosion allowance to compensate for the cut-out disc. If we refer to Figure 4-6, the area replacement rule can be written as

$$\sum_{i=1}^{4} A_i = A_7$$

$A_1$=excess metal area in the run pipe, within a band of radius $d_2$, $in^2$
$A_2$=excess metal area on the branch pipe, within a band of height $L_4$, $in^2$
$A_3$=area branch reinforcement welds, $in^2$
$A_4$=area of branch reinforcement, $in^2$
$A_7$=area of required metal lost by branch cutout, $in^2$

We now need to establish the radius $d_2$ of the reinforcement zone along the header and the height $L_4$ of the reinforcement zone along the branch (dashed line zone in Figure 4-6). The radius of reinforcement $d_2$ measured along the header, from the opening, has to be sufficiently short to effectively reinforce the opening. The height of reinforcement $L_4$ along the branch line is based on the attenuation distance for a uniform moment $M_0$ applied around the circumference of a cylinder, Figure 4-7. The moment attenuation with distance x is

$M(x)=M_o\,(\cos\lambda x+\sin\lambda x)\exp(-\lambda x)$
$\lambda=1.29/(Rt)^{0.5}$

M(x)=moment at a distance x along cylinder, in-lb
$M_o$=applied uniform moment, in-lb
x=distance along cylinder axis, in
R=cylinder diameter, in
t=cylinder thickness, in

The distance $1/\lambda\sim(Rt)^{0.5}$ plays an important role in branch reinforcement as well as spacing of openings. In practice, it can be viewed as the distance beyond which the stress and distortion effects due to an applied point or line load are reduced to about 1/3 their magnitude.

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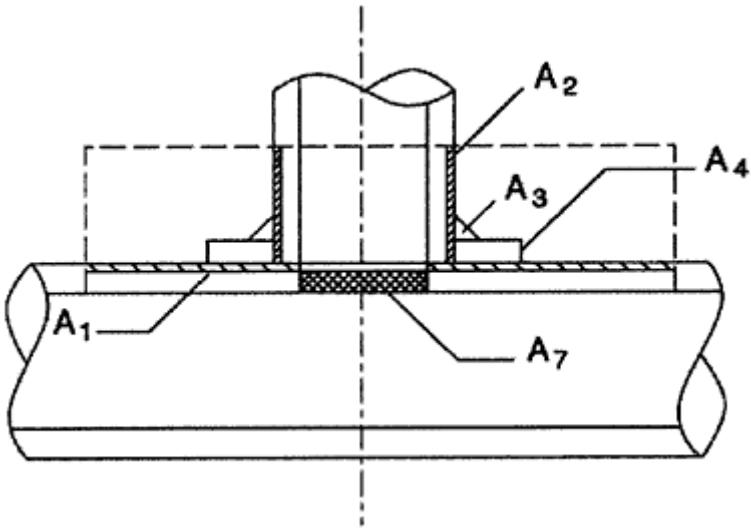


**Figure 4-6** Branch Reinforcement Areas

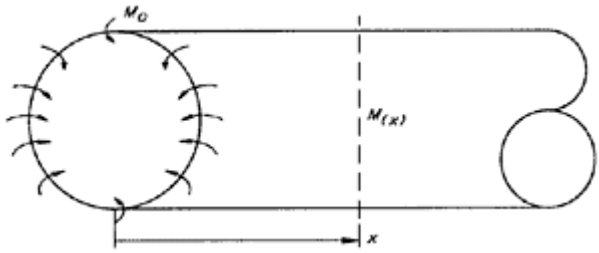


**Figure 4-7** Moment Attenuation Distance

The conditions of applicability of the formulas for radius and height of reinforcement are [ASME B31.3]: no stub-ins under external loads; $D_{run}/t_{run}<100$; $D_{branch}/D_{run}<1$; angle of branch-run≥45°; run and branch axes intersect. Under these conditions

$L_4$=min {$L_4$'; $L_4$"}
$L_4$'=2.5 ($T_h$−c)
$L_4$"=2.5 ($T_b$−c)+$T_r$

$L_4$=height of reinforcement box, in
$T_h$=header wall thickness, in
c=corrosion allowance, in
$T_b$=branch wall thickness, in
$T_r$=minimum thickness of reinforcing ring or saddle, in

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$d_2=\max\{d_1; d_2'\}$

$d_2'=(T_b-c)+(T_h-c)+d_1/2$

$d_2$=radius of reinforcement zone, in

$d_1$=inner diameter of branch pipe, in

The rules for branch reinforcement in gas pipelines are different than for power and process piping, they depend on the run and branch pipe diameters and the maximum allowable operating pressure. At high pressure a full encirclement sleeve becomes necessary.

One advantage in using integrally reinforced branch fittings, Figure 4-8, is that they contain in themselves the area reinforcement that compensates for the disc cut-out from the header [MSS-SP-97].

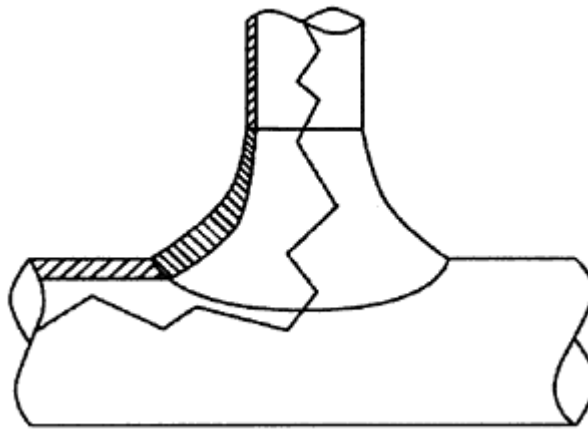

**Figure 4-8** Integrally Reinforced Outlet Fitting

### 4.6.4 End Fillets

It is critical to properly size the fillet welds between the reinforcing pads and the pipe. Too large a fillet weld on a relatively thin header has caused rupture of the header pipe, Figure 4-9 left side. Too small a fillet (usually because of an excessive gap between the pipe and the pad or sleeve) has caused the failure in the weld, Figure 4-9 right side. The rules of the applicable ASME B31 code section should be followed, with the leg of the fillet weld (the weld length along the header) between 1.0 to 1.4 times the header thickness [Rosenfeld].

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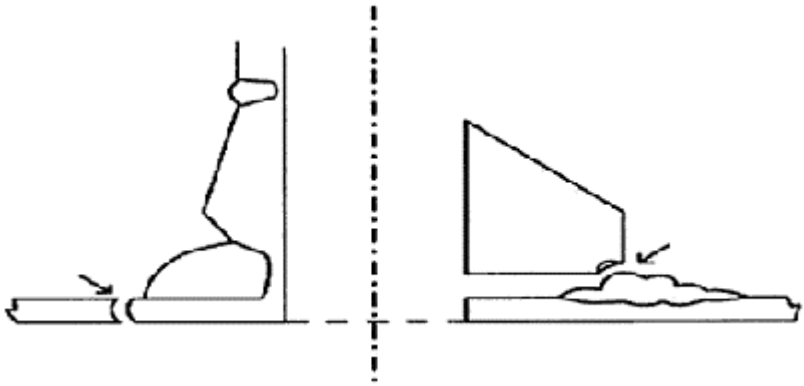

**Figure 4-9** Improperly Sized End Fillet Welds (Excessive Deposit at Left, Excessive Gap at Right)

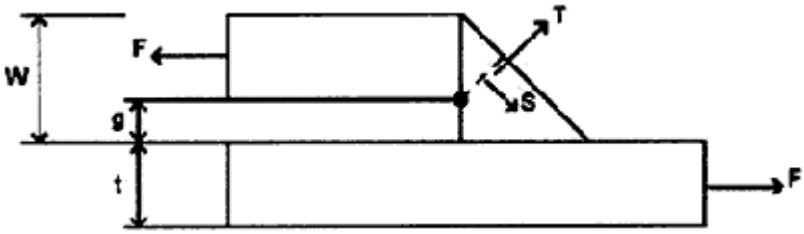


**Figure 4-10** Shear Stress (t) and Normal Stress (s) in Fillet Weld

The shear and principal stress in a sleeve-header end fillet weld depends on their initial gap, Figure 4-10, hence the importance of close fitting the sleeve or pad to the pipe before welding [Rosenfeld]

$\tau_{max}=1.12\ F/[\pi D(w-g)]$

$S_{max}=1.62F/[\pi\ D(w-g)]$

$\tau_{max}$=maximum shear stress in the weld, psi
$S_{max}$=maximum principal stress in the weld, psi
D=pipe diameter, in
w=distance from header OD to sleeve OD, in
g=gap between header OD and sleeve ID, in
F=shear force on weld, lb

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## 4.7 HIGH PRESSURE DESIGN

If Lamé's hoop stress equation used for piping design is applied for high-pressure service, in the order of 6000 psi or more, the pipe wall thickness required would be in many cases prohibitively large. At such high pressures, an alternative design method is used, based on full yield of the pipe wall. The pressure at which the pipe wall of a thick cylinder has completely yielded is [Sims, Nadai]

$$P_{yD} = \frac{2}{\sqrt{3}} S_y \ln \frac{D}{D_i}$$

$P_{yD}$=pressure at which the pipe wall has yielded through the wall, psi
$S_y$=material yield strength, psi
D=outer diameter, in
$D_i$=inner diameter, in

Substituting the allowable stress S for $2S_y/3$, and limiting the design pressure to half the full-section yield pressure $P_{yD}$ we obtain the wall thickness design formula for high pressure piping [ASME B31.3]

$$t = \frac{D}{2}[1 - \exp(-1.155\frac{P}{S})]$$

t=design wall thickness for very high pressure service, in
D=outer diameter, in
P=internal design pressure, psi
S=allowable stress, psi

## 4.8 DESIGN PRESSURE

### 4.8.1 Design Scenarios

As we have seen so far, the minimum design wall thickness is proportional to the design pressure. The design pressure is the pressure that, taken with the concurrent temperature, results in the thickest pipe wall. It is the highest pressure at which the piping system should operate. In practice, it is the system's pressure relief valve set pressure or rupture disc burst pressure. If there is no relief device (valve or rupture disc) the design pressure is the highest credible pressure that can be achieved in the piping system. In choosing a design pressure, one must consider normal and credible abnormal (upset) operating conditions and environments. A system engineer familiar with the system function must develop the normal and abnormal operating scenarios, with consideration of the following conditions [API RP 521]:

(a) Single or multiple operational errors such as opening the wrong isolation valves while in service or inadvertently closing a vessel outlet valve.

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(b) Single or multiple maintenance errors, such as failing to reopen an isolation valve or remove a blind flange after a maintenance outage.
(c) Failure of control devices, such as the failure of a pressure or level switch or interlock.
(d) Electrical failure, loss of instrument air or accidental signals that will erroneously start a pump or close a valve.
(e) Failure of a heat exchanger tube or tank coil that could over-pressurize the lower pressure side of the system.
(f) Waterhammer that could result in an impulse of pressure or several pressure pulses in the system.
(g) Plant fire that could overheat the fluid trapped in the system.

One of the difficulties in this process is to decide which scenarios are credible (and contribute in setting the design pressure) and which ones are incredible (and can be ignored in setting the design pressure) [Stickles, API 752]. As a matter of good practice all potential sources of overpressure with a probability of occurrence of $10^{-2}$ per year or more likely should be considered in setting the design basis and ASME design pressure of a vessel-piping system. In addition, more rare events, those with a probability of occurrence between $10^{-2}$ and $10^{-4}$ per year, should not be ignored if the overpressure scenario could have safety, environmental or financial consequences; in these cases, some deformation, such as bulging or permanent plastic yielding, may be permitted if it has no adverse consequence. If deformation is acceptable, detailed plastic analysis or overpressure testing may be required to qualify the piping system, pipeline or vessel. Events with a probability of occurrence between $10^{-4}$ and $10^{-6}$ per year should not be ignored if they can create significant danger.

In addition to a design pressure, the pipeline codes also define a maximum operating pressure MOP as the highest pressure at which a pipeline is normally operated. The maximum allowable operating pressure MAOP is the maximum pressure at which a pipeline may be operated in accordance with the Code.

The Pitkin, Louisiana, accident illustrates the need to correctly determine overpressure accident scenarios, including the need to consider operator error. On March 4, 1998, preparations were made to purge air from a two-mile pipeline leading to an oil and gas separation facility. The plan was to align valves to permit well fluid to flow into the pipeline, bypass the facility's high-pressure separators, and purge air from the line through an open hatch in the roof of a water tank. The intended flow path from the well (Figure 4-11) was A→B→C→D→E and eventually to atmosphere venting through F. The post-accident investigation indicated that valve alignment was carried out without a written procedure, without Process and Instrumentation Diagrams (P&ID), and without following a checklist [CSHI 02]. As the venting operation started, gas and liquid from the well flowed into the pipeline. One hour into the purging operation, the third stage separator (labeled SE in Figure 4-11) exploded, releasing gases that ignited, killing four.

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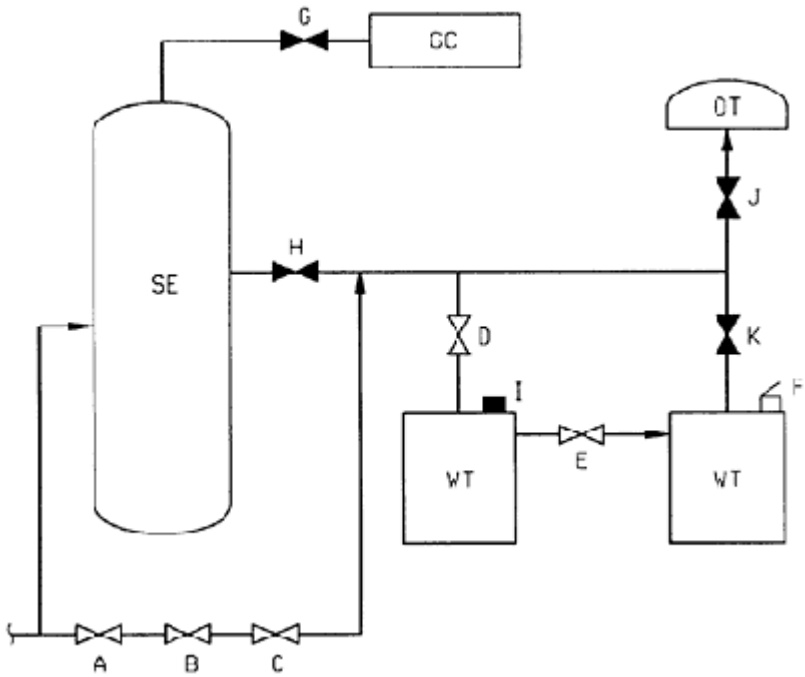


**Figure 4-11** Simplified Diagram of Oil-Gas Separator

The post-accident investigation indicated that valves A and C, believed to be open, were actually closed. Therefore well fluid, gas and oil, was filling and pressurizing the third stage separator. This third stage separator was designed to normally operate at atmospheric pressure, and did not have a pressure relief valve. API 12J [API 12J] recommends that "all separators, regardless of size or pressure, shall be provided with pressure protective devices…". The pressure was estimated to have reached between 135 psi and 400 psi when the separator burst.

### 4.8.2 Pressure Excursions

Because there is margin between the ASME code design pressure $P_D$ and the yield and burst pressures, it is permissible—under certain conditions—to experience short pressure excursions above the design pressure, without damage. The pressure excursion allowance of ASME piping codes is summarized in Table 4-12.

**Table 4-12** Overpressure Allowance in Piping Design

| Piping Code | Overpressure Allowance Beyond design Pressure $P_D$ |
|---|---|
| ASME B31.1 | $P_D$+15% if less than 10% of any 24 hrs. |

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| | |
|---|---|
| | $P_D$+20% if less than 1% of any 24 hrs. |
| ASME B31.3 | $P_D$+33% if less than 10 hrs. once and less than 100 hrs/yr.<br>$P_D$+20% if less than 50 hrs once and less than 500 hrs/yr. |
| ASME B31.4 | $P_D$+10% |
| ASME III | $P_D$+0% during normal operation<br>$P_D$+10% during upset conditions.<br>$P_D$+50% during emergency conditions.<br>$P_D$+100% during faulted conditions, with required shutdown. |

These overpressure excursions are only permitted under certain conditions. For process piping, as an example, the conditions under which a pressure excursion is permitted are: (a) no cast iron fittings in the system, (b) the stress should not exceed yield, (c) the longitudinal stress should not exceed 1.33S, where S is the code stress allowable, and (d) the pressure should not exceed the test pressure.

In addition, one must be careful when allowing overpressure to occur in pipe components such as valves or specialty fittings. In these cases, it is important to consult the manufacturer's specifications to verify that pressure excursions such as those listed in Table 4-12 are permitted.

In the case of pressure vessels, if the vessel design pressure $P_D$ is also the relief valve set point, then:

$P_D$−10%=practical maximum operating pressure.

$P_D$−2%=the relief valve may start to open, the valve simmers.

$P_D$+10%=as the relief valve discharges, the vessel pressure can accumulate another 10% above the valve opening set point. An ASME Section I boiler would only be permitted a 6% overpressure accumulation.

$P_D$+21%=in case of fire, as the relief valve discharges, the vessel pressure can accumulate 21% (or 10% over the previously permitted 10%, in other words, 110%×0%=21%).

$P_D$−7.5%=as the pressure decreases below the set point $P_D$, the valve will start to close but may not completely close upon reaching $P_D$. The valve may reseat as low as 7.5% below $P_D$. This is the blowdown region. For a boiler, blowdown would occur between 2% and 4% of $P_D$.

## 4.9 OVER-PRESSURE PROTECTION

Over-pressure protection of a component or system is most often achieved by the use of a pressure relief device (a pressure relief valve or rupture disc). A pressure relief valve may be (a) a relief valve which lifts gradually and recloses as the overpressure dissipates, which is typical of liquid service, (b) a safety valve that pops open suddenly, remains fully open, and recloses when the overpressure subsides as is typical in gas service, and (c) a safety-relief valve which operates either as a safety or relief valve. The approach to overpressure protection of piping systems is different than for pressure vessels. For piping, the system can be designed to "contain or relieve" the overpressure. In other words, if the pipe, fittings and components are sufficiently thick to contain all credible

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pressure transients, the system does not need a safety or relief device. Control devices, such as pressure or temperature interlocks would assure that the design pressure of the piping system would not be exceeded. Because one has to postulate failure of control devices, these would have to be redundant in certain applications. For example, if the failure of a pressure regulator could over-pressurize a piping system, an alternative to a pressure relief valve would be a second regulator, in series, or an automatic shut-off device, in series. In the case of a centrifugal pump, we would have to assume that a downstream block valve fails closed or is closed by mistake while the pump is running. We would then size the system between the pump and the isolation valve for the full dead-head pressure of the pump.

In the case of a pressure vessel, a relief device is required, even if the vessel is sufficiently thick. This requirement causes serious difficulties for pressure vessels containing fluids that corrode or stick to the relief valve or rupture disk, causing it to malfunction. To address this concern, ASME published Section VIII Code Case 2211, which addresses the question: under what conditions may a pressure vessel be provided with overpressure protection by system design in lieu of a pressure relief device as required by UG-125(a)? In the original issue of Code Case 2211, there were five conditions under which this “overpressure protection by system design” was permitted: (1) The fluid may not be water, air or steam, (2) applying the code case is the owner’s decision, not the contractor, (3) system design must consider all credible overpressure scenarios, (4) the analysis must be made available to the jurisdiction, and (5) the vessel Data Report must refer to the Code Case. Note that in States with pressure vessel laws, the use of this Code Case must typically be authorized by the jurisdiction.

It takes a solid knowledge of a system and its operation to foresee the possible upsets that can occur and the resulting overpressure conditions. Sometimes, a system that has operated well for many years will be placed in an unprecedented overpressure condition. The explosion in a dye plant, in Patterson, New Jersey illustrates this point [CSHI]. On April 8, 1998, a mixing process was underway at the dye plant. The mixing process takes place in a vessel, which is cooled by water or heated by steam circulating into an outer jacket. Figure 4-12 shows the entry of steam (ST) or cooling water (CW), and the exit of condensate (CO) or cooling water (CW). The vessel is protected by two rupture disks mounted in series (RD), and set to relieve at 10 psig [CSHI].

The mixture is introduced into the vessel and heated above 100°F by circulating steam through the jacket. As the reaction progresses, the temperature increases but is kept from reaching 380°F by switching the jacket flow from steam to cooling water. Temperature control is important since beyond 380°F an explosive exothermic reaction would take place.

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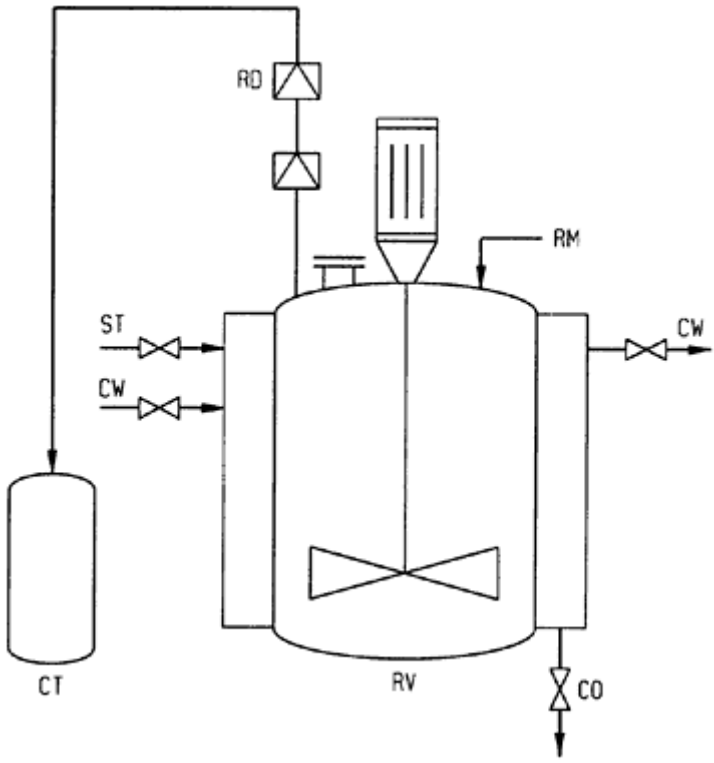


**Figure 4-12** Simplified Diagram of Dye Manufacture Vessel

On the evening of April 8, the operators had introduced the material into the vessel, the temperature of the mixture was close to 115°F, and steam was then introduced to heat the mixture and start the mixing process. The temperature in the vessel started to rise rapidly, reaching 360°F, at which point the rupture disks were blown, the pressure had therefore reached 10 psig. But the reaction did not stop then, and the pressure kept increasing. At 8:18 PM, the vessel's inspection manway was blown-off and the vessel knocked off its supports. Nine were injured, the building walls were destroyed and windows were shattered. An aerosol mixture of hot reactants was dispersed over the entire neighborhood surrounding the facility.

## 4.10 BURST ENERGY

The explosive potential of an overpressure failure in a plant vessel or piping systems must be understood to best realize the importance of pressure design and overpressure protection. If a vessel ruptures by brittle fracture, it will suddenly release all its energy. The energy stored in an ideal gas is [Borzileri]

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$$E_{gas} = \frac{PV}{k-1}\left[1-\left(\frac{P_{amb}}{P}\right)^{\frac{k-1}{k}}\right]$$

V=volume of gas, $in^3$

k=gas constant

$P_{amb}$=ambient pressure, psia

P=pressure of stored gas, psia

The energy stored in a liquid is

$$E_L \sim \frac{1}{2}\frac{P^2V}{\beta}$$

β=bulk modulus of liquid, psi (330,000 psi for water).

For example, to calculate the energy stored in a gas filled bottle made of a 5 ft long section of 4" sch. 40 pipe, pressurized at 150 psi, we set P=164.7 psia, k= 1.4, V=(12.73)(60)=763.8 $in^3$, $P_{amb}$=14.7 psia. Therefore $E_G$=157,247 in-lb= 13,104 ft-lb. The TNT equivalent of the gas filled pipe is $E_G/(1.55\times10^6$ ft-lb)= 0.0085 lb TNT. For the same pipe full of water, $E_L$=31 in-lb, which is negligible.

To calculate the structural or biological effects of the sudden, explosive release of the gas content, the actual distance R and energy stored expressed in equivalent pounds of TNT $E_{TNT}$, are converted to a scaled distance $R_{scaled}$ given by

$$R_{scaled} = \frac{R}{\sqrt[3]{E_{TNT}}}$$

R=distance of target from source, ft

$E_{TNT}$=stored energy in pounds of TNT equivalent, lb TNT.

The $R_{scaled}$ factor is then correlated to damage, as follows shown in Table 4-13.

**Table 4-13** Biological and Structural Effects of Blasts

| Rscaled ($ft/lb^{1/3}$) | Biological Effect | Structural Failure |
|---|---|---|
| 50 | - | Glass windows |
| 30 | Eardrum rupture | Concrete block panels |
| 15 | Lung damage | Brick walls |
| 5 | Mortality | - |

For example, the $R_{scaled}$ factor for a person standing 5 ft away from the 5 ft tall gas filled cylinder is $R_{scaled}=5/(0.0085)^{1/3}$=24. This person could suffer an eardrum rupture.

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Note that in the nuclear power industry, where the effects of pipe breaks in water and steam lines have been studied in detail, “high energy lines” in which “guillotine breaks” (full severance of the pipe cross section) are postulated are not based on energy content, but rather on operating pressure and temperature; a high energy system is one operating at a temperature greater than 200°F or a pressure greater than 275 psi [USNRC].

Finally, we must note that a ductile material, operating above the fragile transition temperature for its thickness (Chapter 3), will tend to fail by leak rather than fracture, and will not release all of its energy suddenly but progressively, without the explosive effect [API 579].

## 4.11 PIPE SPECIFICATION

A pipe specification is a menu of pipe components of the right material and the right rating for a given service. The piping specification is an essential document in each project and at each plant because not only is it used for the initial design and procurement of materials, it will also be used throughout the life of the plant to chose and procure replacement parts, and to find out what is installed, long after the system has been put in service.

Of the five fundamental aspects of piping engineering (materials, design, fabrication, examination, and testing, Figure 2-1), a piping specification only addresses materials and one aspect of design (pressure design). It is essential to understand this limitation: a plant piping specification is not a Code, a series of additional requirements apply beyond material selection and pressure sizing.

Some companies have piping specifications that also address fabrication, examination and testing. Here, we will consider fabrication (welding, flange as-semblies, tubing fittings, threaded joints, etc.), examination (type, extent and acceptance criteria of non-destructive examinations, NDE), and testing (leak testing, hydro-testing, performance testing, etc.) to be covered in separate company or plant procedures, as is often the case.

Because a piping specification is an important design document, it should be backed-up by a detailed report and calculations that explain how the materials, fittings and ratings were selected: why was a certain material specification and grade selected? Why were cast fittings chosen in place of forged fittings, and does it matter? Why was a certain schedule chosen? etc.

The piping specification must be logically tied with the plant’s piping line numbers and the Piping & Instrumentation Diagrams (P&ID’s). For example, a plant line may be labeled “B-CW-6-PABC-NS” to mean that this line is part of process unit B (or zone B, or building B) of the plant, it is part of the Cooling Water system (CW), it has a 6” nominal pipe size, its pipe specification is PABC, and it is not safety related (NS). It is up to the Owner to decide what the line numbering scheme should be; but, in any case, it should include the pipe specification number. With these general points in mind, we can now refer to the example pipe specification shown in Table 4-14.

Line 1 is simply the pipe specification number, to be selected by the Owner or the designer. The letter “P” designates this specification as being a piping specification. Some companies use the name “pipe code” rather than pipe specification, which is fine, provided users understand that it is actually not a Code. The Code is—in the case of our

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example—ASME B31.1 (called out on Line 4), and this specification addresses only two aspects of the Code: material and pressure rating.

Line 2 is the pressure "class", based on the class of flange used in the system, and which can be found in ASME B16.5 tables, knowing the flange material and the system design pressure and temperature. The design pressure is 150 psi (all pressures are to be read as "psig", but it is common practice in industry to label gage pressures as "psi"). This is the highest pressure that may be experienced in service. It is typically the set pressure of the pressure relief device (plus any hydrostatic head to account for changes in elevation between the relief device and the lowest point in the system). The actual operating pressure is typically kept at least 10% below this design pressure so as not to challenge the pressure relief device in normal service. The maximum design pressure is 120°F, a value obtained from the heat transfer, thermal hydraulic design of the system. The minimum design temperature (a point often overlooked) is the lowest temperature the system is permitted to experience in service. It is important to keep in mind that each metal has a nil ductility transition temperature (NDT), which is also a function of thickness, and below which its toughness is very low, and the material can more easily fracture [API 579]. Note that the piping specification does not address the actual operating pressures and temperatures, but only the maximum (or design) values as well as the minimum temperature of the system. The operating pressure-temperature for the various operating modes of the system should be listed in a separate document called a line list.

Line 4 is the corrosion allowance, which has to be selected by the material engineer, based on experience with the same material in similar service. It is the projected value of wall thinning over the service life. For critical service, it will be advisable to periodically take ultrasonic wall thickness readings to confirm that the pipe is not corroding faster than projected during the design phase and reflected in the corrosion allowance. The Code in this case is ASME B31.1. The choice of code must be consistent with the scope defined in the Code itself, industry practice, and regulatory requirements, where applicable.

Line 7 is the pipe material and commercial size. The choice of pipe material is a crucial decision that is based on corrosion resistance (compatibility with fluid and environment, controlled primarily by the material's chemistry, method of fabrication and heat treatment), temperature range in service (compatibility with the strength limit and corrosion behavior at the upper temperature and with fracture toughness at the lower temperature), mechanical properties, and cost. The selected material must have allowable stresses listed for the range of minimum to maximum temperatures. In this case, the material is carbon steel ASTM A 106, Grade B. If an ASME B&PV Section II material is needed, then the specification would be "SA 106, Grade B" where the "S" in front of A 106 indicates that it is a material listed in ASME B&PV Section II. It is also at this stage, that any supplementary requirements would be specified, for example a limit on carbon equivalent for carbon steel, an intergranular corrosion test for stainless steel, etc. The Schedule in line 7 is calculated following the applicable Code equation for minimum wall, including the corrosion allowance and the mill tolerance on wall thickness (12.5% for ASTM A 106). Given the minimum wall, the closest, higher, commercial pipe size is selected. Even though schedule 40 is significantly larger than needed for 150 psi service at 120°F, it remains the minimum schedule of choice for pipe up to 6"in many process

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and power applications. The additional wall thickness will prove useful to counter unexpected corrosion or accidental overpressure in service.

Lines 9 to 14 are standard socket welded (SW) or threaded (THRD) fittings whose dimensions and pressure rating are listed in ASME B 16.11. The material is a forging ASTM A 105, which is compatible with the ASTM A 106 pipe. Lines 15 to 17 are specialty fittings, often times trademarks or registered names developed and fabricated by certain companies. In the case of specialty fittings, size and pressure ratings are obtained not from an ASME B16 standard, but from a vendor catalog or possibly from an MSS-SP standard. In our case, the pipe specification calls out integrally reinforced specialty branch connection fittings (the "o-lets"), and MSS-SP-97 provides dimensional and pressure rating requirements for some of these fittings. There are other suppliers of similar integrally reinforced fittings. The designer is encouraged to explore specialty fittings as they often reflect years of research, innovation and application, and have many practical advantages in hydraulic design (low friction, smoothness of flow), mechanical design (reinforcement is integral to the fitting and branch reinforcement is not required) and ease of assembly and examination. At the same time, the designer should verify that specialty fittings are properly designed and pressure rated by proof testing. They should also be assigned a stress intensification factor "i" (typically established by fatigue testing) to permit design stress analysis. Note that the class for socket welded fittings (in our case class 3000) and that of threaded fittings (in our case class 2000) is not the same as the class for flanges (in our case class 150), which is unfortunate.

Lines 18 to 21 are standard butt-welded fittings whose dimensions and schedule are listed in ASME B16.9. Butt-welded fittings are designated by schedule (the same schedule as the mating pipe) rather than class. The material specification is ASTM A 234, and the grade is WPB, which has chemistry and mechanical properties compatible with Grade B for ASTM A 106. Lines 23 and 24 are specialty fittings.

Lines 25 to 29 are small bore (up to 2" nominal pipe size NPS) and large bore (larger than 2" NPS) flanges permitted by this specification. Note that small bore construction may be threaded or socket welded, and that lap joint flanges (also called Van Stone, Chapter 17) are not permitted in this specification, only slip-ons or weld necks are listed. Lines 30 to 32 are standard carbon steel studs, bolts and nuts. The specification may add that the shape of bolt heads should be heavy hex. The designation RF requires flanges to be raised face. Note that the surface finish of the flange face will depend on the type of gasket selected in line 34 (Chapter 17). Lines 33 and 34 specify the gasket type and thickness. This choice is based on company experience and inquiries with gasket manufacturers.

Other information often encountered in a pipe specification (but not included in the example of Table 4-14) is a matrix of permitted branch connections. The matrix will list a first row of header sizes (NPS) versus a first column of branch sizes NPS, and at the intersection of a given header and branch size it would specify the type of connection permitted: pipe-on-pipe (where the branch is directly welded to the header, with a full penetration weld), or pipe-on-pipe with reinforcing saddle or sleeve (in which case a separate table would provide the size of the saddle or sleeve), or integrally reinforced fitting (the "o-lets" listed in our specification example).

Finally, the foreword to the pipe specification, or a note in each specification, should state that design, fabrication, NDE, and pressure or leak testing shall comply with the

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Code (ASME B31 or NFPA or AWWA, etc.) and refer to the other applicable company procedures. There should also be a means of processing deviations and tracking these on plant drawings. For example, if a lap joint flange is used for line B-CW-6-PABC-NS, this deviation from the pipe specification should be documented, competently reviewed and, if approved and implemented, noted on the P&ID or pipe isometric next to this particular flange joint.

## 4.12 VALVE SPECIFICATION

We can not leave the topic of pipe specification, without addressing valve specifications. Note that valves are not mentioned in this pipe specification, for a good reason: the pipe specification will normally remain unchanged for the life of the system (20 years, 40 years, or more), while valve specifications will change many times over.

The constant improvements in valve materials, design, fabrication and performance (such as trim materials, $C_v$, tightness, cavitation characteristic, etc.) result in continuous changes to valve models. It is better to approach the choice of a new or replacement valve on a case basis, rather than take the cook book approach of the pipe specification (Chapter 25). The parameters that must be specified in a valve procurement include:

Function
Shutoff class (FCI, API, ASME B16)
Size
Pressure and temperature rating (ASME B16)
Flow rate, noise and cavitation restrictions (sizing)
Valve coefficient $C_v$
Style
Ends: screwed, flanged, butt, socket
Material: body and trim
Special qualifications (fire testing, seismic testing, etc.).
Characteristic: linear, quick opening or equal percentage.
Packing: TFE, Graphite, other.
Bolting: standard or custom material.
Fugitive emissions tightness
Actuator style
Size and thrust
Hand jack: top or side

Keep in mind that control valves, check valves and safety and relief valves must be sized for the service (Chapter 25). It is not sufficient to place a 4” valve in a 4” line. A valve sizing calculation should be prepared either by the piping system designer or by the supplier to confirm that the valve will perform its function.

**Table 4-14** Example of Pipe Specification

| 1 | **P-ABC** | | |
|---|---|---|---|
| 2 | Service: CW | | Material: Carbon Steel |
| 3 | Class 150 | $P_D$=150 psi | $T_D$=40 F min, 120 F max |

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| | | | |
|---|---|---|---|
| 4 | Corrosion Allowance 0. 12” | | Code: ASME B31.1 |
| 5 | | **PIPE** | |
| 6 | NPS | SCH. | MAT’L. |
| 7 | ½ to 10 | 40 | ASTM A 106 Gr.B |
| 8 | | **FITTINGS ½” TO 2”** | |
| 9 | 90°, 45° SW or THRD | Class 3000 SW<br>Class 2000 THRD | ASTM A 105, B16.11 |
| 10 | Cross, Tee SW or THRD | Class 3000 SW<br>Class 2000 THRD | ASTM A 105, B16.11 |
| 11 | Coupling SW or THRD | Class 3000 SW<br>Class 2000 THRD | ASTM A 105, B16.11 |
| 12 | Half Coupl’g SW or THRD | Class 3000 SW<br>Class 2000 THRD | ASTM A 105, B16.11 |
| 13 | Cap SW or THRD | Class 3000 SW<br>Class 2000 THRD | ASTM A 105, B16.11 |
| 14 | Plug THRD | Class 2000 | ASTM A 105, B16.11 |
| 15 | Sockolet, Thredolet | Class 3000 | ASTM A 105, MSS-SP-97 |
| 16 | Latrolet SW or THRD | Class 3000 | ASTM A 105 |
| 17 | Elbolet SW | Class 3000 | ASTM A 105 |
| 18 | | **FITTINGS 2–1/2” TO 10”** | |
| 19 | 90°, 45° BW | Sch. 40 | ASTM A 234 WPB, B 16.9 |
| 20 | Cross, Tee BW | Sch. 40 | ASTM A 234 WPB, B 16.9 |
| 21 | Reducer B W | Sch.40 | ASTM A 234 WPB, B16.9 |
| 23 | Weldolet, Sweepolet | Sch.40 | ASTMA 105, MSS-SP-97 |
| 24 | Latrolet BW | Sch.40 | ASTM A 105 |
| 25 | | **FLANGES ½” to 2”** | |
| 26 | SW, THRD or Blind | Class 150 | ASTM A 105, B16.5, RF |
| 27 | | **FLANGES 2–1/2” TO 10”** | |
| 28 | Weld Neck, Slip-On | Class 150 | ASTM A 105, B16.5, RF |
| 29 | Orifice | Class 150 | ASTM A 105, B16.36 SW |
| 30 | | **STUDS, BOLTS AND NUTS** | |
| 31 | Studs or Bolts | - | ASTM A 193 Gr.B7 |
| 32 | Nuts | - | ASTM A 194 Gr.2H |
| 33 | | **GASKET** | |

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| | | | |
|---|---|---|---|
| 34 | All sizes | Class 150 | 1/16” Teflon, B16.21 |

## 4.13 REFERENCES

API 5L, Specification for Line Pipe, American Petroleum Institute, Washington, DC, 1992.
API 12J, Specification for Oil and Gas Separators, 7th ed., American Petroleum Institute, Washington, DC.
API 521, Guide for Pressure-Relieving and Depressurizing Systems, Recommended Practice 521, November 1990, American Petroleum Institute, Washington, D.C.
API 579, Fitness-for-Service, American Petroleum Institute, Washington, D.C.
API 598, Valve Inspection and Test, American Petroleum Institute, Washington, D.C.
API 752, Management of Hazards Associated With Location of Process Plant Buildings, American Petroleum Institute, Washington, D.C.
ASME B31.1, Power Piping, American Society of Mechanical Engineers, New York, NY.
ASME B31.3, Process Piping, American Society of Mechanical Engineers, New York, NY.
ASME B31.4, Liquid Petroleum Transportation Piping, American Society of Mechanical Engineers, New York, NY.
ASME B31.8, Gas Transmission and Distribution Piping, American Society of Mechanical Engineers, New York, NY.
Boardman, H.C., Formulas for the Design of Cylindrical and Spherical Shells to Withstand Uniform Internal Pressure, The Water Tower, Vol. 30. September 1943.
Borzileri, C.V., Department of Energy Pressure Safety Guidelines, M-089, Revision 7, U.S. Department of Energy, Washington, DC.
BS 8010 British Standard, Code of Practice for Pipelines, Part 3. Pipelines subsea: design, construction and installation, British Standard.
Burrows, W.R., Michel R and Rankin, A.W., A Wall Thickness Formula for High-Pressure High-Temperature Piping, American Society of Mechanical Engineers, paper No. 52-A-151, 1952
CFR 49, Code of Federal Regulations CFR 49 CFR Part 192—Transportation of Natural and Other Gas by Pipeline: Minimum Federal Safety Standards, US Department of Transportation, Washington, DC.
Cooper, W.E., The Significance of the Tensile Test to Pressure Vessel Design, Transactions of the ASME, 1957.
CSHI, Catastrophic Vessel Overpressurization, Investigation Report No. 1998–002-I-LA, U.S. Chemical Safety and Hazard Investigation Board, Washington, DC, 1998.
CSHI, Chemical Manufacturing Incident, Investigation Report No. 1998–06-I-NJ, U.S. Chemical Safety and Hazard Investigation Board, Washington, DC, 1998.
Den Hartog, J.P., Strength of Materials, Dover Publications, N.Y.
Fawcett, H.H., Wood, W.S., Safety and Accident Prevention in Chemical Operations, Wiley and Sons.
FCI 70–2, Control Valve Seat Leakage, Flow Control Institute, Cleveland, OH.
Koves, W.J., Nozzle Attachments design Analysis: ASME Code and WRC Bulletins 107, 297, 368, Pressure Vessel Research Council workshop, February 4, 1998, PVRC, New York.
MSS-SP-82, Valve Pressure Testing Methods, Manufacturers Standardization Society of the Valve and Fitting Industry, Vienna, VA.
MSS-SP-97, Integrally Reinforced Forged Carbon Steel Branch Outlet Fittings—Socket Welding, Threaded, and Butt Welding Ends, Manufacturers Standardization Society of the Valve and Fitting Industry, Vienna, VA.
Nadai, A., Theory of Flow and Fracture of Solids, McGraw Hill Book Company, New York.

www.TechnicalBooksPDF.com

Parsons, H.B., Comparison of Rules for Calculating the Strength of Steam Boilers, ASME, December, 1900, Volume XXII of the Transactions of the ASME, American Society of Mechanical Engineers, New York, NY.
Reese, R.T., Kawahara, W.A., ed., Handbook of Structural Testing, Fairmont Press, Lilburn, GA.
Rodabaugh, E.C., A Review of Area Replacement Rules for Pipe Connections in Pressure Vessels and Piping, Welding Research Council Bulletin 335, september, 1998, Pressure Vessel Research Council, N.Y.
Rosenfeld, M.J., Baldwin, R., Evaluation of End Fillet Details for Hot Tap Fittings, Proceedings of the International Pipeline Conference, September 29, 2002, Alberta, Canada.
Sang, Z.F., et. al., Limit Analysis and Burst Test for Large Diameter Intersections, Pressure Vessel Research Council, Project 96–17, Pressure Vessel Research Council, New York.
Sims, J.R., Hantz, B.F., Kuehn, K.E., A Basis for the Fitness for Service Evaluation of Thin Areas in Pressure Vessels and Storage Tanks, PVP—Volume 233, ASME, 1992.
Sims, J.R., Development of Design Criteria for a High Pressure Piping Code, High Pressure Technology—Design, Analysis, and safety of High Pressure Equipment—Proceedings of the Pressure Vessel and Piping Conference, Vol. 110, 1994, ASME, New York.
Stickles, R.P., and Melhem, G.A., How Much Safety is Enough?, Hydrocabon Processing, October, 1998.
USNRC, United States Nuclear Regulatory Commission, Standard Review Plan, NUREG-0800, Section 3.6.1 Plant Design for Protection Against Postulated Piping Failures in Fluid Systems Outside Containment, NRC, Office of Nuclear Reactor Regulation, 1990, Washington, D.C.
Widera, G.E.O., and Wei, Z., Parametric Finite Element Analysis of Large Diameter Shell Intersections, Part-1 Internal Pressure, April, 1997, Pressure Vessel Research Council project report 95–15, PVRC, New York.

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# 5
# External Pressure

## 5.1 BUCKLING PRESSURE

The behavior of piping and tubing subject to external pressure has been well understood since the early 1900's [Jasper, Roark, Saunders, Southwell, Timoshenko]. Consider a long, perfectly circular cylinder subject to uniform external pressure. By long cylinder, we mean a cylinder longer than a critical length given by

$$L_c = 1.11 D \sqrt{\frac{D}{t}}$$

D=diameter, in

t=wall thickness, in

If the external pressure is steadily increased, there will come a point where the cylinder will suddenly buckle. If the cylinder is long and thin, this buckling will occur while the cylinder wall is still elastic. The external pressure at which elastic buckling occurs is called the critical elastic pressure and is given by [Den Hartog]

$$P_{CE} = \frac{1}{1-\upsilon^2} \frac{EI}{R^3} (n^2 - 1)$$

$P_{CE}$=critical elastic external pressure at buckling, psi

v=Poisson ratio of material

E=modulus of elasticity, psi

I=cross section moment of inertia of cylinder wall per unit length ($t^3/12$), $in^4$

t=cylinder wall thickness, in

R=radius of cylinder, in

n=integer equal to 2, 3,…

At the lowest critical pressure, the pressure corresponding to n=2, the cylinder buckles into an oval shape. If this shape is not physically possible because of external constraints, then the pressure will keep increasing till it reaches the value corresponding to n=3 and the cylinder will buckle in a profile with three lobes. For the first buckling mode (n=2) of a thin pipe, the critical pressure can also be written as [Den Hartog, Bednar]

$$P_{CE} = \frac{2E}{1-\upsilon^2} \left( \frac{t}{D} \right)^3$$

The corresponding hoop strain is obtained by dividing the hoop stress by E

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$$\varepsilon_{CE} = \frac{P_{CE}D}{2t}\frac{1}{E} = \frac{1}{1-\upsilon^2}\left(\frac{t}{D}\right)^2$$

$\varepsilon_{CE}$=hoop strain at collapse pressure of a long section of pipe

For steel, v=0.3, therefore

$$P_{CE} = 2.2E\left(\frac{t}{D}\right)^3$$

and the corresponding hoop strain is

$$\varepsilon_{CE} = \frac{P_{CE}D}{2t}\frac{1}{E} = 1.1\left(\frac{t}{D}\right)^2$$

For short sections of pipes ($L<L_C$) with stiff ends, for example a short section between a flange and a valve acting as stiffeners, the stress in the pipe wall may reach the elastic limit before buckling occurs. The pipe will therefore undergo failure by plastic deformation rather than buckling. This occurs when the pressure reaches yield

$$\frac{P_P D}{2t} = S_Y$$

$P_P$=plastic limit pressure for short pipes, psi

$S_Y$=material yield stress, psi.

For intermediate lengths the buckling pressure of steel cylinders can be approximated by [Harvey, Bednar]

$$P_{IC} = \frac{2.6E(t/D)^{2.5}}{L/D}$$

$P_{IC}$=buckling pressure of intermediate length pipe section, psi

The corresponding hoop strain is obtained by dividing the hoop stress by E

$$\varepsilon_{IC} = \frac{P_{IC}D}{2t}\frac{1}{E} = \frac{1.3(t/D)^{1.5}}{L/D}$$

$\varepsilon_{IC}$=hoop strain at collapse pressure for an intermediate section of pipe

In reality, a pipe cross section is not perfectly round, it has an initial ovality, measured by

$$\Delta = \frac{D'}{D} - 1$$

$\Delta$=ovality of cross section

$D'$=maximum measured diameter on oval cross section, in

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D=nominal diameter, in

Unless the initial ovality Δ is negligible, it is intuitive that the oval cross section will buckle more readily than a perfectly circular cross section. A 3% ovality limit is imposed in ASME B31.3 for piping subject to differential external pressure. The critical elastic buckling external pressure of an initially oval pipe is obtained by resolving the following equation

$$P_{OB}^{\ 2} - \left\{ S_Y \frac{t}{R} + (1 + 6\frac{R}{t}\Delta) P_{CE} \right\} P_{OB} + S_Y P_{CE} \frac{t}{R} = 0$$

$P_{OB}$=buckling pressure of initially oval cross section, psi

$S_Y$=material yield stress, psi

## 5.2 ASME CODE DESIGN

The design of piping subject to external pressure usually follows the rules for pressure vessels, as given in the ASME Boiler & Pressure Vessel Code. The ASME design is based on two sets of charts [ASME VIII, ASME II]. The first set of curves, provides the hoop strains $\varepsilon_{CE}$ and $\varepsilon_{IC}$, called factor A, against the ratio L/D of the length L between stiffener divided by the pipe diameter D, plotted for different values of D/t. If L<LC (plastic collapse)

$$A = \varepsilon_{IC} = \frac{1.3(t/D)^{1.5}}{L/D}$$

If $L>L_C$ (elastic collapse)

$$A = \varepsilon_{CE} = 1.1\left(\frac{t}{D}\right)^2$$

For example, for L/D=10 and D/t=20, we read from the ASME chart [ASME II] A=0.003, which corresponds to $A=1.1(1/20)^2$. The second set of ASME curves permits the calculation of the allowable external pressure $P_a$, which is defined as 1/3 the collapse pressure $P_{CE}$ or $P_{IC}$ (called $P_C$) for D/t≥10 [Farr].

$P_a=P_C/3$

Given factor A and the material's Young modulus, we read from the second set of ASME curves a factor B that is half the hoop stress at the critical pressure $P_C$

$B=½[P_CD/(2t)]$

Therefore [ASME VIII]

$P_a=(4/3)\ B/(D/t)$

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For example, given factor A=0.003 for carbon steel at ambient temperature, we read factor B=16,000 psi. Therefore, with B=16,000 psi and D/t=20, we calculate the allowable external pressure as $P_a$=1067 psi.

## 5.3 REFERENCES

ASME II, ASME Boiler and Pressure Vessel Code, Section II, Materials, Part D, Properties, Subpart 3, Charts and Tables for Determining Shell Thickness of Components Under External Pressure, American Society of Mechanical Engineers, New York.

ASME VIII, ASME Boiler and Pressure Vessel Code, Section VIII Division 1, Rules for Construction of Pressure Vessels, AG-28 Thickness of Shells and Tubes under External Pressure, American Society of Mechanical Engineers, New York.

Bednar, H.H., Pressure Vessel Design Handbook, Krieger Publishing Company, Florida.

Den Hartog, Advanced Strength of Materials, Dover Publications, New York.

Farr, J.R., Jawad, M.H., Guidebook for the Design of ASME Section VIII Pressure Vessels, ASME Press, New York.

Harvey, J.F., Theory and Design of Pressure Vessels, Van Nostrand Reinhold.

Jasper, T.M., Sullivan, J.W., The Collapsing Strength of Steel Tubes, Transaction of the American Society of Mechanical Engineers, vol. 53, 1931, American Society of Mechanical Engineers, New York.

Roark, R.J., Young, W.C., Formulas for Stress and Strain, McGraw Hill Book Company, New York.

Saunders, H.E., Windenberg, D.F., Strength of Thin Cylindrical Shells Under External Pressure, Transaction of the American Society of Mechanical Engineers, vol. 53, 1931, American Society of Mechanical Engineers, New York.

Southwell, R.V., On the Collapse of Tubes by External Pressure, Philos. Mag., vol. 29, p. 67, 1915.

Timoshenko, S., Theory of Plates and Shells, Engineering Societies Monograph, McGraw Hill, NewYork.

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# 6
# Layout and Supports

## 6.1 SPACING OF PIPE SUPPORTS

The weight of piping and components must be supported to achieve five objectives: (1) minimize stresses in the piping, (2) maintain the intended layout and slope, (3) avoid excessive sag, (4) minimize reactions on equipment nozzles, and (5) optimize the type, size and location of pipe supports. To achieve these objectives, and given the pipe routing, the design process starts by placing weight supports at regular intervals, following a support spacing guide such as given in Table 6-1.

**Table 6-1** Support Spacing for Steel Pipe [ASME B31.1]

| Pipe Size (in) | Water (ft) | Gas (ft) |
|---|---|---|
| 1 | 7 | 9 |
| 2 | 10 | 13 |
| 3 | 12 | 15 |
| 4 | 14 | 17 |
| 6 | 17 | 21 |
| 8 | 19 | 24 |
| 12 | 23 | 30 |
| 16 | 27 | 35 |
| 20 | 30 | 39 |
| 24 | 32 | 42 |

This is a classic spacing table for steel pipe. It is based on a maximum bending stress of 2300 psi and a maximum sag at mid-span of 0.10”. In practice, longer spans are usually feasible, with a deadweight bending stress in the order of 5000 psi to 10,000 psi; provided the sag between supports remains acceptable.

As a rule of thumb (by close examination of Table 6-1) the spacing of pipe supports for steel pipes in liquid service, expressed in feet, may be taken as the nominal pipe size, expressed in inches, plus ten. For example, the spacing of pipe supports on a 6” line will be approximately 6+10=16 feet.

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This spacing changes at high temperatures and for materials other than steel [Bausbacher]. For example, for copper tubing support spacing varies from 8-ft for 1"tubing to 12-ft for 4" [Grinell]. For PVC pipe the spacing depends on the pipe schedule and operating temperature. The span is 4-ft for ¾" pipe, up to 6-ft for 4" schedule 40 pipe at ambient temperature; approximately 1-ft more for schedule 80, and half that spacing at 150°F [Grinell]. For fiber reinforced plastic pipe (FRP), support spacing in liquid service would vary from around 11-ft for 2" pipe to 22-ft for 8" pipe; and for gas service 17-ft for 2" to 40-ft for 8" [Smith Fiberglass]. For high density polyethylene (HDPE) the spacing would vary from 7-ft for 4" pipe with SDR (diameter over thickness) of 11.0, to 16-ft for 24" pipe SDR 11.0 [Driscopipe].

In the simplest cases, the reactions, moments and deflections of pipe spans due to weight can be estimated using beam formulas. For example, in the cases illustrated in Figure 6-1, the reactions (R), moments ($M_E$ at end, $M_C$ at center and $M_L$ under load) and sag (d) of pipe spans are:

| | | | |
|---|---|---|---|
| Case (a): | R=wL/2 | $M_C=wL^2/8$ | $d=(5wL^4)/(384EI)$ |
| Case (b): | R=wL/2 | $M_E=wL^2/12$ | $d=(wL^4)/(384EI)$ |
| Case (c): | R=wL/2 | $M_C=wL^2/9.3$ | $d=(2.5wL^4)/(384EI)$ |
| Case (d): | R=P | $M_E=Pb$ | $d=Pb^2(3L-b)/(6EI)$ |
| Case (e): | R=5wL/8 | $M_E=wL^2/8$ | $d=wL^4/(185EI)$ |
| Case (f): | R=Pb/L | $M_L=Pab/L$ | $d=Pm_1m_2/(27EIL)$; with |
| | | $m_1=ab(a+2b)$, $m_2=[3a(a+2b)]^{0.5}$ | |

In these simplified representations, a straight pipe between supports acts as a beam, the supports and adjacent pipe spans act as end restraints, and the pipe span behaves as a beam with end conditions somewhere between simply supported and fixed. A reasonable approximation for the bending stress is

$\sigma=M/Z=wL^2/(10\ Z)$

$\sigma$=bending stress, psi

M=approximate bending moment due to pipe weight, in-lb

w=weight of pipe, insulation and contents per unit length, lb/in

Z=section modulus, $in^3$ (Appendix A)

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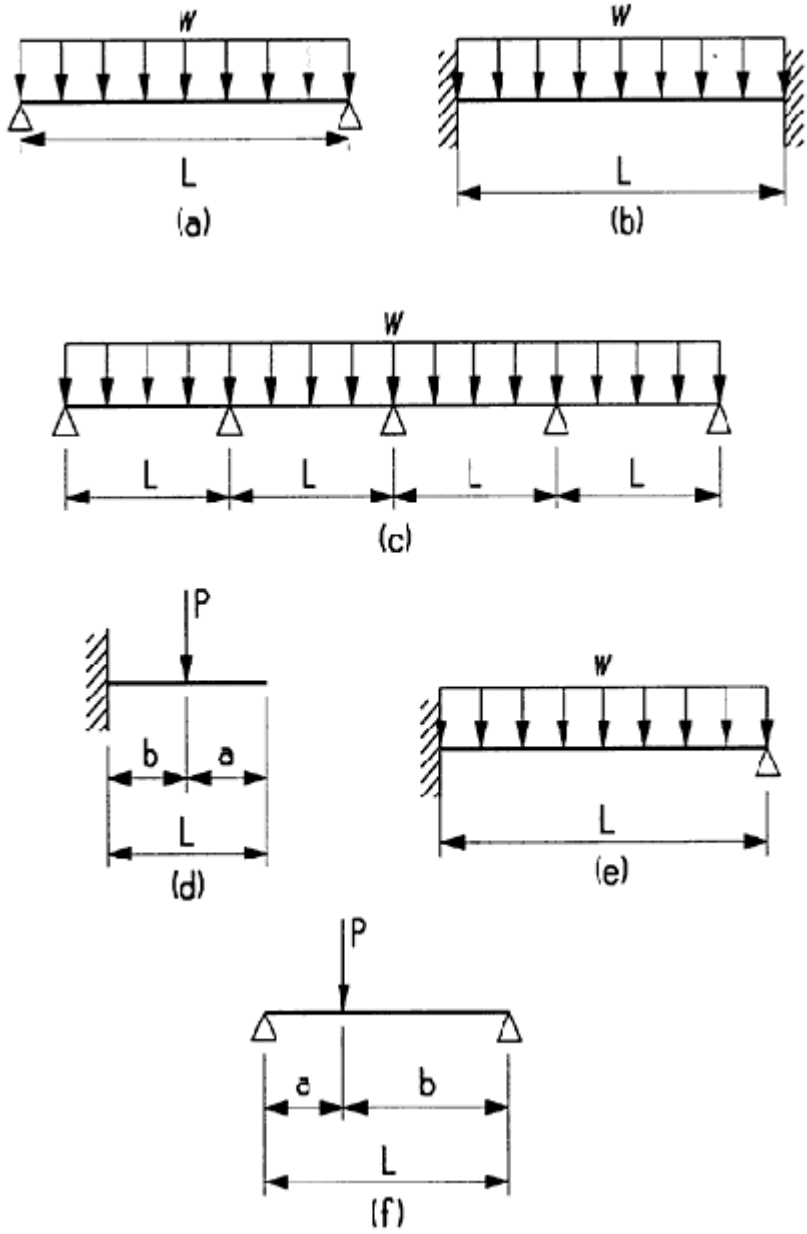


**Figure 6-1** Simplified Pipe Spans

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## 6.2 SUSTAINED STRESS

Under the effect of its own weight, including contents, insulation and in-line components, such as valves or strainers, the pipe will tend to bend downward, causing forces $F_x$, $F_Y$, $F_z$ and moments $M_X$, $M_Y$, $M_Z$, at each point along its length, as illustrated in Figure 6-2.

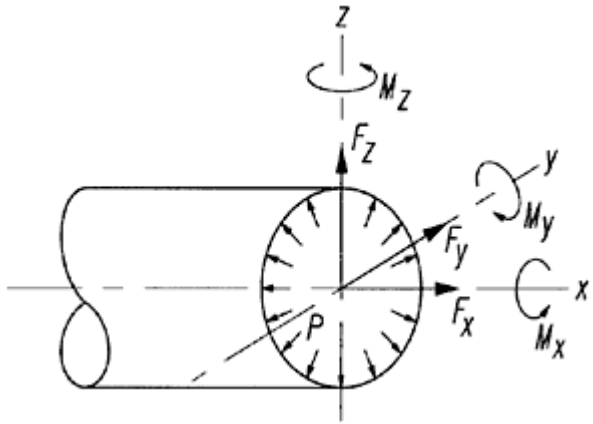


**Figure 6-2** Force and Moment Directions

These weight loads (the axial force $F_x$, shear $F_Y$ and $F_z$, bending $M_Y$ and $M_Z$, and torsion $M_X$) together with the operating pressure are sustained loads causing sustained stresses. In practice, the objective of minimizing deadweight stress is accomplished by limiting the sustained stress to a fraction of the allowable stress *S* at normal operating temperature. A limit on weight bending stress of about 25% of the allowable stress would be a good guideline for steel pipe.

The limit on longitudinal stress due to sustained loads is

$$\frac{PD}{4t} + 0.75i\frac{M_A}{Z} < k_w S$$

P=operating pressure, psi
D=pipe outside diameter, in
t=nominal wall thickness, in
i=stress intensification factor
$M_A$=deadweight moment, in-lb
Z=pipe cross section modulus, $in^3$ (Appendix A)
$k_w$=fraction of allowable weight and pressure stress, for example 0.25
S=allowable stress, psi

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The first term PD/(4t) is the longitudinal stress due to pressure. The second term, 0.75i M/Z [ASME B31.1] is approximately the maximum longitudinal bending stress in the pipe wall due to the combined weight of the pipe, the fluid, the insulation and any component present in the pipe span. This stress is longitudinal, tensile along the bottom fiber of the pipe and compressive along the top fiber. This is an approximate stress; it is neither a bending stress nor a shear stress because, for simplicity, the resultant moment M was selected as the square root of the sum of the squares of the bending and torsional moments.

## 6.3 STRESS INDICES

We will see in Chapter 7 that "i" is a measure of the fatigue capacity of a fitting compared to that of a straight butt weld. But deadweight does not generate fatigue failures, instead, if the pipe weight is excessive, the pipe will bend downward and will eventually deform plastically. For the analysis of non-cyclic loads and movements not causing fatigue, a more appropriate stress should instead protect against excessive plastic deformation to avoid "plastic collapse" of the pipe, that is why the stress index $B_2$, rather than the stress intensity factor "i" is used in ASME III. To understand the stress index $B_2$ we take the case of a pipe elbow or bend [Matzen]. It has been shown that for in-plane bending (bending of an elbow within its plane, bending that tends to open or close the elbow) the maximum stress in the circumferential (hoop) direction is [Larson, Rodabaugh, Schroder]

$S_{max}=(1.8/h^{2/3})M/Z$

$h=tR/r_m^2$

h=flexibility factor

M=nominal, unintensified, moment at elbow, in-lb

Z=pipe section modulus, $in^3$

R=bend radius of elbow at mid-plane, in

$r_m$=mean pipe radius, in

The in-plane limit moment of an elbow (full plastic hinge) is

$M_{in\text{-}plane}=1.5(h^{2/3}/1.8)Z\ S_Y$

For out of plane bending (bending that twists the bend legs in a direction perpendicular to their plane) the maximum stress and limit moment are [Larson, Rodabaugh]

$S_{max}=(1.5/h^{2/3})M/Z$

$M_{out\text{-}of\text{-}plane}=1.5(h^{2/3}/1.5)\ Z\ S_Y$

If all three moments (in-plane and out-of-plane bending, and pure torsion) are combined by SRSS, so that

$M=(M_X^2+M_Y^2+M_Z^2)^{0.5}$

then, the maximum stress would be [Dodge, Rodabaugh]

$S_{max}=(1.95/h^{2/3})\ M/Z$

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The factor $1.95/h^{2/3}$ was originally introduced as a stress index $C_2$ in the 1969 ASME B31.7 piping design code, and was carried over into the current ASME B&PV code section III. With $C_2$ defined as $1.95/h^{2/3}$, the maximum stress in an elbow subject to combined moments is therefore

$$S_{max}=C_2\ M/Z$$
$$C_2=1.95/h^{2/3}$$

The maximum stress reaches the material yield stress $S_Y$ if the bending moment reaches a value $M_Y$ where

$$M_Y=S_Y\ Z/C_2$$

It can also be shown that the moment at plastic collapse of an elbow is 1.5 times the moment at initial yielding of the elbow [Rodabaugh]. The collapse moment is therefore

$$M_{collpase}=1.5\ M_y=S_y Z/[(2/3)C_2]$$

If we define a "plastic collapse" stress index $B_2$ as [Moore]

$$B_2=(2/3)\ C_2$$

Then, collapse occurs when the moment stress reaches a value Mcollpase such that

$$B_2\ M_{collpase}/Z=S_Y$$
$$B_2=2/3\ C_2=1.30/h^{2/3}$$

For conservatism, and because of the scatter in elbow test data, a larger value of $B_2$ equal to $0.75C_2$ instead of $0.67C_2$ was used in the ASME B&PV code, section III [Rodabaugh].

## 6.4 DESIGN STANDARDS

The piping design codes expect the designer to competently support the pipe and design the pipe supports. But, with the exception of ASME B&PV code section III subsection NF for nuclear plant pipe supports, the piping design codes do not provide rules for sizing pipe supports. The designer will have to rely on experience and guidance from publications such as MSS-SP, to select the pipe support types and arrangements. Once the supports have been selected and support loads have been calculated, the support design follows well-established procedures, based on the Manual of Steel Construction and the American Concrete Institute Standards [AISC, ACI, AISI], as we will see later in this chapter.

ASME B31.1 and B31.3 require the design of pipe supports to be based on judgment and simple calculations, with detailed analysis in the more complex cases. Pipe supports can be made of any material suitable for the service, including wood and plastics, but brittle materials, such as cast iron should be avoided in shock applications.

ASME B31.4 refers to MSS-SP-58 and 69, while B31.8 states that piping and equipment must be supported in a "substantial and workmanlike" manner, following good engineering practice. Support materials should be durable and incombustible and

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may be welded to the pipe if the hoop stress in operation (PD/4t) is less than 50% of the pipe material's yield.

The MSS-SP practices for pipe supports cover the following topics:

MSS-SP-58 Materials and Design of Pipe Supports. Allowable stresses and temperatures. Fabrication: forming and welding. Protective coating. Dimensions. Screw threads. Protection saddles and shields. Spring supports and sway braces. Hydraulic and mechanical control devices. Finishing and marking. Inspection and testing

MSS-SP-69 Selection and Description of Pipe Supports. Support selection. Attachment to back-up structure. Support spacing. Insulated lines. Gang supports. Riser supports. Anchors, guides and restraints. Considerations for iron, glass and plastic pipes

MSS-SP-77 Guidelines for Pipe Support Contractual Relationships.

MSS-SP-89 Fabrication and Installation of Pipe Supports. Bill of materials. Fabrication tolerances. Shop fabrication and coating. Testing: proof, qualification and calibration test. Shop inspection. Packaging, marking, shipping, receiving and storage. Installation practice. Field inspection before and after hydrostatic testing. Inspection in operation.

MSS-SP-90 Guidelines on Terminology for Pipe Hangers and Supports. A listing of common pipe support terms and their definition.

MSS-SP-127 Bracing for Piping Systems Seismic—Wind—namic Design, Selection, Application: Selection of braces for dynamic loads and minimum design loads.

Pipe Fabrication Institute standard PFI-ES-26 addresses welded attachments between pipes and supports.

## 6.5 SELECTION OF PIPE SUPPORTS

The term pipe support should be used to refer to a device or assembly that supports the pipe weight. Pipe restraint should be used to refer to a device or assembly that restrains the pipe against lateral or axial movement. Sway brace is used in fire protection sprinkler systems to refer to a device that restrains the pipe against lateral sway. In everyday practice, pipe support is often used to refer to any type of support, restraint or sway brace.

There are basically two categories of pipe supports: the standard (catalog) support and the custom support. The standard support is made from prefabricated components listed in vendor catalogs, rated and qualified by the vendor for use within a certain range of load and movement. The custom support is a custom designed frame or assembly, such as a welded steel frame or pipe rack.

### 6.5.1 Variable Spring

A variable spring hanger supports a hung pipe from above, Figure 6-3. A variable spring support is placed below the pipe and supports the pipe from underneath. Variable spring hangers, Figure 6-3, are standard supports, used to support the pipe's deadweight while allowing vertical movement due to expansion or contraction. The reaction force on a spring hanger varies as the pipe moves vertically, which makes it best suited where vertical expansion and contraction movements are not too large, causing the force in the spring not to vary by more than 25% from cold to hot position.

As the pipe moves horizontally, the spring will swing. A limit of the swing angle is typically specified in the vendor catalog. The spring is selected based on its weight carrying capability and its cold-to-hot travel range. For piping systems operating at temperature, it is advisable to verify, once the line is placed in service, that the calculated movements at the springs do correspond to the observed movements, and that the support contracts and expands within its travel range.

Piping systems solely supported by spring hangers or rod hangers are flexible and are therefore vulnerable to large displacements under vibration or fluid transients.

To illustrate the sizing process for a spring hanger, consider Figure 6-4, with a load during hot operation $F_H$=1000 lb and an ambient to hot movement $D_H$= 0.5" upward. Using the vendor catalog, we select a spring with a hot load of 1000 lb at about midpoint of the spring operating range. For example, in our case, the vendor catalog indicates that there is a spring, Type "ABC", with 1000 lb capacity close to mid travel range. The catalog also indicates a total spring travel range of 2.5", and a spring stiffness of 300 lb/in.

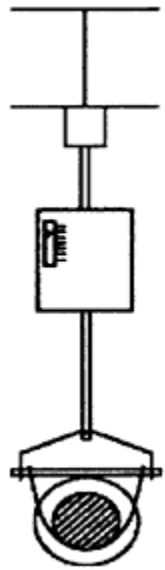

**Figure 6-3** Variable Spring Hanger

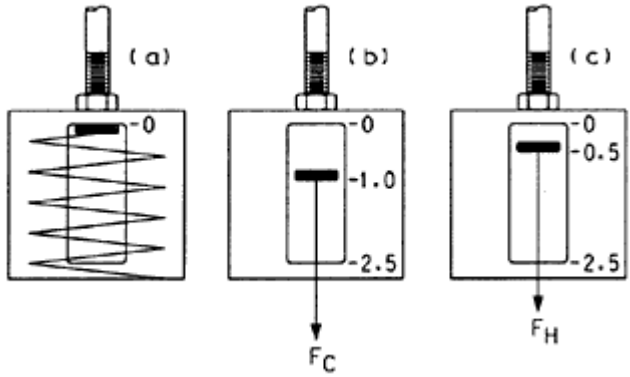


**Figure 6-4** Variable Spring Selection Example

Next, as a matter of good practice, we check that the "spring variability" is less than 25%, where the variability is defined as

$V=100\ (D_H K/F_H)$

V=variability, %

$D_H$=pipe movement from ambient to hot condition, in

K=spring stiffness, lb/in

$F_H$=load in the hot operating condition, lb

In our example V=100 (0.5×0/1000)=15%, which is less than 25%. When the pipe cools down from hot operating to ambient condition, the pipe will move downward by an amount $-D_H$, compressing the spring, and the cold load will be $F_H+K\ D_H$ or, in our case, 1000+(300×0.5)=1150 lb. At 1150 lb, the vendor catalog indicates that spring type "ABC" will be 1" below its zero, no load, top position, well within its total travel allowance of 2.5".

We can now summarize the spring design parameters: when procured, the spring load indicator will be at its top, no load position (see Figure 6-4 (a)). When installed at ambient temperature, holding the pipe weight, the cold load will be $F_C$ =1150 lb and the load indicator will be 1" down from its top zero position, Figure 6-4 (b). As the line heats-up, the load indicator will move up 0.5", Figure 6-4 (c), and the hot load on the spring will be 1000 lb.

### 6.5.2 Constant Load Hanger

A constant load hanger supports a hung pipe from above, Figure 6-5. A constant load support is placed below the pipe and supports the pipe from underneath. Like a variable spring, the constant load hanger, Figure 6-5, is used to support the pipe's deadweight while allowing vertical movement due to expansion or contraction. Unlike a variable spring, the constant load hanger maintains a nearly constant upward load on the pipe as the pipe moves up or down, over a certain range. Constant load hangers are used where the pipe will undergo large vertical movements from thermal expansion, where a variable

spring may have seen load variations in excess of 25%. Limits on swing angle and travel range are similar to variable springs.

The constant spring assembly is usually larger and heavier than a variable spring, and is used primarily where the pipe cold-to-hot vertical travel is too large to be accommodated by a variable spring. For piping systems operating at temperature, it is advisable to verify, once the line is placed in service, that the calculated movements at the hangers do correspond to the observed movements, and that the support operates in its design range.

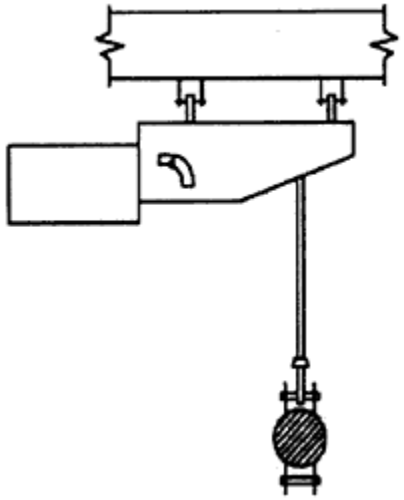

**Figure 6-5** Constant Load Hanger

### 6.5.3 Rigid Frames

Rigid frames, Figure 6-6, are custom designed structures. The quality of materials and fabrication has to be verified during construction. Common materials for carbon steel frames include ASTM A 36 for shapes, ASTM A 53 or A 106 for carbon steel pipe members, ASTM A 500 for tubing, ASTM A 570 for channels, and ASTM A 307 or A 193 for bolts. Support members are typically galvanized or painted. An ASTM A 240 stainless steel plate can be used between stainless pipe and carbon steel supports.

When used to react large loads, the frame has to be sufficiently stiff, with a pipe-to-frame gap ("g" in Figure 6-6) typically no larger than 1/16"+D α AT, where D=pipe diameter, α=coefficient of thermal expansion (Chapter 3), ΔT= rise in temperature. To be rigid, the frame itself may be designed to deflect no more than 1/16" under maximum load [WRC 347].

Friction between the moving pipe and the frame must be minimized, for example by use of pipe rolls or slide plates such as Teflon™ or Lubrite™, or the friction force should be included in the design load.

In outdoor applications, pipe rolls (Section 6.5.5), slide plates or T-shoes also help eliminate line contacts between pipe and steel members that, in the presence of rain water

or condensation, can be the source of crevice corrosion. When frames are used to support multiple pipes, sometimes referred to as “gang supports”, they must be designed to support or restrain the combined tributary load from each pipe. Frames are often made from standard steel shapes or standard catalog struts. For small bore piping and tubing, a common support consists of struts bolted to the wall or ceiling, with the pipe or tube clamped to the strut railing, as shown in Figure 6-7.

Steel frames and their weld details are typically designed to the rules of the American Institute of Steel Construction (AISC) Manual for Steel Construction, and American Iron and Steel Institute standards. The base plate anchor bolts that anchor the steel frame to the structure are designed to the rules of the American Concrete Institute (ACI) standards.

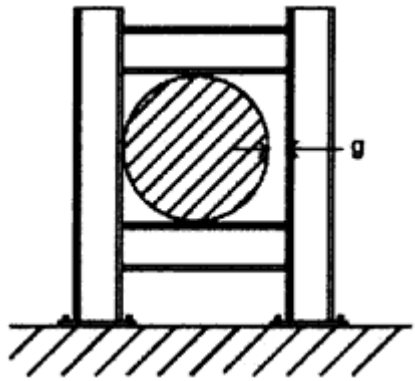


**Figure 6-6** Steel Frame with a Lateral Gap

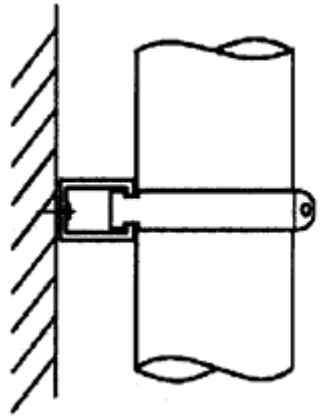

**Figure 6-7** Strut Support Rail and Clamp

### 6.5.4 Rod Hangers

Rod hangers are smooth or threaded rods used to support the weight of hung pipes. They usually have turnbuckles to permit vertical adjustment. Rod hangers are standard catalog components, sized to act as tension members and can buckle in compression. They are load rated for tension and used where upward thermal expansion is not expected. Rod hangers can accommodate some horizontal pipe expansion or sway, which is typically specified in the manufacturer's catalog.

A rod hanger does not accommodate the cold-to-hot vertical movement of the pipe. It is used for systems operating at ambient temperature or at in hot systems at points where the vertical travel is negligible. Limits on swing angle are similar to spring supports. It is often difficult to adjust rod hangers to assure that they are carrying their proper share of the weight, particularly when several rods are installed next to each other on 6" or larger lines. But, unlike variable springs, well-installed rod hangers are quite useful in setting and maintaining a fixed pipe elevation and slope.

In critical applications, for example the first two hangers from rotating equipment, rod hangers can be adjusted at the end of construction to carry the design cold load by (a) using a dynamometer with turnbuckle adjustment, (b) torquing the turnbuckles or rod nuts to a predetermined value based on load and rod diameter.

### 6.5.5 Pipe Rolls

Pipe rolls, Figure 6-8, are used where a pipe undergoes large longitudinal movement and little vertical or lateral movement, for example on long straight steam lines which will be permitted to expand axially thanks to the low friction of the rolls. With time, particularly outdoors, some rolls can corrode or be dislodged. They need to be periodically inspected and maintained.

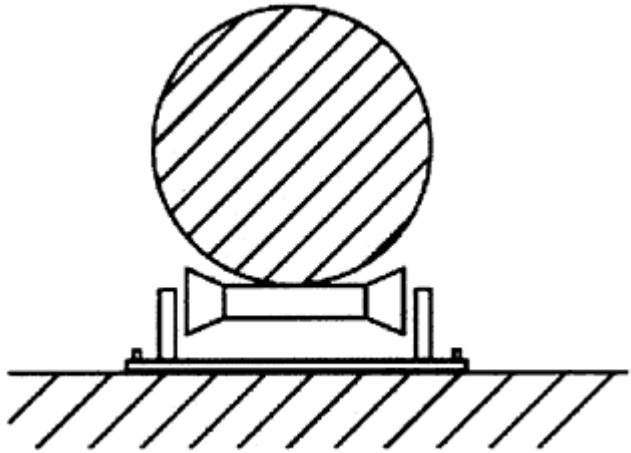

**Figure 6-8** Pipe Roll

### 6.5.6 Rigid Struts

Rigid struts, Figure 6-9, act in tension and compression along their axis. They can be sized to react cyclic and dynamic loads, in which case they are often referred to as "restraints". When used on fire protection sprinkler systems, they are referred to as "sway braces". Two braces placed in a V-shape, will restrain the pipe from moving in the plane of the V.

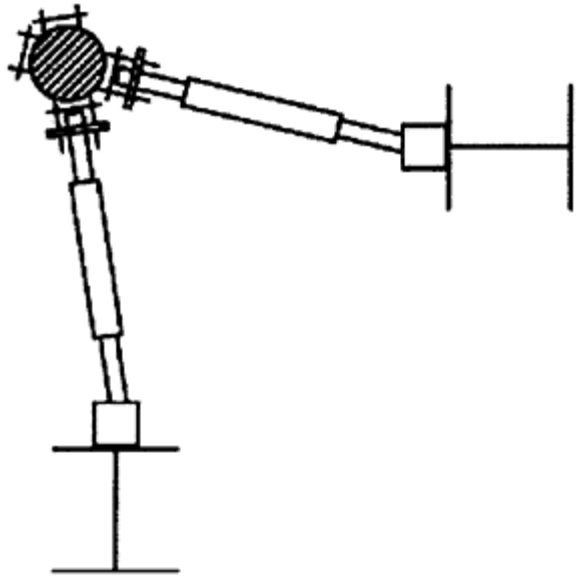

**Figure 6-9** Rigid Strut Tandem

### 6.5.7 Vibration Dampers

Vibration dampers are standard devices that absorb and dampen pipe vibration. They are rated for travel, load and stiffness. It is usually preferable to eliminate the source of vibration rather than dampen its effects, but where this is not feasible, a vibration damper can be used to reduce vibration amplitude. This should be done with care since residual vibration may, in time, fail the damper itself or its attachments by fatigue (Chapter 8). One particular type of damper is the viscous damper. It is an assembly that consists of a plunger attached at one end to the pipe by means of a pipe clamp, and placed at the other end inside an oilfilled cylinder. The viscous damper provides damping and frictional restraint to the piping system (Chapter 8).

### 6.5.8 Snubbers

Snubbers, Figure 6-10, are shock absorbers that act somewhat like a seat belt: they extend or retract to accommodate a slow movement of the pipe (due to thermal expansion or

contraction), but lock under shock (seismic or waterhammer load). They are either hydraulic or mechanical devices, rated based on dynamic load and range of static motion.

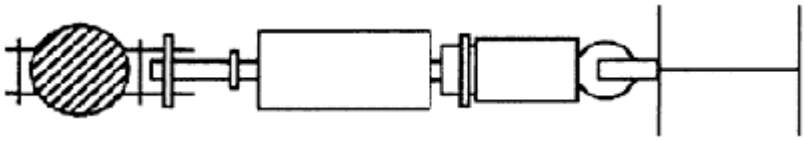

**Figure 6-10** Snubber Installed Horizontally

Depending on the locking mechanism, snubbers are activated either by acceleration or by velocity. Because they contain moving parts or hydraulic fluids, they need to be periodically inspected or tested. They are most useful on hot lines subject to dynamic loads, and are commonly used in nuclear power plants as seismic restraint on hot piping systems. But snubbers must be used judiciously, as their failure by lockup could result in excessive thermal binding of the piping system.

For piping systems operating at temperatures above approximately 150°F, it is advisable to verify, once the line is placed in service, that the calculated design movements do correspond to the observed movements, and that the snubber expands and contracts within its design range.

### 6.5.9 Anchors

Anchors, Figure 6-11, are not to be confused with concrete anchor bolts (section 6.8). Pipe anchors are restraints that constrain the pipe in all degrees of freedom: three translations and three rotations. They are custom made, often times by welding the pipe to a rigid support structure through sleeves, rectangular lugs or round trunions. The construction and stiffness cautions of rigid frames also apply to anchors. Because anchors often involve welding to the pipe wall, caution should be exercised to avoid excessive stresses at the welded attachments. PFI standard ES-26 provides guidance for welding a support to the pipe wall (Chapter 1). The anchor has to be sized to withstand the concurrent forces and moments from both sides (upstream and downstream pipe) of the anchor point.

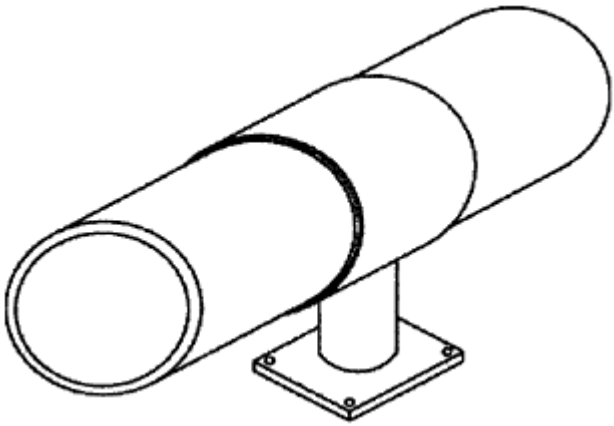

**Figure 6-11** Pipe Anchor with Full Encirclement Sleeve

### 6.5.10 Saddles

Saddles and piers, Figure 6-12, are weight supports and, if sufficiently deep, also act as lateral restraints. U-bolts or straps can be added to provide upward and lateral restrain.

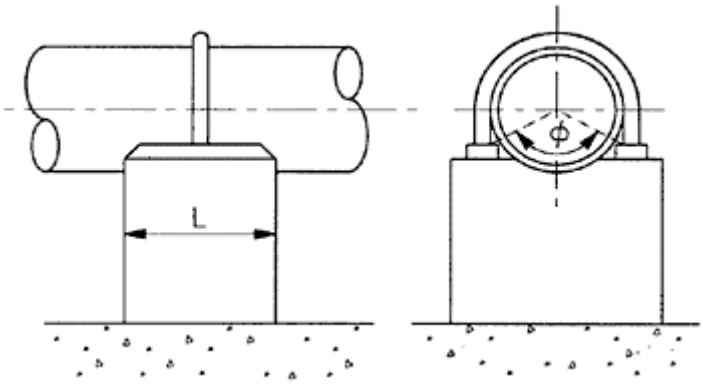


**Figure 6-12** Pier Support with Strap

To avoid excessive contact stresses between pipe and saddle, AWWA recommends limiting the weight supported by a saddle to [AWWA M11]

$$F < \frac{2S_y t^2}{k} \frac{1}{\ln(R/t)}$$

F=maximum pipe load reacted by the saddle, lb

$S_Y$=yield stress of the pipe material, psi
t=pipe wall thickness, in
k=0.02–0.00012(Φ−90)
Φ=saddle-pipe contact angle, degrees
R=pipe radius, in

## 6.6 DESIGN OF STANDARD SUPPORT

Standard supports are designed in accordance with vendor catalogs, given the load applied by the pipe and the movement of the pipe. In all cases the vendor catalog will provide detailed dimensions and installation guidelines. Constant load hangers are listed by travel and load carrying capacity. Variable spring supports are listed by spring deflection (travel range) and load range. Rigid hangers are listed by maximum recommended load. Vibration dampers are listed by stiffness and spring travel. Rigid struts are listed by load rating. U-bolts, clevis hangers, saddles, clamps, upper attachment brackets, turnbuckles, couplings and pipe rolls are listed by maximum recommended load.

## 6.7 DESIGN OF STEEL FRAMES

### 6.7.1 Design

Given the load applied by the pipe, the designer calculates the load distribution (forces and moments) on each member of the frame structure. For the simpler frames this can be done by hand calculations. For more complex and statically indeterminate frames this load distribution would be obtained by computer analysis. Having calculated the load on each member, the designer verifies the members' adequacy [AISC, AISI]. The designer finally sizes the member joints, typically welds, and the concrete anchor bolts [ACI, AISC, AWS, Blodgett].

For example, having determined the forces and moments at a the weld between the rectangular tube steel and a support base plate, Figure 6-14, the weld size is determined treating the weld as a line [Blodgett, AWS]. For simplicity, we assume that the base plate is only subject to shear $F_1$ and bending $M_3$. The weld linear area is $A_w$=2(a+b) (in), the shear force per unit length of weld is $f_1=F_1/ A_w$ (lb/in). The weld bending section modulus reacting $M_3$ is $S_w=ab+b^2/3$ ($in^2$), the bending force per unit length of weld is $f_2=M_3/S_w$ (lb/in). The resultant load per unit length of weld is $f=(f_1^2+f_2^2)^{0.5}$ (lb/in). The required weld size w is finally obtained as $w=f/f_{allowable}$ where $f_{allowable}$ is the allowable stress (psi) determined in accordance with the applicable design code.

The calculated weld size is within a minimum (for example a weld size of at least the tube steel thickness or 3/16" should be used with a ¼" to ½" thick plate) and a maximum (for example 1/16" less than the plate thickness for ¼" or thicker plates) [AISC].

### 6.7.2 Construction

The structural welding code [AWS D1.1] provides requirements for the welding and field erection of steel support frames. Typically, steel support frames are welded by AWS qualified welders, using stick welding and AWS weld details (Chapter 15). The welds are visually inspected for workmanship, with different acceptance criteria depending on whether the structure will be statically loaded or dynamically loaded.

## 6.8 ANCHORAGE TO CONCRETE

Continuing with the example in Figure 6-13, and assuming again that only a shear force $F_1$ and bending moment $M_3$ apply, the four anchor bolts will be subject to shear ($F_1/4$ on each bolt) and the two bolts on the left-hand side of Figure 6-13 will be subject to tension while the concrete on the right-hand side will be in compression. The magnitude of the tension force is

$P=½(M_3/d)$

P=tension force on one of the two bolts in tension, lb

$M_3$=applied bending moment at base plate, in-lb

d=distance between bolt in tension and resultant of compressive force on concrete, in

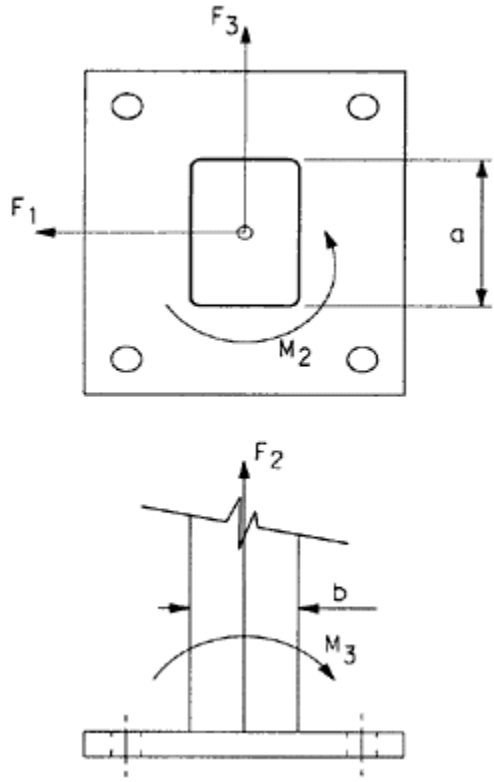


**Figure 6-13** Base Plate Example

The bolt tension depends on the distance “d”, which depends on the assumed distribution of the compressive force, as illustrated in Figure 6-14, cases (a) to (d). The compression can be assumed to have a resultant along the axis of the anchor bolt, as in case (a); or to be a triangular compression on the concrete (with a resultant at 1/3 of the distance from the right-hand outer edge to the centerline), as in case (b), or if the base plate is flexible, as a concentrated reaction at the edge of the steel member, as in case (c) which yields the shortest distance “d” and therefore the largest tension force “P” in the bolts, or as a triangular reaction on the concrete starting at the member’s edge, as in case (d). Having determined the ap-plied shear and tension on concrete anchor bolts (demand), we now compare this demand to the bolt capacity.

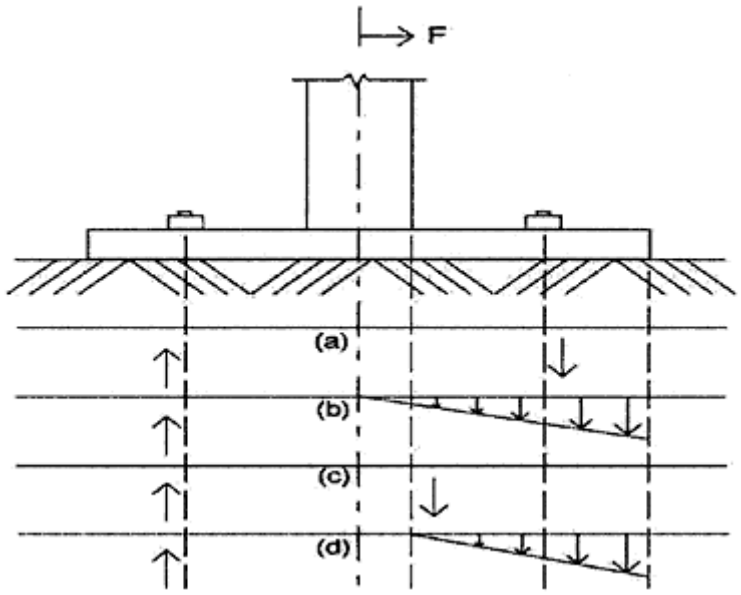


**Figure 6-14** Distribution of Compressive Force on Concrete

First, we note that there are two large classes of concrete anchor bolts: post-installed anchors (Figure 6-15, top row) and cast-in-place anchors (Figure 6-15, bottom row) [ACI 318]. Post-installed anchors are drilled into the concrete after the concrete has been poured. This is the case when installing anchor bolts in existing walls, floors or ceilings, and is common for pipe supports. Cast-in-place anchors are installed with the re-bar and concrete is poured around the bolt during construction. This is common for anchorage of large equipment such as large compressors, pumps and tanks.

There are two groups of post-installed anchors: shell (Figure 6-15, top left) and non-shell (Figure 6-15, top right). There are, in turn, three categories of shell anchors: (1) Self-Drilling: the shell is the drill bit. Once the hole is drilled, it is cleaned and a plug is placed into the hole. The shell is reinserted, expanding over the plug. (2) Non-Drill: same as self-drill, but the shell is hammered over the plug. (3) Drop-In: the hole is drilled and the shell hammered into place. A setting tool expands the shell against the concrete. Non-shell anchors (Figure 6-15, top right) are anchors that penetrate directly the concrete, without a shell surrounding the bolt.

There are two categories of non-shell anchors: (1) Wedge: as the nut is torqued, the bolt pulls up, wedging a clip into the concrete. (2) Sleeve: same as a wedge anchor, but the expanding clip is replaced by an expanding sleeve.

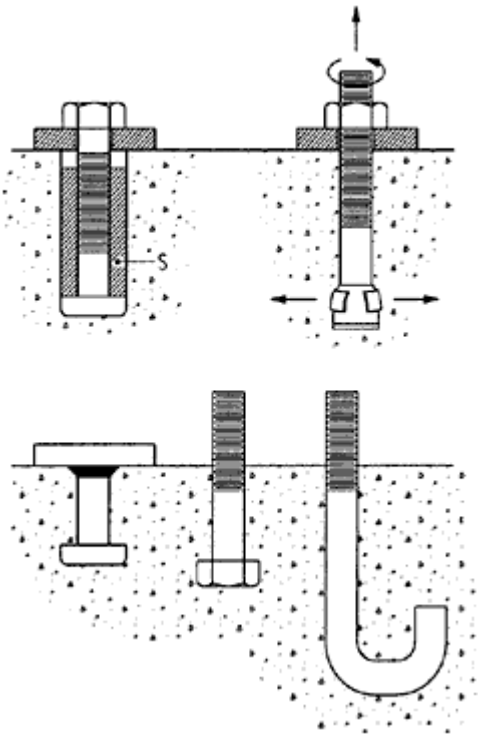


**Figure 6-15** Types of Concrete Anchor Bolts

There are two categories of cast-in-place concrete anchor bolts: (1) Headed Stud (Figure 6-15, bottom left and center), which is a straight bolt with head (the head diameter should have a width of at least 1.5 times the bolt diameter) embedded in concrete or grout, and (2) L- or J-bolt (Figure 6-15, bottom right), which is a steel bar L or J shaped embedded in concrete. The 3/8" to 1" diameter bolts have typically a 3D radius; larger bolts have a 4D radius.

Anchor bolts are typically made of high strength carbon steel, with a yield stress of 75 to 115 ksi and an ultimate strength of 90 to 150 ksi. Material specifications for anchor bolts include ASTM A 193, A 307, A 325, A 354, A 449, A 490 and A 687. Cast-in-place rods are usually carbon steel with a yield stress of 36 to 46 ksi and an ultimate strength of 58 to 70 ksi. Material specifications include ASTM A 36, A 572, A 588, F 1554. High strength rods can also be used, with a yield stress of 105 ksi and an ultimate strength of 125 ksi (ASTM A 193 and F 1554 Gr.105). Concrete anchor bolts can be protected against corrosion by galvanizing (zinc coating) or epoxy coating [ASTM A 153, ASTM B 633].

The total capacity of an anchor bolt in tension and in shear is equal to a nominal value multiplied by penalty factors, where applicable, to account for embedment length, anchor spacing, edge distance, concrete strength and concrete cracks.

$P_C=P_N (X_{EM} X_{AS} X_{ED} X_{CS} X_{CC})$
$V_C=V_N (Y_{EM} Y_{AS} Y_{ED} Y_{CS} Y_{CC})$

$P_C$=tensile capacity, lb
$P_N$=nominal tensile capacity, lb
VC=shear capacity, lb
$V_N$=nominal shear capacity, lb
$X_{EM}$, $Y_{EM}$=embedment length penalty factors for tension and shear
$X_{AS}$, $Y_{AS}$=anchor spacing penalty factors for tension and shear
$X_{ED}$, $Y_{ED}$=edge distance penalty factors for tension and shear
$X_{CS}$, $Y_{CS}$=concrete strength penalty factors for tension and shear
$X_{CC}$, $Y_{CC}$=concrete cracking penalty factors for tension and shear

The nominal tensile and shear capacities are usually established by manufacturer testing [ASTM E 1190, ASTM E 488, ASTM E 1512]. From tensile and shear tests, the manufacturer may define the nominal capacity as

Nominal Capacity=Mean Capacity from Tests/Safety Factor

where the safety factor is typically 3 to 5, depending on the application. Alternatively, the nominal capacity can be defined as

Nominal=Mean (1−2 CV)/(SF)

CV=coefficient of variation of test data=standard deviation/mean
SF=safety factor.

The penalty factors X and Y are determined in accordance with the applicable standards [ACI] and are often specified in anchor bolt vendor catalogs. The embedment length penalty factor ($X_{EM}$ $Y_{EM}$) applies if the bolt is not embedded sufficiently deeply in the concrete, in this case the bolt and a cone of concrete may pull out (Figure 6-16) at a load below the nominal value. The anchor spacing penalty factor ($X_{AS}$ $Y_{AS}$) applies if two bolts are too close to each other. The penalty factors depend on the type of anchor and the manufacturer's recommendations. For example, for undercut anchors [Hilti]

| Anchor Spacing/Embedment Depth | 0.75 | 2.5 |
|---|---|---|
| $X_{AS}$ | 0.65 | 1.0 |
| $Y_{AS}$ | 0.65 | 1.0 |

The edge distance penalty factor ($X_{ED}$ $Y_{ED}$) applies if the bolt is too close to the edge of the concrete structure, in which case the bolt may crack or blow out the concrete edge (Figure 6-17). For example, for undercut anchors [Hilti]

| Edge Distance/Embedment Depth | 0.75 | 2.0 |
|---|---|---|
| $X_{ED}$ | 0.60 | 1.0 |
| $Y_{ED}$ | 0.30 | 1.0 |

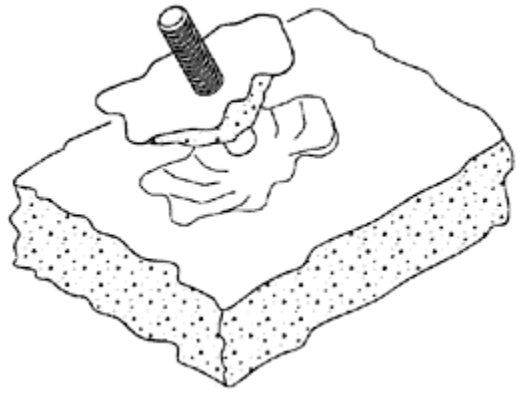

**Figure 6-16** Concrete Failure by Tension, Insufficient Embedment Depth

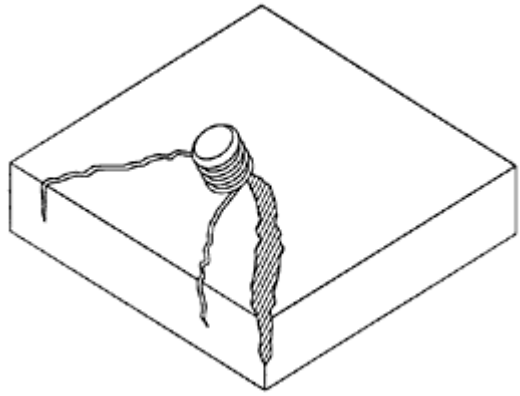

**Figure 6-17** Concrete Failure, Insufficient Edge Distance

The concrete strength penalty factor ($X_{CS}$ $Y_{CS}$) applies if the concrete is too weak (typically if less than 3500 psi strength). The concrete cracking penalty ($X_{CC}$ $Y_{CC}$) applies if the concrete is cracked in the vicinity of the bolt hole.

The penalty factors are applied to the nominal capacities from vendor catalogs to obtain the allowable tension and shear $P_C$ and $V_C$. The calculated tensile and shear force on the bolt P and V (applied demand) are then compared to $P_C$ and $V_C$ using an interaction formula, such as the parabolic interaction (factor 5/3) or a tri-linear approximation to the parabolic interaction [ACI 318]

$$\left(\frac{P}{P_c}\right)^{5/3}+\left(\frac{V}{V_c}\right)^{5/3}\le 1$$

## 6.9 LAYOUT RULES OF GOOD PRACTICE

When laying out equipment and piping systems or pipelines, there are certain rules of good practice that should be followed. These rules are not spelled out in codes or standards, but learned by experience, often times the hard way [Bausbacher, Carucci, Lamit, Payne, Rase].

### 6.9.1 Equipment Elevations

Place large, heavy equipment at grade.
Provide housekeeping pads under equipment to facilitate maintenance.
Provide 3’ to 5’ clearance at bottom of vessels, heat exchangers and tanks.
Provide at least 6” clearance under pipes.

### 6.9.2 Equipment Spacing

Space storage tanks with flammable contents following NFPA rules.
Provide about 3-diameter spacing between vertical vessels.
Provide at least 4’ spacing between horizontal vessels.
Provide room to remove vessel internals and heat exchanger tube bundles.
Provide at least 3’ space all around pumps.
In pump houses, provide at least 6’ wide aisles on either side of pump row.
Provide at least 8’ spacing between compressors.
Make provisions for overhead crane to move or service equipment.

### 6.9.3 Piping

Provide 7” clearance between NPS 2 pipes.
Provide 12” clearance between NPS 6 pipes.
Provide 20” clearance between NPS 12 pipes.
Group pipes to minimize supports and congestion.
Route pipes east-west and north-south, not at angles or diagonals.
On racks, place pipe below electrical conduit or cable trays.
Enter and exit vessels at opposite ends for mixing.
Inlet at top and exit at bottom, unless process dictates otherwise.
Inlet at bottom of inner tubes or coils to keep full on loss of pressure.
Place manholes facing same direction.

### 6.9.4 Valves

Control, check and relief valves must be sized by calculation.
Locate and group valves for ease of access.
Provide retaining clamp to avoid fall of valve chain operators.
Provide extension stems out of confined space.
Place isolation block valves at headers and equipment nozzles.
Chose cast steel body rather than cast iron where there is a risk of dynamic load.

### 6.9.5 Pump Piping

Support pipe at suction and discharge nozzles.
Verify pump alignment before and after connecting pipe.
Install pipe by starting at pump nozzle and piping away.
Provide temporary supports during pipe installation.
Use eccentric reducer with flat atop to avoid trapped air.
Provide 5 straight pipe diameters at inlet and outlet of pump.
Use long radius elbows passed inlet and outlet straights.
Provide strainer at pump suction.
Place check valve between outlet nozzle and outlet isolation valve.
Place relief valve between positive displacement pump and isolation valve.
Mate flat face pump flange to flat face pipe flange.
Align flanges accurately before bolting (Chapter 17).
Follow a written bolt-up procedure.
Provided braided hose or bellows at nozzles if vibration expected (Chapter 8).

### 6.9.6 Compressor Piping

Refer to pump piping.
Check risk of acoustic resonance by analysis (Chapter 8).
Provide for surge chambers (Chapter 8).
Allow for expansion of hot pipeline downstream of compressor.
Avoid condensate pockets at inlet.

## 6.10 REFERENCES

ACI-318 Building Code Requirements for Reinforced Concrete, Appendix D, Anchorage to Concrete, American Concrete Institute, Farmington Hills, Michigan.
ACI-349 Requirements for Nuclear Safety Related Concrete Structures, American Concrete Institute, Farmington Hills, Michigan.
ACI-355 State of the Art Report on Anchorage to Concrete, American Concrete Institute.
AISC, Manual of Steel Construction, Allowable Stress Design, American Institute of Steel Construction, Chicago, IL.
AISI, Specification for the Design of Cold-Formed Steel Structural Members, American Iron and Steel Institute, Washington, DC.

ASME B31.1 Power Piping, American Society of Mechanical Engineers, New York.
ASME B31.7, Nuclear Piping, discontinued, American Society of Mechanical Engineers, New York, 1969.
ASME Boiler and Pressure Vessel Code, Section III, Nuclear Components, 2000 Addenda, American Society of Mechanical Engineers, New York.
ASTM A 153, Standard Specification for Zinc Coating (hot-dip) on Iron and Steel Hardware, ASTM International, West Conshohocken, PA.
ASTM B 633, Standard Specification for Electrodeposited Coatings of Zinc on Iron and Steel, ASTM International, West Conshohocken, PA.
ASTM E 488, Standard Test methods for Strength of Anchors in Concrete and Masonry Elements, ASTM International, West Conshohocken, PA.
ASTM E 1190, Standard Test Methods for Strength of Power-Actuated Fasteners Installed in Structural Members, ASTM, West Conshohocken, PA.
ASTM E 1512, Standard Test Methods for Testing Bond Performance of Bonded Anchors, ASTM International, West Conshohocken, PA.
ASTM F 1554, Standard Specification for Anchor Bolts, Steel, 36, 55, and 105-ksi Yield Strength, ASTM International, West Conshohocken, PA.
AWWA M11, Steel Pipe—A Guide for Design and Installation, American Water Works Association, Denver, CO, 1989.
AWS D1.1, Structural Welding Code, American Welding Society, Miami, FL.
Bausbacher, E., Hunt, R., Process Plant Layout and Piping Design, Prentice Hall.
Blodgett, O.W., Design of Welded Structures, The James F.Lincoln Arc Welding Foundation, Cleveland, OH.
Carucci, V.A., Payne, J.R., Guidelines for the Design and Installation of Pump Piping Systems, Welding Research Council Bulletin 449, New York, 2000.
Crane Publication 53, Valves, Fittings, Pipe, Fabricated Piping, Crane Company, 1952.
Dodge, W.G., Moore, S.A., Stress Indices and Flexibility Factors for Moment Loadings on Elbows and Curved Pipe, WRC Bulletin 179, Welding Research Council, New York.
Driscopipe, Systems Design, Phillips Petroleum Company, Richardson, TX, 1992.
Ellyin, F., Experimental Investigation of Limit Loads of Nozzles in Cylindrical Vessels, WRC Bulletin 219, Welding Research Council, 1976.
Grinell Pipe Hangers, Grinell Corporation, Exeter, NH, 1989.
Gross, N., Experiments on Short Radius Pipe Bends, Institute of Mechanical Engineers, Vol. 1, Series B, 1952.
Hilti, Product Technical Guide, Hilti Corp., Tulsa, OK.
IBC, International Building Code, International Code Council, Falls Church, VA.
Lamit, L.G., Piping Systems Drafting and Design, Prentice Hall, Englewood Cliffs, NJ.
Larson, L.D., Stokey, W.F., Panarelli, J.E., Limit Analysis of Thin-Walled Tube Under Internal Pressure, Bending Moment, Axial Force and Torsion, Transactions of the ASME, Journal of Applied Mechanics, September, 1974.
Matzen, V.C., Tan, Y., Using Finite Element Analysis to Determine Piping Elbow Bending Moment ($B_2$) Stress Indices, WRC Bulletin 472, Welding Research Council, New York, 2002.
Moore, S.E., Rodabaugh, E.C., Pressure Vessel and Piping Codes—Background Changes in the 1981 Edition of the ASME Nuclear Power Plant Components Code for Controlling Primary Loads in Piping Systems, Journal of Pressure Vessel Technology, Vol. 104, 1982, American Society of Mechanical Engineers, New York.
Payne, J.R., PVRC Centrifugal Pump-Piping Interaction Experience Survey, Pressure Vessel Research Council, WRC Bulletin 317, Welding Research Council, 1986.
Rase, H.F., Piping Design for Process Plants, R.E.Krieger Publishing, Florida
Rodabaugh, E.C., and Moore, S.E., Evaluation of Plastic Characteristics of Piping Products in Relation to ASME Code Criteria, NUREG/CR-0261, U.S. Nuclear Regulatory Commission, Washington, D.C., 1978.

Rodabaugh, E.C., Moore, S.E., End Effects on Elbows Subjected to Moment Loading, Pressure Vessels and Piping Conference and Exhibition, Orlando, PVP Vol. 56, 1982.
Schroder, J., Experimental Limit Couples for Branch Moment Loads on 4-inch ANSI B16.9 Tees, Welding Research Council Bulletin, WRC 304, May, 1985, Pressure Vessel Research Council, New York.
Smith Fiberglass, Engineering and Design Guide, Tables 3.1 and 3.2, Red Thread II, Smith Fiberglass Products, Inc. Little Rock, AR, 1992.
WRC Bulletin 353, Position Paper on Nuclear Plant Supports, Welding Research Council, New York, 1990.

# 7
# Flexibility and Fatigue

## 7.1 LAYOUT FOR FLEXIBILITY

Changes in fluid or ambient temperature can have five effects on a piping system: (1) a global or flexibility effect in the form of movements and stresses as the pipe expands and contracts, (2) a local effect in the form of local temperature gradients in the pipe wall as the temperature changes locally, for example when injecting cold water in a hot line, (3) at sufficiently high temperature, creep will take place accompanied by metallurgical changes, (4) changes in mechanical properties, with a loss of toughness at low temperature and a softening at high temperatures, and (5) changes in corrosion mechanisms or corrosion rate. The global effects will be reviewed first. The local and creep effects will be discussed later in this chapter. The changes in mechanical properties are addressed in Chapter 2, and corrosion is addressed in Chapter 20.

The global or flexibility effects that take place as a piping system or pipeline expands or contracts are: (1) movement of the line, (2) forces and moments along the pipe, (3) stresses in the pipe, (4) reactions on supports, and (5) reactions on equipment nozzles.

The first step in any good design and layout process is to understand the first of these five effects: movement of the line. The good designer immerses himself or herself into a three dimensional sketch (isometric) of the line, and tries to intuitively predict how the system will expand or contract as it is placed into service. This first step is often ignored by "analysts" whose understanding of flexibility analysis is to (1) make a model, (2) click on the "run" icon, and (3) check the last output line to read if "stress below code allowable".

Having a good feel for how the line should move, the designer then quantifies the magnitude of movement. Until the 1960's this was accomplished using thermal expansion charts and tables [Spielvogel]. As a tip of the hat to a bygone era, we illustrate how an expansion loop was designed. A vertical expansion loop has a width a and a height h in the middle of a span of total length L (L includes the width a), with an anchor at the both ends. The axial reaction force $F_x$ at each anchor and the maximum bending stress $\sigma_b$ in the loop are [Spielvogel, Grinnell]

$F_x=k_X\ c\ I_p/L^2$

$\sigma_b=k_C\ c\ D/L$

$c=\Delta L\ E/172{,}800$

$F_X$=axial force, lb

c=tabulated expansion factor (for example c=310 for mild steel at 300°F)

$I_P$=moment of inertia of pipe cross section, $in^4$

L=total anchor to anchor length of loop, ft

D=pipe diameter, in

E=Young's modulus of material, psi

ΔL=expansion per 100-ft, in

$k_x$=tabulated force coefficient (for example $k_x$=31.2 if L/a=3 and L/h=3)

$k_b$=tabulated stress coefficient (for example $k_b$=37.4 if L/a=3 and L/h=3)

Today, the designer uses piping analysis computer software to efficiently model the pipe, apply the temperatures and obtain movements and loads throughout the system. A first computerized flexibility analysis should be conducted, with the temperatures applied to the model and including only the restraining effect of weight supports. On the basis of this first analysis, the designer will gain a good understanding of the system flexibility. At this point, restraints, guides and anchors can be added. It is a fallacy to believe that anchors (support arrangement that fully restrain all six degrees of freedom of the pipe, Chapter 6) must be avoided on hot lines. Rather, a few well-placed anchors help balance and direct pipe movements evenly in all directions. In this process it may be necessary to add expansion loops, expansion joints or changes in direction to increase the system flexibility. This is an iterative process, until an optimum configuration is achieved.

To verify that a design is sufficiently flexible, the ASME B31 power piping codes require that the longitudinal moment stress range due to thermal expansion from a cold to a hot condition be limited to a certain allowable value $S_a$. This is written as

$$i\frac{M}{Z} < S_a$$

i=stress intensification factor

M=resultant moment range, in-lb

Z=pipe section modulus, $in^3$ (Appendix A)

$S_A$=allowable stress for thermal expansion, psi

$S_A=f(1.25S_C+0.25S_h)$

$S_C$=allowable stress at minimum (cold) metal temperature, psi

$S_h$=allowable stress at maximum (hot) metal temperature, psi

N=number of cold-hot temperature fluctuations.

f=stress range reduction factor

$f=6(N)^{-0.2}\leq 1$, with the following values of "f" [ASME B31.3]

| | |
|---|---|
| 7,000 and fewer cycles, | f=1 |
| 7,000 to 14,000 cycles, | f=0.9 |
| 14,000 to 22,000 cycles, | f=0.8 |
| 22,000 to 45,000 cycles, | f=0.7 |
| 45,000 to 100,000 cycles, | f=0.6 |
| 100,000 to 200,000 cycles, | f=0.5 |
| 200,000 to 700,000 cycles, | f=0.4 |
| 700,000 to 2,000,000 cycles, | f=0.3 |

If the system undergoes a series of cycles $N_i$ at different stress ranges $S_i$, then an equivalent number of cycles N must be calculated to select f, with

$$N = N_E + \sum_i r_i^5 N_i$$

N=equivalent number of cycles

$N_E$=number of cycles at the maximum stress range $S_E$

$N_i$=number of cycles at stress range $S_i$

$r_i=S_i/S_E$

$S_E$=maximum stress range, psi

$S_i$=stress range with $N_i$ cycles, psi

The equation $iM/Z<S_a$ is also referred to as the flexibility stress equation or secondary stress equation. Let's first explain what is the "moment range". Consider a piping system or pipeline installed at an ambient temperature of 70°F. Assuming that the construction crew did not force the pipe into alignment during assembly, the bending moment at any point of the installed pipe is nearly zero. If the pipe is now put in service at say 150°F, the line expands and the bending moment at a given point, which we call point P, reaches say–30 in-kips. The negative sign is a convention, which simply means that the line bends, for example, upward or left rather than downward or right. If the line is later put out-of-service on a cold day, with an ambient temperature of say 40°F, the moment may reach for example +11 in-kips at that same point. Note that in this case the contraction mo-ment at 40°F (+11 in-kips) has an opposite sign to the expansion moment at 150°F (−30 in-kips). The moment range at point P is the largest absolute difference in moments from the cold to hot condition, or

$M_P$=absolute value ($M_{P,150}-M_{P,40}$)=41 in-kips

A change in moment magnitude occurs in each of three moment directions: two bending moments $M_x$, $M_y$ and a torsional moment $M_Z$. The resultant moment range is the range of variation of the resultant moment M where

$$M = \sqrt{M_X^2 + M_Y^2 + M_Z^2}$$

The distribution of resultant moments M along a piping system, and therefore the stress are usually calculated by computer analysis of the piping system between anchor points. Today, charts and hand calculations are rarely used, and only in the simplest of system configurations.

## 7.2 SIMPLIFIED FLEXIBILITY ANALYSIS

ASME B31.3 gives an approximate formula for judging the adequacy of the pipe flexibility between anchor points. It applies under the following conditions: (a) the pipe is of uniform diameter, and (b) the pipe is of very simple layout (a bend or two) with terminal anchors. This approximate formula is questionable and of little practical use in an age where piping analysis software is readily available. The formula is presented here for information

$D\,Y/[U^2\,(R-1)^2]<0.03$

$S_E=33.3\,D\,Y\,S_A/[U^2(R-1)^2]$

D=pipe diameter, in
Y=thermal growth of end points, in
U=length of line from one anchor to the other, ft
R=ratio of developed pipe length / U
$S_E$=maximum stress range, psi
$S_A$=allowable expansion stress, psi

Another simplified equation can be applied to evaluate the effect of the thermal growth of a header on an attached branch pipe. We conservatively assume, as illustrated in Figure 7-1, that the branch is stiff, rigidly fixed at A (the second rigid restraint away from the header) and B (the header). If one more rigid restraint C is located between A and B, then if the header point B moves a distance d in the direction restrained by A and C, the reactions in the line are

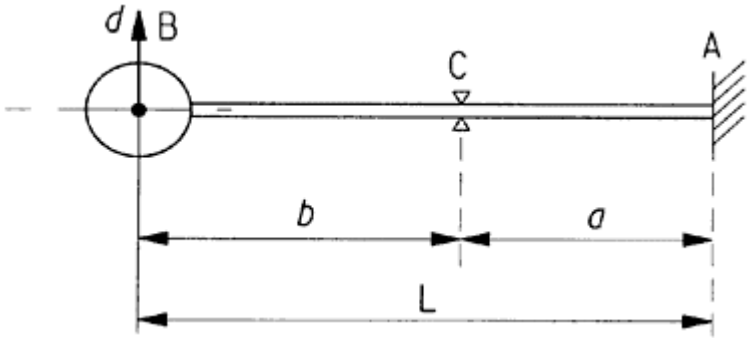


**Figure 7-1** Evaluation of Header Movement

$$R_C = \frac{3EId}{L^3}\left(\frac{3L}{a} - 2\right)$$

$$M_B = \frac{3EId}{L^2}\left(\frac{3L - 2a}{b^2}a + 2\right)$$

$R_C$=reaction load on support C, lb
E=Young's modulus of the pipe, psi
I=moment of inertia of the pipe cross section, $in^4$
d=movement of the header, in
$M_B$=bending moment at the header junction B, in-lb
L, a, b=lengths as indicated in Figure 7-1, in
If there is no support at C, and A and B are still built-in points
$R_A$=12 $EId/L^3$
$M_A$=6 $EId/L^2$

If there is no support at C, and A is built-in but B is simply supported
$R_A$=$R_B$=3 $EId/L^3$

$M_A=3\ EId/L^2$
$M_B=0$

Beam approximations are also useful to resolve field misalignment questions. When two ends of a piping system or pipeline are misaligned, the engineer is called upon to decide if one end can be pulled into alignment with the other, Figure 7-2. Given the pull distance d, we calculate the reactions $R_A$ and RC, and the stresses $iM_A/Z$, $iM_B/Z$ and $iM_C/Z$. The reactions are compared to the capacity of existing restraints at A and C, and the stresses are compared to a reasonably low limit for sustained loads, such as 5000 psi for steel at normal ambient temperature. For conservatism, it is also advisable to increase the calculated reactions and stresses by 1/3.

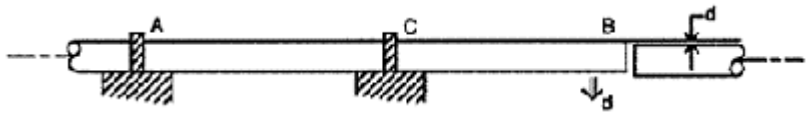


**Figure 7-2** Field Misalignment of Joint

A few rules of good practice are in order when selecting a layout for a piping system operating at temperature: (a) Avoid stiff branches close to expanding headers. (b) A completely free pipe, although very flexible, is not necessarily best because the thermal growth of the pipe has to be guided and balanced, for example to avoid overloading nozzles. (c) Bends and elbows add flexibility to the piping system, reducing the bending moments.

## 7.3 FATIGUE

The allowable stress $S_a$ in the flexibility design equation $iM_C/Z<S_a$, is intended to avoid fatigue failure of the piping system as it undergoes cold-to-hot cycles through its service life. To understand the fatigue limit, we first review the process of fatigue and fatigue failure, in five stages:

Stage 1, slip bands: when metal components are subject to alternating stresses, microscopic slip bands occur at the surface of the metal [Fuchs, Suresh]. The slip bands occur along the planes of maximum shear stress, at 45 degrees from the applied tensile load, shown as stage 1 in Figure 7-3.

Stage 2, microcracks: under continuing alternating stresses, microscopic cracks, in the order of microns in length, form along a slip plane at grain boundaries, shown as stage 2 in Figure 7-3. The duration of stages 1 and 2, the crack initiation phase, can be short if (a) the component contains initial cracks (for example at weld joints) and (b) the local stress is large (stress riser at a discontinuity such as a nozzle or fillet weld).

Stage 3, small cracks: if the applied stress range is sufficiently large, the micro-cracks will grow into small cracks, now visible with dye penetrant. At this point, illustrated as stage 3 in Figure 7-3, the fatigue crack starts to extend through the grain (transgranular)

along the plane of maximum tensile stress (perpendicular to the applied tensile load) and, viewed under a microscope, exhibits the typical fatigue striation surfaces (beach marks).

Stage 4, visible cracks: the crack progresses and is now visible without the help of dye penetrant, Figure 7-3. If fatigue is due to cyclic bending the crack will progress from the outer pipe diameter (maximum tensile stress) towards the inner diameter and progressively around the circumference. The component will spring a leak when the crack has progressed through the wall, and, if the leak can be detected in time, the system may be shutdown before reaching the fracture stage (stage 5). Stages 3 and 4 constitute the crack propagation stages.

Stage 5, fracture: As the visible crack progresses around the circumference, the remaining ligament of metal becomes too small to resist the applied tensile load and will fracture, as indicated by stage 5 in Figure 7-3. The fracture surface in phase 5 has a distinctly coarse look when compared to the crack propagation striations of phase 4.

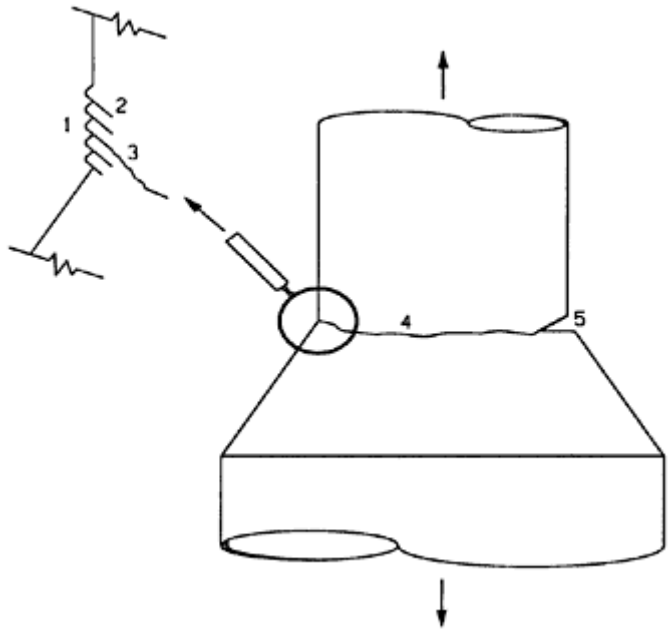


**Figure 7-3** The Five Stages of Fatigue Crack Formation and Propagation

In practice, the duration of the fatigue failure process, from stage 1 to stage 5, the fatigue life of the component, depends on several factors:

(1) The pre-existence of cracks: if the component has a sufficiently large pre-existing base metal or fabrication flaw, such as a weld crack or casting defect, the crack initiation stage is practically non-existent and failure will be reached earlier. Fatigue design standards will therefore differ depending on whether they are based on smooth specimens, without preexisting flaws [ASME II, ASME III, ASME VIII Div.2], or on welded specimens with their inevitable flaws [AASHTO, API RP2A, AWS D1.1,

Barsom, BS 5500, BS 7608]. Similarly, fatigue cracks initiate more readily on a rough surface, particularly for high strength steels, and for low-stress high-cycle fatigue [Bannantine].

(2) The magnitude of the applied cyclic stress: if the applied stress is sufficiently low, it is possible that the fatigue damage mechanism will not progress beyond the microcrack stage and failure will not occur. This is the case when the applied stress is below the endurance limit of the metal. Conversely, if the component has sharp geometric discontinuities, the applied stress will be locally intensified and will lead to the early formation and propagation of fatigue cracks. The endurance limit condition (infinite fatigue life) can be written as

$i\,M/Z<S_{el}$

i=stress intensification factor

M=moment amplitude due to cyclic load, in-lb

Z=section modulus of pipe, $in^3$

$S_{el}$=endurance limit amplitude, psi (from Table 7-1, at $10^{11}$ cycles)

(3) Overloads: Because the endurance limit is caused by the pinning of dislocations by interstitial elements (carbon, nitrogen), it can be affected by overloads, which unpin the dislocation [Bannantine].

(4) Weld quality: the endurance limit $S_{el}$ of a single-side butt weld depends on the quality of the root pass. For constant amplitude fatigue stress cycles, the endurance limit of API 5L girth butt welds is reported to be 2.6 ksi to 3.4 ksi [UK DOE].

(5) Residual stress: Compressive residual stresses on the surface tend to prevent crack initiation, while tensile residual stresses can cause stress corrosion cracking or accelerate the progression of fatigue cracks (Chapters 3 and 21). As-is (non-heat treated) chrome and nickel plating a surface can cause tensile stresses that reduce the endurance limit [Bannantine].

(6) The material: different metals, steels with different mechanical properties (yield, ultimate strength, elongation at rupture, Young's modulus) and different shapes of stress-strain curves have different fatigue properties. Temperature will also affect the mechanical properties and the fatigue life. In carbon steel, differ-ences in local hardness and microstructure also affect fatigue life. Brittle cementite zones are more prone to cracking.

(7) Corrosion: as a fatigue crack propagates, it will reach a size where the fluid can seep into the crack causing corrosion in the crack crevice. We are now faced with the combined effects of corrosion and fatigue, and the resulting acceleration of the fatigue failure process.

(8) Surface finish: Surface finish affects the endurance limit. For example, $S_e$(as forged)=55% $S_e$(mirror polished) for a steel with $S_U$=60 ksi, and $S_e$(as forged)= 30% $S_e$(mirror polished) for steel with $S_U$=140 ksi.

(9) Loading: The endurance limit varies with the type of loading [Bannantine]

$S_e$(axial)~70% $S_e$(bending)

$S_e$(torsion)~58% $S_e$(bending)

## 7.4 SMOOTH SPECIMEN FATIGUE

Fatigue curves used in design are plots of applied alternating stress amplitude S against the number of cycles to failure N. They are often referred to as S-N curves (Figure 7-5). The ASME B&PV Code fatigue curves are presented in ASME B&PV Code Section II, Section III Division 1 and Section VIII Division 2, Appendix I. The ASME fatigue curves are based on cyclic testing of smooth machined specimens of length lo, in air, at ambient temperature. The specimens were subjected to fully reversed cyclic displacements of a fixed amplitude dl, as illustrated in Figure 7-4, or cyclic bending. Note that the applied displacement amplitude dl is constant, for this reason these tests are called displacement controlled. The applied total strain range is then calculated as

$$\Delta\varepsilon_t = \frac{dl}{l_o}$$

$\Delta\varepsilon_t$=total strain range applied to the fatigue specimen

dl=range of elongation of the fatigue specimen, in

$l_o$=initial length of the fatigue specimen, in

For a given material, a series of tests were conducted, each test at a different value of the imposed displacement dl, which corresponds to an imposed strain $\Delta\varepsilon_t$. In each case, the total strain range and corresponding number of cycles required to achieve a 25% drop in applied stress were recorded for the imposed displacement dl. Rather than reporting cycles versus strain ($N_{25\%}$, $\Delta\varepsilon_t$), the ASME fatigue curves report cycle versus an "elastically calculated" fatigue stress amplitude S, which is a pseudo-stress, a number S defined as

$S=E\,\Delta\varepsilon_t/2$

S=elastically calculated fatigue stress amplitude, psi

E=Young's modulus, psi

$\Delta\varepsilon_t$=actual total strain range of the fatigue specimen

This is not the real stress in the specimen since the actual plastic stress is not equal to the strain multiplied by E. The use of this elastically calculated stress is a convention followed in the design of pressure vessels and piping systems. It has been shown that the number of fatigue cycles to failure N is related to the stress amplitude S by a relationship in the form [ASME Criteria, Klanins]

$$S = S_e + \frac{E}{4\sqrt{N}} \ln \frac{100}{100 - RA}$$

S=elastically calculated stress amplitude (actual strain times E), psi

N=number of cycles to failure

RA=percentage reduction in area at fracture, percent

$S_e$=stress amplitude for an infinite number of cycles (endurance limit), psi

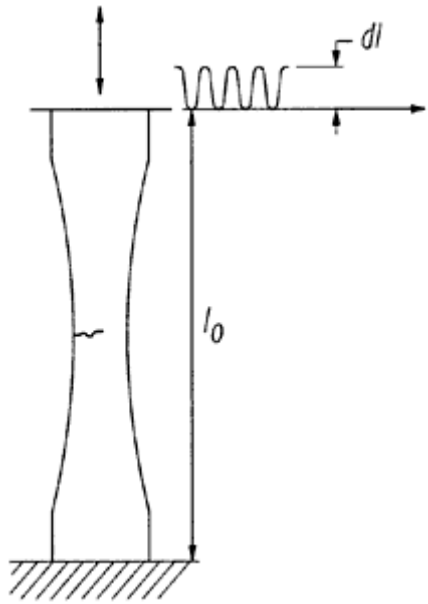


**Figure 7–4** Smooth Specimen Fatigue Test

This expression best fits the experimental data if we use E=26 $10^3$ ksi, RA =68.5% and $S_e$=21.6 ksi for carbon steel; E=26 $10^3$ ksi, RA=61.4% and $S_e$= 38.5 ksi for low alloy steel; and E=26 $10^3$ ksi, RA=72.6% and $S_e$=43.5 ksi for stainless steel.

Having established an actual S-N fatigue curve for a given material, the ASME design fatigue curve can now be plotted by dividing each cycle point of the actual curve by 20 and each stress by 2, and drawing the lower envelope through these reduced values. The curve has the general form shown in Figure 7-5, ranging from 10 cycles to a maximum of $10^6$ to $10^{11}$ cycles depending on the material. Some values of the ASME design fatigue curves, below 700°F, are given in Table 7-1. The range 23.7 ksi to 13.6 ksi depends on the magnitude of local stresses at discontinuities, for large local stresses, the lower stress amplitude applies. The decrease in fatigue life with temperature is accounted for by multiplying the stress amplitude by the ratio of Young's modulus $E_{hot}/E_{ambient}$.

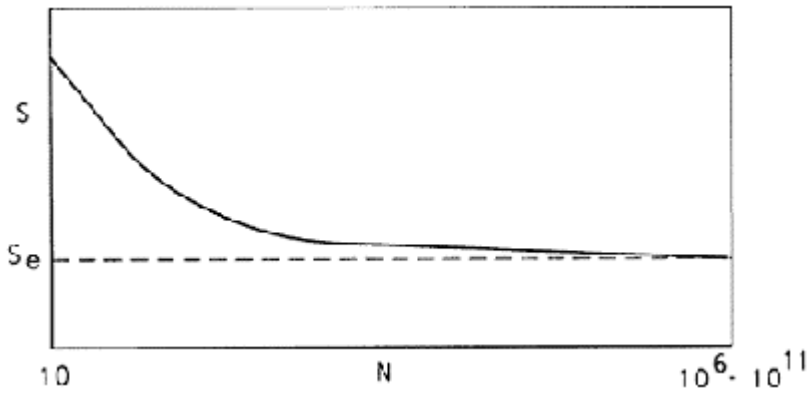


**Figure 7-5** Form of the S (Stress Amplitude) -N (Cycles) Fatigue Curve

The entries in Table 7-1 can cause consternation to engineers used to elastic design. Are entries such as 580 ksi or 420 ksi typographical errors? To understand the very large stress values of Table 7-1, we must explain what is meant by elastically calculated stresses. Elastically calculated means that the stress was computed with no regard for plasticity, as if the specimen subject to a strain ε stayed elastic with a stress Eε, even if ε is a plastic strain. The elastically calculated stress is therefore a fictitious quantity, the product of a real (actual) strain ε times Young's modulus. This explains numbers such as 580 ksi in Table 7-1, a fictitious stress corresponding to an actual strain ε=S/E=580,000 psi/30,000,000 psi=0.019= 1.9%, which is clearly a plastic strain, but a reasonable one (compared to the apparently unreasonable 580 ksi).

**Table 7-1** Elastically Calculated Design Stress Amplitude (ksi) [ASME III]

| Fatigue Cycles→ | 10 | $10^3$ | $10^6$ | $10^{11}$ |
|---|---|---|---|---|
| Carbon steel $S_u$<80 ksi | 580 | 83 | 12.5 | 7 |
| Carbon steel $S_u$=155 to 130 ksi | 420 | 78 | 20 | 11.2 |
| Austenitic Steel | 708 | 119 | 28.3 | 23.7−13.6 |

In many practical cases, the cyclic stress amplitude S fluctuates around an applied mean stress value. For example, the weight of the pipe causes a constant mean stress around which could be superimposed a cyclic stress due to vibration in service, or expansion and contraction. For high cycle fatigue, where the alternating stress amplitude is small (the right hand tail of the S-N curve) an imposed mean stress will result in a downward shift of the fatigue curve (reduced fatigue life). For low cycle fatigue, with large, plastic fluctuating stresses, the mean stress has little effect on the fatigue curve. To understand the effect of mean stress on fatigue life, consider Figure 7-6. The stress cycles between a maximum $S_{max}$ and a minimum $S_{min}$. Its mean (average) value is

$S_m=(S_{max}+S_{min})/2$

and the stress amplitude is

$S_a=(S_{max}-S_{min})/2$

The question at hand is whether, for the cycling shown in Figure 7-6, failure will occur after the same number of cycles N regardless of the value of the mean stress $S_m$. In other words, will fully positive cycles $S_{min}=0$ and $S_m=S_{max}/2$ fail the component by fatigue after the same number of cycles N as fully reversed cycles $S_{min}=-S_{max}$ and $S_m=0$. The answer is yes, if the stress amplitude $S_a$ is larger than yield; no, if the stress amplitude $S_a$ is below yield. This conclusion is based on the following empirical correlation referred to as Goodman's correlation

$(S_N'/S_N)+(S_m/S_U)=1$

$S_N'$=stress amplitude corresponding to failure in N cycles, with mean stress correction, psi

$S_N$=stress amplitude corresponding to failure in N cycles, without mean stress correction, psi

$S_m$=mean stress, psi

$S_U$=ultimate strength of material, psi

For a number of cycles to failure N, the largest mean stress is that for which the maximum stress $S_{max}$ has reached yield

$S_{max}=S_m+S_N'=S_Y$

$S_{max}$=maximum stress, psi

$S_Y$=material yield stress, psi

By substitution into Goodman's correlation, we obtain the stress amplitude for failure in N cycles, corrected for mean stress. If the stress amplitude is less than yield, $S_N<S_Y$, the corrected stress amplitude is [Rodabaugh, ASME Criteria]

$$S_N' = S_N \frac{S_u - S_Y}{S_u - S_N}$$

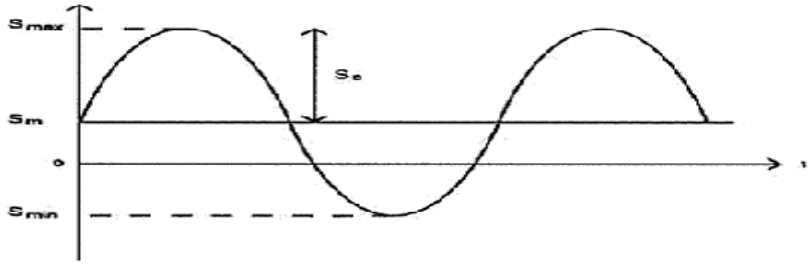


**Figure 7-6** Stress Amplitude $S_a$ and Mean Stress $S_m$

If $S_N \geq S_Y$ the mean stress has no effect on fatigue life

$S_N' = S_N$

Intuitively, if the applied stress amplitude is larger than yield, it overcomes the mean stress, as when a large load overcomes a preload. Goodman's empirical formula is the most common, but not the only one. If in Goodman's formula we replace SU by SY we obtain Sodeberg's correlation [Bannantine]

$(S_N'/S_N)+(S_m/S_Y)=1$

and Gerber's correlation is

$(S_N'/S_N)+(S_m/S_U)^2=1$

To illustrate the effect of mean stress on fatigue life, consider a high strength steel (such as a bolt steel) with an ultimate strength $S_U$=130 ksi and a yield stress $S_Y$=110 ksi. If the steel is subject to cyclic stress with a stress amplitude of 110 ksi (yield) or more, then the mean stress has no effect on the number of cycles to failure since $S_N$ (110ksi)$\geq S_Y$ (110 ksi). Let's say that at 110 ksi it takes $N_{110}$= 2000 cycles to fail the material by fatigue. But, if the applied stress amplitude is less than 110 ksi, for example SN=90 ksi, then the fatigue life must be corrected for mean stress. This is done by noting first the uncorrected life, for example $N_{90}$= 10,000 cycles, and then writing that at 10,000 cycles the failure stress amplitude, with the effect of mean stress, will not be $S_N$=90 ksi but instead it will be

$S_N' = 90\ (130-110)/(130-90) = 45$ ksi

## 7.5 PIPE COMPONENT FATIGUE

The relationship between imposed cyclic stress and cycles to failure in piping components was first investigated in the 1940's by Markl and George [Markl], who took a different approach than the boiler and pressure vessel code S-N curve described in Section 7.4. In Markl's approach, the fatigue life of pipefittings and components was determined by applying a fatigue penalty factor to the calculated alternating stress. This penalty factor, called stress intensification factor, and labeled i, was determined by fatigue tests, first conducted by Markl in the 1940's.

In Markl's tests, the piping component under investigation (a tee, a butt weld, an elbow, etc.) was welded to a straight pipe and rigidly bolted to a wall or floor, as illustrated in Figure 7-7 for a standard butt welded ASME B16.9 tee. The test specimen was then filled with tap water and fully reversible cyclic displacements of a given amplitude d were imposed at the free end of the cantilever arm. As was the case for the ASME B&PV S-N curves, these are displacement-controlled tests. From the cyclic displacement, Markl elastically calculated a stress amplitude S, half the stress range $S_a=(S_{max}+S_{min})/2$ in Figure 7-6.

At some point in the test (in Markl's tests, after 1000 to 100,000 cycles) a crack had progressed through the wall and the water started to leak. As soon as the water started to leak, the test was stopped and the number of cycles N recorded. For ASTM A106 Gr.B pipe butt welds, Markl's tests led to the following relationship between applied stress

amplitude and cycles to failure by through-wall crack of butt-welded straight pipe sections

$S_{ampl}=245{,}000/N^{0.2}$

$S_{ampl.}$=elastically calculated amplitude, psi

N=number of cycles from start of cyclic test to onset of leakage.

As Markl tested standard pipe fittings, such as miter bends, elbows, tees, etc., he noted that the relationship between the recorded failure stress vs. cycles at failure had the same form for all fittings, provided the elastic stress amplitude S was replaced by an intensified stress amplitude iS, where i is a stress intensification factor that depends on the shape and size of the fitting tested. Markl's S-N fatigue failure relationship can be written as

$iS_{ampl}=245{,}000/N^{0.2}$

$iS_{range}=490{,}000/N^{0.2}$

i=stress intensification factor

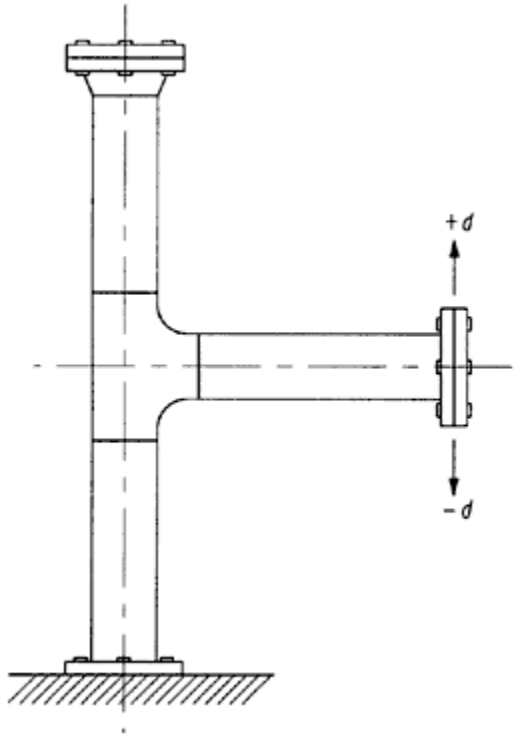


**Figure 7-7** Stress Intensification Factor Test

In Table 7-2 we compare the elastically calculated stress amplitude to failure, Markl's relationship $iS_{ampl}=245{,}000/N^{0.2}$, to the allowable stress amplitude from the design

equation $(iM_{range}/Z)/2<S_a/2=f(1.25\ S_C+0.25\ S_h)/2$, for ASME B31.3 carbon steel $S_C=S_h=20{,}000$ psi. The safety factor between the code design equation and failure is large (10.3) for low cycle/large amplitude fatigue, and reduces (1.4) for high cycle/low amplitude fatigue.

**Table 7-2** Fatigue Stress Amplitude and Cycles

| N cycles | Markl Stress ampl. to failure (ksi) | B31 Allowable stress ampl. (ksi) | Stress ratio Failure/Code |
|---|---|---|---|
| 10 | 155 | 15 | 10.3 |
| 1000 | 62 | 15 | 4.1 |
| 7000 | 42 | 15 | 2.8 |
| 14,000 (f=0.9) | 36 | 13.5 | 2.7 |
| 22,000 (f=0.8) | 33 | 12 | 2.8 |
| 2,000,000 (f=0.3) | 13 | 9 | 1.4 |

Markl's approach to the fatigue design of pipe components, the concept of stress intensification factor and Markl's formulas for i have remained largely unchanged to this day [Rodabaugh, Heald, Wais, Woods]. In particular, testing by Heald and Kiss on Stainless Steel showed that the 245,000 factor applied well to stainless steel, and was practically independent of temperature (they obtained 268,000 at 70°F and 201,000 at 550°F).

For materials other than steel, Markl formula can be applied if adjusted for the material's Young modulus:

$$iS_{ampl}=(E/E_{ref})\ 245{,}000\ N^{-0.2}$$

i=stress intensity factor

$S_{ampl}$=applied stress amplitude, psi

E=Young modulus of the material, psi

$E_{ref}$=Young modulus of steel, 29.3 E6 psi

N=cycles to failure (through-wall crack)

## 7.6 FATIGUE STRENGTH OF SOCKET WELDS

The fatigue failure of socket welded pipe joints has long been a source of concern in vibratory service. Several fatigue tests of socket-welded joints resulted in new relationships between stress amplitude *S* and cycles to failure *N* for small diameter pipe [Higuchi, Kishida, Lida], for example

$$S_{ampl}=145.1\ [(35{,}027/N^{0.458})+41]$$

$S_{ampl}$=alternating stress amplitude, psi

N=number of cycles to failure

In another series of tests, socket welded specimens filled with 50 psi water were tested [Ricardella]. The tests included (a) fillet welds with equal legs (Figure 7-8, with w=h) and an initial pipe end-to-socket gap of 1/16" as required by the ASME B31 design codes, (b) fillet welds with a pipe side leg twice as long as the socket side leg (Figure 7-8, with h=2w), (c) post-weld heat treated welds, (d) welds without the 1/16" gap, and (e) welds with an additional toe weld pass. Based on these tests, it became clear that a fillet weld with a pipe side leg (h) twice as long as the socket side leg (w) had an endurance limit higher than a fillet weld with equal legs. The unintensified endurance limits for equal leg fillets (h=w) compared to h=2w were established at respectively 14 ksi vs 16 ksi for ¾" stainless steel specimen; 8 ksi vs. over 13 ksi for 2" stainless steel pipe; 6 ksi vs. over 12 ksi for 2" carbon steel pipe [Ricardella].

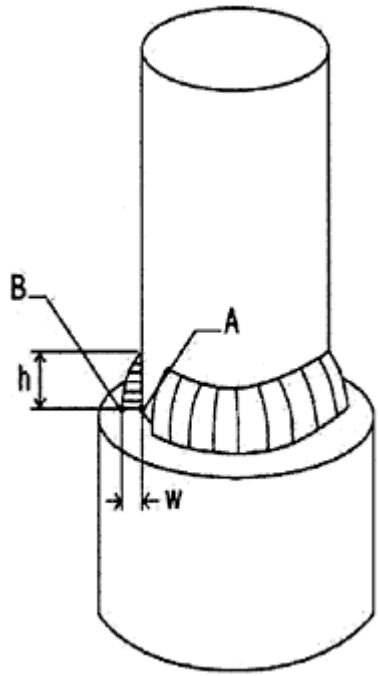


**Figure 7-8** Fillet Weld Legs (w and h), Root (A) and Toe (B)

## 7.7 FATIGUE STRENGTH OF BUTT WELDS

By definition, because Markl compared failure cycles in fittings to failure of butt-welded pipes, a fully penetrated flush butt weld has a stress intensification i= 1.0. Tests on structural steel welds indicate that a flush weld (without weld protrusions) has a fatigue strength close to twice that of a reinforced weld (protruding beyond the pipe outer diameter), and 4 times that of a partially penetrated weld [Reemsnyder].

## 7.8 ASME B31 FATIGUE RULES

The ASME B31 design rules for flexibility are a direct result of Markl, George and Rodabaugh's work on stress intensification factors. The flexibility design equation in ASME B31 is

$iM_C/Z<S_a=f(1.25\ S_C+0.25\ S_h)$

i=stress intensification factor (given in the ASME B31 codes)

$M_C$=moment range, in-lb

Z=pipe section modulus, $in^3$

f=cycle factor

$S_C$=allowable stress at cold temperature, psi

$S_h$=allowable stress at hot temperature, psi

The factor f is 1.0 for a number of cycles less than 7000 and, above 7000 cycles, and is obtained by

$$f = \frac{6}{N^{0.2}}$$

This form of f deserves an explanation. We first see that for a small number of cycles, say N=10, the allowable stress would be very large since f=3.8 in this case. But, for the purpose of design, Markl placed a limit of 1.0 maximum on f, a value that corresponds to N=7000 cycles, which is roughly one cycle a day for 20 years, well above the cyclic duty of most power or process systems.

The formula for the allowable stress range $1.25\ S_C+0.25\ S_h$ is not self-evident. It results from Markl's proposition that the yield stress SY is a conservative limit for bending moment stresses amplitude in piping [Markl]. At that time, the allowable stress for most materials was $S=5/8\ S_Y$. Thus, Markl proposal amounted to an allowable stress $S_Y=8/5\ S=1.6S$. Referring back to Figure 7-6, if the stress amplitude is $S_a=(S_{max}-S_{min})/2=1.6S$, then the stress range is $Smax- S_{min}=2\times1.68$. Since during the hot cycle $S=S_h$ and during the cold cycle $S=S_C$ then, rather than $2\times1.6S$, the range limit is $1.6S_h+1.6S_C$. Further, limiting this range to 80% as an added conservatism leads to a limiting stress range of $1.25S_h+ 1.25\ S_C$. Since this is the total allowable bending stress range, and given that the sustained bending stresses are permitted to reach $1S_h$, the maximum sustained stress of $1S_h$ must be deducted to obtain the allowable stress range for the expansion stress alone, which leads to $1.25S_C+0.25S_h$. For a piping system operating between ambient temperature and a hot operating temperature, the cold condition corresponds to the ambient temperature at initial construction. This temperature is typically selected as 70°F, but in cold outdoor environments a lower value is in order.

A newly constructed carbon steel (ASTM A 106 Grade B) piping system is placed into hot service for the first time. The line is observed during heatup to make sure it moves as predicted. The startup group notices a large bowing at a tee on the tallest vertical leg, Figure 7-9. It did not take long to realize that the two restraints at A and B where installed by mistake, they should not have been there; the tee was supposed to expand freely to the left to accommodate the expansion of the long horizontal run of pipe CD. The line had expanded from ambient temperature, as constructed condition $S_{min}=0$, to the

hot bowed condition, for which the bending stress, elastically calculated, was $S_{max}$=M/Z=64 ksi at the tee. The stress intensification factor for the tee is 1.4 and therefore the intensified stress at the tee in the hot condition was 1.4×64=90 ksi. The intensified stress range from shutdown to hot condition is $S_{max}-S_{min}$=90 ksi−0=90 ksi, and the stress amplitude is 90/2=45 ksi.

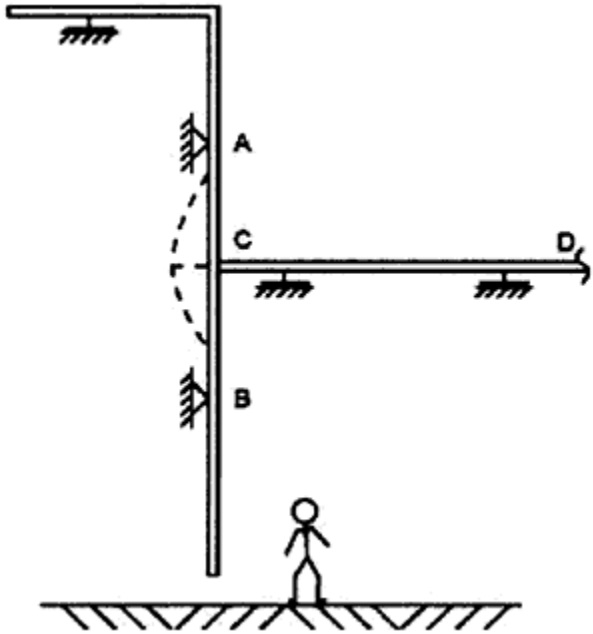


**Figure 7-9** Bowing of Hot Line During Startup

First, we note that the stress range of 90 ksi exceeds the allowable stress for carbon steel f(1.25 $S_h$+0.25 $S_C$)=30 ksi. Because of the design safety factors, and because $S_a$ is a fatigue limit based on 7000 cycles of heat-up and cool-down, the stress range of 90 ksi does not mean that the pipe is in imminent danger of rupture.

Second, we note that the tee has plastically deformed since the stress of 90 ksi exceeds the material's yield stress of 30 ksi. Here again, displacement limited plastic deformation does not mean that the pipe is in imminent danger of rupture. Neither does plastic deformation necessarily mean that when the line is cooled down it will have a permanent deformation. In the case of Figure 7-9, when the line will cool down the contraction of leg CD will re-straighten the vertical leg AB, even though the metal has been plastically deformed at the tee.

Third, we apply Markl's fatigue failure relationship i$S_{ampl}$=245,000/$N^{0.2}$ and note that, if supports at A and B are not removed, the tee would fail by through-wall fatigue crack after a number of heat-ups and cool-downs equal to N =(245,000/45,000)$^5$=4780 cycles.

In light of these observations, the prudent thing to do, unless regulatory requirements dictate otherwise, is to

(a) Cut out supports A and B to bring the line to its intended design configuration. If the supports at A and B are cut while in hot service, the vertical leg will jump to the left. The line was therefore shutdown and cooled down before cutting the supports.

(b) Remove the insulation and verify, visually and by touch, that the pipe has not buckled or wrinkled around the tee

(c) Inspect the tee's base metal and three welds by a nondestructive technique (Chapter 16) to verify that the plastic deformation has not caused the material to crack by opening small pre-existing flaws.

## 7.9 FRACTURE MECHANICS APPROACH

In fracture mechanics, the growth of fatigue cracks can be represented by a crack growth curve that has three regions (regions I, II and III in Figure 7-10). In region I there is a threshold stress intensity factor, below which the fatigue crack does not form or propagate (equivalent to the endurance limit). The threshold stress intensity at a notch depends on the shape of the notch and can be approximated by [Barsom]

$\Delta K_{th}=10\ (r\ S_Y)^{0.2}$

$\Delta K_{th}$=threshold range of stress intensity factor, region I, ksi(in)$^{0.5}$

r=notch radius, in

$S_Y$=yield stress, ksi

The threshold range is between 5 and 15 ksi(in)$^{0.5}$ for steel and 3 to 6 ksi(in)$^{0.5}$ for aluminum [Bannantine]. In region II, which corresponds to the propagation of a crack of depth "a" as a function of applied fatigue cycles "N", the relationship between the crack growth rate da/dN and the range of applied stress intensity is

$$\frac{da}{dN} = C(\Delta K)^m$$

da/dN=crack growth per cycle, in/cycle

m=material and environmental exponent

C=material and environmental coefficient

$\Delta K$=range of applied stress intensity factor, ksi(in)$^{0.5}$ (Chapter 21)

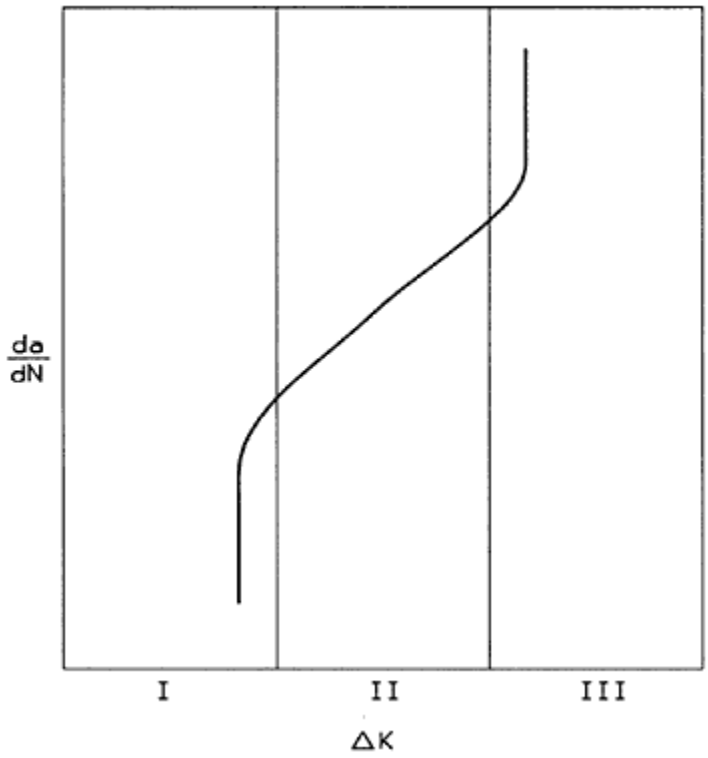


**Figure 7-10** Fatigue Crack Growth Curve log(da/dN) vs. log(ΔK)

The coefficient C depend on the material and the environment, for example in a non-corrosive environment [Fuchs], (C, m)=(3.0 $10^{-10}$, 1) for austenitic stainless steel.

This equation can be modified to capture the shape of the fatigue crack growth curve in regions I and III, leading to

$$\frac{da}{dN} = C(\Delta K)^m \frac{\left(1 - \frac{\Delta K_{th}}{\Delta K}\right)^p}{\left(1 - \frac{K_{max}}{K_c}\right)^q}$$

$K_{max}$=maximum value of the stress intensity, ksi(in)$^{0.5}$

$K_C$=critical stress intensity, fracture toughness, ksi(in)$^{0.5}$

In region III the crack has progressed to the point where the stress intensity K reaches a critical value $K_C$, resulting in fracture of the remaining ligament in the cracked cross section (Chapter 21).

## 7.10 CORROSION FATIGUE

If the formation and propagation of a fatigue crack takes place in the presence of a corrosive fluid, then the existing crack which has been exposed to the corrosive fluid is corroded while the plane of metal just exposed during the last stress cycle is bare. The corroded region, with its passive oxide film, acts as the cathodic (or more noble) pole, while the recently exposed bare steel is anodic and corrodes (Chapter 20). The rate at which the fatigue crack progresses in this case is different than the growth rate in the absence of corrosion. Corrosion therefore influences the shape of the corrosion fatigue curve, da/dN vs. $\Delta K$. Corrosion fatigue curves have been developed for a large number of metals and environments [ASM, Battelle].

## 7.11 SHAKEDOWN

Fatigue cracks tend to originate at sharp structural discontinuities or at existing crack flaws, where the local peak stress is large. This concentrated stress is often well above the material yield stress, creating a local plastic zone in the component. Fatigue design analysis rules generally require as a prerequisite that the material shakes-down to elastic behavior. What is meant is that for elastically based fatigue design rules to apply, the plastic zone must be small, surrounded by elastic material, so that plastic deformation remains confined; and as the cyclic deformation takes place, the strain will not continue to increase, it will not ratchet. In practice it is difficult to prove that shakedown occurs and simplified formulas have been introduced in the ASME Code to check for this condition of confined plasticity.

## 7.12 COLD SPRING

In principle, one way to reduce pipe expansion stresses and reaction loads on equipment nozzles in high temperature service is to cut short the pipe during construction, and pull or push it into alignment with the nozzle during erection, Figure 7-11. The pipe is said to be cold sprung. Then as the system heats up, the pipe will expand and relieve the cold spring imposed during construction, resulting in low stresses or reaction loads in service.

Ideally, if the pipe is cut short and cold sprung by the exact amount of hot expansion, then the reaction load would be zero in hot service. For example, in Figure 7-11, if the pipe will freely expand upwards an amount $\Delta L$, it is cut short by $\Delta L$ during construction, forced into alignment with the nozzle by upward pull during erection, and bolted. Prior to service, while the line is still at ambient temperature, there is an initial downward force applied by the pipe to the nozzle.

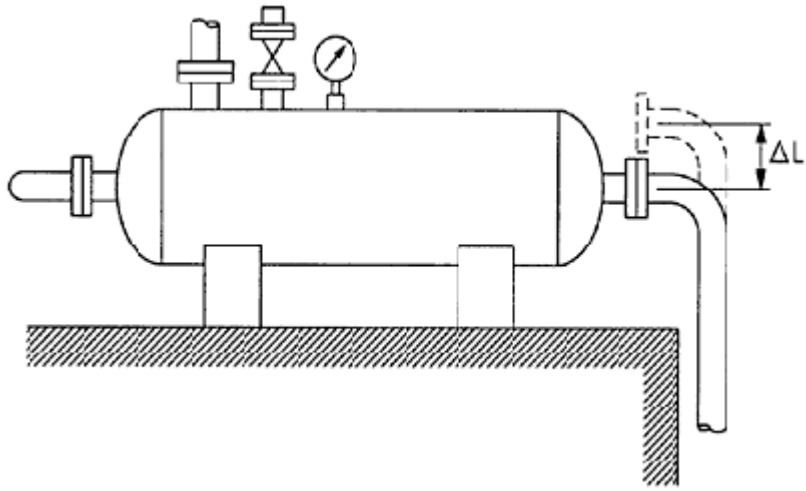


**Figure 7-11** Cut Short Pipe to Reduce Equipment Nozzle Loads

As the line goes into service and expands an amount ΔL, the thermal expansion compensates for the cut short and will relieve the nozzle force, theoretically down to zero. The ASME B31 code only permits to take credit for 2/3 of the beneficial effect of cold spring. This implies that the reaction with cold spring $R_m$ can be obtained from the reaction without cold spring R using the following equation [ASME B31.3]

$$R_m = R\left(1 - \frac{2C}{3}\right)\frac{E_m}{E_a}$$

$R_m$=cold sprung reaction, force or moment, lb or in-lb

R=reaction without cold spring, force or moment, lb or in-lb

C=percent of cold spring

$E_m$=Young's modulus at operating temperature, psi

$E_a$=Young's modulus at ambient temperature, psi

How is cold spring implemented in the field? For small bore pipe, the pipe spool is fabricated short, it is jacked open or closed and installed with the jack. Following tie-in of the spool to the system, the jack is removed, resulting in the applied cold spring load. For larger pipe, threaded studs and welded lugs are used to hold the two pipe ends within cold spring distance.

Cold springing does withstand the test of time. In one case, a cut-short cold sprung steam line was reopened at a flange after 50 years of operation. The two flange faces sprung open and separated by the exact amount of original cold spring marked on the 50-year old design drawing.

## 7.13 THROUGH-WALL TEMPERATURES

In the common case of a steady flow of fluid at constant temperature, with the pipe inner diameter at the temperature $T_a$, the temperature $T(r)$ at any point in the pipe wall is given by [Harvey]

$$T(r) = T_a \frac{\ln \frac{b}{r}}{\ln \frac{b}{a}}$$

$T_a$=temperature at the inside diameter, °F
a=inner radius of pipe, in
b=outer radius of pipe, in
r=radial distance, in

This logarithmic variation of temperature from the inside diameter to the outside diameter approaches a straight line as the pipe gets thinner. For thicker pipe or for quick changes in fluid temperature, the temperature gradient causes stress gradients through the wall. With hot fluid flowing in the pipe, the maximum tangential and axial stresses ($\sigma_t$ and $\sigma_z$) occur at the outer diameter (tension) and inner diameter (compression), while the maximum radial stress occurs at mid-wall (compression).

To illustrate the effects of local temperature gradients on the fatigue life of piping components, consider the 30” diameter header, (b) in Figure 7-12, in which flows hot pressurized water at 600°F and 2000 psi [WRC 376, NRC 88–08]. A 6” branch line, (a) in Figure 7-12, drops vertically 10 ft before reaching a 6 ft horizontal run. An isolation valve is located on the 6 ft horizontal run, (c) in Figure 712, and is normally closed in service. The valve has a piped leak-off tube, (d) in Figure 7-12, to collect any potential packing leaks. Following a few months of operation, the 6” line cracked on the intrados (inside part of the bend, or crotch) of the first 6” elbow, (e) in Figure 7-12.

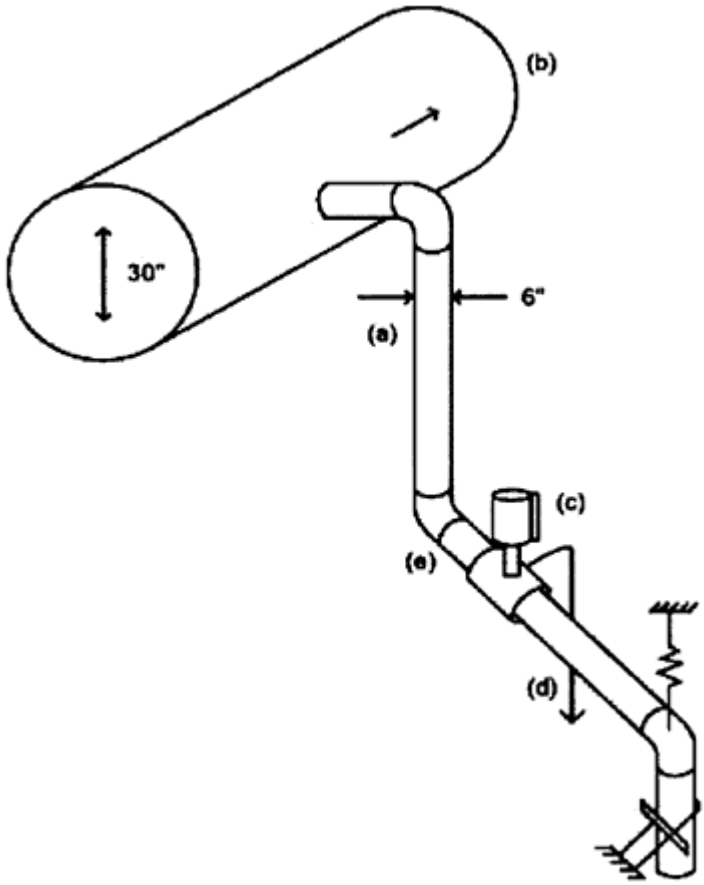


**Figure 7-12** Layout of Header and Branch

The failure was due to local temperature gradients that developed as follows: a small clearance in the isolation valve's gate-to-body seat allowed a trickle flow of 600°F water to flow from the header, Figure 7-12 (b), through the valve stem packing, to the valve leak-off tube, Figure 7-12 (d), to a collection tank. The flow was so slow that the 600°F water stratified in the horizontal leg. The trickle flow at 600°F had a lower density and was slowly flowing towards the valve along the top of the pipe, while at the bottom of the pipe the fluid remained close to ambient temperature. But since the gate was in contact with the 600°F water, its metal expanded and closed the gate-to-body gap, stopping the leakage of hot water through the stem packing. As the leakage flow stopped, the 6" leg cooled down back to ambient temperature. At that point, the valve gate contracted, the gate-to-body gap reopened and the leakage started again. This continuous cyclic heating and cooling of the top of the horizontal 6" branch line caused alternating through-wall stresses and resulted in the initiation and propagation of a fatigue crack.

The onset of flow stratification may be estimated by calculating the Richardson number. If Ri is larger than 1, stratification is likely

$Ri=(d\rho/\rho)\ (gd/v^2)$

Ri=Richardson number
$\rho$=fluid density, $lb/in^3$
$d\rho$=change in fluid density between top and bottom, $lb/in^3$
g=gravity (386 $in/sec^2$)
d=inside diameter, in
v=flow velocity, in/sec

## 7.14 CREEP DAMAGE

Creep manifests itself in two ways: through metallurgical changes and through mechanical changes. The metallurgical changes are addressed in Chapter 20. The first mechanical change is the classic reduction in strength as the temperature increases, illustrated in Table 7-3 for a nickel alloy. The second mechanical change is the reduction of rupture strength with time and temperature, as illustrated in Table 7-4 for a nickel alloy. This effect is even more pronounced for steels. At elevated temperature, if the metal is subject to a constant elastic stress, it will continuously stretch with time, the material creeps until it eventually ruptures [API 579, API 530, ASME NH]. The design rules of ASME III NH-3200 consist in limiting stresses to a time dependent allowable stress $S_t$ obtained as the minimum of (1) the average stress causing a total strain of 1%, (2) 80% of the stress to cause tertiary creep, and (3) 2/3 of the stress to cause rupture.

If, as is often the case in practice, the component is not always held at the creep temperature, but instead cycles in and out of the creep regime, cumulative creep damage must be evaluated. For the first heat-up into the creep regime at a temperature $T_i$ for a period of time $t_i$, the temperature distribution, time, and other concurrent loads are applied to a plastic model of the component (non-linear strain vs. stress at temperature) including creep behavior (strain vs. time at temperature). An accumulated strain is calculated for this first heat-up cycle $\varepsilon_i(T_i, t_i)$. This strain $\varepsilon_i$ is used to enter the material fatigue curve at temperature and obtain the number of fatigue cycles $N_i$ permitted at this strain. The fatigue usage factor is for the actual number of cycles $n_i$ at this temperature $T_i$ in the form $n_i/N_i$. This calculation is repeated for each heat-up cycle ($T_i$, $t_i$) and the cumulative fatigue usage factor is

$$\sum_i \frac{n_i}{N_i} \le 1.0$$

The calculated stress $\sigma_i$ is used to enter a creep rupture life curve (stress vs. rupture time at temperature) and read a maximum allowed time at temperature $t_{im}$. The condition for cumulative rupture life is then written as

$$\sum_i \frac{t_i}{t_{im}} \le 1.0$$

$t_i$=total time at temperature $T_i$, hrs

$t_{im}$=maximum allowed $t_m$ time at temperature $T_i$, hrs

Finally, the fatigue life and creep life usage are combined into a design criterion of the form [ASME NH]

$$\sum_i \frac{n_i}{N_i} + \sum_i \frac{t_i}{t_{im}} \le D$$

**Table 7-3** Mechanical Properties, Nickel Alloy [Special Metals]

| | 70°F | 1000°F | 2000°F |
|---|---|---|---|
| Coefficient Thermal Expansion, 1/°F | $6.0\ 10^{-6}$ | $7.7\ 10^{-6}$ | $9.2\ 10^{-6}$ |
| Modulus of Elasticity, psi | $30.6\ 10^{6}$ | $25.8\ 10^{6}$ | $18.8\ 10^{6}$ |
| Yield Stress, ksi | 50 | 35 | 10 |
| Ultimate Strength, ksi | 110 | 90 | 15 |
| Elongation at Rupture, % | 60 | 65 | 65 |

**Table 7-4** Rupture Strength (ksi), Nickel Alloy [Special Metals]

| | 100 hrs. | 1000 hrs. | 10,000 hrs. | 100,000 hrs. |
|---|---|---|---|---|
| 1100°F | 80 | 60 | 50 | 40 |
| 1500°F | 20 | 15 | 10 | - |
| 1800°F | 6 | 3.5 | 2 | 1.3 |
| 2000°F | 3 | 1.5 | 0.8 | - |

## 7.15 PIPE INSULATION

There are two principal types of pipe insulation: a closed cell insulation for use in cold service, and an open cell insulation for use in hot service. The closed cell insulates the pipe and also has a tortuous path that stops ambient vapors from penetrating and condensing on a cold pipe wall. It is used with a sealed insulation jacket. Cellular glass is an example of closed cell insulation. The open cell insulates the pipe but is fibrous. It lets moisture out from hot pipes. It is used with an overlap insulation jacket. Calcium silicate is an example of open cell insulation.

Insulation must be selected with close consultation with the manufacturer. Examples of pipe insulation materials include [ASTM]:

Cellular glass (ASTM C 552):- 400°F to 900°F, 9 lb/ft$^3$, supplied in pre-shaped sections, closed cell, water resistant, does not easily ignite, used with mastic or metal jacket.

Phenolic foam (ASTM C 1126): -50°F to 250°F, 2 lb/ft$^3$, water resistant, not used on stainless steel or copper, used on plastic pipe, taped and covered with a jacket.

Elastomeric foam (ASTM C 534): 35°F to 180°F, 6 lb/ft$^3$, most often used in commercial rather than industrial applications, water resistant, can ignite with electricity, develops smoke, comes in pre-shaped sections or sheets, taped or glued.

Fiberglass (ASTM C 553): 35°F to 800°F, 4 lb/ft$^3$, Absorbs leaks, good acoustic insulator, taped or secured with metal bands and covered with a jacket.

Mineral wool (ASTM C 547): 250°F to 1200°F, 8 lb/ft$^3$, good acoustic insulator, taped or secured with metal bands and covered with a jacket.

Calcium silicate (ASTM C 533): 250°F to 1200°F, 15 lb/ft$^3$, supplied in pre-shaped sections, heavy but stiff, fire resistant, secured with metal bands and covered with a jacket.

Industrial jackets (lagging) used as insulation covers are commonly made in aluminum, stainless steel or fiber reinforced plastic.

To select an insulation thickness, it is best to consult the insulation vendor catalog. Insulation thickness depends on the insulating material and varies for example from 1" for small bore pipe operating at 150°F to over 5" for 14" pipe operating at 500°F. Properly selected and sized insulation should be able to isolate a 900°F line so that the temperature of a hanger rod 15" above the pipe has dropped close to ambient temperature.

## 7.16 EXPANSION JOINTS

When layout constraints and congestion do not permit the addition of bends and loops to absorb the thermal growth of a piping system, the designer should consider using expansion joints. There are many types of expansion joints, and they each serve a specific purpose. Some joints absorb axial compression, while others absorb lateral movements of the ends [Becht]. Serious accidents have occurred as a result of using the wrong type of expansion joint in a given service. The Flixborough accident illustrates one such case.

In March 1974, one of six steel vessels at the cyclohexane plant in Flixborough, UK, was removed for repair. A temporary 20" pipe spool was installed between existing bellows, at the outlet of the upstream vessel and inlet of the downstream vessel. The pipe had two bends to accommodate the change in elevation between the adjacent vessels, and was supported by a temporary scaffold, Figure 7-13. The pipe size was checked for flow rate and the wall thickness was verified against the system pressure [Lees]. But the effect on the expansion joints of the newly imposed shear force due to the change in elevation in the new pipe spool was not accounted for, and the repair was not given a hydrostatic test.

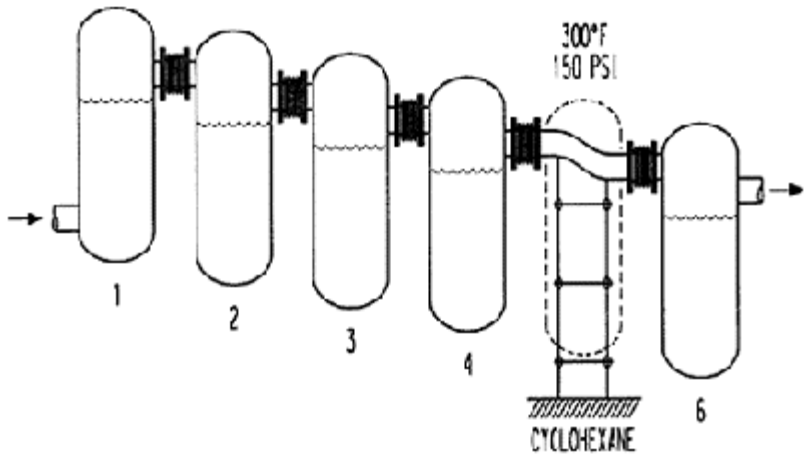


**Figure 7-13** Simplified Vessel and Bellows Arrangement

The repaired system, with the temporary spool, was placed in service for several days, and on June 1, 1974, the bellows squirmed and the pipe ruptured, causing the release of a cloud of cyclohexane. The vapor cloud ignited and exploded, killing 28 and injuring 89. This disaster serves as a vivid reminder that expansion joints are designed and intended to absorb movements in specific direc-tions, within specific limits, and for a specific number of cycles. The engineer should work closely with the supplier in selecting the right type of joint for the service. During the investigation, it became evident that the bellows were never intended to carry shear forces that developed from the bends in the temporary pipe. The modification resulted in a dangerous arrangement. The modification was implemented without following the relevant design standard and without seeking the advice of the expansion joint manufacturer.

The simplest type of expansion joint is a single bellows joint, made of one or several convolutions that can absorb expansion and contraction, Figure 7-14 (a), rotation, Figure 7-14 (b) or shear, Figure 7-14 (c). When using a single bellows expansion joint the piping must be well guided and anchored to resist pressure thrust forces and maintain bellows deflections within limits specified by the bellows manufacturer. There are many other types of expansion joints: two-bellow expansion joints, gimbal expansion joints to absorb rotations, pressure balanced expansion joints, expansion joints with or without tie rods, with or without an internal sleeve, with or without reinforcement, etc. A comprehensive description of the many types of expansion joints is provided in the EJMA standard [EJMA].

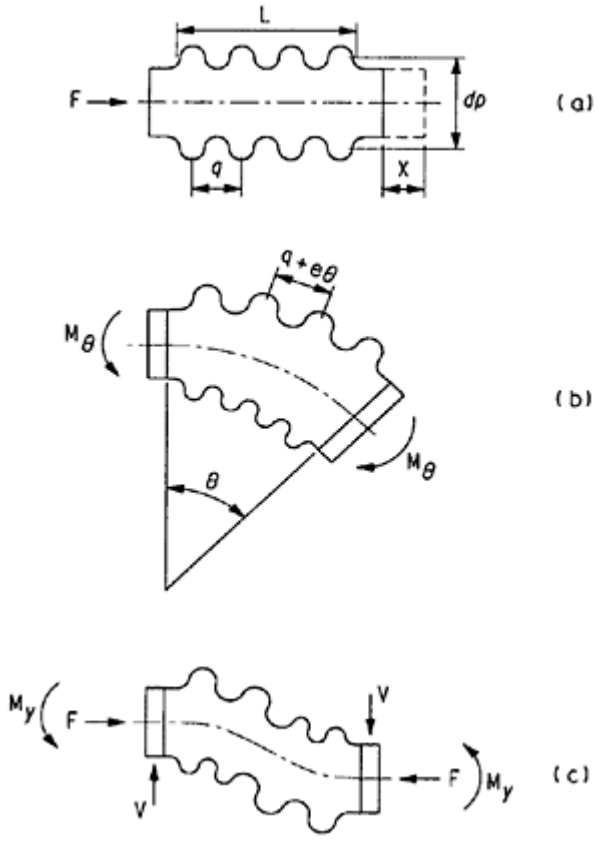


**Figure 7-14** Bellows in (a) Tension, (b) Bending, and (c) Shear

Because different types of expansion joints serve different functions, it is critical to choose the right expansion joint, with the right size for the service. The designer should work with the expansion joint manufacturer to select and size the joint. Typically, the designer will specify pressure, temperature, maximum differential movement of joint ends (translations and rotations), and number of cycles of movement. The manufacturer will help select the right joint for the application. The choice of materials (corrosion resistance, compatibility with process stream) is the designer's responsibility.

## 7.17 REFERENCES

AASHTO LRFD, Guide Specification for Fatigue Design of Steel Bridges, American Association of State Highway and Transportation Officials, Washington, DC.

API RP 2A, Recommended Practice for Planning, Designing and Constructing Fixed Offshore Platforms, American Petroleum Institute, Washington, DC.

API 530, Design of Fire Heater Tubes, American Petroleum Institute, Washington, DC.
API RP 579, Fitness for Service, American Petroleum Institute, Washington, DC.
API RP 1102, Steel Pipelines Crossing railroads and Highways, American Petroleum Institute, Washington, D.C.
ASM, Metals Handbook, ASM International, Metals Park, OH.
ASME II, Boiler and Pressure Vessel Code, Section II, Materials, American Society of Mechanical Engineers, New York.
ASME III, Boiler and Pressure Vessel Code, Section III, Appendix I, American Society of Mechanical Engineers, New York.
ASME VIII, Boiler and Pressure Vessel Code, Section VIII, Division 2, Appendix I, American Society of Mechanical Engineers, New York.
ASME Criteria of the ASME Boiler and Pressure Vessel Code for Design by Analysis in Sections III and VIII Division 2, 1969, American Society of Mechanical Engineers, New York.
ASME B31.3 Process Piping, 1999, American Society of Mechanical Engineers, New York.
ASME NH, ASME Boiler and Pressure Vessel Code, Section III, Subsection NH Class 1 Components in Elevated Temperature Service, American Society of Mechanical Engineers, New York.
ASTM, Annual Book of Standards, Vol. 4.06, Thermal Insulation; Environmental Acoustics, American Society of Testing Materials.
AWS D1.1, Structural Welding Code, American Welding Society, Miami, FL.
Barsom, J.M., Rolfe, S.T., Fracture and fatigue Control in Structures—Applications of Fracture Mechanics, Prentice Hall, Englewood Cliffs, N.J.
Bannantine, J.A., et. al., Fundamentals of Metal Fatigue Analysis, Prentice Hall, Englewood Cliffs, NJ.
Barsom, J.M., Vecchio, R.S., Fatigue of Welded Structures, Welding Research Council Bulletin 422, New York, 1997.
Battelle Columbus Institute, Damage Tolerant Handbook, Metals and Ceramics Information Center, Battelle, Columbus, OH, 1975.
Baumeister, Avallone, and Baumeister, Mark's Standard handbook for Mechanical Engineers, McGraw Hill, Eighth Edition.
Becht, C., B31.3 Appendix X Rules for Expansion Joints, ASME PVP-Vol. 279, developments in a Progressing Technology, 1994, American Society of Mechanical Engineers, New York.
Becht IV, C, Behavior of Bellows, Welding Research Council Bulletin 466, New York, 2001.
Broyles, R.K., EJMA Design Equations, ASME Pressure Vessel and Piping Conference, paper PVP-Vol. 279, 1994, American Society of Mechanical Engineers, New York.
BS 5500, British Standard Specification for Unfired Fusion Welded Pressure Vessels, British Standard Institute.
BS 7608, Code of Practices for Fatigue Design and Assessment of Steel Structures, British Standard Institute.
Buch, A., Fatigue Strength Calculation methods, Trans. Tech. Publications, Aedermannsdorf, Switzerland, 1988.
Coffin, L.F., 1954, A Study of the Effects of Cyclic Thermal Stresses on a Ductile Metal, Trans. ASME, vol. 76, pp. 931–950.
EJMA, Standards of the Expansion Joint Manufacturers Association, Inc., White Plains, New York, 1980.
Fuchs, H.O., Stephens, R.I., Metal Fatigue in Engineering, John Wiley & Sons.
Grinnell, Piping Design and Engineering, Grinnell Company Inc., Providence, RI.
Harvey, J.F., Theory and Design of Pressure Vessels, Van Nostrand Reinhold, New York.
Higuchi et. al., Fatigue Strength of Socket Welded Pipe Joints, ASME PVP 1996, American Society of Mechanical Engineers, New York.
Langer, B.F., Design values for Thermal Stress in Ductile Materials, Welding Research Supplement, September, 1958.

Kishida, K., et al., High Cycle Fatigue Strength of Butt Welded Joints of Small Diameter Pipe, SMIRT Conference, 1991
Klanins, A., Updike, D.P., Park, I., Evaluation of Fatigue in the Knuckle of a Torispherical Head in the Presence of a Nozzle, PVRC 97–24, 97–25, 1997.
Lees, F.P., Loss Prevention in the Process Industries, Butterworth, Heinemann, Reed Educational and Professional Publishing Ltd., 1996.
Lida et al., Fatigue Strength of Socket Welded Pipe Joints, 20th MPA, 1994
Markl, A.R.C., Fatigue Tests of Welding Elbows and Comparable Double-Miter Bends, Transactions of the ASME, Volume 69, No. 8, 1947.
Markl, A.R.C., Fatigue Tests of Piping Components, Transactions of the ASME, Volume 74, No. 3, 1952.
Markl, A.R.C., Piping Flexibility Analysis, Transactions of the ASME, February, 1955.
Massonnet, C, Resistance des Materiaux, Dunod, Paris.
NRC 88–08, Thermal Stresses in Piping Connected to Reactor Coolant System, NRC Bulletin 88–08, Supplement 3, 1989, Nuclear Regulatory Commission, Washington, DC.
PVRC, Pressure Vessel Research Council, Seminar on Practical Aspects of Fitness for Service Evaluation of Process Equipment, October 1999.
Reemsnyder, H., Development and Application of Fatigue Data for Structural Steel Weldments, ASTM STP 648, 1978.
Ricardella, P.C., et. al. ASME PVP vol. 360, 1998, p. 453, American Society of Mechanical Engineers, New York.
Rodabaugh, E.C., George, H.H., Effect of Internal Pressure on Flexibility and Stress-Intensification Factors of Curved Pipe or Welding Elbows, Transactions of the ASME, Vol. 79, pp. 939–948, American Society of Mechanical Engineers, 1957.
Rodabaugh, E.C., Moore, S.E., Stress Indices and Flexibility Factors for Concentric Reducers, Oak Ridge National Laboratory, report ORNL-TM-3795, 1975, Oak Ridge, TN.
Rodabaugh, E.C., Comparison of ASME-Code Fatigue-Evaluation Methods for Nuclear Class 1 Piping with Class 2 or 3 Piping, NURG/CR-3243, June, 1983, US Nuclear Regulatory Commission, Washington, DC.
Special Metals Corp., Data Sheet Inconel 617, Special Metals Corporation, New Hartford, NY.
Spielvogel, S.W., Piping Stress Calculations Simplified, published by S.W.Spielvogel, Lake Success, NY, 1955.
Suresh, S., Fatigue of Materials, Cambridge University Press, UK.
Tavernelli, J.F., and Coffin, L.F., 1959, A Compilation and Interpretation of Cyclic Strain Fatigue Tests on Metals, Trans. ASM, vol. 51, p. 348.
UK DOE, Offshore Installations: Guidance on Design and Construction, UK Department of Energy, HMSO, London, 1984.
Wais, E.A., Rodabaugh, E.C., Background of Stress Intensification Factors in Piping Design, PVP-Vol. 353, ASME Pressure Vessels and Piping Conference, 1997, American Society of Mechanical Engineers, New York.
Woods, G.E., Rodabaugh, E.C., WFI/PVRC Moment Fatigue Tests on 4×3 ANSI B16.9 Tees, Welding Research Council Bulletin 346, New York, 1989.
WRC 376, Metal Fatigue in Operating Nuclear Power Plants, Welding Research Council Bulletin 376, Pressure Vessel Research Council, New York, 1992.

# 8
# Pipe Vibration

## 8.1 ROOT CAUSE

Trying to pinpoint the root cause of pipe vibration, let alone eliminate it, can be a vexing problem. A structured, systematic approach to the problem is required. We start by noting, as illustrated in Figure 8-1, that there are two possible causes for pipe vibration in service: mechanical or hydraulic. A mechanical induced vibration, Figure 8-1 block (2), is due to the mechanical vibration of a piece of equipment such as a pump or compressor which, in turn, causes the pipe to vibrate. The equipment vibration may be transmitted directly to the pipe through its nozzle attachment, Figure 8-1 block (3), or the equipment vibration may cause a structure or floor to vibrate, and the vibrating structure would then transmit the vibration to the pipe through the pipe supports, Figure 8-1 block (4). The pipe may just follow the equipment vibration, Figure 8-1 block (6) or, if the vibration frequency of the mechanical source $f_{MS}$ is close to a mechanical natural frequency of the pipe $f_{MP}$, the pipe may amplify the vibration, Figure 8-1 blocks (5) and (7). A hydraulic induced vibration, Figure 8-1 block (8), is due to continuous pressure pulses that cause the pipe to vibrate. The pressure pulses could be clearly periodical, Figure 8-1 block (9), or more random and turbulent, Figure 8-1 block (10). If the frequency of the pressure pulses, the hydraulic source frequency $f_{HS}$, is close to the acoustic frequency of the pipe cavity $f_{AP}$, Figure 8-1 block (11), the pipe will resonate and amplify the vibration, Figure 8-1 block (13). In this chapter, we will examine the logic behind Figure 8-1 in more detail to understand what causes pipe vibration in service, and how best to solve it.

One more basic point is in order: the vibratory motion of stiff pipe spans (spans with high natural frequencies, in the order of 50 Hz or more) is small, even when in resonance. In piping systems, large motion and therefore the real danger of fatigue failure due to vibration induced bending takes place most often at relatively low frequency, below about 50 Hz.

Finally, experience indicates that to minimize or eliminate piping vibration, it is best to rely on experience during the design stage, follow good construction practices during erection, and include pipe vibration monitoring as part of the preoperational or system startup test. Quantitative analysis at the design stage is complex and, with few exceptions (such as gas compressors [API 618]), questionable.

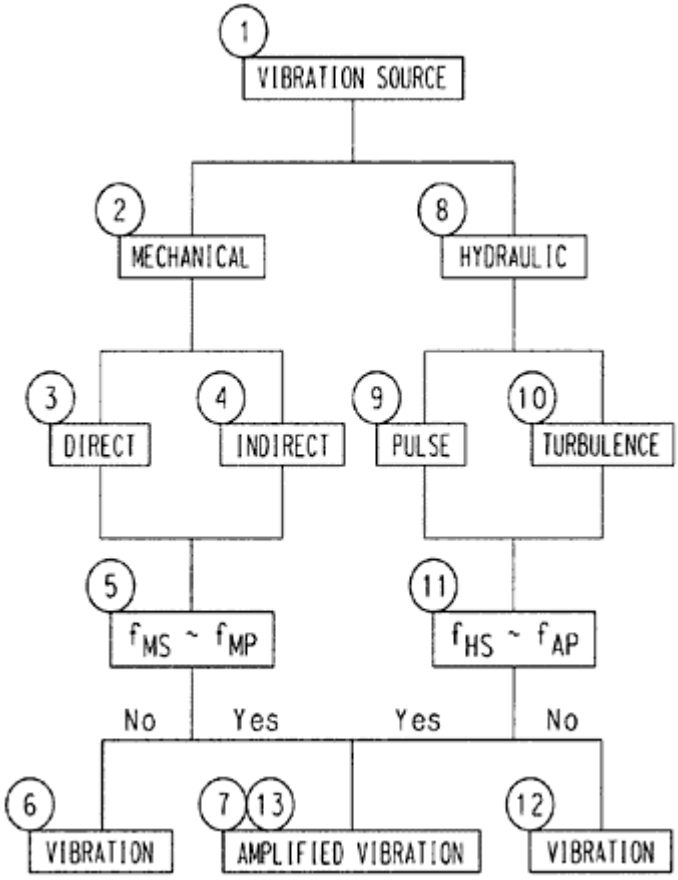


**Figure 8-1** Systematic Approach to Pipe Vibration

## 8.2 MECHANICALLY INDUCED VIBRATION

If the pipe is connected to vibrating equipment, then the pipe will follow the vibration of the equipment. This type of pipe vibration caused by the vibration of attached equipment can be classified as “mechanically induced”. Note that the pipe vibration can be due to two possible sources: it can be directly transmitted to the pipe at the equipment nozzle, or it can be transmitted to the pipe through supports attached to the equipment skid or a flexible floor. The amplitude of the pipe vibration can simply be that of the equipment or, if the equipment vibration frequency is close to a pipe natural frequency, the pipe can amplify the pump or compressor mechanical vibration. In either case, the pipe vibration will tend to die out at a certain distance due to friction and damping, and can be

eliminated by decoupling the piping from the equipment, for example by using flexible hose or bellows at the equipment nozzle.

Well-constructed, well-installed and well-balanced equipment such as pumps, fans or compressors do not vibrate significantly. Equipment vibration is symptomatic of a shortcoming in installation or maintenance, and the frequency of vibration can pinpoint to the cause of vibration, as indicated in Table 8-1. For example, if the rotating equipment is out of balance, it vibrates at one time the shaft rotation frequency. If the rotating equipment is not well anchored to the floor, it is said to have a loose foot and will also vibrate at one time the shaft rotation frequency.

**Table 8-1** Cause, Dominant Frequency and Direction of Vibration

| Cause | Frequency | Direction |
|---|---|---|
| Equipment out of balance | 1×RPM | Radial |
| Shaft axial misalignment | 2×RPM | Radial |
| Shaft angular misalignment | 1×and 2×RPM | Radial and axial |
| Loose foot | 1×RPM | Radial |
| Cracked support frame | 2×RPM | Radial |
| Bearing clearance | Multiples ½RPM | Radial |
| Misaligned belt | 1×RPM | Axial |

In the case of a horizontal centrifugal pump, the axial vibration direction refers to a movement in the direction of the shaft, Figure 8-2. The radial direction refers to a vibratory movement in any direction perpendicular to the axial direction. As the pump casing vibrates, it will transmit this vibration to the inlet and outlet piping through the suction nozzle or the discharge nozzle. In the next section, we will investigate the fundamental equations of motion due to free and forced vibration, but it is important at this stage to gain an intuitive understanding of the factors that control vibration severity, referring to Figure 8-2. If the pump discharge nozzle vibrates at a frequency $f_{MS}$ (mechanical source frequency), the discharge pipe will vibrate, from the discharge nozzle to the first support on the horizontal leg, and possibly beyond. If the mechanical natural frequency of this pipe span $f_{MP}$ is close to $f_{MS}$ the pipe segment is in resonance with the source of vibration and will amplify the vibration. But there are three important practical points to keep in mind:

(1) The more flexible the pipe span, the larger the displacement amplitude of its resonant vibration.

(2) If the pipe is stiff, its fundamental natural frequency is high, and the pipe bending vibration is generally negligible, even if the pipe is in resonance. This is particularly true at fundamental frequencies larger than 50 Hz.

(3) A flexible joint at the pump nozzle, such as a bellow or a braided hose, will decouple the pump nozzle vibration from the pipe.

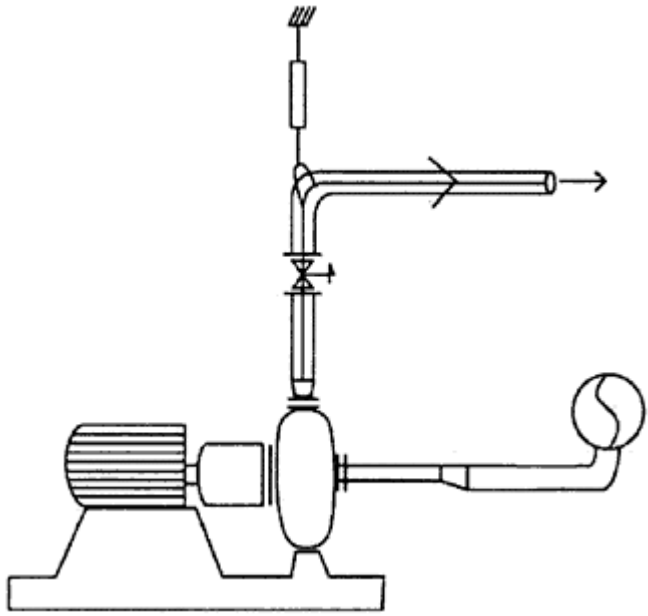

**Figure 8**-2 Pump and Pipe Arrangement

### 8.3 VIBRATION ANALYSIS

The equation of motion of a single degree-of-freedom oscillator (a mass m and restoring spring of stiffness k) with viscous damping c (dashpot), driven by an imposed sinusoidal force of amplitude $P_o$ and circular frequency ω is [Pilckey, Clough, Harris, Vierck]

$$m\ddot{x} + c\dot{x} + kx = P_o \sin \omega t$$

x=displacement of mass relative to driver, in
m=mass, lb-sec$^2$/in
c=damping
k=stiffness, lb/in
$P_o$=amplitude of applied force, lb
ω=circular frequency of applied load, rad/sec
The solution to the differential equation is the movement of the mass m, given by

$$x = e^{-\frac{ct}{2m}}(C_1 \sin \omega_d t + C_2 \cos \omega_d t) + \frac{\frac{P_o}{k}\sin(\omega t - \theta)}{\sqrt{\left(1 - \frac{\omega^2}{\omega_n{}^2}\right)^2 + \left(\frac{2\zeta\omega}{\omega_n}\right)^2}}$$

The first term is a transient motion at the damped natural frequency of the single degree of freedom $\omega_d$. It is a negative exponent of time that will quickly die out. The second term is a continuous sinusoidal motion, as shown in Figure 8-3 (the Figure corresponds to the case $\theta=0$ and hence $x=C_2$ at the initial time $t=0$). It is the forced motion driven by the applied sinusoidal excitation $P_o\sin\omega t$. The circular frequency of the forced motion is $\omega$, equal to the circular frequency of the driving force $\omega$. The damped circular frequency of the single degree of freedom, is

$$\omega_d = \sqrt{\frac{k}{m} - \frac{c^2}{4m^2}} = \omega_n\sqrt{1-\zeta^2}$$

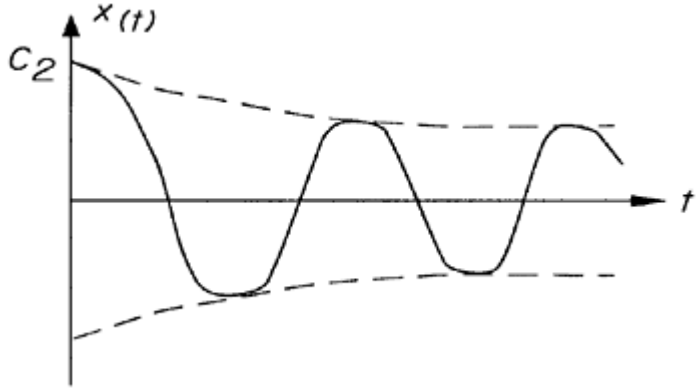


**Figure 8-3** Forced Vibration of Single Degree of Freedom

The undamped circular frequency of the single degree of freedom has the classical form obtained with $c=0$

$$\omega_n = \sqrt{\frac{k}{m}} = 2\pi f_n$$

where $f_n$ is the natural frequency of the $n^{th}$ mode of vibration. The fraction of critical damping is given by

$$\zeta = \frac{c}{c_c}$$

The critical damping is

$$c_c = 2\sqrt{km} = 2m\omega_n$$

The phase angle is

$$\theta = \tan^{-1} \frac{\frac{2\zeta\omega}{\omega_n}}{1 - \frac{\omega^2}{\omega_n^2}}$$

The maximum amplitude of vibration takes place when the frequency of the forcing function ($\omega$) is equal to the natural frequency of the single degree of freedom ($\omega_n$), in this case there is resonance, and the amplitude of this resonant vibration depends on the component's damping. This amplification can be measured by the dynamic magnification factor (DMF), the ratio of the maximum dynamic amplitude ($X_{max}$) to the motion amplitude of the component if it was statically loaded ($P_o/k$). The typical shape of the DMF as a function the frequency ratio $\omega/\omega_n$ for an elastic single degree of freedom is illustrated in Figure 8-4. The magnitude of the peak is a function of the system's damping; the higher the damping, the lower the peak. Damping in a piping system results from the combination of material damping (plasticity) and structural damping (friction, rattle, etc.). System damping is therefore a function of vibration amplitude; the larger the vibration, the more material and structural damping. Vibrations of small amplitude are mostly elastic and involve little friction. In these cases, damping will be very small in the order of 0.5%. Large vibrations that involve plastic deformation and sliding of pipes on supports can have large damping, in the order of 20% or more.

To understand the practical meaning of these equations, consider a single span of pipe on two simple supports. Imagine that the span is pulled upward at mid-span and then let go. The pipe span will start to vibrate around the horizontal, in a bow shape (mode shape) with the largest displacement at mid-span. This is the pipe span's first mode of vibration. For the more common case of a pipe on multiple supports, as shown in Figure 8-5, the natural frequency of the simply supported multi-span pipe is given by [Blevins]

$$f_i = \frac{\lambda_i^2}{2\pi L^2}\sqrt{\frac{EI}{m}}$$

L=length of pipe span, in
E=Young's modulus of pipe material, psi
I=moment of inertia of pipe cross section, $in^4$
m=mass per unit length of pipe and contents, lbm/in
$\lambda_i$=factor given in Table 8-2

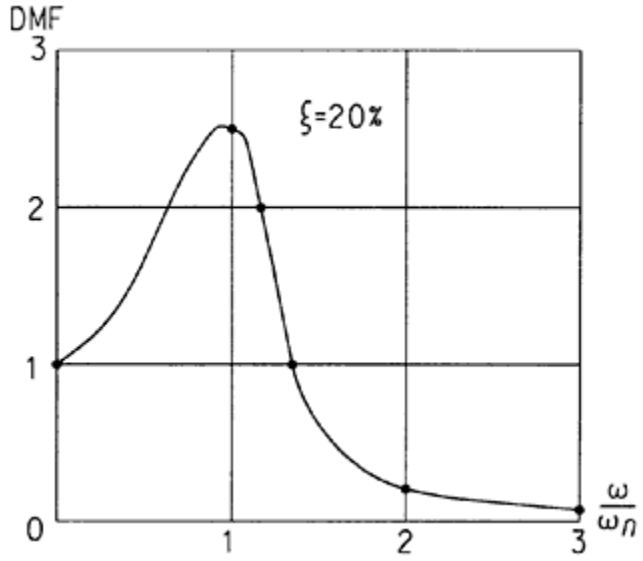


**Figure 8-4** Resonant Response

For example, a 15 ft long span (L=180”), of 4” sch.40 (I=7.2 $in^4$, Appendix A), water filled (m=(1.36 lb/in)/386 $in/sec^2$, Appendix A), carbon steel pipe (E=29 $10^6$ psi), has a first mode natural frequency ($\lambda_1$=3.142), f=11.8 Hz. Therefore, if initially disturbed and then left to vibrate, the single span of pipe will vibrate at a frequency of 11.8 cycles per second. The fundamental (first mode) natural frequency for several other simple configurations illustrated in Figure 8-6 is given in Table 8-3.

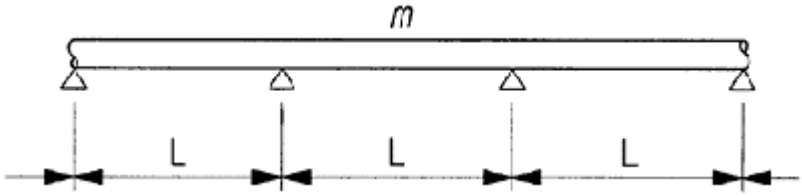


**Figure 8-5** Multispan Beam

**Table 8-2** Frequency Factor $\lambda_i$ [Blevins]

| Spans | Model | Mode 2 | Mode 3 |
|---|---|---|---|
| 1 | 3.142 | 6.283 | 9.425 |
| 2 | 3.142 | 3.927 | 6.283 |
| 3 | 3.142 | 3.557 | 4.297 |
| 4 | 3.142 | 3.393 | 3.928 |

**Table 8-3** Natural Circular Frequency $\omega=2\pi f$ [Blevins, Pilkey]

| | |
|---|---|
| (a) Mass and spring | $\sqrt{\frac{k}{M}}$ |
| (b) Cantilevered beam, of total mass m and concentrated end mass M | $\sqrt{\frac{3EI}{L^3(M+0.23m)}}$ |
| (c) Simply supported beam, of total mass m and concentrated center mass M | $\sqrt{\frac{48EI}{L^3(M+0.5m)}}$ |
| (d) Fixed-fixed beam, with distributed mass m and concentrated center mass M | $8\sqrt{\frac{EI}{L^3(M+0.375m)}}$ |
| (e) Simply supported beam, massless, with concentrated mass M at distance a and b from the two ends (L=a+b) | $\frac{1}{ab}\sqrt{\frac{3EIL}{M}}$ |
| (f) Fixed-fixed beam, massless, with concentrated mass M at distance a and b from the two ends (L=a+b) | $\frac{1}{ab}\sqrt{\frac{3EIL^3}{Mab}}$ |

k=stiffness, lb/in
m=total mass of beam, lbm
E=Young's modulus, psi
I=moment of inertia of cross section, $in^4$
M=concentrated mass, lbm
L=beam length, in
a,b=distance from ends, in

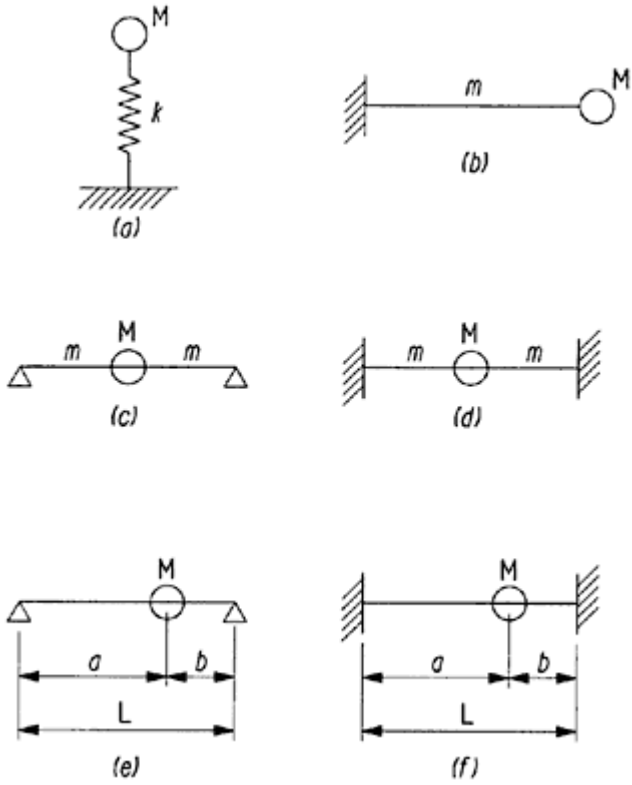


**Figure 8-6** Natural Frequency Models

### 8.4 HYDRAULIC INDUCED VIBRATION

Hydraulic induced vibration starts with a continuous flow disturbance that creates a periodic pressure pulse P(t). At changes in direction (elbow, tee, bend) or changes in flow cross section (valve, orifice, reducer) this pressure pulse P(t) causes pulsating forces F(t)=A P(t) on the pipe and cause it to vibrate. In this section we will review the causes of hydraulic induced vibration in piping and pipelines.

#### 8.4.1 Vane and Piston Motion

Pumps, compressors and fans deliver flow at an average pressure P, with small sinusoidal pressure fluctuations dP(t) around the mean pressure. These pressure pulses reflect the packets of fluid delivered downstream every time a vane passes in front of the outlet nozzle or every time a piston completes its stroke. The dominant frequency of these pressure fluctuations dP(t) is summarized in Table 8-4, where RPM is the pump speed in revolutions per minute, and CPM is a piston's cycles per minute.

**Table 8-4** Frequency of Pressure Fluctuations

| Cause of Vibration | Dominant Frequency | Direction of Vibration |
|---|---|---|
| Vane pass | No. of vanes×RPM | Radial |
| Piston motion | No. of pistons×CPM | Random |

The continuous discharge of fluid pockets causes cyclic pressures P+dP that travel down the pipe, and every time they reach a change of direction or a change in cross section, they cause an unbalanced force dP×A, where dP is the amplitude of pressure fluctuation above the mean pressure P, and A is the cross sectional area of the obstruction. In the majority of cases with centrifugal pumps, these forces are small and do not result in any visible motion of the pipe. Visible vibrations are more common in positive displacement pumps and gas compressors. On every suction stroke of the piston of a positive displacement pump, a pressure drop −dP is created in the suction pipe. On every discharge stroke, a pressure increase +dP is created in the discharge pipe. These pressure fluctuations cause a pulsating flow in the pipe at the same frequency as the stroke frequency. For a single piston with a sinusoidal motion, the peak flow rate is $\pi$ times (3.14 times) the average flow rate [Warwick], which is a significant surge at each stroke.

The vane pass frequency of a centrifugal pump (number of vanes multiplied by the pump's RPM) is the hydraulic forcing frequency of the source fHS in Figure 8-1. If the pipe is sufficiently flexible, the hydraulic pressure pulses (Figure 8-1, block 9) will cause the pipe to vibrate (Figure 8-1, block 12).

For example, during the start-up of the pump illustrated in Figure 8-2, the pump ramps up from 0 RPM to its full running speed. In this startup process, the pump rotation sweeps through the natural frequency of the horizontal discharge pipe and, for a short moment, this section of pipe will be in mechanical resonance with the pressure pulses due to vane passing frequency. If the first support passed the variable spring is sufficiently flexible, the horizontal pipe will sway in a transient vibration, until the pump RPM's, which are steadily increasing, become sufficiently larger than the pipe natural frequency to be out of resonance.

### 8.4.2 Turbulence Induced Vibration

When a steady state flow, zone A in Figure 8-7, reaches an obstruction, vortices will be shed by the obstruction, zone B in Figure 8-7 [Morishita, Dozaki]. The vortices cause pressure fluctuations, at a frequency f given by

$f_{HS}$=n S v/D

$f_{HS}$=hydraulic source frequency of pressure pulses due to vortex shedding, Hz
n=1 for lift (perpendicular to flow) and 2 for drag (flow direction)
S=Strouhal number
Re=Reynolds number
S=0.2 for Re=$10^3$ to $10^5$,
S=0.2 to 0.5 for Re=$10^5$ to 2 $10^6$
S=0.2 to 0.3 for Re=2 $10^6$ to $10^7$

v=flow velocity, in/sec
D=outer diameter of obstruction, in

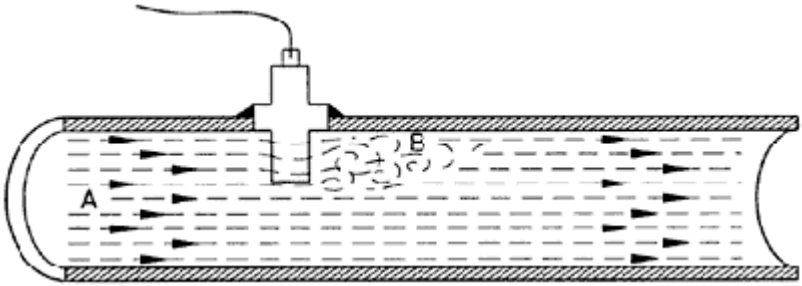


**Figure 8-7** Vortex Shedding around Obstruction

For example, 1075 psi steam at 700°F, flowing at v=70 ft/sec has a Reynolds number Re=v (ID)/v=$3\times10^6$, and therefore its Strouhal number is 0.2 to 0.3. If a 1.5" diameter instrument is inserted into the flow, then the vortex shedding frequency in the lift direction (perpendicular to flow) is $f_{HS}=1\times0.25\times(70\times 12)/1.5=140$ Hz, and 280 Hz in the drag direction (parallel to flow). If the mechanical natural frequency of the instrument is not at the vortex shedding frequency $f_{HS}$ (no resonance) then the amplitude of the lift force is given by

$$F_L=(\rho v^2/2)D\,C_L$$

$F_L$=amplitude of hydraulic lift force applied to protruding instrument, lb
ρ=fluid density, lb/in$^3$
v=flow velocity, in/sec
D=outer diameter of protrusion, in
$C_L$=lift coefficient, approximately 0.5

If the instrument's mechanical natural frequency is close to the hydraulic vortex shedding frequencies 140 Hz or 280 Hz, then the instrument vibration will lock-in at resonance, and the applied force and displacements will be larger. At lock-in frequency, the vibration amplitude is

$$a = D\frac{0.4}{\sqrt{0.06+(2\pi S^2 C)^2}}$$

a=vibration amplitude of obstruction, in
D=diameter of obstruction, in
S=Strouhal number
C=damping factor

$$C=\frac{4\pi\zeta m}{\rho D^2}$$

ζ=damping of instrument relative to critical damping
m=linear mass density of instrument and displaced fluid, lbm/in
ρ=fluid density, lbm/in$^3$
D=outer diameter of instrument, in

Because the vibration amplitudes are small, damping tends to be quite small. For example, with ζ=0.5%=0.005.

The corresponding vibratory bending stress is

i M/Z=4EI a/(L$^2$Z)

i=stress intensification factor
M=bending moment at base of instrument, in-lb
Z=instrument section modulus, in$^3$
E=instrument modulus of elasticity, psi
I=instrument moment of inertia, in$^4$
L=instrument length, in
a=vibration amplitude, in

### 8.4.3 Cavitation and Air Pockets

Another source of turbulent flow that could cause vibration is flashing and cavitation [Tullis, Eitschberger]. Flashing is the formation of vapor bubbles (cavi-ties) in a liquid, as a result of local pressure drops below vapor pressure, particularly where flow is forced through an orifice or a partially closed valve. Cavitation is the subsequent collapse of the vapor bubbles. These phenomena are often accompanied by local popping sounds, as if there was gravel in the pipe, and result in erosion of the pipe wall when the bubbles collapse repeatedly close to the pipe wall. Cavitation is particularly troublesome in centrifugal pumps where it can cause erosion of the pump casing, pressure imbalance and vibration. To avoid cavitation, pump suction must be designed to provide a net positive suction head (NPSH), for example by providing a vertical drop of the pump inlet piping to increase the static head. To further minimize the pressure drop at the pump inlet, the flow rate Q should be minimized and the inlet pipe diameter should be maximized [Crane]. Manufacturers will normally specify inlet pipe 1 or 2 sizes larger than pump nozzle. Eccentric reducers are used at the pump nozzle to avoid the entrapment of air pockets at the top of the pipe.

The accumulation of air, or more generally of a gas entrained by a liquid, upstream of a pump may take some time. However, at a certain point, the air or gas pocket becomes sufficiently large and enters the pump, blocking the liquid flow in the pump. Fluid may continue to pass through the air or gas filled pump, but this will be through a reduced liquid flow cross section and therefore at higher velocity and lower pressure. In the best of cases, the liquid will entrain the gas out of the pump. In either case, the liquid will not be uniformly distributed in the pump, which will lead to imbalance and vibration. Conditions to avoid in pumppiping systems [HI, HI 9.8, Carucci]:

(a) Inlet pipe sloping down to the pump.
(b) Reducer with downward slope above horizontal inlet pipe axis.
(c) Headers with larger diameter than branch pipe to pump nozzle.
(d) Air pockets in housing of vertical stem valves.

(e) Excessive throttle leading to cavitation on the suction side.
(f) In-leakage of air in inlet piping system.
(g) Inlet strainer flow area less than 3 times the area of the suction pipe.
(h) Gas in liquid, coming out of solution if the pressure drops significantly.
(i) Inlet velocities in excess of 8 ft/sec.
(j) Partially open inlet valves.

The inlet flow to a pump should be as even as possible through the pipe cross section to avoid an unbalanced flow in the pump casing. Sources of uneven flow must be avoided in horizontal inlet pipes. There should be a sufficient length of straight pipe (5D to 10D) at the pump inlet to reestablish a uniform flow passed the last bend or elbow, before entering the pump. With vertical inlet pipe, elbows should be long radius. Flow downstream of an orifice or throttling valve is uneven, and there should be a sufficient length of straight pipe passed the reduced cross section (5D to 10D) to reestablish a uniform flow.

### 8.4.4 Acoustic Resonance

When pressure pulses travel from a point of origin (such as a pump or compressor) down the pipe, and reflect at discontinuities (closed valve, orifice, etc.) or large volumes (tank, header, etc.), the superposition of incident and reflecting waves can form standing waves in the piping system. This condition occurs if the frequency of the pressure pulse is equal to the acoustic frequency of the pipe. In an open-open pipe section (for example a branch pipe connecting two headers) or close-close section, the acoustic frequencies are [Blevins]

$f_{AP}$= n a/(2L)

f=acoustic frequency of cavity, 1/sec
n=integer 1,2,3,…
a=acoustic velocity in fluid, ft/sec
L=pipe length, ft

In an open-close section, such as a relief valve stub-in (open at the branch to header junction and closed at the relief valve) or a closed bypass branch, the acoustic frequencies are

$f_{AP}$=(2n−1)a/(4L)

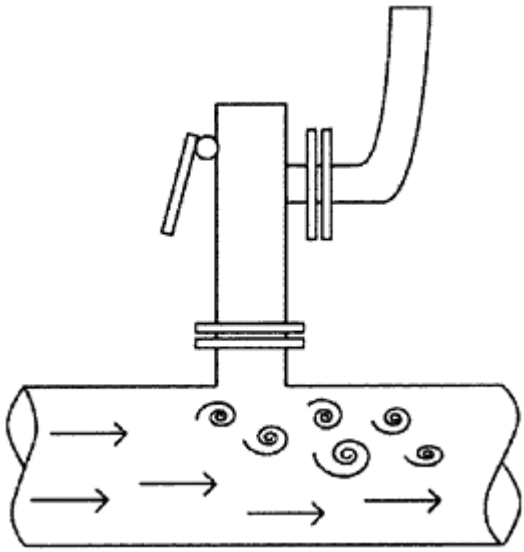

**Figure 8-8** Vortex Shedding at Branch Inlet

Sharp branch openings provide a classic example of acoustic resonance. Referring to Figure 8-8, a sharp branch line (90° pipe-on-pipe connection) causes the formation of flow vortices at the branch entrance edge. The dominant frequency of the vortices $f_{HS}$, may be close (within 20%) to the acoustic frequency of the openclosed branch entrance to the relief valve

$f_{HS} \sim f_{AP}$

$n\ S\ v/D \sim (2n-1)a/(4L)$

In this case, the pressure time history P(t) in the header-to-valve cavity will exhibit a beat with peaks larger than the steady sate pressure in the header, sufficient to intermittently lift the relief valve, usually set at a pressure about 10% above the header operating pressure, and to cause the valve internals to wear prematurely. Such a problem can be resolved by changing the length L of the branch pipe so that $f_{HS}$ is no longer close to $f_{AP}$. A better solution is to replace the source of vortices, the sharp pipe-on-pipe branch connection, by a smooth, well contoured integrally reinforced branch connection, eliminating the sharp branch entrance edge that caused the vortices in the first place [Coffman, Baldwin].

In gas pipelines, the quantitative analysis of acoustically induced vibration is part of the design of a gas compressor stations. The pipe acoustic frequencies are calculated and compared to the compressor piston frequencies, to assure that they are not close, within 20%, otherwise the downstream pipe could amplify the compressor's pressure pulses. The peak-to-peak pressure oscillations, measured at the compressor cylinder flange, should not exceed $P_{cf}$ [API 618] where

$P_{cf}=100\ (dP/P)=\min\ \{3R;\ 7\%\}$

$P_{cf}$=maximum permitted peak-to-peak pressure oscillations, %
dP=maximum permitted peak-to-peak pressure oscillations, psi
P=average line pressure, psi
R=compressor stage pressure ratio
For line pressures between 50 and 3000 psi, the limit is

$$100(dP/P) \leq 300/(P\ d\ f)^{0.5}$$

d=pipe inside diameter, in
f=pressure pulse frequency, Hz

To prevent fatigue damage, the pulse-induced vibration should be below the endurance limit of the pipe material (Chapter 7). To achieve the double objective of limiting dP/P and the cyclic stress amplitude, it may be necessary to install compressor suction and discharge surge cylinders, or acoustic filters. The surge cylinders are typically designed as ASME B&PV pressure vessels, with provisions to drain condensate and other trapped liquids. The minimum volume of a suction surge tank should be [API 618]

$$V_S = 7\ (PD)\ (KT_S/M)^{0.25}$$

$V_S$=minimum suction surge volume, at least 1 $ft^3$, $ft^3$
PD=total net displaced volume per revolution of all cylinders, ft3/revolution
K=isentropic compression exponent at average operating gas pressure and temperature
$T_S$=absolute suction temperature, ºRankine
M=molecular weight of compressed gas
The minimum volume of a discharge surge tank should be [API 618]

$$V_D = 1.6(V_S/R^{0.25})$$

$V_D$=minimum discharge surge volume, at least 1 $ft^3$, and not larger than $V_S$, $ft^3$
R=stage pressure ratio at cylinder flanges

A large gas volume also acts as a reflector or terminal point beyond which the pressure pulse will not propagate. The volume required to create a terminal point is [Sparks]

$$V = V_C\ (95\ e^{-0.085\ m} + 1.5\ m)$$

V=terminal point volume, $in^3$
$V_C$=net displaced volume of a cylinder per revolution, $in^3$
m=molecular weight of gas

### 8.4.5 Breathing Mode

Pressure pulses can cause two types of vibration in a piping system: a beam bending vibration due to the cyclic force dP×A, and a breathing mode of vibration from the radial vibration of the pipe wall. Breathing mode vibration typically occurs at high frequency and is particularly evident in large diameter thin wall pipe (D/t ratio of diameter to wall thickness larger than 100).

The first natural frequency of a cylinder's breathing mode is an extension mode in which the pipe extends uniformly radially around its circumference. The natural frequency of the extension mode for an infinitely long pipe is [Blevins]

$$f = \frac{1}{2\pi R}\sqrt{\frac{E}{\rho(1-\nu^2)}}$$

R=pipe radius, in

ρ=density of pipe material, lb/in$^3$

ν=Poisson modulus of pipe material

E=Young's modulus of pipe material, psi

The first two flexural breathing modes of vibration of a cylinder are shown in Figure 8-9: the two lobe elliptical shape and the three-lobe shape. The first flexural breathing mode shown in Figure 8-9(a) has a natural frequency [Blevins]

$$f = 1.15\frac{t}{R}\frac{1}{2\pi R}\sqrt{\frac{E}{\rho(1-\nu^2)}}$$

t=pipe wall thickness, in

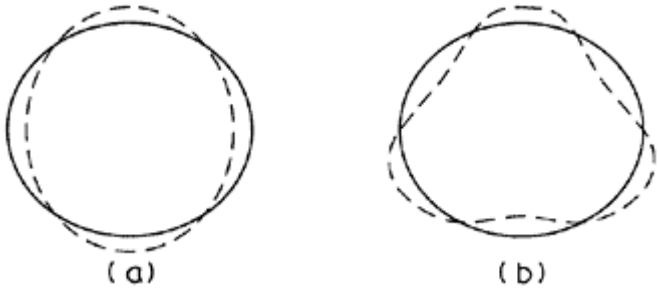


**Figure 8-9** Flexural Breathing Modes of Pipe Wall

### 8.4.6 Valve Noise

In addition to mechanical vibration, pressure changes can cause sound (vibration in the range of 20 Hz to 20 kHz). The sound pressure level is defined as

$db=20 \log_{10} (P_{measured}/0.0002 \text{ microbars})$

db=sound pressure level, decibels

$P_{measuerd}$=measured pressure amplitude, microbars

The human pain threshold is approximately 145 db (pressure pulse amplitudes of 0.004 atmospheres). In the U.S., noise levels are regulated by the Occupational Safety and Health Administration (OSHA). In addition to occupational safety concerns, fatigue cracks in steel valves have been reported at 130 db [Bau-man]. Often times, hydraulically induced piping vibration is accompanied by valve noise, which is a symptom of the pressure fluctuations at the origin of the hydraulic excitation.

Valve noise can be due to one of three causes [Lyons]: (a) pressure fluctuations in a valve causing internal parts to rattle; (b) cavitation due to pressure drop below the vapor pressure; (c) excessively turbulent flow, particularly at high flow velocities and large pressure drops.

The high frequency noise levels in a valve, caused by high frequency pressure pulses, tend to excite the high frequency breathing modes of pipe walls, resulting in failures in straight pipe and branch connections [Carucci]. The elimination of noise in valves can be achieved by replacement of valve internals and trims, or by use of tortuous path trims (section 8.7).

## 8.5 MEASURING VIBRATION

### 8.5.1 Measuring Displacement

When the displacement amplitude is sufficiently large to be visible, it tends to occur at low frequency, 1800 cycles/min (30 Hz) or less. The displacement amplitude can then be measured by several methods [Dove, Harris]. The simplest and least accurate is to visually estimate the displacement, for example by observing a white cardboard marked with ½”×½” squares placed behind the pipe.

Displacement amplitude can also be measured by a dial indicator, which is a mechanical device that records the maximum movement of the pipe relative to a fixed point such as wall or steel structure.

Lanyards can be used to measure vibration amplitude. Lanyards are electromechanical devices with a resistance that measures an electric signal proportional to displacement. They are particularly well suited for low frequencies, below 30 Hz.

A linear variable differential transducer (LVDT) is an electromechanical device that produces an electric signal proportional to the motion of a magnetic core inside coils. It measures relative displacement and therefore one end must be placed at a fixed point.

A proximity probe, or “noncontact transducer”, is an eddy current device that generates an electric current proportional to the gap between the probe and the monitored object. It measures vibration displacements without being in contact with the pipe or equipment, but is limited to small vibration amplitudes such as rotating equipment shafts, rather than pipe bending vibration. They are generally sensitive up to 0.1” in vibration amplitude, with a range of 0 to 10 kHz. The eddy current probes have to be calibrated against the same material as the component being monitored.

Laser systems are used to scan a whole surface, remotely, and generate a plot of vibration amplitudes.

### 8.5.2 Measuring Velocity

Vibration velocity is useful over a range of excitation frequencies, from 1800 cycles/min (30 Hz) up to 60,000 cycles/min (1000 Hz).

A velocity transducer is an electromechanical device that can record vibration velocity, in the range of $10^{-4}$ in/sec (ips) to 100 ips. Velocity transducers are available as

hand held, with a short needle-like probe placed in contact against the vibrating component.

### 8.5.3 Measuring Acceleration

A piezoelectric accelerometer can be used to measure vibration accelerations over a range of frequencies, from 1 Hz up to over 5000 Hz, and in the range of 10 kHz if measuring impacts. They are usually small, can be hand held, or magnetically or mechanically attached to the equipment. Accelerometers are also used to monitor impact shocks and are therefore sensitive to noise resulting from poor contact. The instrument can integrate the measured acceleration signal amplitude a knowing its frequency f, to obtain velocities $v=a/(2\pi f)$ or displacements $d=a/(2\pi f)^2$.

### 8.5.4 Strain Gages

Strain gages can be placed on the vibrating pipe and record directly strain (stress), with proper compensation for surface temperature and moisture.

### 8.5.5 Signal Conditioners and Analyzers

Until the 1990's, vibration monitoring had relied on analog equipment (oscilloscope, strip chart and magnetic tape) for data display and storage. These techniques required continuous recording and the generation of large amounts of raw data. It was also difficult to segregate between background noise and meaningful vibration noise. Data analysis was done manually, often from maximum amplitudes recorded on strip charts.

Today, the PC disk has replaced the magnetic tape, and analog data is automatically recorded by digital means and processed on personal computers. Com-puters also sort and classify vibration data, presented in the form of response spectra (amplitude vs. frequency), power spectral density, maximum amplitude, etc., allowing the analyst to focus on the more interesting task of interpretation and root cause analysis. If necessary, filters and automatic vibration switches can be placed to only record "significant" data and eliminate background noise and very narrow peaks.

## 8.6 ASSESSING VIBRATION SEVERITY

### 8.6.1 Severity Charts

The simplest way to evaluate the significance of vibration is to measure a maximum displacement, velocity or acceleration of vibration and then to enter this maximum value and the vibration frequency in a severity chart. The chart will then classify the observed vibration as smooth, rough, very rough, requiring immediate attention, etc.

The International Standard for Rotating Machinery (ISO 2372, ISO 3945) defines vibration severity as the highest value of the broad band root-mean-square value of the velocity amplitude between 10 to 1000 Hz.

Several national and international standards provide criteria or charts for judging vibration severity in machinery. They include:

| | |
|---|---|
| API 610 | Pumps |
| API 612 | Steam Turbines |
| API 613 | Gear Units |
| API 617 | Centrifugal Compressors |
| API 619 | Positive Displacement Compressors |
| API 541 | Motors |
| Hydraulic Institute | Horizontal Pumps |
| Compressed Air and Gas Institute | Centrifugal Compressors |
| ISO 2954 | Rotating or Reciprocating Machinery |

Examples of vibration setpoints for machinery are provided in Table 8-5. Above these values, machinery vibration is worth recording, analyzing and trending.

The simplicity of the chart approach explains its broad application, but it is only valid if a piping chart is used, not a machinery chart. Indeed a warning is in order: do not apply equipment vibration criteria to piping systems or pipelines. When it comes to vibration, pipes tend to be much more forgiving than machinery, and using a machinery chart to evaluate the severity of pipe vibration will cause many unnecessary rejects of acceptable pipe vibration levels.

**Table 8-5** Vibration Setpoints for Machinery [PMC]

| Equipment | Velocity (in/sec) |
|---|---|
| Turbines and Machine Tools | 0.05 to 0.2 |
| Centrifugal and Gear Pumps | 0.1 to 0.3 |
| Electric Motors | 0.1 to 0.3 |
| Centrifugal compressors | 0.2 to 0.3 |
| Fans and Blowers | 0.2 to 0.4 |
| Reciprocating pumps and compressors | 0.5 to 0.7 |
| Motor generators | 0.5 to 0.7 |

Note that in many cases start-up of a machine is accompanied by relatively large vibration amplitudes, as the machine ramps up to its running speed. This start-up vibration is often non-damaging because it is of such short duration, but can cause an unnecessary shut-down by tripping an automatic vibration switch. To avoid this, time delays of 1 to 20 seconds can be set on solid-state switches.

Piping vibration severity charts [ASME OM, Wachel 1976, Wachel 1981, Wachel 1982, Sparks] are based on pipe size and the configuration and support condition of pipe

spans. A chart method used to check pipe vibration during the startup of nuclear power plants is provided in Part 3 of ASME Operation and Maintenance Standard [ASME OM]. In reality, with today's PC-based pipe analysis tools, a pipe vibration problem worth attention should be investigated by computer analysis, rather than by charts.

### 8.6.2 Pipe Vibration Analysis

The simplest approach to pipe vibration analysis is the static method. This method can only be applied where the vibration mode shape is known, which is typical for low frequency large amplitude vibration, but is more difficult for low amplitude high frequency vibration. For large amplitude low frequency vibration the deformed vibratory shape of the piping is established based on field measurements. The maximum measured vibration amplitudes and concurrent directions (mode shapes) are imposed at various points of a stress analysis model of the piping system, including stress intensification factors, to statically force the pipe into the deflected shape and amplitude of its worst recorded vibration. The resulting bending stresses are obtained as output. These statically calculated vibration stress amplitudes iM/Z are compared to the endurance limit $S_{el}$ [ASME OM, Stoneking], by writing

$$i\,M/Z < S_{el}/SF$$

i=stress intensification factor of the fitting

M=vibration moment amplitude, in-lb

Z=pipe section modulus, $in^3$

$S_{el}$=endurance limit from the ASME fatigue curve, psi (Table 7-1, at $10^{11}$ cycles)

SF=safety factor

A more complex analysis, but one that yields more information, is the modal analysis of the piping system [Sadaoka]. The mode shapes of the piping system model are obtained by dynamic analysis and the vibration response spectrum (accelerations vs. frequencies) is applied to the model of the piping system to yield dynamic displacements and bending stresses. The same equation as in static analysis $2iM/Z < S_{el}/SF$ is used to analyze the results. More complex yet is a time history analysis in which the excitation input (amplitude vs. time) is applied to a finite element model of the piping system [Moussa]. Due to cost, time, complexity, and excessive sensitivity of results to modeling assumptions, time history techniques are rarely justified in vibration analysis of piping systems.

## 8.7 PREVENTION AND MITIGATION

### 8.7.1 Eliminate the Source

The best solution to a vibration problem is obviously to eliminate its source. To that end, it is necessary to systematically follow the logic chart in Figure 8-1. Mechanical sources of vibration are typically easier to pinpoint (refer to the vibration frequency signatures of Table 8-1) and to eliminate by fixing the rotating equipment [Hawkins]. Most

maintenance departments have personnel expert in equipment vibration analysis and diagnostics.

To achieve good alignment it is also necessary to minimize the loads imparted by the pipe on the equipment nozzle, by following some simple guidelines:

(a) Connect piping once the pump has been bolted to the concrete slab.

(b) Run pipe from the pump, not to the pump.

(c) Do not force the pipe into alignment before welding or bolting to the pump nozzle. The pipe and pump nozzle must be axially and angularly aligned before joining. ASME B31.3 calls for an angular alignment of 1/16" per foot of flange face before torquing a flange (Chapter 17).

(d) Do not support the pipe off the pump nozzle, instead use supports or springs to support the pipe weight, as illustrated in Figure 8-2.

(e) Analyze pipe flexibility for hot lines. Add bellows, braided hose, loops and offsets as necessary to reduce expansion loads on equipment nozzle.

(f) Obtain limit on nozzle loads from equipment suppliers.

(g) Pump shaft alignment should not change more than 2 mils (0.002") when connecting the pipe to the pump. In all cases, refer to the vendor installation instructions.

(h) Do not support pipe from pump casing.

### 8.7.2 Good Layout and Supports

A very flexible piping system, such as a spring-hung system, will exhibit large amplitudes of vibration. It is necessary to provide rigid guides in a piping system to prevent excessive vibration. In particular, support heavy in-line components such as large valves. On hot systems, guides have to be placed judiciously to allow for expansion.

Avoid long pipe spans with fundamental frequencies within 20 % of rotating equipment frequencies.

The design of piping systems prone to vibration should avoid points of high stress concentration where vibration induced fatigue failure are known to have occurred, such as unreinforced branch connections. Replace sharp pipe-on-pipe fabricated branch connections by smooth, contoured integrally reinforced branch connections, to avoid the formation of vortices in the flow.

The frequent fatigue failure of pipe socket welds has lead to extensive studies of the effects of mechanical vibrations on the fatigue life of socket welded pipe joints [EPRI, Smith, Higuchi, Ricardella]. The potential for vibration induced fatigue leakage or rupture of pipe socket welds can be minimized in several ways: (a) tie back the vent or drain to the run pipe [Olson]; (b) use thicker wall pipe; (c) avoid lap or slip-on joints, (d) use full penetration welds; (e) insist on quality welding, particularly the first pass for fillet welds; (f) try to size fillet welds with the pipe leg twice the size of the socket leg.

In piping systems prone to vibration, avoid unnecessary changes in direction. Use surge tanks and terminal point tanks where necessary, as explained in section 8.4.

Use vibration damping devices, catalog items specially designed for that purpose [Mays]. Avoid the use of snubbers to eliminate or mitigate vibration. Snubbers are designed to restrain the pipe against large accelerations or velocities such as those due to earthquakes or water hammer, they are generally not meant to mitigate low amplitude high cycle vibration.

### 8.7.3 Preoperational Testing

Of all the prevention and mitigation measures, this one is the most effective: monitor the line when placed in service for the first time or when returned to service after modifications. Monitoring could be visual, followed by the more quantitative techniques described in section 8.5 if made necessary by visual inspections or unusual noise.

### 8.7.4 Reducing Turbulence and Cavitation

The reduction or elimination of cavitation must first rely on the proper hydraulic design of the piping system and sizing of pumps, compressors and valves. One way of reducing pulsation due to vortex shedding is to divide the flow area into narrow channels by using dividing vanes in elbows or valves with a tortuous flow path, as illustrated in Figure 8-10 for a large diameter bend used in water-works, and a ball valve.

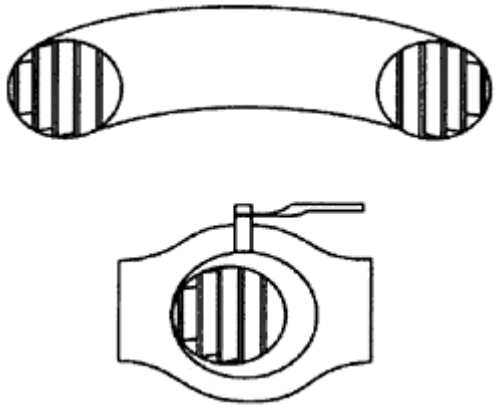

**Figure 8-10** Flow Vanes in Bend and Ball Valve

Control valves should be designed explicitly to prevent cavitation [ISA, Fisher, Zappe, Donavin, Husu]. In a recent application [Johnsen], a large 30" control valve, weighing 25,000 lb, was designed to control start-up flow (line packing) as well as full-flow operation of a pipeline on a North Sea platform. The design imposed limits on exit velocity head ($\rho v^2/2$) of 70 psi and noise to 70 dBA. At start-up, the inlet pressure is 1100 psi with a pressure drop at the valve of 360 psi. During operation, the inlet pressure is 900 psi with a pressure drop of 2 psi. To achieve these design objectives, the valve trim was fabricated with three zones along its 18" vertical stroke. At start-up, when the stroke is less than 40% and the pressure drop is large, the bottom of the trim has a tortuous path, forcing the flow through narrow channels, avoiding cavitation and excessive noise or vibration. At 40% to 60% stroke, the trim also includes V-shaped openings, which provide a larger flow area. From 60% to fully open, the trim has wider openings, with practically no pressure drop.

### 8.7.5 Pulsation Damper

Pressure pulses created by positive displacement pumps can be reduced by reducing the flow rate and pump speed (strokes per minute). In practice, these may not be viable solutions. Instead, a common solution is to add a pulsation damper downstream or upstream of the pump, depending on where the pressure pulses are of concern. A few practical guidelines regarding pulsation dampers [Warwick]:

(a) The damper volume should be a minimum of 15 pump strokes.

(b) The damper can be placed off a full outlet tee.

(c) The damper should be no further than ~40 diameters from the pump.

(d) Any pipe between the tee and damper should be straight, short, preferably no more than ~15 diameters, and should have the full size of the header.

### 8.7.6 Damping

The amplitude of vibration can be reduced by damping the piping system. This may be achieved by adding shims of rubber or other vibration damping materials around the pipe at adjustable supports [DOD], or by using specially designed viscous dampers [Mays]. The damping necessary to reduce the amplitude of vibration from $R_o$ to $R$ is [Kellog]

$$\zeta = \frac{1}{2} \frac{1 - \left(\frac{\omega}{\omega_n}\right)^2}{\frac{\omega}{\omega_n}} \sqrt{\left(\frac{R_o}{R}\right)^2 - 1}$$

$\zeta$=damping coefficient

$\omega$=circular frequency of the forcing function, 1/sec

$\omega_n$=circular frequency of natural frequency, 1/sec

$R_o$=amplitude of vibration without damping

R=amplitude of vibration at damping $\zeta$

Note that unless $\omega \sim \omega_n$ the damping $\zeta$ required to achieve a given reduction $R/R_o$ will be difficult to achieve in practice. This means that viscous damping is only practical close to resonance $\omega \sim \omega_n$.

### 8.7.7 Flexible Connections

Flexible hose, braided hose, corrugated metal tubes, and flanged elastomeric bellows are often used at pump inlet and outlet nozzles to protect pump alignment and decouple the pipe from the pump's mechanical vibration. Flexible connections are weaker than the pipe, and should not support weight or load. They should be installed following the supplier's catalog procedures. They should be rated to the full system design pressure, with a safety factor consistent with the pipe design code. The material should be compatible with the fluid, environment and operating temperature. They should be regularly inspected and replaced as necessary.

## 8.8 REFERENCES

API 541, Form-Wound Spiral Cage Induction Motors, American Petroleum Institute, Washington, DC.
API 610, Centrifugal Pumps for Petroleum, Heavy Duty Chemical, and Gas Industry Services, American Petroleum Institute, Washington, DC.
API 612, Special Purpose Steam Turbines for Petroleum, Chemical and Gas Industry Service, American Petroleum Institute, Washington, DC.
API 613, Special Purpose Gear Units for Refinery Service, American Petroleum Institute, Washington, DC.
API 617, Centrifugal Compressors for General Refinery Service, American Petroleum Institute, Washington, DC.
API 618, Reciprocating Compressors for Petroleum, Chemical, and Gas Industry Services, American Petroleum Institute, Washington, DC.
API 619, Rotary-Type Positive Displacement Compressors, General Refinery Services, American Petroleum Institute, Washington, DC.
ASME B31.1, Process Piping, American Society of Mechanical Engineers, New York.
ASME OM, Standards and Guides for Operation and Maintenance of Nuclear Power Plants, ASME OM-S/G-2000, American Society of Mechanical Engineers, New York.
Baldwin, R.M., Simmons, H.R., Flow-Induced Vibration in Safety Relief Valves: Design and Troubleshooting Methods, ASME Publication 84-PVP-8, PVP Conference and Exhibition, san Antonio, TX, June, 1984, American Society of Mechanical Engineers, New York.
Bauman, H.D., Solve Valve Noise and Cavitation Problems, Hydrocarbon Processing, March, 1997.
Blevins, R., Natural Frequencies & Mode Shapes, Van Nostrand.
Carucci, V.A., Mueller, R.T., Acoustically Induced Piping Vibration in High capacity Pressure Reducing Systems, 82-WA/PVP-8, The American Society of Mechanical Engineers, New York.
Clough, R.W., Penzien, J., Dynamics of Structures, McGraw-Hill Book Company, New York.
Coffman, J.T., Bernstein, M.D., Failure of safety valves due to Flow-Induced Vibration, Journal of Pressure Vessel Technology, Vol. 102, Feb. 1980, American Society of Mechanical Engineers, New York.
Crandall, H.S., Mark, W.D., Random Vibration in Mechanical Systems, Academic Press, New York.
Crane, Flow of Fluids through Valves, Fittings and Pipe, Technical Paper No. 410, Crane Co., Joliet, IL.
DOD MIL-STD-2148, Vibration Damping Materials, Procedures for Installation, Maintenance, and Repairs, U.S. Department of Defense.
Donavin, P.R., et. al., Feedpump Discharge Line Vibration Solved at Byron Station, Unit 2, Proceedings of the American Power Conference, Vol. 55-II, Illinois Institute of Technology, Chicago, IL, April, 1993.
Dove, R.C., Adams, P.H., Experimental Stress Analysis and Motion Measurement, Charles E.Merrill Books, Columbus, OH.
Dozaki, K., et. al., Modification and Design Guide of Thermowell for FBR, ASME PVP Conference paper PVP-Vol. 363, American Society of Mechanical Engineers, 1998.
Eitschberger, H., Revesz, Z., Cavitation-Induced Vibration in Piping, Structural Mechanics in Reactor Technology, 8th International Conference, Brussels, Belgium, 1985, North-Holland, 1985.
EPRI, EPRI Fatigue Management Handbook, TR-104534, December 1994, Electric Power Research Institute, Charlotte, NC.
Fisher, Control Valve Handbook, Fisher Controls, Marshaltown, Iowa.

Harris, C.M., Crede, C.E., Shock and Vibration Handbook, McGraw Hill Book Company, New York.
Hawkins, K., Minimizing Amplification of a Pumping Unit's Critical Resonance Frequencies, Pumps and Systems Magazine, June, 1999.
HI 9.8, Pump Intake Design, Hydraulic Institute, Parsippany, NJ.
HI, Hydraulic Institute Standards for Centrifugal, Rotary & Reciprocating Pumps, Hydraulic Institute, Parsippany, NJ.
Higuchi, M., et. al., High Cycle Fatigue Strength of the Fatigue Strength Reduction Factor of Socket Welded Pipe Joint, ASME 1997 PVP Conference, Vol. 353, July 1997, American Society of Mechanical Engineers, New York.
Husu, M., New, Simple Solutions for Vibration, Noise and Rangeability Problems in Natural Gas Control Valves, Energy Processing/Canada, May–June, 1998.
ISA, Handbook of Control Valves, Instrument Society of America, Pittsburgh, PA.
ISA S75.02, Standard Test Procedures for Control Valve Capacity, Instrument Society of America, Pittsburgh, PA.
ISO 2954, Mechanical Vibration of Rotating and Reciprocating Machinery—Requirements for Instruments for Measuring Vibration Severity, International Standards Organization.
Johnsen, O., Smirt, P.A., Valve Serves both Start-up, Full-flow Operation Functions, Pipeline & Gas Industry, May, 1999.
Kellog, Design of Piping Systems, the M.W.Kellog Company, 1956.
Lyons, J.L., Askland, C.L., Lyon's Encyclopedia of Valves, Van Nostrand Reinhold Company, New York.
Mays, B., Rencher, D., Viscous Dampers Cut Vibration in Heater Drain Line, Power Engineering, June, 1993.
Morishita, M., Wada, Y., Fatigue Analysis of Thermowell due to Flow-Induced Vibration, ASME PVP Conference paper PVP-Vol. 363, American Society of Mechanical Engineers, 1998.
Moussa, W.A., AbdelHamid, A.N., On the Evaluation of Dynamic Stresses in Pipelines Using Limited Vibration Measurements and FEA in the Time Domain, ASME PVP-Vol. 368, 1998, American Society of Mechanical Engineers, New York.
Olson, D.E., Chun, H.S., Avoiding Tap Line Vibration, ASME Publication 82-PVP-54, Pressure Vessel and Piping Conference, Orlando, FL, June, 1982, American Society of Mechanical Engineers, New York.
Olson, D.E., Piping Vibration Experience in Power Plants, Pressure Vessel and Piping Technology 1985—A Decade of Progress, Sundararajan, C, ed., American Society of mechanical Engineers, New York.
Pilckey, W.D., Formulas for Stress, Strain, and Structural Matrices, John Wiley & Sons.
PMC/Beta, All Solid State Vibration Switches, PMC/Beta, Natick, MA.
Ricardella, P.C., et. al., Vibration Fatigue Testing of Socket Welds, ASME PVP-Vol. 360, 1998, American Society of Mechanical Engineers, New York.
Sadaoka, N., et. al., Development of Analysis System for Flow-Induced Vibrations in Piping Systems, ASME PVP Conference paper PVP-Vol. 363, American Society of Mechanical Engineers, 1998.
Smith, J.K., et. al., Development of Screening Procedure for Vibrational Fatigue of Small Bore Piping, ASME 1995 PVP Conference, Vol. 313–2, July 1995, American Society of Mechanical Engineers, New York.
Smith, J.K., Vibrational fatigue Failures in Short Cantilevered Piping with Socket-Welded Fittings, ASME 1996 PVP Conference, Vol. 338, July 1996, American Society of Mechanical Engineers, New York.
Sparks, C.R., et. al., SGA-PCRC Seminar on Controlling the Effects of Pulsations and Fluid Transients in Piping Systems, Report No. 160, South West Research Institute, San Antonio, TX, November, 1979.

Stoneking, J.E., Kryter, R.C., Screening Procedures for Vibrational Qualification of Nuclear Plant Piping, ASME Publication 80-C2/PVP-4, Century 2 Pressure Vessel and Piping Conference, San Francisco, CA, August, 1980, American Society of Mechanical Engineers, New York.
Tullis, J.P., Hydraulics of Pipelines, Pumps, Valves, Cavitation, Transients, Wiley Inter-Science.
Vierck, R.K., Vibration Analysis, International Textbook Company, Scranton, PA.
Wachel, J.C., Bates, C.L., Techniques for Controlling Piping Vibration and Failure, ASME publication 76-Pet-18, Joint Petroleum Mechanical Engineering and Pressure Vessels and Piping Conference, Mexico City, September, 1976, American Society of Mechanical Engineers, New York.
Wachel, J.L., Piping Vibration and Stress, Proceedings Machinery Vibration Monitoring and Analysis Seminar Meeting, New Orleans, LA, April 7–9, 1981.
Wachel, J.L., Piping Vibration and Stress, Machinery Vibration and Analysis Seminar, Vibration Institute, New Orleans, LA, 1982.
Warwick, E., Minimizing Pressure and Flow Pulsations from Piston/Diaphragm Metering Pumps, Pumps and Systems Magazines, March, 1999.
Zapp, R.W., Valve Selection Handbook, Gulf Publishing Company, Houston, TX.

# 9
# Fluid Transients

Fluid transients are sudden changes in flow velocity and pressure. These transients are important because they can cause large forces and overpressure that can fail piping and support systems. To best understand these effects, we refer to Figure 9-1. As shown in the top right box, fluid transients become evident when operators hear loud noises coming from the piping, or witness violent movements, in some cases the pipe jumps off its supports and visibly deforms, leaks or even ruptures. Pipe ruptures from fluid transients are due either to a large and sudden overpressure that burst the pipe or from excessive tension or bending forces caused by large pressure imbalance in the system.

Fluid transients can be due to one of four causes, as indicated by the four columns of Figure 9-1. This figure charts the logical framework for understanding fluid transients and will be followed in this chapter, progressing from left to right. In the first case (left hand column in Figure 9-1) the system is full of flowing liquid. The transient in a full liquid system is due to sudden changes in flow velocity (dv) that result in large pressure changes (dP). These changes in pressure cause forces (F) and overpressure (P+dP) in the system. In the second case (second column from left in Figure 9-1), the system contains liquid and its vapor (for example water and steam). Here, the transient is due to the cooling and subsequent collapse of the vapor, causing what is often referred to as bubble collapse water-hammer. The larger the bubbles, the more violent the transient. The third case (third column from left in Figure 9-1) is that of a flow of mixed liquid and gas, causing pressure fluctuations and transient forces in the line. A down to earth example of such liquid-gas transients is the gushing of a mixture of water and air when first turning on a garden hose. In the last case (the right hand column in Figure 9-1) a purely gaseous flow causes a fluid transient when the gas suddenly discharges through an orifice, which is for example the case during the discharge of a safety valve in a gas or steam system.

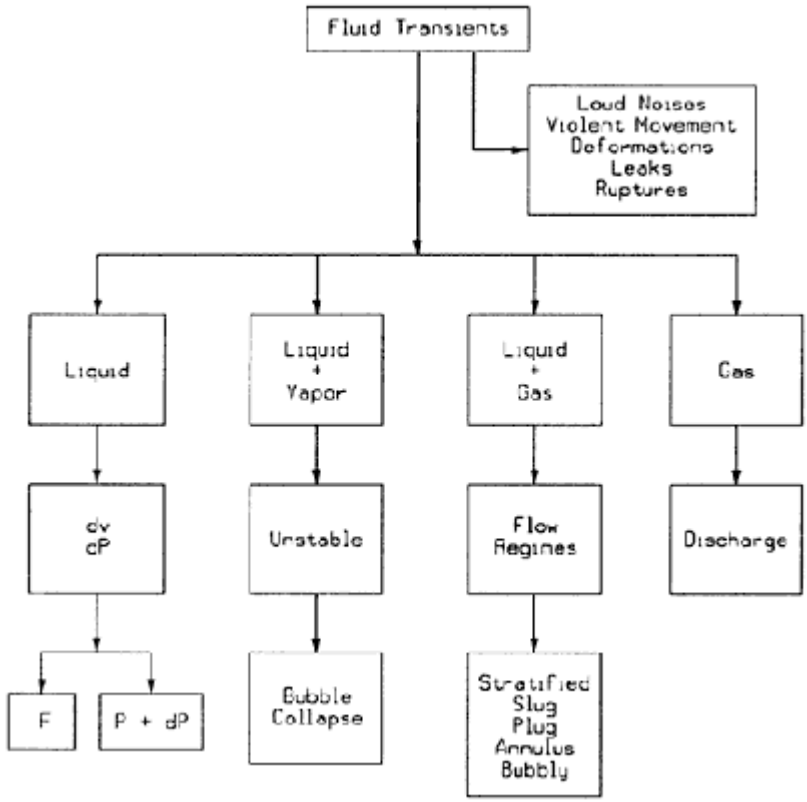


**Figure 9-1** The Big Picture for Fluid Transients

## 9.1 SINGLE LIQUID PHASE

### 9.1.1 Bulk or Propagative Flow

A liquid flows steadily at a velocity v and pressure P from a reservoir through a valve, as illustrated in Figure 9-2(a). If the valve is closed, the liquid close to the valve is brought to a stop, its velocity is v=0 and its pressure has increased to P+dP. If we could take a snapshot of the fluid velocity and its pressure, a short instant after the valve has closed, we would see an interface between the stopped liquid (v=0 and P+dP) and the rest of the fluid still flowing left to right in Figure 9-2(b) at a velocity v and pressure P. This interface is the front of a pressure wave with pressures P+dP on one side and P on the other, moving upstream, right to left in Figure 9-2, at the speed of sound in the liquid.

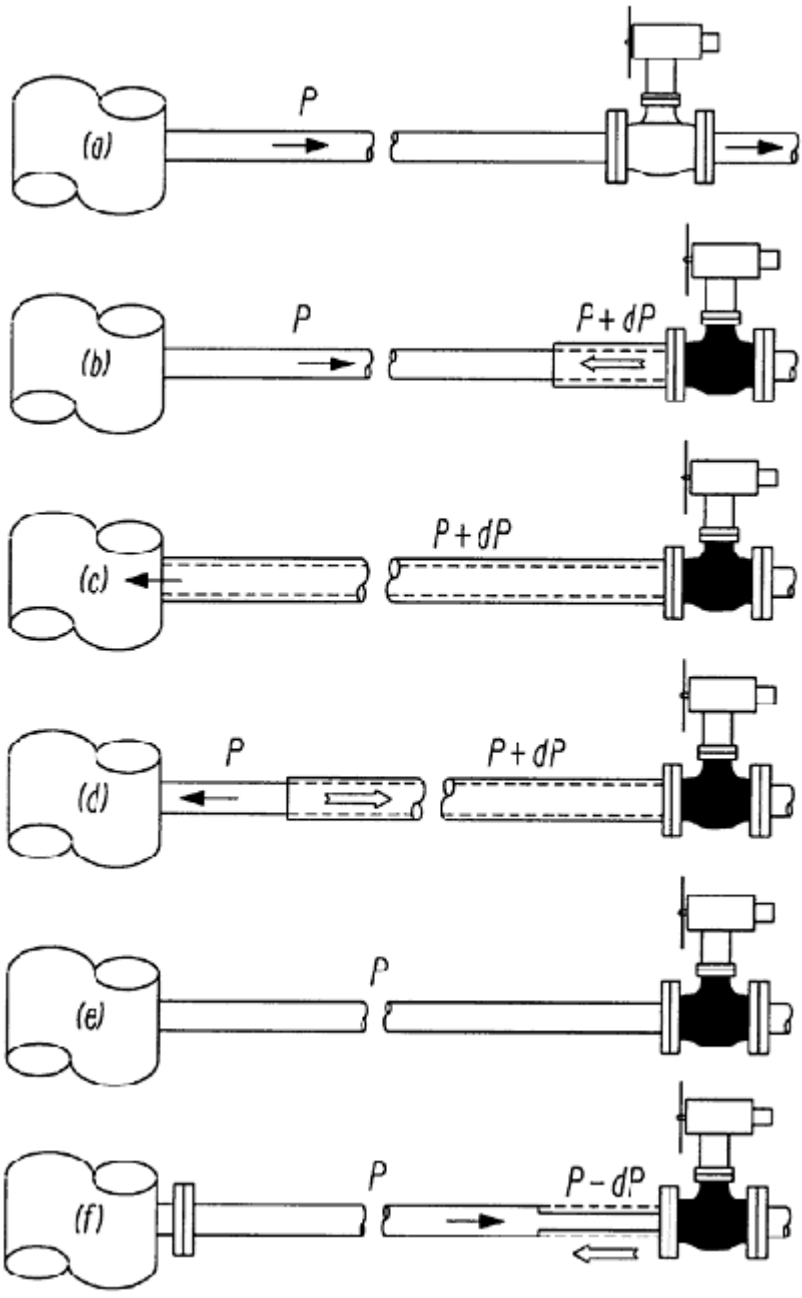


**Figure 9-2** Travel of Pressure Wave

When the wave front reaches the reservoir to the left, the whole pipe is at pressure P+dP, Figure 9-2(c). At this point, the pressure in the pipe at the reservoir inlet drops back to P. The pressure has dropped to P at the reservoir to the left, and this pressure drop from P+dP to P travels back towards the valve, Figure 92(d), eventually bringing the whole line back down to pressure P, left to right.

As the pressure drop reaches the closed valve, Figure 9-2(e), the whole pipe is at pressure P, but an important effect, which is not intuitively evident, takes place now: the pressure wave "reflects" at the valve and, having dropped from P+ dP down to P, it continues to drop by an equal amount dP, from P down to P−dP. This low pressure P−dP now travels back towards the reservoir, right to left on Figure 9-2(f).

This back and forth movement of the pressure wave, which varies between P+dP and P−dP, continues for a few cycles, before dying out from friction and loss of energy through the pipe wall.

If L is the distance between the reservoir and the valve, and since the pressure wave travels at the speed of sound a, it will take a time $t_p$ for the pressure wave to travel from the valve to the reservoir and back, a distance 2L, with

$$t_P = \frac{2L}{a}$$

$t_P$=propagation time, sec

L=distance from the pressure disturbance, in

a=speed of sound in the fluid, in/sec

Let $t_D$ be the time during which the initial pressure disturbance occurs, for example the time it takes the valve to close. If the disturbance time is much larger than $t_P$ it takes the pressure wave to take a round trip, $t_D >> t_P$ the valve closure is slow, the flow transient condition is referred to as bulk flow and is accompanied by a small pressure surge. If $t_D$ is equal or shorter than the wave travel time $t_P$ the transient is a waterhammer accompanied by a much larger pressure increase, as summarized in Table 9-1.

**Table 9-1** Description and Classification of Transients

| $f_p$ | Description | Classification |
|---|---|---|
| <2L/a | Instantaneous | waterhammer |
| ~2L/a | Rapid | waterhammer |
| >>2L/a | Slow | surge |

The speed of sound in a fluid filled pipe is given by [Streeter]

$$a = \sqrt{\frac{\kappa/\rho}{1 + \frac{D}{e}\frac{\kappa}{E}}}$$

a=sonic velocity, in/sec

κ=bulk modulus, psi
ρ=liquid density, lb/in$^3$
D=pipe inside diameter, in
e=pipe wall thickness, in
E=Young modulus of the pipe material, psi

Examples of liquid properties and sonic velocities at room temperature, neglecting the pipe elasticity, are listed in Table 9-2.

**Table 9-2** Example of Liquid Properties at Room Temperature

| Liquid | ρ (slugs/ft$^3$) | κ (psi) | a (ft/sec) |
|---|---|---|---|
| Ethyl alcohol | 1.532 | 130,000 | 3,810 |
| Benzene | 1.705 | 154,000 | 4,340 |
| Glycerin | 2.447 | 654,000 | 6,510 |
| Kerosene | 1.564 | 188,000 | 4,390 |
| Mercury | 26.283 | 3,590,000 | 4,770 |
| Machine oil | 1.752 | 189,000 | 4,240 |
| Fresh water | 1.937 | 316,000 | 4,860 |
| Salt water | 1.988 | 339,000 | 4,990 |

Note: For example, for fresh water, 1.937 (32.2)=62.4 lb/ft$^3$

### 9.1.2 Pressure Change in Bulk Flow

When the change of velocity is slow (bulk flow) the pressure rise is given by [Parmakian, Moody]

$$dP = \frac{\rho(\Delta v)^2}{2g}$$

dP=pressure change at disturbance, psi
ρ=fluid density, lb/in$^3$
Δv=change of fluid velocity at disturbance, in/sec
g=gravity=386 in/sec$^2$

For example, if a valve is closed slowly on water at ambient temperature (ρ =62.4 lb/ft$^3$) flowing at 10 ft/sec, the pressure rise in bulk flow is dP=0.7 psi, practically negligible.

The momentum force exerted on a gradually and linearly closing valve in bulk flow is [Moody]

$$F_B = \frac{\rho A v}{g}\frac{L}{t_D}$$

$F_B$=force on closed surface in bulk flow, lb
v=initial velocity, in/sec
A=cross section area of closed surface, $in^2$
L=length of pipe, in
For a slow closing valve, with a finite closing time T, the pressure rise is given by [Lyon]
$H_R=M-H-(M^2-m_1^2)^{0.5}$
$M=m_1+m_2$
$m_1=H+H_{R,max}$
$m_2=v^2\,(a-2L/T)^2/(2g^2\,H)$

$H_R$=pressure rise, ft
$H_{R,max}$=maximum pressure rise from a waterhammer (Joukowski), ft
v=flow velocity, ft/sec
L=pipe length, ft
T=valve closing time, sec
H=system static head, ft (2.3 $1\times P_{psi}$ for water)
g=32.3 $ft/sec^2$
Conversely, for a slow opening valve, the pressure drop is
$H_D=[n_S(2H+n_S)]^{0.5}-n_s$
$n_s=(2/H)[v\,L/(gT)]^2$

$H_D$=pressure drop, ft
T=valve opening time, sec

### 9.1.3 Waterhammer

If the change in flow velocity occurs over a short period of time $t_D$, of the order of magnitude or shorter than the propagation time $t_P$, then the flow transient is a waterhammer (propagative flow). In this case, the pressure rise due to the sudden, instantaneous valve closure ($t_D<t_P$) is given by Joukowsky's formula [Streeter]

$$dP = \frac{\rho a(\Delta v)}{g}$$

dP=pressure change due to instantaneous valve closing, psi
ρ=fluid density, $lb/in^3$
Δv=change of fluid velocity at disturbance, in/sec
g=gravity=386 $in/sec^2$
This formula was first derived by Professor N.Joukowski, in Moscow in 1898, and published in English in the Proceedings of the American Water Works Association in 1904. It has come to be known as Joukowski's waterhammer formula. Going back to the

example in section 9.1.2, for water flowing at ambient temperature at 10 ft/sec ($\rho$=62.4 lb/ft$^3$, a=4860 ft/sec, $\Delta$v=10 ft/sec, g=32.2 ft/sec$^2$) the sudden closure of a valve will cause a pressure rise dP=654 psi, a very significant pressure spike. Note that the pressure rise does not depend on the initial pressure, only on the initial velocity.

Conversely, for an instantaneously opening valve, the pressure drop is [Lyon]

$H_D=[n_F(2H+n_F)]^{0.5}-n_F$

$n_F=H_{R,max}{}^2/(2H)$

$H_D$=pressure drop, ft

$H_{R,max}$=maximum pressure rise from a waterhammer (Joukowski), ft

H=system static head, ft

The large pressure rise dP that accompanies a waterhammer has two damaging effects: first, some pipe fittings or components, for example cast iron valve bodies, instruments or expansion joints, or even the pipe itself, may not be able to withstand the increased pressure P+dP, and may fail by bursting under the transient pressure. Second, at changes in direction or cross section, the pressure rise will cause unbalanced axial loads A(dP) where A is the cross section area of the obstruction or the pipe area at an elbow. Under the effect of this unbalanced force, the pipe may move and bend, potentially causing the rupture of pipe hangers and guides, plastic deformation, and possibly rupture of the pipe. Because this effect is at the root of serious accidents, we will examine it now in more detail. We will consider part of a piping system in which a waterhammer is taking place, Figure 93, where F is the axial force in the horizontal pipe run and t is time. We have seen that the pressure wave P+dP is moving upstream at the speed of sound. At one point the pressure wave passes around an elbow, as shown in Figure 9-3, position B. The elbow to the left sees a force (P+dP)A, where A is the flow area of the pipe, while the downstream elbow at right still sees a force PA.

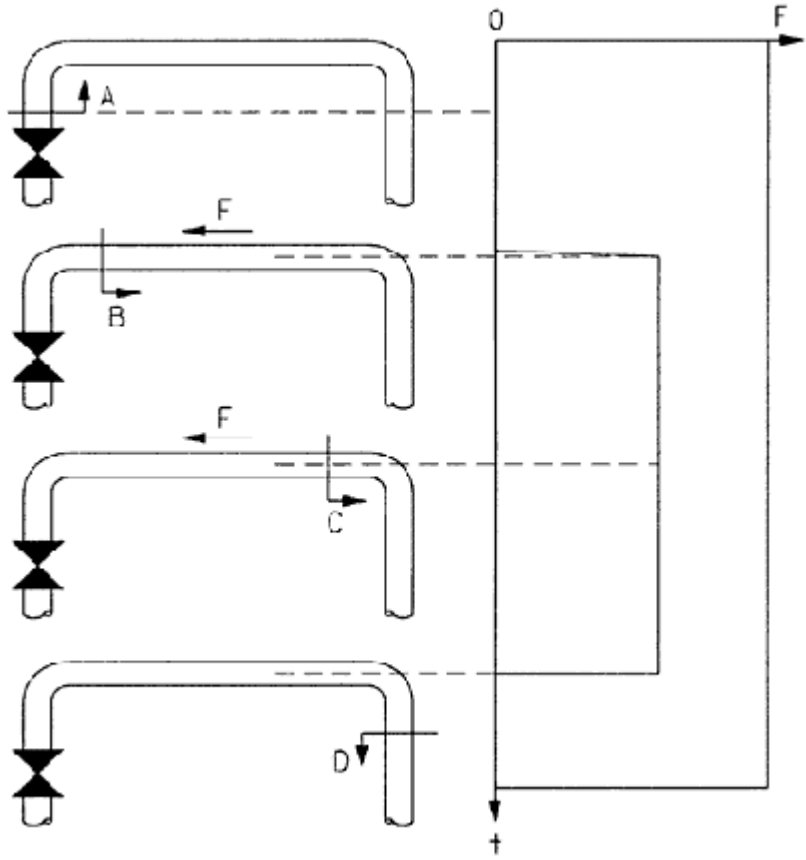


**Figure 9-3** Waterhammer Force Imbalance F Between Elbows

If the distance between the two opposite elbows is L, the unbalance of pressure between the elbow at P+dP and the elbow at P will last for a short time $t_E$= L/a, until the pressure wave reaches the second elbow, in position C. During this period of time, as the pressure wave goes from position B to position C, the segment of pipe is subject to a force (P+dP)A pushing the left elbow to the left, and a force PA pushing the right elbow to the right. The resultant is an axial force F=(P +dP)A−PA=(dP)A towards the left. This force is applied very quickly (dynamically) and lasts for a period of time $t_E$ until the pressure wave clears the elbow and reaches position D of Figure 9-3. At this point the right elbow is at a pressure P+ dP and the corresponding force (P+dP)A balances the force at the left elbow. It is in fact a very short impulse force, as if the pipe was banged along its axis, between the two elbows, with a giant hammer. All the rules of impact analysis apply in this case (Chapter 12). If the duration of the impulse $t_E$ is sufficiently long compared to the pipe's natural period (the inverse of its natural frequency), then the pipe has time to deform and damage can be significant. The dynamic force applied to the pipe is

$F_{Dyn}$=F(DLF)

$F_{Dyn}$=dynamic force applied to the pipe, lb
F=maximum static force (dP)A, lb
DLF=dynamic load factor (Chapter 12)

### 9.1.4 Valve Characteristics

In practice, the valve closing time $t_D$ depends on the valve characteristic, the relationship between stem position (or disk angle or ball angle) and flow [Dickenson].

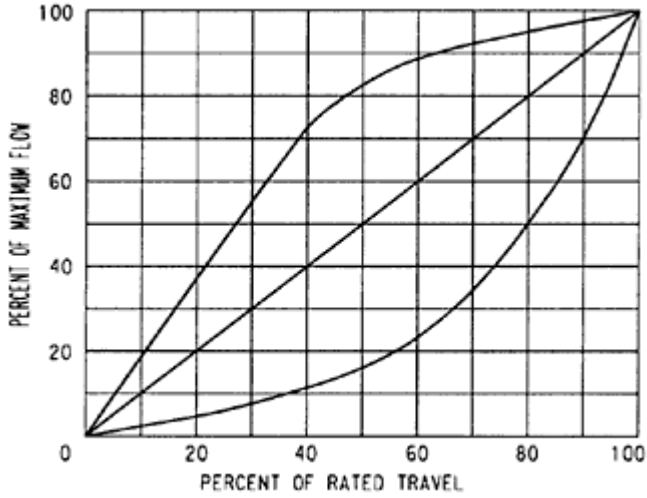


**Figure 9-4** Typical valve Characteristics

Some common valve characteristics include quick opening (top curve), linear (middle line), equal percentage and hyperbolic (bottom curve). In practice, many gate, plug, butterfly and ball valves have characteristics between linear and equal percentage. If a valve still passes 90% of the flow rate even when the valve stem has traveled 50% of its distance. In this case, the valve closing time is, for all practical purposes, less than half the actual closing time.

### 9.1.5 One-to-Two Phase Transient

We have seen that a pressure transient due to valve closure results in a pressure wave that suddenly increases the pressure from its initial value P up to P+dP. This increase is followed by a drop to P−dP, with back and forth fluctuations between P+dP and P−dP, until the pressure fluctuations gradually cease. This is illustrated by the solid line in Figure 9-5, which is the measurement of pressure in the line at one fixed point as a function of time. With the line pressure starting at $P_2$, rising to $P_3$=$P_2$+dP at the onset of

the waterhammer and then fluctuating around $P_2$ until the pressure surges gradually ceases.

In the case where the pressure rise dP is larger than the initial pressure P ($P_1$ to $P_4=P_1+dP$ in Figure 9-5) then, when the transient pressure drops to $P_1-dP$, it will drop to a negative pressure, below the liquid's vapor pressure, and part of the liquid will vaporize during the transient.

We now have a vapor pocket surrounded by liquid undergoing the transient pressure change. This is an unstable condition, and is illustrated by the dashed line in Figure 9-5, with the line pressure starting at $P_1$, rising to $P_4$ at the onset of the waterhammer, and then decreasing to a vapor pressure of 0 psi and vaporizing. The liquid surrounding the vapor pocket will at some point rush into the vapor and collapse the vapor bubble. This collapse will give rise to a large pressure spike, and is often times accompanied by a loud "bang".

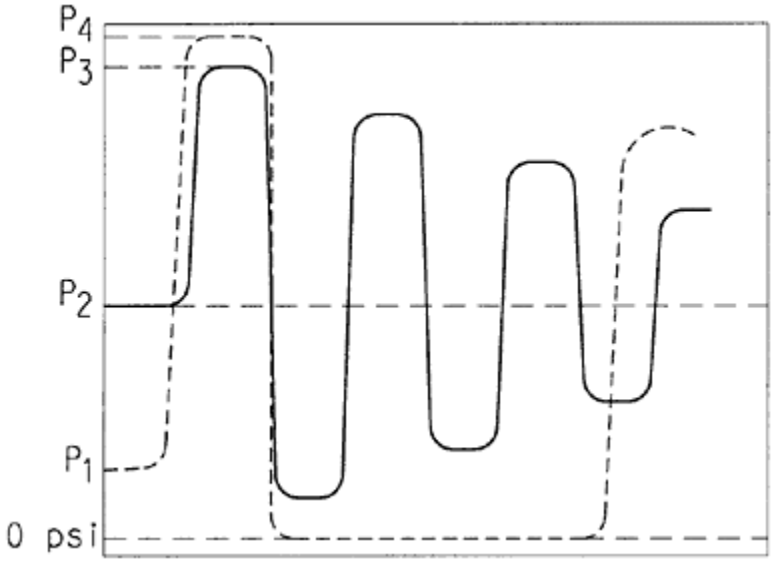


**Figure 9-5** Pressure Trace During a Waterhammer

The single-phase liquid transient, damaging on its own account, has evolved into a yet more damaging two-phase liquid-vapor transient, which will be analyzed later in this chapter. The fact that the time elapsed when the pressure is at zero after the vapor has formed (bottom part of dashed line at 0 psi) is longer than the low pressure period with the solid liquid pipe (bottom part of solid line) is because the speed of sound is much slower in the liquid-vapor two-phase condition (dashed line) than in the liquid phase (solid line). The velocity of sound in a liquid-gas mixture is

$$a = \frac{1}{\sqrt{\rho_m\left(\frac{\alpha}{P_G} + \frac{1-\alpha}{K_L} + \frac{D}{tE}\right)}}$$

$\rho_m$=density of liquid-gas mixture, lb/in$^3$
$P_G$=gas pressure, psi
$K_L$=bulk modulus of elasticity of liquid, 318,000 psi water at 70°F
t=pipe wall thickness, in
E=Young's modulus of pipe, psi

### 9.1.6 Pump Fill Rate

If a pump fill rate is set at [Moody]

$$Q(t) = Q_{max}\left(1 - e^{-t/\tau}\right) = Av$$

$Q_{max}$=maximum flow rate, in$^3$/sec
$\tau$=pump start time, sec
A=pipe flow area, in$^2$
v=flow velocity, in/sec
the liquid position in the pipeline, at a given time t is given by

$$x = \frac{Q_{max}}{A}\left(t - \tau + \tau e^{-\frac{t}{\tau}}\right)$$

and the resulting force on a pipe elbow is

$$F = \frac{\rho A}{g}\left(\frac{Q_{max}}{A}\right)^2\left[1 + 3e^{-\frac{2t}{\tau}} - \left(3 - \frac{t}{\tau}\right)e^{-\frac{t}{\tau}}\right]$$

### 9.1.7 Prevention of Liquid Waterhammer

The first step in preventing a liquid waterhammer is to recognize its likelihood. An understanding of Figure 9-1 goes a long way in achieving this objective. Then, layout and components must be selected to prevent fast flow transients. Finally, the systems must be operated to avoid waterhammer. It is clear from the preceding sections that it is advantageous to slow down the closure of isolation valves. By doing so, the valve closure transient will be in the form of bulk flow, with an insignificant pressure surge. With check valves, it is advantageous to have the valve close on a short travel stroke so as to limit the amount of back-flow, and therefore the speed of valve closure and the magnitude of the slamming force.

An atmospheric surge tank can be added upstream of fast closing isolation valves to absorb most of the pressure surge caused by valve closure or opening [Parmakian]. During normal operation, the water level in the surge tank matches the hydraulic grade

line at the point of tie-in. Rather than travel through the line, the pressure increase in the pipe due to valve closure would be absorbed in the tank, by a temporary increase of tank level. Conversely, a pressure drop at valve opening will be compensated by a temporary decrease in surge tank level. To avoid building a tall surge tank to match the pipeline pressure, a second type of surge tank, a one-way surge tank, can be used. In this case, the tank is filled below normal operating pressure. A check valve closed by line pressure retains the tank's contents. When the line pressure drops, the surge tank contents are discharged into the line, to compensate for the pressure drop. A third type of surge tank is the air chamber. Here, the contents of the surge tank are kept at pressure by compressed air or nitrogen.

## 9.2 TWO-PHASE VAPOR-LIQUID WATERHAMMER

### 9.2.1 Steam-Water Waterhammer

Two-phase vapor-liquid transients are transients that take place when the liquid coexists in the pipe with its vapor, one phase converting into the other as the pressure rises or drops. The classic example occurs in steam lines that contain water condensate [Van Duyn, Bjorge, EPRI]. The transients that take place under these conditions can be extremely violent, leading to rupture of pipe and, in the case of steam, because of the high temperatures and energies involved, to severe injuries and even fatalities.

The phenomenon of steam-water waterhammer has been well understood since the early years of the industrial revolution with the expanding use of steam power and steam heating. Indeed, little can be added today to the excellent description dating back to 1883 [Thurston]:

"When a pipe is filled with steam, and then has introduced into it a quantity of cold water, or when a pipe, itself cold, and containing cold water, even in very small quantity, and without pressure, has steam turned into it, the first contact of the two fluids is accompanied by a sudden condensation which causes a sharp blow to be struck, usually at the point of entrance; and sometimes a succession of such blows, which are the heavier as the pipe is large, and which may be startling, and even dangerous. The steam, at entrance, passes over, or comes in contact with the surface of the cold water standing in the pipe. Condensation occurs, at first very slowly, but presently more quickly, and then so rapidly that the surface of the contact between the two fluids is broken, and condensation is completed with a suddenness that produces a vacuum. The water surrounding this vacuum is next projected violently from all sides into the vacuous space, and, crossing it, strikes upon the surface surrounding it. As water is nearly incompressible, the blow thus struck is like that of a solid body… Where pipes are not burst by this action, it is common to see them sprung and twisted out of line, torn from their connections and, when a succession of shocks occur, as is often the case, the whole line writhes and jumps lengthwise to an extent that is sufficiently serious to cause well-grounded alarm." "It seems very certain that we may consider it as proven that the pressures produced by "water-hammer" are often enormously in excess of those familiar to us in the use of steam, and that they have, in many cases exceeded 1000 pounds per square inch. It is, then, evident that it is not often safe to calculate upon meeting these

tremendous stresses by weight and thickness of metal, but that the engineer must rely principally, if not solely, upon complete and certain drainage of the pipe at all times as the only means of safely handling steam in long pipes, such, especially, as are now coming into use in the heating of cities by steam led through the streets in underground mains."

The wise advise in Mr. Thurston's last sentence is echoed, 100 years later, in the U.S. Nuclear Regulatory Commission report, NUREG-0927, 1984, which states: "State-of-the-art mechanistic or quantitative two-phase analysis of water-hammer phenomena is not a practical means of resolving all waterhammer…the extensiveness of possible plant conditions, alignments, and computer code calculational limits preclude analyzing all possible scenarios… Anticipated water-hammer events, caused by components performing in their intended manner should be included as occasional loads in the design basis of piping and their support systems." This correctly recognizes that it is practically impossible to consider at the design stage all the possible malfunctions and errors that can lead to waterhammer and the resulting overpressure and loads. The right approach is to focus on good system design (slope, drip legs, drains, vents, traps, filters, valves, etc.) and the use of correct and clear operating procedures to prevent the water-hammer from occurring in the first place.

### 9.2.2 Case Histories

To best illustrate liquid-vapor two-phase waterhammer, we investigate three actual incidents. The first incident resulted in extensive damage. The second case resulted in the death of a plant operator. The third case illustrates well the unpredictable nature of liquid-vapor transients.

In the first case, saturated steam is supplied at 150 psig and 366°F to a series of branch lines off two parallel headers, as shown in Figure 9-6. A cross-over leg (VD) ties the two headers [Andrews]. The headers include expansion joints to allow for the expansion of the straight pipe runs at the operating temperature of 366°F. The headers are mostly rod hung, with three anchors on each header, closely following the standard layout guidance of the Expansion Joints Manufacturers Association [EJMA]. The 10" header (isolation valve VB to blind flange BF) had been isolated for a period of four weeks, with valves VA and VB and cross-over VD closed, and the isolation valves on the branch lines between VB and blind flange BF also closed. With the line section VA-to-VB still out of service, steam supply was to resume in the branch line between VE and BF. Cross-over valve VD was slowly opened to feed steam from one header to the other; then the operator slowly opened the branch line's isolation valve to restart steam flow. It is at this point, when opening the branch line valve, that a loud waterhammer occurred.

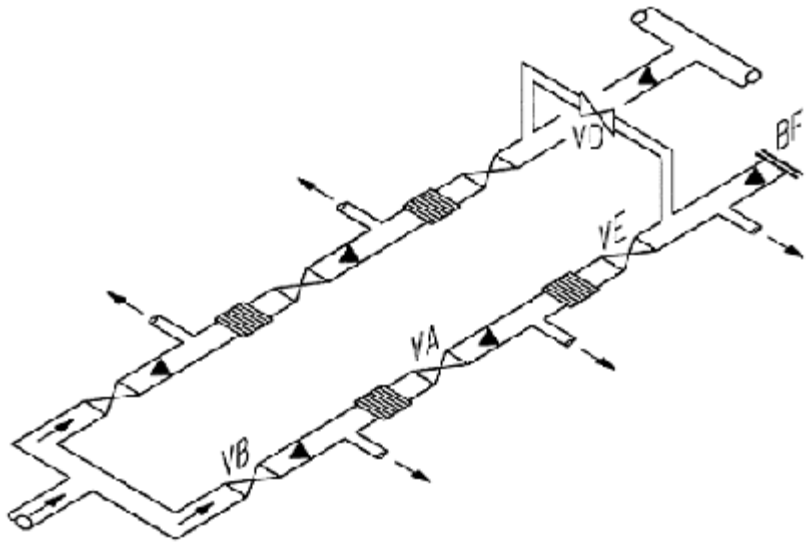


**Figure 9-6** Steam System Layout, First Case Study

The branch line valve was promptly closed, but the waterhammer continued for several minutes. The waterhammer was finally stopped by closing the cross-over valve VD. What happened? When the line was shutdown for the four-week period, steam condensed into water between VA and BF. The water was not drained before the steam flow was reintroduced into the branch line, through VD. We now have hot steam flowing over and into the water, the water cools pockets of steam that condense into water and leave in their place a large vacuum bubble. The water rushes into this vacuum bubble causing violent impacts against the pipe wall and local but large pressure spikes. Note an important point: When valve VD was first opened, steam was slowly bubbled into the trapped condensate and gradually heated the water. The steam was in equilibrium with the hot water. Once the branch valve was opened and flow started, the steam mixed rapidly with the bulk of condensate, cooling rapidly and collapsing causing the bubble collapse waterhammer.

In the second case, a section of 150 psi steam line had been shutdown for a period of several months, and steam condensate had formed upstream of the isolation valve, as shown by W (water) in the top sketch of Figure 9-7. When came time to restart steam flow into the lower line, the work order specified that the operator was to first slightly crack open the isolation valve and permit a small flow of condensate to slowly mix with and be entrained by the steam in the upper header.

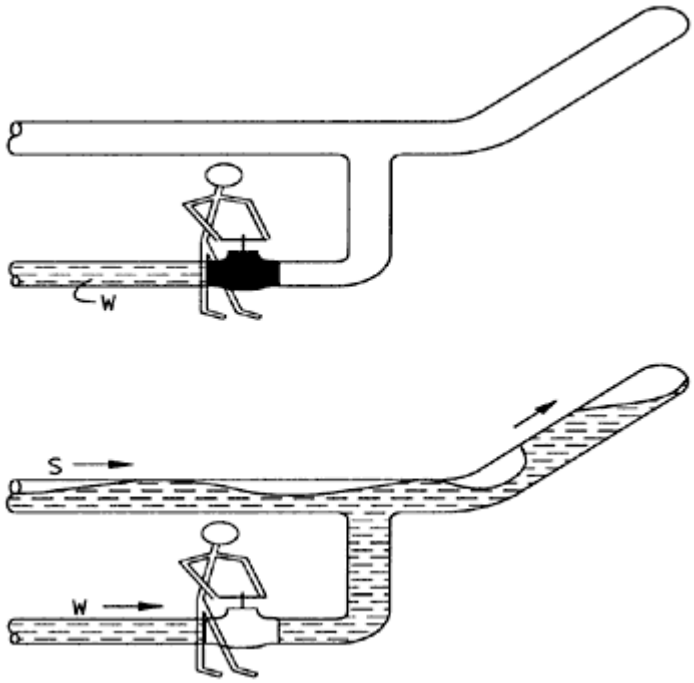


**Figure 9-7** Steam System Layout, Second Case Study

Shortly after starting the operation, there was a violent explosion that ruptured the body of a cast iron valve, releasing steam in the valve pit. The operator died from burns to the lungs caused by inhalation of steam escaping from the ruptured cast iron valve body. Post-accident investigations indicated that as the valve was opened, water entered the downstream header in which steam was circulating (S in Figure 9-7), the water-steam mixture had formed steam bubbles trapped at the top of the pipe by the wavy surface of the water; the steam bubbles cooled down and collapsed, causing pressure spikes measured to be twenty times the initial steam pressure, a pressure spike sufficiently large to rupture the cast iron body of a valve. Interestingly, the steel components, including the body of cast steel valves, did not rupture. Only the body of a cast iron valve ruptured.

In the third case, steam was introduced into a line with condensate. No one was aware that there was condensate trapped in the line. The steam flow was very slow, resulting in a stratified condition: the steam flowing slowly on top of the condensate trapped at the bottom of the pipe, which results in a top-to-bottom temperature gradient, with the top hot, at steam temperature, and the bottom cooler, at water temperature. To the operators' surprise the line visibly bowed upward and lifted several inches off its supports over a distance of close to 100 ft. In this case, fortunately for the operators, a bubble collapse waterhammer did not occur. Instead a stable stratified flow condition occurred, with steam flowing on top of condensate. This stratified flow is difficult to predict in two-phase water-steam conditions since the steam could condensate and collapse at any

moment. It illustrates that, sometimes, steam and water can co-exist without causing a waterhammer.

### 9.2.3 Predicting the Effects of Two-Phase Transients

A liquid loop operating at 60 psig and ambient temperature is supplied by a pump and discharges into an atmospheric tank, Figure 9-8. When the pump shuts down, the column of liquid in the 50 ft vertical leg on the side of the atmospheric discharge tank will drain down, settling down from 50 ft (point B) to 34 ft (point A) since a 34 ft water column corresponds to 1 atmosphere. As it drains down, the liquid will leave above it in the vertical leg a vapor space AB that is 50−34=16 ft tall. If a vacuum breaker was placed at the top of the riser, point B, this drop of liquid in the vertical leg would automatically suck in ambient air to fill the vacuum. Whether the 16 ft are filled with vapor or air, it is said that the column of liquid has separated; there is column separation between A and B. The check valve at C will keep CB filled with liquid, unless the check valve is leaking back towards the pump.

As the pump restarts, the pressure rises quickly to the operating pressure of 60 psi. If the 16 ft long section AB is filled with vapor (no vacuum breaker at B) the upstream liquid column (interface B) rushes forward very rapidly, traveling 16 ft towards the downstream liquid column (interface A), collapsing the vapor pocket from B to A. An impact of the liquid columns, a waterhammer, takes place when the two water columns rejoin as B reaches A.

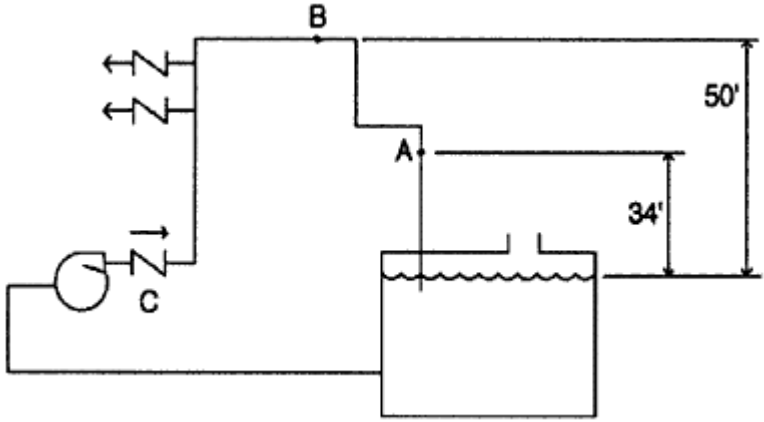


**Figure 9-8** Siphon Column Separation

The same effect takes place in a liquid line as a result of waterhammer if the pressure rise dP due to waterhammer is sufficiently large. The reflected pressure, P-dP, could be sufficiently low to reach the vapor pressure of the liquid and form a vapor bubble that separates the liquid into two columns. This is also a column separation. The vapor bubble will collapse as the pressure tends to recover. This collapse of the vapor bubble will cause the two columns of water to rush towards each other to fill the vacuum created by the

condensation of vapor. The separated columns will rapidly slam into each others causing a waterhammer.

If separated left and right hand columns rush towards each other at a velocity $v_L$ and $V_R$ respectively, they will slam and the pressure will rise at the point of impact to [Moody]

$$P = \frac{\rho a}{2g}(v_L + v_R)$$

P=pressure at water column collision, psi

ρ=fluid density, lb/in$^3$

a=speed of sound, in/sec

g=gravity, 386 in/sec$^2$

$V_L$=velocity of left column, in/sec

$V_R$=velocity of right column, in/sec

If one liquid column is stationary and the other column slams into it at velocity u, the pressure rise is half that predicted by Joukowski's equation for a water column slamming into a solid wall

$$P = \frac{\rho a}{2g} u$$

When a steady flow of liquid at a velocity $u_o$ is interrupted by a sudden valve closure, the upstream pressure will rise to the value predicted by Joukowski's equation. Downstream, the flow inertia may cause the liquid column to separate from the valve, forming a vapor pocket. The time period that it takes the water column to decelerate and start reversing itself to return towards the valve is [Moody]

$$\Delta t = \frac{\rho u_o L_R}{g(P_R - P_v)}$$

ρ=fluid density, lb/in$^3$

$u_o$=initial flow velocity, in/sec

$L_R$=length of downstream pipe to constant pressure discharge, in

$P_R$=constant pressure reservoir, psi

$P_v$=vapor pressure, psi

g=gravity, 386 in/sec$^2$

At the instant the reversed downstream column is ready to impact the valve, its velocity at impact is, interestingly, $u=-u_o$ and the pressure rise at impact downstream is the same as a Joukowski prediction $dP=\rho\, a\, u_o/g$.

### 9.2.4 Steam System Layout

There are rules of good practice to layout steam systems and reduce the risk of bubble collapse waterhammer:

(1) Know the starting and end points of the line and keep the layout as simple as possible, with east-west and north-south runs, with vertical or 45° off-vertical loops over roads and crossings.

(2) Anchor the line between expansion loops to balance the thermal expansion equally among consecutive loops.

(3) In process steam (up to approximately 400°F) keep maximum line growth within 5".

(4) Support line to avoid low points at middle of spans.

(5) Support line to avoid uplift off supports, in particular provide variable spring hangers at top of risers.

(6) Verify that pipe slope is maintained when steam piping expands.

(7) Keep supports simple, chose rods or one type of sliding shoes, with springs atop risers.

(8) Place steam traps and drains (free blows) at low points to eliminate condensate at startup and during operation.

(9) Heat-up steam lines using low flow through bypass with all drains open.

(10) Frequently inspect the proper function of steam traps.

(11) Repair insulation to avoid heat losses and formation of condensate.

(12) Locate condensate accumulation using thermocouples or thermography.

(13) Vent high points when draining low points.

(14) Properly size steam traps.

(15) Place steam traps at 100 ft to 200 ft distance (consult steam trap catalog).

(16) Use half the steam trap distance when steam flows uphill.

(17) Place drip legs to collect condensate at steam traps.

(18) Drain condensate from both sides of closed isolation valves.

(19) Slope lines to drain condensate.

(20) Avoid pockets that could trap condensate, such as low points in valve bonnets.

(21) Place relief valves on low-pressure side of pressure regulators.

(22) Size relief valves for fully open regulator or fully open bypass valve.

(23) Drain the trap discharge condensate.

(24) If water-steam mixing is intentional in certain processes, mix through a specially designed mixing valve.

(25) Periodically inspect the condition of pipe supports.

(26) Familiarize operators with the causes of waterhammer, its symptoms, and corrective actions should it occur.

(27) Avoid cracking open valves as a means of draining large quantities of condensate into steam.

(28) Be attentive to isolated metallic "banging" sounds coming from the pipe (small bubble collapse). They can eventually become large transients.

(29) Outdoors, regularly cut grass and growth around steam line.

## 9.3 NON-CONDENSABLE TWO-PHASE WATERHAMMER

### 9.3.1 Flow Regime

A two-phase pipeline or piping system contains liquids and noncondensable gases in a common stream. There could be more than one liquid (such as oil and water) and more than one gas (such as a mixture of natural gas and air). If the pipe was transparent, we would see that the liquid-gas two-phase flow can have one of several flow regimes, as illustrated in Figure 9-9.

Bubbly Flow: Bubbles of gas are carried along in the liquid, Figure 9-9(a). The fluid acts as an equivalent liquid with a reduced speed of sound.

Plug Flow: The bubbles of gas are sufficiently large to combine into plugs that are entrained with the liquid, at the liquid velocity, Figure 9-9(b).

Stratified Flow: The gas flows on top of the liquid, in the same direction but at higher velocity, Figure 9-9(c). The gas slips by the fluid. This slippage tends to slow-down the liquid and increase its pressure where it bulks up.

Wavy Flow: As the velocity increases, waves form at the surface of a stratified liquid, causing a wavy stratified flow, Figure 9-9(d). This flow regime reduces slippage between gas and liquid.

Slug Flow: The liquid waves are sufficiently high to entrap pockets of gas, forming alternating pockets of gas and liquid, Figure 9-9(e). This effect is called slugging. It causes large pressure pulses of pressure and can induce severe vibration, for example in offshore platform risers and separation facilities fed by a liquid-gas mixture from subsea wells. It is a flow regime to be avoided. Slug catchers may be installed to protect equipment from slugging loads and vibration.

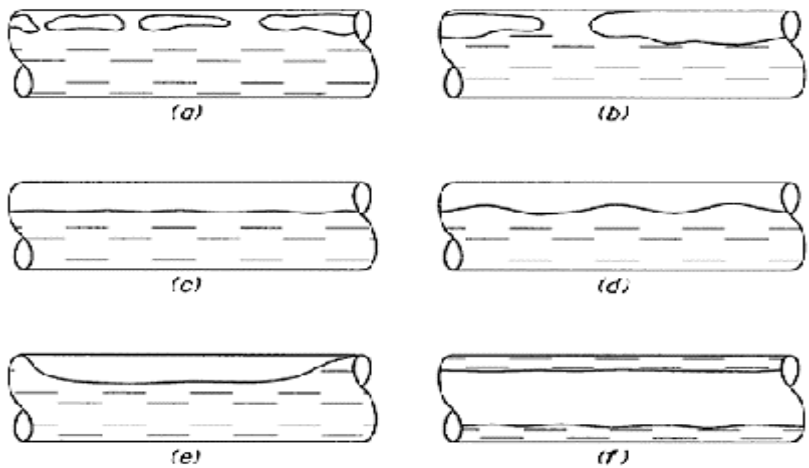


**Figure 9-9** Liquid-Gas Flow Regime

Annular Flow: The annular flow develops when the proportion of gas is large and the gas is at high velocity, Figure 9-9(f). The liquid is pushed out as an annulus against the inner pipe wall while the gas flows in the center of the pipe.

Today, the prediction of multi-phase flow regimes is done by computational fluid dynamics analysis, as well as various simplified models developed for this purpose [Oliemans]. The earliest hydrodynamic model for two-phase liquid-gas flow was developed in the 1950's; it had the advantage of showing the parameters that control the transition from one flow regime to another. In the early models the two-phase flow regime was predicted by plotting a point with ordinates $B_x$ and $B_y$ (the Baker parameters) on a flow regime map

$$B_x = 531 \frac{W_L}{W_v} \frac{\sqrt{\rho_L \rho_v}}{\rho_L^{2/3}} \frac{\mu_L^{1/3}}{\sigma_L}$$

$$B_y = \frac{2.16 W_v}{A\sqrt{\rho_L \rho_v}}$$

$W_L$=liquid flow rate, lb/hr
$W_V$=vapor flow rate, lb/hr
A=pie cross section area, $ft^2$
$\rho_L$=liquid density, $lb/ft^3$
$\rho_V$=vapor density, $lb/ft^3$
$\mu_L$=liquid viscosity, cP
$\sigma_L$=liquid surface tension, dyn/cm

Other maps have been developed to predict two-phase flow regimes. For example, the map by Spedding and Nguyen [Spedding] is based on the parameters

$$\sqrt{\frac{v_T}{\sqrt{gD}}}$$

$$\frac{Q_L}{Q_G}$$

$V_T$=average total velocity in the pipe
g=gravity
D=pipe inner diameter
$Q_L$=liquid volumetric flow rate
$Q_G$=gas volumetric flow rate

### 9.3.2 Analysis of Slug Flow

The reaction exerted by a slug of water as it discharges past an elbow is

$$F_X = F_Y = \frac{\rho A v^2}{g}$$

$F_x$=force along the incoming leg, lb
$F_Y$=force along the exit leg, lb
$F=F_X=F_Y$
ρ=liquid density, lb/in$^3$
a=velocity of sound in liquid-gas mixture, in/sec
A=pipe cross sectional area, in$^2$
v=slug velocity, in/sec
g=gravity, 386 in/sec$^2$
The velocity of the slug is approximately

$$v = \sqrt{\frac{(P_U - P_D)g}{\rho_L(1-\alpha)}}$$

v=slug velocity, in/sec
$P_U$=upstream pressure, psi
$P_D$=downstream pressure, psi
$\rho_L$=liquid density, lb/in$^3$
α=void fraction

Another concern with slug flow is related to multiphase pumps used to pump liquid and gas, such as raw mixtures of oil, gas and water, without having to separate the liquid and gaseous phase. The pumps can be sized for the gas-liquid mix, however a higher horsepower (higher torque) is needed when a slug of liquid reaches the pump. A costly solution would be to size the pumps for liquid flow, which means that they would run underpowered with a mixture of gas and liquid. A second solution would be to use a slug catcher. A third solution would be to detect the slug ahead of time and, upon detection, automatically reduce the pump speed to reduce the torque on the pump.

### 9.3.3 Trapped Air

Air pockets trapped in a liquid line will be forced along by the flow and may accumulate at high points in the line, this will reduce the flow area at those high points, and will force an increase in pump pressure to maintain the same flow rate. These air pockets may be vented at high points, but the liquid will rush behind the air, and as it enters the narrow vent pipe, will suddenly decelerate, causing a rapid drop in liquid velocity and possibly a waterhammer, as can be predicted from section 9.1.

To avoid these problems, the line should be vented during startup when it is being slowly and steadily filled from its low point. Air and vacuum release valves are used to exhaust air during filling operations and then close when approached by liquid, through the action of a buoyant float or the flow of water against a closing element. If reversible, the same valves could act as vacuum breakers when the pressure in the line drops below atmospheric pressure, to avoid column separation.

The presence of entrapped air in a piping system, even in small amounts, can cause serious complications when filling the line for hydrostatic testing. The volume of water $dV_T$ required to pressurize a line of internal volume V, containing a trapped volume of air nV, from an initial absolute pressure $P_a$ to a final absolute pressure $P_a+dP$ is

$dV_T=dV+dV_{add}=V\ dP\ [1/\kappa+n/(P_a+dP)]$

$dV_T$=total added volume of liquid, $in^3$

dV=added volume if system full of liquid (no air), $in^3$

$dV_{add}$=additional volume of liquid due to the presence of the air bubble, $in^3$

dP=pressure increase due to the addition of $dV_T$, psi

$\kappa$=bulk modulus of elasticity of liquid, Table 9-2.

$P_a$=initial pressure, psia

n=proportion of trapped air, volumetric

For example, to increase the pressure of a 4" sch.40 water filled line (bulk modulus $\kappa$=316,000 psi), 500 ft long (internal line volume 76,382 $in^3$) from atmospheric (14.7 psia) to 150 psi (164.7 psia), if the water initially contains n=1% trapped air, the volume of water needed is, first 76,382 $in^3\times$%=328 gallons to fill the line at atmospheric pressure, then to raise its pressure to 150 psi the volume needed would be

$dV_T=76{,}382\ (150)\ [1/316{,}000+0.01/164.7]\ (7.48052/12^3)$

$dV_T=0.16$ gal+3.0 gal

The first term, 0.16 gallons, is the volume of water needed to pressurize the water filled line from atmospheric pressure to 150 psi, if there were no air pockets. The second term is the additional volume of water needed to pressurize the line if there was a pocket of water 1% the volume of the line.

The air pockets and the water in the line are at the same pressure, but the energy contained in these air pockets at hydrostatic pressure (the TNT equivalent of the air pockets, Chapter 4) is much higher than the energy contained in the surrounding water at the same pressure. If a leak occurs where the air pocket is located, it could result in a violent explosion releasing a large energy, compared to the same leak in the water filled portion of the line that is at the same test pressure.

The sudden opening of a valve, or the sudden start of a pump, in a liquid filled line containing an entrapped air bubble causes liquid to rush towards the air bubble, compress the bubble and cause pressure fluctuations in the line. The volume of the air bubble $\Omega$, its pressure P, and the liquid flow velocity can be obtained by resolving a set of differential equations [Martin]

$d\Omega/dt=-Av$

$d\Omega/dt=-g\ (P-P_o)/L-fV^2/(2D)$

$dP'/dt=-(nP'/\Omega)\ (d\Omega/dt)$

$\Omega$=bubble volume at time t, $in^3$

$\Omega_o$=initial volume of air bubble, $in^3$

t=time, sec

A=pipe flow cross sectional area, $in^2$

v=flow velocity, in/sec

g=gravity, $in/sec^2$

P=gage pressure, psi
P'=absolute pressure, psi
$P_o$=absolute value of applied pressure, psi
n=polytropic exponent
f=friction coefficient
L=distance from the pressure source to the air bubble, in

To illustrate this effect, consider a 21 ft long air bubble trapped in a 3" water filled pipeline at atmospheric pressure. The line is suddenly brought to 27 psig pressure, for example by the start of a pump against a closed valve. Using a friction factor f=0.018 and a polytropic coefficient n=1.2, the solution of the differential equations indicates that the pressure in the trapped air will fluctuate, reaching a maximum of 87 psig, compressing the entrapped air from a 21 ft long bubble down to 4 ft. The compressed air bubble then pushes back the water column, reducing the air pressure, and causes a back and forth motion of the water with the corresponding pressure fluctuations.

## 9.4 STRESS ANALYSIS

The evaluation of adequacy of a piping system subject to postulated or actual waterhammer loads must address (a) the effect of overpressure, (b) the force induced stresses and movements in the system, and (c) the integrity of supports. The overpressure should be compared to the design code allowances for short overpressure conditions, and the pressure rating of fittings and components (Chapter 4). The stresses induced in the piping system due to the transient force imbalance should be evaluated as occasional loads and be limited to the corresponding code allowable stress

$$\frac{PD}{4t} + 0.75i\frac{M_A + M_B}{Z} < kS$$

P=operating pressure concurrent with the transient, psi
D=pipe outer diameter, in
t=pipe wall thickness, in
i=stress intensity factor
$M_A$=sustained moment (for example weight), in-lb
$M_B$=occasional moment (in this case waterhammer), in-lb
$Z$=pipe section modulus, $in^3$
k=factor, depends on applicable ASME code
S=ASME code allowable stress at transient temperature, psi

The allowable stress kS should be selected with a good understanding of the basis for the dynamic allowable stresses in the code [Slagis]. An allowable stress $kS=S_Y$ would be appropriate for one-time waterhammer loads in ductile piping systems. Even plastic deformation may be permitted by detailed stress and func-tionality analyses (Chapter 12). Pipe supports should be evaluated (a) for pipe displacements to make sure that the pipe will not disengage from its support, and (2) for stresses using the steel and anchor bolt design standards (Chapter 6).

## 9.5 REFERENCES

Andrews, P.B., et. al., Condensation Induced Waterhammer in Steam Supply System, ASME Pressure Vessel and Piping Conference, 1995.

Bjorge, R.W., Griffith, P., Initiation of Waterhammer in Horizontal and Nearly Horizontal Pipes Containing Steam and Subcooled Water, ASME Journal of Heat Transfer, 106, November, 1984.

Dickenson, T.C., Valves, Piping and Pipelines Handbook, Elsevier Advanced Technology, Oxford, UK.

Daugherty, R.L., Franzini, J.B., Fluid Mechanics with Engineering Applications, McGrawHill, 1965.

EPRI, Waterhammer Prevention, Mitigation, and Accommodation, Electric Power Research Institute report NP-6766, Palo Alto, CA, 1992.

EJMA, Standards of the Expansion Joint Manufacturers Association, Expansion Joint manufacturers Association, White Plains, NY.

Fuji, T., Waterhammer phenomena in a one-component two-phase bubbly flow, Proceedings of the ASME PVP conference, PVP-Vol. 360, 1998.

Lyons, J.L., Lyons' Handbook of Control Valves, Van Nostrand Reinhold, New York.

Martin, C.S., Entrapped Air in Pipelines, BHRA Fluid Engineering, Paper F2, Second International Conference on Pressure Surges, Cranfield, Bedford, England, September, 1976.

Martin, C.S. et. al., Pressure Pulse Propagation in Two-Component Slug Flow, Journal of Fluids Engineering, Transactions of the ASME, Series I, Vol. 101, No. 1, 1979.

McKetta, J.J., "Piping Design Handbook, M.Dekker Inc.

Moody, F.J., Introduction to Unsteady Thermofluid Mechanics, John Wiley & Sons, New York.

Oliemans, R.V.A, Multiphase Science and Technology for Oil/Gas Production and Transport, SPE Paper No. 27958, University of Tulsa Petroleum Engineering Symposium, Tulsa, 1994.

Parmakian, J., Waterhammer Analysis, Prentice Hall, New York, 1955.

Slagis, G.C., Basis of Current Dynamic Stress Criteria for Piping, Welding Research Council Bulletin 367, New York, 1991.

Spedding, P.L. and Nguyen, V.T., Regime Maps for Air-Water Two-Phase Flow, Chem. Eng. Sci, 35, 779–793, 1980.

Streeter, V.L., and Wylie, E.B., "Fluid Mechanics in Systems", Prentice Hall.

Thurston, R.H., Hoboken, New Jersey, article published in the Transactions of the American Society of Mechanical Engineers, 1883.

Van Duyn, et. al., Waterhammer Events Under Two-Phase Flow Conditions, ASME Winter Annual Meeting, December 10–15, 1989.

# 10
# Wind Design

## 10.1 WIND DAMAGE

On August 24, 1992, Hurricane Andrew passed directly over the Turkey Point power plant site in Florida, leaving in its wake a true testimony to the behavior of well-designed industrial facilities. The Turkey Point site consists of two fossil-fired units and two nuclear-powered units. The site saw wind speeds of 145 mph, with gusts of over 175 mph. The "class 1" nuclear structures sustained practically no damage. Other structures sustained some damage, although relatively minor in light of the extremely high wind speeds [INPO/NRC]:

(a) The plant lost offsite power for five days. The emergency diesel-generators automatically picked-up the essential loads. Two days after the storm hit, diesel fuel was being delivered to the site by truck.

(b) Offsite communications were lost for four hours.

(c) A 100,000-gallon water tower collapsed, destroying portion of the fire protection piping. The failure of the elevated water tank was attributed to a wind-borne missile that probably struck compression struts or tension rods causing the tank support column to fail.

(d) Part of the plant lighting was made inoperable.

(e) Electrical conduit failed when the containment vent stack lifted 3/8".

(f) Water pipes were bent and dented when struck by flying debris.

(g) Several sheet metal buildings were damaged.

Interestingly, this damage primarily affected equipment and distribution systems. It brings up the importance of emergency procedures that should consider equipment failures, such as loss of emergency power and electrical distribution. There are primarily three aspects to the wind design of outdoor piping systems and pipelines: (1) The wind pressure, (2) the vibration of the pipe due to vortex shedding around the pipe, and (3) for very high winds and tornadoes, the impact of windborne missiles.

## 10.2 WIND PRESSURE

The design of outdoor piping and pipelines for wind pressure typically follows the provisions of civil structural or construction codes [ASCE 7, IBC]. The methods were originally developed for buildings and civil structures, and later expanded to apply to large equipment and distribution systems such as piping systems on racks and pipelines. The lateral drag pressure on the pipe due to wind speed, called velocity pressure, is

$q_z = 0.00256\ K_z\ K_{zt}\ V^2\ I$

$q_z$=velocity pressure, lb/ft$^2$

$K_z$=velocity pressure exposure coefficient

$K_{zt}$=topographic factor

V=wind speed, mph

I=importance factor

The various factors are defined in design codes [ASCE 7], and are briefly described here. The velocity pressure factor $K_z$ is a function of the height of the pipe above ground and its location (city, open country, seashore, etc.). $K_z$ varies from 0.32 at ground level in a city to 1.22 at 40 ft above ground at seashore. The topographic factor $K_{zt}$ is a function of the pipe location relative to hilly terrain, and varies between 1 and 2. The wind speed V is typically selected from building code wind maps. The ASCE-7, 1995, maps provide basic wind speeds based on 3-second gust, for an elevation of 33 ft above ground in open country. The mapped wind speeds correspond to a 2% probability of exceedance in 50 years. In other words, the chance that the mapped wind speeds will be exceeded is less than 2% in a 50-year period. Values of V are in the order of 140 mph (coastal eastern United States) to 80 mph (western U.S.). The importance factor I depends on the consequence of failure of the facility or system, and varies from 0.87 to 1.15. The designer must consult the applicable design code edition when selecting the velocity pressure factors and design wind speed because they are periodically readjusted to reflect most recent wind data.

The design wind force per foot of pipe is [ASCE 7]

$$F=q_z\, G\, C_f\, A_f$$

F=design wind force, in the wind direction, per foot of pipe length, lb/ft

$q_z$=velocity pressure, lb/ft$^2$

G=gust factor

$C_f$=force coefficient

$A_f$=area of pipe cross section perpendicular to wind, per foot of pipe length, ft$^2$/ft

The gust factor G depends on the location of the piping system, and is generally around 0.80. The force coefficient $C_f$ is a function of (a) the height of the pipe h divided by its diameter D, (b) the smoothness of the pipe, and (c) the product $D(q_z)^{0.5}$ where the pipe diameter D includes the insulation thickness. $C_f$ varies from 0.5 for a smooth profile (typical of piping systems) at ground, to 0.7 for an elevation at 25 times the diameter. For large $D(q_z)^{0.5}$ and large elevations, $C_f$ can be as large as 1.2.

The design wind force can therefore be calculated in a straightforward manner. This force will then be applied as a uniform lateral load along the pipe spans that are perpendicular to the wind direction. If the pipe is suspended or is placed on sliding plates or rollers, the pipe will have to be guided laterally against excessive bending caused by the lateral wind force. If the pipe is resting on a flat steel member, the pipe-support friction force $\mu w$ (where $\mu$ is the coefficient of friction and w the linear weight of the pipe) may be larger than the lateral wind force. In this case, friction alone will keep the pipe from sliding sideways. It is however prudent not to rely solely on friction and to guide the pipe laterally every third or fourth weight support. For a hot line, the system flexibility will have to be verified where lateral guides are added.

## 10.3 VORTEX SHEDDING

As steady winds flow across a pipe span, they cause vortices behind the pipe. The vortices generate pressure fluctuations that cause the pipe to oscillate in the direction of the wind (drag) and in the direction perpendicular to the wind (lift). The vortex turbulence has a dominant frequency given by

$f_{vs}=n\ S\ v/D$

n=1 for lift, 2 for drag

$f_{vs}$=dominant vortex shedding frequency, Hz

S=Strouhal number

v=wind velocity, in/sec

D=pipe outside diameter (including insulation), in

The Strouhal number for wind vortices around a cylinder is in the order of 0.2 [Simiu]. The pipe is therefore subjected to a sinusoidal lift force per unit length, of frequency $f_{vs}$ with amplitude F given by [Simiu]

$F=C_L\ \rho\ V^2\ D/2$

F=lift force per unit length of pipe, lb/in

$C_L$=lift coefficient

ρ=air mass density, $lbm/in^3$

V=wind velocity, in/sec

D=insulated pipe outside diameter, in

There is a critical velocity $v_{crit}$ at which the vortex shedding frequency will be equal to the natural frequency (first mode) of the pipe span $f_{pipe}$. In practice, within approximately 20% of this wind speed, the pipe span will resonate with the wind vortices resulting in significant, visible movement of the line, unless the line is very stiff (supported on short spans). In order to avoid resonant cross-flow vibration due to vortex shedding, the pipe span natural frequency $f_{pipe}$ should differ by at least 20% from the wind vortex shedding frequency $f_{vs}$. This can be achieved by stiffening the line (placing more supports) or by using tuned vibration absorbers (a mass, spring and damper device) at mid-span [Hart, Norris]. A detailed analysis is in order for critical lines such as gas or oil pipelines in open windy areas.

## 10.4 WIND-BORNE MISSILES

The study of actual windstorm damage provides a logical starting point for understanding the types of objects that can become wind-borne missiles [McDonald]. Sheet metal or wood buildings and structures are toppled by wind forces but do not readily fly. Flying missiles typically include wood planks, poles and pipes, and, in tornadoes, steel beams. Automobiles and trucks are also considered missiles that can roll and tumble in high winds. Three types of missiles have been considered design basis tornado missiles [McDonald]: a 2×4 timber plank, a 3-in diameter steel pipe, and an automobile. For straight winds and hurricanes (little vertical speed compared to tornadoes) a 2×4 timber

plank weighing 15-lb is considered a missile, with a design impact speed in the order of 100 mph from ground level to a height of 200 ft.

The effects of missiles on structures, walls and equipment are experimentally studied using air cannons that propel missiles into a target at the desired speed. In addition, simple experiments have been conducted by applying a sudden blow to the side of steel pipes [McLean]. The data indicates that welded steel pipes struck by wind-borne missile will deform (ovalize and dent locally). A gouge (sharp cut) will form at the point of impact unless the pipe is insulated. Rupture or leakage is likely if the struck pipe contains mechanical joints (threaded, swaged, grooved, etc.) that tend to pull open under impact. Detailed analyses of the effects of impact can be conducted by hand calculations or finite element analysis [Stronge, Jones].

## 10.5 REFERENCES

ASCE 7, Minimum Design Loads for Buildings and Other Structures, American Society of Civil Engineers, New York.

Hart, J.D., et. al., Mitigation of Wind-Induced Vibration in Arctic Pipeline Systems, Proceedings of the 11th International Conference on Offshore Mechanics and Arctic Engineering.

IBC, International Building Code, International Code Council, Virginia.

INPO/NRC, Effect of Hurricane Andrew on the Turkey Point Nuclear Generating Station from August 20–30, 1992, report sponsored by the Institute of Nuclear Power Operations and the U.S. Nuclear Regulatory Commission, March 1993.

Jones, N., Structural Impact, Cambridge Press.

McDonald, J.R., Rationale for Wind-Borne Missile Criteria for DOE Facilities, Lawrence Livermore National Laboratory report UCRL-CR-135687, S/C B505188, September 1999.

McLean, J.L., Beazley, P.K., Manhardt, A.H., The Dynamic Deformation of Piping, Welding Research Council Bulletin 321, January 1987, Welding Research Council, New York.

Norris, M., et. Al., Implementation of Tuned Vibration Absorbers for Above Ground Pipeline Vibration Control, Proceedings of the IPC 2000, ASME International Pipeline Conference, October 1–5, 2000, American Society of Mechanical Engineers, new York.

Simiu, E., Scanlan, R.H., Wind Effects on Structures, Wiley-Interscience.

Stronge, W.J., Impact Mechanics, Cambridge Press.

# 11
# Seismic Design and Retrofit

## 11.1 THE SEISMIC CHALLENGE

The challenge in seismic design of new equipment or seismic retrofit of existing equipment is to do all that is necessary, but only what is necessary. It is essential to avoid doing too little or doing too much. An example of doing too little starts with an owner who specifies the seismic requirement as a one-liner: "Seismic design in accordance with building code", with no further thought as to the scope and purpose of the seismic effort (what is to be seismically designed and why?); and a designer with no experience who in turn would follow a cook book [ASHRAE, NFPA 13, SMACNA], and place some sway braces here and there, with little understanding of seismic design. An example of doing too much is that of a designer who would spend an inordinate amount of time and effort in seismic analysis or testing, taking a purely academic or unnecessarily complex approach to seismic qualification, or concluding that a prohibitive amount of new hardware and construction are needed to qualify the system.

The responsibility for competent seismic engineering (doing all that is necessary, but only what is necessary) starts with the owner who must clearly define the objectives of the seismic design or retrofit effort. To help in this process, section 11.2 provides the outline for a project specification that defines the rules and nomenclature for the seismic qualification of piping systems.

## 11.2 SEISMIC SPECIFICATION

Preparing a Project Specification is the first step to a seismic design or qualification activity. It forces the owner to think through and clearly specify the scope of work. It lets the designer determine the level of effort with a sound understanding of the scope of work.

### 11.2.1 Project Specification

Before all else, the owner should prepare a Project Specification, which specifies, as a minimum

(a) The scope and boundaries of systems to be seismically designed or retrofitted.

(b) The applicable design and construction code (such as ASME B31.1, B31.3, B31.4, or B31.8), and any exceptions to the Code.

(c) The required seismic function of the piping system (position retention, leak tightness, or operability).

(d) The free field seismic input for the design basis earthquake, by reference to the applicable building code, or as established through a site-specific seismicity study or by regulation.

(e) The responsibility for developing the in-structure seismic response spectra, if the piping is not at grade level.

(f) The operating and design conditions concurrent with the seismic load.

(g) The responsibility for the evaluation of seismic interactions.

The owner or the designer should be aware of federal, state or local codes or regulations that impose explicit requirements and methods for seismic qualification of piping systems, these would replace or supplement the provisions in this section.

### 11.2.2 Seismic Input

The seismic input excitation may be defined as horizontal and vertical seismic static coefficients, or as horizontal and vertical seismic response spectra. The seismic acceleration at an upper floor in a building is larger than at ground. The increase in seismic excitation from ground level to floor level is the in-structure amplification. The in-structure amplification can be determined by existing seismic standards, such as the in-structure seismic coefficients of the International Building Code (IBC), or by a facility specific dynamic evaluation. The IBC accounts for in-structure amplification through a factor 1+2 z/h, where z is the elevation in the building and h is the total building height. This factor implies that the maximum in-structure amplification, the acceleration at the top floor, is (1+2 h/h) =3, or three times the ground acceleration.

The damping value to be used in the seismic analysis of piping systems and equipment subject to large earthquakes should be 5%.

### 11.2.3 Seismic Qualification

The seismic qualification requirements differ depending on the seismic function of the piping system: operability, leak tightness, or position retention (refer to definitions in section 11.2.8).

#### *11.2.3.1 Operability*

The seismic qualification of piping systems that must remain operable during or following the design basis earthquake must be established by static or dynamic analysis or by testing. The seismic qualification of piping systems for operability must demonstrate the seismic adequacy of the piping itself, the pipe supports and their attachment to the building structure, and the equipment and components within the scope of seismic qualification. Table 11-3 is a summary of the applicable criteria that follow.

Pipe Stress—The elastically calculated longitudinal stresses due to the design basis earthquake, calculated by static or dynamic analysis, and the concurrent sustained operating loads should comply with the following equation

$PD/(4t)+0.75\ i(M_w+M_S)/Z<1.5\ S_Y$

P=normal operating pressure, psi

D=pipe outer diameter, in

t=nominal pipe wall thickness, in

i=stress intensity factor (from the applicable ASME B31 Code)

$M_W$=resultant moment due to weight, in-lb

$M_S$=resultant seismic moment amplitude (inertia and anchor motion), in-lb

Z=pipe section modulus, $in^3$

$S_Y$=minimum specified material yield stress at operating temperature, psi

Where the elastic longitudinal stress limit of $1.5S_Y$ can not be met, the piping system may be qualified by more detailed analysis techniques [WRC 379].

The maximum permitted seismic loads and displacements on unlisted pipe components (components for which no stress intensification factor is listed in the applicable ASME B31 Code) should be based on limits established by test or analysis, with a safety factor of 3 against leakage.

We will see in Chapter 12 that at very high strain rates, in the order of 1 $sec^{-1}$ and above, the yield and ultimate strength of steel increase compared to the classical properties established by ASTM quasi-static testing. Is this increase in strength relevant to understanding seismic behavior? In other words, are strain rates in the order of 1 $sec^{-1}$ achieved in earthquakes? Consider, for example, the strain rate for the dynamic response of a structure or system to earthquakes. With a peak spectral acceleration around 10 Hz, the structure will cycle through its maximum strain amplitude 4×10 Hz=40 times per second. In a large earthquake, if the steel reaches a plastic strain of 20% (at the threshold of ductile tensile failure for mild carbon steel), the strain rate would be 20% strain 40 times a second=0.20 strain in 1/40 of a second=8 $sec^{-1}$. On the other hand, if the structure remains elastic, the strain amplitude would be less than 0.2% and the strain rate less than 0.08 $sec^{-1}$. Therefore, for small earthquakes, when the structure or component respond in the elastic regime, there is practically no increase in strength due to strain rate. But in large earthquakes, if the structure or component respond well within the plastic regime, at onset of tensile failure, there is an increase in tensile strength of the material. This increase alone is however misleading, for two reasons: (1) the fact that maximum strain is achieved at the instant of zero strain rate ($d\varepsilon/dt=0$ when $\varepsilon$ is maximum), and (2) the drop in fracture toughness under dynamic load, more than strength, controls the behavior of actual materials that inevitably contain flaws.

Active Equipment—The operability of active equipment (defined in section 11.2.8) during or following the design basis earthquake may be established by testing, analysis where applicable, or by similarity of earthquake experience as defined in seismic qualification standards [ICBO AC156, IEEE-344, IEEE-382 or ASME QME-1].

Static Equipment—The adequacy of static equipment should be established in accordance with the applicable design and construction code, such as the ASME Boiler and Pressure Vessel Code for pressure vessels, API 650 for oil storage tanks, and AWWA D100 for water storage tanks.

Pipe and Equipment Restraints—Seismic qualification must establish the seismic adequacy of pipe supports and restraints and their attachment to the building structure. The seismic load on each pipe support and restraint should be calculated by seismic static or dynamic analysis, and their seismic adequacy must be determined in accordance with a structural design and construction code, such as AISC or AISI for steel members, and IBC or ACI for concrete anchor bolts (Chapter 6).

### *11.2.3.2 Leak Tightness*

The requirements for seismic qualification of piping systems that must remain leak tight during or following the design basis earthquake vary with pipe size and the magnitude of seismic input (Table 11-3). For pipe larger than 2" nominal pipe size (NPS) and for a design basis earthquake with a peak spectral acceleration (PSA, defined in section 11.6.1) larger than 0.3g, it is recommended that the seismic design and retrofit requirements for leak tightness be identical to the operability requirements of section 11.2.3.1, except for the operability requirements of active equipment, which are not applicable. For piping 2" NPS and smaller, or where the PSA is below 0.3g, the position retention rules of 11.2.3.3 may apply for leak tightness, with the additional requirement that the loads imposed on non-welded and non-flanged pipe joints (for example swage fittings, groove couplings, etc.) be within vendor limits. Table 11-3 is a summary of the applicable criteria.

### *11.2.3.3 Position Retention*

Pipe Stress—The seismic qualification of piping systems that must retain their position, but need not be leak tight or perform a function, may be established by sway bracing following standard support and restraint spacing criteria.

One possible spacing criterion consists in spacing weight supports in accordance with the support spacing of ASME B31.1, reproduced in Table 11-1, and placing a lateral seismic support approximately every 40 ft and a longitudinal support approximately every 80 ft [MSS-SP-127]. A lateral support within 2 ft of an elbow or bend may provide longitudinal seismic restraint to the pipe run passed the elbow or bend.

**Table 11-1** Spacing of Weight Supports [ASME B31.1]

| NPS | Water | Gas |
|---|---|---|
| 1 | 7 | 9 |
| 2 | 10 | 13 |
| 3 | 12 | 15 |
| 4 | 14 | 17 |
| 6 | 17 | 21 |
| 8 | 19 | 24 |
| 12 | 23 | 30 |
| 16 | 27 | 35 |
| 20 | 30 | 39 |
| 24 | 32 | 42 |

Pipe and Equipment Supports—The seismic qualification of piping systems for position retention must establish the seismic adequacy of the pipe supports and their attachment to the building structure. The seismic load on each pipe support should be calculated by

seismic static or dynamic analysis, and the seismic adequacy of supports and anchorage for position retention should be demonstrated with a safety factor of 2 or more against failure modes that could cause loss of position.

The permanent deformation of supports is acceptable in this case, provided it does not cause the pipe to disengage and fall off.

### 11.2.4 Material Condition

The seismic retrofit of existing piping systems should take into account the material condition of the system. Where corrosion or environmental cracking are suspected, the piping should be inspected by non-destructive volumetric techniques. The quality of construction and the maintenance condition of the system should be inspected in the field, and the maintenance record of equipment and components should be investigated with the facility engineer to assess their adequacy, operability and structural integrity.

### 11.2.5 Interactions

All seismically qualified piping systems should be evaluated for seismic interactions. Credible and significant interactions should be identified and resolved by analysis, testing or hardware modification, as described in section 11.9.

### 11.2.6 Documentation

The designer should prepare a Qualification Report, certified by a Professional Engineer experienced in the field of piping systems design and construction, and in seismic qualification. The Qualification Report should include, as a minimum:

(a) Drawing, sketches and (for existing systems) photographs, showing the scope of work.

(b) Final pipe support arrangement.

(c) Calculations showing design input (acceleration, static force, or response spectra) and code compliance for piping, equipment, and supports.

(d) Documentation of qualification of equipment operability where applicable.

(e) Drawings for new or modified supports, with dimensions, weld and anchor bolt details, bill of materials, and information necessary for material procurement and construction.

### 11.2.7 Maintenance

The owner is responsible for maintaining the configuration of the seismically qualified piping system. In particular, changes to layout, supports, components or function, as well as material degradation in service must be reconciled to verify the continued seismic adequacy of the system.

### 11.2.8 Definition of Common Terms

Active equipment (also referred to as dynamic equipment): mechanical or electrical equipment or components that must perform an active function (involving moving parts or signals) during or following the design basis earthquake. Examples of active mechanical equipment include valve operators, pumps, compressors, etc. that must change state (open, close, start-up, shutdown, throttle, etc.) during or following the earthquake.

Design basis earthquake: the level of earthquake for which the system must be qualified to perform a seismic function (position retention, leak tightness or operability).

Free field seismic input: the facility's seismic input (typically static coefficients or acceleration response spectra) in the free field, away from the influence of structures.

In-structure seismic response spectra: the seismic excitation (typically static coefficients or acceleration response spectra) in the various buildings and structures, at grade or floor elevations.

Leak tightness: the ability of a piping system to not leak during and after an earthquake.

Operability: the ability of a piping system to deliver and control flow during or after an earthquake.

Peak spectral acceleration (PSA): the maximum acceleration value of a seismic response spectrum.

Position retention: the ability of a piping system not to fall or collapse in case of earthquake.

Seismic response spectra: a plot or table of accelerations, velocities or displacements versus frequencies or periods (typically, for piping systems, accelerations vs. frequencies), for each of three orthogonal directions (typically east-west, north-south, and vertical), specified for one or several damping values. An example of seismic response spectrum is shown in Figure 11-1.

Seismic function: the ability of a piping system, as specified by the owner in the Project Specification, to either retain its position (position retention), remain leak tight (leak tightness), or perform a function (operability).

Seismic interactions: spatial interactions (such as seismically induced falling or swinging of overhead structures, or spray from adjacent ruptured or leaking lines) or system interactions (such as unintended seismically induced signals to components, or loss of contents from the rupture of an un-isolable branch line) that could affect the seismic function of the piping system. Credible interactions are interactions that could occur (for example a tall block wall may collapse and hit the piping system). Significant interactions are interactions that, should they occur, would affect the seismic function of the piping system (for example the fall of a small instrument on a large pipe may be credible but not significant, while the fall of a block wall on the same pipe is significant).

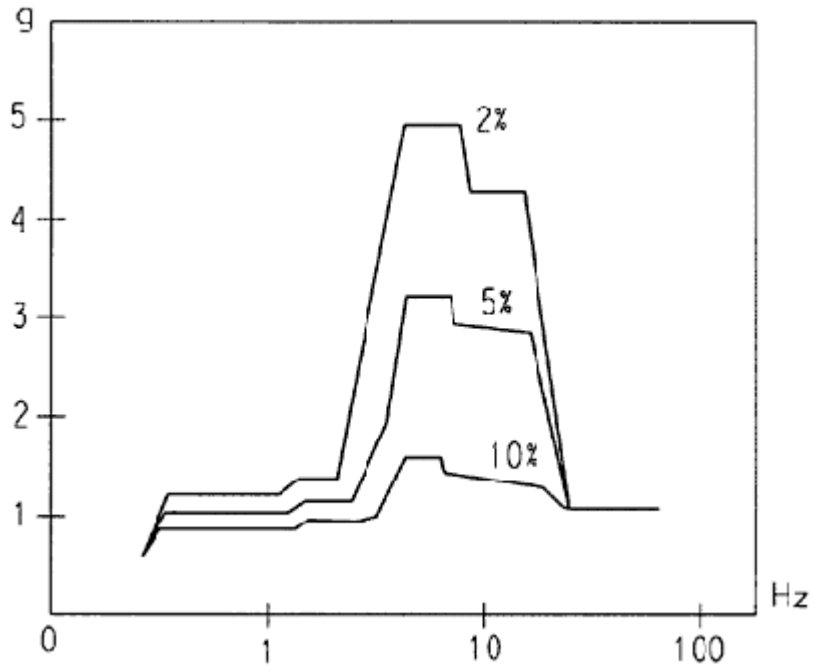


**Figure 11–1** In-Structure Response Spectra, 2%, 5% and 10% Damping Acceleration (g) vs. Frequency (Hz)

Seismic qualification: the activities necessary to demonstrate that the system can perform its seismic function in case of design basis earthquake. Seismic qualification may be achieved by static or dynamic analysis, testing, or use of earthquake experience data in accordance with IEEE 344 or ASME QME-1.

Seismic retrofit: the activities involved in evaluating the seismic adequacy of an existing piping system and identifying the changes or upgrades required to seismically qualify the system.

Seismic static coefficient: an acceleration value to be applied to the piping system in each of three directions (typically two horizontal directions, east-west and north-south, and a vertical direction).

Static equipment: mechanical equipment or component that does not perform an active function (involving moving parts) during or following the design basis earthquake. Examples of static equipment include storage tanks, pressure vessels, heat exchangers, and manual valves.

Sway bracing: the selection and placement of lateral and longitudinal restraints (braces) to brace the pipe against excessive lateral and longitudinal movement during a design basis earthquake.

## 11.3 RULES OF GOOD PRACTICE

In order to make the right design or retrofit decisions, and before proceeding with seismic analysis or testing, the designer should be familiar with lessons learned from seismic tests [Slagis] and real earthquakes to recognize and avoid features and conditions that caused failure.

Anchor motion—Many pipe failures in earthquakes are due to sliding or rocking of unanchored equipment, vessels and tanks to which the pipe is attached, or to differential movement of two building structures or floors spanned by the pipe. Some building codes advocate the use of flexible assemblies to decouple the seismic displacement of one side of a piping system from the other side, for example when crossing building joints.

Stiff branch lines—When a large pipe header is suspended on rods, it will tend to sway sideways during an earthquake. This would cause branch lines attached to the header to snap if they are rigidly tied to the building.

Inertia—The seismic load will force the pipe to sway sideways and, for large earthquakes, to uplift off its supports. It is therefore necessary to brace the pipe against sideway swaying and, for large earthquakes, to provide vertical seismic restraints. A preliminary bracing scheme, prior to proceeding with computer analysis for final design, would consist in placing lateral horizontal and vertical bracing every third or fourth weight support, leading to seismic spans (distance between lateral braces) three to four times those in Table 11-1. Other preliminary spacing of seismic supports (also referred to as sway bracing) are provided in MSS-SP-127 and NFPA-13. For very long horizontal pipe spans, such as encountered in straight pipe racks, an axial support should be added to restrain the longitudinal movement of the pipe. Where leak tightness or function are required by the Project Specification, it is advisable to brace the pipe close to equipment nozzles to limit the load applied by the pipe on the equipment. Next to load sensitive equipment it may be necessary to design an anchor (a support that resists loads in all degrees of freedom, Chapter 6).

Valve operators—Large valve operators must be seismically qualified if the valve has to operate following the earthquake [IEEE 344, IEEE 382, ASME QME-1].

Material conditions—Heavy corrosion or environmental cracking could cause a pipe to fail during an earthquake.

Construction quality—Poor weld or joint assembly, poor maintenance (missing bolts, deformed or poorly engaged pipe supports, loose concrete anchor bolts, etc.) could cause a piping system to fail during an earthquake.

Pipe supports—Undersized pipe support members or welds, and supports that rely on friction (such as C-clamps attached to I-beams) may fail or slide off during an earthquake.

Interactions—During an earthquake, a piping system may fail as a result of the collapse of a concrete block wall or the fall of a large overhead component. It is therefore essential that any seismic design or retrofit effort include a review and assessment of seismic interactions. It is not uncommon for the upgrade of seismic interactions (such as bracing block walls, or tying-down a suspended ceiling, or anchoring a tank) to be costlier than the seismic design and construction of the piping system itself. Section 11.9 provides further guidance regarding seismic interactions.

## 11.4 SEISMIC ANALYSIS TECHNIQUES

There are two important decisions to be made regarding seismic analysis techniques: the seismic input and the type of analysis. The seismic input may either be a static coefficient or a dynamic input. In either case, the seismic input is obtained, as described later in this section, from building code seismic maps or from a detailed geotechnical investigation of the site seismicity. The static coefficient is typically a single horizontal acceleration value and a single vertical acceleration value, specified as a fraction of "g". For example the seismic input may consist of a horizontal static coefficient $a_H$=0.3g and a vertical static coefficient $a_v$=0.2g. The dynamic input is typically a horizontal and vertical acceleration response spectrum, which is a plot of acceleration or velocity or displacement against frequency or period (typically, acceleration vs. frequency is the most common form of seismic response spectrum, as illustrated in Figure 11-1). Another type of seismic input would be a time history, which is a curve of accelera-tion or velocity or displacement vs. time. But the complexity and sensitivity of the time-history analysis technique is seldom justified and very rarely used in seismic design and retrofit.

Having established the seismic input as a static coefficient or a response spectrum, the type of analysis may be selected as (a) a "cook-book", (b) a static hand calculation, (c) a static analysis of a piping model, or (d) a computerized response spectrum analysis of a piping model.

(a) In a cook-book approach, the designer selects seismic support locations at fixed intervals, following a recipe, for example: a lateral sway brace is placed every 40 ft along the pipe and a longitudinal restraint every 80 ft. The braces may be pre-designed based on the specified spacing. The technique has the great advantage of simplicity, but has two important drawbacks: first, the cook-book methods tend to be conservative and over-predict the number and size of seismic supports. Second, cook books can be so simple that they may be used by engineers who have little, if any, understanding of piping systems and seismic design. The dangers of this situation are obvious.

(b) With a hand calculation technique, the pipe is divided into individual spans or into a series of simple U or Z configurations. The seismic load is applied as a lateral force distributed along the span, and bending stresses and support reactions are calculated using beam formulas. This technique was quite useful in the 1960's and early 1970's. However, with the advent of interactive PC-based piping design software, the system analysis techniques described in (c) or (d) are recommended as more precise and faster than the hand calculation techniques. The hand calculation techniques are still useful to provide an intuitive means to predict and interpret the output of a computer analysis.

(c) The third analysis technique consists in preparing a piping model of the system, using commercially available piping design software (the use of general finite element analysis software is not recommended in piping design, except in the rare case of elastic-plastic analysis). The seismic coefficient is applied statically and uniformly as a distributed force to the whole model in each of three directions (typically east-west, north-south and vertical). The output consists of the full distribution of seismic movements, stresses and support loads in the system.

(d) The computerized response spectra analysis technique is similar to the static technique described in (c) above, but the seismic input is a response spectrum in each of three directions (typically east-west, north-south and vertical), as illustrated in Figure 11-

1. The movements, loads and stress results are more precise and usually (but not always) lower than the static analysis method. However, since the analysis is frequency dependent, the piping model needs to be more accurate to properly reflect the system's natural frequencies.

## 11.5 SEISMIC INPUT BASED ON IBC

In many cases, the seismic input is based on the rules of the International Building Code (IBC) or ASCE 7. This section summarizes the IBC method and is based on the 2000 edition of the code. The user is cautioned to use the latest applicable edition of the Building Code since the equations, coefficients and parameters do change. The International Building Code provides two types of input, a static coefficient for static analysis, or a seismic response spectrum for dynamic analysis. The IBC method for equipment and piping consists of three parts: (1) site ground motion, in which a site acceleration is defined based on seismic maps and soil characteristics, (2) seismic load for equipment and piping located inside a structure or on racks, and (3) seismic load for equipment and piping located at grade, for example in the plant yard.

The site ground motion at the facility is first determined by the following nine steps:

Step-1: in this example, the site ground motion will be selected from the IBC seismic maps, and not from a site-specific seismicity study.

Step-2: to obtain the IBC site ground motion, the facility location is first placed on the IBC map (IBC Figures F1615(1) to (10)), and the mapped maximum considered earthquake spectral acceleration (MCESA) is read from the contour intervals as

$S_S$

$S_1$

$S_S$=MCESA at short period, and 5% damping in a site class B.

$S_1$=MCESA at 1 sec, and 5% damping in a site class B.

Step-3: the soil characteristics of the site are determined as: hard rock, dense clay, sand, etc., and the shear wave velocity $v_s$ is estimated.

Step-4: the soil is classified as class A to E, according to IBC 1615.1.1.

Step-5: the site coefficients $F_A$ and $F_v$ are determined from IBC Tables 1615.1.2(1) and (2), given the site class and $S_S$ and $S_1$

Step-6: the mapped spectral acceleration for short period $S_M$S and the mapped spectral acceleration for 1-second period $S_{M1}$ are calculated as

$S_{MS}=F_a S_s$

$S_{M1}=F_V S_1$

Step-7: the design spectral response accelerations for short period and 1-second are calculated as

$S_{DS}=(2/3) S_{MS}$

$S_{D1}=(2/3) S_{M1}$

Step-8: two reference spectral periods are defined as

$T_o= 0.2 S_{D1}/S_{DS}$

$T_S=S_{D1}/S_{DS}$

Step-9: the design response spectrum of the facility, at 5% damping, can now be traced. It consists of three regions. For frequencies between 0 and $f_s$ ($1/T_S$) the acceleration varies as $S_{DS}$ ($f/f_s$). From $f_s$ to $f_o$ ($1/T_o$) the acceleration has the peak spectral value $S_{DS}$; from $f_o$ to infinity, the acceleration is $0.6S_{DS}$ ($f_o/f$)+$0.4S_{DS}$. An infinitely stiff component will experience a zero period acceleration (ZPA) of 0.4 SDS. This IBC response spectrum is shown in Figure 11-2.

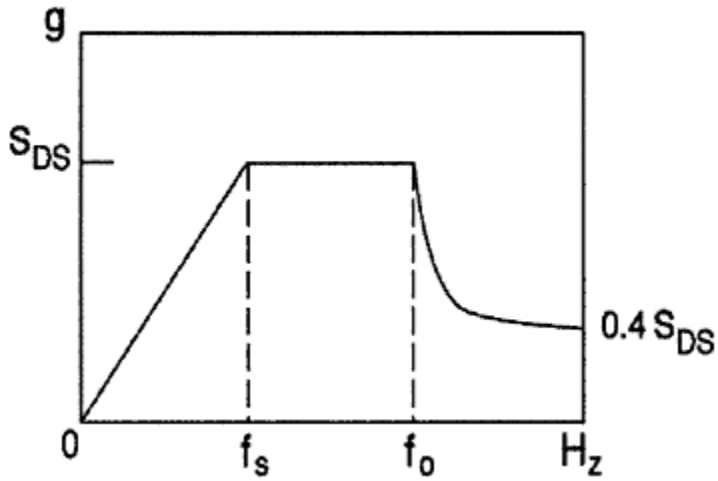


**Figure 11-2** The IBC Horizontal Seismic Response Spectrum Acceleration (g) vs. Frequency (Hz)

We have just developed the seismic input at ground level. We will now develop the seismic input applied to a piping system contained inside a structure (building or steel frame structure), in six steps:

Step—1: based on the consequence of failure of the system (failure effect), the system is assigned a Seismic Use Group I, II or III (IBC 1616.2), and an importance factor I=1.0 or 1.5 (IBC 1621.1.6).

Step—2: given the Seismic Use Group (SUG I, II or III) and the values of $S_{DS}$, $S_{D1}$ and $S_1$, the system is assigned a Seismic Design Category (SDC) A to F (IBC 1616.3). The extent of seismic design and qualification will increase from SDC A to SDC F.

Step—3: at this point, according to IBC (IBC 1621.1.1), several types of systems or components can be exempted from seismic design, as shown in Table 11-2.

**Table 11-2** Exemption from Seismic Design [IBC]

| SDC | I | W | FC | H |
|---|---|---|---|---|
| A, B | any | any | any | any |

| | | | | |
|---|---|---|---|---|
| C | 1.0 | any | any | any |
| D, E, F | 1.0 | 20 lb | Yes | any |
| All | 1.0 | 400 lb | Yes | 4 ft |

Nomenclature:
SDC=seismic design category
I=importance factor (1.0 or 1.5)
W=maximum weight, component below this weight can be exempted.
FC=distributed systems (piping, HVAC, conduit) with flexible connections
H=maximum height above floor, component below this height can be exempted.

Step—4: the horizontal seismic load applies separately for the longitudinal and lateral directions, it is given by $F_P$ where (IBC 1621.1.4)

$0.3\ S_{DS}\ I\ W \leq F_P = [0.4\ a_P\ S_{DS}\ W\ I/R_P]\ (1+2\ z/h) \leq 1.6\ S_{DS}\ I\ W$

$S_{DS}$=design spectral acceleration for short period, g
I=importance factor (1.0 or 1.5)
W=weight, lb
$F_P$=horizontal load, lb
$a_P$=component amplification factor
$R_P$=component response modification factor
z=height of attachment to structure, ft
h=height of structure, ft

The component amplification factor $a_P$ is meant to amplify the zero period acceleration $0.4\ S_{DS}$ to account for dynamic coupling between the structure and the flexible piping system. It varies from 1.0 to 2.5. In IBC 2000 it is 1.0 for piping systems. The component response modification factor $R_P$ accounts for the ductility (ability to deform and absorb plastic energy) and redundancy (ability to redistribute load) of the piping system. It varies from 1.0 to 5.0. In IBC 2000 it is 1.25 for low deformability piping systems, 2.5 for limited deformability piping system, 3.5 for high deformability piping systems.

Step—5: the effect of the horizontal seismic load $F_P$ (applied separately in the lateral and longitudinal direction) is added to the effect of the vertical seismic load Fv given by (IBC 1617.1.1, 1621.1.4)

$F_V = 0.2\ S_{DS}\ W$

$F_v$=vertical component of seismic load

The total seismic load is therefore the horizontal load $F_P$ plus the vertical load $F_v$. This is a vectorial addition, in other words, the effects of the horizontal load are vectorially added to the effects of the vertical load to obtain the total seismic effect on the system (IBC 1617.1.1, 1621.1.4)

$E = F_P + F_V$

Step—6: the total load is the sum of the seismic load E and the weight W. If the allowable stress design method (also called working stress design method) is used to qualify the piping system, as is the common practice, then the seismic load E may be divided by 1.4 (IBC 1605.3.2), the total load is therefore

$F_T$=W+E/1.4

## 11.6 SEISMIC RESPONSE SPECTRA

A seismic response spectrum is a plot of accelerations, velocities or displacements against periods or frequencies, for a given damping value, as illustrated in Figure 11-1.

For piping analysis, a seismic response spectrum plot of acceleration vs. frequency plot is commonly used (Figures 11-1 and 11-2). The practical meaning of the seismic response spectrum is the following: for a given frequency $f_N$ on the horizontal axis, the seismic response spectrum gives the maximum acceleration reached by a single degree of freedom (a mass m, spring stiffness k and dashpot with damping ζ, Chapter 8) if it is subject to the input excitation (acceleration vs. time) of a given earthquake. As a prerequisite to seismic response spectra analysis, seismic response spectra must developed in the free field (ground spectra) and in the structure or building at elevations of pipe support attachment (floor response spectra).

### 11.6.1 Seismic Input

To further understand seismic response spectra and introduce the terminology commonly used in seismic design and analysis, consider the equation of mo-tion of a single degree of freedom of mass m, stiffness k and damping c, subject to a sinusoidal excitation $P_o$sinωt

$$m\ddot{x} + c\dot{x} + kx = P_o \sin \omega t$$

x=displacement of mass relative to ground, in

m=mass, lb-sec$^2$/in

c=damping

k=stiffness, lb/in

$P_o$=magnitude of applied force, lb

ω=circular frequency of applied load, 1/sec

In the case of an earthquake, the input forcing function P(t) is not a single frequency sinusoid Posinωt but a time-dependent forcing function, the ground or structure motion

$$P(t) = m\ddot{x}_g(t)$$

m=mass of single degree of freedom oscillator' lb-sec$^2$/in

$x_g$(t)=amplitude of ground motion at time t, in

The displacement of a SDOF subject to the seismic forcing function of general form P(t) is obtained by the Duhamel integral

$$x(t) = \int_0^t \frac{P(\tau)}{M\omega_D} e^{-\zeta\omega(t-\tau)} \sin \omega_D (t-\tau) d\tau$$

Given the displacement function x(t), the response spectrum can be developed, and the following parameters can be defined:

Spectral displacement—the maximum value of the displacement x(t) of the SDOF oscillator of natural frequency $\omega_D$ and damping $\zeta$ subject to an earthquake P(t) is the spectral displacement at frequency $f=\omega_D/2\pi$ and damping $\zeta$.

Spectral velocity and acceleration—the maximum value of the first and second derivatives of x(t) are the spectral velocity and acceleration respectively. The spectral acceleration will be noted $a(f,\zeta)$.

Peak spectral acceleration (PSA)—the maximum spectral acceleration for a given damping $\zeta$; $a(\zeta)=\max a(f, \zeta)$. For a given damping $\zeta$, it is the maximum acceleration (the upper plateau) in Figure 11-1. It is $S_{DS}$ in Figure 11-2.

Peak ground acceleration (PGA)—the maximum seismic acceleration of a SDOF oscillator with infinite frequency $a(f=\infty,\zeta)$ placed on the ground. Note that in the "rigid range" (large frequencies f) the acceleration does not depend much on damping. The maximum acceleration of a rigid SDOF $a(f=\infty,\zeta)$ is the maximum acceleration of the ground since the rigid SDOF does nothing more than follow the ground motion, hence the name "peak ground acceleration". In earthquakes, the "rigid range" typically starts between 20 Hz and 33 Hz. The PGA is 0.4 $S_{DS}$ in Figure 11-2.

Zero period acceleration (ZPA)—the spectral acceleration at zero period, i.e. at infinite frequency. At ground level, the ZPA is the PGA. It is the acceleration of the right-hand side tail of the response spectrum in Figure 11-1. Note that it is practically independent of damping. It is 0.4 $S_{DS}$ in Figure 11-2.

Design basis earthquake response spectra—over the years, engineers have used some classical shapes as seismic response spectra. These spectra are scaled up or down to match the particular site's peak ground acceleration (PGA). Classical DBE spectral shapes have been developed and scaled to a site-specific peak ground acceleration [Housner, Newmark, US NRC].

### 11.6.2 Modal and Directional Combinations

In the response spectra analysis of a piping system, displacements and loads are first calculated separately for each natural frequency (mode) of the system, in each of three directions (north-south, east-west, vertical). The modal results and directional results are then combined to obtain a total, resultant response of the system. The engineer has a choice of modal and directional combination techniques. In the original modal analysis work, Newmark had chosen as resultant response the square root sum of the squares (SRSS) of the individual modal responses (loads or displacements at the various points along the piping system) [Newmark]

$$R = \sqrt{\sum_{1}^{N} R_i^2}$$

R=resultant response

$R_I$=response in mode i

Studies by Singh et. al. concluded that the SRSS combination could underestimate the total response if some modal frequencies of the equipment were closely spaced [Singh]. Closely spaced modes are modes with frequencies within about 10% of each other. Several techniques exist to combine closely spaced modes [US NRC RG 1.92], including the ten percent method

$$R = \sqrt{\sum_{1}^{N} R_k^{\,2} + 2\sum_{i,j} \left| R_i R_j \right|}$$

R=resultant response

$R_k$=response in mode k

$R_I$ and $R_j$=response in two closely spaced modes

The resultant load and displacement from three-directional seismic input is typically obtained by square root sum of squares of the response in each direction

$R=[(R_{EW})^2+(R_{NS})^2+(R_V)^2]^{0.5}$

R=resultant response

$R_{EW}$=east-west response

$R_{NS}$=north-south response

$R_V$=vertical response

Or by the so-called 100−40−40 technique:

$R_{100-40-40}$=100% $R_{EW}$+40% $R_{NS}$+40% $R_V$

R=max $\{R_{100-40-40}; R_{40-100-40}; R_{40-40-100}\}$

## 11.7 SEISMIC QUALIFICATION

At each point along the piping system model, the output from the seismic analysis consists of loads (forces and moments), total longitudinal stress $P_OD/(4t)$ $+0.75i(M_W+M_S)/Z$, displacements and rotations, and loads on supports, restraints and equipment nozzles. Some software also provides acceleration at every point. The extent of seismic qualification depends on the required function, as illustrated in Table 11-3, and has been discussed in section 11.2.

**Table 11-3** Seismic Qualification Criteria

| Criterion | Operability | Leak Tight NPS>2” PSA>0.3 g | Leak Tight NPS≤2” or PSA≤0.3 g | Position Retention |
|---|---|---|---|---|
| Pipe Stress | Y | Y | N | sway bracing |
| Mech. Joint | Y | Y | Y | N |
| Eq’t. Anchored | Y | Y | Y | Y |
| Eq’t. Operable | Y | N | N | N |
| Restraints | Y | Y | Y | Y |
| Interactions | Y | Y | Y | Y |

## 11.8 SHAKE TABLE TESTING

The most direct method to seismically qualify an active component that must perform a function during or after a design basis earthquake is through shake table testing. The designer specifies the required response spectrum (RRS) at 5% damping for which the equipment must be qualified. The designer also specifies, on drawing, the equipment mounting details. The test laboratory develops an artificial seismic input motion x(t) programmed into the shake table servomechanism of the shake table. The equipment is mounted on the shake table according to the installation details supplied by the designer, and the equipment is shaken. The measured table spectrum is the test response spectrum (TRS) and it must envelope the RRS for the test to be valid. The equipment integrity and operation may be verified during and after the test. Seismic testing is particularly well suited to qualify electrical equipment and active mechanical equipment, which must operate during or following the earthquake. A seismic test must be well planned and entrusted to a test facility experienced in applying the test methods of ICBO AC156, IEEE 344 and IEEE 382. Planning and conduct of a shake table test may be summarized in eight steps:

Step 1—Select testing method: equipment is seismically tested and qualified by one of three methods: (i) proof testing (test the equipment to an RRS equal to or slightly larger than the design basis earthquake), (ii) generic testing (test the equipment to a larger RRS than required by the DBE), (iii) fragility testing (test with steadily increasing input excitation, until failure of the equipment or until the table capacity is reached).

Step 2—Decide whether to test the assembly or a device. When testing an assembly such as a pump skid, the test arrangement must accurately simulate the equipment mounting and its attachments. When testing a device, such as a valve motor operator alone without the valve, the test arrangement must accurately simulate the amplification of seismic input that will take place through the pipe span and the valve body.

Step 3—Specify the test input. The applicable test standard, such as ICBO AC156, will normally specify the type of test: single frequency, sine sweep or response spectrum test. The single frequency test is suitable for equipment with single dominant frequency and narrow in-line or in-structure input. The test should be sufficiently long, in the order of 20 seconds. The sine-sweep test consists of a sinusoidal input with varying frequency, sweeping the frequency range of the spectrum. The table dwells on certain frequencies, for example dwell points at 2–4–8–16–32 Hertz. The test is valuable in identifying the equipment natural frequencies. The response spectrum test is a test at the specified 5% damped required response spectra (RRS) in each direction. The test facility will have to provide a plot of the measured test response spectra (TRS) at 5% damping, showing that they equal or exceed the RRS.

Step 4—Choose single or multiple axis test. The applicable test standard, such as ICBO AC156, will normally specify the type of test: single axis, bi-axial or triaxial test. In the single-axis test, the equipment is shaken in a single direction. It is a useful test for studies and research because the response is not complicated by multi-directional input. The bi-axial test consists of a horizontal direction run simultaneously with the vertical direction. The equipment is then rotated 90° horizontally and the test is repeated. The tri-

axial test consists of statistically independent input in all three directions, and is commonly used in qualification tests.

Step 5—Specify interface requirements. These include mounting and means of hold-down, instrumentation and applied piping loads on the component.

Step 6—Specify Inspections. The designer should specify the desired function during and/or after testing, and what to inspect at the test facility, prior to, during and following the test.

Step 7—Specify instrumentation and records. Typically, the test instrumentation includes accelerometers on the table. The table accelerometers record the table input and confirm that the required input (RRS) is enveloped by the table test response spectra (TRS), over a certain frequency range (such as 1 Hz to 100 Hz). In addition, accelerometers are mounted on the equipment.

Step 8—Specify the contents of the test report. The applicable standard will normally specify the contents of the test reports. The results must be readable and easy to interpret, accompanied by photographs (and videotape) of the test. The test report will normally include pre- and post-inspections, results of the functional test, photographs, drawings of test setup, plots of RRS vs. TRS at same damping (typically 5%), report of anomalies, and certifications.

## 11.9 SEISMIC INTERACTIONS

### 11.9.1 Description

An interaction is the seismic induced failure of a structure, system or component, other than the piping systems being qualified, that affects the function of the piping system. An interaction source is the component or structure that could fail and interact with a target. An interaction target is a component that is being impacted, sprayed or accidentally activated. A credible interaction is one that can take place. A significant interaction is one that can result in damage to the target. There are four types of seismic interactions:

Falling—A falling interaction is an impact on a critical component due to the fall of overhead or adjacent equipment or structure.

Swing—A swing or sway interaction is an impact due to the swing or rocking of adjacent component or suspended system.

Spray—A spray interaction is spray or flooding due to the leakage or rupture of overhead or adjacent piping or vessels.

System—A system interaction is an accidental or erroneous signal resulting in unanticipated operating conditions, such as the unintended start-up of a pump or closure of a valve.

### 11.9.2 Interaction Review

Having clearly identified the interaction targets, an interaction review consists of a walk-down, photographic record, and supporting calculations to document credible and significant sources of interactions.

In practice, it is only necessary to document credible and significant sources of interaction. It is not necessary to list and evaluate every single overhead or adjacent component in the area around the target; only those that could interact (credible interaction) and whose interaction could damage the target (significant interaction). In all cases, a photographic record of the interaction walk-down should be obtained.

Because only credible and significant sources of interaction are documented, an important aspect of the interaction review is engineering judgment. As a minimum, a team of two reviewers, each with at least 5 years experience in seismic design, must reach consensus on credible and significant interactions. The review team must be familiar with all three aspects of seismic engineering: analysis, testing and earthquake experience.

Where system interactions are of concern (erroneous signals, flow isolation, etc.), the written input of a system engineer is in order. An owner may also perform an independent third party review to verify the conclusions of the interaction evaluation.

### 11.9.3 Falling Interactions

In most cases, judgment is sufficient to establish whether a falling object can reach a target and be a credible interaction. Where judgment is insufficient, one can calculate the radius R of the zone in which a falling object can strike. This zone is called the zone of influence [Flanders]

$R=V_H \{[(V_v/g)^2+2H/g]^{0.5}-V_v/g\}$

R=radius of the zone of influence, in

$V_H$=horizontal spectral velocity, in/sec

$V_v$=vertical spectral velocity, in/sec

g=gravity=386 in/sec$^2$

H=height of fall, in

Earthquake experience indicates that suspended ceilings and block walls are a credible and significant source of interaction. When a falling body of weight W falls from a height h and impacts a target of weight $W_b$ and stiffness k, the impact force and deflection can be calculated based on energy conservation [Pilkey], where P is an overestimate of the impact force because it does not account for rebound, deformation of the source or friction and heat loss at impact

$$P = W + W_b + \sqrt{W_b^2 + 2W(W_b + kh)}$$

$$d = d_{st} + \sqrt{d_{st}^2 + 2h(d_{st} - d_s) - d_s^2}$$

P=impact force, lb

W=weight of falling body, lb

$W_b$=weight of elastic member, lb

k=stiffness of elastic member, referenced to point of impact, lb/in

h=height of free fall, in

d=maximum displacement at impact, in

$d_s$=static displacement of elastic member due to its own weight $W_b$, in
$d_{st}$=static displacement of member due to W plus $W_b$, in

### 11.9.4 Rocking or Swing Impact

The potential for sliding, rocking or overturning of free standing, unanchored equipment can be predicted based on the slenderness ratio (the height of the equipment's center of gravity relative to the width of its base), the coefficient of friction between the equipment and floor, and the horizontal and vertical acceleration [Aslam, Gates, Shao, Zhu]. The swing amplitude of a suspended system (suspended piping, HVAC, cable trays, etc.) can be estimated by

$d=1.3\ S_a/\omega^2$
$f_a=(g/L)^{0.5}/(2\pi)$

d=swing amplitude, in
$S_a$=spectral acceleration at frequency $f_a$, in/sec$^2$
$\omega$=natural circular frequency of swing motion=$2\pi f_a$, sec$^{-1}$
$f_a$=swing frequency, sec$^{-1}$
L=pendulum length, in
g=gravity, 386 in/sec$^2$

### 11.9.5 Significant Impact

Credible impacts that are significant must be documented. They include impacts on active components such as a pump or valve, instruments and impact sensitive components, pipe impacts by a pipe larger than the target pipe, collapse of portion of a wall or structure, impact by a heavy component, an overhead architectural feature or ceiling, or grating.

## 11.10 REFERENCES

AISC, Manual of Steel Construction, American Institute of Steel Construction, Chicago, IL.
AISI, Specification for the Design of Cold-Formed Steel Structural Members, American Iron and Steel Institute, Washington D.C.
ASCE 7, Minimum Design Loads for Buildings and Other Structures, American Society of Civil Engineers, Reston, VA.
ASHRAE CH 53, Seismic and Wind Restraint Design, American Society for Heating, Refrigeration and Air-Conditioning Engineers, Atlanta, GA.
ASHRAE RP-812, A Practical Guide to Seismic Restraint, American Society for Heating, Refrigeration and Air-Conditioning Engineers, Atlanta, GA.
Aslam, M, et. al., Earthquake Rocking Response of Rigid Bodies, ASCE Journal of the Structural Division, Vol. 106, No. ST2, February, 1980.
ASME B31.1, Power Piping, American Society of Mechanical Engineers, New York.
ASME B31.3, Process Piping, American Society of Mechanical Engineers, New York.
ASME B31.4, Pipeline Transportation Systems for Liquid Hydrocarbons and Other Liquids, American Society of Mechanical Engineers, New York.

ASME B31.5, Refrigerant Piping and Heat Transfer Components, American Society of Mechanical Engineers, New York.
ASME B31.8, Gas Transmission and Distribution Piping Systems, American Society of Mechanical Engineers, New York.
ASME B31.9, Building Services Piping, American Society of Mechanical Engineers, New York.
ASME B31.11, Slurry Transportation Piping, American Society of Mechanical Engineers, New York.
ASME Boiler and Pressure Vessel Code Section III, Division 1, Nuclear Components, Appendix F, American Society of Mechanical Engineers, New York.
ASME QME-1, Qualification of Active Mechanical Equipment Used in Nuclear Facilities, American Society of Mechanical Engineers, New York.
Flanders, H.E., Antaki, G.A., Thomas, B.D., Evaluation of Seismic Interactions: Guidelines and Application, PVP-Vol. 237, Seismic Engineering, Volume 2, ASME, 1992, American Society of Mechanical Engineers, New York.
Gates, W.E, Scawthorn, C., Mitigation of Earthquake Effects on Data Processing Equipment, Proceedings ASCE National Spring Convention, 1982.
Housner, G.W., Design Spectrum, Earthquake Engineering, Chapter 5, R.L.Weigel, ed., Prentice Hall, 1970.
IBC, International Building Code, International Code Council, Falls Church, VA.
ICBO AC156, Acceptance Criteria for the Seismic Qualification Testing of Nonstructural Components, International Conference of Building Officials, Whittier, CA.
IEEE-344, Recommended Practice for the Seismic Qualification of Class 1E Equipment in Nuclear Power Generating Stations, Institute of Electrical and Electronics Engineers, New York, NY.
IEEE-382, Standard for the Qualification of Actuator for Power-Operated Valve Assemblies with Safety-Related Functions for Nuclear Power Plants, Institute of Electrical and Electronics Engineers, New York, NY.
MSS-SP-127, Bracing for Piping Systems Seismic—Wind—Dynamic Design, Selection, Application, Manufacturers Standardization Society of the Valve and Fittings Industry, Vienna, VA.
Newmark, N.M., and Hall, W.J., Procedures and Criteria for Earthquake Resistant Design, Building Practices for Disaster Mitigation, National Bureau of Standards, 1973.
Newmark, N.M., and Hall, W.J., Earthquake Spectra and Design, Earthquake Engineering Research Institute, Monograph Series, reprinted, 1987.
NFPA-13, Installation of Sprinkler Systems, National Fire Protection Association, Quincy, MA.
Pilkey, W.D., Formulas for Stress, Strain, and Structural Matrices, John Wiley & Sons
Shao, Y., and Tang, C.C., North Carolina State University, Center for Nuclear Power Plant Structures, Equipment and Piping, report C-NPP-SEP-23/98, 1998 Seismic Response of Unanchored Structures and Equipment.
Singh, A.K., et. al., Influence of Closely Spaced Modes in Response Spectrum Method of Analysis, Proceedings of the Specialty Conference on Structural Design of Nuclear Power Plant Facilities, ASCE, 1973.
Slagis, G.C., Evaluation of Seismic Response Data for Piping, Welding Research Council Bulletin 423, New York, 1997.
SMACNA, Seismic Restraint Manual Guidelines for Mechanical Systems, Sheet Metal and Air Conditioning Contractors National Association, Chantilly, VA.
US NRC RG 1.60, Horizontal Design Response Spectra, US Nuclear Regulatory Commission Regulatory Guide 1.60.
US NRC RG 1.92, Combining Modal Responses and Spatial Components in Seismic Analysis, Regulatory Guide 1.92.

WRC 379, Alternative Methods for Seismic Analysis of Piping Systems, Welding Research Council Bulletin 379, New York, 1993.
Zhu, Z.Y., Soong, T.T., Toppling Fragility of Unrestrained Equipment, Earthquake Spectra, Volume 14, No 4, November 1998.

# 12
# Explosions

## 12.1 DEFLAGRATION AND DETONATION

Deflagration and detonation are the propagation of a combustion zone at subsonic and supersonic velocity, respectively. Explosion is generally defined as the resulting burst of the containing tank, vessel or pipe [NFPA]. In this chapter we will refer to explosion as the sudden release of energy that accompanies either a deflagration or a detonation. We will review the effects of explosions occurring inside vessels and pipes, as well as the effects of external explosions where the pressure wave hits the pipe from the outside.

The pressure wave that accompanies a deflagration progresses relatively slowly, for example at about 3 ft/sec (for hydrocarbon-air mixtures) and lasts from milliseconds to seconds. The pressure wave that accompanies detonation travels at supersonic velocities, near Mach 5, in the order of 5000 to 9,000 ft/sec [Lees] and even 2700 ft/sec for high explosives [Zukas], and lasts microseconds to milliseconds. As the high pressure that accompanies the combustion zone fills a vessel or travels down a pipe, the wall will experience large hoop and longitudinal stresses which may permanently deform, peal or burst it into fragments. As a deflagration flame front accelerates in the first few feet along a pipe (at 2 to 3 ft/sec, over 10 to 20 ft), it compresses the downstream gas causing piling pressures in the order of 300 psi. In a long pipe, beyond a run-up distance [Steen], the deflagration will transition to a supersonic detonation, with velocities in the order of 6000 ft/sec and pressures of several thousands psi. Peak pressures of 50 times the initial pressure have been measured at the instant of transition to detonation [Engebresten]. After reaching the peak pressure, the detonation pressure drops to a stable value. The phenomenon of transition from deflagration to detonation is not fully understood quantitatively [Shepherd], but there is a sufficient number of experiments to indicate that the transition from deflagration to detonation in piping and vessels depends on the type of gas mixture, its initial pressure, the pipe diameter, the bound-ary conditions (open, closed or partially closed flow path), and the layout. It occurs particularly quickly if the initial pressure is large and the pipe surface is rough, or in the presence of flow obstructions such as a valve or orifice. For example, a 490 ft/sec and 5.5 psi flame front in a straight pipe will transition into a 3800 ft/sec and 240 psi flame front at an elbow [Enardo]. The smaller the pipe and the higher the initial pressure, the faster the transition from deflagration to detonation [Lees]. For acetylene, the deflagration-detonation transition distance can be as short as 2 ft at an initial pressure of 80 psi, while it is 300 ft at an initial pressure of 10 psi.

As the explosion takes place, any point inside the pipe or vessel will sense a first incident pressure pulse, followed by smaller pulses due to the reflection and return of the initial pulse, as illustrated in Figure 12-1.

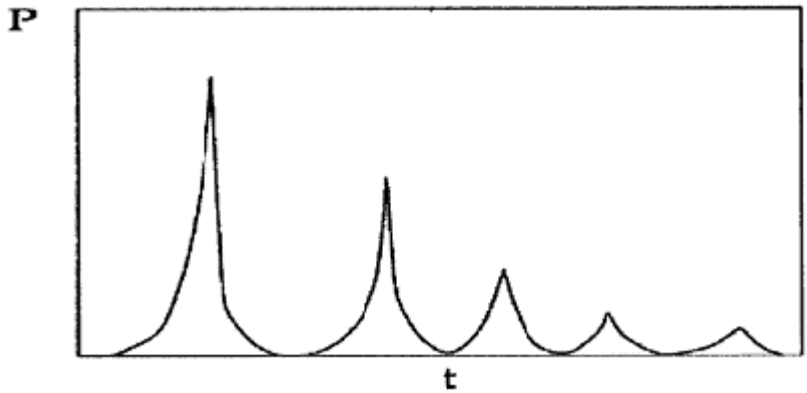


**Figure 12-1** Pressure Time History at a Given Point

The prediction of the shape of the pressure pulse, the pressure time history (pressure vs. time), is achieved using specialized thermo-hydraulic codes, or hydro-codes, benchmarked against experiments. The pressure pulse reflections, beyond the first peak, will depend on the pipe or vessel geometry, and are obtained by analysis of a coupled hydraulic-structural model [Duffey].

## 12.2 DYNAMIC LOADS

The severity of the dynamic pressurization of a piping system or vessel will depend on the duration of the pressure pulse compared to the natural period (the reverse of the natural frequency, T=1/f) of the piping or vessel, where T is a measure of how quickly the material will respond to the pressure pulse. The natural frequency of interest is the one that corresponds to the expected dynamic deformation of the pipe or vessel subject to the pressure pulse. For a cylinder (pipe or vessel shell), it is typically one of the two breathing modes: longitudinal (axial stretch) or radial (radial bulging). At a shell to head junction, the outward breathing mode of the shell and head will cause bending of the knuckle between the shell and head. The fundamental frequency of the axial mode of a pipe treated as an open cylinder is [Blevins]

$$f = \frac{1}{2L}\sqrt{\frac{E}{\mu}}$$

f=natural frequency of the first axial mode of cylinder, Hz

L=pipe length, in

E=Young's modulus, psi

μ=mass density of pipe material, lbm/in$^3$

The fundamental frequency of the radial breathing mode of a pipe treated as an open cylinder is [Blevins]

$$f = \frac{1}{2\pi R}\sqrt{\frac{E}{\mu(1-\nu^2)}}$$

$\nu$=Poisson ratio of material

R=radius of cylinder, in

In general, three load regimes can be defined when a load, in our case the explosive pressure, is applied dynamically to a structure or component: quasi static, impulsive or dynamic. The load regime that takes place in any particular case will depend on the natural period T of the component compared to the duration $T_I$ of the dynamic pressure pulse [Baker]:

If $T_I$>40 T, we are in the presence of a relatively stiff component (low natural period T compared to $T_I$) and a relatively long duration at pressure. The load is said to be quasi-static. The rigid component has time to fully deflect before the explosive pressure dissipates. For a rectangular pressure pulse, the maximum dynamic deflection is twice the static deflection under maximum pressure. For a triangular pressure pulse, the maximum dynamic deflection is 1.5 times the static deflection under maximum pressure

If $T_I$<0.4 T, we are in the presence of a relatively flexible component (large natural period T compared to $T_I$) and a relatively short duration at pressure. The load is said to be impulsive. The load dissipates before the flexible component has had time to fully deflect outward or stretch. The maximum deflection is lower than would be predicted for a static deflection under maximum pressure. Peak pressure is no longer sufficient to predict the pipe or vessel response; instead the deforma-tion will depend on the magnitude of the impulse, which is the area under the pressure vs. time curve. For example, under impulsive loading a triangular pressure pulse of 6500 psi lasting 70 microseconds has the same effect as a rectangular pressure pulse of 2000 psi lasting 130milliseconds [Duffey].

For $T_I$ between 0.4 T and 40 T, the load regime is neither quasi-static nor impulsive, it is called dynamic.

When the pipe or vessel experiences the first pressure pulse, its immediate response is an outward bulging and stretching (first breathing mode), If the pipe or vessel have not burst under the effect of the first pulse, they will rebound and reverberate. The reflection of pressure waves adds to this vibratory response. As the reflected pressure pulses subside, so will the vibration, unless—of course—the vessel or pipe has already burst. There is little material damping during the vibratory phase, typically less than 1% critical damping, because the vibration amplitude is small. This vibratory motion that follows the first expansion of the vessel or pipe can be calculated by finite element time history analysis. In practice, the first outward bulging can burst the component, while the following vibratory response of decreasing amplitude can contribute to fatigue failure, and they are therefore of interest mostly in the case of repetitive explosions. For a single explosion, fatigue is typically not a concern; the analysis of the first breathing mode response (radial and axial stretch, including bending at discontinuities) is usually sufficient to assess the integrity of the component.

## 12.3 DYNAMIC PROPERTIES

At high strain rates, such as encountered in explosions, there is an increase in yield stress and ultimate strength of steel, and a decrease in notch toughness.

The strain rate for a standard tensile test [ASTM E 8] is 100 psi/min, which for a material with a Young's modulus of E=30,000 psi, such as steel, corresponds to a strain rate of $100/(60\times30{,}000)=5\ 10^{-5}$ $sec^{-1}$. At room temperature, the yield stress and ultimate strength of a low carbon steel increase with strain rate, by approximately 10% between strain rates of $10^{-6}$ to $10^{-3}$ $sec^{-1}$ [Boyer]. Room temperature tensile tests conducted on mild carbon steel [Horger] indicate that yield and ultimate are practically unchanged at strain rates between $10^{-6}$ $sec^{-1}$ and $10^{-3}$ $sec^{-1}$, but yield increased by 30% at 0.5 $sec^{-1}$ while ultimate remained practically unchanged. At 100 $sec^{-1}$ and 300 $sec^{-1}$ yield had more than doubled, and ultimate strength increased by 50%. In a series of tests conducted at high temperature on 304 stainless steel, results at 1000°F showed no change in yield and ultimate between $3\ 10^{-5}$ $sec^{-1}$ and $10^{2}$ $sec^{-1}$, but at 1600°F yield doubled and ultimate strength more than tripled [Steichen].

A general form of the dynamic yield and ultimate stress as a function of strain rate, for low carbon steel at ambient temperature at strain rates between 1 and 100 $sec^{-1}$ is[DOE/TIC]

$S_{YD}/S_Y=1.3+0.25 \log (d\varepsilon/dt)$

$S_{UD}/S_U=1.1+0.1 \log (d\varepsilon/dt)$

$S_{YD}$=dynamic yield stress, psi

$S_Y$=static yield stress, psi

$d\varepsilon/dt$=strain rate, $sec^{-1}$

$S_{UD}$=dynamic ultimate strength, psi

$S_U$=static ultimate strength, psi

Other relationships for dynamic yield stress of steel are [Manjoine, Bodner]

$$S_{YD}/S_Y=\left(\frac{1}{40.4}d\varepsilon/dt\right)^{0.2}+1$$

and [Beazley]

$$S_{YD}/S_Y=\left(\frac{1}{100}d\varepsilon/dt\right)^{0.1}+1$$

For Aluminum alloys [Bodner]

$$S_{YD}/S_Y=\left(\frac{1}{6500}d\varepsilon/dt\right)^{0.25}+1$$

In reality, since the condition of maximum is dε/dt=0, the maximum strain is reached when the strain rate is theoretically nil and the strain is reversing. It is therefore incorrect to take credit for the increase in yield and ultimate strength in defining the maximum allowable stress or strain.

The increase in yield and tensile strength may however be accounted for in the material property model used in a finite element analysis [US NRC]. It is true that at strain rates of 1 $sec^{-1}$ and above, yield stress and ultimate strength of steels do increase. But, as discussed in Chapter 3, notch toughness decreases with increased strain rate. The material will behave in a more brittle manner at high strain rates, such as those encountered during explosions.

## 12.4 PRESSURE LIMITS

Having calculated the pressure applied to he pipe or vessel by an explosion, it becomes necessary to compare this applied pressure to the pressure that could cause the component to burst. The burst pressure of a cylinder under static or quasi-static pressure is [Cooper]

$$P_U = 4(3)^{-\frac{n+1}{2}} \frac{t}{D} S_U$$

$P_u$=burst pressure, psi
$S_U$=material ultimate strength
n=material parameter
t=wall thickness of cylinder, in
D=diameter of cylinder, in

The pressure that could cause the fragmentation of a cylindrical vessel or a pipe, in case of detonation is [Price]

$$P_{ex} \geq t\sqrt{2S_f \rho \varepsilon} / T_i$$

$P_{ex}$=pressure leading to explosion of container, psi
t=wall thickness of container, in
$S_f$=flow stress of material, psi,
ε=strain at failure
$T_I$=time during which the material is subject to the pressure wave, sec
ρ=material density, $lb/in^3$

## 12.5 DESIGN CRITERIA

### 12.5.1 Quasi-Static Load

To assess the integrity of a piping system or vessel under quasi-static internal pressure, we must first model the component, apply the deflagration or detonation pressure and obtain the resulting stresses, strains or deformations. In the simplest case, if the bending

stresses due to rebound and vibration are insignificant compared to the tensile and bending stresses from the radial and axial breathing deformations, the analysis may be static, with the pressure equal to

$P_{applied}$=(DMF) $P_{peak}$

$P_{applied}$=internal pressure applied to the model, psi

DMF=dynamic magnification factor

$P_{peak}$=peak (maximum) internal pressure during the explosion, psi

An accurate value of DMF can be obtained for a quasi-static load ($T_l$>40 T). This static pressure may be applied to an elastic or a plastic model of the component. A second analysis approach is to apply the time history of the pressure pulse to an elastic or plastic finite element model of the component. The dynamic magnification effect would then be automatically accounted for in the dynamic analysis process. Once stresses, strains or deflections are calculated (the demand on the component), the acceptance criterion (demand vs. capacity of the component) would depend on the type of analysis performed, and the design code. The ASME B31 pressure piping codes do not provide design rules for deflagration or detonation. The pressure vessel code ASME B&PV Section VIII Division 1, Appendix H [ASME VIII] does address deflagration and refers back to the rules of ASME B&PV Section III Appendix F [ASME III]. ASME B&PV Section VIII Division 2 contains general rules for stress qualification that can be applied to explosion analysis. ASME VIII Division 3 (high pressure vessels) is particularly well suited for explosion analysis since it addresses strength, fatigue and toughness.

To apply the ASME B&PV design rules, we must first define certain terms: Primary general membrane stress intensity ($P_m$) is an average stress across a section, excluding discontinuities (such as a transition in diameter or a shell-head knuckle) and stress concentrations (such as a sharp shell-nozzle corner or the edge of an opening). Primary bending stress intensity ($P_b$) is the linear stress gradient that goes through zero at mid-thickness, excluding discontinuities and stress concentrations. Primary local membrane stress intensity ($P_L$) is the average stress across a section, considering discontinuities (gradual size and shape changes) but not stress concentrations (sharp corners or openings). Secondary membrane plus bending stress intensity (Q) is the stress at a structural discontinuity, excluding stress concentrations. Peak stress (F) is the incremental stress due to a concentration.

In the static or quasi-static regime, the rules of ASME B&PV Section VIII, Division 2 are based on elastically calculated stresses. They depend on the code allowable stress $S_m$, which is a function of the material and its temperature at time of loading. $S_m$ is listed in ASME B&PV Section II Part D [ASME II]. The ASME B&PV Section VIII Division 2 rules state that the elastically calculated stresses in a vessel must be limited to $P_m$<k $S_m$, $P_L$<1.5 k $S_m$, $P_L+P_b$<1.5 k $S_m$, $P_L+P_b+Q$<3 $S_m$, $P_L+P_b+Q+F$<3 $S_m$.

In the static or quasi-static regime, as an alternative to ASME VIII Division 2, the rules of ASME B&PV Section III Appendix F provide five criteria for the evaluation of extreme load conditions, illustrated as (a) to (e) in Figure 12-2.

Elastic analysis: in the case of an elastic analysis, Figure 12-2 (a), the stress is proportional to strain and the elastically calculated primary membrane stress is limited to 70% of the material's ultimate strength (or "ultimate" $S_U$), the primary membrane and

bending stresses are limited to 105% of ultimate and the shear stress is limited to 42% of ultimate.

Plastic analysis: in a plastic analysis, Figure 12-2 (b), using the actual stress-strain curve of the material to predict stresses, the maximum primary stress intensity is limited to 90% of ultimate, and the primary membrane and shear stresses are limited to 70% and 42% of ultimate respectively, as in elastic analysis.

Limit collapse analysis: in a limit collapse analysis, Figure 12-2 (c), where the material is modeled as elastic-perfectly plastic with an elastic stress-strain followed by a flat horizontal stress-strain line beyond yield, the applied load is limited to 90% of the load at collapse (the load that causes excessively large deformations).

Plastic collapse analysis: in a plastic collapse analysis, Figure 12-2 (d), the limit load is that at which the deformation reaches twice the deformation at the onset of yield, as defined by $\Phi_2=2\ \tan^{-1}\Phi_1$.

Plastic instability analysis: in a plastic instability analysis, Figure 12-2 (e), the plastic model is progressively deformed, forming multiple hinges, to the point where plastic instability occurs; the limit load is defined as 70% of the instability load.

### 12.5.2 Impulsive Loads

In the impulsive load regime, when the pressure pulse is much shorter than the component's natural period, the evaluation of integrity could be based on deformations and stresses as for the quasi-static regime, but a more accurate evaluation should account for the time-dependent interaction between the impulse and the component, and the acceptance criteria should be based on the plastic strain in the component, compared to the plastic instability strain of the material, n+0.15 effective strain and 0.87(n+0.15) hoop strain [Duffey].

### 12.5.3 Fracture

The analysis of explosions, following the rules of ASME III or ASME VIII should be supplemented by a fracture analysis, in which a crack sufficiently large to have been undoubtedly detected during fabrication or later inspections is postulated at various critical points, and the fracture stability of the flawed vessel subject to the explosive pressure is analyzed with due consideration for the reduction in fracture toughness due to the dynamic nature of the applied pressure [ASME VIII Div.3, ASME XI, API 579].

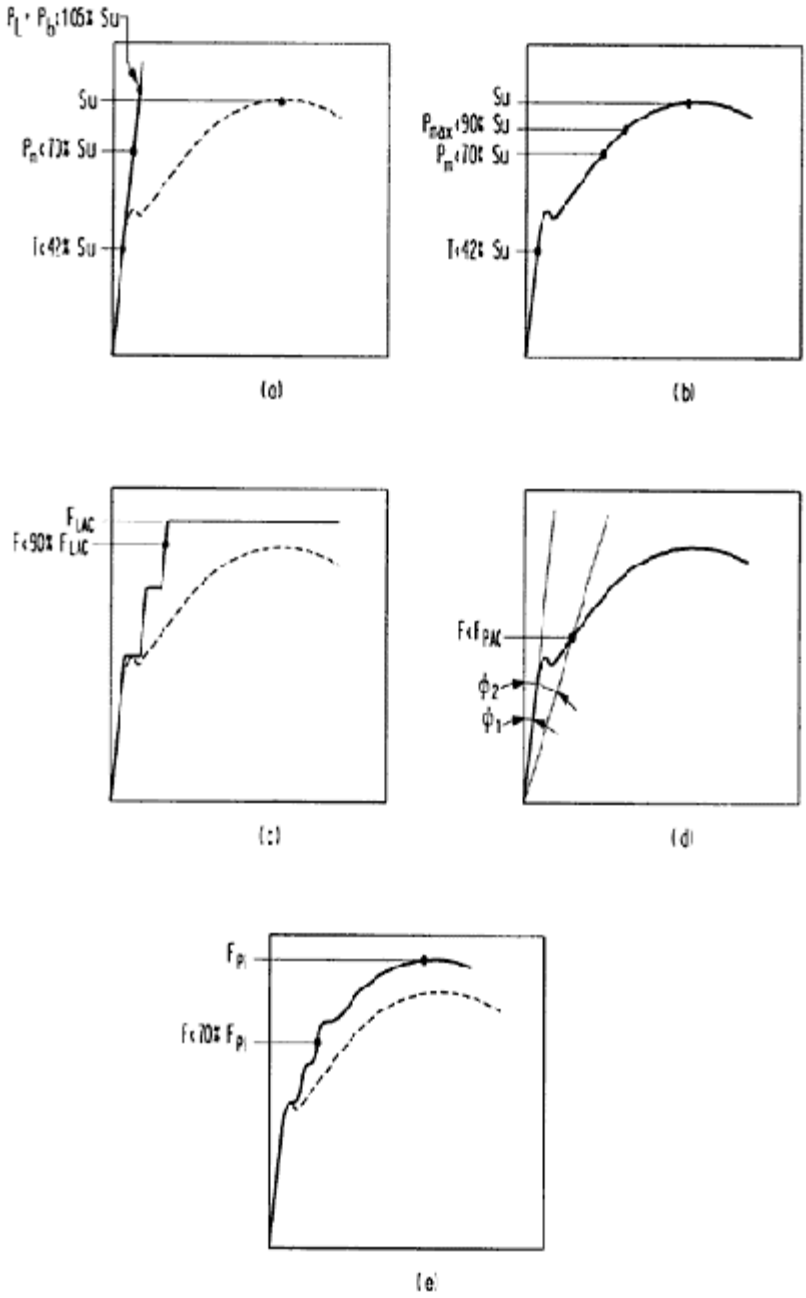


**Figure 12–2** ASME III Appendix F Criteria

## 12.6 EXPLOSION PROTECTION

Ideally, explosions should be prevented from occurring by system design. When this is not feasible, the designer or the owner would resort to explosion mitigation devices such as flame and detonation arresters [API 2028, API 2210, UL], explosion isolation valves [Fike], suppression systems, explosion vents [NFPA], or rupture discs [ASME VIII].

Flame arresters are typically passive devices consisting of wound metallic channels that permit relatively free passage of gas under normal operating conditions, but act as heat exchangers in case of explosion and absorb the flame's heat. Some flame arresters have liquid seals and baffles. Flame arresters must be properly selected and are sized for the fluid, pressure and type of explosion (deflagration or detonation) [Mendoza].

Explosion isolation valves detect a pressure increase through pressure sensors mounted on vessels, tanks or pipes. The sensors trigger the discharge of a nitrogen cylinder that pneumatically closes an isolation valve, in a matter of milliseconds [NFPA, Fike].

Suppression systems are somewhat similar to fire protection systems. A sensor detects a deflagration pressure spike and sends a signal to discharge Halon or a powdered chemical suppressant through nozzles, into the system.

Explosion vents are openings through which explosion gases are vented. They are typically used for venting deflagrations in atmospheric storage tanks and large enclosures. The vent area is [NFPA 68]

$$A_V = \frac{CA_S}{\sqrt{P_{red}}}$$

$A_v$=vent area, $ft^2$

C=venting equation constant (0.05 to 0.20 depending on fuel).

$A_S$=internal surface area of enclosure, $ft^2$

$P_{red}$=maximum design pressure of enclosure, psi

Deflagrations and detonations can be vented through rupture discs. These are non-reclosing devices designed to burst open at a predetermined pressure [AIChE]. They are available in a wide range of sizes (up to 44" from catalog, and larger if custom made), and burst pressures (up to 100,000 psi). In the 1920's, the first rupture discs were flat sheets of metal used as "fuse points" to blow open and release pressure, these discs were however prone to fatigue failure from pressure fluctuations in service. The flat sheet design was replaced by dome shaped discs, first produced in the 1930's, made from a solid sheet of thin metal, tension loaded (pressure on the concave side). Current discs are reverse buckled (pressure on the convex side), the dome reverses by buckling at the burst pressure and shears along score lines or by pressing against knife blades.

Rupture discs are ASME stamped "UD" and national Board stamped "NB". They have a burst pressure, with a tolerance (typically the larger of 5% or 2 psi), a burst time in the order of milliseconds, and a certified flow resistance coefficient. Some discs can be used for gas or liquid, while others are to be used exclusively in gas service. Because rupture discs are made of thin sheet metal, they must be periodically inspected or replaced to verify that they have not thinned by corrosion. Damage to the disc surface (dents, gouges) will also affect its behavior. For example, a dented 100 psi disc may reverse buckle with

a light popping sound at 60 psi, but the now reversed dome (with pressure on the concave side) will not burst until reaching pressures well above 100 psi.

A rupture disc installed upside-down (pressure on concave side rather than the intended pressure on the convex side) may burst at 1.5 times the intended pressure. It is therefore essential that the manufacturer provide a clearly visible indication of flow direction, and that instructions be followed by the owner during installation. Rupture discs are installed in threaded, welded or bolted disc holders. They can be used as the sole overpressure protection device, or in series with a second disc, or upstream of a pressure relief valve under certain conditions [ASME VIII].

## 12.7 EXTERNAL EXPLOSIONS

The effects of external explosions on buildings and structures in an industrial complex have been addressed by the American Society of Civil Engineers [ASCE]. An equivalent static analysis may be used to evaluate the effect an external explosion may have on a piping system. First, the duration of the pressure pulse ($T_l$) is compared to the natural lateral bending period (T) of the piping system. In the quasi-static regime ($T_l$>40 T) the lateral explosion force is equivalent to a static acceleration given by

$$a = \frac{(DMF)P_o DC_f}{w}$$

a=equivalent quasi-static acceleration, g's

DMF=dynamic magnification factor, 2 maximum

$P_o$=explosion pressure at the pipe, psi

D=pipe diameter, including insulation thickness, in

$C_f$=force coefficient for a cylinder [ASCE]

w=weight per unit length of pipe, lb/in

The lateral acceleration is applied laterally to the pipe as a distributed acceleration a, and deflections, bending moments and stresses are calculated.

In the impulsive regime ($T_l$<0.4T) the lateral deflection can be calculated by [Baker]

$$\Delta = \frac{iDL^2}{\sqrt{50\rho AEI}}$$

Δ=lateral elastic-plastic deflection of pipe span subject to explosion, in

i=specific impulse, integral of pressure-time curve (area under curve), psi-sec

L=length of pipe span, in

ρ=mass density of pipe, lb/in$^3$

A=metal area of cross section, in$^2$

E=Young's modulus of pipe material, psi

I=cross section moment of inertia of pipe, in$^4$

The designer should keep in mind that the incident pressure pulse will reflect on a wall and therefore some wall mounted supports and anchors that were in compression during

the passage of the first incident pressure wave, will be in tension, generally more vulnerable, when subject to the reflected wave.

An important question is how to obtain the external explosion pressure $P_o$ applied to the pipe. There are three solutions to this question: First, the system could be designed for the maximum pressure at the point of explosion. But this would ignore the decay of the pressure wave, and may be totally unrealistic at a certain distance from the blast source. Second, the pressure time history may be numerically calculated as a function of time and distance using hydro-codes. Third, a closed form solution could be used to bound, realistically, the problem.

An elegant closed form solution, developed by Chowdhury and Suri [Chowdhury], is summarized here. As the blast front passes through a point at a distance r and time t—for example the location of the pipe or pipe rack—the front will cause the air to move an amount u(r,t). The propagating wave equation is

$$\frac{\partial^2 u}{\partial t^2} = c^2\left(\frac{\partial^2 u}{\partial r^2} + \frac{1}{r}\frac{\partial u}{\partial r}\right)$$

u(r,t)=displacement of air caused by passage of explosion, in
c=velocity of sound in air, in/sec
r=distance of analysis point from source of explosion, in
t=time after the initiation of the explosion, sec
By separation of variables
$u(r,t)=\Phi(r)\, q(t)$

and with the boundary conditions
$u(r=R, t)=0$
$u(r,t=0)=0$
$v(r=0, t=0)=v_o$

v(r,t)=velocity of air in pressure wave, in/sec
$v_o$=initial velocity at source of explosion, in/sec
Solving the differential equation, with boundary conditions, leads to

$$u(r,t) = \left[J_o(\alpha_m \frac{r}{R})\right]\left(\frac{v_o R}{c\alpha_m}\sin\frac{c\alpha_m}{R}t\right)$$

$$v(r,t) = \left[J_o(\alpha_m \frac{r}{R})\right]\left(v_o \cos\frac{c\alpha_m}{R}t\right)$$

$$P(r,t) = \rho\frac{v^2}{k}$$

$$R = v_o t_{expl.}$$

$J_o$=Bessel function of order 0

$a_m$=solutions to $J_o(\alpha_m r/R)=0$
P(r,t)=applied pressure due to air compression, psi
$\rho$=mass density of air, $lb/in^3$
k=ratio of specific heats, 1.4 for air
R=blast wave length (distance beyond which P is small), in
$t_{expl.}$=duration of explosion, sec

The free field wave reflected on the incident surface, for example the pipe or rack, will cause a total pressure $P_r$ (kPa) and a stagnation residual pressure $P_S$

$P_r(r)=(2+0.0073\ P)P$

$P_s(r)=P_r\{1-[(k-1)/2]M^2]^{k(k-1)}$

$P_r$=peak total pressure, including reflection on target surface, kPa
$P_S$=residual stagnation pressure, after decay of $P_r$, psi
M=Mach number at distance r, v(r)/c

## 12.8 REFERENCES


AIChE, Guidelines for Pressure Relief and Effluent Handling Systems, American Institute of Chemical Engineers, New York.

API 579, Fitness for Service, American Petroleum Institute, Washington, DC.

API 2028, Flame Arresters in Piping Systems, American Petroleum Institute, Washington, DC.

API 2210, Flame Arresters for Storage Tanks Storing Petroleum Products, American Petroleum Institute, Washington, DC.

ASCE 7, Minimum Design Loads for Buildings and Other Structures, American Society of Civil Engineers, Reston, VA.

ASCE, Design of Blast Resistant Buildings in Petrochemical Facilities, American Society of Civil Engineers, Reston, VA.

ASME Boiler & Pressure Vessel Code, Section II, Materials, American Society of Mechanical Engineers, New York

ASME Boiler & Pressure Vessel Code, Section III, Nuclear Components, American Society of Mechanical Engineers, New York.

ASME Boiler & Pressure Vessel Code, Section VIII, Pressure Vessels, American Society of Mechanical Engineers, New York.

ASME Boiler & Pressure Vessel Code, Section XI, In-Service Inspection of Nuclear Power Plant Components, American Society of Mechanical Engineers, New York.

ASTM E 8, Tension Testing of Elastic Materials, American Society of Test and Materials, PA.

Baker, W.E., et. al., Explosion Hazards and Evaluation, Elsevier, New York, 1983.

Beazley, P.K., Large Elastic-Plastic deformation of Pipes, Analytical Phase II for Long Pipe Specimens, Pressure Vessel Research Council, New York, 1983.

Blevins, R.D., Formulas for Natural Frequency and Mode Shapes, Van Nostrand, 1979.

Bodner, S.R., Symonds, P.S., Experimental and Theoretical Investigation of Plastic Pipe Deformation of Cantilever Beams Subject to Impulsive Loading, Trans. Journal of Applied Mechanics, December, 1962.

Boyer, H.E., ed., Atlas of Stress-Strain Curves, ASM International, American Society of Metals, Metal Park, OH.

Chowdhury, I., Suri, S.A.K., Improve Estimation for Dynamic Pressure Waves from an Explosion, Hydrocarbon Processing, October, 2002.

Cooper, W.E., The Significance of Tensile Test to Pressure Vessel Design, Welding Research, Supplement, January 1957.
DOE/TIC-11268, A Manual for the Prediction of Blast and Fragment Loadings on Structures, U.S. Department of Energy, Washington, D.C., 1981.
Duffey, T.A., Rodriguez, E.A., Romero, C., Design of Pressure Vessels for High Strain Rate Loading: Dynamic Pressure and Failure Criteria, Welding Research Council Bulletin WRC 477, WRC, New York.
Engebresten, T., Propagation of Gaseous Detonation Through Regions of Low Reactivity", PhD. Thesis, 1991, ITE NTH, Trondheim, Norway.
Enardo Manufacturing Company, Catalog, Tulsa, Oklahoma.
Fike, Catalog, Blue Springs, MO.
Gillis, P.P., Gross, T.S., Effect of Strain Rate on Flow Properties, American Society of Metals, ASM, 1995.
Horger, O.J., Metals Engineering Design, McGraw-Hill, New York.
Lees, F.P., Loss Prevention in the Process Industries, Butterworth-Heinemann, Oxford, England.
Manjoine, M.J., Influence of Rate of Strain and Temperature on Yield Stress of Mild Steel, Journal of Applied mechanics, Vol. 11, ASME Trans. Vol. 66, 1944.
Mendoza, V.A., et. al., Do Your Flame Arresters Provide Adequate Protection?, Hydrocarbon processing, October, 1998.
NFPA 69 Explosion Prevention Systems, National Fire Protection Association, Quincy, MA.
Price, J.W.H., An Acetylene Gas Cylinder Explosion, Transactions of the ASME, Vol. 120, February, 1998.
Steen, H., Schampel, K., Experimental Investigation of the Run-Up Distance of Gaseous Detonations in Large Pipes, Pergamon Press, UK, 1983.
Steichen, J.M., High Strain Rate Tensile Properties of AISI Type 304 Stainless Steel, Transactions of the ASME, July, 1973, American Society of Mechanical Engineers, New York.
Shepherd, J.E., et. al., Unconfined Vapor Cloud Explosions: A New Perspective, International Conference on Modeling and Mitigating the Consequences of Accidental Releases, 1991.
UL Publication 525, Flame Arresters for Use on Vents of Storage tanks for Petroleum, Oil and Gasoline, Underwriters Laboratories.
US NRC, NUREG-1061, Volume 4, Report of the U.S. Nuclear Regulatory Commission Piping Review Committee, Evaluation of Other Dynamic Loads and Load Combinations, US. NRC, Washington, DC.
Zukas, J.A., Walters, W.P., Explosive Effects and Applications, Springer Verlag, New York.

# 13
# Subsea Pipelines

## 13.1 SUBSEA PIPELINE SAFETY

In the United States, offshore production of oil and gas began in the early 1950's. Today, in the Gulf of Mexico, there are over 17,000 miles of subsea pipelines gathering and carrying oil and gas from offshore wells to platforms and to facilities onshore. Most of these pipelines are regulated by the Office of Pipeline Safety of the U.S. Department of Transportation (under Code of Federal Regulations 49 CFR), and some subsea pipelines are regulated by the Minerals Management Services of the U.S. Department of the Interior (under Code of Federal Regulations 30 CFR). State regulations also apply in certain locations. Because of the potential significance of leaks in offshore pipelines, there are continuous efforts to report, trend and analyze failures. From the investigation of nearly 1000 incidents in the period 1960 to 1990, the following conclusions were reached [NRC]: 49% of offshore pipeline failures are due to corrosion, but these corrosion induced failures have only caused 2% of the pollution and resulted in no fatalities. In contrast, failures from maritime traffic account for only 14% of the failures but have caused 90% of the pollution damage and have resulted in several fatalities. Fatal accidents include the 1987 Sea Chief vessel accident in which the vessel struck an 8" pipeline operating at 480 psi, causing an explosion that killed two. The struck pipeline had been installed in 1968 under a 3 ft cover of sediments, but erosion had removed much of the initial cover and the line was only under 6" of mud at the time of the accident. In another case, in 1989, the Northumberland vessel struck a 16" gas pipeline. The resulting fire killed eleven. The line, installed in 1972 under 8 ft of cover, was uncovered at the time of the accident. Today, depth of burial is closely regulated in zones of Heavy maritime activity. For example, in the U.S., a minimum cover of 3 ft below bottom or 18" below consolidated rock are required in depths shallower than 12 ft. In deepwater ports these depths become 6 ft and 3 ft respectively.

In the continuing search for cost-effective safety, risk-informed design criteria are being established. In one case, four limit states are defined [Sotberg]: serviceability (normal operation), ultimate (leak tightness, but the line may not be operable), fatigue (crack propagation), and accident (rupture). At the same time, a safety classification is established based on the consequence of failure, as shown in Table 13-1. Given the safety class, an annual probability of exceedance (the permissible yearly probability of not meeting a particular limit state) is established for each of the four limit states, as shown in Table 13-2.

**Table 13-1** Safety Classification [Sotberg]

| | Location Class I(1) | Location Class II(2) |
|---|---|---|
| Non-flammable, Non-toxic | Low Safety | Low Safety |
| Flammable or Toxic Liquid | Normal Safety | High Safety |
| Flammable or Toxic Gas | Normal safety | High safety |

(1) No human activity
(2) Within 500 m of human activity

**Table 13-2** Annual probability of Exceedance [Sotberg]

| | Low Safety | Normal Safety | High Safety |
|---|---|---|---|
| Serviceability | $10^{-1}$ to $10^{-2}$ | $10^{-2}$ to $10^{-3}$ | $10^{-2}$ to $10^{-3}$ |
| Ultimate | $10^{-2}$ to $10^{-3}$ | $10^{-3}$ to $10^{-4}$ | $10^{-4}$ to $10^{-5}$ |
| Fatigue | $10^{-3}$ | $10^{-4}$ | $10^{-5}$ |
| Accident | $10^{-4}$ | $10^{-5}$ | $10^{-6}$ |

## 13.2 DESIGN PROCESS

The first step in the design process of subsea pipelines is very similar to onshore pipelines: the design process starts with the thermo-hydraulic analysis of the line to determine the required flow area (inner diameter) and pumping or compressing capacity, taking into consideration the fluid properties, flow regime, flow rates and pressure drops. This step is complicated in the presence of two-phase flow (liquid and gas), with the liquid phase being itself comprised of several constituents (for example oil and water). In the second design step, a wall thickness is selected based on the maximum internal pressure, temperature, and material. The design for internal pressure follows the same rules as for onshore pipelines. Beyond this second step, the design process becomes unique to subsea pipelines. In the third step, the wall thickness is checked against the external hydrostatic pressure applied to the pipeline, to verify that the pipe will not buckle. The fourth step considers installation loads as the pipe is welded and handled in long sections then lowered to the sea or lakebed. In the fifth step, the subsea pipeline is analyzed for on-bottom stability. Several iterations are usually necessary to arrive at the optimum choice of pipe material, grade and wall thickness.

## 13.3 INTERNAL PRESSURE

The selection of wall thickness, given the maximum pressure and temperature, and the pipe material specification and grade, follows the rules of ASME B31.4 for liquid

pipelines and ASME B31.8 for gas or two-phase (liquid-gas) pipelines, and is covered in Chapter 4.

## 13.4 EXTERNAL PRESSURE

In the presence of external pressure, and in the absence of other loads, the buckling mode of an ideally round and straight pipe depends on its D/t ratio (diameter divided by thickness), in the same way that the buckling of a column in pure compression depends on its slenderness ratio. For large D/t (thin wall pipe) the buckling occurs while the material is still elastic (elastic buckling). The elastic collapse external pressure is $P_E$ given in Chapter 5 as

$$P_E = \frac{2E}{1-\upsilon^2}\left(\frac{t}{D}\right)^3$$

$P_E$=elastic collapse external pressure, psi
t=pipe wall thickness, in
D=pipe outer diameter, in
E=Young's modulus, psi
v=Poisson ratio

At small D/t (thick pipe) buckling results from yielding of the cross section. Yielding occurs at a pressure $P_Y$, given in Chapter 5 as

$$P_Y = 2\frac{t}{D}S_{Yh}$$

$P_Y$=external pressure at yielding, psi
$S_{Yh}$=minimum material yield strength in the hoop direction, psi

At intermediate values of D/t the buckling regime transitions from elastic collapse $P_E$ to yield $P_Y$, with a collapse pressure [Murphey]

$$P_C = \frac{P_Y P_E}{\sqrt{P_Y^{\,2} + P_E^{\,2}}}$$

$P_C$=collapse pressure, psi

Therefore, a subsea pipeline of diameter D and thickness t, can resist—without buckling—a hydrostatic pressure equal to $P_C$. If the same pipeline contains a buckle, this buckle will propagate if the hydrostatic pressure exceeds a value $P_P$, given by [Fowler, Bai]

$$P_P = 24\frac{S_{Yh}}{(D/t)^{2.4}}$$

$P_P$=buckle propagation pressure, psi

As an example of deep-water pipeline, at a depth of 6500 ft the external hydrostatic pressure on the pipeline is approximately 2850 psi. If the outer diameter of the pipeline is 12.75" and the minimum yield stress is 52,000 psi (API 5L X52), then the wall thickness would have to be at least 0.5" to avoid buckling ($P_C$ >2850 psi). The same pipe will necessitate a wall thickness of at least 1" to prevent buckle propagation ($P_P$>2850 psi). This example illustrates that in deep waters it becomes uneconomical to size a pipe to resist buckle propagation. Instead, the wall thickness may be selected sufficiently large to prevent buckle formation in the first place, and buckle propagation, should it occur, would be mitigated by the use of buckle arrestors.

The length $L_B$ and thickness $t_B$ of buckle arrestors of diameter $D_B$ can be calculated from [Bai]

$$P_h - P_p = (P_a - P_p)[1 - \exp(-15\frac{t_B L_B}{D_B^2})]$$

$P_h$=hydrostatic pressure, psi=ρ g ($h_{max}+h_t+h_s$)
ρ=sea water density, lbm/in$^3$
g=gravity, in/sec$^2$
$h_{max}$=maximum depth for pipe thickness, in
$h_t$=tidal amplitude, in
$h_s$=storm surge, in
$P_a$=propagating pressure for buckle arrestor, psi
$t_B$=thickness of buckle arrestor, in
$L_B$=length of buckle arrestor, in
$D_B$=diameter of buckle arrestor, in

$$P_a = 34\frac{S_{Yh}}{(D_B/t_B)^{2.5}}$$

If the pipeline cross section is not perfectly round, then the collapse pressure can be obtained from

$$P_C = \frac{A}{2} - \sqrt{\frac{A^2}{4} + P_E P_Y}$$

$$A = P_Y + P_E + 3P_E\delta\frac{D}{t}$$

δ=initial ovality of cross section=100 $[(D_{max}-D_{min})/(D_{max}+D_{min})]$

If the pipeline is subject to longitudinal stresses concurrent with the external pressure, then the equation for $P_C$ may still be used, with an adjusted value $S_{Yh}$' for the hoop yield stress [Fowler]

$$S_{Yh}' = S_{Yh}\left[\sqrt{1 - \frac{3}{4}\left(\frac{\sigma_L}{S_{YL}}\right)^2} - \frac{1}{2}\frac{\sigma_L}{S_{YL}}\right]$$

$S_{Yh}$'=adjusted hoop stress, psi

$\sigma_L$=longitudinal stress in pipe wall, psi

$S_{YL}$=longitudinal yield stress, psi

A pipeline can be subjected to displacement limited bending stresses (bending stresses due to an imposed curvature of the line) concurrent with the external pressure, for example when bending the pipe around a reel, or if the pipeline is entrained by the sea bed over a finite distance. In this case, Murphey and Langner proposed the following criterion for avoiding collapse [Murphey]

$$\frac{P}{P_c}+\frac{D^2\kappa}{t}\le 1$$

P=external hydrostatic pressure, psi

$\kappa$=curvature, 1/in=1/R

R=radius of curvature due to bending, in

If the pipeline is subjected to load controlled (as opposed to displacement limited) bending stresses, then the Von Mises equivalent stress $S_e$ must be kept low, below the elastic limit, by applying the rules of the applicable design code [ASME B31.4, ASME B31.8, or BS 8010]. With the bending stresses kept low, the collapse pressure is not significantly different than the case without bending

$$S_e=\sqrt{\sigma_h^{\,2}+\sigma_L^{\,2}-\sigma_h\sigma_L+3\tau^2}\le f_d S_Y$$

$S_e$=equivalent stress, psi

$\sigma_h$=hoop stress, psi

$\sigma_L$=longitudinal stress, psi

$\tau$=shear stress, psi

$f_d$=design safety factor (refer to applicable code)

$S_y$=yield stress, psi

## 13.5 PIPE LOWERING

The first significant load on the pipeline is experienced while joining and lowering the pipeline to the seabed. These installation stresses may even control the pipeline design. If the lowering operation is not properly designed, planned and controlled it may damage the pipe or its coating. Concrete coating is particularly sensitive to overbending during the lowering operation [API 17A]. Figure 131 illustrates several methods for lowering a pipeline:

S-curve: The pipeline is assembled horizontally on the lay vessel and lowered with a smooth S shape.

Tow: The pipeline is assembled on shore then towed on buoys to its final location and sunk in place

J-curve: The pipeline is assembled vertically on the lay vessel and lowered with a smooth J shape.

Reel: The pipeline is assembled and rolled on shore and then unrolled from the reel onto the seabed. While this is a common technique for pipelines made of soft materials such as polyethylene, it is also used for carbon and alloy steel pipes up to 12” in diameter [Kenawy, Crome]. As it is lowered to the seabed, the radius of curvature of the pipeline profile should be sufficiently large to prevent buckling. Several formulas have been proposed to calculate the minimum radius of curvature of a pipeline to avoid buckling. First, we note that the maximum bending strain in the curved pipe is

$$\varepsilon = \frac{D}{2R}$$

ε=maximum bending strain

D=pipe outer diameter, in

R=radius of curvature of bent pipe, in

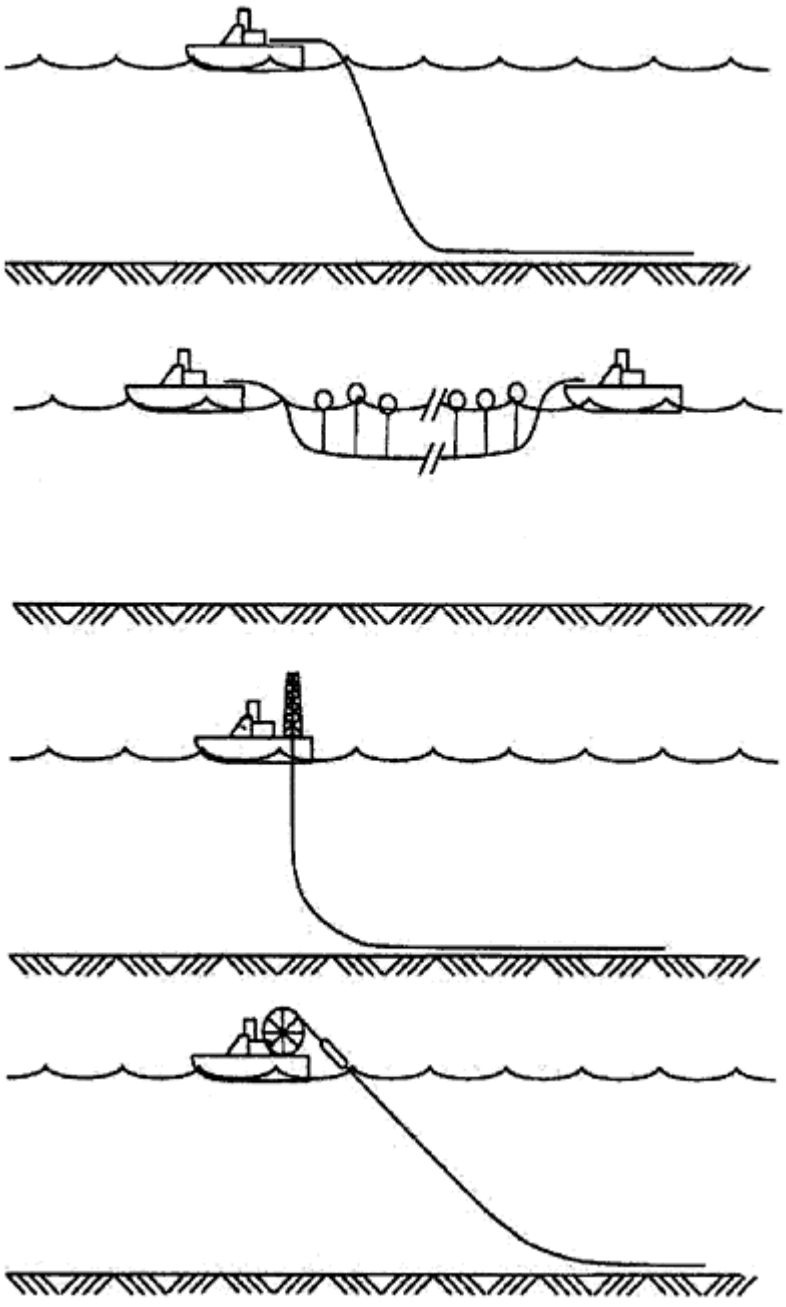

**Figure 13–1** Methods for Lowering Pipelines to the Seabed

In the absence of significant external pressure, buckling can be avoided if the maximum bending strain is limited to the following values [Plonski]

$\varepsilon_{TL}<\min\{3(t/D)^2; 1\%\}$

$\varepsilon_{DC}<\min\{6(t/D)^2; 1.5\%\}$

$\varepsilon_{LD}<\min\{15(t/D)^2; 2\%\}$

$\varepsilon_{TL}$=total longitudinal strain

$\varepsilon_{DC}$=displacement controlled strain

$\varepsilon_{LD}$=local discontinuity strain

D=pipe outer diameter, in

t=pipe wall thickness, in

For the purpose of pipe lay operations, the design strain limit is reported to be 0.2% [Bai].

In the presence of external pressure, the rules of section 13.4 would apply. A difficulty arises at field coated welds: unlike mill-applied insulation and weight coating, which is thick and stiff, the weld cover coating applied in the field or lay vessel tends to be more flexible. As the pipe is reeled or lowered it will have a tendency to hinge (and therefore buckle) around the field joints, which are more flexible than the adjacent pipe and coating. This condition can be investigated by elastic-plastic finite element analysis to determine if and where buckling may occur and establish preventive measures [Crome].

## 13.6 ON-BOTTOM STABILITY

### 13.6.1 Objective

Offshore pipelines are subject to wave and current forces that can cause the pipeline to shift on the seabed [Zhang]. This is a particular concern near well connections, platform risers, and seabed discontinuities. In these critical locations the pipe has to be stable on the seabed. For other locations, where stability is not essential, it may still be necessary to limit pipe movement to 10 times or 20 times the diameter to avoid buckling [Plonski]. The pipeline must therefore be designed to remain stable under the most adverse combination of its lightest weight (lowest density of contents) and the most severe waves and currents (such as a combination of 100 year return period for waves with a 10 year return period current, and vice-versa). The analysis may show that the pipe is unstable on the sea bed, and it may be necessary to stabilize the line with concrete coating, rock cover, sand bags, stabilizing mattresses, trenching, or tie-down anchors.

### 13.6.2 Static Analysis

The criterion for on-bottom stability, illustrated in Figure 13-2, can be written as [Damgaard, Bai]

$F_H<\mu(W_S-F_L)+F_P$

$F_H$=sum of the horizontal drag and inertia forces on the pipe, lb/ft

$\mu$=friction coefficient

$W_S$=submerged weight of the pipeline, lb/ft
$F_L$=hydrodynamic lift force, lb/ft
$F_P$=passive soil friction due to embedment and trenching, lb/ft

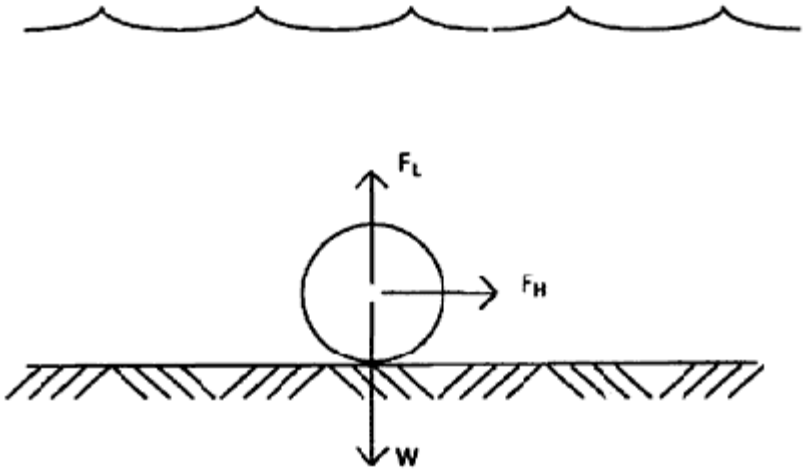


**Figure 13-2** Forces Applied to Subsea Pipeline

Following API 1111, the friction coefficient μ is 0.5 to 0.7 for a sandy sea-bed, 0.5 for gravel, and 0.3 to 0.6 for clay [API 1111]. BS 8010 provides lateral friction factors for North Sea applications of 0.5 to 0.9 for a non-cohesive bed (sand) and 0.3 to 0.75 for a cohesive bed (clay) [BS 8010].

The horizontal force $F_H$ is comprised of the drag force $F_D$ and the inertia force $F_I$, with $F_H(t)=F_D(t)+F_I(t)$

$$F_D(t) = \frac{1}{2}\rho_m D C_D U(t)|U(t)|$$

$$F_I(t) = \frac{\pi D^2}{4}\rho_m (C_M - 1)a(t)$$

$F_H$=total horizontal force, lb
$F_D$=drag force, lb
$\rho_m$=mass density of water, lbm/ft$^3$
D=pipe outer diameter (including coating), ft
$C_D$=transverse drag coefficient
U(t)=free stream water particle velocity, ft/sec
$F_I$=inertia force, lb
$C_M$=inertia coefficient
a(t)=free stream water particle acceleration, ft/sec$^2$

The pipe in contact with the seabed is also subject to a vertical lift force $F_L$, with

$$F_L(t) = \frac{1}{2}\rho_m C_L [U(t)]^2$$

$F_L$=lift force, lb

$C_L$=lift coefficient

Typical values for the drag, inertia and lift coefficients can be obtained from standards [API 1111, BS 8010]. Extensive testing conducted on full scale and scaled down models indicate that the coefficients $C_D$, $C_M$ and $C_L$ depend on several factors that can be represented as dimensionless parameters [Justesen, Fredsoe]: e/D the ratio of the gap between the bottom of the pipe and the seabed surface; KC the Keulegan-Carpenter ratio=$U_m$ T/D where $U_m$ is the wave velocity amplitude and T its period; The ratio $U_C/U_m$ of the current velocity to the wave velocity; Re the Reynolds number=U D/v where v is the kinematic viscosity of the water. The lift coefficient $C_L$ in shear flow varies in a non-linear manner with e/D [Fredsoe], Tests have confirmed the horizontal force equations $F_D$ and $F_I$ with $C_D$ between 1 and 2.5 depending on KC and e/D, and $C_M$ between 1 and 5 depending on KC and e/D [Justesen]. However, as written, the drag force $F_D$ does not explain the observed lateral oscillations (in the direction of flow) and vertical oscillations (cross flow) caused by vortex shedding [Torum, Skomedal]. The shedding of vortices due to cross flow around the pipe causes periodic hydraulic forces with a dominant frequency $f_s$ equal to (Chapter 8)

$$f_s = \frac{Sv}{D}$$

$f_s$=vortex shedding frequency, Hz

S=Strouhal number, in the order of 0.2 for Reynolds numbers of 500 to 500,000

V=cross flow velocity at mid span, ft/sec

D=pipe outer diameter, ft

If the frequency of vortex shedding is close to the natural frequency of the immersed pipe span $f_P$, the span oscillation will resonate with the vortex shedding force. Tests on an immersed 18 ft long span of 6" pipe have recorded resonant oscillation amplitudes in the order of 20% of the pipe diameter in the direction of flow (in-line drag), and 150% of the pipe diameter in the vertical direction (cross flow lift) [Humphries]. The vibration amplitude has been correlated to a stability parameter KS [Humphries, King]

$$KS = \frac{2m_e\beta}{\rho_m D^2}$$

KS=stability parameter

$m_e$=linear mass of span, lbm/ft

$\rho_m$=mass density of water, lbm/ft$^3$

$\beta$=damping of pipe span in air

D=pipe outer diameter, ft

If KS>18 the cross-flow oscillation will tend to disappear, and if KS>1.4, the in-line oscillation will tend to disappear. For short spans (L<20D) end effects will also affect the vibration amplitude. The vertical lift force $F_L$ has proven to be more complex than

$F_L=\rho C_L U(t)^2/2$. While this equation always predicts a positive (upward) lift force (the velocity being squared), the lift force has been observed to be negative (downward) for low values of KC. It then becomes positive for intermediate values of KC (increasing as e/D decreases, i.e. the closer the pipe is to the sea bed, the higher the lift force), and finally the lift force converges to zero as KC increases.

## 13.7 PIPELINE FLOTATION

The condition to prevent flotation of a pipeline in water can be written as [AWWA]

$$\frac{\pi}{4}(D^2-d^2)w_p+HD(1-\frac{1}{g_e})w_e \geq (SF)\frac{\pi D^2}{4}w_w$$

D=outside pipe diameter, ft
d=inside pipe diameter, ft
$w_P$=unit weight of pipe material in air, lb/ft$^3$
H=soil cover over pipe, ft
$g_e$=specific gravity of backfill particles
$w_e$=bulk unit weight of dry backfill, lb/ft$^3$
SF=safety factor
$w_w$=unit weight of water, lb/ft$^3$

Flotation can also be avoided by means of cement weight coating (140 lb/ft$^3$). The coating may be (a) sprayed, (b) rolled around the pipe, with one or two layers of embedded steel wire reinforcement, or (c) injected in a layer 1" to 5" thick, with a steel cage. The cement coating also protects the anticorrosion coating from mechanical damage such as boat anchors or fishing gear. The reinforcement should be electrically isolated from the pipe and from cathodic protection anodes.

Other options to avoid flotation, and to limit movement, is to dump rock at various points along the line, place sand filled membranes over the pipeline [McGill], or use commercially available or specially designed anchors to tie the pipe to the sea bed [Anchor Pipe].

## 13.8 FATIGUE DESIGN

Risers and free spanning (uncovered) spans in subsea pipelines are subject to cyclic loads from waves, platform movement, etc. API 17A recommends the fatigue life to be three times the pipeline service life. BS 8010 provides a straightforward method for the evaluation of fatigue life under cyclic loads, in the form

$$\sum_i S_i C_i < 2.1\times 10^6$$

$S_i$=stress range of load condition i, ksi

$C_i$=fatigue life constant (0.0 for $S_i$=0 to 5 ksi, 0.2 for $S_i$=5 to 10 ksi, 0.6 for $S_i$= 10 to 15 ksi, and 1.0 for $S_i$=15 to 18 ksi).

## 13.9 HOOK AND PULL

As indicated in section 13.1, most of the damage from subsea pipeline ruptures is caused by maritime traffic, particularly hook and pull by fishing gear and ship anchors. The route, marking and layout of subsea pipelines, particularly in shallow waters is regulated to minimize the risk of such accidents. Despite the best precautions, hook and pull accidents still occur. In this case, a thorough inspection of the line is in order. The inspection may be guided by an analysis of the accident to help estimate the extent and severity of damage. The analysis model would con-sist of three parts: the local pipe wall stiffness at the point of contact, the ship trawl stiffness, and the pipe-soil stiffness. The indentation of a pipe wall when struck by a force F, ignoring the coating, is [Bai]

$$\Delta = \frac{F^2}{25 S_Y{}^2 t^3}$$

Δ=indentation of pipe wall, in
F=impact force, lb
$S_Y$=yield stress of pipe steel, psi
t=pipe wall thickness, in
Expressed as the energy required by a sharp edge to cause an indentation Δ is

$$E = 25 S_Y t^2 \sqrt{\frac{\Delta^3}{D}}$$

E=impact energy, in-lb
D=pipe outside diameter, in

## 13.10 REFERENCES

AGA, Submarine Pipeline On-Bottom Stability, project PR-178–9333, 1993, American Gas Association.
Anchor Pipe, Screw Piles, Anchor Pipe International, Houston, TX.
API 17A, Recommended Practice for Design and Operation of Subsea Production Systems, American Petroleum Institute, Washington D.C.
API 1111, Design, Construction, Operation and Maintenance of Offshore Hydrocarbon Pipelines, American Petroleum Institute, Washington D.C.
ASME B31.4, Liquid Petroleum Transportation Piping, American Society of Mechanical Engineers, New York, NY.
ASME B31.8, Gas Transmission and Distribution Piping, American Society of Mechanical Engineers, New York, NY.
AWWA M-9–95, Concrete Pressure Pipe, Manual of Water Supply, American Water Works Association, Denver, CO.
Bai, Y., Pipelines and Risers, Elsevier Ocean Engineering Book Series, Vol. 3, Elsevier.
BS 8010, Code of Practice for Pipelines, Part 3, Pipelines Subsea: Design, Construction and Installation, British Standard.
Crome, T., Method Analyzes Buckling for Reeled Insulated Line Pipe, Oil & Gas Journal, February 7, 2000.

Damgaard, J.S., and Whitehouse, R.J.S., Evaluation of Marine Pipeline On-Bottom Stability, Pipeline and Gas Journal, Part I Hydrodynamic Aspects, Part II Soil and Sediment Aspects, March, 1999 and April, 1999.
Fowler, J.R., Langner, C.G., Performance Limits for Deepwater Pipelines, Offshore Technology Conference, OTC 6757, May 6–9, 1991.
Fredsoe, B., Sumner, B.M., Hydrodynamics Around Cylindrical Structures, World Scientific Publishing Co, 1997.
Humphries, J.A., and Walker, D.H., Vortex Shedding of large Scale Cylinders in Sheared Flow, Offshore, Marine and Arctic Engineering Symposium, Volume V, ASME, 1996.
Justesen, P., et. al., Forces on and Flow around Near-Bed Pipelines in Waves and Current, Offshore, Marine and Arctic Engineering Symposium, Volume V, ASME, 1996.
Kenawy, F.A., Ellaithy, W.F., Cost, Lay Method Major Factors in Subsea Coiled-Tubing Pipeline; Oil & Gas Journal, November 1, 1999.
King, R., A Review of Vortex Shedding Research and its Application, Ocean Engineering, Vol. 14, 1977.
McGill, J., Novel Approach to Pipeline Weighting, Water Engineering and Management, April, 2002.
Murphey, C.E., Langner, C.G., Ultimate Pipe Strength Under Bending Collapse and Fatigue, Proceedings Offshore Mechanics and Arctic Engineering Symposium, American Society of Mechanical Engineers, 1985.
NRC, National Research Council, Improving the Safety of Marine Pipelines, Washington, D.C., 1994.
Plonski, T., GL Rules for Subsea Pipelines, Proceedings of OMAE Conference, Volume V Pipeline Technology, 1996, American Society of Mechanical Engineers, New York.
Skomedal, E., Static Calculations of Pipeline Free Spans, International Journal Offshore and Polar Engineering, Vol. 2, No. 2, 1992.
Sotberg, T., et. al., The Superb Project: A New Safety Philosophy for Submarine Pipeline Design, 1996 OMAE Conference, Volume V Pipeline Technology, American Society of Mechanical Engineers, New York, NY.
Torum, A., et. al., Current Induced In-Line Oscillation of Pipelines in Free-Spans, Proceedings of the 1996 International Conference OMAE, American Society of Mechanical Engineers, New York, NY.
Zhang, J., ed., Ocean Wave Kinematics, Dynamics and Loads on Structures, Proceedings of the 1998 International OTRC Symposium, American Society of Civil Engineers, Reston, VA.

# 14
# Buried Pipe

## 14.1 TO BURY OR NOT TO BURY

The decision to bury a pipe or place it above ground depends on several factors: on one hand, a buried pipe (a) reduces plant congestion, (b) allows for the shortest route (fewer bends) from point to point, (c) avoids existing above ground obstructions, (d) is protected from ambient temperature changes, (e) is protected from wind loads, and (f) if buried deeply, is protected from surface traffic and activities. In certain cases, burying the pipe may be the only viable alternative.

On the other hand, a buried pipe (a) has unique corrosion challenges that may dictate the use of coating and cathodic protection, (b) requires more elaborate repairs, with the need to locate the pipe, locate the leak, open the trench or resort to specialized trenchless repair techniques, (c) can be accidentally damaged by digging, (d) may leak for some time before the leak is detected, (e) requires careful trenching and backfill to avoid excessive soil settlement, and (f) has to be designed for soil and surface loads, which requires a good understanding of the soil condition and properties. In certain plant yard applications, burying the pipe has proven to be a costly decision, and it is not uncommon, after years of corrosion and leakage, to see buried plant piping abandoned in place, and replaced by above ground systems.

Buried pipes (either buried underground or covered by an embankment) can experience a broad range of loads that must be accounted for in design. Normal service loads include internal pressure, constrained expansion or contraction due to changes in fluid temperature, soil weight, surface traffic, and normal soil settlement. Abnormal (accidental) loads include large pressure transients (such as waterhammer), large soil settlement (soil failure), and seismic forces.

## 14.2 INTERNAL PRESSURE

The design of buried pipe for internal pressure consists in selecting the minimum required wall thickness, given the material, diameter, design pressure, and corrosion allowance. The equations for pressure design are given in the applicable ASME B31 code for pressure piping, or AWWA for water works systems. The wall thickness sizing equation for buried pipe is the same as if the pipe was placed above ground (Chapter 4). No credit is taken for the external bearing pressure applied by the soil to counter the internal pressure, no matter how well compacted. Internal pressure will also cause a thrust force in bell and spigot construction. This thrust force must be restrained through tie rods bridging the joint or thrust blocks at changes in direction [DIPRA].

## 14.3 SOIL LOADS

The study of the effect of soil loads on buried pipe dates back to the early 1900's, a time when the first large scale irrigation projects were developed, relying on underground clay tiles to carry and distribute water to fields. In a pioneering study published in 1913, Professor A.Marston presented the experimentally based "Theory of Loads on Pipes in Ditches" and the formula for predicting soil loads on buried pipes [Marston]. More recent publications in this field have confirmed the wisdom of Marston's theory [Moser, Watkins]. Design equations for soil loads can be found in AWWA manuals and standards [AWWA C150, AWWA C900, AWWA M11, AWWA M23]. The simplest design rule for pipes installed in a trench with backfill, is to apply the prism formula, which states that the earth load on the pipe is equal to the weight of the soil prism right above the pipe, as shown in Figure 14-1

$P_v=\gamma H$

$P_v$=earth load pressure on buried pipe, psi

$\gamma$=unit weight of backfill, lb/in$^3$

H=burial depth, in

If the pipe is below the water table, then the soil pressure is reduced by buoyancy and increased by the weight of water

$$P_v = \gamma H - 0.33\frac{h}{H}\gamma H + \gamma_w h$$

h=height of water above pipe, in

$\gamma_w$=unit weight of water, lb/in$^3$

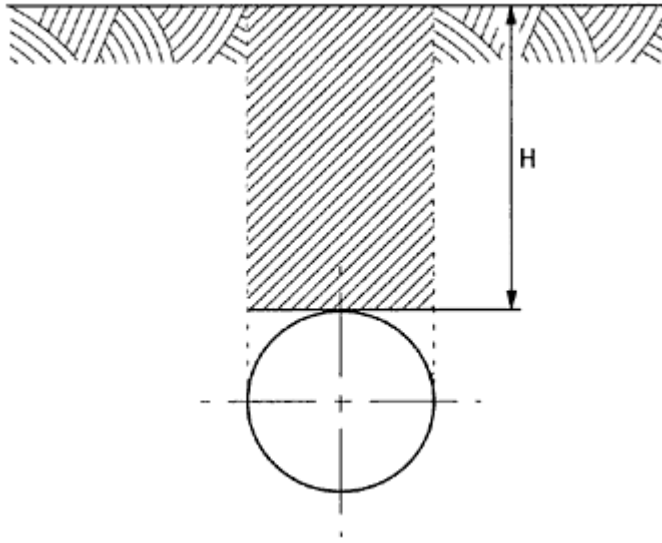


**Figure 14-1** Soil Prism Above Pipe

If, instead of being placed in a ditch with backfill, the pipe is tunneled through undisturbed soil, the earth pressure is lower by a factor 2c (H/D) where c is the soil cohesion [Moser]. Whether in a ditch or tunneled into place, the soil load on steel pressure piping is small. For example, under 10 ft of dry soil ($\gamma$=0.07 lb/in$^3$) the pressure on the pipe is 0.07×120=8.4 psi. Soil loads become important for large diameter thin pipes (large diameter/thickness ratio) as encountered in waterworks (water conduits made of corrugated sheet metal) and for materials with limited ductility, prone to fracture under external loads (concrete, cast iron).

## 14.4 SURFACE LOADS

Buried pipes crossing highways, runways, railroad tracks, construction sites, are exposed to loads due to the passage of heavy surface traffic. The pressure transmitted to the buried pipe by a surface load is [ALA, Moser, WRC]

$$P_p = 0.48 \frac{P_s}{H^2 \left[1 + (d/H)^2\right]^{2.5}}$$

$P_S$=surface load, lb

d=offset distance from surface load to buried pipe, in

For example, an 180,000 lb surface load right above a pipe (d=0) will cause a pressure of 6 psi if the pipe is buried 10 ft below ground. Under the effect of soil and external loads, the buried pipe will tend to ovalize, causing through-wall bending stresses, with [ALA]

$$\sigma_b = 4E \frac{\Delta}{D} \frac{t}{D}$$

$$\frac{\Delta}{D} = \frac{0.15P}{\frac{EI}{R^3} + 0.061E'}$$

$\sigma_b$=through-wall bending stress, psi

E=modulus of elasticity of pipe, psi

$\Delta$=change in pipe diameter due to ovalization, in

D=pipe diameter, in

t=pipe wall thickness, in

EI=pipe wall stiffness, in-lb

E'=modulus of soil reaction, psi [AWWA C150]

For example, a downward pressure of 6 psi on a carbon steel pipe buried in poorly compacted soil will cause it to ovalize 0.8%, with a through-wall bending stress of 14.5 ksi. Where the surface load is both significant and repetitive, fatigue considerations may dictate a deeper soil cover. This subject has been extensively investigated for steel pipelines crossing highways and railroad tracks, and techniques have been developed to predict fatigue life and minimum depth of cover in this case [API 1102].

## 14.5 THERMAL EXPANSION AND CONTRACTION

When the fluid temperature conveyed in a buried pipe differs from the soil temperature, the pipe will tend to contract or expand. In a straight pipe, fully restrained by the surrounding soil, unable to expand or contract, the temperature change will cause an axial stress [B31.4, B31.8, ALA]

$$\sigma_A = E\alpha(T_2 - T_1) - \upsilon\frac{PD}{2t}$$

$\sigma_A$=axial stress in a pipe fully constrained by the surrounding soil, psi
E=modulus of elasticity of pipe material, psi
α=coefficient of thermal expansion of pipe 1/°F
$T_2$=fluid temperature, °F
$T_1$=burial installation temperature, °F
v=Poisson ratio of pipe material
P=internal pressure, psi
D=pipe diameter, in
t=pipe wall thickness, in

The situation is different if the buried pipe contains a bend, and is not assumed to be fully restrained by an infinitely stiff soil at the bend. In this case, the pipe will tend to flex around the bend, with the surrounding soil exerting a restraining force. The pipe acts as a beam on elastic foundation, as illustrated in Figure 14-2. The hand calculation of stresses is only possible in the simplest of configurations [ASME B31.1, ALA, WRC 425].

In most cases, it will be necessary to analyze the expansion or contraction around the bend using a pipe stress analysis program, with the soil modeled as spring elements around the pipe. The stiffness and spacing of soil springs depends on the soil and compaction properties [ALA, WRC 425, ASCE]. The bending stresses in the pipe are calculated and compared to an allowable stress, such as defined in Appendix VII of ASME B31.1.

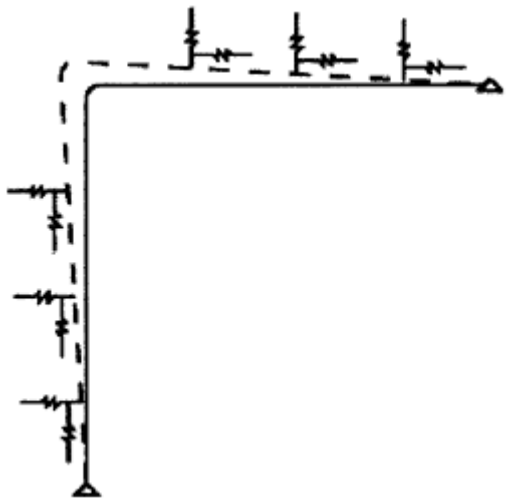

**Figure 14-2** Partially Restrained Expansion at a Buried Pipe Bend

If the temperature rise in a pipeline becomes significant, the compressive stress in the line will increase and could cause the pipe to buckle up, what is referred to as upheaval buckling. The compressive force in the buried pipeline is

$$N = \sigma_A A_{pipe} = [E\alpha(T_2 - T_1) - \upsilon \frac{PD}{2t}]\pi Dt$$

The critical compressive buckling force in a perfectly straight pipeline is [Friedman, WRC 425]

$$N_{critical,perfect} = 2\sqrt{EIk_e}$$

$N_{critical,perfect}$=compressive buckling force for a perfectly straight buried pipe, lb

E=pipe material Young modulus, psi

I=pipe cross section moment of inertia, $in^4$

$k_e$=stiffness of soil cover, lb/in

The critical compressive buckling force in a real pipeline with an initial curvature is a fraction of

$N_{critical,actual}=\lambda N_{critical,perfect}$

$N_{critical,actual}$=compressive buckling force for an initially deformed pipeline, lb

$\lambda$=fraction that depends on initial curvature of the pipeline

The uplift resistance of the soil can also be expressed in terms of force per linear foot of pipeline. In a cohesionless soil [Schaminee]

$$P = \gamma HD(1 + f_d \frac{H}{D})$$

$\gamma$=soil density, lb/in$^3$
H=burial depth, in
D=pipeline diameter, in
$F_d$=load factor 0.6 for gravel or rock dump, down to 0.15 for very loose soil
For cohesive soils

$$P = DC_u(1 + f_C \frac{H}{D}) \le 5.14DC_u$$

$C_u$=shear strength of soil, psi

## 14.6 GROUND MOVEMENT

Ground movement (either a gradual settlement or spread, or a sudden failure due for example to a landslide, an earthquake or mining operations) could cause a buried pipe to fail by plastic tension or by compressive buckling. The assessment of ground movement consists of two parts: first, the prediction of the deformed pipe profile; second, the resulting stresses or strains in the deformed pipe. The first part, predicting the pipe profile, is not a simple proposition. The civil engineer must estimate the magnitude of movement and the distance over which it will take place. Given the soil deformation profile, the stresses or strains in the pipe can be estimated by computer analysis or hand calculation. A computer analysis will generally consist of an elastic-plastic model of the pipe, restrained by non-linear soil springs, with the ground movement imposed at the base of the soil springs [ALA, ASCE]. The stresses are obtained directly as output. An elastic analysis, in which the total stress in the pipe is kept below a fraction $F_D$ of the material yield stress, can be accomplished by hand calculations. A pipe settlement X would be judged acceptable if it occurs over a distance at least equal to L, where [API 1117]

$$L = \sqrt{\frac{3.87 \times 10^7 DX + 7.74 \times 10^7 X^2}{F_D S_Y - S_E}}$$

L=minimum required length ($L=2L_1$ in Figure 14-3), ft
D=outside pipe diameter, in
X=mid-span deflection, ft
$F_D$=design factor
$S_Y$=minimum yield stress of pipe material, psi
$S_E$=longitudinal stress in pipe prior to ground movement, psi

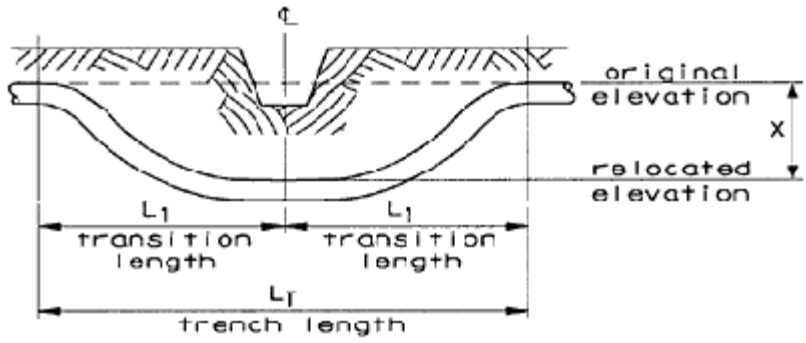


**Figure 14-3** Mid-Span Deflection

The profile, along the pipe should be at least as gradual as that given by [API 1117]

$$X_a = \frac{16a^2X(L-a)^2}{L^4}$$

$X_a$=vertical deflection, at a distance a, ft

a=distance along the trench from origin of deflection, ft

In addition to limiting the stress to $F_D S_Y$, the strain on the compressive side of the bent pipe should be less than the buckling strain [WRC 425]

$$\varepsilon_b = \frac{4\Delta D}{L^2} = \frac{D\kappa}{2} < 2.42\left(\frac{t}{D}\right)^{1.6}$$

$\varepsilon_b$=maximum compressive strain in bent pipe

$\Delta$=maximum bow at mid-span, in

D=pipe outside diameter, in

L=length of pipe segment, in

$\kappa$=curvature of bent pipe (1/R where R is the radius of curvature), 1/in

## 14.7 SEISMIC

Earthquakes can fail buried pipes in one of two ways: (1) a large ground movement that fails the pipe by tension (particularly at corroded sections, poor weld joints and mechanical joints) or by compressive buckling, and (2) a large cyclic movement caused by the passage of the seismic wave. The effect of ground movement can be analyzed following the rules of Section 14.6. It has been argued that wave passage alone could not fail modern (arc welded), well constructed (fabrication, NDE and hydrotest per ASME B31 code), and well maintained (little corrosion) steel pipe. Where wave passage must be analyzed, the upper bound of the strain in the pipe can be obtained by assuming that it is equal to the soil strain caused by wave passage [ALA, ASCE]

$$\varepsilon_a = \frac{V_g}{\alpha C_s}$$

$\varepsilon_a$=soil strain
$V_g$=peak ground velocity due to wave passage, ft/sec
$\alpha$=factor 2 for shear waves, 1 for other seismic waves
$C_S$=apparent propagation velocity for seismic waves, 6560 ft/sec

## 14.8 REFERENCES

ALA, American Lifelines Alliance, Guidelines for the Design of Buried Steel Pipe, American Society of Civil Engineers, Reston, VA.
API 1102, Steel Pipelines Crossing Railroads and Highways, American Petroleum Institute, Washington, D.C.
API 1117, Movement of In-Service Pipelines, American Petroleum Institute, Washington, D.C.
ASCE Guidelines for the Seismic Design of Oil and Gas Pipeline Systems, American Society of Civil Engineers, Reston, VA.
ASME B31.1, Power Piping, American Society of Mechanical Engineers, New York.
ASME B31.4, Pipeline Transportation Systems for Liquid Hydrocarbon and Other Liquids, American Society of Mechanical Engineers, New York.
ASME B31.8, Gas Transmission and Distribution Piping Systems, American Society of Mechanical Engineers, New York.
AWWA C150, Thickness Design of Ductile-Iron Pipe, American Water Works Association, Denver, CO.
AWWA C900, PVC Pressure Pipe 4-in Through 12-in for Water Distribution, American Water Works Association, Denver, CO.
AWWA M11, Steel Pipe, American Water Works Association, Denver, CO.
AWWA M23, PVC Pipe—Design and Installation, American Water Works Association, Denver, CO.
DIPRA, Thrust Restraint Design for Ductile Iron Pipe, Ductile Iron Pipe Research Association, Birmingham, AL.
Friedman, Y., Debouvry, B., Analytical Design Method Helps Prevent Buried Pipe Upheaval, Pipe Line Indutsry, November, 1992.
Marston, A., and Anderson, A.O., The Theory of Loads on Pipes in Ditches, and Tests of Cement Clay Drain Tile and Sewer Pipe, Bulletin 31, Iowa Engineering Experiment Station, Iowa State University, Ames, Iowa, 1913.
Moser, A.P., Buried Pipe Design, McGraw Hill, New York.
Schaminee, E.L., et. al., Soil Response for Pipeline Upheaval Buckling Analyses: Full-Scale Laboratory Tests and Modelling, 22nd Offshore Technology Conference, Proceedings, Volume 4, OTC 6486, 1990
Watkins, R.K., Anderson, L.R., Structural Mechanics of Buried Pipes, CRC Press, New York.
WRC 425, Welding Research Council Bulletin 425, A Review of Methods for the Analysis of Buried Pressure Piping, G.Antaki, Pressure Vessel Research Council, New York.

# 15
# Welding

## 15.1 SHOP AND FIELD WELDING

Welding is a joining process in which the parts to be joined are heated above their melting temperature. The bond between the two parts is metallurgical. The melted parts may be joined with a filler metal also melted in the joining process, or without a filler metal (autogenous welding). The term welding also applies to thermoplastics, which can be heated, melted and fused together, typically by compressing the two heated ends. Brazing applies to a joining process in which a filler metal is melted at a temperature above 840°F but generally below the melting temperature of the parts to be joined. The molten filler metal is deposited at the junction between the two solid parts. The bond is formed by diffusion of braze metal into the parts being joined. Soldering is similar to brazing, but the filler metal (solder) melts at a temperature below 840°F (tin, lead and zinc melt below 840°F, Table 15-1). The bond is formed by adhesion of melted solder between the parts. There are several welding processes: arc welding, resistance welding, flash welding, oxyacetylene torch welding, friction welding, electron or laser beam welding [Weisman]. For the latest knowledge on welding the engineer should refer to research and publications from institutes or societies such as the Edison Welding Institute (EWI, Columbus, OH), the Welding Institute (TWI, UK), the Pressure Vessel Research Council (PVRC), or the American Welding Society (AWS, Miami, FL).

Welding of piping and pipelines takes place first in the pipe mill, where skelp (sheets of steel) are rounded and seam welded, longitudinally or spirally. This pipe mill weld is a seam weld. Second, pipes and components (fittings, valves, etc.) are welded together in the shop or in the field. The shop or field weld is generally a girth weld (weld around the circumference) or a fillet weld (for example at socket welded joints, or when welding an external attachment to the pipe). A multi-pass girth weld consists of a root pass (from the inside or outside diameter), a hot pass (root cover), and before the root pass completely cools, one or several fill passes, and a wider cap pass.

A common welding technique for shop or field welding metallic piping systems is arc welding. In arc welding, the pipe metal, and the filler metal, if used, are melted by the heat of an electric arc and then solidify to form the weld joint. The arc is formed by the flow of current between the parent metal (work-piece) and an electrode, which can be consumable (melts and becomes part of the joint) or non-consumable (having a higher melting temperature than the arc, the electrode—typically tungsten or carbon—does not melt).

**Table 15-1** Approximate Melting Temperatures

| |
|---|
| Tin: 450°F |
| Lead: 600°F |
| Zinc: 790°F |
| Aluminum alloys: 1000°F |
| Magnesium alloys: 1100°F |
| Copper, brass, bronze: 1800°F |
| Copper-nickel: 1900°F |
| Gray cast iron: 2100°F |
| Stainless steel: 2600°F |
| Nickel alloys: 2600°F |
| Cr-Mo steels: 2700°F |
| Carbon steel: 2800°F |
| Iron oxide: 2900°F |
| Titanium alloys: 3000°F |
| Zirconium: 3380°F |
| Nickel oxide: 3600°F |
| Chromium oxide: 4100°F |
| Molybdenum: 4730°F |
| Tantalum: 5425°F |

In all cases, the molten pool of metal that constitutes the weld must be shielded from contaminants such as dust; oxygen and hydrogen could lead to inclusions, porosities or cracks in the weld. Shielding is achieved by means of a flux deposit or a gas blanket. The common arc welding techniques for shop and field welding of piping systems and pipelines are shielded metal arc welding (SMAW), submerged arc welding (SAW), gas metal arc welding (GMAW, MIG), flux core arc welding (FCAW), gas tungsten arc welding (GTAW, TIG). The end bevel may be square (typical for 1/8" thin pipe), 60° to 80° single-V or double-V total bevel angle.

## 15.2 WELDING PROCESSES

### 15.2.1 Shielded Metal Arc Welding

Shielded Metal Arc Welding (SMAW) is also referred to as Manual Metal Arc Welding (MMAW) or stick welding, Figure 15-1. The consumable electrode (a stick or rod

typically 9 to 18 in. long) is covered with a metallic sheath (flux) that stabilizes the arc and produces gases that protect the arc from contamination by the atmosphere.

The advantages of SMAW are: it welds most metals, thick or thin parts, it is common, it permits the use of a variety of flux covers, it can be used for all welding positions, the equipment is light and easy to transport in the field, the power supply can be direct current (dc) or alternating current (ac), it requires no gas or water supply, and it can utilize an electrode that matches the base metal or a special alloy electrode.

The shortcomings of SMAW are: the welding process is slowed down since each stick (rod) must be replaced as it is consumed, the slag needs to be cleaned after each pass, and it tends to spatter molten weld metal.

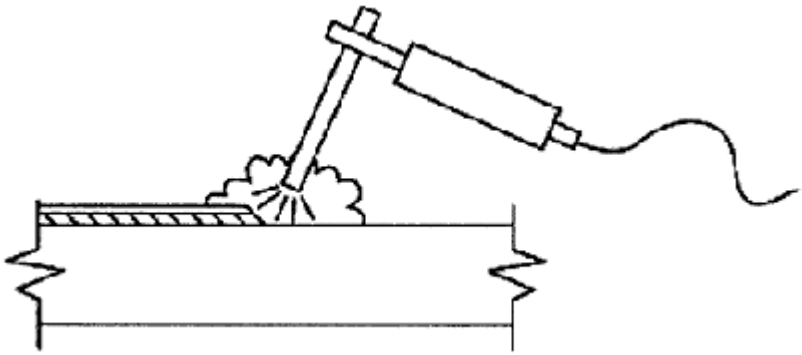

**Figure 15-1** Illustration of Shielded Metal Arc Welding

### 15.2.2 Submerged Arc Welding

In submerged Arc Welding (SAW), the arc between the continuously fed bare metal electrode and the work piece is submerged by the deposition of molten granular flux, which shields the arc from the atmosphere, Figure 15-2. The electrode itself or an additional welding rod serve as filler metal. The flux shields the arc and molten pool from the atmosphere and reduces the cooling rate.

The advantages of SAW are: it can be used to weld large pieces (beams, vessels) because it has a high deposition rate, its deep penetration makes it well suited for thick plates, even with imperfect alignment of weld bevels. Pipe mills commonly employ automated SAW to weld pipe seams.

The shortcomings of SAW are: it is mostly a horizontal (flat) welding technique, the welder can not see the arc that is immediately covered by the flux, the metal stays molten for a longer time (owing to the flux cover) which may require backing, and the welding equipment is large.

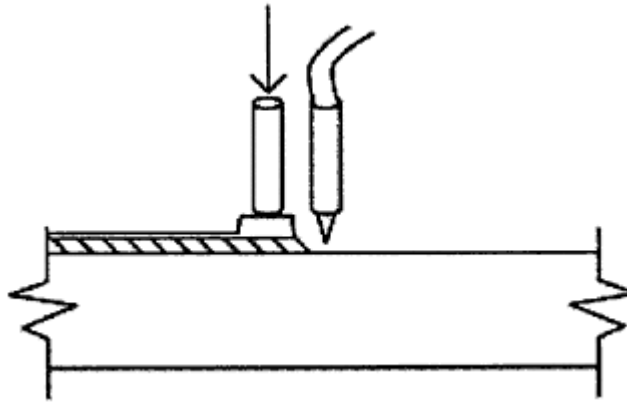

**Figure 15-2** Illustration of Submerged Arc Welding

### 15.2.3 Gas Metal Arc Welding

Gas Metal Arc Welding (GMAW) is also called Metal Inert Gas welding (MIG) when shielded by an inert gas such as helium, and Metal Active Gas welding (MAG) when shielded by a reactive gas such as $CO_2$, Figure 15-3. The electrode is consumable, varies from 20 mils to 3/32" diameter, and is fed through the center of a gas nozzle (torch tips) that continuously supplies a shielding gas such as argon, helium or carbon dioxide. It is mostly used with direct current supply.

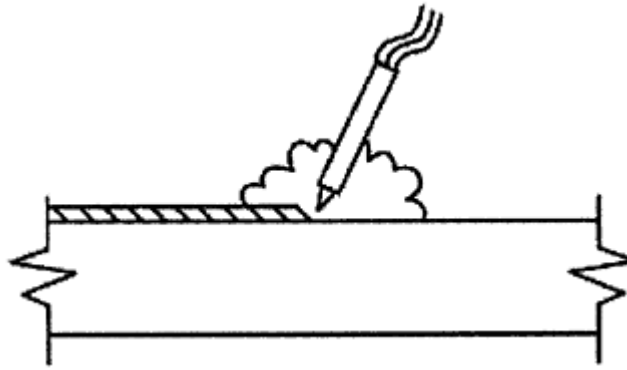

**Figure 15-3** Illustration of Gas Metal Arc Welding

The electrode may be in fast alternating contact with the weld pool (short circuit dip transfer) or kept at short distance from the base metal (globular or spray transfer for all positions).

The advantages of MIG are: it is slag free, it can be used to weld most materials, its continuous feed makes it efficient, and the weld bead can be controlled by the transfer method and the choice of shielding gas.

The shortcomings of MIG are: it requires large equipment, shorts can lead to lack of fusion, it requires a gas source (argon, helium, carbon dioxide), when welding outdoors wind can blow the shielding gas away, and it tends to spatter weld metal.

### 15.2.4 Flux Core Arc Welding

In Flux Core Arc Welding (FCAW), the electrode is consumable and contains in its center either a flux that automatically shields the molten pool (self-shielding FCAW), or minerals or alloys, in which case gas shielding is necessary (gas shielded FCAW), Figure 15-4.

The advantages of FCAW are: it can be used to weld many metals, it can be used in all positions, it has a high deposition rate, and its slag is relatively thin and readily removed.

The shortcomings of FCAW are: the need to chip-off flux at the end of a weld pass, and welding is accompanied by smoke.

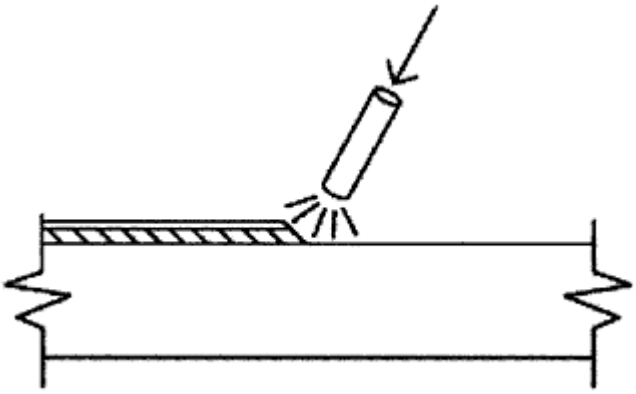

**Figure 15-4** Illustration of Flux Core Arc Welding

### 15.2.5 Gas Tungsten Arc Welding

Gas Tungsten Arc Welding (GTAW) is also called Tungsten Inert Gas welding (TIG). The non-consumable electrode is made of tungsten (pure tungsten, zirconium tungsten, or thoriated tungsten) and is contained at the center of a gas nozzle, Figure 15-5. Welding may be autogenous (no filler metal) or a separate welding rod may be used to supply filler metal. The arc and weld pool are protected by inert gas (welding grade argon or helium). TIG can be used with ac or dc power supply. On carbon steel, TIG is usually limited to thin wall pipe or tubing, and root and hot pass (the first two passes) on standard size pipe.

The advantages of TIG are: the electrode and welding rod being separate can be separately controlled leading to good quality welds (smooth, uniform) in most metals, it does not require weld cleanup, it is spatter free. It is particularly well suited for standard wall pipe, or for a first pass on thick wall piping and pipeline.

The shortcomings of TIG are: it is a relatively slow welding process, the tungsten electrode, which remains solid during welding may chip and form an inclusion, the shielding gas can be affected by wind.

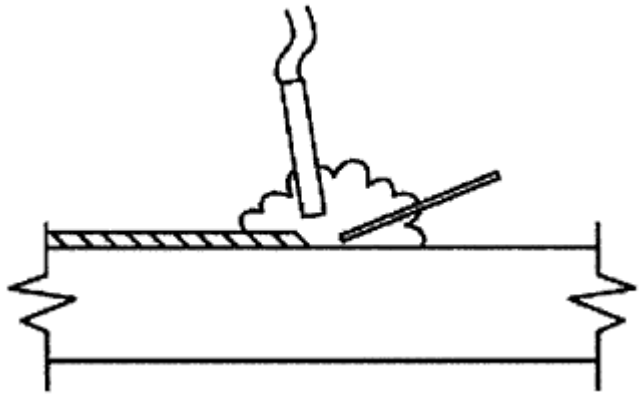

**Figure 15-5** Illustration of Gas Tungsten Arc Welding

### 15.2.6 Welding Parameters

The quality of a weld will depend on the choice of the welding process, the welder's skills and the welding parameters, which include: weld joint design, welding position, fixture, weld backing, composition of filler metal and flux, type of electrode, electrode diameter, welding current (ac, dc, polarity), arc length (electrode-work gap), travel speed, welding technique (oscillatory weave or straight stringer, arc starting and stopping), arc voltage, shielding gas flow, pre-heat, inter-pass temperature control, and post-weld heat treatment. Example of TIG welding parameters:

| | |
|---|---|
| Thickness | 0.25" |
| Joint | 37.5° V bevel |
| Current | 100 amps |
| Polarity | DC electrode negative |
| Voltage | 12 volts |
| Filler metal | ER70S-3 (carbon steel), ER308 (stainless steel) |
| Filler size | 3/32 |
| Shielding gas | Welding grade Argon |

| | |
|---|---|
| Flow rate | 5 $ft^3$/hr (carbon steel) 20 $ft^3$/hr (stainless steel) |
| Nozzle size | 3/8” |
| Nozzle-pipe distance 0.5” | |
| Preheat | ASME B31.3=50°F minimum with SU<60 ksi |
| Post-Weld Heat | ASME B31.3=none for 0.25” wall |
| Welding positions Flat, horizontal, vertical, overhead | |

### 15.2.7 Gas Purging

When making the root pass in alloy steels or non-ferrous alloys, the air contained in the pipe tends to form oxides, hard spots, cracks and porosities. To avoid these problems, the air is purged from inside the pipe with argon, helium or nitrogen. Nitrogen purging is also used to avoid surface discoloration in stainless steel welds.

### 15.2.8 Mechanized Welding

Mechanized welding (automatic or semiautomatic) is commonly used for repetitive welding of large diameter pipe, such as in the case of oil or gas pipelines. Five pass welds in a 36” diameter, 0.5” wall pipeline are reported to be accomplished three times faster with mechanized welding than with manual stick welding [Share].

## 15.3 WELD DEFECTS

### 15.3.1 Weld Metallurgy

Chapter 3 addressed the metallurgy of steel, and the formation of grains while cooling molten metal. A similar process takes place in a weld, where the molten weld pool consists of the molten base metal and filler metal, if used. The solidified molten metal is the weld itself.

The weld is a mini casting with the molten metal poured into the weld bevel. The adjacent base metal, heated to the point of changing its microstructure but not to the point of melting, is the heat-affected zone (HAZ).

Beyond the HAZ is the parent metal, which is unaffected by the weld process. A weldment therefore exhibits three distinct microstructures: the weld itself (a cast microstructure, with weld grains elongated in the direction of the heat flux created while the weld pool solidifies), the heat affected zone (typically larger grains that have been altered by the heat of the welding process), and the base metal (unaffected by the weld process).

Microstructure, grain size and mechanical properties are controlled by the welding technique and parameters, the automatic heat treatment provided by subsequent passes and the heating and cooling rate, and can therefore be influenced by preheat and post-weld heat treatment.

All welds contain discontinuities (indications) detectable by surface or volumetric examination techniques. Where these discontinuities exceed a certain size specified by construction or welding standards, such as API 1104 or ASME B31, these discontinuities are labeled defects or flaws. Figure 15-6 illustrates common weld indications: (a) lack of penetration where the two pipe ends have not fused completely through the thickness, (b) gas porosities or non-metallic or oxide inclusions, (c) root concavity or undercut that may be blunt or sharp, (d) excessive penetration or projection and splatter of metal into the pipe, (e) cracks and sharp crack-like flaws, (f) lack of fusion between consecutive weld passes or between the weld and the adjacent base metal (lack of side wall fusion). Examination techniques used to detect these indications and acceptance criteria are discussed in Chapter 16.

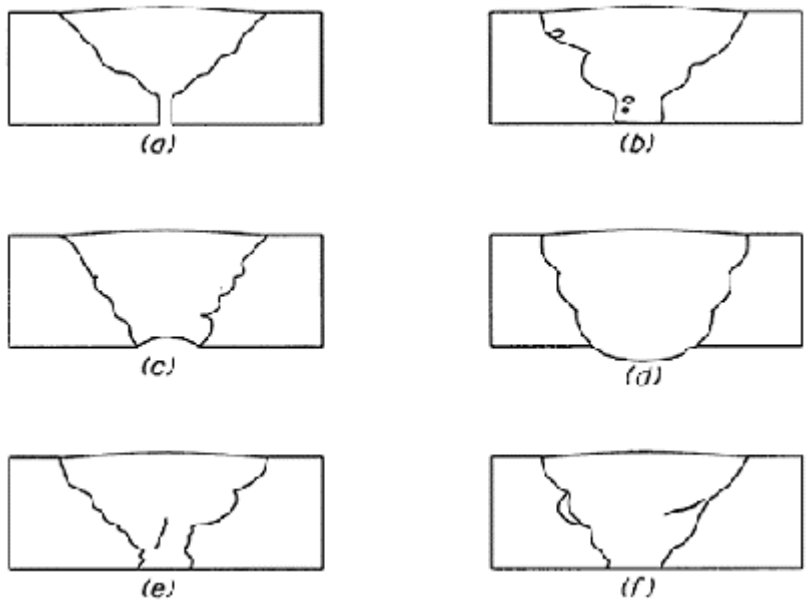


**Figure 15-6** Indications in Pipe Welds

### 15.3.2 Porosities

Porosities are gas pockets or voids trapped in the weld metal (Figure 15-6(b)). For example, porosities can form if atmospheric oxygen mixes with the molten steel to form bubbles of CO gas. This can be prevented by the presence of deoxidizing elements such as silicon or aluminum that tend to deplete the oxygen available for the formation of CO by forming their own oxides $SiO_2$ or $Al_2O_3$. The weld area and therefore its strength are reduced by the presence of porosities, but too much silicon or aluminum also reduces the strength and toughness of steel.

### 15.3.3 Cracks

Cracks are sharp fractures in the weldment (Figure 15-6(e)). They can be formed during the welding process (hot cracks), or can be delayed, appearing hours or days after the weld is completed (cold cracks).

#### *15.3.3.1 Hot Cracking*

Hot cracking can be caused by chemical compounds with a low melting temperature (such as iron sulfides) that are swept towards the grain boundary during the cooling process, causing intergranular cracks, particularly in coarse grain welds. Hot cracking can also form in deep weld beads as the center of the weld pool solidifies last and is pulled apart by the adjacent contracting metal.

#### *15.3.3.2 Delayed Cracking*

Delayed cracking can be caused by a combination of three adverse conditions: (1) large tensile stresses in the weld, (2) local hard martensite spots or inclusions, and (3) gas diffusion in the steel. A classical case of delayed cracking in welds is hydrogen cracking. In this case, water vapor (humidity) in the atmosphere or in the filler metal is decomposed into oxygen and hydrogen during welding. The hydrogen is adsorbed into the carbon steel weld pool and diffuses in the solid austenite formed as the steel starts to cool. But as cooling continues, the austenite transforms to ferrite and carbide where the hydrogen is much less soluble. If cooling is too rapid, the hydrogen is locked in the steel, diffuses and concentrates forming cracks in the weld, often times many hours after the weld has been completed (delayed cracking). This cracking takes place in zones of high hardness where the weld is more brittle, and where the weld is subject to tensile stresses that tend to open the crack lips. Where the presence of hydrogen can not be excluded, it may be necessary to volumetrically examine the weld for hydrogen induced cracking, at the time of welding and again several hours after welding. Low hydrogen electrodes, stored above 250°F (to keep them moisture free) are required where hydrogen induced cracking is a concern.

### 15.3.4 Inclusions

Inclusions are non-metallic particles (such as tungsten from a TIG electrode), slag, impurity or oxide trapped in the weld (Figure 15-6(b)). Because the melting point of oxides is higher than base metal, they form inclusions in the weld. On the other hand, oxides or carbides with a high melting point act as nuclei for grain growth, and therefore lead to the formation of a large number of smaller grains, acting as grain refiners. Other grain refiners include aluminum, nitrogen, vanadium and zirconium.

### 15.3.5 Root Concavity and Undercut

Root concavity (Figure 15-6(c)) is a recess in the weld root and an undercut is a recess at the edge of the weld bead, either on the pipe inner or outer diameter. They are often a consequence of inappropriate welding parameters or technique. A root concavity and an undercut reduce the cross sectional area of the weld metal. An undercut also acts as a stress riser or as a corrosion crevice.

### 15.3.6 Incomplete Penetration

An incomplete penetration (or lack of penetration) occurs if the first weld pass does not fully penetrate the pipe wall to melt its inner diameter (Figure 15-6(a)). Looking at the completed weld from the inside, the edge of the inner diameter weld bevels would still be visible, not fused. An incomplete penetration reduces the weld cross section (its strength), acts as a stress riser (it is actually a circumferential crack), behaves as a corrosion crevice, and could cause local turbulence of the flow stream.

### 15.3.7 Lack of Fusion

Lack of fusion (or incomplete fusion) takes place when, locally, the weld process does not fuse the parent metal or the previous weld pass (Figure 15-6(f)). A lack of fusion acts as an embedded crack.

### 15.3.8 Shrinkage

Girth welds cause the pipe to shrink longitudinally by approximately 1/8” and to peak (angular misalignment). Shrinkage and distortion can be minimized by controlling fit-up and tack welding, by reducing the root opening and the number of weld passes, by increasing the weld speed, and by optimizing the weld sequence. On large diameter cylinders, particularly vessels and tanks, it may be necessary to restrain the part during welding to avoid distortion. This can be done by placing radial beams inside the vessel. The welds may have to be progressively stress relieved afterwards to reduce residual stresses caused by these constraints.

## 15.4 CODES, STANDARDS AND PRACTICE

### 15.4.1 ASME B31 and API 1104

The requirements for welding piping systems and pipelines are specified in the applicable ASME B31 code, API 1104 for pipelines, ASME Boiler & Pressure Vessel Code Section III for nuclear power plant piping, and NBIC NB-23 for repairs. These codes address the following key requirements:

(a) Weld type: with few exceptions, full penetration butt welds are the standard practice for girth welds and branch to header welds. Fillet welds are used for socket fittings, slip-on flanges and external attachments.

(b) Weld size: full penetration welds have a depth equal to the pipe wall thickness. Minimum dimensions of fillet welds are explicitly specified in the applicable ASME B31 code.

(c) Welding process: typically, most qualified welding process or combination of processes which produce welds that meet the procedure qualification requirements are acceptable. In practice, arc welding is the preferred welding process for piping and pipelines.

(d) Qualifications: the qualification of welders and weld procedures (Welding Procedure Specifications WPS, Procedure Qualification Record PQR) follow either the rules of ASME Boiler & Pressure Vessel code Section IX (typical for power and chemical process applications) or API 1104 (typical for pipeline applications), as specified in the applicable ASME B31 code section.

(e) Weld end preparation: the ASME B31 codes will typically address weld bevel profile (U, V or J, single or double groove, angle, flat landing), alignment, root gap, backing rings, transition and mismatch.

(f) Pre-heat and post-weld heat treatment: minimum pre-welding temperatures and post-weld heat treatment (PWHT) temperature, hold time and heating and cooling rate are typically specified as a function of pipe material (P group number or chemistry), wall thickness, and in some cases ultimate strength. The heating method and means of control are left to the user, and can be accomplished by electric induction, electric resistance jacket, fuel fired ring burners, fuel fired torch, furnace, etc. For certain piping alloys in chemical process applications, PWHT is required to be followed by hardness testing to verify the efficiency of the heat treatment process (soundness of the microstructure and adequate toughness) [ASME B31.3, API 582].

(g) Choice of welding consumables: Guidance for the selection of consumables (filler metal) is provided in welding standards [ASME OM, API 582, AWS].

### 15.4.2 American Welding Society

The AWS multi-volume Welding Handbook is a comprehensive treatment of the art and technology of welding [AWS]. It includes:

Volume 1—Fundamentals of Welding

Volumes 2 and 3—Welding Processes

Volume 4—Engineering Applications—Materials

Volume 5—Engineering Applications—Design

The welding of structural (non-pressure boundary) members typically complies with either AWS D1.1 or AWS D1.6. The American Welding Society (AWS) standards of interest to welding piping and pipelines include:

AWS A 3.0—Welding terms and definitions, including terms for brazing, soldering, thermal spraying and thermal cutting.

AWS A 5.01—Filler metal procurement guidelines.

AWS A 5.XX—Series of filler metal specifications.

AWS D1.1—Structural welding code—steel.

AWS D1.6—Structural welding code—stainless steel.

AWS D 10.4—Recommended practices for welding austenitic Cr-Ni stainless steel piping and tubing.

AWS D 10.6—Recommended practices for gas tungsten arc welding of titanium pipe and tubing.

AWS D 10.7—Recommended practices for gas shielded arc welding of aluminum and Aluminum alloy pipe.

AWS D 10.8—Recommended practices for welding of Cr-Mo steel piping and tubing.

AWS D 10.9—Specification for qualification of welding procedures and welders for piping and tubing.

AWS D 10.10—Recommended practices for local heating of welds in piping and tubing.

AWS D 10.11—Recommended practices for root pass welding of pipe without backing.

AWS D 10.12—Recommended practices and procedures for welding low carbon steel pipe.

AWS D 10.13—Recommended practices for brazing of copper pipe and tubing for medical gas systems.

### 15.4.3 Electrode Nomenclature

The nomenclature for weld electrode is defined in AWS standards. By necessity, it is rather complex since it must reflect many different parameters. The following examples illustrate this point:

E7018-A1 (SMAW): E=electrode, 70=70 ksi ultimate strength, 1=all positions, 18=AC or DC current, low H powder cover, A1=C-Mo steel electrode.

F7P6-EM12K (SAW): F=flux, 7=70 ksi ultimate strength, P=PWHT (1150°F 1 hr.), 6=Charpy toughness >20 ft-lb at −60°F, E=electrode, M=medium manganese, 12=0.12% nominal carbon, K=Silicon killed molten steel.

ER-70S-2 (GMAW or GTAW): E=electrode, R=rod, 70=70 ksi ultimate strength, S=bare, solid electrode, 2=composition per AWS.

E70T-1 (FCAW): E=electrode, 70=70 ksi ultimate strength, 0=flat position, T =tubular flux core electrode, 1=$CO_2$ shielding, spray transfer mode, DC current, electrode+

### 15.4.4 Welder and Weld Procedure Qualification

The quality of welding has greatly improved over the years not only because of improvements in technology but also because of the better skills and qualifications of welders. The complaint in 1935 was that "this particular method of fabrication [welding] was frowned upon by the various rating societies, mainly because of the Board of Trade's prohibitions concerning the use of this process in the making of longitudinal joints for steam boilers. Failures in pressure vessels had contributed in no small measure to the widespread suspicion surrounding the welding process applied to pressure vessels, and this could be readily understood when one remembers that anyone could buy a welding set and call himself a welder. As one commentator recently put it: There is nothing to prevent a cook, having discovered a new method of boiling hams, from going to such a person and having a vessel constructed which would be a danger not only to himself but to every other person who came in contact with it" [Davy].

Today, welders and welding procedures undergo a stringent qualification process. A weld procedure specification documents the essential variables of the welding process, including [API 1104]: weld process, pipe or fitting material, diameter and wall thickness, joint design (weld end preparation), filler metal and number of beads, shielding gas or flux, electrical or arc characteristics, travel speed, position and welding direction, pre-heat, and time between passes. To reduce the number of weld procedure qualifications, base metals and electrodes or welding rods are grouped into P groups (base metal) or F groups (electrodes and welding rods). One procedure qualification would apply to all materials within a group. The P groupings are listed in Table 15-2.

**Table 15-2** P Numbers and Groupings for Base Metals

| | |
|---|---|
| P No. 1 | Group 1—Carbon steels $S_U$=40 to 70 ksi<br>Group 2—C-Mn, C-Si, C-Mn-Si steels $S_U$=70 to 75 ksi<br>Group 3—C-Mn-Si steel $S_U$ to 80 ksi |
| P No. 3 | Group 1—Steel Cr>0.75%, alloy≤2%, $S_U$=50 to 65 ksi<br>Group 2—P No. 3 Group 1 but with $S_U$=70 to 75 ksi<br>Group 3—Steel Mn-Mo, Mn-Mo-Si, Ni-Mo-Cr-V, $S_U$ 80 to 90 ksi |
| P No. 4 | Group 1—Steel Cr=0.75 to 2%, alloy ≤3%. $S_U$=60 to 75 ksi<br>Group 2—Steel with Cr, Cu, Ni, Al, Mo, alloy <3%, $S_U$≥60 ksi |
| P No. 5 | Group 1—Steel, Cr-Mo, alloy <5%, $S_U$=60 to 75 ksi<br>Group 2—Steel, Cr-Mo, alloy <10%, $S_U$=60 to 100 ksi |
| P No. 6 | Group 1—Martensitic, 11–13% Cr stainless $S_U$=60 to 70 ksi<br>Group 2—Martensitic, 12–17% Cr stainless<br>Group 3—martensitic, Cr, Cr-Mo, Cr-Mo-Ni<br>Group 4—13% Cr—4% Ni castings |
| P No. 7 | Group 1—Ferritic, 11–13% Cr<br>Group 2—Ferritic, 17–18% Cr |
| P No. 8 | Group 1—Austenitic stainless steel<br>Group 3—17%–20% Cr—4%–6% Ni—6%–9% Mn |
| P No. 9A | Group 1—Ni steels, 2.5% Ni max. |

| | |
|---|---|
| P No. 9B | Group 1—Ni steels, 3.5% Ni max. |
| P No. 10A | Group 1—Mn-V steels |
| P No. 10B | Group 2—Cr-V tubes |
| P No. 10C | Group 3—Cr-Mn-Si plate |
| P No. 10D | Group 4—20% Cr—1% Cu tubes |
| P No. 10B | Group 5—26%–27% Cr alloys |
| P No. 10F | Group 6—Mn-Mo-V, Mn-Cr, Mo-V alloys |
| P No. 10G | Group 7—36% Ni |
| P No. 10H | Group 8—18% Cr-5% Ni-3% Mo alloys |
| P No. 101 | Group 9–26% Cr—1% Mo alloys |
| P No. 11A | Group 1—8–9% Ni alloys<br>Group 2—5% Ni-Mo alloys<br>Group 3—Casings Ni-Cr-Mo-V<br>Group 4—Mn-Mo, Mn-Mo-Ni |
| P No. 21 | Al alloys $S_U$=8 to 18 ksi |
| P No. 22 | Al alloys $S_U$=20 to 32 ksi |
| P No. 23 | Al alloys with Si |
| P No. 25 | Al alloys with $S_U$=33 to 45 ksi |
| P No. 31 | Cu alloys with $S_U$=30 to 45 ksi |
| P No. 32 | Cu alloys with $S_U$=40 to 50 ksi |
| P No. 33 | Cu alloys with Si |
| P No. 34 | Cu alloys with Ni |
| P No. 35 | Bronze |
| P No. 41 | Ni |
| P No. 42 | Ni-Cu alloys |
| P No. 43 | Ni-Cr-Fe, Ni-Cr-Mo-Cb, Ni-Cr-Fe-Mo |
| P No. 44 | Ni-Mo, Ni-Cr-Mo alloys |
| P No. 45 | Ni-Cr-Fe, Ni-Fe-Cr-Mo-Cu, Ni-Fe-Cr-Si alloys |
| P No. 51 | Ti with SU=35 to 50 ksi |
| P No. 52 | Ti with SU=65 ksi min. |
| P No. 61 | Zr |

Samples of completed welds are destructively tested after heat treatment where required by the construction code [API 1104]. Tests include a tensile test to verify that the weld is stronger than the minimum base metal tensile strength, a break test on a notched weld to

verify the soundness of the broken weld face, and a bend test to verify the ductility of the weld (no cracks after bending the weld specimen in a U shape). In some cases, where permitted by the welding code, radiography is substituted for destructive testing.

## 15.5 POST-WELD HEAT TREATMENT

Various post-weld heat treatments are applied to a weld in order to improve microstructure, mechanical properties, resistance to corrosion, resistance to environmental cracking, and reduce residual stresses (Figure 15-7). For shop and filed girth and fillet welds, ASME B31.1 and B31.3 impose post-weld heat treatment on the basis of the material (P group number), wall thickness, carbon content and ultimate strength. The pipeline codes ASME B31.4 and B31.8 impose post-weld heat treatment on the basis of carbon, carbon equivalent and wall thickness.

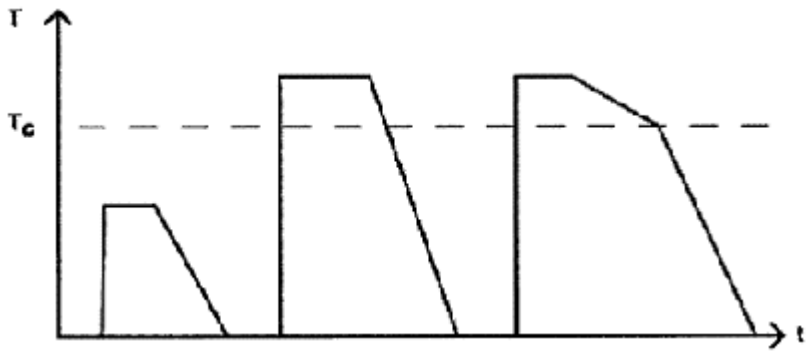


**Figure 15-7** Post-Weld Heat Treatment

Left to Right: Stress Relieving, Normalizing, Annealing

Stress Relief: heating below the critical temperature (the phase transformation temperature, for example the ferritic-austenitic transformation temperature in carbon steel, Figure 3-5). Holding the temperature long enough to reduce residual stresses and then cooling slowly. Stress relieving reduces residual stresses, and generally improves ductility and toughness, and reduces fabrication distortions.

Normalizing (heating above approximately 1550°F, then cooling in still air): “heating a steel object to a suitable temperature above the transformation range and then cooling it in air to a temperature substantially below the transformation range” [ASTM A 941]. Normalizing achieves specified mechanical properties, improves ductility, and homogenizes the material.

Annealing (normalizing but with furnace cooling): “a generic term covering any of several heat treatments” [ASTM A 941]. The objectives of annealing include reducing

hardness, facilitating machining and cold working, producing a desired microstructure, or desired mechanical properties.

Quenching (rapid cooling in oil or water): “rapid cooling” [ASTM A 941]. Quenching hardens the steel, but at the expense of a loss of ductility.

Solution annealing: “Heating to and holding at a suitable temperature to place constituents into solid solution, and then cooling rapidly to hold the constituent in solution” [ASTM A 941]. Typical heat treatment for austenitic stainless steel to place chromium into solution and then cooling rapidly to avoid the precipitation of chromium carbides.

Tempering (stress relieving after quenching or normalizing): “reheating a quench hardened or normalized steel object to a temperature below the transformation range, and then cooling at any desired rate” [ASTM A 941]. The objective of tempering is to restore ductility to steel.

Case Hardening and carbonizing (carburization): Heating carbon steel above the critical temperature in a carbon rich environment. The carbon adsorbed on the surface makes it harder without reducing the ductility of the inner metal.

## 15.6 IN-SERVICE WELDING

It may become necessary to weld on a piping system or pipeline while the line is in service, flowing a liquid or a gas. This is the case for example when making a hot tap, or when depositing a weld overlay to repair a flaw in the pipe wall (Chapter 23) [API 1107, API 2201]. In addition to the usual parameters and cautions considered when welding on an empty pipe, cracking and burn-through are two key concerns when welding in-service.

There are three keys to avoid cracking when welding on in-service steel pipe: (1) avoid hard spots (martensite) which are the source of hydrogen cracking, (2) reduce the risk of intake of hydrogen, and (3) reduce tensile stress. To avoid the formation of martensite, the weld must be cooled slowly; therefore, the heat input should not be too low, a small hot spot will cool too quickly, particularly with the rapid cooling effect of the flowing fluid. To reduce the risk of hydrogen intake, it is necessary to use low hydrogen electrodes and it may also be necessary to pre-heat the in-service pipe to reduce moisture. To reduce tensile stresses, it is often necessary to reduce the operating pressure while welding in order to reduce the tensile hoop stress $PD/(2t)$.

Reducing the operating pressure also reduces the risk of burn-through and blow-out. It is also useful to have some flow in the line, because if the fluid in the line is not flowing, the weld heat will increase the pressure of the trapped fluid to a potentially dangerous level. The heat input must be kept sufficiently low. If the heat input is too large, the weld will burn through the pipe wall, and fluid will burst out under the effect of internal pressure. It is therefore preferable to use a small electrode (3/32” or less for the first pass [API 2201]) on wall thickness below 0.5”. We can see that the heat input must therefore be qualified to a near optimum level, not too low to avoid hydrogen cracking, and not too high to avoid burn-through. In conclusion, before welding in service, it is essential to (1) know the chemistry of the in-service pipe to assure compatible weld procedure and filler, (2) know the thickness of the in-service pipe to qualify the welder and the welding

procedure to the same thickness, (3) know the fluid pressure and flow rate to select the heat input accordingly.

## 15.7 SURFACING TECHNIQUES

Surfacing consists in depositing weld metal on the surface of a base metal, in order to achieve one of several objectives:

(a) Cladding: deposition of a corrosion resistant alloy on a less costly base metal (for example nickel alloy on carbon steel).

(b) Hardfacing: deposition of a hard metal (erosion resistance) on a softer base metal.

(c) Overlay: deposition of a filler metal to replace wall thickness lost by corrosion.

(d) Buttering: deposition of a filler metal in preparation for welding. For example buttering of stainless steel on a carbon steel part in preparation for welding to a new stainless steel component.

(e) Spraying: Particles of molten metal wire are sprayed on the surface of the parent metal using an oxyfuel gun. The sprayed surface is then machined or ground to the required surface finish.

## 15.8 REFERENCES

API 582, Welding Guidelines for the Chemical, Oil, and Gas Industries, American Petroleum Institute, Washington, DC.

API 1104, Welding of Pipelines and Related Facilities, American Petroleum Institute, Washington, DC.

API 1107, Pipeline Maintenance Welding Practices, American Petroleum Institute, Washington, DC.

API 2201, Procedures for Welding or Hot Tapping on Equipment in Service, American Petroleum Institute, Washington, DC.

ASME OM, Boiler & Pressure Vessel Section III, Operation and Maintenance, American Society of Mechanical Engineers, New York.

ASME Boiler & Pressure Vessel Section IX, American Society of Mechanical Engineers, New York.

ASTM A 941, Terminology Relating to Steel, Stainless Steel, Related Alloys, and Ferroalloys, American Society for Testing and Materials, West Conshohocken, PA.

AWS, Welding Handbook, American Welding Society, Miami, FL.

Davy, C.H., Modern Methods of Welding, Transactions of the Institution of Chemical Engineers, Volume 13, 1935, London.

NBIC, National Board Inspection Code, ANSI/NB-23, The National Board of Boiler and Pressure Vessel Inspectors, Columbus, OH.

Share, J., Alliance Pipeline Finishes ’99 on Schedule, within Budget, Pipeline & Gas Journal, January 2000

Weisman, C., ed., Welding Handbook, American Welding Society, Miami, FL.

# 16
# Examination

## 16.1 VISUAL EXAMINATION

In the ASME piping and pipeline codes, examination refers to the actual act of examining materials, components or joints. It is the quality control function (QC). Inspection refers to the verification of activities or documentation. It is the quality assurance function (QA), the oversight. But in practice these words are often used interchangeably.

Of all the examination techniques, visual examination appears to be the simplest, only requiring lighting and a good pair of eyes. In reality, visual examiners must be experienced in many aspects of fabrication, welding and degradation mechanisms, to be able to judge what they see and to call upon additional examination techniques if they detect anomalies. Examiners are classified in increasing order of qualification from VT-1 to VT-3.

Visual examination is used to detect surface defects of base material such as casting porosities, surface fabrication defects such as weld burn through, excessive reinforcement, undercut, root concavity, misalignment, workmanship and visible in-service degradation such as wear, general and local corrosion, pitting, erosion, and large surface cracks.

Pipe Fabrication Institute standard ES-27 [PFI] defines visual examination as examination with the "unaided eye", other than the use of corrective lenses, within 24". A broader definition would include the use of a magnifying glass or remote examination with borescope or camera snaked into the pipe or equipment. The shortcomings of visual examination are evident: it is limited to the component's surface, it requires visual access to the surface, directly or remotely, and it requires an experienced inspector. Visual examination may also be limited by lighting. A light meter can be used to measure the light intensity. Detailed exami-nations may require twice the light intensity needed for general visual examination.

The tools used for visual inspections include ruler, tape, micrometer, gages, $\pi$ tape to measure pipe size, dial indicator, mirrors, flash light, illuminated magnifier, borescope, fiber optic scope, etc.

Historically, visual examination was relied upon in the first issue of the ASME Boiler Code, in 1915, to verify that boilers were "free of injurious effects" and were of good "workmanship" [Cross]. In the 1920's, a remote viewing instrument started to be used for the inspection of cannon bores, and came to be known as borescope. Today, remote visual examinations using fiber optics and crawlers have been adapted to a multitude of field conditions.

## 16.2 MAGNETIC PARTICLES TESTING

In magnetic particle testing (MT) a magnetic field is created on the surface of the part. A yoke is typically used for the examination of piping. The magnetic field is induced by a horseshoe yoke magnetized by ac current circulating in wound coils. The yoke is placed in contact with the pipe surface, generating magnetic lines of force in the pipe, as shown in Figure 16-1. A powder (dry method) or solution (wet method) of magnetic particles is dispersed on the surface. The magnetic particles orient themselves along the magnetic lines. Surface discontinuities become visible as the magnetic lines appear disturbed [Huber].

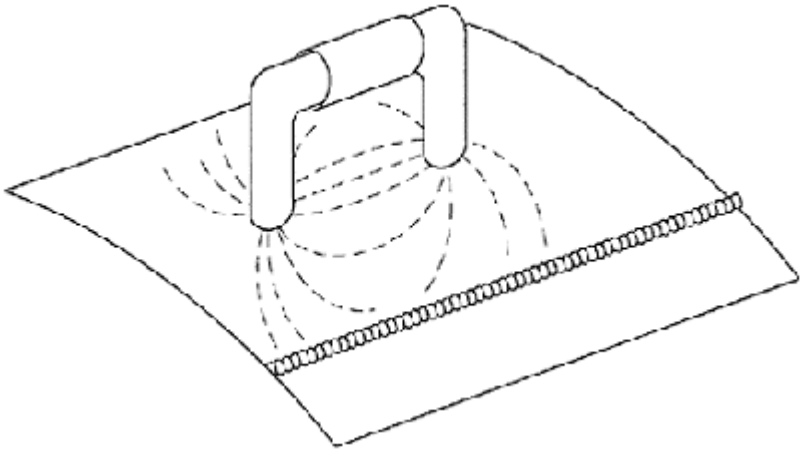

**Figure 16-1** Magnetic Particle Testing Using a Yoke

Other methods of magnetization include the head shot, the longitudinal field and the prod methods. In the case of a head shot, current is passed straight through the part, inducing a circular magnetic field around the part. The longitudinal field method is the reverse of the head shot method. Here, the current passes in a wire wound around the part and a longitudinal field is generated in the part. The last magnetization method, the prod method consists in placing two conductors in contact with the part, and passing a dc current through the part.

Magnetic particles testing is used to detect surface or near surface defects or flaws, including base material flaws such as casting porosities and hot tears, surface cracks, laps and seams in forgings; fabrication flaws, such as surface cracks and incomplete penetration; or in-service degradation such as surface cracks.

The advantages of MT: it can detect subsurface flaws, the flaw does not have to be open (for example, cracks can be filled with rust), and it is protable.

The shortcomings of MT: it is a surface and near surface examination technique, it applies only to ferromagnetic materials, such as carbon steel but not austenitic stainless steel. Discontinuities are better detected if perpendicular to the field, and it therefore becomes necessary to take readings with the magnet placed in at least two perpendicular directions. The surface examined must be smooth to permit the free flow of magnetic particles. The part may have to be demagnetized. The method does not generate a permanent record. Typically, ac wet methods are suited for shallow surface cracks, while dry dc methods are suited for detecting subsurface discontinuities. With non-fluorescent particle examination, it is necessary to have a minimum light intensity.

Magnetic particle testing was first attempted in the 1920's. The technique was introduced into the 1931 ASME Code as a supplement to radiographic testing. The applicable standards are ASME B&PV Section V Article 7, ASTM E 269, and ASTM E 709.

## 16.3 LIQUID PENETRANT TESTING

In liquid penetrant testing (PT) the surface to be examined is cleaned and dried by forced hot air or simply by atmospheric drying. A penetrant is applied to the surface, as shown in Figure 16-2. The penetrant is either a visible dye, usually red, or a fluorescent dye that glows under ultraviolet or black light. The dye is left to soak for a period of time, called dwell time, typically 10 to 15 minutes for steel, as shown in Figure 16-2, top view. Excess penetrant is then removed using a rag lightly sprayed with remover (for solvent removable penetrant) or with water (for water washable penetrant), as shown in Figure 16-2, middle view. A contrasting spray or powder developer is then applied to draw the penetrant into the developer layer by capillary action, as shown in Figure 16-2, bottom view. The capillary suction of the developer must overcome the capillary retention of the crack.

The method detects surface flaws. These include: Base material flaws such as casting porosities and hot tears, surface cracks, laps and seams in forgings, fabrication flaws such as surface cracks, incomplete penetration and porosity, or inservice degradation such as pitting, local corrosion, and surface cracks.

The advantages of PT: it is very sensitive to surface cracks, it can be used on uneven surfaces (nozzles, fillet welds, bolt threads, etc.). It is portable and relatively simple to apply.

The shortcomings of PT: it is a surface examination method and is affected by surface cleanliness, roughness or porosity. It leaves no permanent record. The flaw must be able to accumulate penetrant by capillary action, and must therefore be open to the surface. If the flaw is filled with dirt or oxide, it may not be detected. The surface must be cleaned after penetrant application. Cleaning may be achieved by vapor degreasing or by chemical means [Grendahl, Siegel].

Historically, colored dyes were first used as liquid penetrant in 1938. Fluorescent dyes, first used in advertising in the 1930's, were patented as penetrant in the early 1940's. The applicable standards are ASME B&PV Section V Article 6, ASTM E 165, ASTM E 260, and ASTM E 433 and ASTM E 1417 (replacing MIL-STD-6866).

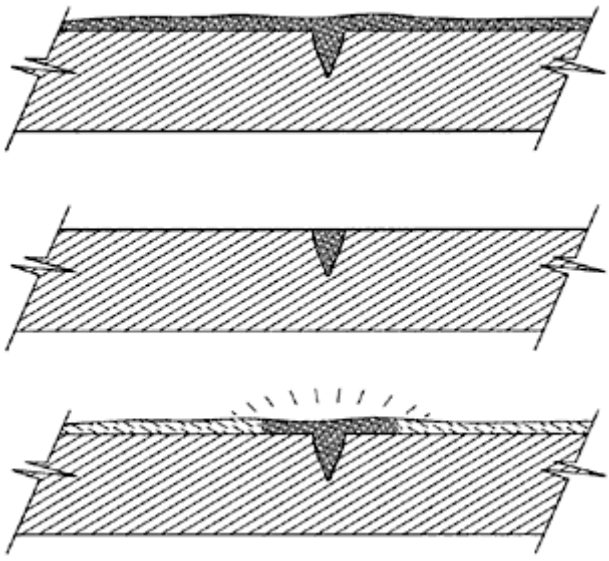

**Figure 16-2** Liquid Penetrant Testing

## 16.4 RADIOGRAPHIC TESTING

In radiographic testing (RT) a source of X-rays or gamma-rays (from radioactive elements such as Iridium 192, or Cobalt 60) is placed on one side of the specimen or pipe wall, and a film or a fluoroscope (for real time examination) is placed on the opposite face, as close as possible to the wall.

There are many ways of placing source and film, as illustrated in Figure 16-3. Whether X-rays or gamma rays are used, the radiation energy and radiographic techniques must achieve a specified quality of resolution. Current developments are in the area of filmless radiography (instant radiographic imaging) where the radiographic image is captured on a phosphor screen then stored on a digital file for immediate viewing and filing.

Radiographic testing equipment is calibrated using a penetrameter with a thickness of about T=2% t where t is the thickness of the weld to be examined, and with holes with diameter 1T, 2T and 4T. For example a *"2–2T"* penetrameter has a thickness of T=2% t and a 2T hole. Real-time non-film radiography has a sensitivity in the order of 2%-2T.

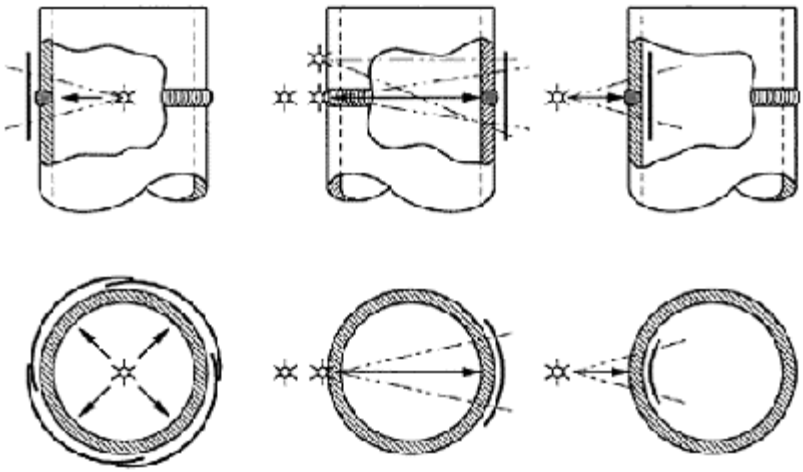

**Figure 16-3** Source and Film Arrangements for Radiographic Testing

The advantages of RT: it is effective for detecting surface and volumetric flaws, including base material flaws such as inclusions and porosities, fabrication flaws such as excessive reinforcement, undercut, incomplete penetration, concavity, porosity, inclusion or in-service degradation such as pitting, local corrosion, and cracks. It results in a permanent radiographic film record.

The shortcomings of RT: because of the use of radioactive materials or X-rays, radiographic testing requires special precautions such as evacuating and roping off the area. Radiographic testing requires access to two sides of the pipe wall. If the shape of the specimen is too irregular, it becomes practically impossible to decipher the radiography; this is the case, for example, at pipe branch welds. RT can locate a crack-like flaw and show its length, but not its depth.

The first reports of the use of X-rays for the examination of weld joints go back to the 1920's where radiography was used in shipbuilding. In 1930, it was used in boiler fabrication shops and, in 1931, it was introduced into the ASME Code. It is the most widely used volumetric inspection technique for pipe butt welds. The applicable standards are ASME B&PV Section V Article 2, ASTM E 94, ASTM E 142, ASTM E 242, ASTM E 747, ASTM E 999, ASTM E 1025, ASTM E 1030, and ASTM E 1079.

## 16.5 ULTRASONIC TESTING

In ultrasonic testing (UT) the surface to be examined is cleaned and made smooth. A couplant (C, Figure 16-4) is applied to the surface and a transducer (T, Figure 16-4) is placed over the couplant. For angle beam examination a solid wedge (W, Figure 16-4) is used to set a fixed beam angle. The transducer emits ultrasonic waves, which are waves

with a frequency between 20,000 Hz and $2010^6$ Hz. Note that subsonic waves have a frequency range below 20 Hz, while sonic waves have a frequency between 20 Hz and 20,000 Hz. The waves emitted by the transducer are either normal to the surface (straight beam) to check wall thickness and detect flaws, or at an angle (angle beam) to detect flaws and particularly cracks. Reflected waves (the pulse echo) indicate the presence of a reflector also referred to as indication [ASTM E 500, Blitz, Carlin, Frederick, McGonagel]. The amplitude and timing of the ultrasonic echo characterizes the size and depth of the indication.

Another UT technique is through-transmission or time of flight diffraction (TOFD). In this case, a discontinuity is detected from the change in amplitude of the ultrasonic wave emitted by a transmitting transducer and collected by a receiving unit.

UT results can be represented as A-scans (distance or time plotted against amplitude of signal, this is typically the raw signal on the UT instrument's CRT screen), B-scans (longitudinal cross section of the wall showing the wall thinning profile), or C-scans (plan view of wall thinning showing contour lines of equal wall thickness).

UT can also be used for other non-destructive examinations, such as measuring the embedment depth of anchor bolts (by shooting a straight beam from the bolt head down to the bolt tip) or measuring the fill height of pipes or tanks.

The advantages of UT: only one side of the component must be accessed, it measures the length and depth of a crack-like flaw, and by use of different beam angles it can be used complex shapes.

The shortcomings of UT: it is not easy to use on complex shapes, it requires a smooth surface and a couplant. Because the transducer covers only a small area, it takes time to scan a large component. Also, multiple flaws may hide each other. UT requires skill and experience. It does not leave a direct permanent record.

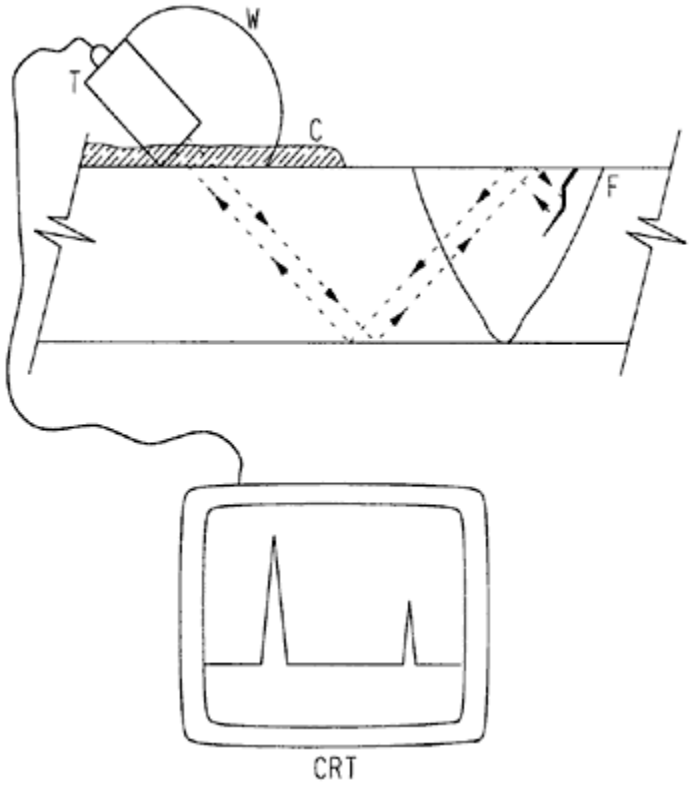


**Figure 16-4** Angle Beam UT Wedge (W), Transducer (T), Couplant (C)

Ultrasonic testing is also commonly used for in-service inspection of piping wall thickness and service induced degradation such as cracks and blisters. With the right tools, three-dimensional information can be obtained and plotted to help understand the wall-thinning pattern [Share]. On-stream (in-service) wall thickness inspections using UT have been done at temperatures up to 1000°F.

Ultrasonic testing dates back to 1929, when Russian engineer Sokolov used ultrasonic waves to measure the wall thickness of metals. The method was patented in Germany in 1931, and further improved and patented in 1940 in the United States. Straight beams were applied for the examination of welds around 1945, and angle beams started to be used in 1947. The applicable standards are ASME B&PV Section V Articles 4 (in-service examination) and Article 5 (examination of materials and fabrication), ASTM E 114, ASTM E 164, ASTM E 213, ASTM E 428, ASTM E 500, and ASTM E 797.

## 16.6 EDDY CURRENT TESTING

In eddy current testing (ET) a high frequency current is applied to a probe or encircling coils. Eddy currents are generated in the specimen which, in turn, affect the primary current. The presence of a flaw or a change in wall thickness will be detected by a disturbance of the current. Eddy currents are used to detect surface or near surface flaws.

The depth of penetration depends on the metal and the frequency of the magnetizing current. Eddy currents at frequencies of 100 Hz to 10 kHz can detect surface cracks. At 10 kHz to 6 MHz ET can also detect subsurface flaws [ASM]. ET is commonly used for the examination of heat exchanger tubes during manufacture or at outages. Eddy currents can also be used to measure grain size, heat treatment and hardness. Eddy current testing requires experienced operators to apply the technique and interpret its results because ET is sensitive to changes in thickness of the pipe wall or weld, surface roughness and calibration.

The applicable standards are ASME B&PV Section V Article 8 and ASTM E 243. Low frequency, deeply penetrating pulsed eddy currents are used to measure wall thinning under insulation [Cohn]. Changes in wall thickness can be detected on pipe wall thickness from 0.3” to 1.5”, with an accuracy in the order of a few mils. This ability to measure wall thickness while the line is in-service and without removing insulation can lead to significant cost savings.

## 16.7 ACOUSTIC EMISSION TESTING

In acoustic emission testing (AE) piezoelectric sensors are placed in contact with the pipe, which is then pressurized. The internal pressure causes tensile stresses in the pipe wall, which tend to open existing cracks. The sensors can detect stress waves from propagating cracks or from plastic deformation. The signal is sent to an amplifier and analyzer. Using several sensors, the source of stress waves can be located by triangulation of the arrival signal time.

The method can be applied to detect crack growth during pressure testing; it can also be used to locate loose parts, leaks, cavitation in liquids, or to study the stressing of fiberglass materials.

Acoustic emission testing requires contact with the pipe and an applied stress, typically from hydrostatic test pressure. The applied stress must be in the right direction to cause a crack to emit a signal. The pressure must also be sufficiently large to challenge the crack, without leading to failure. Precautions must be in place to avoid measuring background waves or external disturbances. AE is used in the first pressurization of the component; subsequent pressurization to the same pressure may not emit significant acoustic signals. Ambient noise must be filtered out.

Acoustic emission was used in the 1920’s for seismological explorations, and in the 1950’s for the study of the failure mechanisms in tensile specimen. It has gained acceptance in the vessel and pipe industry since the 1970’s. The applicable standards are

ASME B&PV Section V Articles 11, 12 and 13, ASTM E 569, ASTM E 749, ASTM E 751, ASTM E 1067, ASTM E 1118.

## 16.8 THERMOGRAPHY

Thermography is the optical recording of infrared radiation to locate material anomalies through a change in heat flow (refer to table 16-1 for the frequency range of infrared radiation). Infrared thermography, where the heat source emanates from the part, is used to inspect such items as boiler tubes, leaks of warm or hot fluid, and power insulators.

**Table 16-1** The Electromagnetic Spectrum

| Wave Type | Approximate Frequency (Hz) |
|---|---|
| Radio waves | $10^3$ to $10^9$ |
| Microwaves | $10^9$ to $10^{12}$ |
| Infrared | $10^{12}$ to $10^{15}$ |
| Visible | $10^{15}$ |
| Ultraviolet | $10^{15}$ to $10^{19}$ |
| X rays | $10^{16}$ to $10^{20}$ |
| Gamma rays | $10^{19}$ to $10^{23}$ |
| Cosmic rays | Above $10^{22}$ |

Thermography can be used to detect liquid level in a tank. In this case, on a hot day, a thermography will clearly indicate a change of color between the tank wall above the liquid level (warmer) and that below the liquid level (cooler). Thermography is also well suited for use in-service since it can detect leaks of fluids warmer or cooler than atmosphere, or wetness under the insulation from leaks or rain water, which make the wet spots under the insulation cooler than the dry insulation.

## 16.9 MEASUREMENT ACCURACY

The value of a non-destructive examination technique can be measured by its accuracy and its probability of detection [NTIAC]. Tables 16-2 to 16-4 are examples of the accuracy of various examination techniques. In practice, each condition is unique and accuracy depends on many factors: the flaw, the object, the method, the material, the equipment, procedure and process, the calibration, the acceptance criterion, and the human factors [Rummel, NTIAC]. To illustrate the accuracy of common NDE techniques, we refer to recent inspections of 21 welded carbon steel specimens, from 0.24” to 0.60” wall thickness, containing flaws, inspected by different examiners and techniques. The detection rate is summarized in Table 16-5 [Lilley].

**Table 16-2** Detectable Flaw Size (mils) in Polished Laboratory Specimen [Pettit]

| Technique | Fab. Surface Flaw | Fatigue Crack | Internal Void | Internal Crack | Lack of Penetration |
|---|---|---|---|---|---|
| Visual | 50 | 30 | NA | NA | NA |
| UT | 5 | 5 | 15 | 80 | 30 |
| MT | 30 | 30 | 300 | 300 | NA |
| PT | 10 | 20 | NA | NA | NA |
| RT | 20 | 20 | 10 | 30 | 30 |
| ET | 10 | 10 | NA | NA | NA |

**Table 16-3** Detectable Flaw Size (mils) in Finely Machined Parts [Pettit]

| Technique | Fab. Surface Flaw | Fatigue Crack | Internal Void | Internal Crack | Lack of Penetration |
|---|---|---|---|---|---|
| Visual | 100 | 60 | NA | NA | NA |
| UT | 10 | 10 | 30 | 160 | 60 |
| MT | 60 | 60 | 600 | 600 | NA |
| PT | 20 | 40 | NA | NA | NA |
| RT | 40 | 40 | 20 | 60 | 60 |
| ET | 20 | 20 | NA | NA | NA |

**Table 16-4** Detectable Crack Size (mils) [Wurm]

| | Crack Width | Crack Length | Crack Depth | Comment |
|---|---|---|---|---|
| Visual | 4 | 80 | - | Clean surface and optical devices, simple shape. |
| PT | 0.4 | 40 | 20 | Non-porous material with cracks open to the surface. |
| MT | 0.04 | 40 | 4 | Ferromagnetic materials, surface or subsurface flaw. |
| ET | 0.4 | 40 | 4 | Electrically conductive materials. |
| UT | 0.04 | 40 | 40 | Surface or volumetric crack. Simple geometry. |
| RT | 4 | 40 | 2% wall | Surface or volumetric cracks, limited thickness. |

**Table 16-5** Flaw Detection Rates (per cent) [Lilley]

| Wall Thickness | 0.24”–0.31” | 0.31”–0.39” | 0.39”–0.47” | 0.60” |
|---|---|---|---|---|
| X-radiography | 69 | 63 | 66 | 67 |
| γ-radiography | 63 | 53 | 54 | 71 |
| Manual UT | 46 | 46 | 48 | 69 |
| Automated UT | 82 | 84 | 82 | 86 |
| TOFD UT | 80 | 79 | 75 | 96 |

## 16.10 TYPE AND EXTENT OF EXAMINATIONS

When constructing a piping system, it becomes necessary to decide (a) what to examine, (b) how to examine it, and (c) how to assess the results. The ASME B31 codes (and API 1104 for petrochemical and pipeline applications) provide requirements for the extent of examinations (what to examine), the method of examination (how to examine), and the acceptance criteria (how to judge the results). An indication that does not meet the specified acceptance criteria is a flaw.

The method and extent of examinations depend on the applicable code. For process piping (ASME B31.3), the materials and assembly are visually inspected for compliance with the code and the design. In addition to visual examinations, 5% of girth butt and miter welds (or brazed joints) including a joint from each welder, must be examined by RT. But RT is not mandatory for ASME B31.3 Category D piping (pressure below 150 psi, temperature below 366°F, non-flammable, non-toxic and non-damaging to human tissue). All butt welds in ASME B31.3 Severe Cyclic piping service must be examined by RT. For ASME B31.3 Category M piping (piping containing harmful fluids) at least 20% of welds must be examined by RT. Ultrasonic examination is often permitted as an alternative to RT. Where RT is not feasible, for example on socket welds or branch connections, PT or MT may be used instead.

Note the following points:

(a) In-process examination (quality control oversight of the weld preparation and weld process) supplemented by surface NDE (MT or PT) may substitute RT for a specific weld that can not be accessed for RT.

(b) The fabricator may select the joints to be examined unless the owner contractually specifies otherwise.

(c) The percentage of examinations is based on judgment, not probabilistic analysis.

(d) Visual examinations need not be recorded by individual weld.

(e) Where a project involves more than one contractor, the percentage of examinations applies to each fabricator separately.

(f) It is advisable to increase the 20% rate of radiography for ASME B31.3 Category M service [Becht].

For oil and gas pipelines, the percentage of welds to be examined by RT is based on the hoop stress during operation (PD/(2t)) and on the pipeline location. In most cases, radiographic examinations will be performed on all the welds of a high-pressure

transmission pipeline if the line crosses a high consequence area (HCA). Federal, state and local regulations apply in addition to examination requirements of ASME B31.4 and ASME B31.8 [49CFR].

## 16.11 ACCEPTANCE CRITERIA

The ASME B31 codes and API RP 1104 provide criteria to accept or reject indications detected during the visual, surface (MT or PT) and volumetric (UT or RT) examinations of piping system and pipeline welds.

The ASME B31 acceptance criteria for welding indications are called "workmanship standards" because they are based on fabrication experience and reflect what can be expected from an experienced welder following an appropriate welding procedure. They are not based on fitness-for-service principles; in other words, a weld with a flaw larger than the workmanship acceptance criteria may still be shown by stress and fracture analysis or testing to be acceptable for the service, not leading to leakage or failure. API RP 1104 is also a workmanship standard, but it does permit fitness-for-service analyses to be applied to evaluate flaws that do not meet the workmanship standards.

## 16.12 PERSONNEL CERTIFICATION

NDE requires certified experienced operators. In the United States, the American Welding Society (AWS) and the American Society of Nondestructive Testing (ASNT) certify non-destructive examination personnel. The ASNT certification is through the inspector's employer, following the rules of SNT-TC-1A and ASNT CP-89. In contrast, the European Union's Pressure Equipment Directive requires NDT personnel to be certified by a third party organization, a notified body.

Certification of inspection (examination) personnel is recommended but not required by the pressure piping construction codes ASME B31.

## 16.13 PIPELINE PIGS

A pig is a tool that moves through a pipeline for the purpose of inspection, dimensioning, or cleaning. Pigs are usually divided into two broad categories: utility pigs and smart pigs (also called intelligent pigs or in-line inspection (ILI) tools) [NACE].

### 16.13.1 Utility Pigs

Swab pigs are soft pigs that can be run initially in a pipeline to verify the effective size of the inner diameter.

Mandrel pigs with driving cups, scraper plates or wire brushes are typically propelled using compressed air, run in multiple passes for cleaning mill scale and removing debris from a pipeline, Figure 16-5.

The brushes may be magnetized to collect debris. Pig diameter and stiffness can be increased as the pipe gets cleaner. Thin and flexible wire brushes are used to clean inside narrow pits. The driving speed is set by the pressure differential across the pig, which can be controlled by by-pass ports in the driving cups.

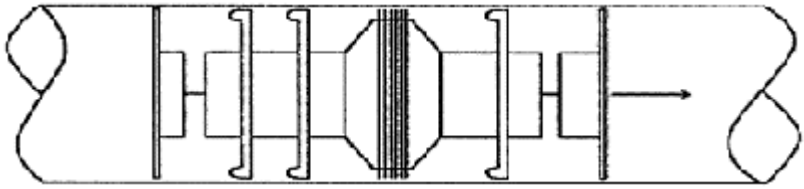

**Figure 16-5** Mandrel Pig with Driving Cups and Wire Brush

Another cleaning pig is the foam pig, Figure 16-6. It is typically bullet shaped, with a polyurethane core, a cover of abrasive silicone carbide chips, nail straps, urethane scrapers or steel studs.

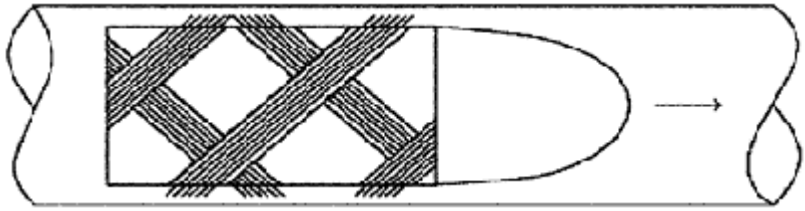

**Figure 16-6** Foam Pig

Gauging pigs, Figure 16-7, typically with aluminum plates, are used after debris removal to verify that there are no obstructions or significant ovalities in the pipeline. They can be fitted with impact loggers to record the position along the pipe where the plates have been impacted by an obstruction.

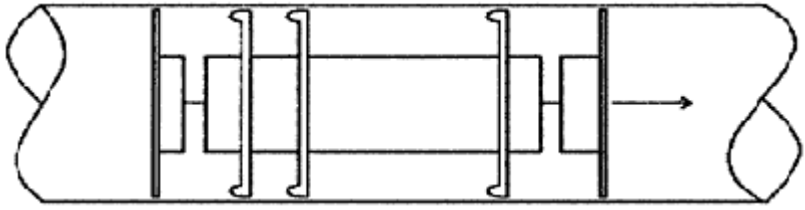

**Figure 16-7** Gauging Pig with Driving Cups and Gauging Plate

Geometry or caliper pigs record the location and size of obstructions through deflection of odometer wheels. They are typically propelled by liquid such as when dewatering following hydrotest.

Venting pigs are plugs pushed through a pipeline by the introduction of hydrostatic water to sweep out air from the line prior to hydrotest.

Dewatering pigs are typically foam pigs used to swab the line, as a precursor to more thorough drying of a pipeline by air, vacuum (which boils away any condensate in the line), nitrogen or methanol.

Commissioning pigs sandwich slugs of product to avoid mixing when filling the pipeline.

Batching pigs, Figure 16-8, are spheres used to separate multiple products or to contain batch inhibitors between two pigs. Gel pigs are viscous mixtures capable of adsorbing and carrying out debris.

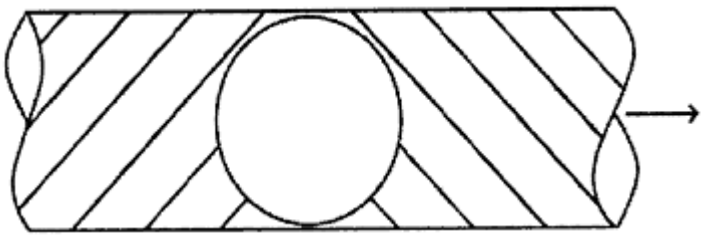

**Figure 16-8** Batching Sphere

### 16.13.2 Smart Pigs

Geometry tools consist of electro-mechanical spring loaded wheels mounted all around the pig that can record changes in pipe shape (dents) and ovality.

Gyroscopic pigs are fitted with a navigational system that records a three-dimensional isometric of the pipe routing.

Magnetic flux leakage pigs were first used in the 1960's. Wire brush magnets apply a longitudinal magnetic field to the pipe wall. Sensors detect changes in magnetic field caused by changes in wall thickness, Figure 16-9. The tool can size the length and depth of metal loss. Transverse field inspection has also been developed more recently. High resolution can be achieved by increasing the number of magnets and sensors. High-resolution tools can discriminate between ID and OD wall thinning up to approximately 10% of the wall thickness, with 80% confidence. The unit is typically fitted with wire brush magnets, sensors, odometer wheels, drive cups, data collection and storage capability. Travel speed varies but is typically around 5 miles per hour (7 ft/sec).

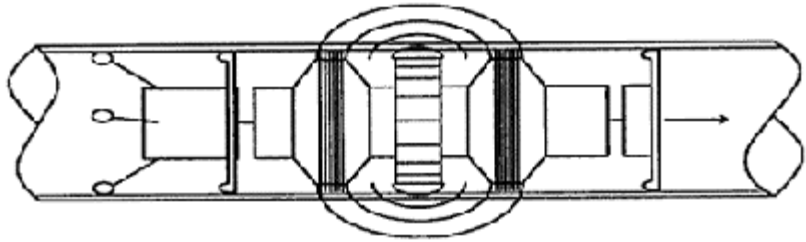

**Figure 16-9** Magnetic Flux Leakage Pig

Ultrasonic pigs require a very clean ID and a liquid couplant between the transducers and the pipeline wall, Figure 16-10, with the array of UT transducers at the back of the pig. Accuracy can be 20 mils flaw depth with 95% confidence, and 100 mils length with 300 mils width, but these accuracies are constantly improving. Travel speed is around 2 miles per hour (3 ft/sec). Angle beam ultrasonic transducers have been used since the mid-1990's to detect cracks.

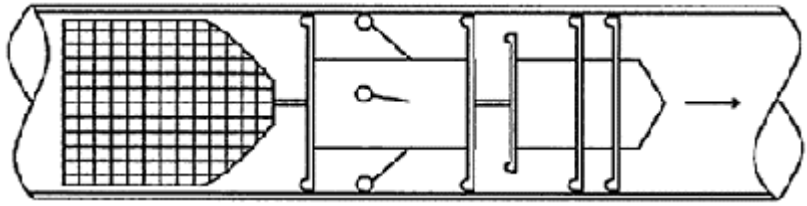

**Figure 16-10** Ultrasonic Pig at Back of Pig

## 16.14 REFERENCES


49 CFR Transportation, Part 192 Transportation of Natural Gas and Other Gas by Pipeline: Minimum Federal Safety (gas pipelines, ASME B31.8). Part 193 Liquefied Natural Gas Facilities: Federal Safety Standards. Part 194 Response Plans for Onshore Oil Pipelines. Part 195 Transportation of Hazardous Liquids Pipelines, Code of Federal Regulations, Washington, DC.

API RP 1104 Steel Pipelines Crossing railroads and Highways, American Petroleum Institute, Washington, DC, 1993.

ASM, Metals Reference Book, ASM International, Metals Park, OH.

ASME V, Nondestructive Examination, American Society of Mechanical Engineers, New York.

ASNT, The Nondestructive Testing Handbook on Radiography and Radiation Testing, American Society of Nondestructive Testing, Columbus, OH.

ASNT 2055, Recommended Practice No. SNT-TC-1A, American Society of Nondestructive Testing, Columbus, OH.

ASNT 2505, Standard for Qualification and Certification of Nondestructive Testing Personnel Also known as CP-189, American Society of Nondestructive Testing, Columbus, OH.
ASTM A 609, Standard Practice for Castings, Carbon, Low-Alloy, and Martensitic Stainless Steel, Ultrasonic Examination Thereof, American Society for Testing and Materials, West Conshohocken, PA.
ASTM E 94 Standard Guide for Radiographic Testing, American Society for Testing and Materials, West Conshohocken, PA.
ASTM E 114, Standard Practice for Ultrasonic Pulse-Echo Straight-Beam Examination by the Contact Method, American Society for Testing and Materials, West Conshohocken, PA.
ASTM E 142, Standard Method for Controlling Quality of Radiographic Testing, American Society for Testing and Materials, West Conshohocken, PA.
ASTM E 164, Standard Practice for Ultrasonic Contact Examination of Weldments, American Society for Testing and Materials, West Conshohocken, PA.
ASTM E 165 Liquid Penetrant Inspection Method, American Society for Testing and Materials, West Conshohocken, PA.
ASTM E 213, Standard Practice for Ultrasonic Examination of Metal Pipe and Tubing, American Society for Testing and Materials, West Conshohocken, PA.
ASTM E 242, Standard Reference Radiographs for Appearances of Radiographic Images as Certain Parameters are Changed, American Society for Testing and Materials, West Conshohocken, PA.
ASTM E 243, Practice for Electromagnetic (Eddy-Current) testing of Seamless copper and Copper-Alloy Tubes, American Society for Testing and Materials, West Conshohocken, PA.
ASTM E 260, Standard Practice for Packed Column Gas Chromatography, American Society for Testing and Materials, West Conshohocken, PA.
ASTM E 269, Standard Definitions of Terms Relating to Magnetic Particle Examination, American Society for Testing and Materials, West Conshohocken, PA.
ASTM E 428, Standard Practice for Fabrication and Control of Steel Reference Blocks Used in Ultrasonic Examination, American Society for Testing and Materials, West Conshohocken, PA.
ASTM E 433, Standard Reference Photographs for Liquid Penetrant Inspection, American Society for Testing and Materials, West Conshohocken, PA.
ASTM E 500, Standard Terminology Relating to Ultrasonic Examination, American Society for Testing and Materials, West Conshohocken, PA.
ASTM E 569, Standard Practice for Acoustic Emission Monitoring of Structures During Controlled Stimulation, American Society for Testing and Materials, West Conshohocken, PA.
ASTM E 709, Practice for Magnetic Particle Examination, American Society for Testing and Materials, West Conshohocken, PA.
ASTM E 747, Standard Practice for Design, Manufacture, and Material Grouping Classification of Wire Image Quality Indicators (IQI) Used for Radiology, American Society for Testing and Materials, West Conshohocken, PA.
ASTM E 749, Standard Practice for Acoustic Emission Monitoring During Welding, American Society for Testing and Materials, West Conshohocken, PA.
ASTM E 751, Standard Practice for Acoustic Emission Monitoring During Resistance Spot Welding, American Society for Testing and Materials, West Conshohocken, PA.
ASTM E 797, Standard Practice for Measuring Thickness by Manual Ultrasonic Pulse-Echo Contact Method, American Society for Testing and Materials, West Conshohocken, PA.
ASTM E 999, Standard Guide for Controlling the Quality of Industrial Radiographic Film Processing, American Society for Testing and Materials, West Conshohocken, PA.
ASTM E 1025, Standard Practice for Design, Manufacture, and Material Grouping Classification of Hole-Type Image Quality Indicators (IQI) Used for Radiology, American Society for Testing and Materials, West Conshohocken, PA.
ASTM E 1030, Standard Test Method for Radiographic Examination of Metallic Castings, American Society for Testing and Materials, West Conshohocken, PA.

ASTM E 1067, Standard Practice for Acoustic Emission Examination of Fiberglass Reinforced Plastic Resin, American Society for Testing and Materials, West Conshohocken, PA.
ASTM E 1079, Standard Practice for Calibration of Transmission Densitometers, American Society for Testing and Materials, West Conshohocken, PA.
ASTM E 1118, Standard Practice for Acoustic Emission Examination of Reinforced Thermosetting Resin Pipe, American Society for Testing and Materials, West Conshohocken, PA.
ASTM E 1417, Standard Practice for Liquid Penetrant Examination, American Society for Testing and Materials, West Conshohocken, PA.
Becht IV, C, Process Piping—the Complete Guide to ASME B31.3, ASME Press, New York.
Blitz, J., Fundamentals of Ultrasonics, Plenum Press, New York, 1967.
Bush, S.H., Interpretive Report on Nondestructive Examination Techniques, Procedures for Piping and Heavy Section Vessels, Welding Research Council Bulletin 420, New York, 1997.
Carlin, B., Ultrasonics, McGraw-Hill, New York, 1960.
Cohn, M.J., de Raad, J.A., Nonintrusive Inspection for Flow-Accelerated Corrosion Detection, PVP-Vol. 359, ASME 1997, American Society of Mechanical Engineers, New York, NY.
Cross, W., The Code, American Society of Mechanical Engineers, New York.
Frederick, J.R., Ultrasonic Engineering, Wiley, New York, 1965.
Grendahl, S., Alternative Quantitative Methods for Screening Penetrant Testing Cleaning Solutions, Materials Evaluation, January 2001.
Huber, O.J., Fundamentals of Nondestructive Testing, Metals Engineering Institute, Metals Park, Ohio, 1984.
Lilley, J.R., Perrie, C.D., Developments in NDT, AEA Technology, October, 1997. NIL Project: Non-Destructive Testing of Thin Plates, Document No. NDP 93–40, Stelwagen, March, 1995.
McGonagel, W.J., Nondestructive Testing, McGraw-Hill, New York, 1961.
NACE, Standard RP0102, In-Line Inspection of Pipelines, NACE International, Houston, TX.
NTIAC, NDE Capabilities Data Book, Nondestructive Testing Information Analysis Center, Austin, TX.
NTIAC, Magnetic Particle Inspections: Capabilities and Problems, Nondestructive Testing Information Analysis Center, Austin, TX.
NTIAC, Probability of Detection (POD) for Nondestructive Evaluation, Nondestructive Testing Information Analysis Center, Austin, TX.
Pettit, E.D., et. al., "Fatigue Flaw Growth and NDI Evaluation for Preventing Through Cracks in Space Craft Tankage Structures, NASA CR-1285600, September, 1972.
PFI Standard ES-27, Visual Examination, The Purpose, Meaning and Limitation of the Term, Pipe Fabrication Institute, Springdale, PA.
Rummel, W.D., What you Should Expect from your Penetrant Inspections, ASNT Fall Conference and Quality Testing Show—2002, Paper Summaries Book, San Diego, Nov. 2002, American Society for Non-Destructive testing, Columbus, OH.
Share, J., Alliance Pipeline Finishes '99 on Schedule, within Budget, Pipeline & Gas Journal, January 2000.
Siegel, R., Sherwin, A.G., Verifying Cleaner Effectiveness Prior to Inspection Penetrant Processing, ASNT Fall Conference and Quality Testing Show—2002, Paper Summaries
Book, San Diego, Nov. 2002, American Society for Non-Destructive Testing, Columbus, OH.
Wurm, B., et. al., Fracture Mechanics Assessment Procedure for Engineering Components, Comparative Evaluation of International FFS Standards, Pressure Vessel Research Council, Workshop, Houston, October, 2000.

# 17
# Pipe Flange

## 17.1 FLANGE STANDARDS

A pipe flange is a common joint in piping systems. The ease of disassembly and re-assembly of flange joints makes them particularly well suited for maintenance, line breaks and pipe connections to equipment and components. Pipe flanges are often taken for granted, and relatively little attention is paid to flange joining compared for example to the attention accorded to welding. Flange joint sizes and ratings have been standardized since the 1920's. Today's common pipe flange standards are:

| | |
|---|---|
| ANSI B16.5 | Pipe Flanges and Flanged Fittings. |
| ANSI B16.24 | Cast Copper Alloy Pipe Flanges and Flanged Fittings. |
| ANSI B16.34 | Valves—Flanged, Threaded, and Welding End. |
| ANSI B16.36 | Orifice Flanges. |
| ANSI B16.47 | Large Diameter Steel Flanges NPS 26–60. |
| API 6A | Pressure Rating. |
| API 601 | Metallic Gaskets for Raised Face Pipe Flanges. |
| API 605 | Large Diameter Carbon Steel Flanges (26" to 60"). |
| MSS SP-44 | Steel Pipeline Flanges. |
| ASME PCC-1 | Guidelines for Pressure Boundary Bolted Flange Joint Assembly |

## 17.2 FLANGE TYPES

There are several types of pipe flanges, as illustrated in Figure 17-1. To choose the best suited flange joint for a given service, we must consider the system operating pressure, the loads applied in service (pressure, bending), the fluid, the ease of assembly, and past experience with similar flanges.

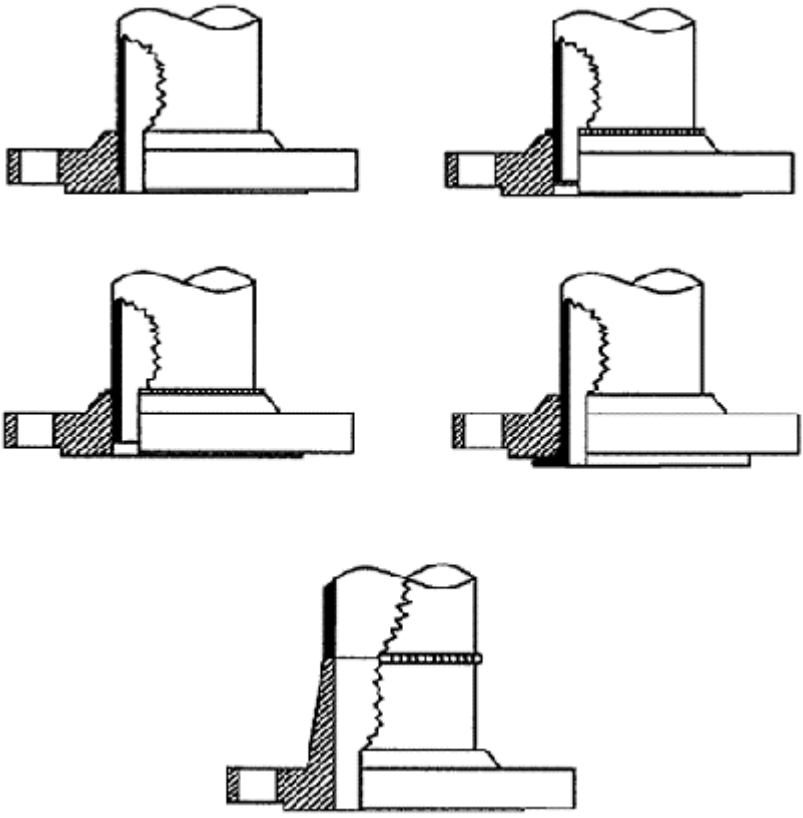

**Figure 17-1** Common Flange Types

Clockwise from top left: Threaded, Slip-On, Lap, Weld Neck, Socket

Threaded: the threaded flange is mostly used with small-bore piping (2" NPS and smaller). It is easy to assemble, but is susceptible to crevice corrosion since process fluid penetrates the thread region. It is also susceptible to leakage in vibratory service, particularly if installed misaligned, or under large pipe bending moments. A threaded joint has a stress intensification factor of 2.3 (Chapter 7), which means that under cyclic stress it takes $2.3^5$~64 times fewer cycles to leak a threaded joint than a butt weld.

Slip on: the flange is slipped onto the pipe and double fillet welded to the pipe. It requires less skill to assemble and permits aligning bolt holes by rotating the flange before welding. It is generally used in moderate pressure service, for example by limiting its use to Class 300 for pipe larger than 4". A slip-on flange with a single outward weld is similar to a socket welded flange, and—if used at all—should be limited to non-corrosive, non-critical, 2" and smaller pipe.

Socket welded: the socket-welded flange is commonly used for small-bore piping. Because the gap at the pipe to socket interface is a crevice exposed to the process fluid, the socket-welded flange is susceptible to crevice corrosion.

Lap joint: the lap joint flange slips onto the pipe and abuts against the pipe lap. The flange can rotate to align bolt holes and is easily dismantled. It is commonly used in mild service, when the loads applied by the pipe to the flange connection are small.

Welding neck: the tapered neck of a welding neck is butt welded to the pipe providing a smooth transition with little stress concentration and no gaps or recesses that could lead to crevice corrosion. The welded connection to the pipe is strong and can be readily radiographed. The welding neck flange is well suited for severe service at high temperature and pressure, or under large external loads.

## 17.3 FLANGE GASKETS

### 17.3.1 Selection Factors

There are three main groups of flange gaskets: (1) non-metallic sheet gaskets, (2) semi-metallic jacketed or spiral wound gaskets, and (3) metallic ring gaskets. To select a gasket, one must consider the following factors: (a) limitations specified in the gasket manufacturer's catalog, (b) operating experience with the same gasket in similar service, (c) maximum operating pressure and temperature, sometimes specified as a maximum value of PxT, such as 20,000 psixF or 350,000 psixF, (d) compatibility of the gasket with the fluid, and its permeability to the fluid, (e) compatibility of the gasket with the flange material, (f) availability and cost of gaskets, (g) resistance to creep or flow [ASTM F 38], (h) ability to recover [ASTM F 36A], (i) sealability [ASTM F 37], (j) compatibility of the gasket with the environment, (k) strength [ASTM F 152], and (l) smoothness or roughness of the flange face finish.

In some cases the gasket must withstand a fire, for example in petrochemical applications, or be compatible with exacting cleanliness requirements, for example in the food and drug industries. An example will help illustrate the need for caution when using gasket catalogs: a gasket was originally listed and sold by a ven-dor for steam service up to 450°F. The facility engineer specified the gasket for use in 350 psi saturated steam, with a temperature of 430°F. Within hours of startup, two gaskets blew out causing steam to jet radially out from the flange joint. While investigating the incident, the engineer ordered the latest gasket catalog and, to everyone's surprise, the new catalog limited the use of this type of gasket to 400°F "continuous" service and permitted 450°F only "for short duration". Catalogs do change without warning.

### 17.3.2 Non-Metallic Gaskets

The following gasket applications are provided as rough guidance. The choice of gasket for a given fluid, pressure and temperature, must be based on the plant's experience and up-to-date information from the gasket manufacturer.

Nitrile (NBR, Buna-N) is a synthetic rubber, typically used from −50°F to 250°F, and up to 150 psi. If reinforced with Aramid fiber (an organic, aromatic amid fiber) it can withstand higher temperatures and pressures up to 500°F and 1200 psi.

Natural rubber has good strength against tearing, and is typically used from −60°F to 250°F.

Butyl, isobutylene, is an elastomer used in gas service, and is typically used from—50°F to 300°F.

Neoprene is typically used from −50°F to 250°F, and up to 150 psi.

EPDM (ethylene propylenediene) is suitable for steam service, typically used from −50°F to 300°F. If reinforced with Aramid fiber it can withstand higher temperatures: −50°F to 550°F, at 1200 psi.

Viton, typically used from −20°F to 390°F, and up to 150 psi.

Reinforced polytetrafluorocarbon (PTFE, Teflon™) typically used from −350°F to 50°F, and up to 800 psi. Reinforced Teflon™ is preferable to virgin Teflon™, which tends to flow and lose the joint preload. Glass reinforced PTFE resin, typically used from −350°F to 480°F, 700 psi.

Graphite, typically used from −300°F to 850°F, and up to 2000 psi.

SBR (styrene butadiene) is a synthetic rubber, with Aramid fiber it is typically used from −60°F to 250°F, and up to 1200 psi.

### 17.3.3 Semi-Metallic Gaskets

A semi-metallic gasket consists of a filler material (such as graphite or Teflon™) sandwiched in a metal jacket (such as stainless steel) or spirally wound. With spiral wound gaskets, an outer metal ring serves to center the gasket and avoid blowout, and an inner metal ring serves to avoid inward buckling of the gasket inner windings if overly compressed, Figure 17-2. The metal used can be carbon or stainless steel, as well as nickel, copper or aluminum alloys. Semi-metallic gaskets can generally be used at higher pressures and temperatures than the non-metallic gaskets of group.

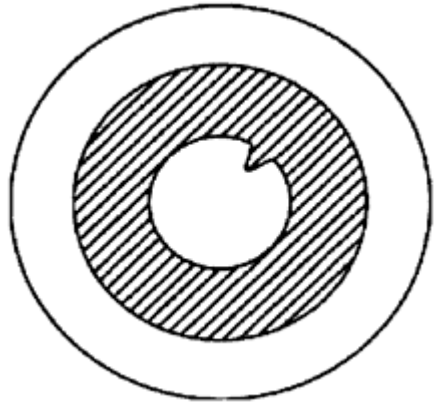

**Figure 17-2** Buckling of Spiral Wound Gasket Without Inner Ring

Spiral wound Teflon™ is typically used from −350°F to 500°F, and up to 6000 psi. Spiral wound graphite is typically used from −350°F to 850°F, and up to 6000 psi. Above 850°F, there may be a tendency for the graphite in some spiral wound graphite gaskets to deteriorate and, in the presence of hydrogen, form methane. Metal jacketed gaskets: the metallic jacket envelops a non-metallic sheet gasket.

### 17.3.4 Metallic Gaskets

Metallic gaskets are made of a solid metal ring that lodges into a grove in both flange faces. It is used for high pressure and temperature service, typically above 5000 psi, on pipe as well as vessel flanges. The bolt torque has to be very high to yield the metal ring inside the groove and fulfill the sealing function. Also, the ring hardness has to be lower than the flange hardness so that the ring, and not the flange, will yield at the point of contact. In addition, the flange groove must be sufficiently smooth to avoid stress concentration. Some large flanges have cracked at the edge of the groove corner due to large bearing stresses imposed by the ring on the groove.

## 17.4 FLANGE FACES

There are several types of flange faces, as illustrated in Figure 17-3: (a) flat face, (b) raised face, (c) lap joint, (d) metallic ring joint.

Flat face: flat faces are typical in cast iron class 125 and 250 flanges. The full-faced gaskets have a large contact area that extends beyond the bolt circle. The reason for having a flat face on a cast iron flange is to distribute the compressive force over the widest area (beyond the bolt circle) avoiding bending of the flange face that would cause the cast iron to crack. For this reason, when mating a raised face pipe flange to a flat face cast iron equipment flange, it is advisable to machine down the raised face so as to evenly distribute the load on the cast iron face. Conversely, on steel pumps, if the steel pump flange has a flat face it could be mated to a raised face steel pipe flange since—with the correct preload—the steel pump flange can accommodate bending.

Raised face: a raised face is common in pipe flanges. The face is typically raised 0.06” for classes 150 and 300, and 0.25” for class 400. Because the face is raised, the gasket diameter is smaller than the bolt circle.

Lap joint: the pipe lap protrudes from the flange and acts as a raised face.

Ring face: the metallic ring can be oval or octagonal. The ring groove in the flange face is machined with precision to a very smooth finish, to avoid leaks in service.

The type of gasket will dictate the finish of the flange face. In general, softer gaskets require a rough serrated surface finish to grip the gasket in place, while metallic ring gaskets will require a very smooth, mirror-like finish. Typical surface finishes are defined in centerline average (CLA) or roughness average $R_a$, which is the arithmetic average of the absolute values of the profile height deviations from the mean line. Always refer to the gasket vendor instructions for the flange surface finish required for a given gasket.

Typical surface finish for flange faces are:

| | |
|---|---|
| Thin (1/16") soft gaskets | 100 to 250 μ-inch CLA |
| Thick (over 1/16") soft gaskets | 100 to 500 μ-inch CLA |
| Spiral wound | 100 to 250 μ-inch CLA |
| Metallic flat | 30 to 100 μ-inch CLA |
| Metal ring joint | 60 μ-inch CLA |

Surface finish is typically judged by visual comparison or by instruments and lasers [ASME B46.1]. An experienced machinist is able to judge surface roughness by touch and visual inspection. The serrations should be circular, concentric or spiral, and free of scratches to avoid radial leak paths (Table 17-1).

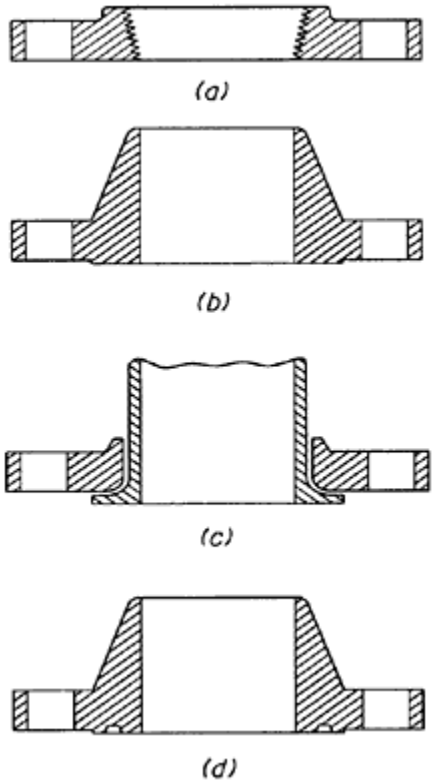


**Figure 17-3** Typical Flange Faces

(a) Threaded, (b) Raised Face, (c) Lap Joint, (d) Metallic Ring

**Table 17-1** Length and Depth of Imperfections in Flange Faces [B16.5]

| Pipe size (in) | Radial (in) | Depth (in) |
|---|---|---|
| 0.5 to 2.5 | 0.12 | 0.06 |
| 3 | 0.18 | 0.06 |
| 3.5 to 6 | 0.25 | 0.12 |
| 8 to 14 | 0.31 | 0.18 |
| 16 | 0.38 | 0.18 |
| 18 to 24 | 0.50 | 0.25 |

## 17.5 FLANGE RATING

Flanges are grouped in "classes" by "ratings" according to their design pressure. The flange rating is expressed in "psi" or, as a common practice, in "pounds". We speak for example of "a 300 pound flange". It is however preferable to simply refer to a "class 300 flange", dropping altogether the "pound" or "psi" nomenclature since a class 300 flange does not mean that the flange rated pressure is 300 psi. The exact definition of flange rating is given in B16.5. For class 300 and heavier, the rated working pressure $P_T$ is a function of the pressure rating $P_r$ and the allowable stress $S_T$, defined as the lowest of $0.6S_Y$ at 100°F, $0.6S_Y$ at temperature, and 1.25 S from ASME Boiler and Pressure Vessel code Section I at temperature, with

$P_T=P_r\ S_T/8750$

$P_T$=rated working pressure for the specified material at temperature T, psi

$P_r$=pressure rating class index

$S_T$=flange material allowable stress at temperature T, psi

As a rule of thumb, at ambient temperature, the maximum operating pressure of a carbon steel flange of class X is approximately 2.4X. This rule works particularly well for class 300 lb and heavier. For example, a class 300 carbon steel flange will have a design pressure of ~2.4×300=720 psi at ambient temperature.

As an example, the maximum working pressure of class 150 is shown in Table 17-2 for several materials, and in Table 17-3 for classes 150 to 4500 ASTM A 105 flanges.

**Table 17-2** Pressure Rating (psi) Class 150 Flanges [ASME B16.5]

| Temp. °F | Carbon steel | 2–1/4Cr—1 Mo | 304 Stainless |
|---|---|---|---|
| −20 to 100 | 235–285 | 290 | 275 |
| 200 | 215–260 | 260 | 235 |

| | | | |
|---|---|---|---|
| 300 | 210–230 | 230 | 205 |
| 400 | 200 | 200 | 180 |
| 500 | 170 | 170 | 170 |
| 600 | 140 | 140 | 140 |

**Table 17-3** Ratings (psi) for A 105 Carbon Steel Flange [ASME B16.5]

| Temp. °F | 150 | 300 | 400 | 600 | 900 | 1500 | 2500 | 4500 |
|---|---|---|---|---|---|---|---|---|
| −20 to 100 | 285 | 740 | 990 | 1480 | 2220 | 3705 | 6170 | 11110 |
| 200 | 260 | 675 | 900 | 1350 | 2025 | 3375 | 5625 | 10120 |
| 300 | 230 | 655 | 875 | 1315 | 1970 | 3280 | 5470 | 9845 |
| 400 | 200 | 635 | 845 | 1270 | 1900 | 3170 | 5280 | 9505 |
| 500 | 170 | 600 | 800 | 1200 | 1795 | 2995 | 4990 | 8980 |
| 600 | 140 | 550 | 730 | 1095 | 1640 | 2735 | 4560 | 8210 |
| 700 | 110 | 535 | 710 | 1065 | 1600 | 2665 | 4440 | 7990 |
| 800 | 80 | 410 | 550 | 825 | 1235 | 2060 | 3430 | 6170 |
| 900 | 50 | 170 | 230 | 345 | 515 | 860 | 1430 | 2570 |
| 1000 | 20 | 50 | 70 | 105 | 155 | 260 | 430 | 770 |

A type 304 stainless steel flange class 150, operating at ambient temperature can be placed in a system with a design (maximum) pressure of 275 psi. This system, including the flange, can be hydrostatically tested at 1.5 times 275 psi=413 psi. In other words, it is acceptable to exceed the flange pressure rating for the purpose of hydrostatic testing.

## 17.6 FLANGE BOLT TORQUE

On July 6, 1988, on the Piper Alpha platform in the North Sea, a condensate injection pump (pump CIP-A, Figure 17-4) was shutdown for routine maintenance. As part of the maintenance activity, the 4" pressure safety valve for pump A was removed (PSV-504) and sent to the platform's valve shop for routine calibration. A pair of blind flanges was fitted in its place as temporary covers.

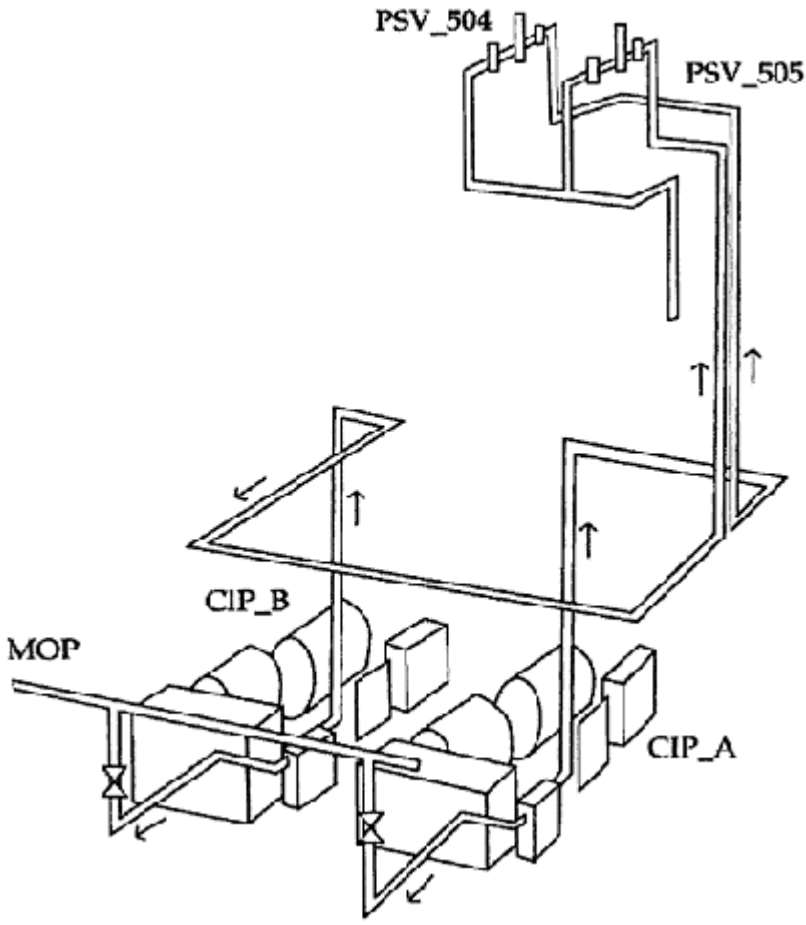


**Figure 17-4** Simplified Schematic of Condensate Injection Pumps

That same day, the second condensate injection pump (pump CIP-B) tripped unexpectedly. With the two condensate pumps out of service, condensate started to accumulate in hold tanks, triggering high-level alarms. As the operators tried in vain to repair and restart pump CIP-B, the decision was made to promptly terminate the maintenance of pump CIP-A and return it to service.

Apparently, pump A's maintenance work package did not clearly point out that the pressure relief valve had been removed, and the temporary blind flanges were at a higher elevation, not visible from the pump deck [Cullen, Lancaster]. Pump A was therefore started, with—unknown to the operators—blind flanges still installed in place of PSV-504. As the pump started, the line was filled with 670 psi condensate, which started to leak around the blind flange gasket, with a loud hissing sound.

The investigation indicates that the blind flanges, installed as a temporary measure on an empty line while the relief valve was being tested, had not been properly tightened [Cullen]. In this type of service, flange bolts have to be tightened in a controlled manner. The escaping vapor ignited, leading to a series of explosions, including the burst of the gas pipe riser to the platform. The destruction of the platform was nearly total, and 165 men lost their lives.

The rule for correctly bolting a pipe flange can be summarized as follows: the flange gasket must be uniformly compressed to a certain stress level, not less and not more; and, in the process of compressing the gasket, the flange bolts should not yield [ASME PCC-1]. This objective is achieved by following a step-by-step calculation process described in this section [ASME VIII, ASME B16.5]. We take as an example a 4" carbon steel water piping system, with a design pressure of 165 psi and a design temperature of 70°F.

Given the flange material (carbon steel forging A105), ASME B16.5 Table 1A will assign the flange to group 1.1.

Given the group (1.1), the design temperature (ambient) and design pressure (165 psi), ASME B16.5 Table 2 will assign a Class 150.

Given the pipe size (4") and flange class (150), ASME B16.5 Table 8 tells us that the flange will require 8 bolts of 5/8" diameter.

We now chose a gasket that is compatible with the fluid and compatible with the operating pressure and temperature.

Having chosen a 1/8" thick rubber gasket, ASME B16.5 Fig.E1 (or ASME VIII Div.1 Appendix 2 Table 2–5.1) tells us that the gasket is group Ia, and our choice is compatible with the recommendation of ASME B16.5 section 5.4.2 to use group Ia gaskets with class 150 flanges. Both ASME B16.5 Appendix E and ASME VIII Div.1 Appendix 2 give us two gasket factors: m=gasket factor=2.00 and $\gamma$ (called "y" in ASME VIII)=minimum design seating pressure=1,600 psi.

The "m" factor is an experimentally determined factor. It is the ratio of the compressive pressure to be exerted on the gasket during assembly, to the highest system pressure in service. In this case, an "m" of 2 means that, when assembling the flange, we need to exert a compressive pressure on the gasket of at least twice the design pressure.

The "y" factor is the compressive stress that must be generated in the gasket for proper seating. The seating stress is a function of the gasket thickness. For example a catalog may specify y=1600 psi for a 3/16" gasket, 2100 psi for ¼", 2600 psi for 3/8" and 3000 psi for ½". The thicker gaskets require larger compressive stresses for proper seating. The values listed in ASME B16.5 Appendix E and ASME B&PV VIII Appendix 2 were established based on tests of a number of gasket groups. More accurate values for specific gaskets may be obtained from gasket vendors.

Knowing that the gasket belongs to group Ia and that the pipe is 4", ASME B16.5 Fig.E2 gives a flange face width W=0.84". In ASME VIII Appendix 2, the width W is labeled N. We can verify this width by noting that the OD of a 4" pipe is 4.5" and that the radius of a 150 lb raised face is, from B16.5 Table 4, equal to 6.19". Therefore the width of the raised face is W=N=(6.19−4.5)/2=0.845".

An important assumption in ASME VIII flange design is that the compression on the gasket is not uniform, it is assumed that as flange bolts are torqued, the flange faces tend to bend and pinch the gasket and only portion of the total gasket width N is effectively compressed.

The "effective" gasket width "b", which will be smaller than the actual gasket width "N", is obtained as follows. Given the gasket width N, and since the gasket is completely contained within the raised faces, ASME VIII Appendix 2 Table 2-5.2 (or ASME III Appendix E) defines a "basic gasket seating width" $b_o$ as $b_0=N/2=0.42$". If $b_o \leq 0.25$" then $b=b_0$. If $b_o>0.25$" then $b=0.5(b_0)^{0.5}$. In our case, $b_0=0.42">0.25$" and therefore $b=0.5(0.42)^{0.5}=0.32$".

The effective width "b" extends inward from the outer edge of the raised face to a diameter "G", which is the diameter of load reaction. If $b_o \leq 0.25$" then G is the mean diameter of the gasket contact face. If $b_o>0.25$" then G is the outside diameter of gasket contact face minus 2b. In our case, $b_0=0.42">0.25$" and therefore, G=6.19"−2 (0.32")=5.6".

The required preload $W_m$ on the gasket must be sufficient to achieve two objectives: (a) in service, the preload must be sufficient to sustain the internal pressure, this is the preload $W_{m1}$, and (b) at assembly, the preload must be sufficient to seat the gasket between the flange faces, this is the preload $W_{m2}$.

Let's first look at the preload $W_{m1}$. The minimum required preload to sustain the design pressure is given in ASME VIII Div. 1 Appendix 2 as

$W_{ml}=H+H_P$

$W_{m1}$=bolt preload to sustain the design pressure, lb

H=bolt preload to resist the hydrostatic load in the pressurized pipe, lb

$H_P$=experience based preload to ensure leak tight joint, lb

To calculate H, we consider that only a band of width "b" is sealed at the periphery of the gasket, beyond the diameter of load reaction "G". With this approximation, the area that is not sealed, an area $\pi G^2/4$, will be subject to the operating pressure and, as a maximum, the design pressure. Therefore, the hydrostatic load trying to open the flange joint is

$H=(\pi G^{2/}4)\ P=0.785\ G^2\ P$

P=design pressure, psi

The second load term, $H_P$, was determined experimentally to be proportional to the system pressure; the proportionality constant was labeled "m". A pressure mP needs to be applied over a ring of diameter G and width 2b. This can be written as

$H_P=(2b)\ (\pi G)\ m\ P=7{,}238$ lb

We now consider the load $W_{m2}$ needed to stress the gasket to a value "y" and seat the gasket. In our case y=1,600 psi. As before, we consider the compression to occur over a ring band of diameter G and width b. The minimum required bolt load to achieve a stress y is

$W_{m2}=(\pi G)\ b\ y=9{,}008$ lb

The minimum bolt preload is the largest of $W_{m1}$ and $W_{m2}$, which corresponds to approximately 9000 lb. At this stage, we know that we need a class 150 flange, an 8-bolt flange assembly, which can sustain a preload $W_m$ of 9,000 lb. At installation, each bolt must see a tension of at least 9,000/8=1,125 lb. The standard bolt for a 4" class 150

flange is 5/8" (Table 8 of ASME B16.5). A standard 150 lb, 4" flange has eight 5/8" bolts. The root diameter of the 5/8" bolt is 0.5168", which corresponds to a bolt root area

$A_{bolt}=\pi d^2/4= 0.210$ in$^2$

The bolt stress area (slightly different from the root area) is [ASTM A 574]

$$A = 0.7854\left(D - \frac{0.9743}{n}\right)^2$$

D=nominal diameter, in

n=number of threads per inch, 1/in

With this formula, a 5/8"—11-thread/inch bolt has a stress area of 0.226 in$^2$

Given the installation tension of 1,125 lb per bolt, over an area of 0.21 in$^2$, the tensile stress applied to the bolt during preload to compress the joint is

$\sigma=W/A_{bolt}=5{,}357$ psi

We can now select a bolt material. We need a bolt that is 5/8" in diameter, can sustain at least 5,357 psi, and is compatible with a carbon steel forging flange body. We chose an ASTM A193 Gr.B7 bolt, commonly used with carbon steel flanges. It is a high strength bolt compatible with the fluid, and listed in B16.5 Table 1B. At ambient temperature, its allowable stress is [ASME II]

$S_{ambient}=23$ ksi

The bolt yield and ultimate strength are [ASME II]

$S_y=95$ ksi

$S_U=115$ ksi

In this case the bolt preload stress of 5,357 psi is well within the material allowable stress. Often times, at higher pressures or temperatures, or with gasket with larger seating stress y, the calculated bolt tension may exceed the material allowable stress. In practice, it is common to torque the bolt close to its yield point, for example 85% to 90% of yield [Faires]. We know that we need to torque each of the flange four bolts to achieve a tension of 1,125 lb in each bolt. The minimum bolt torque T needed to achieve a preload $F_i$ is

$T_{min}=K\ F_i\ d$

$T_{min}$=minimum bolt torque, in-lb

K=nut factor

$F_i$=required preload, lb

d=nominal bolt diameter, in

The nut factor is a factor established experimentally and depends on the metal and lubricant [ASME PCC-1]. It typically ranges from 0.10 to 0.20 for lubricated bolts, and can be as high as 0.5 for unlubricated bolts [EPRI, Bickford]. With K=0.20, $W_m$=1,125 lb, d=5/8", the minimum torque needed to seat the gasket and react the hydrostatic load in

service is $T_{min}$~12 ft-lb. We compare this minimum torque to the torque required to reach the bolt allowable stress and 80% of the bolt yield stress

$T_{max}$~60 ft-lb (to reach S=23 ksi)

$T_{max}$~170 ft-lb (to reach 80% of $S_y$=95 ksi)

ASME B&PV Section VIII Division 1, Appendix N reports the approximate bolt stress achieved by using a normal hand wrench is

$$S = \frac{45{,}000}{\sqrt{d}}$$

S=stress in bolt using normal hand wrench, psi

D=nominal bolt diameter, in

In the case of the 5/8" bolt the stress achieved with a manual wrench would be 57 ksi, well above the minimum needed of 10.7 ksi, also above the bolt allowable stress of 26.7 ksi, but below 80% yield=0.8 (95)=76 ksi.

Obvious but important warning: bolt torque values only make sense assuming the faces are in uniform contact before torquing. Otherwise, part of the torque would serve to pull the two flange faces closer together rather than cause a uniform preload on the gasket.

## 17.7 EXTERNAL LOADS

While in operation, the piping system will apply forces and moments on the pipe flange. These loads can be due to weight, expansion, vibration, settlement, etc. How to account for the effect of applied loads in designing and preloading a flange joint? The first and simplest method is to convert the applied load into an equivalent pressure, which is added to the concurrent pressure. The flange is then qualified to the total pressure. An equivalent pressure formula is provided in ASME B&PV Section III, Division 1, NB-3658, as [ASME III]

$$P_{eq} = \frac{16M}{\pi D^3}$$

$P_{eq}$=equivalent pressure, psi

M=applied moment, in-lb

D=mean gasket diameter, in

A second approach would consist in limiting the applied bending and torsional moment $M_{fs}$ to the following value [ASME III, Rodabaugh]

$$M_{fs} \le 3125 \frac{S_y}{36000} C A_b$$

$M_{fs}$=bending and torsional moment, in-lb

$S_y$=yield stress of flange material, psi

C=diameter of bolt circle, in

$A_b$=total cross sectional area of bolts at root of threads, $in^2$

The normal, shear and Von Mises stresses in any flange bolt when the flange assembly is subject to moments and forces are [Lu]

$$\sigma_{11i} = \frac{K_b}{K_b + K_m}\left[\frac{F_1}{NA} - \frac{M_3}{I} R \sin\frac{2\pi}{N}(i-1) + \frac{M_2}{I} R \cos\frac{2\pi}{N}(i-1)\right] + \sigma_p$$

$$\tau_{12i} = \frac{1}{A}\left[\frac{F_2}{A} - \frac{2M_1}{RN}\cos\frac{2\pi}{N}(i-1)\right]$$

$$\tau_{13i} = \frac{1}{A}\left[\frac{F_3}{A} + \frac{2M_1}{RN}\sin\frac{2\pi}{N}(i-1)\right]$$

$$\sigma_{ei} = \sqrt{\sigma_{11i}^2 + 3(\tau_{12i}^2 + \tau_{13i}^2)}$$

A=effective cross section area of flange bolt, $in^2$

$F_1$=axial load along x axis, lb

$F_2$=shear force along y axis, lb

$F_3$=shear force along z axis, lb

I=moment of inertia of flange, $in^4$

$K_b$=stiffness of bolt i, lb/in

$K_m$=stiffness of flange, lb/in

$M_1$=torsion on flange around x axis, in-lb

$M_2$=bending on flange, around y axis, in-lb

$M_3$=bending on flange, around z axis, in-lb

N=number of bolts

R=radius of bolt circle, in

x=longitudinal axis of pipe

y, z=transverse axes of pipe

$\sigma_P$=preload stress, psi

$\sigma_{11}$=normal stress in bolt i, psi

$\sigma_{ei}$=Von Mises equivalent stress in bolt i, psi

$\tau_{12i}$=shear stress in bolt i, psi

$\tau_{13i}$=shear stress in bolt i, psi

A bending moment applied to a flange joint can also be translated into an edge load, compressive over half the circumference of the gasket and tensile on the opposite half [Bouzid]

$N = M\cos\alpha/(\pi B^2)$

N=distributed edge force, lb/in

M=applied bending moment, in-lb

α=angle around the circumference, deg

B=inside diameter of flange, in

## 17.8 ASSEMBLY OF PIPE FLANGES

### 17.8.1 Assembly Steps

The proper assembly of a pipe flange joint follows several steps described here [Bibel, ASME PCC-1].

(a) Select a standard flange, of the right rating, compatible with the pipe and the service. Refer to Sections 17.1, 17.2 and 17.5.

(b) Select a gasket compatible with the flange and the service. Refer to Section17.3.

(c) Verify that the flange and gasket are not damaged. Refer to Section 17.4.

(d) Align the two flange faces to achieve parallelism of the faces, and axial alignment of bolt holes. An initial parallelism of flange faces of 1/16" per foot of flange face, or even tighter, is required by piping codes. This is achieved by feeler gages and dial indicators or laser alignment at pipe-nozzle of certain rotating equipment. In practice, the precision of alignment should depend on the flexibility of the pipe span and sensitivity of the connected component to nozzle loads. For example, a pipe flange on flexible small bore tubing lines does not need to be as tightly aligned because the flexibility of the tubing spans will accommodate some level of rotation while bolting the flange. On the other hand, a flange between a stiff pipe, for example a large diameter or rigidly supported pipe, and the nozzle of a pump, compressor or heat exchanger on the other side, will have to be very well aligned before applying a torque to the bolts.

(e) Lubricate the bolts, unless they are coated. Lubricating the bolts will assure that the applied torque serves to preload the bolt and is not lost in friction. As much as 30% of the torque is used to overcome friction in new non-lubricated bolts, and this loss is as much as three times larger with galled threads. The gasket vendor should be contacted for compatibility of lubricant. When reusing bolts, make sure that they are wire brush cleaned before being lubricated.

(f) Establish a torque range by specifying a maximum and minimum torque, as described in Sections 17.6 and 17.7. In the field, the mechanic will have to tighten the bolts within this range. The method used to tighten flange bolts varies: the simplest and least accurate is hand wrench tightening based on the mechanic's feel and experience. Next in accuracy is the use of a calibrated torque wrench. The most accurate tightening is achieved by controlled hydraulic tensioning of the bolt and running down the nut against the flange face while the bolt is stretched, Figure 17-5. The nut is tightened against the flange (a), then the stud is hydraulically stretched (b), and the nut is turned down against the flange (c). Preload is then obtained as the stud is released from the tensioning machine. In this case the accuracy of bolt preload can be 10% or better. Note that with hydraulic tensioning some plastic deformation at the nut-flange bearing surface will cause some loss of preload. This can be minimized by using washers.

The choice of flange torque methods depends on the type of gasket, the service, the mechanic's training and experience, and the company's policy. For example, manual torque may be sufficient for mild service, such as a non-critical class 150 water line. Hydraulic tensioning with elongation control may be necessary for large flanges in critical service, for example a heat exchanger head to shell flange [ASME PCC-1].

There are two alternatives to torque control: (1) when using spiral wound gaskets and (2) when using soft gaskets. In the first case, the proper preload of spiral wound gaskets can be verified by visual inspection of the side of the gasket to assure that the gasket has been fully compressed and the outer metal ring is in contact with the flange faces all around the circumference. In the second case, when using a soft gasket, the flange faces can be aligned and placed evenly in contact with the gasket before bolt-up; the gap "G" between flange faces is measured, then the bolts are tightened to bring the gap to a fraction "mG" of the initial gap. The value of "m" is obtained from the gasket manufacturer. For example, if the gasket is to be compressed 25%, the gap between flange faces after bolt tightening would be 0.75G.

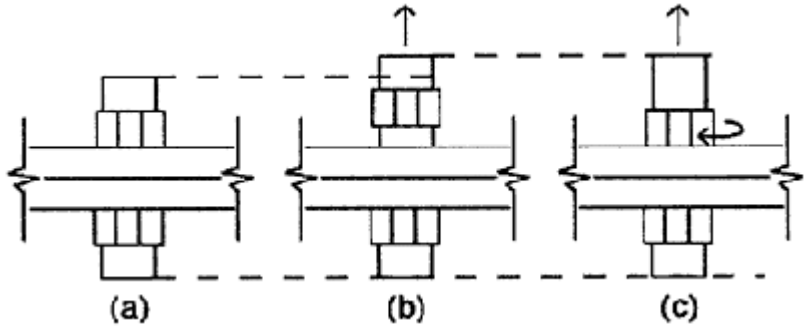


**Figure 17-5** Hydraulic Tensioning

(g) Torque bolts in a star pattern, particularly for flanges with 8 or more bolts (torque the bolt at 12-o'clock, followed by 6-o'clock, followed by 3-o'clock, followed by 9-o'clock, etc.) [ASME PCC-1]. For very large flange assemblies (such as bolted heads of heat exchangers) the star pattern would apply to groups of bolts (for example, bolt first three bolts at 12-o'clock, then three bolts at 6-o'clock, etc.). Torque in incremental passes since, as a new bolt is torqued, the adjacent bolt preload is partially lost, an effect called elastic interaction [Bibel].

Bolting in three passes (at 33%, 66% and 100% of the required torque) helps achieve an even preload of all bolts around the flange. Sometimes the three passes are followed by two last passes at 100% torque. Pressure Mylar film that changes color with the intensity of compression at each point of its surface can be used during trials to visualize the distribution of preload on a gasket.

(h) After bolt-up, visually verify proper assembly and engagement of the bolt, screw or stud threads into the nut. As a matter of good practice, one or two threads should be visibly protruding from the nut to confirm complete engagement of the bolt or stud into the nut.

Piping codes will specify minimum thread engagement for flange joints. While the gas pipeline code (B31.8) requires full engagement, the process piping code (B31.3) would permit to be one thread short of full engagement. The minimum thread engagement necessary to develop the joint capacity is [ASME B1.1]

$$LE_{ext} = \frac{\left[D - \frac{0.9743}{1/P}\right]^2}{2(1/P)D_{min}\left[\frac{1}{2(1/P)} + 0.57735(d_{2min} - D_{1max})\right]}$$

$LE_{ext}$=length of minimum engagement based on external threads, in

1/P=number of threads per inch, 1/in

D=basic major diameter, in

$D_{min}$=minimum major diameter of external threads, in

$D_{1max}$=maximum minor diameter of internal thread, in

$d_{2min}$=minimum pitch diameter of external thread, in

(i) Verify leak tightness during a system pressure test, or at start-up as specified in the construction or maintenance code.

(j) Hot torquing: ASME VIII Division 2, article 3.5, acknowledges that preloads do relax in service and may have to be reestablished by torquing in service. This is particularly true for hot service since the line tends to expand and rotate at flange joints. Re-torquing in-service may also become necessary when a leak develops. If done at all, torquing a flange in service must be approached with the utmost care. It is advisable to avoid the practice altogether for steam, high pressure or temperature, and hazardous fluids. A shield should be used for high pressure, flashing fluid, highly corrosive or damaging fluids.

### 17.8.2 Closing the Gap

The following situation is not uncommon: the piping spools have been assembled in the field and when comes time to bolt-up the final flange the gap between flange faces is wider than the gasket. For example, the gap between raised faces is 1.5” while the gasket is 1/8” thick. The ideal solution would be of course to replace a pipe spool to add the missing 1.5” length of pipe. But sometimes this may not be feasible. In this case, the assembler has two options: pull the line or use a Dutchman. The first option, line pull is equivalent to a cold spring procedure (Chapter 7). The second option consists in making a steel ring the size of the raised faces (a Dutchman or, as it is called in Spanish, a pancake) with serrated finish on both faces, and inserting the Dutchman in the following sequence: flange face—gasket—Dutchman—gasket—flange face. The bolt torque would have to be calculated to achieve the desired preload of the assembly. Piling up gaskets in contact one against the other is not an option, and is explicitly prohibited by ASME B31 construction codes.

## 17.9 NUTS AND BOLTS

### 17.9.1 Definitions

A bolt is an externally threaded fastener, typically headed, generally used with a nut. It fits in an oversized hole and is tightened by an end nut. A stud is an externally threaded fastener, typically headless, generally used with one or two nuts. A screw is an externally threaded fastener, generally used without a nut. It normally fits in a threaded hole. A nut is an internally threaded fastener to mate with bolts, screws or studs.

There are three thread classes. Class 2 has a large allowance (play), class 3 has a smaller allowance and class 5 has a negative allowance and must be force twisted. In defining thread class, a letter A refers to external threads, B to internal threads.

Unified thread sizes are identified by "UN" symbols [ASME B1.1, ASME B18.2.1]. UN may have a flat or rounded root contour. UNR and UNJ have rounded root contours. The prefix "C" refers to coarse threads (fewer threads per inch than the standard thread), "F" refers to fine threads (more threads per inch) and "EF" refers to extra fine. For example 3/8–20UN-2A stands for 3/8" nominal diameter, 20 threads per inch, UN standard thread, class 2A loose fitting.

### 17.9.2 Bolt Fabrication

Bolt materials conform to one of several ASTM specifications, which will be reviewed in 17.9.3. The bolt manufacturer buys rolls of wire of specified diameter, chemistry, mechanical properties and heat treatment. The bolt, screw or stud can then be cold formed, warm formed (around 500°F), forged (around 2200°F) and machined. The head is formed round by blows then cut to hexagonal "hex" shape. The shank is extruded to proper diameter then threaded. The fabrication process strives to maintain the original wire's strain hardened properties to achieve high yield and tensile strength. Hardness is a standard material specification requirement for many grades of bolt material. Charpy impact toughness is a supplementary requirement to be specified by the buyer.

### 17.9.3 Bolt Specifications

(a) ASTM A 193 Alloy Steel Bolting Material for High-Temperature Service, comes in several grades, for example:

B7 is a low alloy (0.37 to 0.49 C, 0.75 to 1.20 Cr, 0.15 to 0.25 Mo). It is a popular choice, with high strength and ductility, for use between 0°F and 550°F. Often chosen as the standard bolt for carbon steel and low alloy steel applications. For size 2.5" and under, $S_u$=125 ksi, $S_y$=105 ksi. Rockwell C28 to C30, and if heat treated C40. High hardness makes it susceptible to hydrogen stress corrosion cracking (SCC, Chapter 20) at −150°F to 250°F. Can be a problem in solutions of anhydrous ammonia ($NH_3$), hydrogen cyanide (HCN), hydrogen fluoride (HF), hydrogen sulfide (HS). The approximate price ratio is 1 (reference) uncoated, and 2 if Teflon™ coated.

B7M is similar to B7, but "M" denotes modified steel because it has the same chemistry as B7 but is heat tempered at 1150°F after threading to reduce risk of stress corrosion cracking (SCC). Recommended use~−400°F to 1000°F. In critical applications may impose supplementary requirement of 100% hardness testing. May bow or form an

oxide layer at tempering. May be dual marked “B7M, L7 and L7M” if meets the low temperature Charpy test requirements of A 320. The approximate price ratio is 1 uncoated, 2 Teflon coated, 2 uncoated 100% tested, 3 Teflon™ coated 100% tested.

B8 class 1 is a 304 Stainless Steel (0.08 max C, 18 to 20 Cr, 8 to 10.5 Ni) with $S_u$=75 ksi, $S_y$=30 ksi. “Class 1” refers to Cr carbide solution treatment and “Class 1 A” refers to solution annealed in the finished condition for maximum corrosion resistance. Its approximate price ratio is 3.

B8 class 2 is similar to class 1, but in addition the material is strain hardened after carbide solution treatment to $S_y$=100 ksi, this increase in yield stress comes with increased risk of Hydrogen SCC. Its approximate price ratio is 4.

B8M class 1 is 316 Stainless Steel (0.08 max C, 16 to 18 Cr, 10 to 14 Ni, 2 to 3 Mo) where “M” denotes Molybdenum. Its mechanical properties are $S_u$=75 ksi, $S_y$=30 ksi. “Class 1” refers to Cr carbide solution treatment and “Class 1 A” is solution annealed in the finished condition for maximum corrosion resistance. Its approximate price ratio is 6.

B8M class 2 is similar to class 1, but in addition it is strain hardened after carbide solution treatment (with increased risk of Hydrogen SCC). Its approximate price ratio is 7.

(b) ASTM A 307 Carbon Steel Bolts and Studs, 60,000 psi Tensile Strength. Grades A and B are unalloyed carbon steels. $S_u$=60 ksi, $S_y$ not specified. Grade A =0.06% max P, with B241 max. Grade B=0.15% max P, with B212 max (cast iron flanges). Black finish or galvanizing.

(c) ASTM A 320 Alloy Steel Bolting Materials for Low Temperature Service. Alloy steel (L7 series, L43 and L1) and stainless steel (B8 304 and B8C 347) bolts with a minimum Charpy impact energy requirement. L7 series, L43 and L1; $S_u$= 100 to 125 ksi, $S_y$=80 to 105 ksi. B8 series; $S_u$=75 to 125 ksi; $S_y$=30 to 100 ksi. Recommended use of L7~0°F to 550°F.

(d) ASTM A 325 High Strength Bolts for Structural Steel Joints. Alloy steel, Type 1 has limits on C, Mn, P, Su; Type 3 also has limits on Si, Cu, Ni, Cr, Va, Mo, Ti. $S_u$=105 ksi, $S_y$=81 ksi.

(e) ASTM A 354 Quenched and Tempered Alloy Steel Bolts, Studs and Other Externally Threaded fasteners. Bolts and studs made of C=0.3% to 0.5% steel with a maximum on P and S. $S_u$=115 ksi to 150 ksi; $S_y$=99 ksi to 130 ksi depending on size and grade. Often chosen when 150 ksi strength is needed.

(f) ASTM A 449 Quenched and Tempered Steel Bolts and Studs. Bolts and studs made of C=0.3% to 0.5% steel with a min on Mn, and a maximum on P and S. $S_u$ =90 ksi to 120 ksi; $S_y$=58 ksi to 92 ksi depending on size and grade.

(g) ASTM A 453 Bolting Materials, High Temperature, 50 to 120 ksi Yield Strength With Expansion Coefficients Comparable to Austenitic Steels. Stainless steel bolts. Come in 4 grades (660 most commonly used, also available as 651, 662 and 665). $S_u$=95 to 155 ksi, $S_y$=50 to 120 ksi. May be used in the creep range (above 800°F) with a 1200°F 100 hours tension test.

(h) ASTM A 490 Heat-Treated Steel Structural Bolts, 150 ksi Minimum Tensile Strength. Three types of high strength steel bolts, with $S_u$=150 to 170 ksi. Type 1 (alloy steel) is most common. Type 3 has good resistance to atmospheric corrosion.

(i) ASTM A 540 Alloy Steel Bolting Materials for Special Applications. Five grades of alloy steel B21 (Cr-Mo-Va), B22 (Cr-Mo), B23 (Cr-Ni-Mo), B24 (Cr-Ni-Mo) and

B24V (Cr-Ni-Mo-Va). $S_u$=120 to 165 ksi and $S_y$=105 to 150 ksi, with Charpy impact test. Several supplementary requirements include UT, fracture, MT, Charpy at elevated temperature (212°F).

(j) ASTM A 564 Hot-Rolled and Cold Finished Age-Hardening Stainless Steel Bars and Shapes. Stainless steel (for example, type 630 is 0.07 max C, 15 to 17.5 Cr, 3 to 5 Ni), although not a bolt specification, can be considered for stainless steel, corrosion resistant, and high strength applications. They can be machined in the solution-annealed condition (1900°F no $S_u/S_y$ requirement), then can be age hardened (800°F 1 hr. to 1100°F 4 hrs) to the required mechanical properties ($S_u/S_y$ =190/170 ksi to 140/125 ksi).

(k) ASTM F 468 Nonferrous Bolts, Hex cap Screws, and Studs for General use, covers copper, brass, aluminum, bronze, nickel and titanium alloys.

(l) ASTM F 593 Stainless Steel Bolts, Hex cap Screws, and Studs, covers 304, 316, 321 and other common stainless steel alloys. Mechanical properties include yield, tensile and hardness.

### 17.9.4 Nut, Washer Specifications

(a) ASTM A 194 Carbon and Alloy Steel Nuts for Bolts for High-Pressure and High-Temperature Service. This specification for nuts parallels ASTM A 193, with nut grades 1 to 8, including 7, 7M, 8 and 8M. Grades 2H (carbon steel) and 7 (alloy steel) are often used for in high pressure or temperature applications.

(b) ASTM A 563 Carbon and Alloy Steel Nuts. Covers several grades of carbon steel and alloy steel nuts. Addresses zinc plating. A non-mandatory appendix gives guidance regarding bolt-nut combinations.

(c) ASTM F 436 Hardened Steel Washers, carbon or low alloy steels, have a prescribed hardness.

(d) ASTM F 467 Nonferrous Nuts for General Use, covers copper, brass, aluminum, bronze, nickel and titanium alloys.

(e) ASTM F 436 Hardened Steel Washers. Applies to fasteners ½" to 4" diameter. The required mechanical properties are limited to a hardness range. Supplementary requirements apply to surface roughness of 750 micro-inches. Is a common choice for carbon steel washers.

(f) ASTM F 594 Stainless Steel Nuts, covers 304, 316, 321 and other stainless steel alloys. Mechanical requirements are proof stress and hardness.

Washers, while not required, help achieve and maintain the proper bolt stretch by reducing friction and embedment of the bolt head in the flange. Washers are typically ASTM F 436 Type 1 material for carbon steel bolts with ASTM A 194 Gr.7 nuts. For stainless steel bolts (ASTM A 193 Gr. B8 or B8M) and nuts (ASTM A 194 Gr.8 or 8M and ASTM F 594) the washer is 18/8 stainless steel.

### 17.9.5 Restrictions

Restrictions are imposed by the gas pipeline code on the use of bolt materials. ASTM A 193, ASTM A 320, ASTM A 354, ASTM A 449 and ASTM A 1 94 are permitted. ASTM A 307 Grade B is permitted only for class 150 and 300 flanges, from −20°F to 450°F and with A307 nuts.

### 17.9.6 Corrosion Prevention

Often times, it is necessary to protect flange bolts against corrosion. This can be accomplished by galvanic protection, such as the application of a 2 to 4 mils layer of zinc or cadmium. Thicker coatings will require re-tapping the nut to a larger diameter. In highly corrosive environment this coating may seize and be difficult to disassemble without cutting the bolt. If the temperature does not exceed 400°F, coatings of polyamide have been used with PTFE fill. For temperatures between −400°F and 550°F, a bonding layer of phosphate can be applied over the steel, followed by a polymer top coat about 1 mil thick. These coatings have the side advantage of reducing friction during assembly. Lubrication is usually not necessary for new-coated bolts. Many coats are sufficiently thin so that the nut need not be re-tapped.

## 17.10 MAINTENANCE

### 17.10.1 Flange Assembly Sequence

Having reviewed many aspects of flange joining, let's list a work sequence for removing and reinstalling a valve-to-pipe flange joint.

Drain and vent the line.

Make sure the line is supported on both sides of the flange to support the pipe and maintain alignment as the flange is unbolted. Add temporary supports if necessary.

Double check that the line is drained and vented (depressurized).

Loosen nuts, progressively and evenly all around.

Remove nuts and bolts, studs or screws.

Separate flange faces, using a wedge if necessary, without scratching the flange faces.

Completely remove old gasket.

Reface flange if necessary.

Confirm valve type, size and packing.

Check bolts or studs types, length.

Clean bolts or studs as needed, with a compatible solvent.

Wire brush bolts, studs and nuts as needed, with a compatible wire material.

Confirm gasket type, style and size.

Visually inspect flange faces for imperfections (section 17.4).

Confirm flange faces are smooth or serrated as required by gasket.

Clean flange faces as needed, with a compatible solvent.

Visually inspect inside the pipe.

Clean pipe of scale, dirt, etc.

Clean valve with air or flush with water.

Check manual valve, open and close.

Align and support pipe.

Lubricate bolts and nut threads, unless the bolt is a new-coated bolt.

Place valve between pipe flanges.

Align flanges with drift pin.

Align flange faces to within 1/16" per foot of face (section 17.8).

Align bolt holes to within 1/8".

Insert bottom bolts on one flange, finger tight.
Use washers as necessary.
Select new gasket (section 17.3), do not reuse gasket.
May lightly spray the gasket with compatible adhesive.
Insert gasket (keep gasket horizontal on flat board until assembly).
Insert all bolts, finger tight.
Repeat on opposite valve nozzle.
Torque bolts (section 17.8).
Tighten opposite valve nozzle.
It is good practice to have at least one thread protrude from nut (section 17.8).
Use two more passes: 1 clockwise and 1 counter-clockwise.
Re-torque after 24 hours if necessary and permitted (section 17.8).

### 17.10.2 Replacing a Gasket

Be sure line is empty.
Loosen bolts, opposite to mechanic.
Be sure line is empty (again).
Spread flange faces open, use flange spreader if necessary.
Remove gasket.
Scrape out gaskets parts if any left (do not scratch flange face).
Cement gasket to both sides of blank.
Slip gasketed blank between flanges.
Insert freshly lubricated bolts (may need longer bolts because of blank).
Tighten in cross-over pattern, in three passes.

### 17.10.3 Welding a Slip-On Flange

Clamp and support free pipe end.
Slide flange, leaving an offset between the flange and the pipe end. Typically, ¼” for small bore pipe (2” NPS and smaller), 3/8” for 3” to 10”, and 5/8” above 10”.
Measure offset all around.
Use square to assure a 90° angle between the flange face and the pipe.
Four tack welds at 90° around the pipe.
Weld flange to pipe (section 17.2).
Inspect welds.

### 17.10.4 Leakage Diagnostics

The following checklist could help pinpoint the cause, or causes, of a pipe flange leak.
Wrong pressure and temperature rating of gasket.
Wrong flange class.
Wrong fluid service for gasket.
Incorrect gasket thickness.
Incorrect gasket modulus (too soft or too hard).
Incorrect bolt material, may have yielded.

Flange surface too smooth or defective.
Bolt torque not controlled during construction.
Bolt torque too small, insufficient preload, gasket creep.
Bolt torque too large, overstress of bolt or gasket, bent flange.
Bolting sequence not followed, uneven preload.
Cold spring load on flange.
Overpressure transient.
Operating forces on flange not considered.

### 17.10.5 Refinishing Flange Faces

When a flange face is scratched or gouged, it will be necessary to either repair the face or replace the flange. Repair of the flange face can be done by refinishing the face using a pre-qualified tool radius and feed rate to avoid reducing the flange thickness to a value below that specified in ASME B16.5. If the flange face is below the thickness specified in ASME B16.5, it will be necessary to increase the flange thickness by weld metal build-up before machining the surface. The weld metal, welding technique, welder qualifications and heat treatment, if required, must be properly selected and pre-qualified. Impurities remaining on the flange face during weld build up, such as pieces of gasket material or corrosion deposit, must be removed since they can result in weld defects. After repair, the finished surface should be examined to the requirements of ASME B16.5. Once the repaired flange is reassembled in the field, the joint should be leak tested as required by the construction Code.

## 17.11 REFERENCES

API 6A—Pressure Rating, American Petroleum Institute, Washington, D.C.
API 601—Metallic Gaskets for Raised face Pipe Flanges, American Petroleum Institute, Washington, D.C.
API 605—Large Diameter Carbon Steel Flanges (26” to 60”), American Petroleum Institute, Washington, D.C.
ASME II, Boiler and Pressure Vessel Code, Section II, Materials, Part D, table 3, American Society of Mechanical Engineers, New York.
ASME VIII, Boiler and Pressure Vessel Code, Section VIII, Pressure Vessels, Division 2, Appendix 2, American Society of Mechanical Engineers, New York.
ASME B1.1—Unified Inch Screw Threads, American Society of Mechanical Engineers, New York.
ASME B16.5—Pipe Flanges and Flanged Fittings, American Society of Mechanical Engineers, New York.
ASME B 16.24—Cast Copper Alloy Pipe Flanges and Flanged Fittings, American Society of Mechanical Engineers, New York.
ASME B 16.34—Valves—Flanged, Threaded, and Welding End, American Society of Mechanical Engineers, New York.
ASME B 16.36—Orifice Flanges, American Society of Mechanical Engineers, New York.
ASME B 16.47—Large Diameter Steel Flanges NPS 26–60, American Society of Mechanical Engineers, New York.

ASME B 18.2.1—Square and Hex Bolts and Screws Inch Series, American Society of Mechanical Engineers, New York.
ASME B31.8—Light Weight Flanges (25 psi), American Society of Mechanical Engineers, New York.
ASME B46.1, Surface Texture (Surface Roughness, Waviness, and Lay), American Society of Mechanical Engineers, New York.
ASME Boiler & Pressure Vessel Code, Section III, Division 1, Nuclear Components, American Society of Mechanical Engineers, New York.
ASME PCC-1 Guidelines for Pressure Boundary Bolted Flange Assembly, American Society of Mechanical Engineers, New York, NY.
ASTM A 574 Standard Specification for Alloy Steel Socket-Head Cap Screws, American Society for Testing and Materials, West Conshohocken, PA.
ASTM F 36, Standard Test Method for Compressibility and Recovery of Gasket Materials, American Society for Testing and Materials, West Conshohocken, PA.
ASTM F 37, Standard Test Methods for Sealability of Gasket Materials, American Society for Testing and Materials, West Conshohocken, PA.
ASTM F 152, Standard Test Methods for Tension Testing of Nonmetallic Gasket Materials, American Society for Testing and Materials, West Conshohocken, PA.
Bibel, G., Ezell, R., Bolted Flange Assembly: Preliminary Elastic Interaction data and Improved Bolt-Up Procedures, WRC Bulletin 408, January 1996, Welding Research Council, New York.
Bickford, J.H., Nassar, S., "Handbook of Bolts and Bolted Joints, Marcel Dekker, New York.
Bouzid, A.H., et. al., Tightness Prediction of Bolted Flanged Connections Subjected to External Bending Moments, PVP-Vol. 367, Analysis of Bolted Joints, American Society of Mechanical Engineers, New York.
Cullen, W.D., The Public Inquiry into the Piper Alpha Disaster, Cm 1310, Her Majesty's Stationary Office, London, 1991.
EPRI, Bolted Joint Maintenance and Applications Guide, Electric Power Research Institute, Electric Power Research Institute TR-104213s, December, 1995.
Faires, V.M., Design of Machine Elements, Fourth Edition, 1965.
Lancaster, J., Engineering Catastrophes, Causes Effects of Major Accidents, CRC Press, 2000.
Lu, C. et. al., Seismic Response of a Regulator in Memphis Gas Transmission System, Technical Council of Lifeline Earthquake Engineering, Monograph no. 6, ASCE, 1995.
MSS SP-44, Steel Pipeline Flanges, Manufacturers Standardization Society of the Valves and Fittings Industry, Vienna, VA.
Rodabaugh, E.C., Moore, S.E., Evaluation of Bolting and Flanges of ANSI B16.5 Flanged Joints—ASME Part A Design Rules, ORNL/SUB/2913–3, September 30, 1976, Oak Ridge National Laboratory, Oak Ridge, TN.

# 18
# Mechanical Joints

## 18.1 WHAT THEY ARE

Mechanical joints are pipe joints other than welded, brazed or soldered joints (Chapter 15) or flange joints (Chapter 17). Mechanical joints are versatile specialized fittings, developed and optimized for specific applications. They include swage fittings, easily assembled and disassembled, used on tubing systems; grooved couplings common in water lines and fire protection piping systems; quick disconnects used on hose ends, braided hose used at pump-pipe nozzles, flared tube fittings, and threaded joints common in small bore piping (NPS 2 and smaller).

Mechanical fittings are often labeled by the brand name or trademark of a company. From a user perspective, these fittings are quite useful since they do not involve welding, yet they provide leak tight service, and many of these mechanical joints can be easily disassembled and reassembled during maintenance.

Going back to the seven fundamentals of Chapter 2, before using a new type of mechanical joint, the user should obtain the following assurances:

(1) Materials: To avoid corrosion or degradation in service, the fitting materials (metallic parts and non-metallic O-rings, gaskets, etc.) and finish must be compatible with the existing pipe, the fluid, and the operating temperature. Materials must comply with a listed specification. Some fittings must also pass a fire test for service in refining or petrochemical applications, or a radiation compatibility test for service with radioactive fluids.

(2) Design: The fittings must be pressure rated in accordance with the applicable design code. The “rated pressure” listed in the vendor catalog should have been established by proof testing on a production sample to prove either (a) that the fitting can withstand without leak an operating pressure equal to three times the rated pressure, with allowance for a temperature derating factor, if the fitting is used at temperatures other than ambient, or (b) that the fitting is stronger than the matching pipe, in which case it will be assigned a pipe schedule number. In addition, the fabricator of a mechanical joint fitting should provide the user a stress intensification factor for the fitting (Chapter 7). This factor is used to design a system for flexibility if it operates above or below the ambient temperature, or to assess fatigue life in vibratory service. Many reputable vendors do conduct Markl type tests to establish their products’ stress intensification factor values (Chapter 7).

(3) Fabrication: in this case, fabrication consists in assembly, disassembly and reassembly of the joint, following precise step-by-step written instructions. For critical service, maintenance mechanics should be qualified for assembly of mechanical joints, usually by a representative of the manufacturer. If the fabrication of a mechanical joint

involves welding (for example, welding of end flanges to a braided hose), welding should follow a code-qualified procedure and be done by a code-qualified welder.

(4) Examination: the vendor should provide, together with the assembly instructions, the means for verifying the adequacy of the joint. This may take the form of go-no go gages, torque range, visual markings, etc.

(5) Leak testing: in the case of new construction, the joints are either pressure tested or leak tested with the system, depending on the requirements of the construction code. During maintenance, for example when making or remaking instrument tubing to periodically test an instrument, or to reassemble a manifold, it is not necessary to re-hydrotest the system every time, since there have been no modifications to the system. In critical systems, where the consequence of leakage is unacceptable, the user relies on the accuracy of the joining and examination process, steps (3) and (4), and an independent quality control of these steps is in order. In critical applications, the owner has the option to require a pressure test following maintenance work, if deemed necessary.

(6) Operation: operator rounds should include verification of leak tightness of mechanical joints in critical systems.

(7) Maintenance: in critical applications, maintenance mechanics should be trained in the making, remaking and examination of mechanical joints, preferably by the vendor's representative, or by a person certified by the vendor.

A serious accident resulted from poor maintenance joining: in March 2000, seven workers inhaled radioactively contaminated gas that leaked through a tube fitting. A controlled negative pressure of inert gas was normally maintained in a header (TN), by alternatively drawing a vacuum (TN to VAC through C1 and C2) or introducing inert gas (IG to TN through O2 and O1), Figure 18-1. The manifolds consist of 3/8" tubing, assembled by swage fittings, as is commonly the practice for tubing systems. After many years of service, the vacuum line was contaminated with radioactive elements. With time and radiation, the seat in isolation valve C1 had deteriorated and the valve leaked through the seat, allowing the entry of pressurized gas in section C1–C2, normally under vacuum. The gas would have been contained if it were not for a leak out of the swage fitting at the inlet of valve C1. The investigation revealed that the swage fitting was not properly assembled.

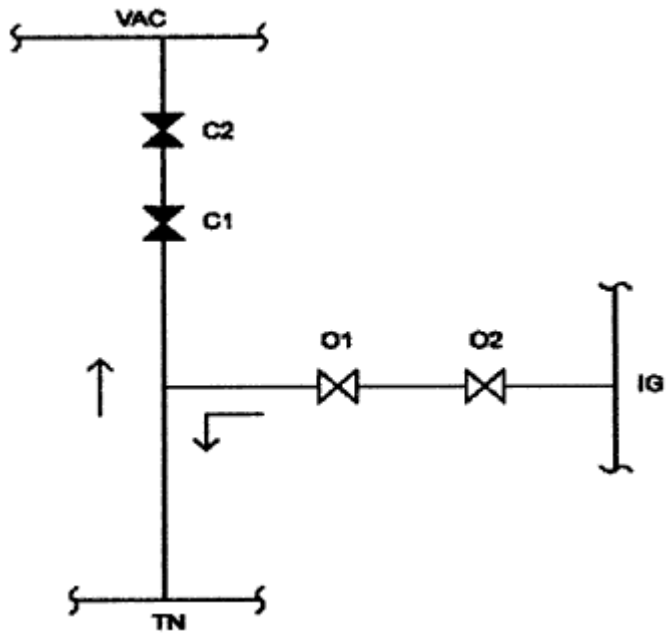


**Figure 18-1** General layout of Inert Gas and Vacuum Tube

## 18.2 SWAGE FITTINGS

To swage is "to reduce or taper an object as by forging or squeezing". This is indeed what swage fittings do. Swage fittings are commonly used to join tubing systems. They are typically made of a body (B, Figure 18-2), a ferrule (F, Figure 18-2), and a nut (N, Figure 18-2). As the nut threads into the body, it compresses the ferrule on the tube surface. The ferrule bites into the surface providing a fixed point to tighten the joint.

Since these joints were first patented in the 1940's, they have been continuously improved to achieve ease of installation, better gripping action and achieve their leak tightness. Today swage fittings are available with single and double ferrules and in a wide range of materials, coatings and heat treatment. Common ma-terials for swage fittings are stainless steel, copper, carbon steel, aluminum, and nickel alloys. To harden the ferrule, which permits a better gripping of the tube outer diameter, the ferrule is either work hardened (drawn), case hardened (carburized, Chapter 15), or plated (for example with chromium). Caution is in order since the hardening process may alter the metallurgy of the ferrule and lead to accelerated corrosion. It is therefore advisable, in corrosive service, to inquire not only on the chemical composition of the parts, but also on the type and effect of ferrule hardening.

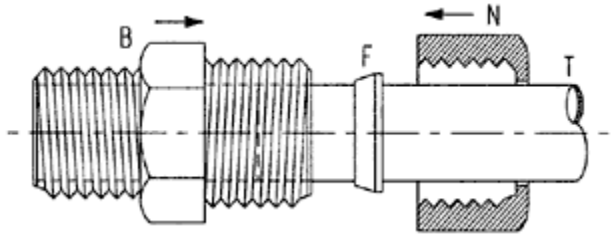


**Figure 18-2** Example of a Swage Fitting

The compression-fitting vendor will provide a catalog or an installation procedure with step-by-step instructions. These must be followed to achieve a leak tight joint. They typically include the following steps:

(1) Verify that the joint parts are correct.
(2) Cut square the tube ends.
(2) Clean and smooth the tube end.
(3) Align the tube straight with the fitting.
(4) Insert the tube into body to a prescribed depth.
(5) Tighten finger tight the nut into the body.
(6) Torque the nut a prescribed number of turns beyond step (5).
(7) Use a go-no go gage to verify the joining process.
(8) Steps (6) and (7) may vary when remaking a joint (reassembly after disassembly, as is typical with instrument or manifold tube joints).

## 18.3 GROOVED FITTINGS

A circular groove is cut or rolled into each end of two pipes. A soft gasket (G) overlaps the two pipe ends. The gasket has two shoulders (protrusions) that fit into each groove. The gasket bridges the two pipes ends. The two half shells of a mechanical clamp (MC) are placed over the gasket and bolted together, squeezing the gasket shoulders into the groove at the end of each pipe, providing a leak tight joint, Figure 18-3.

The seven fundamentals apply: (1) materials, in particular the long-term performance of the gasket. (2) Design, the pressure rating and the stress intensification factor, and—in this case—the flexibility of the joint where lateral loads are important, for example if grooved fittings are used in a fire sprinkler system that must be seismically qualified to suppress a fire in case of earthquake. Under lateral loads the pipe spans may rotate around the groove fittings, which acts as hinge point with finite flexibility. (3) Fabrication, field erection is quite efficient and must be done by qualified persons. There is typically a limit on misalignment of successive spans joined by grooved fittings. (4) Examination consists in verifying the adequacy of the groove, the gasket and the mechanical clamp, and the proper torque of the clamp bolts. (5) Testing, pressure or leak testing of new construction would follow the construction code requirements. Testing of maintenance activities (opening and remaking the joint with no change other than

possibly the gasket) may not need to be pressure tested. In critical systems, independent quality control of the machining and joining process is in order. (6) Operations should include visual inspection for leak tightness as part of routine operator rounds. (7) Maintenance mechanics should be trained in the disassembly and reassembly of the joints.

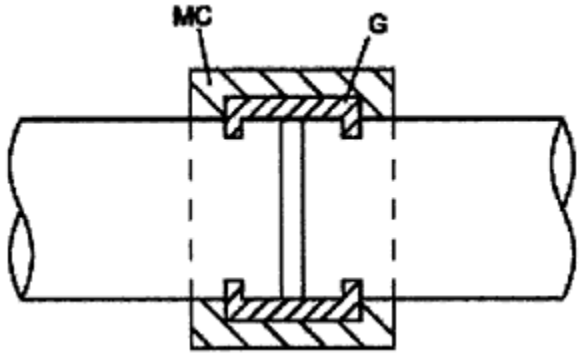


**Figure 18-3** Grooved Fitting

## 18.4 IN CONCLUSION

We have touched on only two of a multitude of specialty mechanical joints. The product line continues to grow to fill the needs of users who want an easily installed, easily disassembled and re-assembled joint with superior leak tightness, ideally as reliable as a welded joint. Reliable performance will depend on the manufacturer's and user's understanding of the seven fundamental points listed in section 18.1.

# 19
# Leak and Pressure Test

## 19.1 LEAK TEST AND PRESSURE TEST

A leak test is conducted at or below operating pressure to verify that a piping system is leak tight or, for certain applications such as waterworks or fire mains, that leakage is within prescribed limits. A pressure test is conducted above operating pressure, often times at 1.1 times to 1.5 times the system design pressure to verify the strength and leak tightness of a system. The leak and pressure tests defined by the construction codes are shell tests; their purpose is to verify that the fluid in the piping system does not leak out of the system. The construction code leak test is not a closure test meant to confirm that a closed valve does not leak through its seat. Seat leak tightness is addressed in Chapter 25. The leak test is a control on the quality of assembly during construction or maintenance. Most often, when leaks occur during testing they originate at mechanical joints, such as flanges or threaded connections. Leaks have also occurred, but much more rarely, in a defective weld or base metal, such as a cracked elbow, or in components, such as at a repair weld in a valve body.

The hydrostatic and pneumatic tests are pressure tests that challenge the system above the design pressure. But they do not replace pressure design of piping or components. The pipe size, fitting and component rating must still be sized in accordance with the design rules or qualified by proof testing to a pressure higher than the system hydrostatic or pneumatic test pressure.

In some cases, hydrostatic tests of pressure vessels have resulted in catastrophic ruptures of the vessel while conducting the test. In most cases, post-failure analyses indicated an unusually low impact toughness of the steel at room temperature (low Charpy V-notch toughness) aggravated by the presence of a weld flaw undetected by earlier examination. In these cases, the pressure test was instrumental in uncovering a material of poor quality or a weld defect. This is however an unacceptable and costly way of uncovering such problems.

In the oil and gas pipeline industries, hydrostatic testing is used to assess the fitness-for-service of a pipeline, based on fracture mechanics analysis. The logic being that if a pipeline can withstand a certain hydrostatic pressure it can be concluded that it does not contain flaws larger than a certain size. In Chapter 21 we will see that a pipeline tested at a hoop stress of 100% its specified minimum yield stress (SMYS) will not contain a flaw longer or deeper than the limits of ASME B31G, otherwise the line would have failed at these flaws during hydrotest.

## 19.2 LEAK AND PRESSURE TEST METHODS

There are several methods for leak and pressure testing a piping system. ASME B&PV Code, Section V Article 10, provides a good reference for leak testing techniques. Following are the most common leak test methods used for industrial piping systems and pipelines.

Bubble test (ASME B&PV Section V Article 10, Appendix I): air or an inert gas is introduced in the pipe, at a relatively low pressure, in the order of 15 psig. A bubble solution is applied at welds and joints. The joints are then visually inspected for signs of bubbles, which would indicate a leak. The bubble solution may contain inhibitors to avoid corrosion. The sensitivity of the method is in the order of $10^{-4}$ to $10^{-5}$ std. cc/sec.

Vacuum box (ASME B&PV Section V Article 10, Appendix II): a bubble forming solution is applied at welds and joints. The weld or joint is then covered with a sealed box with transparent windows, as shown in Figure 19-1. A slight vacuum of about—2 psid is drawn into the box and held for a short time, typically less than a minute. The joints in the vacuum box are then visually inspected for signs of bubbles, which would indicate a leak. The method is particularly well suited for the leak testing girth welds in pipes. The sensitivity of the method is in the order of 1 to $10^{-2}$ std. cc/sec.

Sensitive leak test (ASME B&PV Section V Article 10, Appendix IV and V): the sensitive leak test technique consists in detecting leakage of a tracer gas, such as Helium or $SF_6$, into or out of the tested system [ASTM E 432]. There are four sensitive leak test methods, illustrated in Figure 19-2. Either a gas is introduced inside the component and the air is sampled around it for traces of gas (sniffing probe or bell jar), or the component is immersed in gas or joints are sprayed with gas, and the air inside the component or system is sampled for traces of gas (hood or tracer spray). A detector probe is used to detect small amounts of leakage by ionization and magnetic field deflection of the tracer gas. The sensitiv-ity of the method ranges from $10^{-2}$ to $10^{-10}$ std. cc/sec, depending on the particular option and method of leak detection selected.

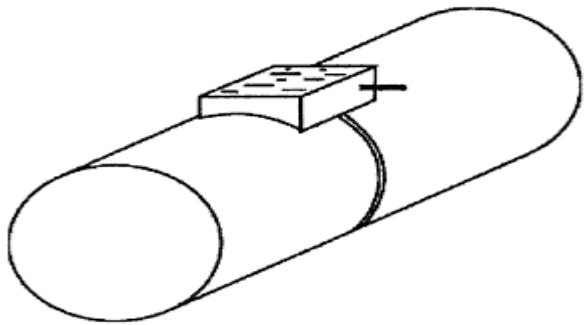

**Figure 19-1** Leak Box Test Arrangement

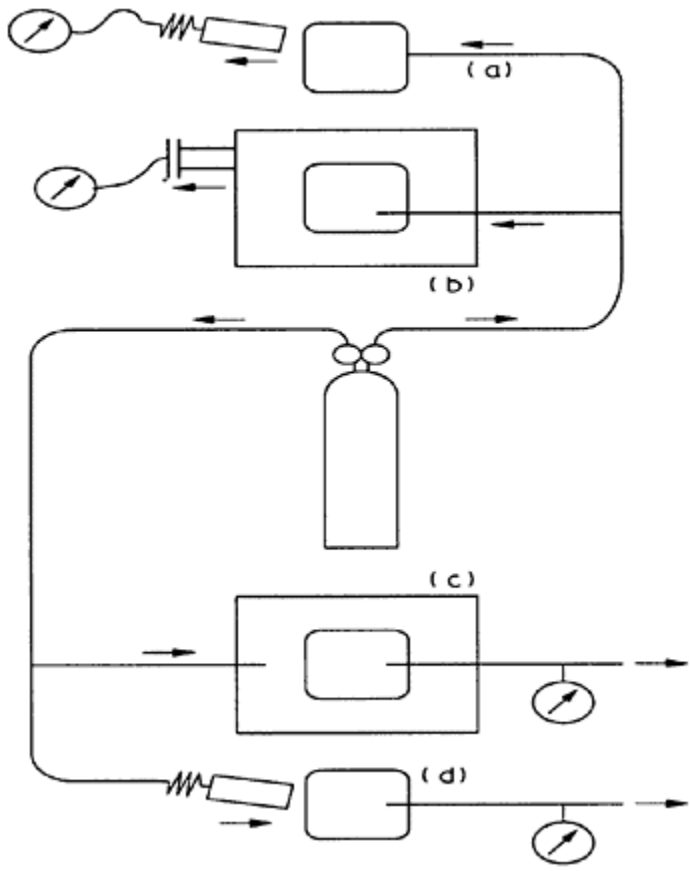


**Figure 19-2** Sensitive Leak Test Arrangements

(a) Sniffing Probe, (b) Bell Jar, (c) Hood, (d) Tracer Spray

Hydrostatic test: The piping system is filled with water to a pressure specified by the construction code, Figure 19-3 at top. Note the isolation valve, the recommended relief valve, and the easily visible pressure gages. Joints are inspected for evidence of leakage. To this end, the joints should be visible, bare, and uninsulated. The sensitivity of the method is in the range of 1 to $10^{-2}$ std. cc/sec. As a practical point of reference, a liquid leak rate of 1 std. cc/sec corresponds approximately to 1 gallon per hour. The power piping code, ASME B31.1 would permit the leak test to be conducted after insulation is installed, provided the hydrostatic pressure is maintained for a sufficiently long time; low points should be left uninsulated to permit the detection of leaking water. The hydrostatic test pressure is defined in ASME B31.3 as

$P_{test}=1.5\ P_{design}\ (S_{test}/S_{design})$

$P_{test}$=test pressure, psi
$P_{design}$=design pressure, psi
$S_{test}$=allowable stress at test temperature, psi
$S_{design}$=allowable stress at design temperature, psi

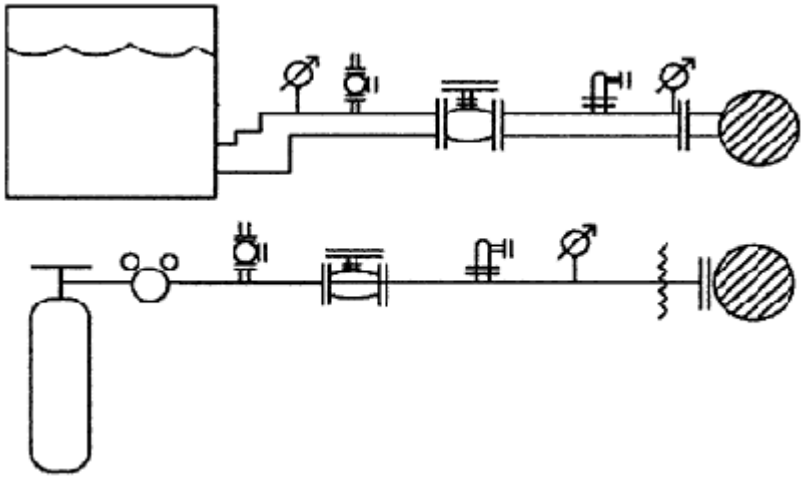

**Figure 19-3** Hydrostatic (top) and Pneumatic (bottom) Test Arrangements.

The correction $S_{test}/S_{design}$ has an important practical consequence when hydrostatically testing piping systems in the field: the leakage of valve packing during hydrotest. Indeed, take for example a carbon steel system with a design temperature of 650°F. The correction factor is 20 ksi/17 ksi=1.18. The piping system will be tested at a pressure of $1.5\times1.18\times P_{design}$. Yet, the vendor has only tested the valve at 1.5 times its rated pressure [ASME B 16.34]. Therefore, if the system design pressure is equal to the valve pressure rating, the valve will be exposed for the first time to a pressure $1.5\times1.18\times P_{rated}$, in this case the field hy-drotest pressure is 18% larger than its manufacturer tested pressure of $1.5\times P_{rated}$, and as a result there are cases where the valve packing leaked during the field hydrotest.

New pipelines are hydrostatically tested before commissioning and service. Hazardous liquid pipelines operating at a hoop stress larger than 20% of SMYS (minimum specified yield stress) are tested in accordance with ASME B31.4 at 1.25 times the design pressure for a minimum of four hours [ASME B31.4, 49CFR195]. If the line is covered and the joints are not visible, the test is extended for four more hours at a pressure of 1.1 times the design pressure. A pipeline designed for the maximum hoop stress of 72% SMYS will therefore experience a stress of 1.25×72%=90% SMYS during hydrotest. Conversely, a pipeline cannot be operated at a pressure that exceeds 80% of the hydrostatic test pressure, which corresponds to 80%×90% SMYS=72% SMYS.

Up to the 1950's, many gas transmission pipelines were tested with air or gas slightly above maximum operating pressure. As a result of failures, operating and insurance companies and regulators imposed a hydrostatic test at 1.1 to 1.4 times the maximum allowable operating pressure (MAOP) depending on the location class (population density) [ASME B31.8, 49CFR192].

A pig is inserted in front of the hydrostatic water to sweep all the air out of the line through open-air vent valves at high points. This will insure that the pipeline is full of liquid, with no trapped air pockets. There are three reasons why air has to be removed when hydrotesting a pipeline: (1) to reduce the energy trapped in the line, so that a leak will not turn into a large rupture (Chapter 4), (2) to avoid pressure fluctuations caused by pumping water against trapped air bubbles (Chapter 9), (3) to maintain the siphon effect to balance static heads caused by ups and downs in elevation. A pipeline can be tested in sections, to reuse the test water.

When testing pipelines, a pressure-volume (P-V) chart is used together with a calculation of projected volume vs. pressure to compare volume of water pumped (pump strokes) to corresponding pressure increase [API 1110, Kiefner]. If the pressure does not rise as predicted with pumped volume, this would indicate either a leak or plastic deformation of the pipe. The volume of water required to fill a pipeline and raise its pressure from atmospheric pressure to a test pressure P is [McAllister]

$V_{test}=V_{pipeline}\times F_{WP}\times F_{PP}\times F_{PWT}$

$V_{pipeline}=0.0408\ d^2L$

$F_{WP}=[1–4.5\ 10^{-5}\ (P/14.7)]^{-1}$

$F_{PP}=1+(D/t)(0.91P/30)\ 10^{-6}+3.6\ 10^{-6}\ (T-60)$

$F_{PWT}=[1+(T-60)\ 18.2\ 10^{-6}]/FWT$

$V_{test}$=volume of test water to fill and pressurize the line, gallons

$V_{pipe}$=pipe volume, gallons

$F_{WP}$=correction factor for water compressibility

$F_{PP}$=correction factor pipe volume increase with pressure

$F_{PT}$=correction factor for pipe and water temperature change

D=pipe inner diameter, in

L=pipe length, ft

P=test pressure, psi

D=pipe outside diameter, in

T=pipe temperature, °F

$F_{WT}$=factor 1.0 at 60°F [McAllister]

Pneumatic test: the piping system is filled with air or gas, and slowly pressurized to a pressure specified by the construction code, such as 110% of the design pressure for ASME B31.3 pneumatic tests. An arrangement of the test assembly is shown in Figure 19-3, at bottom. Note the pressure regulator, the isolation valve, the mandatory safety valve, the easily visible pressure gages, and the barrier or rope placed as a safety precaution. Joints are inspected for evidence of leaks, either using bubble solution or by monitoring the pressure drop.

Pressure change test (ASME B&PV Section V Article 10, Appendix VI): this pressure testing technique is also referred to as pressure decay (rate of fall) test. The system is pressurized, typically with water, and sealed. The pressure is monitored at least every

hour for large systems, during a sufficiently long period of time to detect any pressure drop that would indicate leakage. For testing small volumes (such as a small vessel) 15 minutes may be sufficient. The sensitivity of a rate of fall test is 1 to $10^{-2}$ std. cc/sec. Note that the rate of fall has to be corrected because the pressure in the sealed system will increase if the ambient temperature increases and vice-versa. The increase in pressure due to a temperature rise $\Delta T$ of a trapped liquid can be obtained by solving the following set of algebraic equations [Mohitpour, McAllister]

$A-B\ C+D=0$

$A=d^2\ L\ F\Delta T$

$B=\{d+\alpha\Delta Td+(2PR/E)\ [R^2(1+v)+r^2(1-2v)]/(R^2-r^2)\}^2-d^2$

$C=L+L\alpha\Delta T+(PL/E)\ r^2\ (1-2v)/(R^2-r^2)$

$D=P\ d^2\ L/B$

d=internal pipe diameter, in

E=Young's modulus of pipe material, psi

F=thermal expansion of fluid

L=length of isolated pipe, in

P=pressure, psi

R=outer radius, in

r=inside radius, in

$\alpha$=coefficient of thermal expansion of pipe material, 1/°F

$\Delta T$=change pf temperature of trapped fluid, °F

v=Poisson ratio of pipe material

For vacuum systems, the test is reversed, and the pressure increase (rate of rise) would be measured to detect in-leakage. The sensitivity of a rate of rise test is in the range of 1 to $10^{-5}$ std. cc/sec.

Ultrasonic leak detection (ASME B&PV Section V Article 10, Appendix X): a probe is used to detect ultrasounds, in the range of 20 to 100 kHz, emitted by gas leakage through a small hole or crack.

In-service leak test: the in-service leak test is a visual inspection of pipe joints during system startup, at normal operating pressure.

## 19.3 CHOICE OF TEST METHOD

Neither the constructor nor the owner has much latitude when it comes to leak or pressure testing. The type of test and the test pressure are typically specified by the construction code, federal, state or local regulations. In the construction of pressure vessels or piping systems, pressure testing is typically a hold point witnessed by the owner's inspector, or a third party inspector. Leak testing must include all the new joints of a piping system. Even if new piping subassemblies are tested in the pipe shop, the field joints between these subassemblies must be tested in the field.

In power and chemical process plant piping, the leak test is a pressure test typically a hydrostatic test at 150% of the design pressure. For systems with low design pressure, typically below 150 psi, and non-flammable, non-toxic fluids, the pressure test may be replaced by an in-service leak test during system startup, if permitted by the construction

code or the applicable regulation. For example, an in-service leak test is typical for building services and in non-critical process systems operating below 150 psi and 366°F. In industrial process plants, for piping systems operating above 150 psi, when the hydrostatic or pneumatic tests cannot be conducted, they may be substituted by full radiography of girth butt welds, PT or MT of fillet welds and an in-service leak test. This may be the case for closure welds, which are the last welds in a new construction, and the tie-ins between new and existing piping. Where the tie-in is at a flange or mechanical joint that can not be hydrostatically tested, then the joint assembly should follow written instructions and be witnessed by an independent inspector.

Oil and hazardous liquids pipelines (ASME B31.4) are hydrostatically tested, typically at 125% the design pressure, for at least 4 hours if the system operates at a hoop stress PD/(2t) in excess of 20% of the yield stress. If the joints are not visible, the first test is to be followed by a second test for at least four hours at 110% the design pressure [49 CFR 195, ASME B31.4].

Leak testing of gas pipelines and distribution systems depends on the location of the pipeline and the operating hoop stress. For a hoop stress below 100 psi, an in-service soap bubble leak test is often sufficient. For hoop stresses above 30% of yield, the pipeline is hydrostatically tested at pressures ranging from 110% to 140% of design pressure, depending on the pipeline location [49 CFR 192, ASME B31.8].

Keep in mind that the hydrostatic test pressure is the minimum pressure to be achieved in the system, therefore, because of elevation heads, low points will see a pressure in excess of the hydrostatic test pressure. But in no case should the hydrostatic hoop stress in the pipe exceed its yield stress or, preferably, 90% of the yield stress, except when the hydrotest is used as a means to check for flaws in pipelines (Chapter 21).

## 19.4 CONDUCT OF TEST

A leak test has to be competently planned and conducted [ASTM E 479, API 1110]. The following guidance applies to hydrostatic tests.

### 19.4.1 Plan the Test

(1) Select and train test inspectors, preferably personnel qualified for visual inspections.

(2) Prepare written test procedures and develop forms to record date, system boundaries, fluid, pressure, examiner certification.

(3) Decide on hold points for quality or owner inspections.

(4) Walk down the system to verify that construction is correct and complete.

(5) Verify proper location and condition of vents and drains.

(6) Make sure supports are completed and temporary supports are in place, if required.

(7) Spring hangers should be pinned if not sized to support the water weight.

(8) If the test pressure exceeds the design pressure, restrain expansion joints to safely sustain the test pressure.

(9) Verify test boundaries and valve alignments, use a piping line diagram to indicate test boundaries, valve status, vents, drains, pressure gages and test pressures.

(10) Verify the pressure rating of all components and fittings against the test pressure. Note that for the purpose of hydrostatic testing, a B16.5 flange can withstand 1.5 times its rated pressure, but an instrument should not be tested above its indicated range.

(11) Disconnect equipment, components and gages that can not withstand the test pressure.

(12) Make sure joints are visible, uninsulated. They could be painted. Joints that were successfully tested in the shop need not be reexamined in the field.

(13) Make provisions for a relief valve close to the fill connection. Temporarily remove or seal system relief valves, since the test pressure will exceed the design pressure and relief valve set point.

(14) Install test relief valves. For process piping a test relief valve is recommended during hydrotest and required during a pneumatic test.

(15) Chose a clean fluid compatible with the pipe. For example, use water with less than 250 ppm chlorides with stainless steel. In one case, untreated hydrotest water was permitted to stay stagnant inside several stainless steel systems for a period of five months, resulting in widespread pitting corrosion and costly replacements.

(16) Avoid testing in cold weather, below 40°F for carbon steel. If using antifreeze, plan for its disposition.

(17) For pneumatic tests, rope off, and shield if necessary, the test area. To select the rope-off distance on pneumatic tests, consider the equivalent TNT approach described in Chapter 4.

### 19.4.2 Conduct the Test

(1) Fill from bottom and vent air at high points. If air is entrapped, it will be difficult to reach the test pressure, and the trapped pressurized air bubble could be a source of catastrophic rupture of the pipe during hydrostatic testing.

(2) Consider a first plateau at low pressure.

(3) Pressurize slowly, in plateaus, to permit the system temperature to reach equilibrium.

(4) Watch for solar heating, refer to formulas for pressure increase with ambient temperature.

(5) Maintain the test pressure for 10 minutes during visual inspection.

(6) Wipe condensation as it occurs. It is preferable to heat the fluid slightly or test above the dew point. Dew points are given in Table 19-1. For example, from Table 19-1, if the ambient temperature is 70°F and the humidity is 50%, dew will form if the pipe wall is at 50°F.

(7) Examine all joints.

**Table 19-1** Dew Point in Air (°F), given Ambient Temperature and Humidity

| Humidity % | 30°F | 50°F | 70°F | 80°F |
|---|---|---|---|---|
| 90% | 28 | 47 | 67 | 87 |
| 70% | 22 | 40 | 60 | 78 |

| | | | | |
|---|---|---|---|---|
| 50% | 15 | 31 | 50 | 67 |
| 40% | 11 | 26 | 43 | 61 |
| 30% | 4 | 20 | 36 | 52 |

### 19.4.3 Plan for Leaks

(1) If leakage occurs, drain, repair and reexamine the piping, then retest.

(2) Unless otherwise required by regulations, the owner may wave retesting for minor repairs, such as flange retightening, or welded repairs that do not penetrate the pipe wall.

(3) For bolted joints, bolts may be retightened if permitted by the owner. Retightening bolted joints should not be permitted during pneumatic testing, instead, the test should be interrupted and the system depressurized to repair the leak.

### 19.4.4 Drain and Dry

(1) Flush and dispose of the test water.

(2) Dry by reducing the line pressure below atmospheric pressure, close to vacuum. Any water is then entrained as a low-pressure vapor outside the pipe. Drying may also be accomplished by blowing dry air or nitrogen in the system. Drying of long pipelines may be accomplished by several runs of polyurethane foam pigs followed by blowing dry air. The drying operation is completed when the outlet air is measured to be dry. In pipelines, drying may be accomplished by running batches of liquid methanol to absorb moisture.

## 19.5 ISOLATION

In practice, it is sometimes difficult to find an isolation point, such as a gate valve, to isolate a new subsystem to be tested from the rest of an existing piping system. Inflatable plugs, Figure 19-4 top, can be used to isolate sections of low-pressure systems. They are commonly used to test water lines.

High-pressure pipe stoppers, discs or pistons with neoprene seal, have been used to isolate sections for test pressures as high as 5000 psi, Figure 19-4. Line stops, Figure 19-4 bottom, can be inserted using hot taps (Chapter 23). They have been used to isolate pipelines with diameter as high as 36” and pressures as high as 1000 psi. Plugs with inflatable seals have been used in subsea pipelines [Farque].

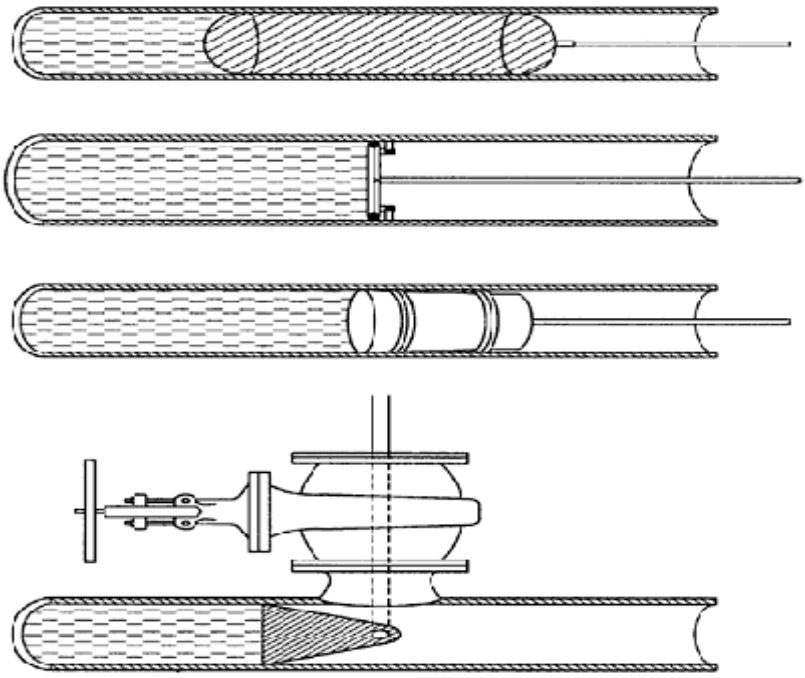

**Figure 19-4** Line Isolation Techniques

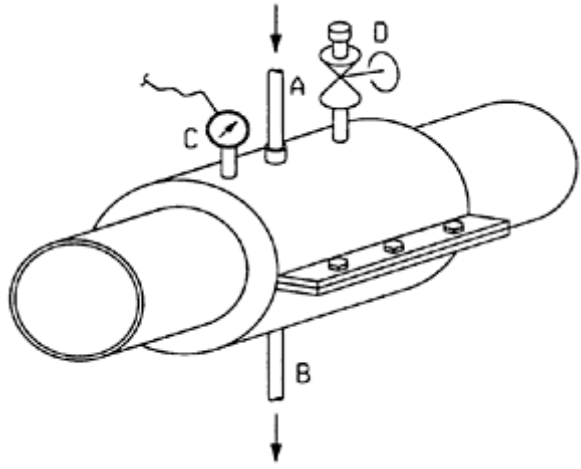


**Figure 19-5** Freeze Plug Arrangement.

(A) Refrigerant Inlet, (B) Outlet, (C) Thermocouple and (D) Vent

An alternative method for isolating liquid filled pipe sections is to form a freeze plug, by circulating liquid nitrogen or Freon in coils or through a jacket placed around the pipe, as illustrated in Figure 19-5, or by packing dry ice (carbon dioxide) in a cradle around the pipe. Precautions are in order to assure that (1) the plug will not fail at test pressure, (2) the frozen pipe will not fracture [EPRI, Bowen], and (3) the trapped liquid will not over-pressurize and burst the pipe. For these reasons, it is advisable to retain the services of a specialized contractors or certified technicians to perform the freeze plug. When the activity is complete, it will be necessary to thaw the freeze plug from the center towards the periphery so that the plug will not detach from the pipe wall while still under differential pressure, and be propelled down the line.

## 19.6 LOCATING LEAKS UNDERGOUND

Ideally, and as is often required by construction codes, the joints of underground pipelines should be kept uncovered during hydrostatic testing. But sometimes a hydrostatic test is conducted with the line fully buried and leak detection becomes more difficult. In this case, the simplest way of detecting liquid leaks, is to inspect the ground surface along the route of the buried pipeline, on a dry day, looking for wet spots. Dye (for liquids) or odorant (for gases) can be added to the water to facilitate the detection of leaks at the ground surface. Tracer gas may be added to the hydrotest water and leaks detected by hand held sniffing probes at the ground surface or, for long pipelines, by a riding cart containing the sniffing probes [Tracer].

As an alternative to direct inspection of joints, the prediction of pump strokes required to reach the test pressure, starting with a water-filled line at atmospheric pressure, can be compared to the number of pump strokes actually needed in the field to determine whether the line is leaking.

Small digs can be made to locally expose the pipe, or manholes used to place listening devices to detect sound that will indicate leaks. To isolate the leak in a buried line, it may be necessary dig and cut the line at mid-point, and hydrostatically test each half of the line. A leaking half would in turn be divided in two halves and the procedure is repeated until the leaking area is isolated. Alternatively, a freeze plug may be used, as described section 19.5, to isolate pipe sections.

Small chronicle leaks in hydrocarbon pipelines can be detected by sampling hydrocarbon through a collection tube run next to the buried pipeline [Bryce]. In the same manner, leaks of toxic fluid can be detected by using double-containment piping (carrier or core pipe within an outer or jacket pipe) and sampling the vapor in the annular space between the inner and outer pipe [Ziu].

## 19.7 REFERENCES

49CFR192, Transportation of Natural and Other Gas by Pipeline: Minimum Federal Safety, Code of federal Regulations, Washington, DC.

49CFR195, Transportation of Hazardous Liquids by Pipelines, Code of federal Regulations, Washington, DC.

API 598, Valve Inspection and Testing, American Petroleum Institute, Washington, D.C.

API 1110, Pressure Testing of Liquid Petroleum Pipelines, American Petroleum Institute, Washington, DC.
ASME B31.4, Pipeline Transportation Systems for Liquid Hydrocarbons and Other Liquids, American Society of Mechanical Engineers, New York, NY.
ASME B31.8, Gas Transmission and Distribution Piping Systems, American Society of Mechanical Engineers, New York, NY.
ASME B 16.34, Valves—Flanged, Threaded, and Welding End, American Society of Mechanical Engineers, New York.
ASME Boiler and Pressure Vessel Code, Section V, Article 10 Leak testing, ASME, New York.
ASTM E 432, Standard Recommended Guide for the Selection of a Leak testing Method, American Society for Testing and Materials, West Conshohocken, PA.
ASTM E 479, Recommended Guide for Preparation of a Leak testing Specification, American Society for Testing and Materials, West Conshohocken, PA.
Bowen, R.J., et. al., Pipe Freezing Operations Offshore—Some Safety Considerations, Offshore Marine and Arctic Engineering Symposium, Volume V Pipeline Technology, ASME, 1996.
Bryce, P.W., et. al., Leak-Detection System to Catch Slow Leaks in Offshore Alaska Line, Oil & Gas Journal, December 9, 2002.
EPRI, Freeze Sealing (Ice Plugging) of Piping, report TR-016384R1, Electric Power Research Institute, November, 1997.
Farque, J.A., Remotely Operated Hydroplug Keeps Vital Pipeline Online, Pipeline & Gas Journal, February, 1999.
FCI, Flow Control Institute, Control Valve Seat Leakage, ANSI/FCI 70–2, Cleveland, Ohio.
FCI, Flow Control Institute, Standard for Solenoid Valve Seat Leakage, ANSI/FCI 91–2, Cleveland, Ohio.
Howard, G.J., The Ice-O-Lator: A Proven Way to Determine Pipeline Leaks, OCG Journal, April 19, 1987.
McAllister, E.W. editor, Pipelines Rule of Thumb Handbook, Gulf Publishing, Third Edition.
Mohitpour, M, Golshan, H., and Murray A., Pipeline Design and Construction A Practical Approach, ASME Press, New York.
MSS-SP-61, Pressure Testing of Steel Valves, Manufacturers Standardization Society, Vienna, Virginia.
Ziu, C.G., Handbook of Double Containment Piping Systems, McGraw Hill, New York.

# 20
# Degradation in Service

## 20.1 A CRITICAL DECISION

Correctly predicting degradation mechanisms, and selecting the proper materials for the service is a critical decision in any project, large or small. The wrong decision will plague operations for years to come.

Understanding corrosion mechanisms, let alone predicting future corrosion, is quite a challenge because corrosion depends on such a large number of variables: the fluid inside the pipe, the atmosphere around the pipe, the temperature, the concentration of constituents in the fluid, the flow velocity, the amount of dissolved oxygen, the phase (liquid, vapor, gas), the pH, the contaminants in the fluid (sometimes a few parts per million of a contaminant make a difference), the process condition (flowing, shutdown, wash, open to atmosphere, etc.), the metallurgy of pipe (fabrication process, grain type and size, heat treatment, inclusions, hard spots, etc.), the mechanical properties (strength, notch toughness), the condition of the weld and heat affected zone, the geometry (crevices, local turbulence, cavitation, etc.), the condition of coating and lining, the soil resistivity and cathodic protection in buried pipe.

To help solve difficult corrosion questions and select the right material for a service, the engineer has several tools. First, the engineer will refer to company standards and pipe specifications that capture years of practical knowledge for a specific service. Second, the engineer may tap the industry's knowledge by contacting colleagues at other facilities or through industry sponsored groups and research institutes such as the Electric Power Research Institute (EPRI), the Gas Research Institute (GRI), the American Gas Association (AGA), the Pressure Vessel Research Council (PVRC), the Materials Properties Council (MPC), the Edison Welding Institute (EWI), the American Petroleum Institute (API), the Nickel Development Institute (NiDI), etc. Third, the engineer may research industry stan-dards, reports and guides published by societies such as the American Society of Mechanical Engineers (ASME), the National Association of Corrosion Engineers (NACE), ASM International, and the American Welding Society (AWS). Fourth, the engineer may contact material and component suppliers for advice on material selection (for example referring to a gasket supplier's catalog in choosing a gasket material compatible with a certain service). Fifth, the engineer may secure the expert services of a consultant, an engineering company or a laboratory that specialize in materials and corrosion. Last but not least, the engineer should continuously learn about materials and degradation through textbooks [ASM Handbook, Dillon, Galka, Landrum, Shackelford, Treseder, Van Droffelaar, etc.], and through publications and training seminars, in order to best assimilate and apply the information obtained from the other sources mentioned.

In this chapter, we will review degradation mechanisms, grouped into eight categories:

(1) General Corrosion.
(2) Local Corrosion.
(3) Galvanic Corrosion.
(4) Erosion Corrosion.
(5) Environmental Effects.
(6) Microbiologically Induced Corrosion.
(7) High Temperature Degradation.
(8) Mechanical Damage.

## 20.2 GENERAL CORROSION

In the broadest sense, corrosion refers to the deterioration of a metal by chemical or electrochemical action ("corodere" in Latin is to "eat away"). General corrosion is an even and uniform deterioration in the form of wall thinning. It is an electro-chemical mechanism, which, in the simplest terms, can be illustrated as follows (Figure 20-1): two different pieces of metal (for example a steel plate and a copper plate) are connected by a metal wire and immersed in a liquid solution (the electrolyte, for example hydrochloric acid HCl). Spontaneously, iron from the steel plate will go into solution (the steel corrodes away), while—in the case of an HCl electrolyte—hydrogen gas is formed at the copper plate.

We will see in a moment why this happens. For now, it is important to note that for corrosion to take place we needed a circuit consisting of the following four parts: (1) an anode (the steel plate), (2) a cathode (the copper plate), (3) an electrical conductor (the metal wire), and (4) an electrolyte (the HCl solution).

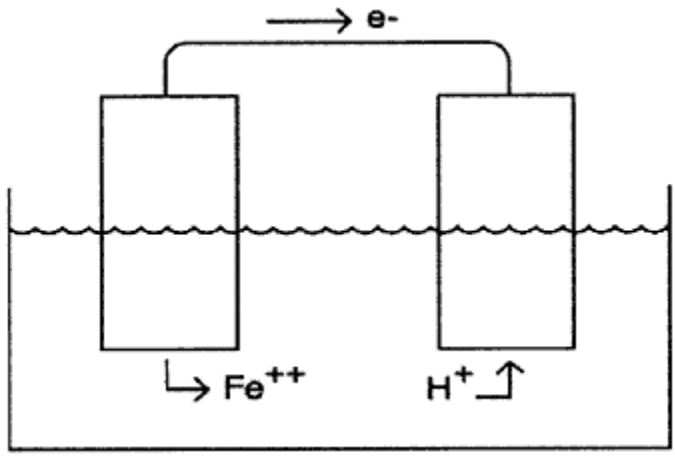


**Figure 20-1** A Corrosion Cell, with Anode (left) and Cathode (right)

Next, we note that the two metal plates could also be two separate zones of the same plate. One area of the steel will be more “anodic” than another that acts as a “cathode”. This difference between anodic and cathodic regions of the same metal is the result of inevitable differences in grain structure, local chemistry, impurities, heat treatment, cold working, and residual stress in the metal. In fact in a metallic plate there would be a multitude of small anodes and cathodes next to each other, Figure 20-2. The conductor between the two plates is the plate itself, and the electrolyte could be sea water, rain water, ground around buried pipes, an acid solution, etc.

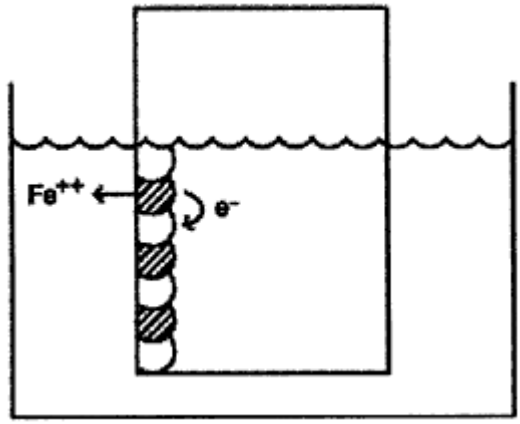


**Figure 20-2** Multitude of Anodes and Cathodes

If any part of the four-part circuit is removed, the corrosion mechanism would be arrested. For example, if the steel is painted, its surface is shielded from the electrolyte, and corrosion is arrested. But if the paint is scratched and uncovers the bare metal, corrosion will proceed at that spot.

### 20.2.1 Progressive Corrosion

In progressive corrosion, the corrosion products are soluble or otherwise continuously removed into the fluid or atmosphere eliminating the protective barrier that shields the pipe against future corrosion. A classic example of progressive corrosion is bare (uncoated, unpainted) carbon steel pipe in aerated water (natural water at .ambient conditions equilibrated with air contains 7 to 8 ppm of oxygen [ASM Handbook]). In this case, a corrosion cell forms between two areas of the bare steel pipe, in contact with the water, Figure 20-2. At the anodic area of steel, iron atoms will lose electrons to become ions $Fe^{++}$ that go in solution (the iron is oxidized), while the electrons e- are released into the metal

$$Fe \rightarrow Fe^{++} + 2e^{-}$$

More generally, the anodic reaction produces positively charged metal ions and free electrons. The metal is oxidized. At the surface of the cathodic area, the electrons released from the anodic area combine with water and oxygen to form hydroxyl ions $OH^-$

$$4e^- + O_2 + 2H_2O \rightarrow 4\ OH^-$$

The hydroxyl ions will, in turn, combine with the oxidized iron $Fe^{++}$ to form ferrous hydroxide

$$2\ Fe^{++} + 4\ OH^- \rightarrow 2\ Fe(OH)_2$$

The ferrous hydroxide precipitates as ferric salt (ferric hydroxide) $Fe(OH)_3$ that is carried in solution. If dried, water will evaporate and the ferric hydroxide $Fe(OH)_3$ will turn to rust $Fe_2O_3$, the natural form of iron ore

$$2\ Fe(OH)_3 \rightarrow Fe_2O_3 + 3\ H_2O$$

What is important is the outcome: iron that was part of the pipe has been removed in the form of rust and the wall thickness has been reduced. Oxidation of iron (loss of electrons and increase in algebraic charge from Fe to $Fe^{++}$) took place at the anode. What happened at the cathode (the formation of hydroxide ion $OH^-$) is specific to the electrolyte environment. In our example, with water, hydroxide ions $OH^-$ are formed. If instead of aerated water we had an acid solution (for example HCl), then the cathodic reaction would consist in the formation of atomic hydrogen

$$H^+ + e^- \rightarrow H$$

The fact that the carbon steel was unprotected (no paint and no coating), leaves a large number of anodic areas exposed, and therefore the oxidation and dissolution of iron occurs over a large area, resulting in general and progressive corrosion and generalized wall thinning.

If hydrogen accumulates around the cathode it will shield the cathode and reduce the corrosion rate. But if there is dissolved oxygen in solution, the oxygen will combine with the hydrogen and prevent this shielding effect [Dillon].

### 20.2.2 Passivating Coating

The second mode of general corrosion is the case where the oxide film is stable and stays attached to the metal surface, naturally forming a passivating coating. It does not erode or dilute away. A classic example of passivating coating is $Cr_2O_3$, the chromium oxide layer that forms at the surface of stainless steel. The chromium oxide is stable and protects the stainless steel from further generalized corrosion. This does not mean that the metal is immune to all forms of corrosion. The passive film can be penetrated by contaminants such as ferric chlorides, PVC or chlorides from marking materials. These penetrations constitute pitting initiation sites. For true corrosion resistance it is therefore essential to keep the stainless steel surface clean from contaminants. This can be achieved by cleaning with of petroleum-based solvents or non-toxic solvents [Hill]. High temperature scale formed during welding has lower corrosion resistance than the base metal and can be removed by chemical means with hydrofluoric and nitric acid, followed by rinsing (pickling [ASTM A 380]). The clean stainless steel surface then reacts with ambient

oxygen to regenerate the passivating chromium oxide layer. This step is called passivation and is enhanced by nitric acid. After fabrication and installation, acid-based cleaners such as phosphoric-based and citric acid-based surfactants can be used to clean stainless steel surfaces in the field.

Other stable oxides include $Al_2O_3$ and $SiO_2$. Iron oxide is also a protective oxide; hot iron (such as steel heated by a gas flame or coming out from a heat treatment furnace) reacts with ambient oxygen to form a black iron oxide layer at the surface. Unlike rust, the iron oxide layer FeO is a passivating coat that protects the iron from further corrosion.

$Fe \rightarrow Fe^{++} + 2e^{-}$

$2e^{-} + O \rightarrow O^{--}$

$Fe^{++} + O^{--} \rightarrow FeO$

The tendency to form stable oxide layers is measured by a factor, the Pilling-Bedworth ratio [Shackelford]. For example, the following metals tend to form stable oxides: Copper, aluminum, silicon, chromium, and lead. Corrosion mechanisms are influenced by differences in fluid, temperature, concentration, PH, and flow rate. It becomes difficult to predict whether progressive corrosion will take place or a passivating coating will form. For example, the corrosion of carbon steel in wet $H_2S$ service results in the formation of a protective iron sulfide film

$Fe + H_2S \rightarrow FeS + 2\,H$

However, above a certain concentration of chlorides or ammonia in the fluid stream, flaky iron-ammonia compounds will form. They dissolve more easily and are not as protective as FeS. Corrosion will then progress deeper into the metal. This illustrates an essential, and vexing, aspect of corrosion engineering: the likelihood of corrosion, its form, and its rate depend on the multitude of factors mentioned in section 20.1. Despite obvious difficulties in predicting corrosion, the designer may find some generic corrosion rates in reference handbooks. For example, corrosion rates in sea water have been reported in the range of 3 to 13 mils/year for carbon steel and 3 to 4 mils/year for 5Cr steels [ASM]. Structural steel corrosion rates are reported in the following ranges: in rural areas 0.15 to 0.30 mils/year, in industrial areas 0.17 to 0.73 mils/year, in marine environment 0.37 to 0.90 mils/year, and in severe marine environment 7 to 9 mils/year [ASM]. Because of the many factors that influence corrosion rates, these are often analyzed and reported in a statistical manner, for example as a yearly corrosion rate vs. its probability of exceedance. A study of corrosion in oil and gas pipelines indicates a corrosion rate of the form [Worthingham]

$P = 10^{-aX}$

P=probability of exceedance of a corrosion rate X, $0<P\leq 1$

a=factor that depends on the corrosion mechanism

X=corrosion rate, mils/year

For example, with a corrosion mechanism for which a=3/5, the probability of exceedance of a corrosion rate of 5 mils/year is $P=10^{-(3/5)5}=10^{-3}$. Interestingly, the same study indicates that the future corrosion rate will be faster in zones that are currently deeply corroded than in zones with little corrosion. With our nomenclature, this means that factor a is smaller for deeper corrosion. In other words, we could have a=2/5 for deep

corrosion zones and a=1 for light corrosion zones; a certain corrosion rate X (for example 5 mils/year) will be more likely in a deeply corroded zone (probability of exceedance of 5 mils/year is $P=10^{-(2/5)5}=10^{-2}$ which is quite likely) than in the zone of shallow corrosion ($P=10^{-1\times5}=10^{-5}$ which is unlikely).

## 20.3 LOCAL CORROSION

Local corrosion is, as its name indicates, corrosion that occurs in small areas. An example of local corrosion is pitting at points where coating has been accidentally removed from a coated steel surface. The local loss of protective coating exposes a zone of metal (in itself a multitude of anodes-cathodes) to an electrolyte (for example rain water or soil); this permits corrosion to take place at that narrow bare metal surface. The loss of iron from the multitude of small anodes will cause the narrow and deep pits characteristic of pitting corrosion [Dillon]. Localized corrosion is referred to as pitting if the metal loss is roughly as deep as it is wide. A pitting resistance equivalent (PRE) is sometimes used to rank the pitting corrosion resistance of alloy steels, molybdenum and nickel alloys, where PRE is defined as

PRE=%Cr+3.3 %Mo+16 %N

Which leads to PRE of stainless steel 304=18, 304L=18, 316=25.2, 317 =30.5; and for duplex steels 2205=33.5, 2505=37.3, 2507 alloy 100=41; and molybdenum alloys=40 to 45; and nickel alloy 2000=76.

Another common example of localized corrosion is crevice corrosion, for example at pipe-support contact points, under solid deposits in storage tanks, under name plates in outdoor vessels, at the interstices of a threaded joint, in the minute gaps of a socket welded fitting, or at the inner diameter of a partially penetrated girth weld. Crevice corrosion often takes place when the fluid in the crevice becomes depleted in oxygen, and the metal in the oxygen depleted zone is anodic relative to the metal in contact with the free flowing liquid. This effect is clearly illustrated by the oxygen concentration cell or differential aeration cell, Figure 20-3: two identical steel electrodes are connected by a conducting wire and immersed in water. Nitrogen is bubbled around one steel electrode and oxygen is bubbled around the second steel electrode. The steel on the nitrogen side becomes anodic and corrodes relative to the steel on the oxygen side. The same happens at metal areas covered by solid deposits, the area covered by the deposit is starved of oxygen and becomes anodic (corrodes) relative to the surrounding uncovered metal that acts as a cathode.

Crevice corrosion is a costly problem that must be avoided through good design details that exclude crevices. For example, when outdoor, a fiberglass edge can be placed between the pipe and a flat support surface to slightly elevate the pipe off the support surface, avoiding the formation of a pipe-support crevice space where rain water will accumulate. If local corrosion is due to base metal or weld defects, such as grooving corrosion in older pipelines, these must be removed. Metals, such as iron or nickel alloys, can be tested for susceptibility to crevice corrosion [ASTM G 79].

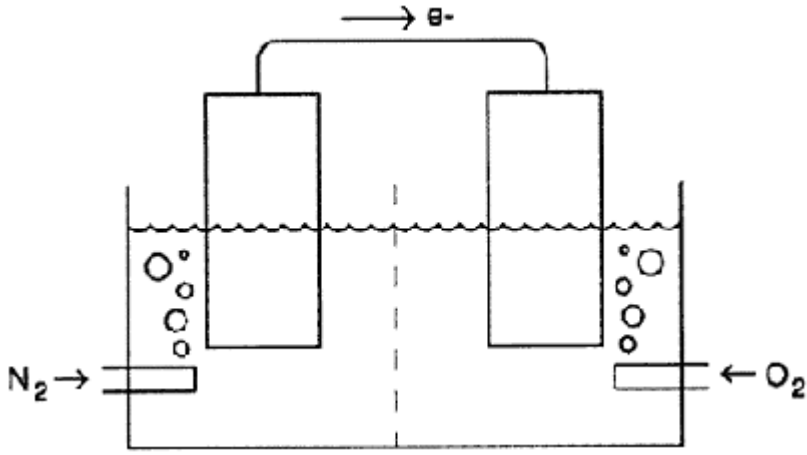


**Figure 20-3** Oxygen Concentration Cell

Local corrosion could be caused by poor base metal or weld quality. This is the case for cold stitched seams on older pipelines (Chapter 3). These defective welds have been prone to selective localized corrosion in the form of deep grooves, referred to as grooving corrosion [Kiefner].

## 20.4 GALVANIC CORROSION

When two metals are electrically coupled through the same electrolyte, the more anodic metal in the galvanic series for that particular electrolyte, Table 20-1, will corrode [ASTM G 92]. A galvanic series such as the one in Table 20-1 only holds true for its specific environment. While magnesium is anodic to steel in sea water, it is cathodic to steel in hydrofluoric acid. Placed in an electrolyte, the more anodic metal (least noble) will corrode in favor of the more cathodic (more noble) metal. This is for example the case if a copper plate (more noble, cathode) is coupled to a carbon steel plate (less noble, anode) through a wire, and immersed in sea water (Figure 20-1). As electrons are released from the carbon steel plate where iron oxidizes $Fe \rightarrow Fe^{++}+2e^{-}$, they flow through the wire towards the copper plate. The current, which by convention flows opposite to the electrons, flows from the carbon steel plate towards the electrolyte. Cathodic protection is the means by which the corrosion of the carbon steel plate is countered in one of two ways:

(1) An impressed direct current is forced to flow from the electrolyte (for example the ground in buried pipelines or the water in subsea pipelines) towards the carbon steel and copper plates. The carbon steel acts as a cathode and does not corrode. This is the principle of impressed current cathodic protection.

(2) The carbon steel plate is connected to an even less noble (more anodic) metal, such as magnesium or zinc. This is the principle of sacrificial anodes. The magnesium or zinc

metal is sacrificed (corrodes away) to protect the relatively more noble carbon steel. Galvanic protection of carbon steel is based on the same principle: zinc, which is more anodic than steel, is applied as coating to steel, and will corrode first, forming a stable zinc oxide protective coat.

**Table 20-1** Galvanic Series in Sea Water.

| Anodic (less noble) |
|---|
| Magnesium |
| Magnesium alloys |
| Zinc |
| Galvanized steel |
| Aluminum alloy 5052 |
| Cadmium |
| Aluminum alloy 2024-T4 |
| Low carbon steel |
| Wrought and cast Iron |
| 410 stainless steel (active) |
| 50–50 lead-tin solder |
| 304 and 316 Stainless (active) |
| Lead |
| Tin |
| Some Copper alloys |
| Nickel 200 (active) |
| Inconel alloy 600 (active) |
| Brass, Bronze |
| Inconel 600 |
| Some Copper alloys |
| Monel alloy 400 |
| Nickel and Inconel (passive) |
| 302, 304, 321, 347, 410 Stainless (passive, intact oxide film) |
| Ni—Cu alloy |
| Inconel 625, Hastelloy C |
| 316, 317 Stainless (passive) |
| Silver |

| |
|---|
| Titanium |
| Gold, Platinum |
| Graphite—Carbon |
| Cathodic (most noble) |

## 20.5 EROSION CORROSION

Erosion is the abrasion of the inner diameter of the pipe caused by the impingement of suspended solids in a flowing liquid, or the impingement of suspended liquid droplets in flowing wet steam or gas. If erosion is accompanied by corrosion, the passivating corrosion layer is eroded and entrained into the flow, exposing new material for further corrosion.

Erosion corrosion (flow accelerated corrosion) has received much attention in the power industry as a result of failures in steam piping [EPRI]. The factors that cause erosion are (a) suspended solid particles in liquid or liquid droplets in steam or gas, (b) high velocities (in the order of 10 ft/sec for liquid service in steel and 3 ft/sec in copper tubing), and (c) local flow turbulence.

Erosion can also be caused by cavitation, which is the formation of vapor bubbles at the center of turbulence vortices in liquid lines, and the subsequent collapse of the vapor bubbles. Therefore, cavitation tends to form where the liquid pressure drops below vapor pressure, such as downstream of an orifice, an elbow, or an improperly sized control valve [ISA].

The best solution to erosion and cavitation is to eliminate its cause. This may require changes to valves, bends, orifice plates, layout or flow rate. For example, a common practice to reduce the likelihood of erosion is to provide 5 to 10 diameters of straight pipe downstream of an orifice, a bend or a control valve. Another solution is (1) to use an erosion resisting alloy such as type 440C stainless steel, or (2) to line the pipe with a hard material that resists abrasion or erosion, for example by painting or spraying the inner surface with an epoxy-ceramic mix or hard facing with cobalt based alloys (such as Stellite™), tungsten carbides, or high chromium alloys. Materials can be tested to measure their susceptibility to erosion [ASTM G 32].

Fretting is wall thinning by erosion due to vibratory friction between two surfaces in contact.

## 20.6 ENVIRONMENTAL EFFECTS

The term environmental effects encompasses a number of corrosion mechanisms that are specific to a certain metal in a certain environment. They are most often cracking mechanisms. An example of environmental effect is preferential corrosion along grain boundaries, referred to as intergranular corrosion, Figure 20-4. A classic example of intergranular corrosion occurs with austenitic stainless steel: during heat-up or cool-down of stainless steel (for example during mill fabrication or welding), as the metal traverses

the range of temperatures between ap-proximately 700°F and 1500°F, carbon at the grain boundaries precipitates as chromium carbide, depleting the grain boundaries of much of their chromium. The steel is said to be sensitized. As this sensitized metal is now placed into service, the chromium-depleted grain boundary becomes a preferential area of corrosion in specific environments like sulfuric or phosphoric acids, acetic and formic acids, and high chloride waters such as sea water. The intergranular corrosion proceeds in the form of a multitude of cracks along the grain boundaries. If left undetected, these intergranular cracks will propagate through the wall and result in through-wall leakage or rupture of the pipe or component, with no loss of wall thickness and little advance warning. Austenitic stainless steels can be tested for susceptibility to intergranular corrosion [ASTM A 262, ASTM A 708]. Specimens of stainless steel are immersed in various boiling acid solutions, and then examined to detect any sign of intergranular attack. To prevent sensitization of stainless steel, the metal can be solution annealed and then rapidly cooled (quenched) through the sensitization temperature range (1500°F down to 700°F) to avoid the precipitation of chromium carbides. Sensitization of stainless steel can also be reduced by the addition of titanium or columbium which will cause titanium or columbium carbides to precipitate instead of chromium carbides. This is the case for austenitic stainless steel types 321 (titanium stabilized) or 347 (columbium stabilized). The effects of sensitization can also be reduced by reducing the carbon content, for example down to 0.03% or less. Duplex stainless steel (austenitic-ferritic steel) is particularly resistant to intergranular corrosion. As an alternative, if the operating service temperature permits, a PTFE (Teflon$^{R}$) or glass liner may be used to shield the metal from the electrolyte.

**Figure 20-4** Illustration of Intergranular Corrosion

Stress corrosion cracking is environmental cracking that takes place in the presence of tensile stresses in the material. It may be intergranular (IGSCC) or transgranular (TGSCC).

Intergranular stress corrosion cracking (IGSCC) is a form of environmentally assisted cracking (EAC) in which degradation takes place along the grain boundaries (Figure 20-4) under the combined effect of tensile stress (or cyclic tensile stresses) and a corrosive environment. The tensile stress could be the result of hoop stresses due to internal pressure, or residual tensile stresses from welding. Examples of IGSCC include steel in the bleach stock environment of pulp and paper plants, steel in nitric acid or sulfuric acid service, steel in aqueous alkaline solutions (pH 8 to 11) containing $H_2S$ and $CO_2$ particularly in zones of high hardness, steel gas pipelines in high PH (9 to 13) and warm environment (close to 100°F downstream of compression stations). Some nickel alloys are susceptible to intergranular corrosion by HCl, $H_2SO_4$, alkaline solutions (caustic NaOH and KOH). Aluminum alloys are also susceptible to intergranular corrosion in $H_2S$, $SO_2$, and marine environments.

Transgranular cracks are cracks that do not follow the grain boundaries, but instead tend to follow certain crystallographic planes, cutting through the grains, Figure 20-5. A classic example of trans-granular cracking is corrosion-fatigue cracking, in which the crack progresses in the direction perpendicular to the applied cyclic fatigue load, regardless of the grain orientation. The progression of fatigue cracks is further accelerated in the presence of a corrosive atmosphere because (a) the fatigue crack acts as a narrow crevice, and (b) the cyclic stress continuously drives the crack, exposing fresh metal (non-passivated by an oxide layer) to the corrosive environment.

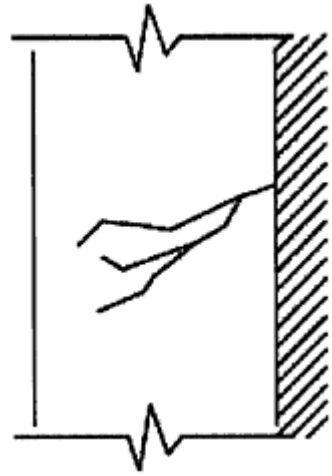

**Figure 20-5** Illustration of Transgranular Corrosion

Transgranular cracks can also develop as a result of corrosion, such as caustic attack of type 316 stainless steel tubes operating at 500°F. In one case, a colony of branched trans-granular cracks were caused by sodium hydroxide (NaOH) attack of steel, which had to be replaced by a nickel alloy.

Transgranular cracking can also be accelerated in the presence of tensile stresses, leading to trans-granular stress corrosion cracking (TGSCC) [Jones].

Trans-granular stress corrosion cracking in stainless steel can also be due to external chlorides, from insulation, rain water or process fluid deposited on the pipe surface, particularly on warm pipes (temperature between 150°F and 212°F).

Another example of trans-granular cracking is that which develops in carbon steel in the presence of wet $H_2S$ [NACE TM 0177, NACE TM 0284], Figure 20-6. This cracking occurs in two forms: (a) sulfide stress cracking (SSC) in the presence of tensile stresses, (b) hydrogen induced cracking (HIC) in which atomic hydrogen diffuses into the steel, forming blisters and cracks along the direction of steel plates. When HIC cracks are closely clustered they can become interconnected and look like steps (step oriented hydrogen induced cracking or SOHIC).

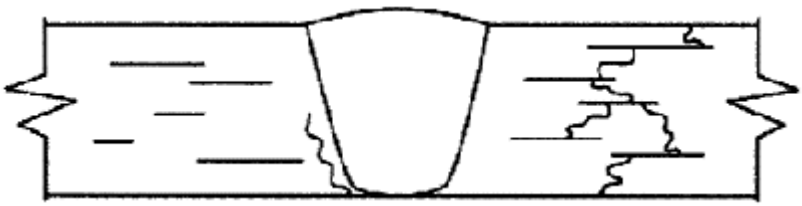

**Figure 20-6** HIC (left), SSC (at left of weld root), SOHIC (right)

Wet $H_2S$ cracking has been linked to zones of high hardness or high residual stresses for example in carbon steel or low alloy steel welds which are not postweld heat treated; damage increases with hydrogen concentration and in the presence of cyanides [API 945]. Special resistant steels have been developed for wet $H_2S$ service. They are characterized by low oxygen and sulfur contents (below 0.002%). While sulfide stress cracking is mostly trans-granular, it has also been detected in intergranular form.

## 20.7 MICROBIOLOGICALLY INFLUENCED CORROSION

Microbiologically influenced corrosion (MIC) is corrosion caused or accelerated by micro-organisms (bacteria, fungus, algae, plankton). There is a multitude of bacteria species, some are anaerobic (do not require oxygen) others are aerobic (require oxygen). They accelerate corrosion in many ways; for example (a) creating anodic regions, (b) causing cathodic depolarization, and (c) leading to under-deposit acid attack. MIC appears as a slimy deposit, which—when washed away—unveils deep pits caused by the micro-organisms, Figure 20-7.

**Figure 20-7** Illustration of Pits Forming Beneath Bacterial Deposit

In the case of anodic reactions, the environment under the colony of microorganisms becomes anodic compared to the rest of the system, causing corrosion to proceed as deep pits through the pipe wall. For example aerobic bacteria causes the depletion of oxygen and formation of sulfuric acid

$$2\ S+3\ O_2+2\ H_2O \rightarrow 2\ H_2S0_4$$

Sulfide producing bacteria causes the formation of iron sulfides

$$4\ Fe^{+}+SO_4^{-}+4\ H_2O \rightarrow 3\ Fe(OH)_2+FeS+2\ OH^{-}$$

In the case of cathodic depolarization, sulfide-reducing bacteria consume hydrogen from cathodic regions, thereby accelerating the cathodic reaction and corrosion by eliminating hydrogen as it is formed.

In the case of under-deposit acid attack, the micro-organisms generate and concentrate acetic acid that corrodes the pipe.

Most bacteria develop in stagnant or low flow systems (below approximately 3 ft/sec), in a neutral environment (pH between 4 and 8), and warm temperatures (70°F to 120°F). These conditions are prevalent in crevices, under disbonded coatings, in low flow cooling water lines, or in stagnant water-filled fire protection sprinkler pipes.

MIC can be detected by visual internal inspection, the colonies appearing as slime or small tubercules. There are quantitative methods to sample, characterize and count microbial colonies [Huchler]. Once the microbes are removed and the surface is cleaned, the damage becomes evident, in the form of heavy pitting and wall thinning [Jones, ASM].

To prevent MIC one must avoid using untreated water under stagnant, warm conditions. Removing MIC once it has taken place requires thorough cleaning to remove bacteria colonies attached to the pipe wall, followed by treatment with biocides. Larger organisms, such as muscles and mollusks, also tend to accumulate in relatively warm waters at low flow rates, such as the intake structures of cooling water systems. [AU: please make sure your original meaning is intact in following sentence.] The flow area is reduced because it is fouled by these larger organisms, which are the starting sites for MIC. Periodic cleaning and mechanical removal, as well as chemical treatment (with bleach or chlorine), become necessary.

Biocides are commonly used to fight MIC. The biocide is selected based on tests of micro-organism cultures, chemical and PH tests of fluid, analysis of the metal surface, and soil analysis in the case of external MIC in buried pipe. Material selection is also an important consideration in the presence of MIC; for example in stainless steel systems,

the use of austenitic stainless steel with 6% molybdenum (6 Mo steel) will generally prove more resistant to MIC.

## 20.8 HIGH TEMPERATURE EFFECTS

Above 30% to 50% of a steel's melting temperature (and 50% for nickel alloys) creep takes place. Creep is a complex phenomenon that depends on many variables, such as the metal chemistry, heat treatment, and hardness. Creep manifests itself in two ways: through metallurgical changes and through mechanical changes. The mechanical changes have been addressed in Chapter 7. The metallurgical changes can cause bulging of the pipe wall or the formation of creep voids. Microscopic creep voids tend to form in base metals, welds and heat affected zones. Over time, the creep voids will coalesce into macro-cracks. There is typically a relatively long incubation period, during which micro-cracks form (in the order of 4 mils), followed by crack growth. During the long incubation period, the micro-cracks are difficult to detect by conventional ultrasonic testing.

Creep typically takes place during long operating periods at or above 750°F for carbon steel, 900°F for Cr-Mo steels, and 1050°F for series 300 stainless steels. The creep temperature may be 30°F lower for weldments. Creep has caused catastrophic failures of longitudinal seams of steel pipes in high-pressure steam systems in power plants. These failures have lead to extensive ultrasonic angle beam inspections of longitudinal seam welds, followed by repair or replacement, in some cases replacing the seam welded pipe by a seamless pipe. Additional means for detecting creep include gages used to detect any bulging of the pipe diameter, hardness readings or infrared thermography. These detection techniques are further complicated if there are scale deposits on the pipe inner diameter causing hot spots. Creep may also take place in conjunction with corrosion.

Another high temperature effect is graphitization. Between 900°F and 1100°F the iron carbides contained in some carbon steels and low alloy C-Mo steels will decompose, releasing carbon in the form of free graphite nodules that are random or in preferential directions. This graphitization significantly reduces the steel's mechanical properties. Spheroidization is a similar mechanism to graphitization, that tends to occur in some C-Mo and Cr-Mo steels, in which carbides combine into small spheres during prolonged service between 900°F and 1400°F.

Temper embrittlement is a loss of toughness in low alloy steels that can occur during long term exposure at temperatures in the range of 600°F to 1100°F, particularly in the presence of elements such as phosphorous, antimony or arsenic. A similar loss of toughness occurs in ferritic and duplex stainless steels between 600°F and 1100°F, and is referred to as 885°F embrittlement. Yet another similar mechanism, referred to as sigma phase embrittlement, can occur in austenitic stainless steels between 1000°F and 1700°F.

High temperature corrosion is an accelerated form of corrosion, such as chlorine and hydrogen chloride corrosion, further aggravated by the low melting point of certain corrosion products.

Because carbon steel is susceptible to high temperature effects, alloy steels are used in such applications, for example 1.25Cr–0.5Mo steels and, above 1500°F, 5Cr or 9Cr steels. Stainless steels are also used to 1500°F but their strength tends to be low. Instead,

high alloys (25Cr–20Ni) with high carbon (0.4% C) are used for temperatures in excess of 1500°F. For high-temperature hydrogen service, the Nelson curves [API 949] provide guidance for the selection of 1Cr–½ Mo, up to 6Cr–½ Mo, and 18–8 stainless steel, depending on factors such as operating temperature and partial pressure of hydrogen. Nickel alloys, in addition to having good corrosion resistance in many environments, maintain good mechanical properties at high temperature; Inconel™ alloy 601 is used in furnace tubes, alloy 617 is used in gas turbines, Haynes™ alloy 230 is used in high temperature petrochemical and power applications. Titanium alloys (ASTM B 337 pipe and ASTM B 363 fittings) also have good high temperature corrosion resistance and strength, and are much lighter than steel.

## 20.9 MECHANICAL DAMAGE

A pipe can be mechanically damaged during transport, construction, while in-service, or during maintenance. Mechanical damage can take the form of accidental bends, buckles (surface ripples), dents (deformation of the cross section), gouges (sharp, knife-like groove), or fatigue cracks.

Dents and gouges are particularly dangerous in high-pressure service, such as oil and gas transmission pipelines, where the hoop stress is in the order of 50% or more of the yield stress. Dents and gouges can be formed during construction or through encroachment by excavation equipment (third party damage) or, offshore, by a ship's keel or anchor. Encroachment is the single largest source of pipeline failures, with 80% of the failures occurring at the instant of damage (when the pipeline is hit) and 20% of the failure occurring later [Rosenfeld]. Encroachment damage can take several forms: first, the impact can be so severe that the pipeline is punctured. Second, the sharp gouge formed by impact acts as a stress riser. Third, the impact spot is a locally cold worked zone with, in some cases, a transfer or welding of the tool's metal onto the pipe surface. Fourth, the pipe wall at the dent tends to bend and breathe radially in and out as the pressure in the pipeline fluctuates, eventually causing it to rupture by fatigue [API 1156]. Fifth, the impact causes the coating to be scraped off the pipe, resulting in accelerated localized corrosion or cracking. Dents and gouges are also caused during pipe lying if the pipe bears against a rock in the ditch, with the weight of the pipeline and soil above it. A rock dent is usually at the bottom of the pipe and causes a more rounded deformation of the pipe cross section than an encroachment dent. If the rock remains wedged under the dent, it will constrain the pipe wall against breathing, which is generally preferable to an unconstrained dent, particularly in liquid pipelines where pressure fluctuations are significant. High-resolution inline-inspection tools are used to help differentiate between an encroachment dent and a rock dent. In process piping, where the hoop stress (PD/2t) is not as large as in pipelines, dents and gouges are not as critical, and can be evaluated for continued service [API 579] or ground smooth if the remaining wall thickness after grinding is larger than required by the construction code, as described in Chapter 23.

Another form of mechanical damage is the accidental bending of a pipe. In this case, the pipe may be left as-is or repaired as described in Chapter 23.

Mechanical fatigue, like temperature-induced fatigue (Chapter 7), is the initiation and propagation of a crack under cyclic stresses. The damage will be accelerated in the

presence of corrosion (corrosion fatigue). When the material contains a pre-existing fabrication crack, then the crack initiation period is non-existent, and fatigue will tend to propagate the crack through the wall. If the material does not contain a pre-existing crack, the cyclic stress—if sufficiently large (above the endurance limit)—will initiate a transgranular crack along the planes of maximum shear stress, and the crack will evolve eventually to propagate in a direction perpendicular to the maximum tensile stress. Two things can happen at this point: if the crack size and material toughness are such that the applied stress intensity exceeds the fracture toughness of the material, the crack will cause the component to suddenly fracture. If the material is sufficiently tough, the crack will progress through the wall and eventually leak when the remaining wall ligament breaks. The latter case is what is referred to as leak-before-break.

## 20.10 LINING AND COATING

### 20.10.1 Properties

Lining the internal diameter of a pipe or coating its outer diameter is a common technique for corrosion protection. The objective is to shield the metal from the electrolyte (air, water, soil, process fluid).

When selecting a liner or coating, a set of performance objectives should be established, and verified through testing, as listed in Table 20-2.

Some cautions regarding standard properties are in order. By necessity, many of the standard properties are established from tests of relatively short duration, and then lifetime performance, 30 years or more, are determined by extrapolation.

Another question is whether field coating (such as used to coat field welds and repairs, often applied in the presence of wind, dust, humidity, insects, etc.) is as good as the mill applied coating. Coating or liner failures have often been attributed to insufficient or improper surface cleaning.

**Table 20-2** Standard Properties of Liners and Coatings.

| | |
|---|---|
| Compressive Strength | ASTM C 109, ASTM D 695 |
| Hardness | ASTM D 2240 |
| Tensile Modulus and Strength | ASTM D 638 |
| Water Absorption | ASTM C 642, ASTM B 117, D 879 |
| Thin Film Water Absorption | ASTM D 570 |
| Dielectric Strength | ASTM D 229 |
| Permeation | ASTM D 1653 |
| Resistance to Acids and Alkalites | ASTM C 581 |
| Solvent Resistance | ASTM D 2792 |
| Temperature Resistance | ASTM D 2488 |

| | |
|---|---|
| Heat Emisivity | ASTM E 307 |
| Adhesion | ASTM D 4541 |
| Cathodic Disbondment | ASTM G 8, ASTM G 42 |
| Impact Resistance | ASTM G 14 |
| Flexibility | NACE RP 0394, ASTM D 1622 |
| Abrasion Resistance | ASTM C 131, ASTM C 1900 |
| Wear Abrasion | ASTM D 4060 |
| Drying Time | ASTM D 1640 |
| Weather Exposure | ASTM D 1014, ASTM D 822 |

### 20.10.2 Liquid Organics

Until the early 1800's, substances made from human or animal organs were considered to have a special origin or vital force, and were labeled organic. Today, organic chemistry is the study of compounds of carbon, whether natural or synthetic. Liquid organic coatings are compounds of carbon, mixed with other elements, and applied in brushed, rolled or sprayed form, in single or multiple layers.

The most common liquid organic material is paint, lacquer and enamels, consisting of alkyd resins made from oils, mixed with pigments to provide color and protection against sunlight, and a volatile thinner and applicant. They are good for weather resistance and fair for acid and water resistance.

Epoxies are polymers (synthetic macromolecules consisting of a repeated structural unit or "mer", made from carbon, hydrogen, oxygen or nitrogen). They are thermosetting, which means that they set and become more rigid when heated. They are mixed with pigments and a curing agent that causes the epoxy to become tighter and more chemically resistant. Fusion bonded epoxy was first introduced in the 1950's. To apply fusion-bonded epoxy, the pipe is blast cleaned and heated. Epoxy is applied to the pipe OD or ID (using remote controlled spray devices if necessary), and heated again to thermoset in layers several mils thick. Epoxies can be mixed with ceramics to improve their erosion resistance.

Coal-tar epoxies and coal tar enamels are epoxies and enamels with coal tar filler that improves their moisture and microbial resistance. Coal tar is a byproduct of the carbonization of coal. It is applied hot to a blast cleaned pipe surface. Coal tar enamels are applied at temperatures close to 400°F, under controlled ventilation because of the release of volatile organic compounds or VOC down to 200°F. Coal tar enamels have been used since the 1920's and have covered thousands of miles of pipelines. They are commonly used underground and if used above ground they must be wrapped to be protected from solar radiation.

Asphaltic bitumen is a dark, viscous aggregate of hydrocarbon, mixed with sand and fillers, which adheres well to the pipe and is waterproof.

### 20.10.3 Multilayer Coating

In their simplest form, multilayer coatings consist of an adhesive, a corrosion resistant inner layer and a tough, wear resistant outer layer. There is a multitude of choices in multiplayer coatings:

Epoxy-urethane coatings consist of a urethane polymer (toughness and abrasion resistance) bonded to an epoxy resin (adhesion and moisture barrier).

Epoxy-adhesive-polyethylene is a three-layer coating applied to a cleaned pipe surface, induction heated to about 400°F, the epoxy primer is sprayed first, followed by the wrapped adhesive layer, and covered by a polyethylene outer jacket.

Epoxy-coal tar enamel-fiberglass-wrap is a four-layer coating, with the epoxy applied as the first layer, followed by a thick coat of coal tar enamel, in the order of 100 mils, a fiberglass wrap, and finally Kraft paper to protect the coating from ultraviolet exposure [Johnson].

Epoxy-polymer-concrete is a three-layer coating, much lighter than a concrete coat, used to protect fusion-bonded epoxy against abrasion and impact.

### 20.10.4 Metallic Coatings

The most common metallic coating is galvanized zinc coating applied to the surface of carbon steel pipe, to protect the pipe against atmospheric corrosion. The zinc is applied by immersion (hot dipped) or by electroplating, and may be alloyed with aluminum for further corrosion resistance. Cladding is the deposit, often by weld overlay, of a metal, typically an alloy, atop a base metal. For example, stainless steel is deposited on the inner surface of a carbon steel vessel to provide corrosion resistance while minimizing the vessel's cost compared to an all-stainless vessel. Another common example of cladding is the deposit of a high chromium alloy on valve internals to improve their wear resistance.

## 20.11 CORROSION INHIBITORS

There are two types of corrosion inhibitors: (a) inhibitors that alter the chemistry of the process stream, and (b) inhibitors that form passive films or enhance naturally occurring passivating films on the pipe surface. Chemical deaeration (removal of dissolved oxygen) is an example of inhibition by modifying the chemistry of the process stream. In hot water and steam service it is often necessary to eliminate dissolved oxygen from water to maintain the stability of the passivating magnetite film $Fe_3O_4$ thereby reducing the rate of corrosion. This can be achieved by the addition of sodium sulfite or hydrazine (above 600°F) to deaerate the water [Jones]:

$Na_2SO_3 + ½\ O_2 \rightarrow Na_2SO_4$

$N_2H_4 + O_2 \rightarrow N_2 + 2\ H_2O$

Inhibitors that act by forming or improving passivating films include chromates (further enhanced by the addition of zinc), calcium and silicates in carbon steel or cast iron

recirculating water systems; phosphates and borates in carbon steel system at near neutral PH.

## 20.12 MATERIAL SELECTION

It is clear from the many considerations discussed so far in this chapter that corrosion depends on so many factors that it can be difficult to predict. Because of this complexity, material selection must based on a company's experience, consultation with corrosion engineers, reference to national standards and guides (NACE, API, etc.), and specific material-environment testing. Chapter 1 provides a list of NACE references, and many excellent guides and handbooks have been published on the subject of corrosion control and material selection, some are listed in the reference section of this chapter.

## 20.13 REFERENCES

API 579, Fitness for Service, American Petroleum Institute, Washington, D.C.
API 945, Avoiding Environmental Cracking in Amine Units, American Petroleum Institute, Washington, D.C.
API 949, American Petroleum Institute, Washington, D.C.
API 1156, Effects of Smooth Dents on Liquid Petroleum Pipelines, American Petroleum Institute, Washington, D.C.
ASM Handbook, Volume 13, Corrosion, American Society of Metals, Materials Park, OH.
ASM, Handbook of Corrosion Data, American Society of Metals, Materials Park, OH.
ASTM A 262, Practices for Detecting Susceptibility to Intergranular Attack in Austenitic Stainless Steel, ASTM International, West Conshohocken, PA.
ASTM A 380, Standard Practice for Cleaning, Descaling, and Passivation of Stainless Steel Parts, Equipment, and Systems, ASTM International, West Conshohocken, PA.
ASTM A 708, Recommended Practice for Detection of Susceptibility to Intergranular Corrosion of Severely Sensitized Austenitic Stainless Steel, ASTM International, West Conshohocken, PA.
ASTM G 32, Standard Method for Vibratory Cavitation Erosion Test, ASTM International, West Conshohocken, PA.
ASTM G 79, Guide for Crevice Corrosion Testing of Iron and Nickel Based Stainless Alloys on Seawater, and other Chloride-Containing Aqueous Environments, ASTM International, West Conshohocken, PA.
ASTM G 92, Guide for the Development and Use of Galvanic Series for Predicting Galvanic Corrosion Performance, ASTM International, West Conshohocken, PA.
Dillon, C.P., editor, Forms of Corrosion Recognition and Prevention, NACE, Houston, TX.
Dillon, C.P., Corrosion Control in the Chemical Process Industry, McGraw Hill.
EPRI, Report TR-108859, Flow Accelerated Corrosion, Electric Power Research Institute, 1997.
Fuchs, H.O., Stephens, R.I., Metal Fatigue in Engineering, John Wiley & Sons.
Galka, R.J., editor, Internal and External Protection of Pipes, Proceedings of the 8th Conference, Gulf Publishing.
Hill, J.W., Chemical Treatment Enhances Stainless Steel Fabrication Quality, Welding Journal, May, 2002.
Huchler, L.A., Monitor Microbiological Populations in Cooling Water, Hydrocarbon Processing, September, 2002.

ISA-RP-75.23 Considerations for Evaluating Control Valve Cavitation, Instrument Society of America, ISA, Research Triangle Park, North Carolina.
Johnson, J.R., et. al., A New Higher Temperature Coal Tar Enamel Pipeline Coating Systems, NACE International Annual Conference and Exposition, 1996.
Jones, D.A., Principles and Prevention of Corrosion, Prentice-Hall.
Jones, R.H., Stress-Corrosion Cracking, ASM International, 1992, American Society of Metals, Materials Park, OH.
Kiefner, J.F., Procedure Analyzes Low-Frequency ERW, Flash-Welded Pipe for HCA Integrity Assessments, Oil & Gas Journal, August 5, 2002.
Landrum, R.J., Fundamentals of Design for Corrosion Control, A Corrosion Aid for the Designer, NACE Publication, Houston, TX.
NACE, Corrosion Data Survey, NACE, Houston, TX.
NACE Corrosion Engineer's Reference Handbook, Edited by R.S.Treseder, NACE, Houston, TX.
NACE Handbook, Forms of Corrosion, Recognition and Prevention, National Association of Corrosion Engineers, Houston, TX.
NACE TM 0177, Standard Test Method for Laboratory Testing of Metals for Resistance to Specific Forms of Environmental Cracking in $H_2S$ Environments, National Association of Corrosion Engineers, Houston, TX.
NACE TM 0284, Standard Test Method for the Evaluation of Pipeline and Pressure Vessel Steels for Resistance to Hydrogen Induced Cracking, National Association of Corrosion Engineers, Houston, TX.
Rosenfeld, M.J., Here are Factors that Govern Evaluation of Mechanical Damage to Pipelines, Oil & Gas Journal, September 9, 2002.
Shackelford, J.F., Introduction to Materials Science for Engineers, Prentice Hall.
Suresh, S., Fatigue of Materials, Cambridge University Press.
Treseder, R.S., Tuttle, R.N., Corrosion Control in Oil and Gas Production, corUPdate, Houston, TX.
Uhlig. H.H., Corrosion and Corrosion Control, Wiley.
Van Droffelaar, et. al., Corrosion and its Control, An Introduction to the Subject, NACE, Houston, TX.
Von Baeckman, W., et. al., Cathodic Corrosion Protection, Theory and Practice of Electrochemical Protection Process, Gulf Publishing, Houston, TX.
Worthingham, R.G., et. al., ILI Data Predicts Corrosion Severity, Pipe Line and Gas Technology, November/December 2002.

# 21
# Fitness-for-Service

## 21.1 FITNESS-FOR-SERVICE

Fitness-for-service may be defined as the quantitative analysis of the adequacy of a component to perform its function in the presence of a defect. Fitness-for-service is a technique used to make safe and cost-effective run-or-repair decisions. “It may be contrasted with the best quality that can be achieved within a given set of circumstances, which may be inadequate for some exacting requirements, and needlessly uneconomic for others which are less demanding” [Wells].

For piping, pipe fittings and components, the defect may be (1) a pipe mill defect (base metal or seam weld defect such as a lamination or a crack), (2) a transportation defect (rail, truck or handling induced fatigue or impact damage), (3) a construction defect (shop or field handling, joining, welding or erection flaw), (4) degradation during operation (Chapter 20). The latter includes (a) corrosion induced defects such as wall thinning, environmental cracking, or blistering (b) fatigue due to cyclic loads, sometimes aggravated by corrosion, and (c) accidental defects such as a dent, gouge or distortion.

The first three types of defects (pipe mill, transportation and construction defects) should be detected during construction, pre-operational testing and commissioning, before the system is turned-over to the operating department. These defects will then be evaluated using construction code acceptance criteria that are based on workmanship standards, typically more restrictive than those based on fitness-for-service analysis. But if these defects are missed during the construction, commissioning and turnover phase, and later discovered during service, the owner—with concurrence from the jurisdictional authority, where applicable—may elect to characterize and evaluate each defect using fitness-for-service analysis.

For the purpose of evaluating the fitness-for-service of a pipe component, degradation can be viewed as illustrated in Figure 21-1. Significant degradation in service may be detected by periodic inspection, or may proceed undetected. The continuing degradation may lead to a leak or a rupture (fracture with separation of the pipe). Fitness-for-service is the understanding and prediction of this progression, with the purpose of making a safe and cost-effective run-or-repair decision. Fitness-for-service analysis also makes it possible to predict the failure mode: leak or break.

In addition to pitting, wall thinning and cracks, there are mechanical defects uniquely important to piping systems operating at large hoop stress, in the order of 50% or more of yield, such as oil and gas transmission pipelines. These mechanical defects are: dents (deformation of the circular cross section), gouges (sharp groove), ovality (flattening of circular cross section into a slightly elliptical shape), buckles or wrinkles (one or more kink or wave on the compressive side of a bend), and ripples (mild buckles or wrinkles).

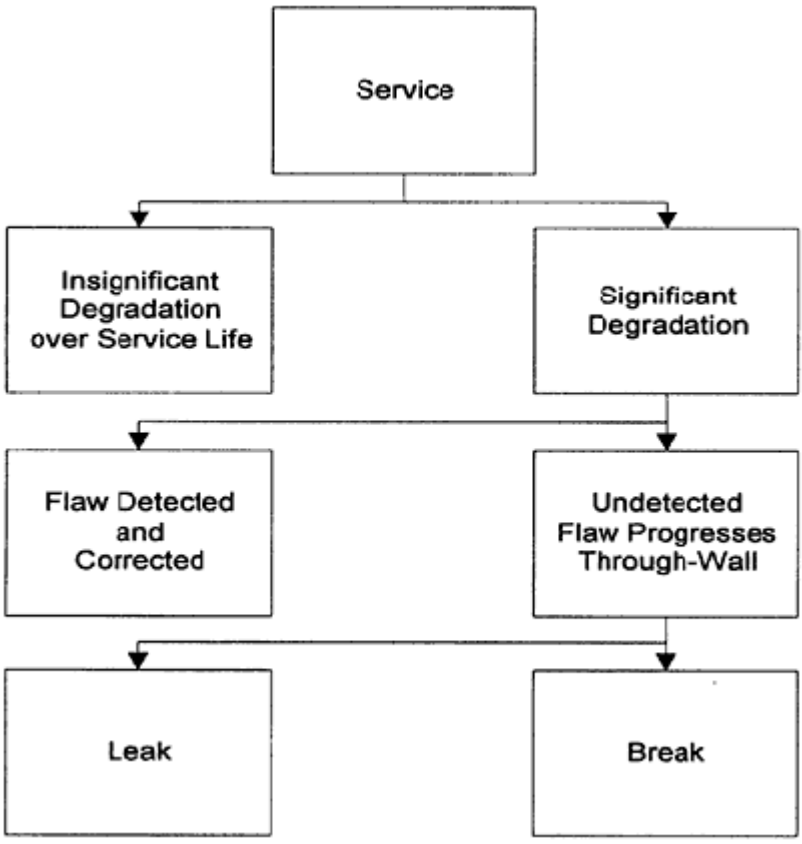


**Figure 21-1** Progression of Degradation Mechanisms

## 21.2 WALL THINNING

### 21.2.1 Measurement

If erosion or corrosion have caused wall thinning, it becomes necessary to evaluate the integrity of the remaining wall and decide whether the pipe is still fit for continued service. If the measured remaining wall thickness, minus the projected future thinning, exceeds the minimum wall required by the applicable design code, then the piping system would be fit for continued service. But if the measured wall thickness, minus the projected future thinning, is less than the minimum required by code, a fitness-for-service analysis is needed to decide whether to continue operation as-is, continue operation at a reduced pressure, plan for a later shutdown, or immediately shutdown and repair. To perform such an analysis, the following parameters must first be determined: the nominal wall thickness of the pipe $t_{nom}$, the minimum thickness required by code $t_{min}$, the

measured minimum wall thickness $t_{mm}$, the longitudinal extent of wall thinning $L_M$, shown in Figure 21-2, and the circumferential extent of wall thinning.

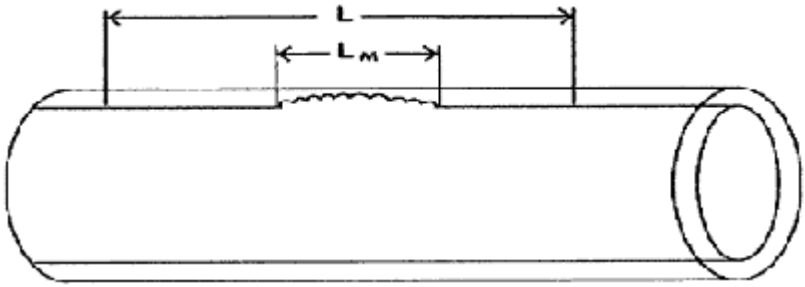


**Figure 21-2** Measured Length of Wall Thinning

The measurement of $t_{mm}$ poses the first difficulty in this evaluation process. In case of internal corrosion, it can be difficult to pinpoint the area of most corrosion, where thickness should be measured. Ideally, the whole line should be inspected, this is for example the case with in-line inspection of pipelines using smart pigs, as described in Chapter 16. In many cases it is not feasible to inspect a whole piping system, and inspection must be limited to a few specific areas [API 570, API 574]. In this case, the selection of areas to be inspected may be based on a visual examination of the pipe using a borescope to inspect the inside of the pipe and visual inspection of the outside of the pipe. The selection of inspection locations should be based on the understanding of the degradation mechanism, experience with similar service, a review of maintenance records and prior history of leaks or repairs.

When in doubt, the inspected area should be made as wide as practically feasible. The longitudinal and circumferential extent of corrosion (the corrosion profile or topography) must also be measured, up to a region where the wall is back to its nominal thickness. The future corrosion allowance FCA must be estimated; this is the amount of corrosion predicted to occur between this inspection and the next one. Wall thinning is typically characterized (extent and depth) by straight beam ultrasonic inspection and (for pipelines) my magnetic flux leakage. The outcome of this inspection would be an inspection report in the form of a profile of the remaining wall (B-scan) or a topographical projection (plan view) with contours of equal wall thickness (C-scan). External pitting can be measured with a pit gauge, a depth micrometer or a laser range sensor [Kania].

### 21.2.2 Ductile Fracture Initiation

The fitness-for-service evaluation of wall thinning in piping systems and pipelines applies to either ductile or brittle fracture. With ductile fracture initiation, the pipe bulges outward at the thinned wall area, undergoing substantial plastic deformation, before eventually failing by longitudinal split, Figure 21-1 case (b). Brittle fracture takes place with little deformation and little, if any, plasticity.

### 21.2.3 Longitudinal Thinning

There are several methods to evaluate the fitness-for-service of local thin areas in ductile materials [API 579; ASME B31G, ASME CC N-513]. The ASME B31G procedure has been commonly used for pipelines since the 1970's. It is based on the hoop stress at ductile fracture in the remaining ligament under the corroded area, Figure 21-3 [Folias, Maxey].

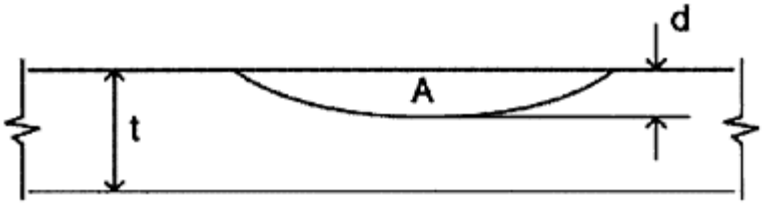


**Figure 21-3** Parameters for B31G Evaluation

If L is the length of corrosion and if $L^2/(Dt) \leq 20$, the hoop stress at ductile fracture is

$$S_P = S_{flow} \frac{1 - A/A_o}{1 - \frac{A/A_o}{M}}$$

$$M = \sqrt{1 + \frac{0.8L^2}{Dt}}$$

$S_P$=hoop stress at ductile fracture, psi

$S_{flow}$=flow stress of the material, psi

A=thinned cross sectional area, $in^2$

$A_o$=nominal cross sectional area, $in^2$

M=Folias bulging factor

If $L^2/(Dt) > 20$ (or A>4 in the nomenclature of ASME B31.G, with $A = 0.893L/(D\,t)^2$), the hoop stress at ductile fracture is the flow stress in the pipe assumed to have uniformly lost a thickness d, all around the circumference

$S_P = S_{flow}\,(1 - d/t)$

D=Pipe diameter, in

t=nominal pipe wall thickness, in

d=depth of penetration of corrosion, in

If the corroded area is approximated by a parabola of area A=2/3 (d t), and if the flow stress is defined as the yield stress plus 10%, $S_{flow} = 1.1\,S_Y$, then failure will occur during a hydrotest conducted at a hoop stress of 100% $S_Y$ if

$$\frac{P_{hydro}D}{2t} = S_Y = 1.1S_Y \frac{1-(2/3)d}{1-\frac{(2/3)d}{M}}$$

By substituting M, we obtain the maximum permissible length of corroded area if $L^2/(Dt) \leq 20$ (what in ASME B31G is written as $A \leq 4$)

$$L = 1.12\sqrt{Dt}\sqrt{\left(\frac{d/t}{1.1(d/t)-0.15}\right)^2 - 1}$$

This ASME B31G equation can be represented by an acceptance curve, as shown in Figure 21-4. Measured wall thinning of depth d and length L that falls beneath the acceptance curve in Figure 21-4 would be acceptable, provided the projected future corrosion would not place it above the curve. Here, acceptable means that the flawed pipe can withstand a hydrostatic test at a pressure given by $P_H D/(2t)=S_Y$.

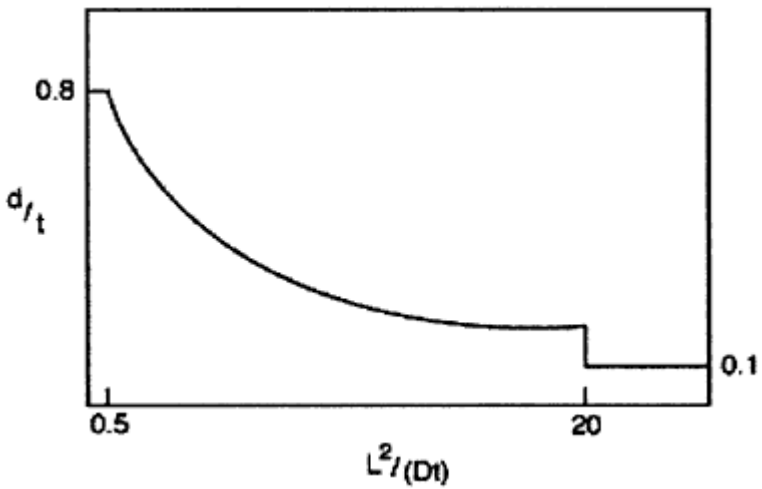


**Figure 21-4** ASME B31G Acceptance Curve

The same ASME B31G rule can be written in terms of maximum safe working pressure P', if $L^2/(Dt) \leq 20$

$$P' = 1.1P_D \frac{1 - \frac{2}{3}\frac{d}{t}}{1 - \frac{2}{3}\frac{d}{t}\frac{1}{\sqrt{1 + \frac{0.8L^2}{Dt}}}}$$

P'=maximum working pressure of a pipe with wall thinning, psi
$P_D$=design pressure of the nominal (unflawed) pipe, psi
d=depth of wall thinning, in
t=nominal wall of pipe, in
L=measured length of wall thinning, in
D=pipe outside diameter, in
and, if $L^2/(Dt)>20$
$P'=1.1\ P_D(1-d/t)$

The pressure that can be sustained in the presence of a flaw of depth d and length L can also be plotted as a function of L and d/t, Figure 21-5. Then, if a pipeline is hydrotested at a pressure $P_T$, all flaws with a combination length L and depth d/t below the horizontal would fail (leak) during the hydrotest, and can be repaired before placing the line in service. Conversely, if a pipeline does not leak during a hydrotest at a pressure $P_T$ it should contain no flaws larger than the combination of lengths L and depth d/t below the $P_T$ horizontal [Kiefner, Leis].

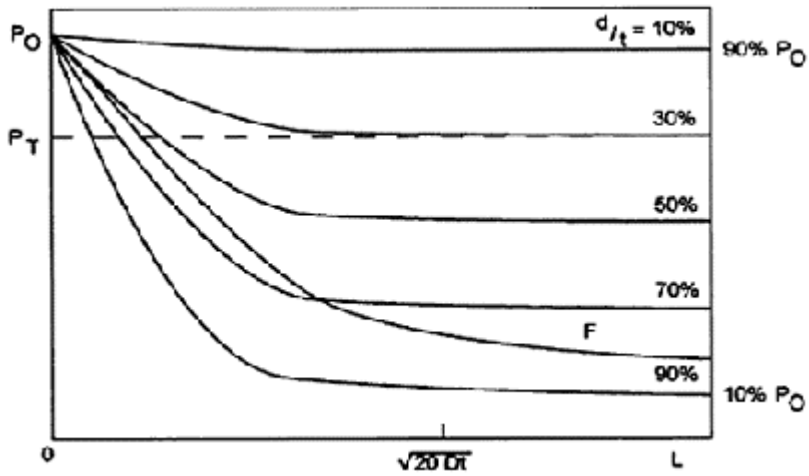


**Figure 21-5** Sustained Pressure as a Function of Flaw Size

A second curve, the failure curve F in Figure 21-5, can be superimposed to the graph. Flaws above this line, should they fail will fail by a large rupture, points above the curve, should they fail, will fail by leakage. In the modified B31G method, the failure stress is [Kiefner 1989, 1990]

$$\left(\frac{PD}{2t}\right)_{failure} = (S_Y + 10ksi)\frac{1-0.85\frac{d}{t}}{1-0.85\frac{d}{t}\frac{1}{M}}$$

The Folias bulging factor is modified to be for $L^2/(Dt) \leq 50$

$$M = \sqrt{1+0.6275\frac{L^2}{Dt}-0.003375\left(\frac{L^2}{Dt}\right)^2}$$

for $L^2/(Dt) > 50$

$$M = 0.032\frac{L^2}{Dt}+3.3$$

ASME B31G is the most commonly used evaluation procedure for wall thinning in ductile steel pipelines, when the only significant stress is the hoop stress due to pressure. But ASME B31G is not the only evaluation procedure. In the NG-18 method, the failure stress is [Kiefner 1978]

$$\left(\frac{PD}{2t}\right)_{failure} = \frac{2S_{flow}}{\pi}\frac{1-\frac{d}{t}}{1-\frac{d}{t}\frac{1}{M}}\cos^{-1}\left(e^{Y}\right)$$

$$Y = \frac{(CVN)E\pi}{4AL(S_{flow})^2}$$

If $L^2/(Dt) \leq 50$

$$M = \sqrt{1+0.625\frac{L^2}{Dt}-0.003375\left(\frac{L^2}{Dt}\right)^2}$$

If $L^2/(Dt) > 50$

$$M = 0.032\frac{L^2}{Dt}+3.3$$

CVN=Charpy V-notch toughness at minimum operating temperature, in-lb
E=Young's modulus of pipe material, psi
A=cross sectional area of Charpy specimen, $in^2$
In the API 579 method, the failure stress is [API 579]

$$\left(\frac{PD}{2t}\right)_{failure} = S_{flow}\frac{1-\frac{d}{t}}{1-\frac{d}{t}\frac{1}{M}}$$

$$M = \sqrt{1+0.8\frac{L^2}{Dt}}$$

In the case of API 579, the flow stress depends on the remaining strength factor (margin) selected.

In the British Gas—DNV method, the failure stress is [Kirkwood, DNV]

$$\left(\frac{PD}{2t}\right)_{failure} = S_U\frac{1-\frac{d}{t}}{1-\frac{d}{t}\frac{1}{M}}$$

$$M = \sqrt{1+0.31\frac{L^2}{Dt}}$$

In the 1999 Battelle method, the failure stress is [Bubenik]

$$\left(\frac{PD}{2t}\right)_{failure} = S_u\left(1-\frac{d}{t}(1-e^{-X})\right)$$

$$X = \frac{0.222L}{\sqrt{D(t-d)}}$$

In ASME Code Case N-513 and ASME Code Case N-480, $L_m$ is measured relative to $t_{min}$, the minimum wall thickness required by the construction code, and the allowable local under-thickness must comply with the following limits

$$L_m \le 2.65\,(R_o t_{min})^{0.5}$$

$$t_{nom} \ge 1.13\,t_{min}$$

$$\frac{t_{aloc}}{t_{min}} \ge \frac{1.5\sqrt{R_o t_{min}}}{L}\left(1-\frac{t_{nom}}{t_{min}}\right)+1$$

$$\frac{t_{aloc}}{t_{min}} \ge \frac{0.353 L_m}{\sqrt{R_o t_{min}}}$$

$L_m$=maximum extent of local thinned area below $t_{min}$, in
$R_o$=outside pipe radius, in
$t_{min}$=minimum wall thickness required by applicable design code, in
$t_{nom}$=nominal wall thickness, in
$t_{aloc}$=allowable local wall thickness, in

### 21.2.4 Circumferential Thinning

Wall thinning along the circumference of the pipe will reduce its net metal cross section area and its section modulus. As a result, the pipe will have a reduced strength against failure by tension or bending. In the case where the applied stress is tensile and axial, the minimum stress to cause failure in the presence of circumferential thinning is [Chell]

$$\sigma_f = S_f(1-d_{eff}/t)$$

$$d_{eff} = \frac{d(1-1/f)}{1-d/(tf)}$$

$$f = \sqrt{1+\frac{(b/t)^2}{2}}$$

$\sigma_f$=lower bound of tensile stress at failure, psi
d=depth of corrosion, in
t=pipe wall thickness, in
b=circumferential width of corrosion, in

### 21.2.5 Cautions

When the applied load is a combination of pressure and bending at the thin area, the integrity of the pipe is more complex than described in sections 21.2.3 and 21.2.4 for pressure alone [Grigory]. API 579 or ASME section XI can be used to evaluate the combined pressure and bending loading condition.

The local metal loss evaluation methods of section 21.2.3 and 21.2.4 are no longer applicable near discontinuities (nozzles, branches, welded attachments, etc.). In this case more elaborate methods (API 579 level 3, finite element analysis or testing) are required [API 579].

If areas of local metal loss are closer to each other than $(Dt)^{0.5}$ they should not be evaluated separately. Instead, they should be treated as a single defect that envelops both local metal loss areas.

If the local metal loss takes place preferentially at a seam weld of questionable quality (for example low frequency ERW seam, Chapter 20), then the pipe or component should be repaired.

Fitness-for-service must account for the future degradation that will continue to take place until the source of corrosion is eliminated.

## 21.3 CRACK FLAWS

Cracks in pipe base metal or welds may be originate during fabrication, transport, or construction; or they may initiate and propagate in service from environmental effects (such as stress corrosion cracking or hydrogen induced cracking), fatigue, or a combination of corrosion and fatigue, as described in Chapter 20. When these cracks are detected in service, it becomes necessary to decide whether the component is fit for continued service, or whether it should be repaired.

When cracks are subject to pressure, residual stresses, or applied loads, large but local stresses exist at the crack tip. Classical stress analysis is of limited use in this case. Fracture mechanics must be used to analyze such conditions. Fracture mechanics had its origins in the work of A.A.Griffith in the early 1900's [Griffith]. It has evolved theoretically, experimentally and numerically to become the key to understanding component failure.

### 21.3.1 Brittle and Ductile Fracture

First, it is important to refer back to Chapter 3 for a description of strength and toughness. Second, it is necessary to summarize the difference between brittle and ductile fracture when trying to predict the fitness-for-service of a cracked component. Several characteristics of brittle and ductile fracture are summarized in Table 21-1, with the following practical notes concerning fracture:

(1) Pipe rupture can be by brittle or ductile fracture. Ductile fracture is preferred because it requires large applied stresses, well into the plastic range; and it is preceded by significant deformation that causes a beneficial load redistribution to the adjoining, more lightly loaded members and metal areas.

(2) Brittle fracture is associated with (a) low temperature service, or (b) high local stress concentration with nearly equal principal stresses, prevalent in thick sections, under plane strain conditions, or (c) dynamic loading (strain rate in the order of 10 $sec^{-1}$).

(3) Brittle fracture can occur below yield, it is elastic, and can be analyzed by linear-elastic fracture mechanics (stress intensity factor K). Ductile fracture is plastic and can be analyzed by elastic-plastic fracture mechanics (J integral or crack tip opening displacement CTOD).

(4) At a given temperature, the static fracture toughness is higher than the dynamic fracture toughness, as illustrated in Figure 21-6. A crack can initiate in a ductile manner under static load (curve S Figure 21-6) and then propagate (dynamic progression of crack) in a brittle manner (curve D, Figure 21-6).

**Table 21-1** Comparing Brittle and Ductile Fracture

| Brittle Fracture | Ductile Fracture |
|---|---|
| Occurs with little elongation | Significant elongation prior to fracture |
| Little plasticity | Substantial plasticity and energy absorption |
| Fracture sudden, little bulging | Pipe wall bulges out under pressure |
| Quick fracture propagation, 7000 ft/sec in steel | Slower fracture propagation |
| Fracture at nominal stress below yield | Fracture at stress exceeding yield |
| Can be caused by high constraint, equal principal stresses, plane strain, thick section. | Caused by maximum shear, plane stress, thin section. |
| Occurs if temperature below NDT | Occurs above NDT |
| Flat cleavage fracture surface | Shear lips |
| V-shaped chevron marks on fracture surface | Ductile tearing, fibrous surface |
| Crack analysis using K | Crack analysis using J or CTOD |
| More prevalent at high strain rate (dynamic impact) | More prevalent at low strain rate (static loading) |

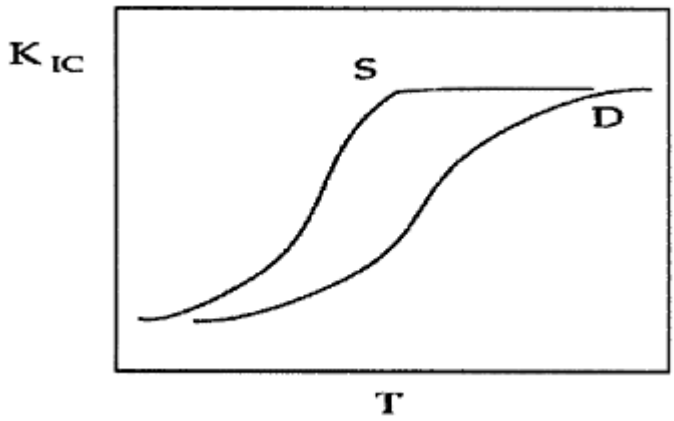


**Figure 21-6** Static (S) vs. Dynamic (D) Fracture Toughness $K_{IC}$

### 21.3.2 Fundamental Approach

There are similarities between classical strength of materials and fracture mechanics. In classical strength of materials, the integrity of a component is established by comparing a calculated stress (for example the maximum shear stress $\tau$) to an allowable stress (for example half the yield stress $S_Y/2$). In linear elastic fracture mechanics, the integrity of a cracked component is analyzed by comparing a stress quantity called the stress intensity factor (K) to an allowable value called the critical stress intensity factor ($K_C$). As in classical stress analysis, fracture mechanics may be elastic (K) or plastic (J), and may be based on stress (K, J) or displacement (crack tip opening displacement or CTOD) [Barsom].

Next, it can be shown theoretically that a pipe crack of size a, subject to an applied nominal elastic stress $\sigma$ is stable under a given stress $\sigma$ if the stress intensity factor $K_I$ is below a critical value $K_{IC}$ called the fracture toughness of the material. $K_{IC}$ is a property of the material, the operating temperature and the state of stress. This condition of crack stability can be written as

$$K_I = \sigma\sqrt{\pi a}F < K_{IC}$$

The geometry factor F depends on the pipe dimension and the crack configuration. Stress intensity factors for a large number of component shapes and cracks are provided in fracture handbooks [Tada, API 579].

### 21.3.3 Stress Intensity

The difficulty in evaluating a crack in the pipe wall arises from the large concentration of stresses at the crack tip. To illustrate this point, we consider the classic problem of an elliptical notch in a flat plate. The plate is loaded in tension, with a nominal stress $\sigma_t$. By

nominal stress we mean the stress in the absence of the notch. The maximum stress at the elliptical notch was first calculated by Kirsch (1898), Kolosoff (1910) and Inglis (1913)

$$\sigma_{max} = \sigma_t\left(1+\frac{2h}{b}\right)$$

$\sigma_{max}$=maximum tensile stress at the side edge of the notch, psi

$\sigma_t$=nominal tensile stress in the absence of notch, psi

h=half width of the ellipse, in

b=height of the ellipse, in

For h=b, we obtain the solution for the stress concentration at the edge of a round hole in a plate $\sigma_{max}=3\sigma_t$. As b→0, the notch becomes a sharp crack and the stress at the edge of the crack becomes infinite. Since an infinite stress is of little use, a different approach is needed to evaluate the effect of stress concentration at the edge of the crack. The solution to this problem was provided by Irwin (1948) and Orowan (1949), on the basis of classical stress theory and one key assumption. First, the classical stress relationships are applied at a point P, a distance r and angle θ from the crack tip. The compatibility equation is

$\Delta(\sigma_{rr}+\sigma_{\theta\theta})=0$

The equilibrium equations in cylindrical coordinates are

$$\sigma_{rr} = \frac{\delta^2 U}{\delta r^2}$$

$$\sigma_{\theta\theta} = \frac{1}{r}\frac{\delta U}{\delta r}+\frac{1}{r^2}\frac{\delta^2 U}{\delta \theta^2}$$

$$\sigma_{r\theta} = -\frac{\delta}{\delta r}\left(\frac{1}{r}\frac{\delta U}{\delta \theta}\right)$$

The condition for finite energy at the crack tip can be written as

$$E(R) = \lim(r \to 0)\int \frac{1}{2}\sigma_{ij}\varepsilon_{ij}dA < \infty$$

The boundary condition for the opening traction $\sigma_{\theta\theta}$ stress and shears stress $\sigma_{r\theta}$ at the crack opening face are

$\sigma_{\theta\theta}=0$

$\sigma_{r\theta}$

To these equations, Irwin added the assumption that, as r→0, the variables r and θ can be separated to calculate the stress components

$\sigma_{rr}(r,\theta)=Ar^n\sigma_{rr}(\theta)$

$\sigma_{\theta\theta}(r,\theta)=Br^n\sigma_{\theta\theta}(\theta)$

$\sigma_{r\theta}(r,\theta)=Cr^n\sigma_{r\theta}(\theta)$

Solving the set of differential equations, as r→0, leads to the general form of the stress at the crack tip

$$\sigma = \frac{K_I}{\sqrt{2\pi r}} f(\theta)$$

σ=nominal stress, ksi

$K_I$=stress intensity, ksi(in)$^{0.5}$

More generally, the stress intensity factor for a crack in a pipe, subject to a nominal stress σ can be written as [Barsom, Blake, Dowling, Liu]

$$K_I = \sigma\sqrt{\pi a}F$$

$K_I$=stress intensity, ksi(in)$^{0.5}$

a=crack dimension, in

F=geometry factor

σ=nominal stress, in the absence of crack, ksi

The stress intensity is a reflection of the state of stress and the geometry at a point along the crack. It is not to be confused with the stress intensity factor established by cyclic fatigue test of pipe fittings and welds (Chapter 7). Note that we have labeled the stress intensity factor $K_I$. The subscript "I" refers to a stress intensity factor that corresponds to a first mode of loading. The first mode is an applied load that tends to open the crack lips by tension, Figure 21-7. There are also mode II and mode III stress intensity factors $K_{II}$ and $K_{III}$ that tend to shear the crack as indicated in Figure 21-7.

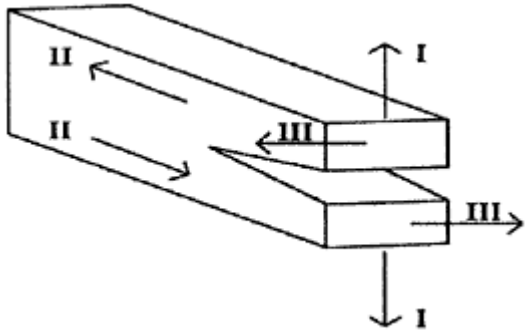


**Figure 21-7** Three Modes of Fracture

In practice, for piping systems, the tensile mode $K_I$, which tends to open the crack in tension, is of primary importance. It follows that for a mode I analysis, the nominal stress of interest is that which acts in tension and perpendicular to the crack. Therefore, for a longitudinal crack the stress of interest is the hoop stress $\sigma_h$ (which is circumferential, perpendicular to the longitudinal crack) and for a circumferential crack it is the longitudinal stress $\sigma_l$ (which is axial and perpendicular to the circumferential crack). For example, for a through-wall longitudinal crack in a straight pipe under internal pressure,

Figure 21-8, and assuming there are no other residual or applied stresses, the stress intensity factor is given by [Tada]

$K_I=\sigma_h (\pi a)^{0.5} (1+0.52\kappa+1.29\kappa^2-0.074\kappa^3)^{0.5}$

$\sigma_h=PD/(2t)$

$\kappa=a/(Rt)^{0.5}$

$K_I$=stress intensity, ksi(in)$^{0.5}$

a=half length of through-wall crack, in

P=internal pressure, psi

D=outer diameter, in

R=pipe radius, in

t=pipe wall thickness, in

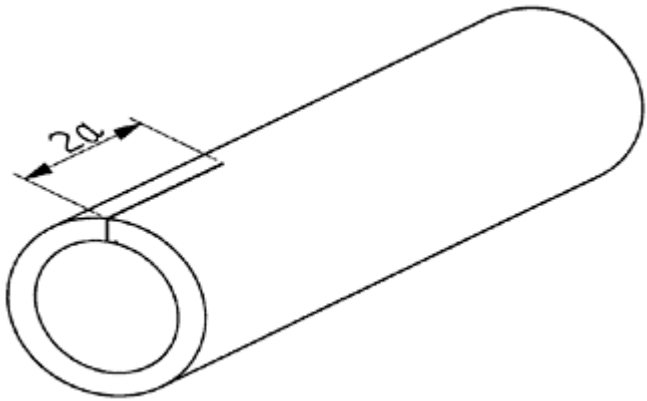


**Figure 21-8** Through-Wall Crack of Length 2a

For example, for a 20" diameter (R=10"), with a 0.25" wall (t=0.25") API 5L pipeline, with a maximum operating pressure of 500 psi (P=500 psi) and a 6" long through wall crack (a=3"), we obtain $\sigma_h$=20 ksi, $(\pi a)^{0.5}$=3.07, $\kappa$=1.90, and $(1+0.52\kappa+1.29\kappa^2-0.074\kappa^3)^{0.5}$=2.48, therefore $K_I$=152 ksi(in)$^{0.5}$

### 21.3.4 Fitness-for-Service Evaluation

The fitness-for-service evaluation of a cracked pipe consists in calculating its fracture toughness ratio (margin against failure by brittle fracture) and its load ratio (margin against failure by ductile fracture), and verifying that both ratios are acceptable. The toughness ratio is

$$K_r = \frac{K_{I,P} + K_{I,S}}{K_{IC}} + \rho$$

$K_r$=toughness ratio, ksi(in)0.5

$K_{I,P}$=stress intensity factor on primary stresses, ksi(in)$^{0.5}$

$K_{I,S}$=stress intensification factor on secondary and residual stress, ksi(in)0.5

$K_{IC}$=fracture toughness, ksi(in)$^{0.5}$

ρ=plasticity interaction factor

The load ratio is

$$L_r = \frac{\sigma_{ref}}{S_{flow}}$$

$\sigma_{ref}$=reference (applied) stress, psi

$S_{flow}$=flow stress of the material, psi

$\sigma_{ref}=\sigma_P F$

$\sigma_P$=applied primary stress, psi

F=dimensionless factor that depends on geometry and crack size

The flow stress is defined in several ways; depending on the standard or author, it is typically either the mean value of the yield and ultimate stress of the material at the evaluation temperature, or the yield stress plus 10 ksi.

Having calculated the toughness ratio $K_r$ and the load ratio $L_r$, we enter the failure assessment diagram (FAD), a curve such as illustrated in Figure 21-9, defined as [API 579]

$K_r=(1-0.14L_r^2)[0.3+0.7\exp(-0.65\,L_r^6)]$

With an upper bound for $L_r$=1.0 for materials with a yield point plateau; $L_r$ =1.2 for C-Mn steel; and $L_r$=1.8 for austenitic stainless steel.

In practice there are several difficulties that must be resolved when using the failure assessment diagram (FAD):

(1) The applied stress intensity factor $K_I$ and reference stress $\sigma_{ref}$ can become difficult to calculate for shapes other than a straight pipe (such as a fitting or nozzle) and for loading other than internal pressure (such as combined torsion, bending and tension).

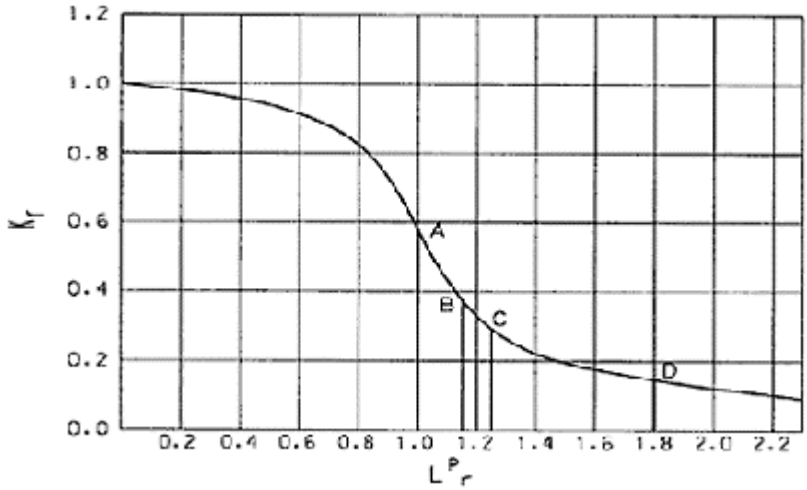


**Figure 21-9** Failure Assessment Diagram (FAD)

(2) The residual stresses and the resulting stress intensity factor $K_{I,S}$ can be difficult to calculate. Residual stress distributions have been published for common weld configurations [API 579], but certain cases will require detailed heat-stress-material coupled numerical analyses and confirmatory testing to obtain a final distribution of residual stresses. In the last few years, the magnitude and distribution of residual stresses have been the subject of extensive testing and thermomechanical finite element analysis [Dong]. Some interesting features of residual stresses are illustrated in Figure 21-10: as the weld cools down and shrinks, tensile hoop stresses develop through the weld section (stress A in Figure 21-10). This circumferential contraction causes bending stresses in the axial direction (compressive stresses at B and tensile at C). A certain distance XY from the weld centerline X, with XY in the order of $(Dt)^{0.5}$, the stress distribution is reversed to equilibrate the stresses in the weld. For the same diameter over thickness (D/t) ratio, if the thickness increases, the distance XY gets shorter and the stresses at Y are now sufficiently close to alter the resultant distribution of residual stresses in the weld at X. The new stress distribution is illustrated at bottom of Figure 21-10. The hoop stress is still tensile through the weld, but is now smaller in the center (A') than at the inner and outer surface (A). The axial residual stress that was a bending stress (B to C) is now self-equilibrating, which means that the tensile stresses at top and bottom (H) are counteracted by compressive stresses at the center of the weld (H'). We can see the importance of these changes in the residual stress field since crack propagation is governed by the magnitude of tensile stresses [Dong].

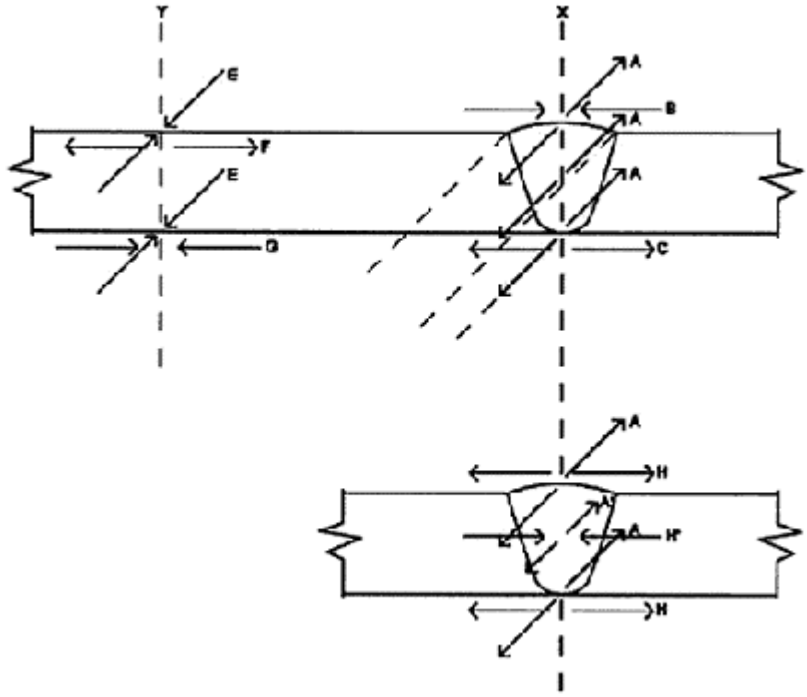


**Figure 21-10** Residual Stresses in a Girth Weld

(3) Testing required to obtain the fracture toughness $K_{IC}$ may not be feasible, due to lack of a sufficiently large specimen or having no access to an equipped test laboratory. A conservative estimate of $K_{IC}$ will then be in order. A relatively simple method, referred to as automated ball indentation (ABI), has been proposed for in-place measurement of fracture toughness. It is a non-destructive technique based on the progressive indentation of the pipe surface [Haggag].

(4) The uncertainty in the calculated stress field, the measured crack length and depth, and the material properties must be estimated case by case, and safety factors must be introduced to compensate for each one of these three unknowns.

(5) The prediction of future crack propagation is essential in projecting the fitness for service, after days, weeks or months of continued service. Crack mechanisms are more difficult to fully understand and project into the future than wall thinning. For this reason alone cracks are, in many cases, immediately repaired rather than analyzed for fitness-for-service.

(6) In the case of applied plastic stress, it becomes more difficult to predict the stress and to measure the elastic-plastic J-integral [Barsom].

### 21.3.5 Crack Arrest

We can now look back at Figure 21-1, and better understand what differentiate a leak (Figure 21-1 (g)) from a long break (Figure 21-1 (h)). A crack will arrest when the energy U driving the crack falls below the energy W absorbed in the fracture; the condition for crack arrest can therefore be written as [Maxey]

$U<E$

with

$U=\sigma^2 \pi R/E$

$W=CVN/A_{CVN}$

U=energy driving the crack propagation, in-lb/in$^2$

σ=tensile stress opening the crack, psi

R=pipe radius, in

E=pipe Young's modulus, psi

W=energy absorbed during the fracture process, in-lb/in$^2$

CVN=Charpy V-notch toughness of the pipe material, in-lb

$A_{CVN}$=cross section of the Charpy specimen, in$^2$

It is evident from the expression of U and W that a crack will arrest if it reaches a section of larger thickness (and therefore lower stress σ) or ofhigher toughness (causing CVN to go up). Also a pressure drop at the crack tip can arrest the crack. In a liquid pipeline, the pressure at the crack tip will very quickly drop, therefore reducing σ and the driving energy U. In contrast, in a gas pipeline, the pressure drop will be slow and may even lag behind the crack propagation, a will therefore remain large and drive the crack a long distance. A classic example is the 1960 crack in a 30" gas pipeline that propagated over eight miles before arresting.

### 21.3.6 Fatigue

The behavior of pipe and pipe fittings subject to fatigue has been addressed in Chapter 7. Fatigue can be due to cyclic mechanical loads (for example pressure pulses), cyclic displacements (for example vibration), or cyclic thermal shocks (for example thermal gradients). Fatigue can also occur in combination with corrosion. The fitness-for-service evaluation of a piping component subject to mechanical fatigue, without corrosion, consists in (a) defining the cause of fatigue and the driving force or displacement, (b) obtaining a conservative estimate of passed and future cycles, (c) calculating the intensified stress range due to fatigue (iM/Z), (d) comparing the calculated stress range to the calculated allowable stress range $S_a= f (1.25 S_C+0.25 S_h)$ where f is a function of the number of cycles. If the number of cycles is infinite, the allowable stress would be the endurance limit of the material. In the absence of an established endurance limit (Chapter 7), $S_a$ may be used with f=0.15 [B31.3]. Where the applied cyclic load is not constant, the lesser cycles are combined into an equivalent number of cycles, as described in Chapter 7.

Corrosive environments (such as hydrogen rich gases, acid or aqueous environments) reduce the fatigue life of components. There are several theories and models to explain the combined action of corrosion and cyclic stress. For example, it is believed that the surface of a crack is strengthened by the oxide layer, and therefore the propagation of a crack exposes non-oxidized weaker metal.

Quantitative models and empirical coefficients have been developed to interpret corrosion fatigue experiments. For example, the crack growth rate of a material can be written as the sum of a purely mechanical contribution plus a stress corrosion contribution [Suresh]

$$\frac{da}{dN} = \left(\frac{da}{dN}\right)_M + \left(\frac{da}{dN}\right)_{SC}$$

a=crack size, in
N=stress cycles
M=denotes mechanically induced
SC=denotes stress corrosion induced

## 21.4 MECHANICAL DAMAGE

### 21.4.1 Ripple

Ripples are mild waves that can form during bending of a pipe or pipeline, Figure 21-11. The ASME B31 design and construction codes provide qualitative requirements generally stating that “bends must be free of buckles”. The Pipe Fabrication Institute recommends that ripple height be no larger than 3% of the diameter [PFI ES-24].

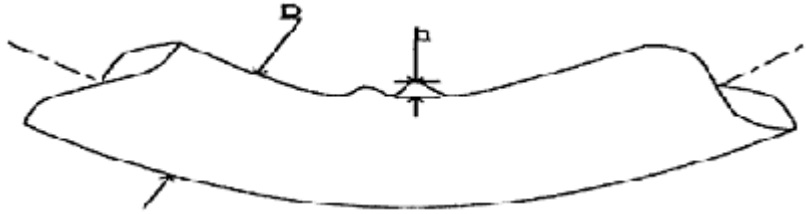

**Figure 21-11** Ripple Height

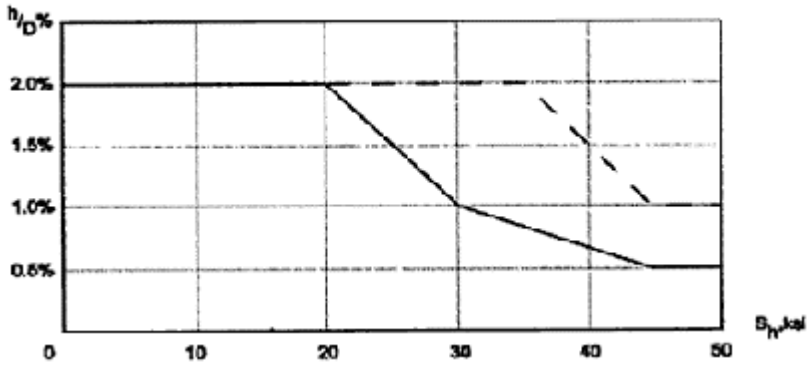


**Figure 21-12** Maximum Ripple Height (h/D) vs. Hoop Stress $S_h$ [Rosenfled]

Ripple bends are particularly costly in oil and gas pipelines because they cause the rejection of a bend or must be evaluated when detected by in-line inspection tools (smart pigs, Chapter 16), and require digs, analysis of the as-found condition, or replacement. Recently, a comprehensive research program was undertaken for the Pipeline Research Council International (PRCI) to determine under what conditions a mild bend ripple can cause rupture in a carbon steel line pipe. A ripple characterized by h/D, where h is the ripple height and D is the pipeline diameter (Figure 21-11) does not pause a risk of fatigue failure if the hoop stress in the line is below the curve shown in Figure 21-12 [Rosenfeld]. This conclusion is based on a conservative assumption of 100-year service life with aggressive pressure cycling. Since liquid pipelines (incompressible fluid) undergo many more cycles than gas pipelines, their acceptance curve is lower.

### 21.4.2 Buckle and Wrinkle

A buckle or a wrinkle is a kink caused by local buckling of the pipe wall, usually as a result of either (a) a tight curvature in a field bend, or (b) soil settle-ment that locally bends the buried pipeline. Construction standards place limits on buckles and wrinkles, for example the height of the buckle or wrinkle must be limited to 1% of the nominal diameter, with a length to depth ratio of 12/1, and consecutive buckles must be no closer than 1 diameter [DNV], or peak-to-valley height of the buckle or wrinkle not to exceed 5% of the diameter [AS 2885]. In addition, the evaluation of a buckle or wrinkle must take into consideration the potential for coating disbondment if the strain exceeds the coating vendor limits, typically in the order of 2% to 5%.

### 21.4.3 Dent

Dents are local deformations of the circular cross section, and gouges are sharp grooves, Figure 21-13. For hazardous liquid pipelines, dents are limited by construction codes to ¼” depth on 4” and smaller pipelines and 6% of the diameter in sizes greater than 4”, with maximum resulting ovality of the cross section ranging from 2.5% to 5% [ASME

B31.4, CSA Z662]. Dents can be constrained (formed for example when the pipe bears against a rock that remains lodged under the pipe) or unconstrained (for example when the dent is caused by an impact), as described in Chapter 20. Two techniques to analyze the fitness-for-service of dented or gouged pipes are API 579 and API 1156. In the API 579 approach, a plain dent (without a sharp gouge) is acceptable if its depth at the instant of damage $d_d$ is less than 5% the pipe diameter. The dent at the instant of damage $d_d$ can be calculated from the dent depth at operating pressure $d_{dp}$ by [API 579]

$$d_d = \frac{d_{dP}}{-0.22 \ln \frac{\sigma_P}{S_f}}$$

$d_d$=dent at instant of damage, in

$d_{dP}$=dent after rebound at pressure P, in

$\sigma_P$=hoop stress at pressure P, psi

$S_f$=flow stress of material, psi

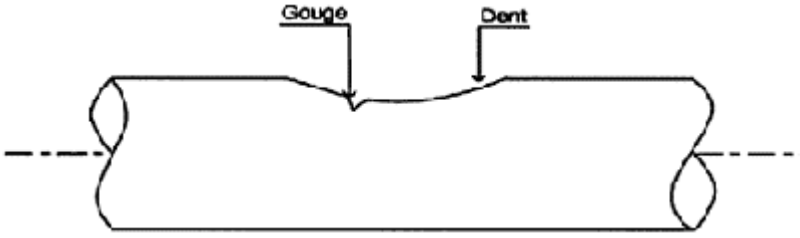


**Figure 21-1**3 Gouge and Dent

Although there is no quantitative fitness-for-service rule for a dent or a gouge in a pipeline, API 1156 provides a method for ranking the severity of dents, in the following manner

P=(A+B+C+W+M+R+K) (d/D)

P=severity ranking.

A=pipe size factor (varies from 1 for D/t<50 to 2 for D/t>100).

B=material quality factor (varies from 1 for modern line pipe to 10 for low toughness, older, low frequency seam welded materials).

C=operating pressure factor (varies from 1 for operating stress below 20% of yield to 4 for cyclic service above 60% of yield).

W=weld factor (varies from 0 away from welds to 10 for a dent at weld).

M=metal loss factor (varies from 0 for no metal loss to 5 if there is metal loss).

R=dent location factor (varies from 5 if dent at bottom of pipe, likely to be constrained by rock, to 30 if dent is at top of pipe, likely due to third-party damage).

K=buckle factor (varies from 0 if buckle is less than 2% of pipe diameter to 5 if buckle exceeds 2% of diameter).

d/D=ratio of buckle height divided by diameter.

The fatigue life of a dented pipe, without a gouge, is given by [API 1156]

$N=\exp [43.944-2.79 \ln (\Delta\sigma/2)]$

$\Delta\sigma=(SCF)\ \Delta P$

N=pressure cycles to failure (through-wall fatigue crack)

$\Delta\sigma$=stress cycle range, psi

SCF=stress concentration factor (a function of dent shape, its constraint, and the ratios d/D and D/t [API 1156])

$\Delta P$=pressure cycle range, psi

### 21.4.4 Dent with Gouge

Dents (deformation of the circular cross section of a pipe, for example as a result of impact by excavation equipment) and gouges (sharp cut on the pipe surface, for example at the precise point where excavation equipment strikes a pipeline) are the leading cause of pipeline failure in service, Figure 21-13. The failure occurs either instantly, by puncture, or at a later time. Delayed fractures are primarily due to pressure cycling during which the dented section breathes in and out as the internal pressure decreases and increases respectively. This breathing mode causes bending in the pipe wall at the dent, as if the dent region is a plate bent back and forth by the fluctuating internal pressure. This cyclic bending is intensi-fied at the sharp gouge, causing local peak stresses that will eventually fail the pipe wall by fatigue. This effect can be worsened by cold working of the pipe wall at the gouge, and the possible presence of a crack at the gouge. If a dent with a gouge is discovered on a line operating at high hoop stress, in the order of $0.5S_Y$ (a hoop stress level common in oil and gas pipelines), the line pressure should be reduced by at least 20% and the dented and gouged section should be repaired either by grinding out the gouge or by wraping the damaged section with a full encirclement sleeve or a stiff overwrap (Chapter 23). The objective of the sleeve or overwrap is to stiffen the dented section and stop the breathing (flexing) of the pipe wall, eliminating the source of cyclic fatigue stresses.

## 21.5 REFERENCES

API 579, Fitness for Service, American Petroleum Institute, Washington D.C.

API 1104, Welding Pipelines and Related Facilities, American Petroleum Institute, Washington D.C.

AS 2885, Pipelines—Gas and Liquid Petroleum, Council of Standards, Australia.

ASME B31G, Manual for Determining the Remaining Strength of Corroded Pipelines, Supplement to ASME B31 code for pressure piping, 1991, American Society of Mechanical Engineers, New York.

ASME B31.3, Process Piping, American Society of Mechanical Engineers, New York.

ASME B31.4, Liquid Petroleum Transportation Piping, American Society of Mechanical Engineers, New York.

ASME CC N-480, Examination Requirements for Pipe Wall Thinning Due to Single Phase Erosion and Corrosion, Section XI, Division 1, American Society of Mechanical Engineers, New York.

ASME CC N-513, Evaluation of Temporary Acceptance of Flaws in Moderate Energy Class 2 and 3 Piping, Boiler & Pressure Vessel Code, Section XI, Division 1, American Society of Mechanical Engineers, New York.
ASTM E 399, Standard for Conducting Elastic Fracture Toughness Test, West Conshohocken, PA.
ASTM E 813, Standard for Conducting Elastic-Plastic Fracture Tests, American Society for Testing of Materials, West Conshohocken, PA.
Barsom, J.M., Rolfe, S.T., Fracture and Fatigue Control in Structures, ASTM, West Conshohocken, PA.
BS 5762, Methods for Crack-Opening Displacement Testing, British Standard Institute, UK.
Blake, A., Practical Fracture Mechanics in Design, Marcel Dekker, New York.
CSA Z662, Oil and Gas Pipeline Systems, Canadian Standard Association, Canada.
Chell, G.G., Elastic-Plastic Fracture Mechanics, Developments in Fracture Mechanics, 1979.
Dong, P., Hong, J.K., Recommendations for Determining Residual Stresses in Fitness-forService Assessment, WRC 476, Welding Research Council Bulletin 476, November, 2002.
Dowling, N.E., Mechanical Behavior of Materials, Prentice Hall, New Jersey.
DNV OS-F101, Submarine Pipeline Systems, Det Norske Veritas, Sweden.
DNV Recommended Practice RP-F101, Corroded Pipelines, Det Norske Veritas, 1999.
Folias, E.S., An axial Crack in a Pressurized Cylindrical Shell, Int. Journal of Fracture Mechanics, Vol. 1(1), 1965.
Griffith, A.A., The Phenomenon of Rupture and Flow of Solids, Philosophical Transactions of the Royal Society, London, 1921.
Grigory, S., et. al., Pipelines Terminals & Storage, Book II Conference Papers, API, 1996.
Haggag, F.M., In-Situ Measurement of the Toughness Properties of Oil and Gas Pipelines, Pipes and Pipelines International, Vol. 48, No. 1, January–February, 2003.
Hertzberg, R.W., Deformation and Fracture Mechanics of Engineering Materials, John Wiley and Sons, New York.
Kania, R., Carroll, L.B., Non-Destructive Techniques for Measurement and Assessment of Corrosion Damage on Pipelines, International Pipeline Conference, Volume I, 1998, American Society of Mechanical Engineers.
Kiefner, J.F., Maxey, W.A., Pressure Ratios Key to Effectiveness, Oil & Gas Journal, July 31, 2000.
Kiefner, J.F., Maxey, W.A., Model Helps Prevent Failures from Pressure Induced Fatigue, Oil & Gas Journal, August 7, 2000.
Kiefner, J.F., Maxey, W.A., Eiber, R.J., Duffy, A.R., Failure Stress Levels of Flaws in Pressurized cylinders, ASTM STP536, 1978.
Kiefner, J.F., Vieth, P.H., A Modified Criterion for Evaluating the Remaining Strength of Corroded Pipe, AGA Project PR3–805, AGA catalog No. L51609, Dec. 22, 1989.
Kiefner, J.F., Vieth, P.H., New Method Corrects Criterion for Evaluating Corroded Pipe, Oil & Gas Journal, August, 1990.
Kirkwood, M., Bin, F., Improved Guidance for Assessing the Integrity of Corroded Pipelines, ASME Pressure Vessel and Piping Conference, Hawaii, 1995
Lies, B.N., Verley, R., Quality Control, NDT Offer Alternatives to Hydrotesting, Pipeline & Gas Industry, August 2000.
Liu, A.F., Structural Life Assessment Methods, ASM International, Metals Park, OH.
Maxey, W.A., et. al., Ductile Fracture Initiation, Propagation and Arrest in Cylindrical Vessels, Fracture Toughness Proceedings of the 1971 National Symposium on Fracture mechanics, Part II, ASTM STP 514, American Society for Testing of materials, 1971.
Maxey, W.A., et. al., Brittle Fracture Arrest in Gas Pipelines, Catalog L51436, American Gas Association, Arlington, VA, 1983.
PFI ES-24, Pipe Bending Methods, Tolerances, Process and Materials Requirements, Pipe Fabrication Institute, Springdale, PA.

Rosenfeld, M.J., Hart, J.D., Zulfiqar, N., Gailing, R.W., Development of Acceptance Criteria for Mild Ripples in Pipeline Field Bends, Proceedings of the International Pipeline Conference, September, 2002, Alberta, Canada, American Society of Mechanical Engineers, New York.

Sims, J.R., et. al., A Basis for the Fitness for Service Evaluation of Thin Areas in Pressure Vessels and Storage Tanks, ASME PVP Volume 233, ASME, 1992, American Society of Mechanical Engineers, New York.

Stephens, D.R., Bubenik, T.A., Francini, R.B., Residual Strength of Pipeline Corrosion Defects Under Combined Pressure and Axial Loads, AGA NDG-18 Report 216, February 1995.

Tada, H., Paris, P.C., Irwin, G.R., The Stress Analysis of Cracks Handbook, ASME Press, New York.

Tetelman, A.S., McEvily, A.J., Fracture of Structural Materials, John Wiley and Sons, New York.

Wells, A.A., The Meaning of Fitness-for-Purpose and Concept of Defect Tolerance, International Conference, Fitness-for-Purpose Validation of Welded Constructions, The Welding Institute, London, 1981.

# 22
# Maintenance, Reliability and Failure Analysis

## 22.1 CASE HISTORY

Monday, December 2, 1984, around 11 PM, things seemed routine when the operators greeted each other for the shift change at a pesticide plant in Bhopal, India. Among its many processes, the plant contained three 15,000-gallon tanks storing methyl isocyanate (MIC), an intermediate ingredient in the fabrication of pesticide. As illustrated in Figure 22-1, MIC is supplied to the tanks through a MIC transfer line (MICTL in Figure 22-1). When needed, MIC is transferred out to the next process unit through underground lines (dashed lines) by pressurizing the tank with nitrogen at about 14 psig (the nitrogen header is noted NH in Figure 22-1).

Each tank was equipped with a pressure relief valve, discharging into a common relief valve vent header (RVYH in Figure 22-1). The vent header was piped to vent gas scrubbers (VGS, lower right in Figure 22-1), then burned and discharged up a flare. These precautions are necessary since the accidental mixing of water and MIC would cause an exothermic reaction, and the discharge of lethal cyanide gases. To further mitigate the exothermic reaction, the tanks are equipped with cooling coils to control the MIC temperature.

Shortly after the shift change, the operators noted a sudden rise of pressure in one of the three tanks. When they hurried to the area, they could hear a rumbling sound coming from the underground tank [Kalelkar, Browning]. Shortly afterward, a cloud of methyl isocyanate gas discharged from the stack, unburned, and enveloped the populated area surrounding the plant, causing a tragedy: over 2,000 fatalities, and countless injuries.

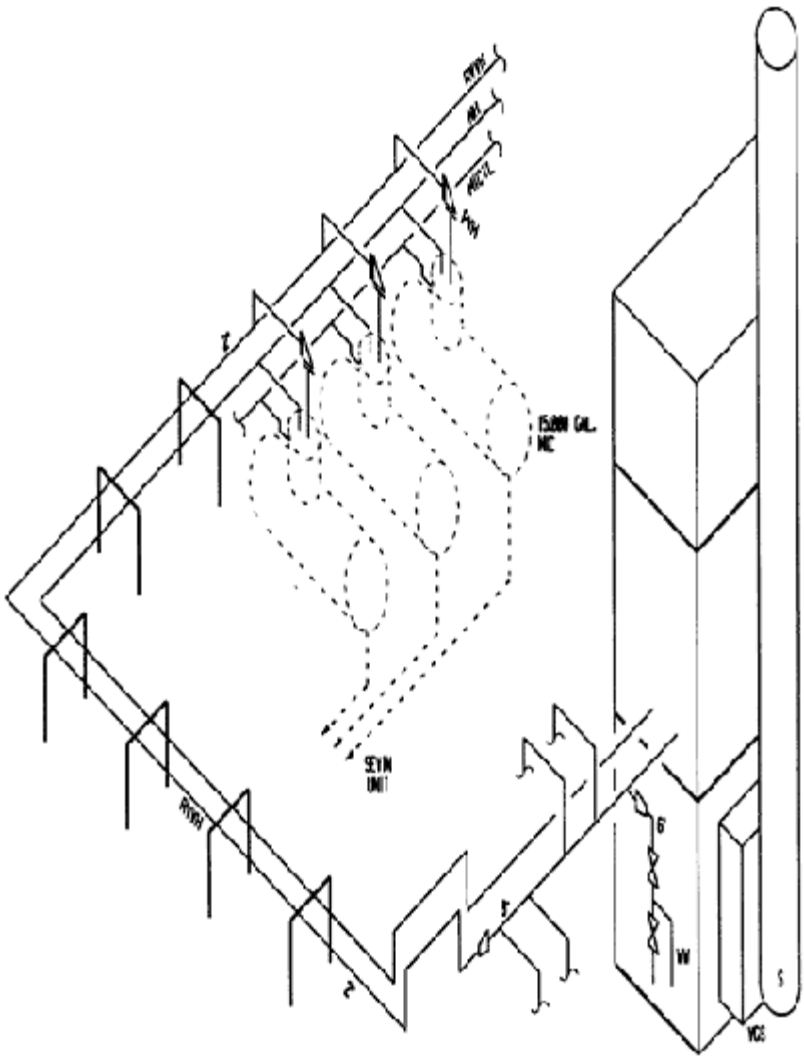

**Figure 22-1** Simplified Illustration of MIC Tank Systems

The source of water that caused the exothermic reaction has been the subject of much controversy. However, once the water-MIC reaction took place, gases escaped through the stack, unscrubbed and unburned, because the tank cooling system was shut down and the flare was out of service [Kletz, Lees]. Such a chain of failures and break-down in conduct of operations and maintenance would seem incredible. Yet, it did occur.

## 22.2 MAINTENANCE OBJECTIVE

The objective of maintenance has been described as "ensuring that physical assets continue to do what their users want them to do" [Moubray] and "to preserve the system function" [Smith]. In the case of piping systems, "physical assets" are pipes, valves, active equipment (pumps, compressors, etc., also referred to as dynamic or rotating equipment), instrumentation, fixed equipment (vessels, heat exchangers, etc., also referred to as static equipment), in-line components (traps, strainers, etc.), insulation and supports. There are also supporting systems that permit the piping system to perform their function: instrumentation and controls, heat tracing, power or air supply to valve operators, etc.

Maintenance is where the hardware speaks to, and at times screams at, the engineer. Whether the system is operating flawlessly, malfunctions or breaks, it is giving the engineer important information. Operating companies handle this information in a variety of manners:

(a) Maintenance information is ignored. Things are fixed as they break, time and again.

(b) Maintenance information is recorded as data, somewhere on a server, on a shelf or in a drawer.

(c) Maintenance data is converted into knowledge for a few, the maintenance mechanic and possibly the system engineer.

(d) Maintenance knowledge is converted into wisdom, by analysis, trending, and communication (in a clear and illustrated manner) to the whole organization.

This progression from nothing, to data, to knowledge, to wisdom differentiates excellent operations from mediocre ones.

## 22.3 MAINTENANCE PLAN

The logical steps of a maintenance plan are:

(1) For each system, determine a maintenance strategy: proactive or reactive, as described in section 22.4.

(2) For each system identified for proactive maintenance, prepare a component list (pipe segments, valves, pumps, compressors, etc.). For each component, specify the required function, its failure mode and failure cause, referring to section 22.6.

(3) For each component failure cause, select the proper inspection technique, referring to sections 22.7 and 22.8. The objective here is to decide what needs to be inspected, when, where and how.

(4) Determine the acceptance criteria that will be used to evaluate the inspection results and to determine the "fitness-for-service" of the system (Chapter 21).

(5) Plan and implement maintenance inspections, either on stream (on line) or during an outage (shutdown).

(6) Document results, intelligently, clearly, and succinctly. Illustrate with digital photographs. Record in a retrievable and sortable maintenance database. Maintain "system health reports".

(7) Issue clear recommendations, for example in three categories: green (ok as-is), yellow (plan for future inspection, degradation is taking place but equipment is fit-for-service till next inspection), or red (repair or replace).

## 22.4 MAINTENANCE STRATEGIES

There are two maintenance strategies: (1) a reactive approach (corrective maintenance, running equipment to failure, or near failure); or (2) a proactive approach (inspecting equipment and taking early steps to overhaul, repair or replace, before failure). Within the pro-active strategy, we can differentiate between preventive maintenance where the inspections are time based, planned at fixed intervals (such as oil change every so many months), and predictive maintenance where the inspections are condition based, resulting from the analysis and trending of inspection results [NTIAC]. The goal of predictive maintenance is to achieve a necessary and sufficient degree of reliability. By necessary we mean that this approach should be implemented only where it is necessary. There are systems where pro-active maintenance is unnecessary, these are systems whose failure would be of little consequence to safety and operations and they can be readily repaired and returned to service. Corrective maintenance would be appropriate in these cases. By sufficient we mean that, where predictive maintenance is judged necessary, it should be conducted in a manner that minimizes costs while achieving the desired level of reliability. The objective is not to have equipment as good as new, but sufficiently good to perform their function, reliably and safely. Systems that are part of a pro-active maintenance strategy include:

(a) Facility safety basis: systems essential to prevent or mitigate credible accidents that would have unacceptable consequences to the workers, the public or the environment.

(b) Production loss: systems essential to maintain an acceptable level of production throughput.

(c) Maintenance cost: systems with equipment that would be costly to replace, or would require long lead times.

(d) Risk of failure: systems at greater risk of failure, for example because of corrosion, operation at high pressure or temperature, operation beyond vendor recommendations, or based on past company or industry experience.

(e) Regulatory requirements: systems or components that are required, by regulation, to be periodically inspected or tested [OSHA, 29CFR, NBIC].

## 22.5 CORRECTIVE MAINTENANCE

Corrective maintenance is reactive maintenance: run to failure, then repair or replace. This maintenance approach is quite common for non-essential systems that do not exhibit any of the characteristics listed in section 22.4. Maintenance managers cite limited manpower and budgets focused first on solving the day's emergencies as an impediment to predictive maintenance. A recent survey reported corrective maintenance at 40% of the maintenance workload [Rockwell]. Well-implemented, corrective maintenance yields a wealth of knowledge. A competent, cost-conscious operation will capture its corrective

maintenance history in an accessible maintenance database. As a minimum, the following information should be recorded by the maintenance mechanic for each corrective maintenance work package:

(a) Equipment make and model.
(b) As-found condition (photographs are recommended).
(c) Mechanics' opinion as to the likely cause of failure.
(d) Corrective action (and possibly recommendation to avoid recurrence).

The value of such maintenance history is self-evident. If a mechanic closes a maintenance work package by stating "gasket leaks, replaced gasket", nothing is learned, nothing is gained. Instead, every leak or failure must be viewed as an opportunity to learn, to improve the system reliability. For the same maintenance work package, a competent documentation would be:

(a) Steam system 300 #, location ABC, Valve DEF, inlet nozzle gasket GHI type JKL.

(b) Part of gasket blown off in service. Gasket appears very brittle, broke into pieces when removed. No other evidence of damage (flange faces, pipe, valve, bolts visually inspected). Photo enclosed.

(c) May not be the right gasket material for 300# steam service. Not used in our other steam systems.

(d) After consultation with engineering (MNO) and gasket supplier (PQR), replaced inlet and outlet nozzle gaskets with gasket STU type VWX.

## 22.6 FAILURE MODES

When a company operates a large number of equipment, corrective maintenance history should go beyond the narrative of section 22.5. Instead, failure mode and failure cause should be captured in a standard format, and regularly sorted, analyzed and trended. The objectives of this continuous analysis is to understand failure cause, and take pre-emptive measures to avoid recurrence, optimize performance, reduce costs, and improve safety. To achieve these objectives, a company may procure a commercially available maintenance tracking and trending software that already includes failure mode and failure cause templates. The company may also join an industry group that has developed tools specific to equipment and operating conditions unique to the industry. Finally, the company may develop its own maintenance history software, based on its experience, its equipment vendors' input, and pertinent publications [Mobley, Moubray, Timmins].

To help in documentation and sorting, each class of equipment (horizontal centrifugal pumps, reciprocating compressors, pipe, flange, etc.) would have a standard list of failure modes and failure causes. Such a standard list would also help the mechanic diagnose and categorize the equipment condition. The standard list also provides a consistent documentation that greatly facilitates retrieval, analysis and trending of maintenance histories. For example, a menu of failure modes and causes for centrifugal pumps follows:

Failure Mode 1—Pumps Fails to Start

Corresponding Failure Causes:

Loss of power.

Internal binding.

Failed bearing.

Failed coupling.

Open or shorted motor.

Start circuit fails.

Failure Mode 2—Pump Delivers Inadequate Flow

Corresponding Failure Causes:

Worn or broken impeller.

Worn wear ring.

Discharge valve closed.

Cavitation.

Seal failure.

Casing cracked.

Gasket leak.

Clogged strainer.

Shaft damage.

Failure Mode 3—Pump Exhibits Abnormal Condition

Corresponding Failure Causes:

Excessive vibration.

Leak of process fluid.

Oil leaks.

Excessive temperature.

Unusual noise.

A second level of failure causes—the cause of the cause—may be necessary to diagnose and correct the failure mode. An example of second level failure causes follows:

Failure Cause 1st Level—Excessive Vibration in Centrifugal Pump

Failure Cause 2nd Level:

Mechanical Cause:

Unbalance.

Eccentric rotor.

Bent shaft.

Axial misalignment at shaft coupling.

Angular misalignment at shaft coupling.

Loose foot.

Rotor rub against fixed part.

Bearing wear.

Oil instabilities.

Gear worn or broken.

Faulty motor.

Belt drive misaligned.

Hydraulic Cause:

Pressure pulsing from vane pass.

Flow turbulence.

Cavitation.

Hydraulic resonance (Helmholtz oscillator).

## 22.7 PRO-ACTIVE MAINTENANCE

### 22.7.1 Preventive or Predictive Maintenance

Where a system cannot be run to failure, it has to be part of the pro-active maintenance program. A choice must now be made between Preventive Maintenance (PM) or Predictive Maintenance (PdM). With PM, also referred to as Scheduled Maintenance [Patton], pre-determined maintenance activities take place at predetermined intervals; for example, replacing pump lube oil every X months, testing a relief valve every X years, etc. The type of maintenance activity and the interval are established based on several factors:

(a) Equipment failure history.
(b) Vendor recommendations.
(c) Industry practice, codes, standards.
(d) Personnel experience.
(e) Risk: likelihood and consequence of malfunction or failure.

Predictive maintenance (PdM) is a name sometimes given to the combination of three activities: vibration analysis, thermography, and oil analysis. In fact, predictive maintenance should be viewed in a much broader sense: it is maintenance based on the expert inspection and analysis—as quantitative as possible—of a component's current and projected condition. While PdM involves more upfront effort and more expertise than PM, its rewards are significant. It is the means towards an optimized operation, one where equipment is repaired only when it must be (before failure or loss of function) and only if it has to be (avoiding costly, unnecessary, and at times harmful repairs or

overhauls). The benefits gained through Predictive Maintenance are significant: early warning of degraded conditions or failure, improvements in operating procedures, improvements in materials, designs and procurement practices, better inventory of spare parts, optimum planning for system outages, gain in staff expertise and motivation through a better understanding of operations, degradation mechanisms and the reasons behind repair decisions.

In practice, many programs start as PM's and evolve into PdM's in search of better reliability. For example, a plant may test its relief valves every three years as part of a PM program, to find out, after several test cycles, that certain valves should be inspected and tested more often, others less often, while a third group—for example smaller utility valves—should not be tested at all, but simply replaced every five years. Analyzing the data, turning data into knowledge, has permitted the plant to evolve to a PdM program, with improvements in costs and reliability. Another example is the periodic volumetric inspection of critical piping, pipelines, tanks and vessels. Base line data is collected first at fixed intervals, then as wall thickness data is analyzed in light of operating parameters, environment and corrosion history, degradation mechanisms start to be better understood, and the inspection intervals and inspection techniques are adjusted accordingly.

### 22.7.2 Inspection Checklists

When the maintenance strategy calls for the periodic inspection of piping systems, pipelines, equipment and their supports, it is advisable to use inspection checklists to guide and document the inspection. This section provides an example of visual inspection checklist for piping, vessels and tanks, and their supports. These visual inspections may be supplemented by periodic surface inspections (typically PT or MT, Chapter 16) or volumetric inspections (typically UT or RT, Chapter 16) based on the system's or component's risk: its likelihood of degradation and the consequence of the degradation on safety (public and worker), profit and the environment.

The visual inspection checklist is written in such a manner as to note "yes" when the condition is satisfactory. The inspection relies heavily on the inspectors' experience in component design, corrosion, mechanical and civil engineering. For guidance on the inspection of piping, vessels, tanks and boilers, the reader may also refer to API standards [API 510, API 570, API 572, API 573, API 574, API 580, API 581, API 653].

The analysis of inspection results to determine the fitness-for-service of piping and pipelines is addressed in Chapter 21. The analysis of degradation of piping, vessels and tanks may follow the rules of API 579 Fitness-for-Service. The analysis of the degradation of piping and equipment supports (steel or concrete structures) must be determined case by case, applying the rules of the construction codes and standards (AISC, AISI and ACI). The analysis of the degradation of liquid or gas pipelines must follow 49CFR, ASME B31G and RSTRENG. The analysis of degradation of nuclear power plant components must follow the ASME Boiler & Pressure Vessel Code Section XI.

#### *22.7.2.1 Piping and Vessels*

No distortion of piping or vessel

No uneven, undersized welds

No unusual or temporary repairs or cutouts

Flanges and gaskets appear in good condition

Flange ratings consistent with design pressure

Bolts and studs fully engaged in nuts

No visible sag

No significant scratch marks, gouges

No deformation of tank or vessel shell

No leakage or signs of leakage

No evident wall thinning or cracking

No excessive corrosion, beyond superficial rust deposit

No unusual discoloration or deposits

Equipment not dislodged from supports

Sliding or rolling supports free to move

All attachments of supports to structure are secure

No missing lagging, insulation or coating

No evidence of paint blisters, peeling or missing

Vessel stamp visible

Relief device on vessel at set pressure below vessel MAWP

Relief device within test date

Pressure gage functioning and within calibration date

No excessive vibration or unusual noises

No other concerns

#### *22.7.2.2 Supports*

Description
Location

(a) Support Components

Missing or Broken Parts [MSS-SP-58]

No missing hanger
No missing clevis, turnbuckle, clamp, bolts

No missing shield
No broken or visibly deformed components

Function

No binding
Insulation does not interfere with movement
No missing roll, shoe
Variable spring travel within range
Snubber travel within range and periodic test recorded
No excessive vibration
No excessive wear

Condition

No excessive corrosion (light surface rust only)
No missing or peeled paint
No missing or peeled coating

(b) Structural Steel

General

No visible settlement
No visible tilt or deflection

Members

No missing members

No visible repairs or cutouts
Load path is complete from equipment down to bolts

Joints (visual)

Good weld condition [AWS D1.1]
Good bolts and Rivets condition

Deterioration

Good paint condition (visual)
Negligible corrosion depth (UT if necessary)

(c) Concrete Attachments [ACI 201.1R]

Alignment of Structure (visual)

No visible settlement
No visible deflection
No visible expansion
No visible contraction

Surface Condition

Good general condition of concrete
No cracks over 2 mm wide
No leaching
No scaling (local flaking >20 mm depth)
No spalling (flake >20 mm deep)

No pop-out (breaking away holes >50 mm dia)
No sign of corrosion and chemical attack (discoloration)
No dampness or liquid stains
No exposed steel
No previous patching

Anchor Bolts

No missing bolts
No missing nuts
No deformed or extended bolts
No cracks in concrete pass through bolt
No cracks between bolt and concrete edge

No other Concern

## 22.8 PDM TECHNIQUES

Because a piping system consists of a large assembly of components with different failure modes and failure causes, there is a host of applicable Predictive Maintenance techniques. To illustrate the point, we take as an example a simple piping system, shown in Figure 22-2. The system consists of 6" pipe, from a vertical vessel to a horizontal heat exchanger, with two welded manual valves, a support and a spring hanger (line hot in service); and a 2" branch line with a threaded manual valve. The maintenance plan follows the outline of section 22.3.

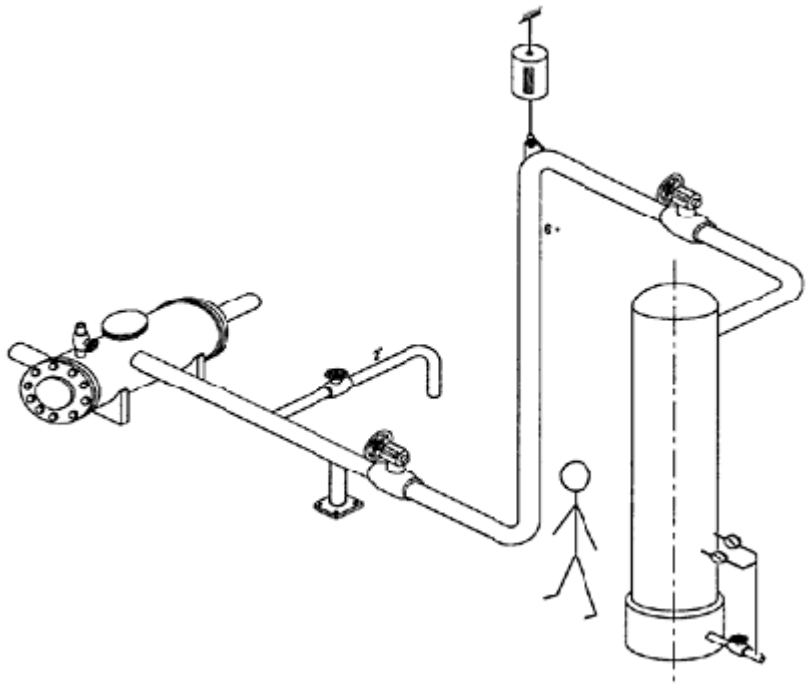

**Figure 22-2** Illustrative Example for PdM

(1) Maintenance strategy: predictive because the system is essential to operations.

(2) Component list, function, failure mode and failure cause:

(2.a) Piping 6" and 2", needs to remain leak tight. Failure mode would be loss of pressure boundary. Failure causes: corrosion or fatigue cracking.

(2.b) Manual valves, normally open, must be able to close for isolation. Failure mode would be loss of operability (hand wheel cannot be turned), loss of leak tightness if closed, loss of pressure boundary through body, bonnet, packing, joints. Failure causes: corrosion buildup, corrosion of wall, packing wear, debris at valve seat, wear of valve disk or plug.

(2.c) Vertical vessel, must remain leak tight. No overpressure. Failure mode would be loss of pressure boundary (leak) or rupture by overpressure. Failure causes: corrosion, failure of pressure relief valve to open and discharge at set pressure.

(2.d) Heat exchanger, must operate at nominal and full flow, need to maintain heat transfer. No overpressure. Failure mode: tube leak, shell and heads leak, head flange leak, overpressure. Failure causes: corrosion (thinning, cracking or plug-ging), inadequate flange gasket, bolts or assembly torque, tube vibration in cross flow, failure of pressure relief valve to open and discharge at set pressure.

(2.e) Supports, must maintain pipe in position, variable spring needs to remain within travel range. Failure mode: support fails, pipe dislodges, spring motion exceeds travel

allowance. Failure causes: corrosion, impact (such as waterhammer) or vibration, wear of support parts, external damage.

(3) Inspection locations and techniques.

(3.a) Piping and vessels: visual inspection of equipment and supports (external or internal during shutdown or by remote techniques such as borescope), surface examination (liquid penetrant or magnetic particles), volumetric (ultrasonic straight or angle beam, portable radiography with X-rays or radioisotopes, pulse eddy current for inspection through insulation), leak detection (visual, acoustic, thermography), pressure test (holding pressure with liquid test, bubble solution with pneumatic test, or tracer gas detection), eddy current of heat exchanger tubes, or acoustic emission. Ultrasonic measurement of flow rate (Doppler effect or time of flight).

A difficult decision is where to inspect. Fatal pipe and vessel ruptures have happened even though the piping and vessels were part of a periodic inspection program. The failure however occurred at a different location than the locations inspected. Industry standards have been developed to help the owner decide where to inspect piping and vessels [API 510, API 570, API 572, API 574, API 579, NBIC]. The decision remains however closely tied to understanding the specific degradation mechanism taking place in the system. In addition to locations susceptible to specific corrosion mechanisms (for example microbiologically induced corrosion in stagnant warm water), inspection locations may include [API 570, API 574]: injection points (1 ft upstream and 25 ft downstream), dead-legs, corrosion under insulation or under coating, points of pipe penetration in soil or structures, where erosion is a concern (elbows, tees, thermowells, etc.), points of audible cavitation (control valves, orifice, pump suction or discharge), vibration points, thermal fatigue (for example where the expansion analysis predicts a cyclic expansion or contraction stress within 80% of the design stress allowable), locations of potential freeze damage, mechanical joints (threaded, swaged, grooved couplings) and flange joints.

The great advantage of in-line inspection (smart pig inspection) of oil and gas pipelines is that it covers the full length of the line and avoids the hit and miss spot inspections common in the power and process industries.

(3.b) Valves: many facilities overhaul valves on a rotating schedule (preventive maintenance PM). If little is known about the valve performance in a specific process, this could be a valid starting point, but the overhaul should be used to gather knowledge on degradation mechanisms, with the intention of switching as soon as feasible from PM to PdM. Valve maintenance will usually entail, as necessary, tightening or replacement of valve packing, visual inspection of seating surfaces and leak paths, cleaning, polishing, lap as necessary, blueing test for leak tightness (blue dye permits viewing non-uniform contact between disk and seat ring), replacing seat ring, tightening body-bonnet (if screwed or flanged). Where leak tightness is critical, the valve may be periodically tested for seat leakage. The manual valve wheel should be periodically cycled open and close.

Acoustic techniques can be used to detect through-seat or pressure boundary leakage of liquid, steam or air. The acoustic signal in a reference transducer (background noise) is compared to the signal at the valve to measure a leak signal. The transducer may be hand held or mounted to the pipe and valve.

(3.c) Heat exchanger tubes: periodic visual (pulling out the tube bundle), eddy current for evidence of wall thinning, pitting or cracking, pressure or leak testing.

(3.d) Supports: on critical systems, support members and anchor bolts visually inspected, for evidence of damage. Variable spring travel should be verified by yearly reading the scale on the spring can and comparing to predicted cold (shutdown) and hot (operating) positions.

(4) Acceptance criteria for the integrity of the pressure boundary of piping, vessels, heat exchangers and valve bodies are based on fitness-for-service procedures, such as discussed in Chapter 21. Acceptance criteria for operability of manual valves are process specific (time to open or close valve, and leak tightness of valve seat). Acceptance criteria for supports are based on vendor catalogs (for variable springs, anchor bolts) and judgment.

## 22.9 RELIABILITY

There are basically three methods to gain knowledge from maintenance activities. The first method is to investigate a failure or malfunction in the field, as it happens. The advantage of this approach is that a lot of first hand information can be gathered regarding failure mode and failure cause. The shortcoming is the difficulty to generalize the findings. The second method is to qualitatively review historical maintenance records, particularly corrective maintenance, for a class of equipment over a period of time. The advantage here is to be able to see general trends, the shortcoming is that the quality of documentation may be poor (the proverbial “Was broken. Fixed it”). The third method is to statistically analyze a large number of maintenance data, to compile and trend quantitative reliability statistics.

A word of caution is in order regarding the use of reliability data. Reliability data is available from various sources [RAC, AIChE, API 581]. Following are some examples of mean failure rates that can be found in the literature, not to promote their use, but rather to discuss their limitations:

Tanks and Vessels:

Tank leakage 1E-7/hour

Vessel rupture 5E-9/hour

Heat exchanger tube leak 1E-6/hour

¼” leak in vessel or storage tank 4E-5/year

4” leak in vessel or storage tank 1E-5/year

Rupture of vessel 6E-6/year

Rupture of storage tank 2E-5/year

Pipe and Fittings:

Leak of metallic straight pipe 0.0268E-6/hour

Leak of metallic fittings 0.57E-6/hour

Flange gasket leak 1E-7/hour

Plugged strainer 3E-6/hour

¼" leak in ¾" pipe 1E-5/year-ft

¼" leak in 6" pipe 4E-7/year-ft

¼" leak in pipe larger than 16" 6E-8/year-ft

Rupture of ¾" pipe 3E-7/year-ft

Rupture 6" pipe 8E-8/year-ft

Rupture pipe larger than 16" 1E-8/year-ft

Valves:

Solenoid valve fails open 3E-6/hour

Solenoid valve fails closed 3E-6/hour

Solenoid fails to respond 2.83E-3/demand

Motor operator fails to respond 5.58E-3/demand

Air operator fails to respond 2.2E-3/demand

Safety relief fails to open 3E-3/day

Safety relief fails to reclose 3E-3/day

Check valve leaks through 1E-6/hour

Check valve leaks through 3.18E-6/hour

Pumps and Compressors:

Compressor fails 1430E-6/hour

Pump motor fails to start 1E-2/demand

Centrifugal pump motor fails to start 18.6E-3/demand

Centrifugal pump motor fails to run at rated speed 920E-6/hour

Centrifugal pump motor fails while running 292E-6/hour

Pump overspeed 3E-5/hour

Pipe leak 3E-9/hour-foot

Pipe rupture 1E-10/hour-foot

This type of reliability data has to be used with the greatest care, as it raises several questions. First, in some cases, equipment is grouped in such broad classes as to be of questionable value. For example, what is "metallic straight pipe"? Is an underground cast iron bell and spigot drain pipe installed forty years ago grouped with a new welded stainless steel pipe in a process building, designed and fabricated to ASME B31.3, joined by certified welders, inspected by 100% radiography?

Second, little attention is paid to the maintenance pedigree of the equipment. For example, what is "centrifugal pump motor fails to start"? Is this a new pump, or a pump hat has been idle for 10 years with no maintenance and overhaul?

Third, the failure mode is, most of the time, not defined. For example, note the same likelihood of 3 $10^{-3}$/day for "Safety relief fails to open" and "Safety relief fails to reclose". Technicians who have spent time in a safety-relief valve shop will undoubtedly be surprised by these figures. While some valves that have been in service may fail to open at the exact set pressure, they will in the vast majority of cases open at 10% to 20% above the set pressure. Is popping open 10% to 20% above set point classified as "safety relief fails to open"? On the contrary, it is not uncommon for safety valves in steam service to not fully re-close after discharge, causing a continuous small. Is this small leak what is meant by "safety valve fails to reclose"?

Fourth, the most complete reliability tables provide not only a mean failure value, but also a statistical minimum and maximum. This is wise, but in many cases, there is a factor of 10 to 100 between the mean and the minimum failure rates.

In summary, reliability data, based on the study of large quantities of failures, has many limitations, and must be applied with the greatest care. Plant, system or component specific data are more useful. A much better use of reliability data is to correct the generic failure values by quantitative correction factors specific to the system and the service. This is the approach followed by the Risk Based Inspection technique in API 581. Correction factors applied to the generic failure rates should account for several variables: (1) damage mechanisms (potential for corrosion and other degradation mechanisms at play, accuracy of predicted corrosion rate), (2) quality and frequency of field inspections, (3) system maintenance condition (vibration monitoring, movement checks, etc.), (4) natural forces (weather, wind, soil stability, seismic activity, etc.), (5) complexity of piping system (layout, number and type of non-welded fittings, number and type of valves, support condition, etc.), (6) quality of initial construction, (7) quality of repairs, (8) operational history (reliability, use factor and downtime, etc.), (9) complexity and stability of process (operating temperature and pressure, steady state or transient operation, likelihood of exothermic reaction, explosion or fire, etc.), (10) condition of process fluid (liquid, flashing liquid, gas, vapor, single or two-phase, etc.), (11) outside factors (encroachment, third party damage, etc.).

## 22.10 MAINTENANCE AND THE CONSTRUCTION CODES

The oil pipeline, gas pipeline and nuclear power piping codes have explicit maintenance requirements [ASME B31.4, ASME B31.8, ASME XI, ASME OM]. This is understandable since these functions are regulated by state and federal requirements [10CFR50, 49CFR] and the codes have a focused scope. In contrast, ASME B31.3, the code for chemical process systems has such a broad scope that it chose not to develop explicit maintenance rules. Instead, specific industries have developed their own guidelines or codes, such as API 570 for refining and petrochemical plants.

In all cases, of the five areas covered by construction codes (Chapter 2) the first three apply to maintenance as-is, with few exceptions: materials (ASTM listed materials used within their code temperature limits), design (sizing, layout and supports) and fabrication (welding and joining). One of the few exceptions to applying the construction code to maintenance: the difficulty encountered in some cases in heat-treating a maintenance or repair weld. This difficulty may be circumvented by pre-qualified local post-weld heat

treatment (bull's eye heating) or temper bead welding techniques where a weld bead heat treats the previous one [NBIC].

The other two areas covered by construction codes, examination and leak testing, can not always be applied as-is, and must therefore be adapted to maintenance. Examinations of non-welded joints (flanges, swage fittings, threaded joints, etc.) are usually performed by a certified mechanic, with independent inspection being limited to critical lines. The need for leak testing, in particular hydrostatic testing of tie-ins between a new subassembly and the existing system should be decided on a case basis.

When it is not practical to isolate an existing line to hydrotest the tie-ins of a maintenance modification, then the tie-in joints may be inspected by a comprehensive in-process examination (looking over during the joining process) and non-destructive examination instead of hydrotest: UT or RT for butt welds, PT or MT for fillet welds, and independent quality control for the re-making of mechanical joints such as flange connections.

## 22.11 ELEMENTS OF FAILURE ANALYSIS

We have seen that corrective maintenance (run to failure) is a viable maintenance strategy when the failure is no more than a nuisance. To the contrary, when the failure could affect profit (production), safety (worker or public) or the environment (stewardship, regulation), then preventive (PM) or predictive (PdM) maintenance are in order. If despite of PM or PdM a failure does occur, it is often necessary to analyze the failure, determine its root causes and make a repair that avoids its recurrence. This is achieved through failure analysis. In the case of piping systems and pipelines, failure analysis relies heavily on the seven fundamental aspects of competent engineering (Chapter 2): materials (metallurgy and corrosion), design (fluid flow and stress), fabrication (joining and welding), examination (non-destructive and destructive testing), leak testing (pressure testing and leak detection), operation (normal or abnormal operating conditions) and maintenance (prior performance).

The failure analysis process relies on the steps shown in Figure 22-3 and described in the following sections.

### 22.11.1 Data Collection

Retrieval of initial design and construction data applicable to the failed system, including, where available, material specifications, material certificates, material test reports, vendor files, system descriptions, design calculations and stress analyses, fabrication and welding records, non-destructive examination construction records, testing records. Interview of designer, metallurgist, quality control personnel. Retrieval of operating log to establish operating conditions at time of failure (flow rate, pressure, temperature) and operating history, with particular attention to abnormal events. Interview operators and system engineer. Maintenance history, prior repairs, equipment or system modifications, overhauls, replacements, reports of abnormal conditions. Interview of maintenance mechanics.

### 22.11.2 Visual Examination, Macrofractography and NDE

Investigation of the failed component in situ (in failed position) with additional lighting and hand-held magnifying glass. Recording failed position and condition, surface appearance, dimensional changes, discolorations and corrosion, paint and coating, supports, environment anomalies. Photographs. Care should be taken not to alter the failure surface by touch.

The visual examination of the fracture surface with magnification, is called “macrofractography”, where the prefix “macro” is used in contrast with “micro” for examination using microscopes. It can be recorded on a macrofractograph (photograph with magnification).

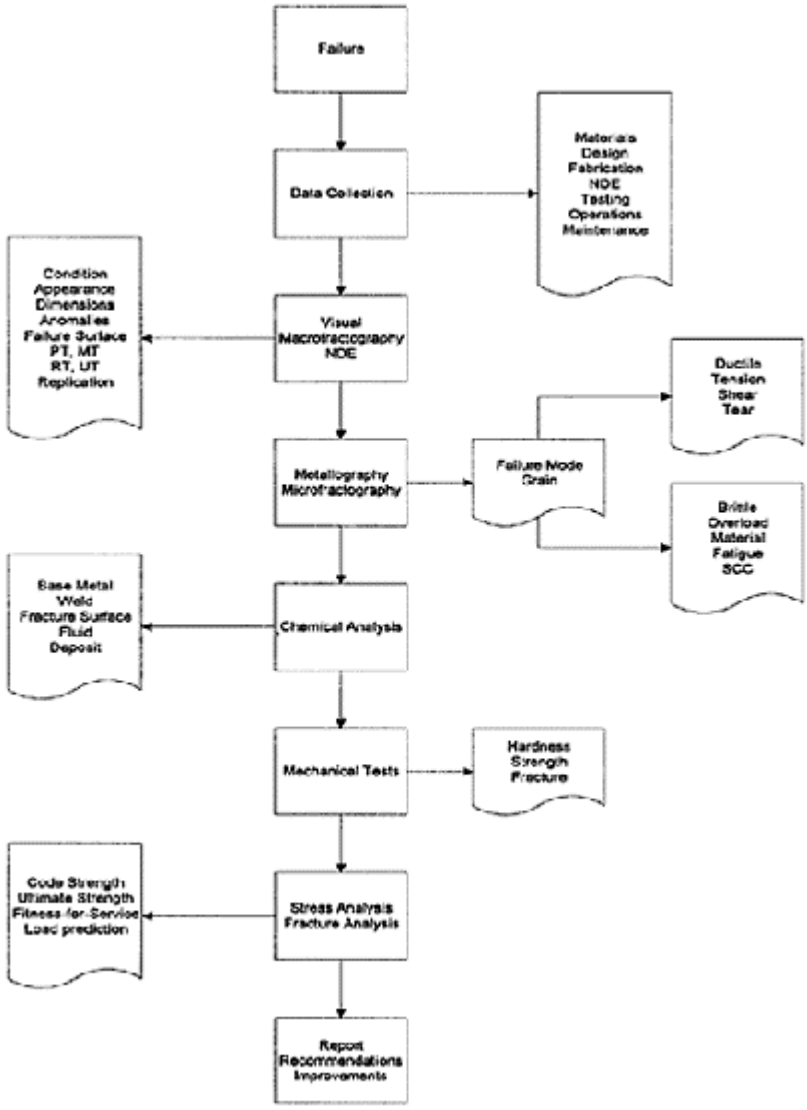


**Figure 22-3** Failure Analysis Logic

The visual examination of the fracture surface can provide valuable information: a ductile fracture has a fibrous, dimpled appearance. A brittle fracture has a featureless, shiny crystalline appearance. Brittle fracture in a thick-wall component has a chevron pattern (V-shaped marks on the fracture surface, with the tip of the V's pointing to the crack origin. A fatigue fracture has a beach mark appearance with a brittle fracture appearance between marks. Most often, the fracture surface has a mixture of ductile and brittle features, but one fracture appearance will dominate.

The failed component can also be examined by non-destructive techniques such as UT, MT, RT, UT (Chapter 16) for evidence of cracking, wall thinning, construction defects or degradation in-service.

### 22.11.3 Metallography and Microfractography

The common techniques for the microscopic analysis of a fracture surface are:

(1) Standard light optical microscopy (LOM) of a prepared specimen
(2) Scanning electron microscopy (SEM) of the as-is specimen.
(3) Transmission electron microscopy (TEM) of a replica.

Metallography provides valuable information on the grain structure and flaws in the base material, the weld and the fracture surface. It indicates whether the fracture was ductile or brittle. Ductile fracture surfaces are dimpled. Ductile fracture is caused by (i) overstress by shear or tension in a thin ductile material (plane stress, i.e. stress constant through the wall thickness), or (ii) overstress at the surface of a thick material, or (iii) overstress by shear. An example of ductile fracture is the standard tensile test illustrated in Figure 22-4(a). The specimen first develops microvoids that coalesce (combine) and eventually fail the specimen by tension in the middle (cup) followed by shear at 45° on the sides (cone), Figure 22-4(b). The dimples are equiaxed (roughly same dimple size in all three directions), which is caused by tension overload, Figures 22-4(c) and 22-5(a).

In ductile tensile fracture, the fracture surface contains three zones: (1) a flat fibrous core (the original dislocation) in a plane perpendicular to the applied tension (Figure 22-4(c)), (2) a fibrous radial zone with striations emanating from the core (this radial zone may be missing in thin sections), and (3) a shear lip at the periphery where final separation occurred, at 45° from the core plane (Figure 22-4(d)). In shear fracture, the dimples may be elongated in opposite directions on the two fracture lips (Figure 22-5(b)), or elongated in the same direction, which is caused by tensile tearing (zipper effect as in Figures 22-4(d) and 22-5(c)).

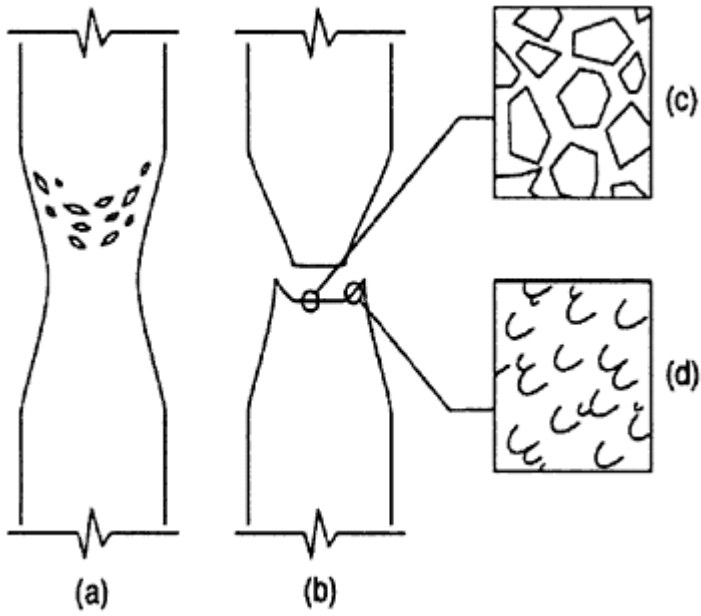


**Figure 22-4** Shear and Tension Overload

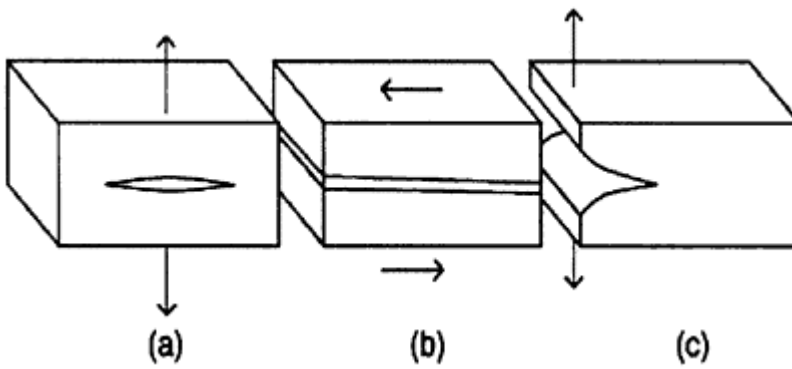


**Figure 22-5** Tension, Shear and Tearing Failures

Brittle fracture surfaces are crystalline, featureless. Brittle fracture is caused by (i) fracture of a brittle material (material with low fracture toughness at operating temperature), which could happen without apparent overstress, while the material is still elastic, or (ii) overstress in a thick material (plane strain, i.e., strain constant near the center of the material which is constrained by the surrounding thick material), or (iii) fatigue crack propagation, or (iv) crack propagation by stress corrosion cracking.

Metallography is also used to study the shape of cracks to determine whether they are intergranular or transgranular (which points to the corrosion or failure mechanism), their branching direction (which points to their direction of propagation since the cracks tend to propagate in the direction of increasing branching), the possibility of multiple initiation sites (cracks tend to intersect at right angles).

### 22.11.4 Chemical Analysis

Chemical analysis can be applied to the base metal and weld to confirm their composition (alloy analyzer for positive material identification), to the fracture surface (Auger electron spectroscopy AES) to determine the nature of corrosion residue, and to the fluid to determine its composition and the presence of impurities. Chemical analysis of metals can be by a wet technique (solution), energy dissipative X-rays (EDX for simultaneous analysis of a spectrum of elements), or wave length dispersive spectrometers (WDS for analysis of one element at a time). These methods can be augmented by accelerated corrosion tests, such as stress corrosion cracking tests or weight loss corrosion tests.

### 22.11.5 Mechanical Tests

Mechanical tests typically consist of (a) microhardness tests to determine unusually hard and brittle spots in the base metal, weld or heat affected zone, (b) strength tests (tensile, yield .and elongation at rupture) to determine conformance to the minimum requirements of the material specification, and (c) toughness tests (Charpy, drop weight tear test, fracture toughness) to determine the ductility of the material.

The difficulty with most mechanical tests is that they are destructive tests. But there are specialized techniques to shave off a small scoop of metal from the surface and conduct tension tests on the small specimen.

### 22.11.6 Stress and Fracture Analysis

The material, dimensional, fracture, and operating facts assembled so far are used to model and analyze the failed component. This step is necessary to confirm the postulated failure mode or to choose among different possible failure modes. For example, if failure appears to be due to overpressure in a ductile material, than a classical stress analysis can be conducted to verify that the stresses in the component at the failure pressure have reached the ultimate strength of the material.

If failure is attributed to fatigue, than a fatigue usage factor can be calculated. If failure is attributed to sudden crack propagation, than a fracture analysis can be used to confirm the plausibility of crack induced fracture.

### 22.11.7 Improvements

The lessons learned through failures must be used to avoid recurrence, to improve performance by improving the engineer’s understanding of the seven fundamentals: materials, design, fabrication, examination, testing, operations and maintenance. For the sake of profits and safety this knowledge must be clearly communicated through the

various departments of a company, and reflected in its engineering, operations and maintenance procedures. Unique knowledge should be communicated to the whole industry, through publications, conferences and technical papers. “The more sand has escaped from the hour glass of our life, the clearer we should see through it”, Jean Paul Sartre.

## 22.12 REFERENCES

10 CFR Energy, Code of Federal regulations, Part 50 Domestic Licensing of Production and Utilization Facilities, Washington, DC.
49 CFR, Code of Federal Regulations, Title 49, Transportation, Part 192 Transportation of Natural and Other Gas by Pipeline, Subpart M—Maintenance, Washington, D.C.
ACI 201.1R, Guide for Making a Condition Survey of Concrete in Service, American Concrete Institute, Detroit, MI.
AIChE, Guidelines for Design Solutions for Process Equipment Failures, American Institute of Chemical Engineers, New York.
AIChE, Guidelines for Process Equipment Reliability Data, , American Institute of Chemical Engineers, New York.
AISC, Manual of Steel Construction, American Institute of Steel Construction, Chicago, IL.
API 510, Pressure Vessel Inspection Code: Maintenance, Inspection, Rating, Repair, and Alteration, American Petroleum Institute, Washington, D.C.
API 570, Piping Inspection Code: Inspection, Repair, Alterations, and Rerating of InService Piping Systems, American Petroleum Institute, Washington, D.C.
API 572, Inspection of Pressure Vessels, American Petroleum Institute, Washington, D.C.
API 573, Inspection of Fired Boilers and Heaters, American Petroleum Institute, Washington DC.
API 574, Inspection of Piping, Tubing, Valves, and Fittings, American Petroleum Institute, Washington, D.C.
API 579, Fitness for Service, American Petroleum Institute, Washington DC.
API 580, Risk-Based Inspection, American Petroleum Institute, Washington, D.C.
API 581, Risk Based Inspection, Base Resource Document, American Petroleum Institute, Washington, D.C.
API 653, Tank Inspection, Repair, Alteration, and Reconstruction Code, American Petroleum Institute, Washington DC.
API RP 750, Management of Process Hazards, Process Safety Management, American Petroleum Institute, Washington, D.C.
ASME B31.1 Power Piping, American Society of Mechanical Engineers, New York.
ASME B31.3 Process Piping, American Society of Mechanical Engineers, New York.
ASME B31.4 Liquid Petroleum Transportation Piping, American Society of Mechanical Engineers, New York.
ASME B31.8 Gas Transmission and Distribution Piping, American Society of Mechanical Engineers, New York.
ASME Boiler & Pressure Vessel Code, Section XI In-Service Inspection, American Society of Mechanical Engineers, New York.
ASME O&M, Boiler & Pressure Vessel Code, Section III, Operation and Maintenance, American Society of Mechanical Engineers, New York.
AWS D1.1, American Welding Society Code D1.1, Steel, Miami, FL.
Browning, J.B., Crisis Response: Inside Stories on Managing Under Siege, ed. J.A.Gottschalk, Visible Ink Press, Gale Research, Detroit, MI, 1993.
EPRI, Predictive Maintenance Primer, NP-7205, April 1991, Electric Power Research Institute, Palo Alto, CA.

Kalelkar, A.S., Investigation of Large-Magnitude Incidents: Bhopal as a Case Study, The Institution of Chemical Engineers, Conference on Preventing Major Chemical Accidents, London, England, 1988.
Kletz, T., What Went Wrong, Case Histories of Process Plant Disasters, Gulf Publishing Company, Houston, 1994.
Lees, F.P., Loss Prevention in the Process Industries, Butterworth, Heinemann, Reed Educational and Professional Publishing Ltd., 1996.
Mobley, R.K., Maintenance Fundamentals, Plant Engineering, Newnes, Boston, MA.
Mobley, R.K., Root Cause Analysis, Plant Engineering, Newnes, Boston, MA.
Moubray, J., Reliability Centered Maintenance, Industrial Press, New York.
NBIC, National Board Inspection Code, ANSI/NB-23, The National Board of Boiler and Pressure Vessel Inspectors, Columbus, OH.
NTIAC, Nondestructive Evaluation for Condition Based Maintenance, Nondestructive Testing Information Analysis Center, Austin, TX.
NTIAC, NDE Methods for Characterization of Corrosion, Nondestructive Testing Information Analysis Center, Austin, TX.
OSHA 29 CFR 1910.119, U.S. Department of Labor, Occupational Safety & health Administration, Washington, D.C.
Patton, J.D. Jr., Preventive Maintenance, Instrument Society of America, Research Triangle, NC.
RAC, Reliability Analysis Center, Failure Mode/Mechanism Distributions, U.S. Department of Defense, Washington, D.C.
Rockwell, 2002 Maintenance and Reliability Practices Survey, Rockwell Automation, Milwaukee, WI.
Smith, A.M., Reliability-Centered Maintenance, McGraw-Hill, New York.
Timmins, P.F., Solutions to Equipment Failures, American Society of Metals, Materials Park, OH.

# 23
# Repair Techniques

## 23.1 REPAIR STRATEGY

The first consideration when planning a repair is understanding the root cause of the problem that led to the repair. On the basis of this understanding, a repair strategy is established, which addresses six key decisions: repair technique (discussed later in this chapter), materials (chemistry, mechanical properties, corrosion resistance), design (flow and system logic, sizing), fabrication (joining, welding, heat treatment), examination (NDE of the failed component and the repair), testing (hydrotest, pneumatic test, leak test).

A repair plan is then developed and implemented. The repaired component remains in service for as long as its design life justifies it. In other words, no repair is temporary or permanent; instead, each repair will last as long as its weakest component. It is therefore essential, with each repair, to define a useful life (or design life) of the repair, and plan periodic inspections to confirm this useful life.

## 23.2 REPLACEMENT

The most common practice for repairing pipe sections or components is, of course, to replace them. The new component or subassembly should fully comply with the construction code, preferably a recent edition of that code. In some cases, regulators or owners will impose the original code edition (the so-called code of record) for materials, design or fabrication of replacement components. This is seldom technically justified and is impractical (where to find a spool fabricated to a 1967 ASTM material specification?). It often leads to costly yet sterile comparisons of changes between code editions, to justify the obvious: the use of the latest edition of codes and standards is preferable to the use of an obsolete one.

An important consideration in planning the removal of the degraded section of pipe or component is to carefully and clearly specify lockout, tagout, depressurization, draining and venting the pipe before cutting or opening the system. The rules of OSHA 29 CFR 1910.147 [OSHA] may apply to such activities. The importance of proper isolation should not be discovered the hard way, as was the case on the morning of Sunday, October 22, 1989, when maintenance workers started to clear three of the six settling legs of a polyethylene reactor near Houston, Texas [OSHA, Lees]. The activity started by first closing the settling leg's air operated isolation valve (Figure 23-1(B)), and then dismantling and cleaning the leg (Figure 23-1(E)), while the system was still at pressure. Corporate procedures required double isolation, but at this facility the settling legs were

isolated by a single valve (Figure 23-1(B)). Matters are further complicated by the fact that the isolation valve is pneumatically operated to either close or open.

It appears that, as the dismantling of the second leg started on Monday morning, the air hoses to the pneumatic valve had been crisscrossed and what should have been air-to-close turned out to be air-to-open (Figure 32-1(C)). The valve was believed to be closed, but was actually open. As the leg was partially dismantled, a mixture of hydrocarbon vapors leaked through and was ignited, followed by two more explosions, leaving 22 dead, and 130 injured.

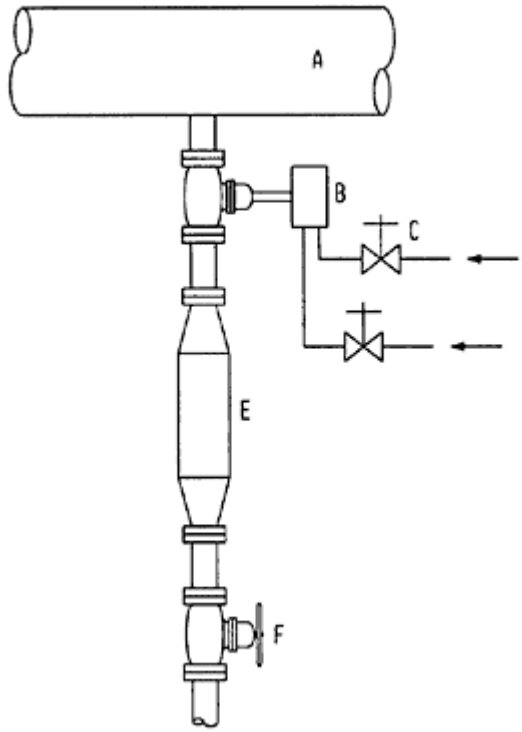


**Figure 23-1** Settling Leg Arrangement

When pressure equipment is readied to be open, the operator should not rely on pressure gage reading alone to confirm that the system is at atmospheric pressure. Confirmation must also rely on open venting to atmosphere and draining [HSE], as evidenced in the following accident. At 2:30 AM on the morning of March 13, 2001, two explosions rocked a plastic manufacturing plant in Augusta, Georgia, killing three workers [USCS]. Twelve hours earlier, an attempt had been made to start a production unit. The attempt was aborted, but a quantity of hot molten plastic had been diverted to a horizontal 750-gallon waste collection vessel, referred to as “catch tank”, Figure 23-2. The hot plastic proceeded to react, foam and expand, eventually forcing its way and solidifying in the relief valve inlet pipe, Figure 23-2(A).

When several hours later, during the night shift, three workers attempted to open the catch tank cover, they first checked the pressure gage, Figure 23-2(B). But the pressure gage was on the vent pipe, and—unknown to the operators—the vent was sealed from the vessel body by the plug of solidified plastic. Reading no pressure, the workers proceeded to remove the bolts on the vessel cover. With 22 of the 44 bolts removed, the 48" diameter, 1,750-lb cover suddenly blew off, spewing hot plastic and killing the three workers. The accident was further compounded when the cover, flying 14 ft, ruptured a hot oil line, forming a flammable vapor cloud that ignited.

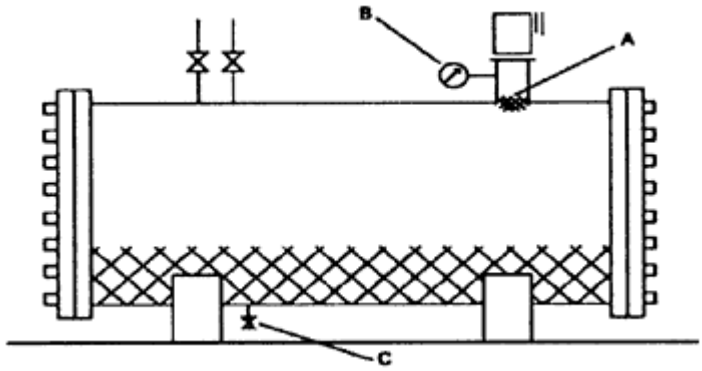


**Figure 23-2** Catch Tank with Plugged Relief Vent Pipe

The plant had a written lockout/tagout procedure, but it relied on the use of drain valves that were often plugged with solidified plastic, Figure 23-2(C). So operators and maintenance technicians relied instead on the pressure gauge in the relief valve pipe to check the vessel pressure.

To drain a pipeline, the system must be shutdown and isolated at block valves (double isolation may be required with hazardous contents). Where this is not feasible, the line can be blocked using inflatable inserts (typically tight to 15 psid), Figure 23-3, or discs with elastomeric seal (to 1500 psid) [COB, Sealfast].

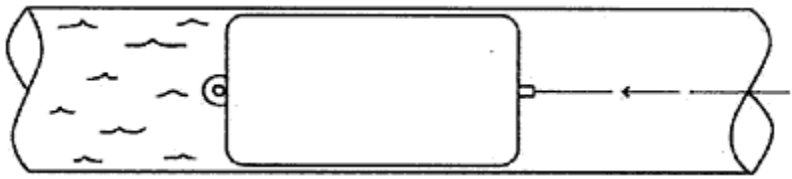

**Figure 23-3** Inflatable Isolation Plug

The line can also be isolated by tapping (referred to as hot tap in the oil and gas industry and wet tap in power, process and waterworks) [API 2201, Cantwell, Goodfellow, Hydrastop, IPSCO, Jandu, TDW, Team, Tubb]. Referring to Figure 23-4, following are

the steps for hot tapping: The full encirclement split sleeve (1) is welded to the pipe. The hot tap gate valve (2) is bolted to the sleeve. The hot tap machine (3) is bolted to the valve. The rotating cutter (4) is lowered against the pipe wall and cuts the pipe. The cutter and cut disc of pipe wall are retracted into the hot tap machine and the gate (5) is closed. The hot tap machine is replaced by the line-stop machine (6). The thick rubber gasket or plug (7) of the line-stop machine is lowered into the pipe at position (8). The flow is now interrupted. The line is drained and vented on one side of the tap and repairs made on the drained side. When repairs are completed and flow has to resume, the plug (7) is retracted. The hot tap valve gate (5) is closed. The line-stop machine is replaced by a plug machine (9). A disc plug (10) is inserted into the sleeve nozzle (1) in position (11). The hot tap valve (2) is removed, and a blind flange (12) is bolted atop of the sleeve nozzle.

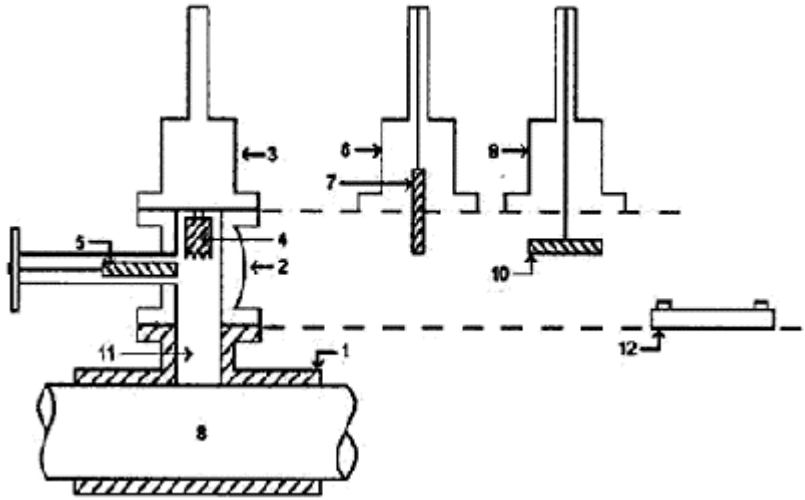


**Figure 23-4** Hot Tap

When replacing a spool as shown in white in Figure 23-5, the new component or subassembly should fully comply with the original code (joining process, NDE, hydrotest). For example, joints B, C and D on the new spool of Figure 23-5 should be hydrotested. It is however difficult in many cases to hydrotest the tie-in joints between the new and old pipe, joints A and E in Figure 23-5. In this case, NDE may be used as an alternative to hydrotest of the tie-ins to the existing system [API 570]. Non-hydrotested butt welded tie-ins may be examined by RT or UT; this is the case for weld A in Figure 23-5. Fillet welded tie-ins (branch connections, socket welds) may be examined by PT or MT (including the first pass in critical service). Mechanical joint tie-ins (flange assemblies, threaded joints, swage fittings, etc.) may be inspected in process (while being assembled), this is the case for flange joint E in Figure 23-5. These alternatives are not meant to apply to joints between new parts, but only to the tie-ins where hydrotest is impractical.

Post-weld heat treatment of the tie-ins should be performed if required by the construction code, unless the alternatives of NBIC are followed [NBIC]. As a matter of

good practice, new welds should not be closer than 6" from existing welds, and the replacement pipe should be no shorter than 12" [API 2200] or one-fourth of the pipe diameter.

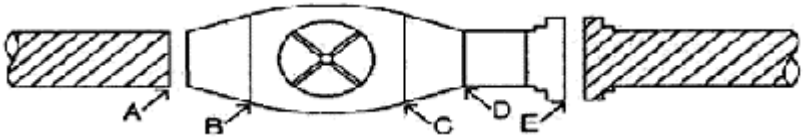


**Figure 23-5** Tie-Ins (A and E) and New Joints (B, C and D)

### 23.3 GRINDING OUT DEFECTS

When base metal or welds contain fabrication or service-induced defects (weld flaws, arc burns, gouges or cracks), these defects can be excavated by grinding, under certain conditions. The ground surface is then examined by liquid penetrant or magnetic particle to confirm that the defect has been removed. The ground out area may be left as-is, without weld deposit, if the remaining wall thickness exceeds the minimum required by code, the ground profile is smooth (3:1 profile or smoother) and the remaining wall is sound. On pipelines, a safe length and depth of grinding are given by [Rosenfeld]

$$L \le 1.12\sqrt{Dt\left[\left(\frac{d/t}{1.1(d/t)-0.11}\right)^2 - 1\right]}$$

L=length of grinding, in
D=pipe diameter, in
t=nominal wall thickness, in
d=depth of grinding, in

If the remaining smooth depression is below the minimum wall, the condition can be analyzed for fitness-for-service as a local thin area (Chapter 21).

Where the grinding process may alter the metallurgy of the pipe or weld (for example on heat treated steels or high strength line pipe), the area should be etched and examined.

Note that many pipe material specifications for new base material allow grinding repairs under certain conditions. For example, in the case of common ASTM A 53 carbon steel, the pipe fabrication specification permits unfilled repairs if a "smooth curved surface" is maintained and the wall thickness is not decreased below specified minimum. ASTM A 530 requires the defects to be "thoroughly chipped or ground out" then welded with a suitable filler metal.

The cavity is ground smooth with beveled sides and edges rounded to facilitate weld deposition. Weld material is deposited in the cavity. The sequence of weld bead overlay is qualified to assure that the base metal and weld are sound and properly tempered. In

certain cases, the repair root pass is examined by liquid penetrant. The completed weld is smoothed and polished, then inspected by PT, UT or RT.

It is highly recommended to shutdown the line before grinding, but when it is inevitable to grind in-service, the system pressure should be reduced by at least 20% to reduce risk of pipe rupture or leak during the grinding operation. In-service welding repair of piping must avoid burn-through, excessive residual stresses and reduced toughness of the weld repair area. These effects are heightened when welding in-service because the flow of fluid in the pipe tends to cool the weld more rapidly. This explains why it is necessary to thoroughly qualify the inservice weld repair procedure and, in certain cases, to limit the procedure to pipe with sufficient wall thickness (Chapter 15). If post-weld heat treatment is required, it may be accomplished after repair or a qualified temper-bead technique may be used [ASME XI, API-510, NBIC].

In nuclear power plant applications, ASME B&PV Section XI, IWA-4700 would exempt hydrostatic testing of repairs that did not penetrate the wall, as well as repairs on 1" and smaller piping and valves [ASME XI]. ASME B31.3 permits the owner to waive the re-hydrotest following "minor repairs" when sound construction practice is followed [ASME B31.3].

## 23.4 WELD OVERLAY

A weld overlay, also referred to as direct weld deposit, consists in depositing weld reinforcement on the surface of a pipe as illustrated in Figure 23-6 [NRC 88–01, ASME N-432]. The weld overlay technique is used in the power, process and pipeline industries to deposit weld reinforcement on the outer wall of a locally corroded or eroded section of pipe. Weld overlay is not recommended to repair cracks, but rather as reinforcement for locally thinned pipe.

Several considerations apply in the selection and application of weld overlay repairs: the compatibility of the weld with the base pipe metal, the sequence of weld bead deposition, the number of passes which can be applied, and the necessity to minimize residual stresses, particularly tensile stresses which can affect crack formation and propagation.

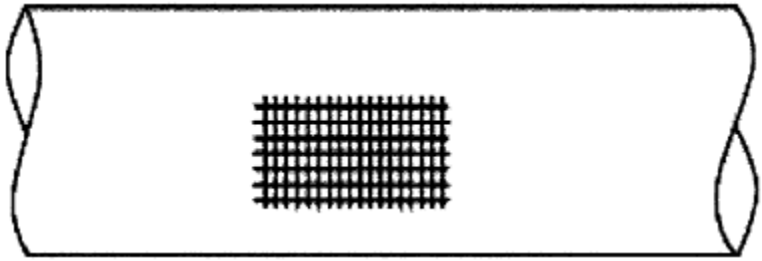

**Figure 23-6** Weld Overlay

The structural integrity of weld overlays over corroded pipe may be qualified by burst test of a mock-up [ASME N-561, N-562]. In most cases, a first bead is deposited along

the perimeter of the area to be repaired. This is followed by two perpendicular layers of weld deposit to achieve sufficient reinforcement.

For gas pipelines, ASME B31.8 section 851.43 permits the repair of small corroded areas by weld deposit with low-hydrogen electrodes [B31.8].

Weld overlay may be performed in service if (1) the technique is qualified under identical conditions (heat input, flow rate, vapor formation, etc.); (2) the fluid is non-flammable; and (3) the remaining wall is sufficient to avoid burnthrough.

If weld overlay is performed while a pipeline is in service, it is necessary to qualify a low-hydrogen welding process that prevents underbead cracking and burn-through. A temper bead technique may be necessary on thicker pipe (t>0.125") whereas each weld pass tempers the previous one [Edison].

## 23.5 FULL ENCIRCLEMENT SLEEVE

A degraded pipe can be repaired with a full encirclement split sleeve, Figure 23-7, with or without the two circumferential fillet welds to seal the ends of the sleeve. In the pipeline industry, a sleeve without the two end closure welds is referred to as type A, and a sleeve with the two end closure welds is referred to as type B.

A type A sleeve (no end closure welds) serves to reinforce a locally corroded area and compress the pipe wall to avoid the formation of an outward bulge that precedes failure by overpressure burst or pressure cycling fatigue in ductile material.

With a type B sleeve (with the two end closure welds) the sleeve serves the same role as the non-welded sleeve but also provides leak tight containment in case of failure of the pipe. The circumferential weld could also be sized to contain the thrust force in case of full separation of the degraded pipe. The repair of pipelines using welded split sleeves is recognized as an acceptable repair technique [49 CFR 192,49 CFR 195, API 570, API 2200, ASME B31.4, ASME B31.8].

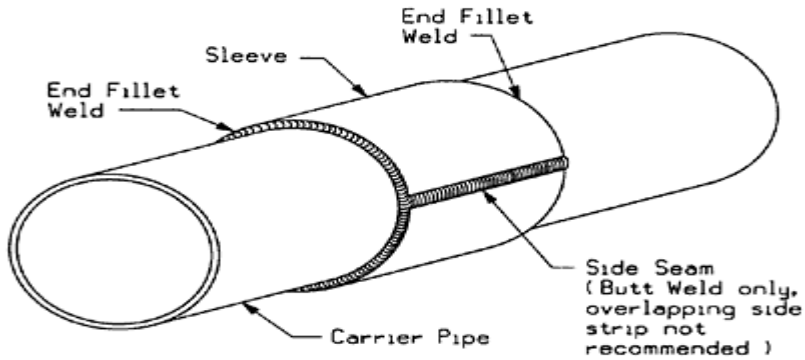


**Figure 23-7** Full Encirclement Split Sleeve, with Circumferential Weld

Welded steel sleeves can be used for internal or external local wall thinning (pitting or broader metal loss). If the corrosion is internal, it will continue to progress to eventually leak through.'

Type A sleeves (no end closure welds) should be as thick as the pipe. They are useful if they place the pipe wall in compression by means of a very tight fit between the sleeve and the pipe, or by heating the sleeve prior to welding the lon-gitudinal seams. A hardenable filler material should be used to fill and bridge any discontinuity under the sleeve.

Tests have shown the benefits of pre-heating the sleeve before encircling the pipe to achieve a tight shrink-fit and reduce tensile stresses. In a recent study, with the hoop stress in the pipe away from the sleeve at 72% yield, the hoop stress in the sleeve was at 35% yield, while the pipe under sleeve was put in slight compression with a compressive hoop stress of 0.8% yield. This result is significant when compared to a sleeve installed in service, which is reported to pick-up at most 10% to 20% of pipe hoop stress, no matter how well fitted to the pipe [Brongers].

The ends of type A sleeve should be sealed to avoid corrosion in the pipe-to-sleeve annulus.

Type B sleeves are used to repair a pipe crack; the crack must be shown to be stable and not able to propagate beyond the sleeve.

In principle, the hoop stress under a type B sleeve may be relieved by hot tapping through the sleeve and pressurizing the annular space between the pipe and sleeve, but this poses several practical problems:

(1) In liquid service, the sleeve-to-pipe annulus may be so tight, if the sleeve was properly installed, as to not allow liquid to evenly fill the annulus.

(2) The pressure of fluid in the annulus (whether by hot tapping or because the pipe has corroded through and leaked) will vary with line pressure and cause fatigue of the sleeve closure weld. Resistance to fatigue makes full penetration longitudinal seams in sleeves preferable to side strips, as noted in Figure 23-7.

(3) If the leaking fluid fills the sleeve-to-pipe annulus, it will be necessary to size the sleeve and the fillet weld to contain the leak pressure. API 570 and ASME B31.8 would require the patch to have a strength "equivalent to a reinforced opening", and the capacity to absorb "membrane strain" in accordance with the principles of the design code.

(4) If the leaking fluid fills the sleeve-to-pipe annulus, it will be necessary to take into consideration the future crevice corrosion in the annulus.

In the nuclear power industry, the full-encirclement sleeve is an option for temporary repair of safety related piping provided the full-encirclement sleeve and its welds have to be capable to fully react the loads resulting from a fully severed pipe (guillotine break).

## 23.6 FILLET WELDED PATCH

The repair of local flaws may be accomplished by fillet welding a circular or square steel patch to the pipe outer diameter, Figure 23-8. Because of it simplicity, this is a common repair technique [API 510, API 570] except for pipelines where patches are not used if the hoop stress exceeds 20% of yield. The patch is of similar or higher-grade material than the pipe, has a thickness comparable to the pipe so that the membrane stresses in the

patch and in the welds do not exceed the design code allowable, and has rounded corners to limit local stresses.

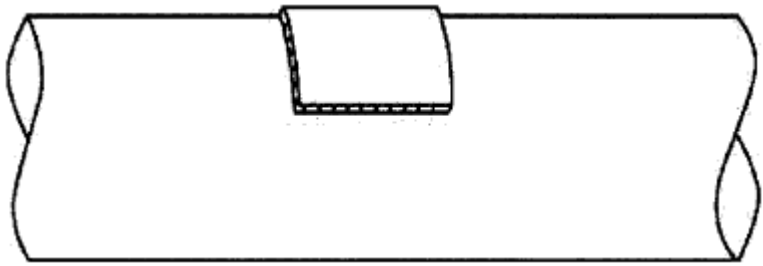

**Figure 23-8** Fillet Welded Patch

API 570 limits the use of patch repairs to pipe base material with a yield strength not more than 40 ksi. Both ASME B31.4 and B31.8 prohibit the use of welded patches on high strength pipe. ASME B31.4 limits the material to API 5L Grade X42 and B31.8 to material with a maximum specified yield of 40 ksi.

For hazardous liquid pipelines, ASME B31.4 limits the patch length to 6" and limits its use to pipe sizes 12" and less.

A variation of the fillet-welded patch is to repair a small pitting leak by fillet welding a small bore fitting (NPS 2 or smaller) to cover the leak.

## 23.7 FLUSH WELDED PATCH

The damaged section is cut out of the tank, vessel or pipe and replaced by a section of the same size, butt welded flush to the existing pipe, Figure 23-9.

This technique is mostly used for the repair of pressure vessels and tanks, and more rarely for piping where the smaller diameter makes butt-welding a patch more difficult. API 570 permits flush patch inserts, welded by groove welds to the pipe and volumetrically examined. The National Board Inspection Code [NBIC] section RD-2060 addresses full penetration patches, rolled or pressed to the proper curvature. If the patch is rectangular, it should have rounded corners to reduce stress concentrations. Post-weld heat treatment will be required if it was mandated by the construction code of the original vessel or if the patch weld results in large residual stresses. The difficulties with flush patches are primarily construction challenges (the alignment of the patch with the existing vessel which requires tight bending and alignment tolerances).

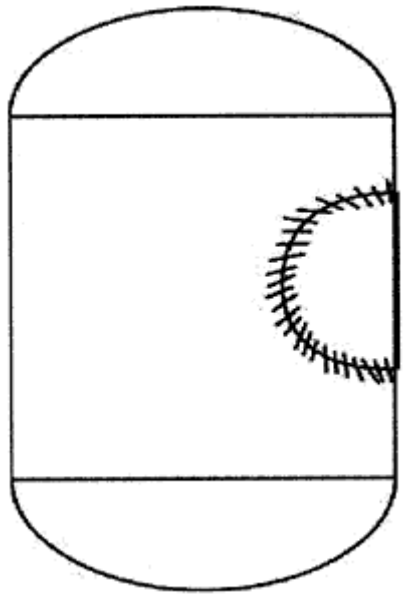

**Figure 23-9** Flush Patch Repair

### 23.8 WELDED LEAK BOX

A welded leak box consists of a pipe jacket with end pieces, placed around the section to be repaired, and seal welded to the pipe, Figure 23-10. If the degraded pipe leaks, the fluid would be contained in the leak box. A leak detection drain or pressure gage can be added to indicate the onset of leakage. The box must be sized to contain the design pressure of the leaking fluid. This is typically achieved in one of two ways: design by analysis or by test. In the case of analysis, the box must meet the applicable code design equations. The difficulty arises at the end pieces. These could be standard pipe caps with a hole opening bored at the center, machined reducers or perforated flat plates. To qualify such an end piece, a detailed analysis is often required (refer for example to ASME VIII Division 2 Appendices 4 or 6).

Alternatively, the leak box can be qualified as an unlisted component by proof test [ASME B16.9, ASME VIII Div.1 UG101, ASME I A-22 or MSS-SP-97]. In this case, a prototype is proof tested until it bursts or deforms. The design pressure is equal to the burst or deformation pressure divided by a safety factor consistent with the design code.

The stability of a through wall crack should be analyzed, with adequate safety margins, to assure that an unstable crack will not longitudinally split and propagate through the repaired pipe beyond the ends of the leak box.

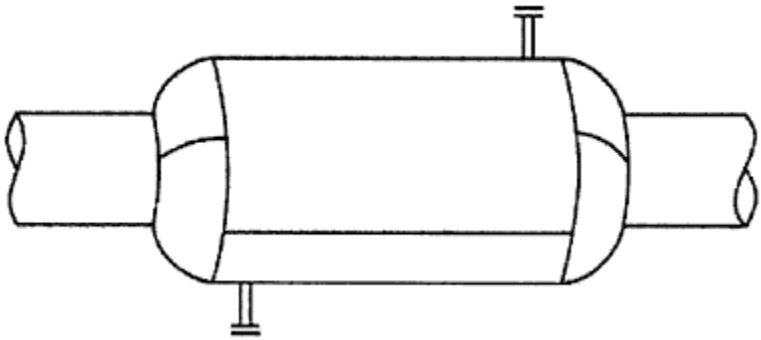

**Figure 23-10** Welded Leak Box

The box and its welds must be sized to react the longitudinal bending stresses due to sustained or occasional design loads (such as weight, thermal expansion, wind, waterhammer, as applicable). In this case, the designer may consider the host pipe to be completely ruptured (circumferential guillotine break) with the leak box reacting all loads. If the loads are excessive, additional pipe restraints may be added on both sides of the leak box. The leak box must be sufficiently long so that the end pieces are welded to sound metal.

## 23.9 MECHANICAL CLAMP

A mechanical clamp is a housing comprised of two half shells, with two end gasket rings that are tightened against the pipe outer diameter by bolting [API 6H], Figure 23-11.

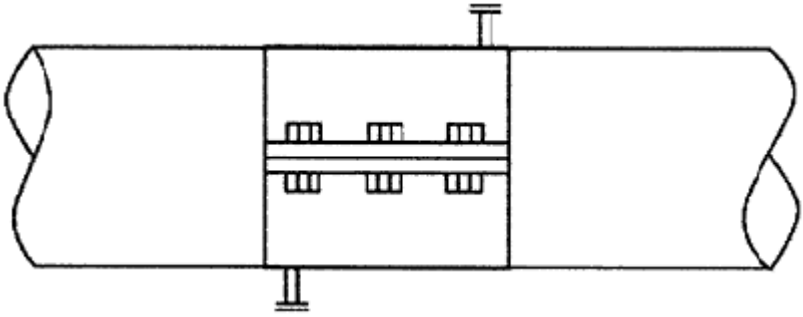

**Figure 23-11** Mechanical Clamp

Mechanical clamps are commonly used in power and process piping systems. Certain clamps may be welded to the pipe, in which case the clamp may be pressure tested and the welds examined. The repair method is listed in API 570 as an acceptable temporary on-stream repair provided the design accounts for the potential for guillotine break and separation of the pipe [API 570].

Mechanical clamps are also permitted for temporary repair of nuclear power plant Class 2 and Class 3 piping systems provided the clamp is designed assuming full severance of the pipe, and the area in the vicinity of the clamp is regularly inspected [ASME XI, NRC 90–05].

A thermosetting sealant can be injected between the leaking pipe and the clamp. The sealant cures and seals the leak. A range of sealants is available to best suit the application.

## 23.10 COMPOSITE OVERWRAP

The use of a non-metallic composite overwrap is a repair technique for corroded or dented pipes or pipelines that does not require welding [FR-60]. Some composite overwraps follow the same principle as a type A full encirclement sleeve. Their objective is to reinforce the corroded wall thus avoiding outward bulging of the pipe wall that precedes burst failure of the thinned area, Figure 23-12.

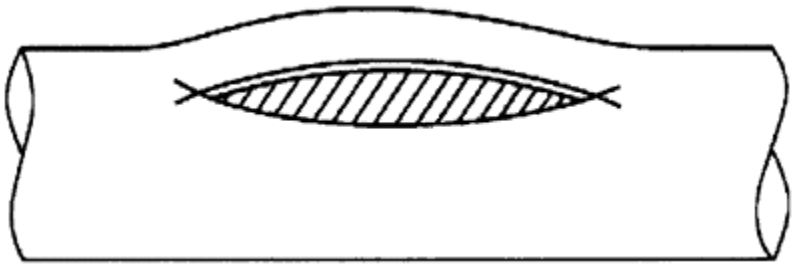

**Figure 23-12** Ductile Burst is Preceded by Outward Bulging

Other overwrap sleeves also provide a leak tight boundary: a resin impregnated fiber glass is wrapped by hand several times around the leaking pipe, and left to cure, Figure 23-13. The cured wrap solidifies around the pipe and seals the leak.

As an alternative, a filler is brushed over the outer diameter of the pipe in the corroded section to fill gaps and assure a tight contact between the pipe and the overwrap. An adhesive is applied over the filler. The sleeve is rolled around the pipe, with adhesive applied on each successive wrap. A strap tightens the wrap before curing. As it cures, the overwrap stiffens and will prevent the outward bulging of the repaired pipe [Boreman, Clock Spring, Citadel Technologies, Mitchell].

The strength of the overwrap should be established by proof testing of wrapped pipe with machined defects. If the mechanical properties of the overwrap will be altered by the environment, a design life should be defined for the repair.

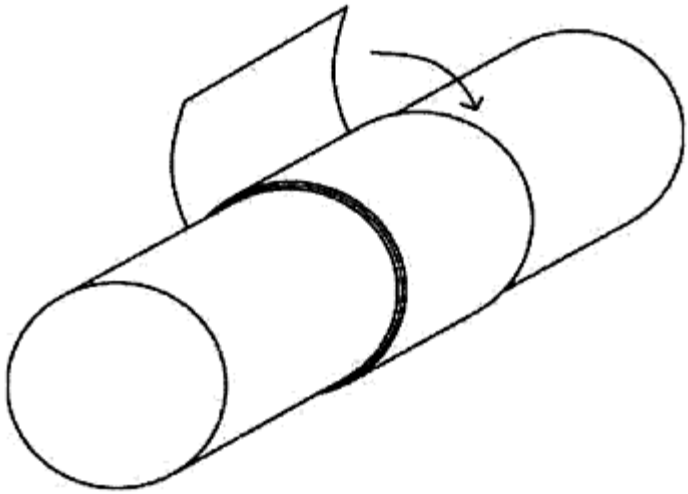

**Figure 23-13** Composite Overwrap Repair

## 23.11 BURIED PIPE REHABILITATION

The need to repair buried pipe without the costly effort of digging, has resulted in the development of several trenchless repair techniques. The first trenchless repair is accomplished by inserting a chemically resistant high-density polyethylene (HDPE) pipe inside the existing degraded pipe.

The HDPE liner is first deformed into a U-shape, and pulled through the existing pipe, Figure 23-14. The U-shaped liner is then reformed to its circular shape, forming a pipe within a pipe [ASTM F 1533]. Two important considerations for this repair technique are (1) the remote cutout and proper sealing of branch openings to avoid infiltration in the pipe-liner annulus, and (2) sealing the pipe-to-liner terminations.

The choice of the liner material is important because polymer liners can lose their function by collapse under annulus pressure by permeation or from chemical dissolution. Gas permeation becomes a concern when the line pressure drops and the annulus pressure becomes sufficient to collapse the liner. Polyamide liners have been successfully used in oil and gas pipelines where HDPE had only lasted 18 months at 130°F and 600 psi. [Berry, Mason, Lebsack].

**Figure 23-14** Insertion Liner Pulled Into Pipe

The trenchless repair of long runs of underground pipe can also be achieved by the insertion of a liner which unfolds inside-out as it is inserted, and then is heated and cured in-place (in-situ) to bond with the existing pipe. This repair technique is used in many applications, including waterworks and gas pipelines [ASTM F 1216]. It has also been used to rehabilitate degraded service water lines in nuclear power plants. The primary component is a tube made of polyester felt which is impregnated with a thermosetting resin containing reinforcing fillers, such as fiberglass, and a curing agent.

First, the liner is selected and designed to restore the strength of the deteriorated pipe. This design depends on the extent of degradation, established by wall thickness and condition inspections of the host pipe, and the properties of the liner. The liner can be sized to provide, by itself, the full strength of the degraded host pipe. The host pipe is cleaned from debris and sharp protrusions that may damage the liner or prevent proper contact between the liner and host pipe.

The resin-saturated tube is introduced at a staging point at one end of the pipe. The liner is secured to the end of the pipe and inverted while being pushed into the degraded host pipe by hydrostatic pressure. Once fully inserted, with the resin now in contact with the host pipe inner diameter, the water in the liner is heated or steam is circulated, which causes the resin to cure and bond with the host pipe. The liner is subjected to a final visual inspection. Surface flaws may be corrected by grinding and trowel of newly mixed resin to fill the repaired area. Voids may be repaired by drilling small access holes through which newly mixed resin is injected. Samples of the cured resin are tested to confirm material properties used in design.

Another trenchless repair technique is live pipe insertion, a method used in gas distribution to repair aging buried lines without interrupting gas flow. In this repair method, a new plastic pipe is inserted into the existing larger diameter metal pipe. The inserted pipe is pressure tested in place while flow continues without interruption through the annular space between the old and new pipe. The new plastic pipe is then connected to carry the flow while the annular space is sealed off by injection of high-density polyurethane.

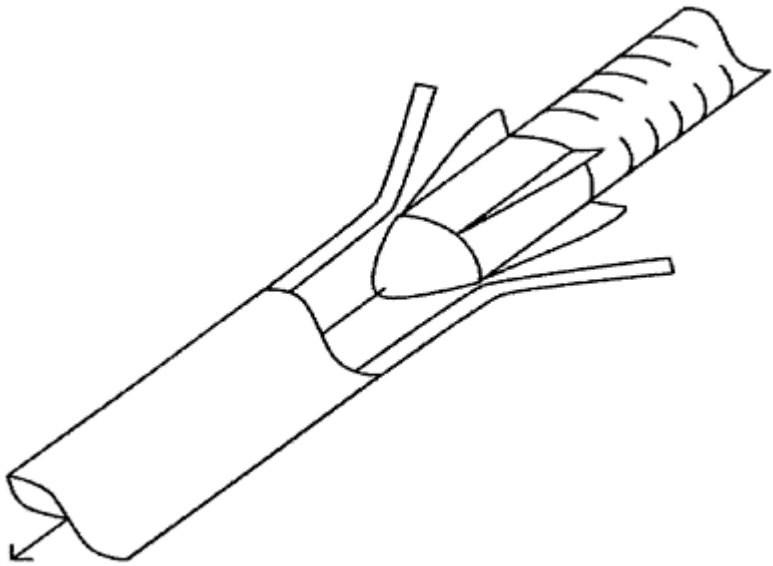

**Figure 23-15** Pipe Splitting with Polyethylene Pipe Follower

In pipe splitting, also called pipe bursting an existing underground pipe is split longitudinally then expanded by a bullet head pulled into the pipe, Figure 23-15. A new pipe, typically polyethylene, is attached to the back of the bullet head and is pulled into the space created by the split pipe. Branch connections must be removed before the operation. A dig is made at the old branch connection and the branch is cut out and removed. A new branch or tee is installed after insertion of the replacement pipe.

## 23.12 BRUSHED AND SPRAYED LINING AND COATING

A number of linings (covering the inner diameter) and coatings (covering the outer diameter) can be rolled, brushed or sprayed on the pipe to stop erosion, corrosion or abrasion of the pipe wall. Typically they are classified as thin (less than 20 to 40 mil paint types) or thick (over 40 mils). The pipe is cleaned by blasting or scraping to white metal with a smooth profile [SSPC]. The pipe may be rinsed with a solvent to clean dust particles and primed to avoid rust on the newly exposed surface. This cleaning step is critical to assure proper adhesion of the new lining to the existing pipe. The lining is then applied. It may consist of a sealer to prepare the surface by filling pits and surface defects and an epoxy based resin with a curing agent, and possibly mineral reinforcements. One or two applications may be necessary to achieve the required thickness.

Linings are factory applied on new pipes or applied in the field. With large diameter pipe, the lining is applied by field spray, brush or roll by a qualified operator wearing the necessary protection and breathing air mask, crawling inside the pipe. Where this is not possible, the operation can be performed by remotely controlled tools such as mechanical

and chemical cleaning pigs followed by a crawler with rotating heads that spray the coating on the pipe, with remote camera inspection of the process. The access of crawlers may be limited by valves, tees or elbows.

If the degradation is due to aggressive abrasion of the inner diameter, a ceramic reinforced epoxy liner may be used. To protect against abrasion of the outer diameter, such as scratching on the surface of pipelines during directional drill insertions, an epoxy based polymer concrete may be used to protect fusion bonded epoxy or coal tar enamel coating. With the right hardener, the concrete-epoxy may be used directly on steel pipe. This concrete-epoxy coating is applied in thickness as low as 10 mils. ASME B31.11, provides some practical guidance regarding protective coatings for buried pipe. The coating must mitigate corrosion, have sufficient adhesion, be ductile, have sufficient strength and be compatible with cathodic protection, if used [ASME B31.11].

## 23.13 PIPE STRAIGHTENING

While it is preferable to replace an accidentally bent pipe, there are times where this may not be feasible. In these cases, an accidentally bent pipe may be straightened back into position, with certain precautions:

(a) The initial accidental bending and subsequent straightening should not cause a dent or gouge.

(b) The pipe cross section should not ovalize more than permitted in the construction code (for example, the ovality should not exceed 8% for ASME B31.3 process piping).

(c) The bent and straightened pipe should not be wrinkled (based on a visual and manual check).

(d) Cold straightening (where the straightening operation takes place below the minimum transition temperature of the metal, for example 1300°F for carbon steel) is possible if permitted by the construction code, otherwise straightening should be performed at the hot bending temperature specified by the construction code. Flame straightening should be limited to 1200°F for plain carbon steel, 1000°F for specially heat-treated steel, and 700°F for stainless steel. The line may be forced into shape, restrained and then heated while restrained for stress relief. The restrains can then be removed once the line is cooled to ambient temperature.

(e) In critical service, if the straightened pipe contains a weld joint, the weld should be volumetrically examined. The integrity of any mechanical joints (flange, tube fitting, etc.) should be evaluated separately.

## 23.14 REFERENCES

49CFR192, Transportation of Natural and Other Gas by Pipeline: Minimum Federal Safety, Code of federal Regulations, Washington, DC.

49CFR195, Transportation of Hazardous Liquids by Pipelines, Code of federal Regulations, Washington, DC.

API 6H, Specification on End Enclosures, Connectors and Swivels, American Petroleum Institute, Washington D.C.

API 510, Pressure Vessel Inspection Code: Maintenance Inspection, Rating, Repair, and Alteration, American Petroleum Institute, Washington D.C.
API 570, Piping Inspection Code, Inspection, Repair, Alteration, and Rerating of In-Service Piping Systems, American Petroleum Institute, Washington D.C.
API 579, Fitness-for-Service, American Petroleum Institute, Washington D.C.
API 1107, Pipeline Maintenance Welding Practices, American Petroleum Institute, 1991.
API 2200, Repairing Crude Oil, Liquefied Petroleum Gas, and Product Pipelines, American Petroleum Institute, Washington D.C.
API 2201, Procedures for Welding or Hot Tapping on equipment in Service, American Petroleum Institute, Washington D.C.
ASME I, Power Boilers, Boiler & Pressure Vessel Code, American Society of Mechanical Engineers, New York, N.Y.
ASME VIII, Pressure Vessels, Boiler & Pressure Vessel Code, American Society of Mechanical Engineers, New York, N.Y.
ASME XI, Inservice Inspection, Boiler & Pressure Vessel Code, American Society of Mechanical Engineers, New York, N.Y.
ASME B16.9, Factory Made Wrought Steel Buttwelding Fittings, American Society of Mechanical Engineers, New York, N.Y.
ASME B31.3, Process Piping, American Society of Mechanical Engineers, New York, N.Y.
ASME B31.4, Pipeline Transportation Systems for Liquid Hydrocarbons and Other Liquids, American Society of Mechanical Engineers, New York, N.Y.
ASME B31.8, Gas Transmission and Distribution Piping Systems, American Society of Mechanical Engineers, New York, N.Y.
ASME B31.11, Slurry Transportation Piping Systems, American Society of Mechanical Engineers, New York, N.Y.
ASME N-432, Code Case, Repair Welding Using Automatic or Machine Gas Tungsten Arc Welding, American Society of Mechanical Engineers, New York, N.Y.
ASME N-561, Alternative Requirements for Wall Thickness Restoration of Class 2 and High Energy Class 3 Carbon Steel Piping, American Society of Mechanical Engineers, New York, N.Y.
ASME N-562, Alternative Requirements for Wall Thickness Restoration of Class 3 Moderate Energy Carbon Steel Piping, American Society of Mechanical Engineers, New York, N.Y.
ASTM F 1216, Standard Practice for Rehabilitation of Existing Pipelines and Conduits by the Inversion and Curing of a Resin-Impregnated Tube, ASTM International, West Conshohocken, PA.
ASTM F 1533, Standard Specification for Deformed Polyethylene (PE) Liner, ASTM International, West Conshohocken, PA.
Berry, A., Polyamide 11 Liners Withstand Hydrocarbons, High Temperature, Pipeline & Gas Journal, December, 1998.
Boreman, D.J., Lewis, K.G., Repair Technologies for Gas Transmission Pipelines, Pipeline & Gas Journal, March 2000.
Brongers, M.P., et. al., Tests, Field Use Support Compression Sleeve for Seam-Weld Repair, Oil & Gas Journal, June 11, 2001.
Cantwell, R.R., The Making of a Permanent Hot Tap Connection, Pipeline & Gas Journal, August, 2000.
Citadel Technologies, Tulsa, OK.
Clock Spring$^{R}$, Clock Spring Company, Houston, TX.
COB Industries Inc., Pressure Test Plugs, Melbourne, FL.
DOT, 49 CFR 192.713(a) and 192.485(a), Transportation of Natural and Other Gas by Pipeline: Minimum Federal Standards, U.S. Department of Transportation, Washington, D.C.
Dresser Industries, DMD Tapping Sleeves, Bradford, PA.

Edison, Guidelines for Weld Deposition Repair on Pipelines, Edison Welding Institute, February 1998.
FR-60, Federal Register Vol. 60 No. 38, Grant Waiver: Repair of Gas Transmission Lines, February 27, 1995, Washington, DC.
Goodfellow, R., Belanger, R, Hot Tap Installed on Operating Sour-Gas Line, Oil & Gas Journal, May 19, 2001.
HSE, 1997, The Safe Isolation of Plants and equipment, Oil Industry Advisory Committee, Norwich, U.K., HSE Books.
Hydrastop, Hydra-Stopping™ and Pipe Tapping Equipment, Blue Island, IL.
IPSCO, International Piping Services Company, Downers Grove, IL.
Jandu, C.S., et. Al., Fitness-for-Purpose Assessment of Encirclement Split-Tees, Proceedings of Onshore Pipelines Conference 2000, IBC 4th International Event, October 2000, Paris.
Lebsack, D., Hawn, D., Internal Pipeline Rehabilitation Using Liners, NACE paper No. 560, Corrosion, 1997.
Lees, F.P., Loss Prevention in the Process Industries, Butterworth, Heinemann, Reed Educational and Professional Publishing Ltd., 1996.
NBIC, ANSI/NB-23, National Board Inspection Code, The National Board of Boiler and Pressure Vessel Inspectors, Columbus, OH.
Mason, J., Pipe Liners for Corrosive High Temperature Oil and Gas Production Applications, NACE paper No. 80, Corrosion, 1997.
Mitchell, J.L., TEPCO Uses Fiberglass Sleeves, MFL Tools to Rehab Liquid Lines, Pipe Line & gas Industry, June, 2000.
MSS-SP-97, Integrally Reinforced Forged Branch Outlet Fittings, Annex B, Manufacturers Standardization Society of the Valves and Fitting Industry, Vienna, VA.
NRC 88–01, Generic Letter, NRC Position on IGSCC in BWR Austenitic Stainless Steel Piping, U.S. Nuclear Regulatory Commission, Washington, D.C.
NRC 90–05, Generic Letter 90–05 Guidance for Performing Temporary Non-Code Repair of ASME Code Class 1, 2, and 3 Piping, U.S. Nuclear Regulatory Commission, Washington D.C.
OSHA, 29 CFR 1910.147 The Control of Hazardous Energy, Code Of Federal Regulations, Washington, D.C.
Rosenfeld, M., Here are Factors that Govern Evaluation of Mechanical Damage to Pipelines, Oil & Gas Journal, September 9, 2002.
Sealfast, Inflatable Pipe Plugs, Portsmouth, VA.
SSPC, Joint Surface Preparation Standards SSPC-SP5, SSPC-SP6 Commercial Blast Cleaning, SSPC-SP10 Near White Blast Cleaning, SSPC-SP14 Industrial Blast Cleaning, Society for Protective Coatings Pittsburgh, PA.
TDW, T.D.Williamson Inc, Tulsa, OK.
Team® Industrial Services, Hot Taps, Line/Freeze Stops, Clamps/Enclosures, Houston, TX.
Tubb, M., TransAlaska Pipeline Restarted After Valve Maintenance Shutdown, Pipeline & Gas Journal, November, 1998.
USCS, Investigation Report, Thermal Decomposition Incident, U.S. Chemical Safety and Hazard Investigation Board, Report 2001–03-I-GA, May, 2002.
WRC 412, Challenges and Solutions in Repair Welding for Power and Process Plants, Welding Research Council Bulletin 412, New York, 1996.

# 24
# Plastic Pipe

## 24.1 PLASTIC FORM

A plastic is a solid material that consists of long, heavy molecules of organic compounds. Vinyl chloride ($C_2H_3Cl$) was first discovered accidentally at the end of the 19$^{th}$ century. When its exceptional corrosion resistance and good mechanical properties were recognized, it started to be produced commercially, including in pipe form, as polyvinyl chloride in the 1930's [UniBell].

There are two forms of plastic pipes: thermoplastics and thermosets. Thermoplastic pipes are pipes that can be repeatedly softened when heated and hardened when cooled, without effect on the material's chemical properties. They are typically manufactured either by extrusion of a melted product through a die, or by injection of a melted product into a mold. Common thermoplastic pipes include PVC (Polyvinyl chloride), polyethylene (PE), high density polyethylene (HDPE), chlorinated PVC (CPVC), acrylonytrile butadiene styrene (ABS), styrene rubber (SR), polybutylene (PB), polypropylene (PP), polyvinyldiene chloride (PVDC, Saran$^R$), fluoroplastics such as polyvinyldine fluoride (PVDF) or polytetrafluorethylene (PTFE, such as Teflon$^R$ or Halon$^R$) or ethylene chlorotrifluoroethylene (Halar$^R$), styrene-rubber (SR), chlorinated polyether (CPE), cellulose acetate butyrate (CAB), and polycarbonate (Lexan$^R$).

The second type of plastic pipe material is thermosetting, which includes epoxies, phenolics, and polyesters. A thermosetting resin sets permanently when cured. It is used as-is for lining or coating, or it can be reinforced with fibers and used to make pipe, fittings and components. Reinforced thermosetting resins can be manufactured either by winding or casting. Winding is accomplished by weaving and then curing a filler (for example, fiberglass filaments) saturated with resin around a mandrel. Casting is accomplished by centrifugally projecting a mixture of chopped fiberglass and resin against the walls of a cylindrical mold. These pipes are commonly called fiber reinforced plastics (FRP). Common resins used in FRP piping include epoxy, polyester, vinylester and phenol-, urea-, melamine-formaldehydes [Chasis, Van Droffelaar].

Rubber is a natural or synthetic material, an elastomer (elastic polymer) [Dillon]. It is a soft material used in piping systems in the form of hose, gasket or packing. Common synthetic rubbers include Buna S (butadiene-styrene), Buna N (butadiene-acrylonitrile), Neoprene (chloroprene), EPDM (ethylene-propylenediene-methylene) and Butyl (isobutylene).

## 24.2 SIZE

Plastic pipe can be procured in several size designations. For industrial pressure service, there are two common size designations: schedules [ASTM D 1785] and pressure ratings [ASTM D 2241]. Common schedules are 40, 80 and 120. A 2" schedule 40 PVC pipe has an outside diameter of 2.375" and a wall thickness of 0.154", same as a steel pipe of the same nominal size and schedule. A rated pipe is designated by its standard dimension ratio or SDR, which is the ratio of pipe diameter to wall thickness. Common SDR values are 64, 51, 41, 32.5, 26, 25, 21, 18, 17 and 13.5. For example, a 2" SDR17 pipe has an outside diameter of 2.275", same as a schedule pipe, but a wall thickness of 2.275"/17=0.140".

## 24.3 CHEMICAL RESISTANCE

The chemical properties of plastic pipes and their corrosion resistance reflect the broad choice of polymers and additives that can be used in their manufacture. Thermoplastics and fiber reinforced pipes are particularly well suited for inorganic fluids such as diluted acids, ammonia gases, alkalites, acids, salts, organic fluids, natural gas and oils. PTFE such as Teflon$^R$ is resistant to most chemicals, with very few exceptions, but can degrade under radiation. Because of its corrosion resistance, PVC is a common material for drinking water pipes. There are however several environments that can corrode plastic pipes, particularly above 120°F. These include acetic acids, benzene, liquid bromine, carbon bisulfide, wet chlorine gas and liquid chlorine, ethers and ethyl esters, and Freon F21 and F22. Plastic pipes are generally resistant to microbiologically induced corrosion (MIC) since they contain no nutrients of interest to biological organisms.

Vinyl based plastics such as PVC, CPVC and VC (vinyl chloride or Saran$^R$) have good corrosion resistance in most services. PVC is however combustible and sensitive to ultraviolet rays, and its service temperature is limited to approximately 140°F. CPVC is more resistant to combustion and can be used up to approximately 210°F. Polyolefins such as HDPE, PP (polypropylene), PB (polybuthylene), and polyurethane have good corrosion resistance, with service temperatures up to 200°F. HDPE is widely used in gas distribution systems. Fluoropolymers such as PVDF, FEP (fluorinated polymer) have good rigidity and wear resistance, with service temperatures up to 350°F.

The use of plastic pipe and plastic liners is more often limited by temperature rather than corrosion resistance. Rather than wall thinning or pitting, corrosion damage in plastics tends to take the form of softening, hardening, swelling or embrittlement [Dillon]. To assess the chemical resistance of plastic pipe it is recommended to start with catalogs and technical bulletins published by plastic pipe fabricators.

## 24.4 PHYSICAL AND MECHANICAL PROPERTIES

Physical properties of plastic pipes include the material specific gravity, the coefficient of thermal expansion in the range of 8 $10^{-6}$ to 80 $10^{-6}$ 1/°F, and whether the material supports combustion (Table 24-1)

**Table 24-1** Physical Properties of Plastic Pipes [Chasis]

| | ASTM Test | PVC | CPVC | PE | FRP General |
|---|---|---|---|---|---|
| Specific gravity | D792 | 1.38 | 1.55 | 0.95 | 1 to 2.5 |
| Expansion ($10^{-6}$ 1/F) | D696 | 30 | 38 | 78 | 8 to 30 |
| Supports combustion | UL | No | No | Yes | Some |

Mechanical properties of plastic pipes are listed in Table 24-2.

**Table 24-2** Mechanical Properties of Plastic Pipes [AWWA M45, Chasis]

| | ASTM Test | PVC | CPVC | PE | FRP General |
|---|---|---|---|---|---|
| Short term tensile strength (ksi) | D638<br>D2105 | 7.2 | 8.4 | 3.3 | 7 to 80 |
| Modulus of Elasticity (ksi) | D638<br>D2105 | 420 | 420 | g<br>120 | 500 to 5000 |
| Short Term Flexural Strength (ksi) | D790 | 14.5 | 15.4 | 3 | 10 to 70 |
| Izod impact (ftlb/in notch) | D256 | 0.7 | 3.0 | 7.0 | 25 |

An important mechanical characteristic of plastics is their viscoelasticity. If a constant stress is applied to the plastic (for example, a weight is hung from one end of a plastic pipe), the pipe will deform instantaneously (short term elasticity), but if the stress remains (the weight is not removed) the deformation will continue (in our example, the pipe holding the suspended weight will keep stretching). The material is said to creep. That is why time under load (pressure, weight or other loads), which means time of service or design life, must be considered when designing plastic systems. Most standards base their plastic design on a 50-year design life, and take into consideration 50-year properties. The creep behavior can be represented by [Janson]

$$\varepsilon=(\sigma/E_S)+(\sigma/E_L)[1-\exp(-Et/\eta_2)]+(\sigma/\eta_3)t$$

$\varepsilon$=total strain of plastic under applied tensile stress

$\sigma$=applied tensile stress, psi

$E_S$=short-term elastic modulus, psi

$E_L$=long-term elastic modulus, psi
t=time at stress, sec
$\eta_2$=material viscosity, psi-sec
$\eta_3$=material viscosity, psi-sec

The first term is the short-term, immediate, elastic strain $\sigma/E_S$. The second term is an elastic-viscous term that, as the time tends to infinity, tends towards the long-term elastic deformation $\sigma/E_L$. The third term is a linear viscous deformation, continuously increasing with time.

## 24.5 PRESSURE DESIGN

The pressure design of plastic pipe is given by one of the following equations [B31.3, FPI]

$t=P(D-t)/(2S)$
$P=2S/(SDR-1)$
$t=PD/(2S+P)$

P=pressure rating, psi
t=pipe wall thickness, in
D=pipe outer diameter, in
S=allowable hoop stress, psi
SDR=standard dimension ratio

The allowable hoop stress S, the maximum permitted tensile stress, is called the hydrostatic design stress (HDS). It is the hydrostatic design basis (HDB) divided by a safety factor [ASME B31.3, UniBell]:

$S=HDS=HDB/SF$

S=allowable hoop stress, psi
HDS=hydrodynamic design stress, psi
HDB=hydrostatic design basis, psi
SF=safety factor
therefore
$P=[2HDB/(SDR-1)]/SF$

The HDB is obtained through a series of long-term rupture tests, as the lowest of the following three values [ASTM D 2837]:

(1) The tensile stress that causes rupture in 100,000 hours (11.4 years) or long term hydrostatic strength (LTHS), obtained by extrapolation of a 10,000 hour (1.14 years) test. The LTHS is then assigned an HDB following a procedure described in ASTM D 2837. For example a measured LTHS between 190 psi and 230 psi is assigned an HDB of 200 psi, LTHS between 240 psi and 290 psi is assigned an HDB of 250 psi.

(2) The 50-year hydrostatic strength, obtained by extrapolation of a 10,000 hour test.

(3) The stress that causes a 5% expansion of the material in 100,000 hours, obtained by extrapolation of a 2,000 hour (83 days) test.

The HDB tests are repeated at several temperatures to obtain HDS=HDB / SF listed in the design code as a function of temperature (Table 24-3).

**Table 24-3** Temperature, HDS and MSTB

| Material | T min (°F) | T max (°F) | HDS 70°F (ksi) | HDS 100°F (ksi) | HDS 180°F (ksi) | MSTB (ksi) |
|---|---|---|---|---|---|---|
| PVC1120 | 0 | 150 | 2 | 1.6 | - | 7.53 |
| CPVC4120 | 0 | 210 | 2 | 1.6 | 0.5 | 7.53 |
| PE3408 | −30 | 180 | 0.8 | 0.5 | - | 2.96 |

The long-term strength of plastics, reflected in HDS, is quite different from their short-term strength, reflected by their mean short-term burst stress (MSTB). The short-term strength PST is typically defined and measured as the pressure that will burst the pipe when raised from 0 ksi to PST ksi in 60 to 70 seconds [ASTM D 1599]. As an example, the short-term tensile strength of PVC is 7 to 8 ksi, the short term flexural strength is 11 to 15 ksi. The long-term tensile and flexural strength is HDB=4 ksi. With a safety factor of 2.5, and operating at ambient temperature (no temperature derating factor) the allowable stress is 1.6 ksi. If the pipe is designed for a pressure hoop stress of 4 ksi/2.5=1.6 ksi, the longitudinal pressure stress will be 1.6/2=0.8 ksi, and therefore the allowable bending stress would be 1.6−0.8=0.8 ksi.

The safety factor SF used to calculate HDS=HDB/SF, or its inverse called the service factor or design factor, varies with service. For example, for water service at ambient temperature (70°F), the safety factor is 2. Given the HDB and SF, the allowable stress is calculated as HDB/SF and the pressure rating is established for each pipe size (D and t). For example, a 12.75" diameter PVC pipe has a HDB of 4000 psi at 70°F. Applying a safety factor of 2, the HDS is 2000 psi. If the design pressure of the system is 200 psi, the design equation is

$2\times2000/(SDR-1)=200$

Therefore, SDR is 21, a standard SDR size, and the corresponding wall thickness is 12.75/21=0.61".

To account for the reduction in tensile strength at temperature, the pressure rating of thermoplastic pipe is reduced with temperature. The temperature correction factor is:

PVC: 1.0 at 70 F down to 0.15 at 150 F.

CPVC: 1.0 at 70 F down to 0.15 at 210 F.

PP: 1.0 at 70 F down to 0.26 at 170 F.

PE: 1.0 at 70 F down to 0.63 at 130 F.

PVDF: 1.0 at 70 F down to 0.25 at 240 F.

For PVC pipe in waterworks, the pressure class is established as [AWWA C 900]

$P=[2\ HDB/(DR-1)]/SF-P_S$

SF=2.5 safety factor

$P_S$=surge allowance based on 2 fps instantaneous variation of flow velocity (typical flow velocity in municipal water mains operating at 80 psi). In special conditions water distribution systems may be designed for flows of 6 to 8 fps. The pres-sure surge is due to the waterhammer that would take place if the flow was stopped instantaneously (Chapter 9).

## 24.6 PRESSURE CYCLING FATIGUE

Failure of plastic pipe or fittings could occur by fatigue under repetitive pressure cycles. For a PVC pipe, the maximum hoop stress S for a number of cycles C is [Vinson, Uni-Bell]

$S=(5.05\ 10^{2`1}/C)^{0.204}$

AWWA M23 reports that PVC pipe can withstand 4000 cycles of 5 ksi hoop stress, and 1.5 million cycles at 1.5 ksi. For reinforced thermosetting resins, ASTM D 2992 procedure A provides a method to determine the hydraulic design basis for cyclic pressure, which results in a fatigue resistance for fiberglass piping of approximately 100 cycles at 20 ksi hoop stress, and 100 million cycles at 8 ksi hoop stress [FPI].

## 24.7 PRESSURE DESIGN OF FITTINGS

For the same schedule, plastic pipe fittings have lower allowable working pressures than straight pipe, as illustrated in Tables 24-4 and 24-5.

**Table 24-4** Pressure Rating of Pipe and Fittings (psi), Schedule 40

| Size | OD | t | BP | PR | MP |
|---|---|---|---|---|---|
| ¾” | 1.050 | 0.113 | 1540 | 482 | 289 |
| 1” | 1.315 | 0.133 | 1440 | 450 | 270 |
| 2” | 2.375 | 0.154 | 890 | 277 | 166 |
| 3” | 3.5 | 0.216 | 840 | 263 | 158 |
| 4” | 4.5 | 0.237 | 710 | 222 | 133 |

**Table 24-5** Pressure Rating of Pipe and Fittings (psi), Schedule 80

| Size | OD | t | BP | PR | MP |
|---|---|---|---|---|---|
| ¾” | 1.050 | 0.154 | 2200 | 688 | 413 |
| 1” | 1.315 | 0.179 | 2020 | 630 | 378 |
| 2” | 2.375 | 0.218 | 1290 | 404 | 243 |

| | | | | | |
|---|---|---|---|---|---|
| 3” | 3.5 | 0.3 | 1200 | 375 | 225 |
| 4” | 4.5 | 0.337 | 1040 | 324 | 194 |

OD=pipe outside diameter, in
t=pipe wall thickness, in
BP=burst pressure of pipe, psi (ASTM D1785, Table 6, test per ASTM D1599).
PR=pressure rating of pipe, psi (ASTM D1785, Table X1.1).
MP=maximum working pressure of fitting, psi

## 24.8 SUPPORT SPACING

The spacing of pipe supports in plastic pipes is governed by the allowable bending stress and mid-span sag. For plastics, the following allowable bending stress has been established based on the material’s hydrostatic design basis HDB [UniBell]

$S_b=(HDB/2)(T/F)$

$S_b$=allowable bending stress, psi
HDB=hydrostatic design basis, psi
T=temperature rating factor, for PVC T=0.88 at 80°F, down to 0.22 at 140°F
F=safety factor=2.0 to 2.5

The spacing L between supports on continuous spans can then be determined from the beam bending stress, as

$wL^2/(10\ Z)<(HDB/2)(T/F)$

w=distributed weight of pipe and contents, lb/in
L=distance between consecutive supports, in
Z=pipe section modulus, $in^3$

In addition to bending stress, pipe sag at mid-span must be addressed in spacing pipe supports. A maximum sag of 0.2% of the span length has been used [AWWA C 900]. This limit can be written as

$wL^4/(128\ EI)<0.2\%\ L$

$L<(0.256\ EI/w)^{1/3}$

or

E=pipe material Young’s modulus, psi
I=moment of inertia of pipe, $in^4$

A closer spacing is required where there are concentrated in-line weights, such as valves. Plastic and FRP pipe have a lower modulus of elasticity than steel (Table 24-2) and must therefore be supported on shorter spans, particu-larly at temperatures above ambient. Table 24-6 is an example of support spacing for PVC pipe, Table 24-7 for fiber reinforced plastic (FRP) and Table 24-8 for high density polyethylene.

**Table 24-6** Support Spacing for PVC Pipe (ft) vs. Pipe Size (in) [Harvel]

| | Sch.40 | | Sch.80 | |
|---|---|---|---|---|
| | 60°F | 140°F | 60°F | 140°F |
| ¾” | 5 | 2.5 | 5.5 | 2.5 |
| 1” | 5.5 | 2.5 | 6 | 3 |
| 1.5” | 6 | 3 | 6.5 | 3.5 |
| 2” | 6 | 3 | 7 | 3.5 |
| 2.5” | 7 | 3.5 | 7.5 | 4 |
| 3” | 7 | 3.5 | 8 | 4 |
| 4” | 7.5 | 4 | 9 | 4.5 |

**Table 24-7** Support Spacing for FRP Pipe (ft) vs. Pipe Size (in)

| | 2” | 3” | 4” | 6” | 8” | 10” | 12” |
|---|---|---|---|---|---|---|---|
| Water | 11 | 14 | 17 | 21 | 23 | 25 | 28 |
| Gas | 17 | 21 | 18 | 34 | 39 | 44 | 48 |

**Table 24-8** Support Spacing for HDPE Pipe (ft) vs. Pipe Size (in)

| | 4” | 12” | 24” |
|---|---|---|---|
| SDR 32.5 | - | 9 | 13 |
| SDR 26 | - | 10 | 13 |
| SDR 11 | 7 | 12 | 16 |

The Fiberglass Pipe Handbook [FPI] provides the following span length equation if there are no in-line concentrated weights

$L_S=[d_m E_b I/(0.013 w)]^{1/4}$

$L_S$=straight length between supports, in

$d_m$=allowable mid-point deflection

$E_b$=bending modulus of elasticity, psi

I=moment of inertia of cross section, $in^4$

w=linear weight, lb/in

In large diameter liquid filled lines, to avoid large local stresses at the point of contact between pipe and support, steel or fiberglass cradles may be used.

For service above normal ambient temperature, above ground FRP pipe is typically anchored (fully constrained against movement or rotation) every few hundred feet, and simply supported between the anchors. In order to avoid compressive buckling as the

pipe tends to expand between anchors, lateral guides should be placed at a distance given by [FPI]

$$L_G=0.2671\ [E_b\ I/(C_t\ A\ T_C\ E_C)]^{0.5}$$

$L_G$=spacing between guides, ft
$C_t$=coefficient of thermal expansion 1/F
A=cross section area of reinforced pipe, $in^2$
$T_C$=temperature change, F
$E_C$=compressive modulus of elasticity, psi (~1.3 $10^6$ psi at 35°F, and 0.6 $10^6$ at 200°F).

## 24.9 FABRICATION AND EXAMINATION

Thermoplastic pipes are joined by solvent-cemented fittings (PVC, ABS, CPVC), coated adhesive (FRP), butt strap adhesive (FRP), mechanical joints, threads or flanges (all), butt fusion (PE, PVDF), compression fittings (PE, PB). The assembly of plastic pipe must follow the pipe fabricator's procedure. For pressure piping (ASME B31 application), assembly must be performed by qualified craft.

### 24.9.1 Solvent Cementing

(1) Break down the gloss. Use a primer or sand paper, then clean the surface.
(2) Use the right cement for the pipe material and schedule.
(3) Apply cement to the pipe and socket.
(4) Insert pipe in socket.
(5) Hold the joint together several seconds.
(6) Avoid cold spring that would pull open the pipe.

### 24.9.2 Coated Adhesive

(1) Break down the gloss. Use a primer or sand paper, then clean the surface.
(2) Mix hardener and adhesive.
(3) Coat adhesive mix to pipe and socket or bell and spigot.
(4) Insert pipe in socket.
(5) Turn ¼ to ½ turn.

### 24.9.3 Butt Strap Adhesive

(1) Break down the gloss using a sand paper, then clean the surface.
(2) Butt pipe ends together. Wrap the mat impregnated with resin around the joint.

### 24.9.4 Hot Plate Butt Fused Joint

(1) 100% visual, daily bend-back test of a joint, data loggers to track and document plate temperature, compression force and time.
(2) Volumetric ultrasonic inspection if critical service.
(3) Watch for grass blades and machine lubricant in fused joint.

### 24.9.5 Hot Plate Socket Joint

(1) Heat male and female ends on hot plate, and insert.

### 24.9.6 Hot Air Welding

(1) Butt joints aligned.
(2) Hot air melts a plastic rod that acts as filler metal.

### 24.9.7 Electrofusion

(1) A prefabricated fitting with electric wires at ends.
(2) Parts assembled and current passed through wire.
(3) Heat from electric resistance melts the joint at the wires.

### 24.9.8 Flange Joints

(1) Typically 1/8" thick full-face gaskets of neoprene or Viton$^{R}$.
(2) Torquing in star pattern (criss-cross) with pre-established torque.

## 24.10 BONDING QUALIFICATION

The qualification of metallic welds relies on bending the welded specimen in a U-shape, and then verifying that it did not crack. Since it is not possible to bend plastics (let alone a bonded plastic joint) in a U-shape, bonder qualification is achieved instead by pressure testing a bonded specimen. Fiber reinforced plastic joints have been qualified by a test at four time the rated pressure of the pipe, for at least one hour. But even with an excellent bonded joint, some thermoplastics could not sustain such high pressures, and a new bonding qualification pressure was introduced in the ASME B31.3 Process Piping code [Bradshaw]:

$$\sigma_b = 0.8\ (0.5\sigma_S + 0.5\sigma_{HDB})$$

$\sigma_b$=stress at bond qualification pressure, psi
$a_s$=stress at short term burst pressure, psi [ASTM D 1599]
$\sigma_{HDB}$=stress at HDB pressure, psi [ASTM D 2837]

## 24.11 REFERENCES

ASME B31.3, Process Piping, American Society of Mechanical Engineers, New York.
ASTM D 256, Standard Test Methods for Determining the Izod Pendulum Impact Resistance of Plastics, ASTM International, West Conshohocken, PA.
ASTM D 638, Standard Test Method for Tensile Properties of Plastics, ASTM International, West Conshohocken, PA.
ASTM D 696, Standard Test Method for Coefficient of Linear Thermal Expansion of Plastics between −30 Degrees C and 30 Degrees C with Vitreous Silica Dilatometer, ASTM International, West Conshohocken, PA.
ASTM D 790, Standard Test Methods for Flexural Properties of Unreinforced and Reinforced Plastics and Electrical Insulating Materials, ASTM International, West Conshohocken, PA.
ASTM D 792, Standard Test Methods for Density and Specific Gravity (Relative Density) of Plastics by Displacement, ASTM International, West Conshohocken, PA.
ASTM D 1599, Standard Test Method for Resistance to Short-Time Hydraulic Pressure of Plastic Pipe, Tubing, and Fittings, ASTM International, West Conshohocken, PA.
ASTM D 1785, Standard Specification for Poly(Vinyl Chloride) (PVC) Plastic Pipe, Schedules 40, 80, and 120, ASTM International, West Conshohocken, PA.
ASTM D 2105, Standard Test Method for Longitudinal Tensile Properties of 'Fiberglass' (Glass-Fiber-Reinforced Thermosetting-Resin) Pipe and Tube, ASTM International, West Conshohocken, PA.
ASTM D 2241, Standard Specification for Polyvinyl Chloride (PVC) Pressure-Rated Pipe (SDR Series), ASTM International, West Conshohocken, PA.
ASTM, D 2837, Standard Test Method for Obtaining Hydrostatic Design Basis for Thermoplastic Pipe Materials, American Society for Testing and Materials, West Conshohocken, PA.
ASTM D 2992, Standard Practice for Obtaining Hydrostatic or Pressure Design Basis for Fiberglass (Glass-Fiber-Reinforced Thermosetting-Resin) Pipe and Fittings, American Society for Testing and Materials, West Conshohocken, PA.
AWWA C900, Polyvinyl Chloride (PVC) Pressure Pipe, 4 in. through 12 in., for Water, American Water Works Association, Denver, CO.
AWWA M 23, PVC Pipe Design and Installation, American Water Works Association, Denver, CO.
Bradshaw, R.D., A Basis for Bonding Qualification Tests for ASME B31.3 Non-Metallic Piping, PVP-Vol. 259, American Society of Mechanical Engineers.
Chasis, D.A., Plastic Piping Systems, Industrial Press Inc.
Dillon, C.P., Corrosion Control in the Chemical Process Industries, Materials Technology Institute, St. Louis, MO.
FPI, Fiberglass Pipe Handbook, Fiberglass Pipe Institute, The Composites Institute of the Society of the Plastics Industry, New York, NY.
Harvel Plastics Inc., Product Bulletin 112/401, 1998, Easton, PA.
Janson, L.E., Plastics Pipes for Water Supply and Sewage Disposal, Borealis, Boras, Sweden.
SPI Society of the Plastics Industry, Inc., Fiberglass Pipe Handbook, Fiberglass Pipe Institute, second Edition, 1992, New York, NY.
UniBell, "Handbook of PVC Pipe Design and Construction", The UniBell PVC Pipe Association, Dallas, TX.
Van Droffelaar, H., Atkinson, J.T.N., Corrosion and its Control, An Introduction to the Subject, NACE International, Houston, TX.
Vinson, H.W., Response of PVC Pipe to Large, Repetitive Pressure Surges, Proceedings of the International Conference on Underground Plastic Pipe, American Society of Civil Engineers, New York, NY, March, 1981.
Vinson, H.W., International Conference on Underground Plastic Pipe, ASCE.

# 25
# Valves

## 25.1 OVERVIEW

In principle, industrial valves are faucets. Like faucets, valves are components used to control flow and pressure in a piping system, from fully open (full flow) to fully closed (isolation), and anywhere in-between (flow regulation). There are four main types of valves: (1) block valves (isolation, on-off service), (2) flow control valves (regulating, throttling), (3) check valves (non-return, back-flow prevention), and (4) pressure relief valves (liquid relief valves, gas safety valves).

The valve itself consists of a body (cast, forged or machined from bar stock) with pipe nozzles (welded, flanged, threaded, or made of specialty fittings), a bonnet (welded, bolted or threaded to the body), a disc (a flat disc, a piston, a cylindrical or tapered plug, or a ball), a seat (against which the seat will close), a stem (a stick) connecting the disc to an actuator, and the actuator (a hand wheel, an air operated diaphragm, a hydraulic piston, or an electric motor, used to move the stem). The wetted parts of the valve, other than the body, are the valve trims: seat, disc, stem, packing, gasket, diaphragm, etc.

Valve specifications (spec. sheets) address the following characteristics:

(a) Construction and qualification standard

ASME B16, NFPA, API, FCI, ASME VIII, National Board, etc.

(b) Function

Throttle, on-off, pressure regulation, pressure relief.

Shutoff class: standard or other. Required seat-disc tightness is best specified by reference to the tightness classifications of FCI 70–2, FCI 91–2, MSS-SP-61, API Std. 527, or API Std. 578.

(c) Hydraulic Characteristics

Line size.

Fluid composition, corrosion characteristics.

Specific gravity.

Inlet temperature.

Inlet pressure.

Inlet vapor pressure.

Normal pressure drop.

Shutoff pressure across valve.

Flow rate.

Required flow coefficients $C_v$ $C_g$ $C_S$

Valve coefficient.

Recovery coefficient $K_m$ $C_l$

Noise level (OSHA limits 90 dba for 8 hours, and 115 dba for 15 minutes).

Control valves have to be sized to achieve the desired function and deliver the desired flow rate and downstream pressure, while reducing the potential for flashing or cavitation. Check valves have to be sized to fully lift or swing open under normal flow conditions, and close without slamming on reverse flow. Safety and relief valves have to be sized to relieve the desired flow rate at the set pressure, and to re-close at the end of the overpressure transient, without shutter.

(d) Valve
Style.
Pressure and temperature rating, typically defined as an ANSI B16 pressure class.
Ends: screwed, flanged, butt, socket.
Material: Iron, CS, SS, etc.
Number of ports: 1 or 2.
Flow direction: down or up.
Any special qualifications (fire testing, seismic testing, etc.).
The relationship between disc lift and flow rate, called valve characteristic, which is most often linear or equal percentage.

(e) Trim
Materials are selected for their resistance to corrosion and cavitation, with due consideration to cost and lead time.
Cage or restrainer material.
Bushing material.
Seat ring material.
Valve plug material.
Valve plug guide: cage, port, stem, top and bottom.
Valve plug balance: balanced or unbalanced.
Port size: full area or partial area.
Characteristic: linear, quick opening or equal percentage (Chapter 9).

(f) Bonnet
Style: standard or other.
Boss: standard or other.
Packing: TFE, Graphite, other.
Bolting: standard or custom material.
Fugitive emissions tightness (standard for leakage to atmosphere, mandated by the Clean Air Act, regulated in the U.S. by the Environmental Protection Agency).

(g) Actuator
Style: piston, diaphragm, other.
Size and thrust.
Air to actuator: 3 top 15 psi, 6 to 30 psi, other.
Close, open or lock on loss of power or air.
Hand jack: top or side.

## 25.2 GATE VALVES

A gate valve, Figure 25-1, is a block valve intended to operate fully open or fully closed. The gate is moved by linear motion of the stem. The flow characteristic of gate valves

(flow vs. gate travel) is very uneven, and flow control with a gate valve is not practical. A gate valve used partially open can cause severe flow cavitation, erosion of the disc, valve and pipe, accompanied by noise or vibration.

There are many types of gate valves. Some gate valves have a solid wedge disc (advantage: simple and sturdy construction. Disadvantage: the solid disc expands at temperature and can bind against the seat; this effect is referred to as thermal binding. Also, the solid disc can bind if the valve body is slightly bent as a result of large pipe loads at the valve nozzles).

Other gate valves have a flexible wedge (advantage: no binding of disc-seat. Disadvantage: fluid migrates into the bonnet, applying a downward pressure on the gate, making it more difficult to open; this effect is referred to as pressure binding).

Gate valves can also have a parallel disc; the disc is specially designed to take advantage of the fluid pressure, sometimes further assisted by a spring inside the disc, to increase the disc-seat sealing force. Unlike other gate valves, the flow direction is critical across parallel disc valves, and, in this case, it becomes essential to install the valve in the right direction of flow.

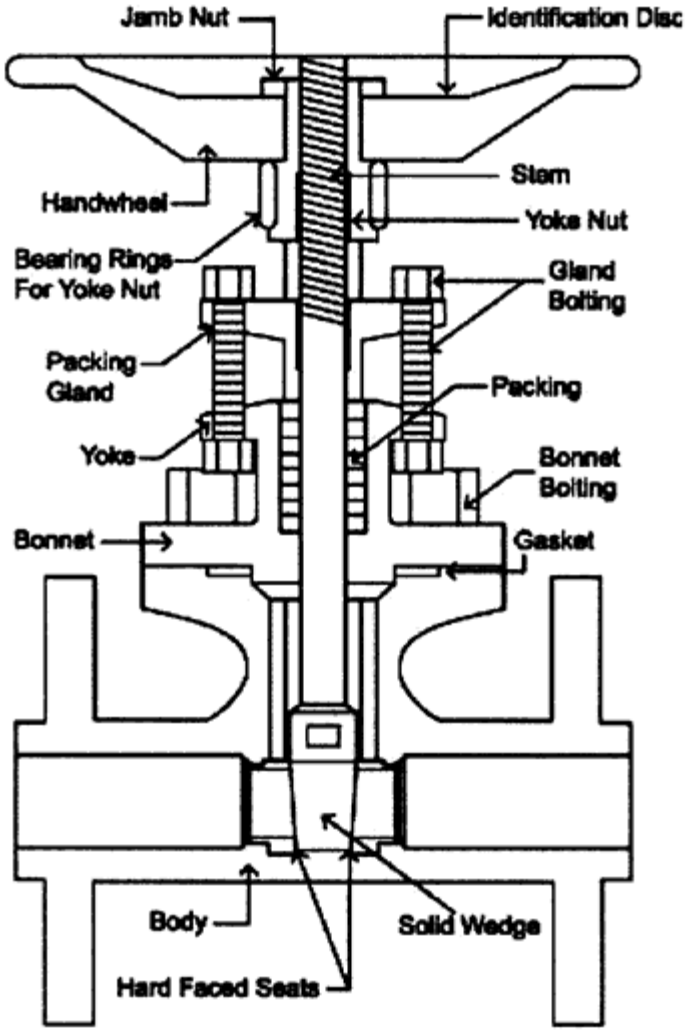


**Figure 25-1** Gate Valve, Solid Wedge

## 25.3 GLOBE VALVES

The disc, ball or plug of a globe valve is moved by linear motion of the stem, Figure 25-2. The flow path and the disc-seat design permit reliable flow and pressure control, which makes the globe valve particularly well suited for regulating service, in addition to its use as a block valve. Flow in a globe valve can be in either direction: flow under the plug (where flow direction tends to open the plug, shielding the stem packing from high pressure when the valve is closed) or flow over the plug (where flow tends to close the

plug, providing better leak tightness in the closed position), it is therefore important to select the proper flow direction for the service and install the globe valve accordingly. Conventional globe valves with stem at right angle (so-called Z body) cause several changes in flow direction and relatively larger pressure drops than most other control valves. The Y body valves have a stem angled relative to the body, a smoother flow profile and less pressure drop. Angle body valves (L shaped) have an outlet nozzle at right angle from the inlet nozzle and stem. The plug, cylinder or seat in a globe valve can be shaped to force the flow through specially designed slots or vanes. These special trims (referred to as whisper trims or tortuous path) reduce turbulence, cavitation and noise across the valve. The design, dimension and flow sizing of globe valves is tightly controlled though the Instrument Society of America's standards ANSI/ISA S75.15 and S75.16.

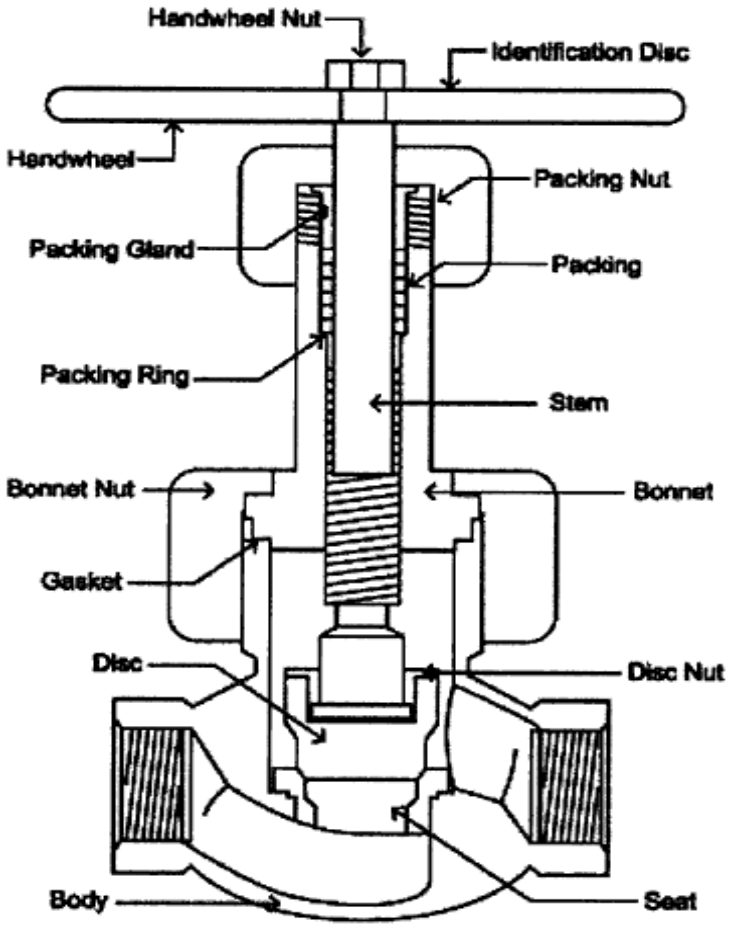


**Figure 25-2** Globe Valve

A needle valve is a small globe valve, with a very narrow flow passage and a needle shaped stem that acts as stem and plug, and very tight threading on the stem drive which permits precise positioning. The body of needle valves is often made of machined bar stock. They are used. for the very precise control of small flow rates (metering).

## 25.4 PLUG VALVE

A plug valve is a quarter turn (rotary) valve with a cylindrical or conical plug and a shaped opening (plug port), Figure 25-3. Plug valves with rectangular ports are used primarily as block valves, taking advantage of the short quarter-turn motion to close. Plug valves with specially shaped ports, and plug valves with an eccentric axis of rotation can also be used for flow control (throttling service). All-metal plug valves can be lubricated to prevent excessive friction, torque and galling as the plug rotates. Alternatively, an elastomer sleeve or liner can be placed around the plug to reduce friction, eliminating the need for lubricant.

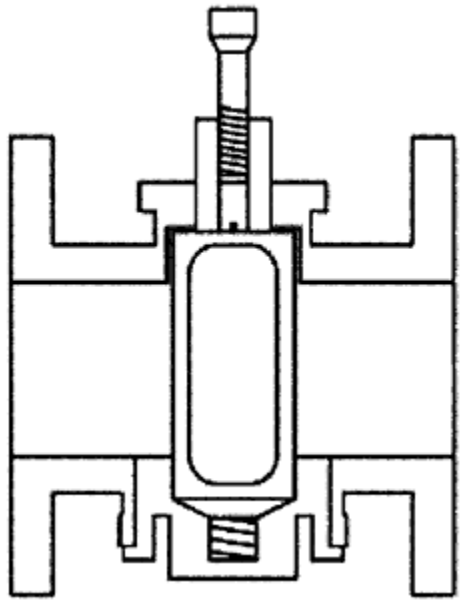

**Figure 25-3** Plug Valve

## 25.5 BALL VALVE

A ball valve is a quarter turn (rotary) valve with a spherical ball and a round or a specially shaped (characterized) opening of the ball. Ball valves can be used as block valves, with good leak tightness, or as flow control valves. The ball valve is useful for

service with clean or dirty fluids since the ball rotates against the body, peeling debris and cleaning itself. However, if the ball is scratched or galled, the valve will leak when closed. Leak tight closure is therefore limited in practice to clean fluids. Like plug valves, ball valves can have a metal or elastomeric seal-ing surface. On some valves, the ball has stops and can only rotate back and forth 90°. Other valves have balls than can rotate 360°.

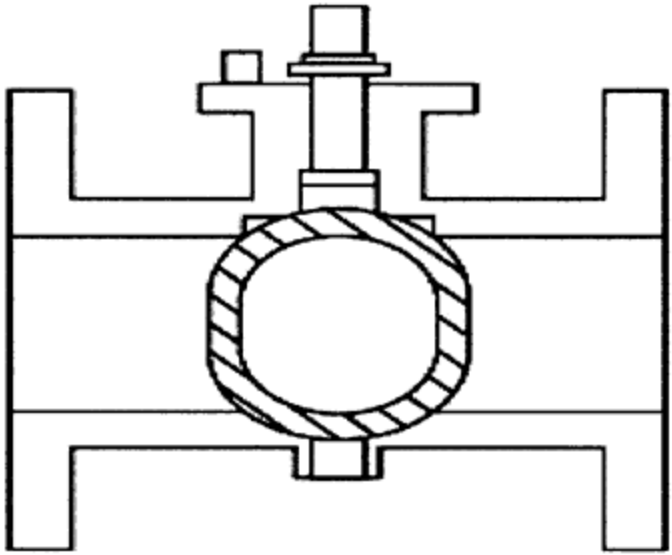

**Figure 25-4** Ball Valve

### 25.6 BUTTERFLY VALVE

A butterfly valve is a quarter turn (rotary) valve with a flat disc rotating around an axis, like the extended wings of a butterfly.

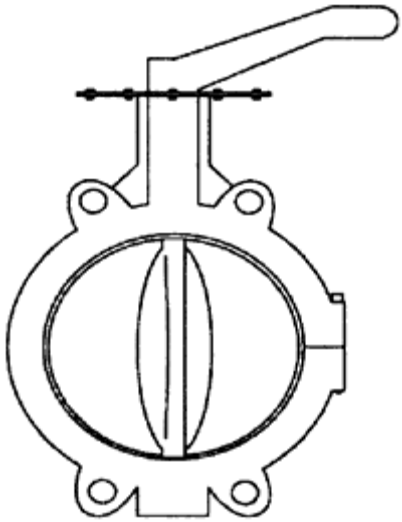

**Figure 25-5** Butterfly Valve

A butterfly acts as a damper; when open, the disc and axis remain in the flow stream. Their great advantage is their narrow width and light weight, low cost, their frictionless rotation that require little torque, and the simplicity of their design. They are well suited for large size, large flows and slurry service, and are often the valve of choice for waterworks. Elastomeric seats are often used, unless leak tightness is not critical, in which case metallic seats are used. Butterfly valves are available with a number of pipe end connections. Wafer butterfly valves are sufficiently narrow to be sandwiched directly between pipe flanges.

## 25.7 DIAPHRAGM VALVE

A diaphragm valve is a valve with a linear motion stem that pushes a flat disc (the compressor) against a diaphragm into the flow stream. The diaphragm seals against a weir in the valve body or against a contoured surface at the bottom of the valve body. The valve can be used as a block valve or in throttling service. In a diaphragm valve, the fluid is not in contact with the valve internals and stem packing, which makes it particularly well suited for very clean service (pharmaceutical or food processing) and for corrosive service (where the diaphragm material can be selected for its corrosion resistance). A pinch valve is similar to a diaphragm valve, but the valve body is simply a cylinder of soft material (for example polyethylene) that can be pinched closed by the linear motion of the stem.

## 25.8 CHECK VALVE

A check valves is a valve designed to automatically permit flow in one direction while preventing reverse flow in the opposite direction. Swing check valves have a disc hinged around a pin at the top of the flow opening, Figure 25-6.

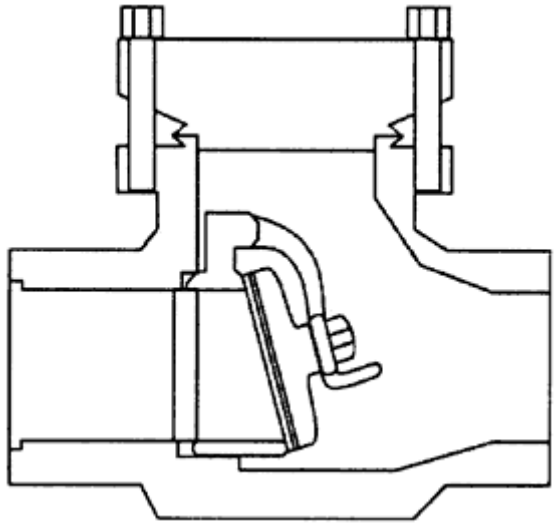

**Figure 25-6** Swing Check Valve

The flow swings the disc open. If the flow stops, the disc weight will drive it to close. If the flow reverses before the disc has fully closed, the disc closure will now be driven by the combined effect of its own weight and the force exerted by the reversing flow, which could be quite large, causing the disc to slam shut, possibly creating a water hammer or breaking the disc by impact. It is therefore important to size a check valve so that (a) the normal flow is sufficient to lift the disc out of the way during normal low, and (b) the disc closes right when the flow stops, before the flow has had time to reverse and slam the valve shut. Note that valve sizing relies on the weight of the disc, it is therefore important not to place the valve on a vertical leg if it was sized for horizontal service.

A tilting disc check valve is hinged around a pin that passes through the disc. The disc has the advantage of a shorter closing swing, allowing less time for flow reversal and slam. The disadvantage of the tilting disc check valve is that the disc remains in the flow stream during normal flow, but its aerodynamic shape is designed to reduce friction losses and pressure drop, while having the right lift and stability characteristic.

A lift check valve is a check valve that relies on the linear motion of a plug, pushed open by flow in one direction, and closing as the flow stops or reverses. A spring assisted lift check valve is a cross between a globe valve, a safety valve and a check valve. The body and flow channels are similar to a globe valve, and the plug is spring assisted for

closure. The flow enters from under the plug. As in the case of a safety-relief valve, the flow automatically lifts the plug. When the flow (and upstream pressure) is reduced, the spring pushes the plug shut against its seat. The valve can be installed horizontal or vertical since it does not rely significantly on the plug weight to re-close.

A butterfly check valve is a butterfly valve with angled and hinged wings. The wings swing towards each other under normal flow conditions, opening the flow area; and then spring back to close the flow path when the flow stops or is reversed. A butterfly check valve can be installed horizontally or vertically.

## 25.9 SAFETY AND RELIEF VALVES

Safety and relief valves are valves that open automatically to relieve overpressure. A safety valve is used in gas or steam service and fully opens at the set pressure. A relief valve is used in liquid service and has an opening characteristic that varies with flow rate, Figure 25-7. Safety and relief valves perform an essential safety function by preventing overpressure ruptures in equipment, pressure vessels, and piping systems. That is why their design and fabrication are closely regulated in most states and in federal facilities to comply with the requirements of the ASME Boiler and Pressure Vessel code. Compliance with the ASME code is evidenced by a stamp on the safety or relief valve. For example, a relief valve that complies with the design, fabrication and quality assurance requirements of the ASME Boiler and Pressure Vessel Code, Section VIII Division 1, is stamped “UV”.

Once placed into service, the relief valve will typically be tested periodically (every year for boiler safety valves, every three to five years for process system valves). The periodic test consists in verifying that the valve pops open at the designated pressure (set-point) and re-closes as expected. Readjusting the set point is most often performed in a valve shop certified by the National Board [ANSI/NB-23], or—if valve removal is impractical—the valve is tested in place.

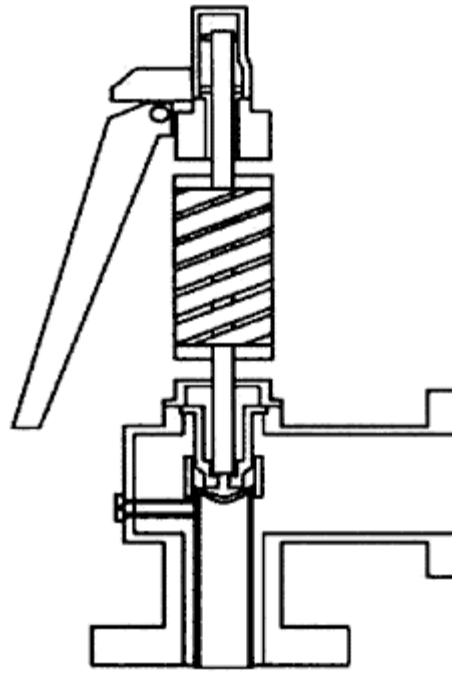

**Figure 25-7** Spring Loaded Relief Valve

In power, pipeline and process applications, the selection and sizing of pressure relief devices (rupture discs, safety valves or relief valves) must comply with the ASME Boiler and Pressure Vessel code. For competent sizing, the engineer must have a thorough knowledge of the requirements of the ASME code [ASME I, ASME VIII, ASME VIII M], be fluent in thermo-hydraulics, and have access to the handful of publications and recommended practices on the subject [Crane, AIChE, API 520, API 521, API 576] and vendor catalogs.

Our objective in this section is to touch on the key criteria for relief valve sizing, with simple illustrations in liquid systems. A full treatment of the subject, including more complex applications for steam and gas systems, is beyond this limited objective.

First criterion: Set point. The relief device should be set at a pressure that does not exceed the pressure rating of the weakest component in the system. Also, preferably, the set pressure should be 10% above the maximum normal operating pressure, to avoid discharge from normal pressure fluctuations during service.

For example, consider a liquid process system with a design pressure of 50 psi (i.e. the lowest rated component is rated for 50 psig). The system normally operates at a maximum pressure of 40 psi. In this case, the relief valve has to be set between a maximum of 50 psi and a minimum of 110%×40 psi=44 psi. In this example, the relief valve will be set at 50 psi.

Second criterion: Capacity. The discharge capacity of the relief device (the minimum flow rate the relief device can discharge) has to be at least the worst credible buildup of flow in the system. To find the worst credible buildup of flow in the system one must consider, at least, the following sources:

(a) The failure of a pressure-regulating valve to a wide-open position, causing the downstream pressure and flow in a system to suddenly rise.

(b) The deadhead of a pump or a compressor against an accidentally closed valve.

(c) The overheating and expansion of trapped fluid due to a rise of ambient temperature or an external fire.

(d) The overheating or overpressure of fluid as a result of an exothermic or explosive reaction in tanks, vessels or pipes.

(e) An external fire.

(f) A runaway reaction.

The question of what constitutes a credible overpressure scenario is a complex one [API 581, API 750, Lees, Deshotels]. It is easily solved if there are applicable regulations mandating certain overpressure accidents be considered as design basis. In the absence of prescriptive rules, the operating company, the regulator and the insurer should carefully address the likelihood and consequence (i.e. the risk) of overpressure events. As a minimum, the system should remain within its code design pressure for overpressures that result from single failure or operator error, or events that have a likelihood of $10^{-2}$/year or more frequent. Overpressure events that have a likelihood of $10^{-2}$/year to $10^{-4}$/year should not cause a rupture or a leak that could endanger workers, the public or the environment. Less likely events (less frequent than $10^{-4}$/year) should also be considered if they can result in significant human or financial loss.

For example, consider a pressure-regulating valve in liquid service. The liquid has a specific gravity of 1. The regulator reduces the liquid pressure from 100 psi to 40 psi. If it fails wide open, the flow rate through the failed regulator will be

$$Q=C_v\,(\Delta P/G)^{0.5}$$

Q=flow rate through regulator, gpm

$C_v$=valve $C_v$ from vendor catalog

$\Delta P$=pressure drop across valve, psi

G=specific gravity of liquid

With a pressure drop from 100 psi down to 40 psi, and if the valve $C_V$ is 5.29, and the specific gravity of the liquid is 1, then

$$Q=5.29\,[(100-40)/1]^{0.5}=41 \text{ gpm}$$

The pressure relief valve must therefore be capable of relieving 41 gpm to counter a fully open regulator failure.

Third criterion: Inlet pressure drop. As the relief device starts to discharge, the upstream fluid in the relief valve inlet pipe starts to flow towards the valve. This sudden flow of fluid to the relief device causes the relief valve inlet pressure to drop. This inlet pressure drop should not be larger than 3% in process systems (or 3 psi at set points of 30 psi or less) [ASME VIII M],

The inlet pressure drop in liquid service is [Crane]

$$\Delta P=0.0000179\,K\,\rho\,Q^2/d^4$$

ΔP=inlet pressure drop, psi

K=resistance coefficient of inlet pipe and fittings

ρ=mass density of fluid, lbm/ft$^3$

Q=flow rate, gpm

d=inner diameter, in

In our example, Q=41 gpm, ρ=62.2 lbm/ft$^3$, d=1.5". If the resistance factor of relief valve inlet pipe and fittings (tees, reducers, elbows, etc.) adds up to 1.6, the pressure drop is $0.0000179\times1.6\times62.2\times41^2/1.5^4$=0.59 psi, which is 1% of the set pressure of 41 psi, and therefore less than 3%. This third criterion is not cast in concrete. It is in a non-mandatory appendix of the ASME code [ASME VIII M]. What is required is that as the inlet pressure drops when the valve opens, the pressure does not drop below the valve reclosing pressure. The difference between the opening set pressure and the reseat pressure is called blowdown. If the inlet pressure drop is too large, it would starve the relief valve and the valve would immediately shut; the pressure would continue to buildup, immediately reopening the valve. As this closing-opening process continues, the valve would chatter, limiting its discharge capability and possibly causing damage. For example, if a relief valve is set at 1000 psi and has a blowdown of 20%, it will open at 1000 psi and reclose when the pressure has dropped down to 800 psi. If the inlet losses are 5%, as the valve opens at 1000 psi the inlet pressure will immediately drop to 950 psi. This pressure of 950 psi is still above the reclosing pressure of 800 psi: the valve will not reclose and will not chatter, even though the pressure drop is larger than the recommended 3%.

Fourth criterion: Accumulation. Pressure accumulation should not exceed the limits permitted by the design code (Chapter 4).

As the relief valve discharges, there will be inlet line losses in the upstream pipe. We go back to our example of a relief valve with 1000 psi set-point and 5% line losses. If the system pressure increases to a static pressure of 1000 psi at the inlet of the closed relief valve, the valve will discharge. The pressure at the valve inlet will drop to 950 psi and then rebuild to 1000 psi. At this point, the pressure in the system being protected has built-up to 1050. In other words, the system pressure has increased from 1000 psi to 1050 psi; it is said to have accumulated 50 psi as the valve discharges. If this is a liquid pipeline (ASME B31.4), the permitted accumulation is 10% (Chapter 4), therefore an accumulation of 5%=50 psi is acceptable.

Fifth criterion: Static backpressure. When the valve is closed, during normal operation, the backpressure at the valve outlet nozzle should be included in the set point.

For example, if a pressure vessel has a maximum allowable working pressure (MAWP stamped on the nameplate) of 500 psi and operates at 300 psi, we have seen in the first criterion that the relief valve set point should be somewhere between 330 psi to 500 psi. Let's say that the set pressure is selected as 400 psi. If during normal operation the relief valve discharge piping sees a static pressure of 30 psi, the set pressure should be set at 430 psi to overcome the superimposed back pressure.

Sixth criterion: Discharge backpressure. As the relief valve discharges, the pressure losses in the downstream piping should be less than 10% of the set pressure.

If the discharge piping is long or constricted, the hydraulic friction in the outlet piping will cause the pressure losses in the downstream pipe to be large and, as a result, the pressure at the relief valve outlet will remain large and tend to close the valve.

## 25.10 CONTROL VALVES

Flow control used to be accomplished by an operator, manually opening or closing a throttle valve, with an eye on a nearby pressure gage. In today's plant, flow control is achieved through automated process control loops. A process control loop includes three principal parts: (1) a measuring device reading and transmitting a process variable, most commonly pressure, temperature, flow rate or level, (2) a computerized controller that takes the measured variable and converts it into a signal to open or close a control valve, and (3) an automated control valve whose electrical or pneumatic operator is actuated by the controller to move the valve stem. A control valve is, in fact, a variable orifice.

A control valve is characterized by its ability to respond to the controller's step signal. The most accurate control valves may respond to signal step changes of 0.5%, while others may require signal changes of 1% or even 5% to respond and change flow area. In particular, the valve will be more sensitive as the actuator is coupled more directly to the stem.

Sizing a control valve consists in selecting a valve that will comply with several rules:

(a) The valve should deliver the required flow rate Q with the required pressure drop dP between its inlet and outlet nozzles.

(b) The valve should control flow in normal service, being 40% to 70% open, and contribute to 10% to 30% of the system pressure drop.

(c) The flow should not be choked.

(d) The valve should not cavitate.

(e) The flow velocity at the valve outlet should not be excessive.

(f) For throttling service, the stem travel vs. flow rate should be well defined (typically linear for liquid systems).

(g) The valve should be able to control flow over the full range of expected flow rate (a valve's rangeability is the ratio of maximum to minimum flow rate that it can control).

(h) For reasons of structural integrity, the valve should not be smaller than one or two sizes below the pipe size.

It may not always be possible to fully comply with rules (c) to (e), and a compromise must be achieved. For example, if some cavitation is inevitable, a hard metallic plug and seat may be used to avoid erosion, or special trims may be used to change the shape of the flow path. To comply with rules (a) and (b), the selected valve must have a valve coefficient Cv such that [Skousen]

$$Q=C_V(1/F_P)(dP/S_P)^{0.5}$$

Q=flow rate, gpm

$C_V$=valve coefficient

$F_P$=inlet/outlet piping geometry factor

$S_P$=specific gravity of the fluid

dP=pressure drop across valve, psi

This relationship between Q and dP shows that, as the control valve closes and the pressure drop increases, the flow rate will increase. There comes a point where the pressure drop is so large that the flow no longer increases with dP, instead the flow rate remains practically constant. It is said that the flow is choked. To comply with rule (c),

the flow across a valve is not choked if the pressure drop across the valve dP is smaller than $dP_{choked}$ where

$$dP_{choked}=F_L^2 (P_1-F_F P_V)$$

$dP_{choked}$=pressure drop across the valve at onset of choked flow, psi

$F_L$=liquid pressure recovery factor (function of valve type, typically 0.7 to 0.9)

$P_1$=valve inlet pressure, psia

$P_V$=vapor pressure of the liquid at inlet temperature, psia

$F_F$=critical pressure factor

$$F_F=0.96-0.28(P_V/P_C)^{0.5}$$

$P_C$=critical pressure of the liquid, psia (for example, 3200 psia for water)

If the pressure drop across the valve dP is too large, the downstream pressure could fall below the vapor pressure of the liquid at the operating temperature and the liquid will flash into vapor bubbles. As the pressure recovers further downstream, the bubbles will collapse, causing popping noises as if there is gravel in the pipe, a phenomenon called cavitation. The multitude of small impingements from bubbles collapsing against the inner diameter of the valve or pipe will erode the metal and form characteristic pits, possibly leading to leaks. To comply with rule (d), cavitation will not occur if the pressure drop across the valve dP is smaller than $dP_{cavitation}$ where

$$dP_{cavitation}=F_L^2 (P_1-P_V)$$

To comply with rule (e), the flow velocity must be below an upper limit, based on experience with similar fluids and pipe and valve materials. Typical flow velocities for process systems in steel pipe are 4 to 10 ft/sec for 2" pipe, up to 7 to 15 ft/sec for 12" pipe. Waterworks and service water systems typically flow at 0 to 7 ft/sec. Control valves should not cause the valve outlet flow to be larger than approximately 30 ft/sec. Given the flow rate Q and the valve flow area $A_v$, the velocity is given by

$$V=0.321 Q/A_V$$

V=flow velocity, ft/sec

Q=flow rate, gpm

$A_V$=flow cross sectional area, $in^2$

## 25.11 SIZING GAS CONTROL VALVES

The mass flow rate of gas through a control valve is

$$w=19.3 F_P C_V P_1 Y [x M_W/(T_1 Z)]^{0.5}$$

w=mass flow rate, lb/hr

$x=dP/P_1$

dP=pressure drop across valve, psia

$P_1$=absolute inlet pressure, psia

$M_W$=molecular weight of gas

$T_1$=upstream temperature, °R (°F+460)

Z=compressibility factor of gas (a function of pressure and temperature)

Y=gas expansion factor

$Y=1-x/[3(k/1.4)x_T]$

k=ratio of specific heats

$x_T$=pressure drop ratio (depends on type of valve)

The flow across the valve will be choked, with the downstream velocity reaching sonic conditions, if

$(dP/P_1)>(k/1.4)x_T$

A control valve in gas service should limit the outlet pressure to Mach (outlet velocity divided by sonic velocity) of 0.5 to avoid excessive noise.

## 25.12 VALVE ACTUATORS

A valve stem may be operated by one of several methods: manually, pneumatically, electrically or electro-hydraulically. Non-manual operators require an activation signal and are called actuators. These are power devices that produce torque or thrust to move the stem [Ulanski].

Manual operation is through a hand wheel, or a handle or lever for quarter turn valves, limiting the torque to approximately 80 ft-lb. A chain is provided to operate elevated valves, with a retaining clamp to avoid the risk of the chain falling on the operator.

Pneumatic actuators rely on air pressure (air operated valves AOV). On receipt of an electric signal (of a few milli-amperes) or a pneumatic signal (3 to 10 psi) from a positioner, plant air (60 to 150 psi pressure, or higher pressures to overcome high pressure flow, regulated to a pressure compatible with the actuator) is introduced into the actuator housing and deforms a diaphragm or drives a piston, which in turn moves the valve stem. Some pneumatic actuators require air to open and air to close (double-acting). Other actuators have a spring that will automatically return the valve to an open or a closed position on loss of air (single acting, fail-open or fail-closed).

Electric motor actuators rely on electric power (often 110 V ac power) to activate a gearbox that drives the stem (motor operated valves MOV). A motor operated valve can develop very high thrust forces to open or close valves against high pressures and high flow rates. The actuators fail in position, which can be an advantage if the fail-open or fail-closed mode of a pneumatic valve is not desired. Motor operated valves have generally a slower motion, which may also be an advantage in avoiding large pressure transients and waterhammer. They are obviously valuable in remote areas, where there is no supply of pressurized air. Motor operators tend to be heavy, and require specialized maintenance.

Electro-hydraulic actuators rely on a motor driving a pump filling either side of a piston with hydraulic fluid; the piston's stroke drives the valve stem. It is gen-erally a fast acting valve that can develop very large opening or closing thrust forces, and can provide fail-safe function.

## 25.13 CLOSURE TEST

A closure test is a test conducted to check for leakage across a closed valve. To conduct a closure test, the valve is closed and pressurized from one side with air, inert gas, water or a non-corrosive liquid of same viscosity as water, the other side being open to atmosphere.

In a broad sense, seat tightness can be viewed in three categories: nominal leakage (valves not required to shut-off), low leakage (a level of tightness sufficient in many industrial applications), and atom-class tightness (extremely high degree of fluid tightness, such as required in aeronautic application) [Zappe]. A more graded approach applies in tightness standards MSS-SP-61, MSS-SP-82, FCI 70–2, API 598, and ASME B 16.34. For a test conducted according to MSSSP-61, each isolation valve or check valve is tested with liquid or gas at 110% the rated pressure, or for certain sizes and ratings with a gas at a pressure of at least 80 psi. The maximum leak rate is 10 ml/hr liquid or 0.1 std.$ft^3$/hr gas per unit of nominal pipe size. Check valves are permitted four times this leak rate. For valves with plastic or elastomeric seal, no visible leak is allowed [MSS-SP-61].

There are three standard seat tightness test techniques in MSS-SP-82: tests W, V and T. In all three cases the acceptance criterion is that there be no "visible leak": no weeping with liquids and no bubbles with air, using a detection technique with a sensitivity of 4.1 $10^{-5}$ $in^3$/sec (6.7 $10^{-4}$ ml/sec) under a differential pressure of 80 psi to 100 psi for 8 NPS and smaller. A type W test is conducted with liquid, air or inert gas at the pressure rating of the valve. A type V test is conducted with air at 80 psi minimum. A type T test is conducted with liquid, air or inert gas at a pressure between 45 psi and 55 psi. In addition to the standard W, V and T tests, a test type S is any test specified, other than W, V or T. A type R designation is where seat tightness is established by quality control measures rather than a test [MSS-SP-82].

There are six standard seat tightness classes in the Flow Control Institute's standard 70–2: Classes I to VI in increasing order of tightness. A Class I valve is not tested for seat tightness. Classes II and III valves are tested with air or water at the maximum operating pressure differential across the seat or 50 psi, whichever is smaller. The permitted leak rate through the seat is 0.5% and 0.1% of rated capacity for Class II and III respectively. Classes IV and V valves are tested with water at the maximum operating pressure differential across the seat. The permitted leak rate through the seat is 0.01% of rated capacity and 5 10"$^4$ ml per minute per inch of seat diameter and per psi differential pressure for Class IV and V respectively. The tightest class is class VI, tested with air or nitrogen gas at maximum operating pressure differential across the seat or 50 psi, whichever is smaller. The permitted leak rate is a function of the port diameter, and varies from 1 bubble/minute for 1" port to 45 bubbles/minute for 8" port [FCI 70–2].

For a test conducted according to API 598, the test pressure is either 60 to 100 psi for low-pressure tests, or 110% of the design pressure for high pressure tests. Leakage of air or gas is detected at the open end by a bubble solution, and leakage of water or liquid can be measured. The seat tightness limits of API 598 are a function of seat type and valve size. For all resilient (elastomeric) seats there should be no visible leak. For metal-to-metal seats the rules are: (a) for valves up to 2" no visible liquid leak or 1 bubble/minute air leak, (b) for 2.5" to 6" valves, 12 drops/minute of water or 24 bubbles/minute of air,

(c) for 8" to 12" valves, 20 drops/minute of water or 40 bubbles/minute of air, (d) for 14" and larger valves, 28 drops/minute of water or 56 bubbles/minutes of air. For check valves, the gas leak rate can not exceed 1.5 std.$ft^3$/hr per inch of nominal size; the liquid leak rate can not exceed 0.18 std.$in^3$/min per inch of nominal pipe size [API 598]. For a test conducted in accordance with ASME B16 [ASME B 16.34], isolation valves are tested with water at 110% of rated pressure, or with gas at 80 psi for 4" and smaller valves. The leakage criteria of MSS-SP-61 or API 598 apply.

Some quarter turn valves (ball, plug, butterfly) have enough play in the closed position that they may not leak in the fabrication shop but will leak in the field. Valves with play in the closed position should not be used where leakage is of concern.

## 25.14 REFERENCES


AIChE, Pressure Relief and Effluent Handling Systems, American Society of Chemical Engineers, New York.

ANSI/ISA S75.15, Instrument Society of America, Research Triangle, NC.

ANSI/ISA S75.16, Instrument Society of America, Research Triangle, NC.

ANSI/NB-23, NBIC, National Board Inspection Code, The National Board of Inspectors, Columbus, Ohio.

API 520, Sizing, Selection, and Installation of Pressure-Relieving Devices in Refineries, American Petroleum Institute, Washington DC.

API 521, Guide for Pressure-Relieving and Depressuring Systems, American Petroleum Institute, Washington DC.

API 527, Seat Tightness of Pressure Relief Valves, American Petroleum Institute, Washington DC.

API 576, Inspection of Pressure-Relieving Devices, American Petroleum Institute, Washington DC.

API 578, Valve Inspection and Testing, American Petroleum Institute, Washington DC.

API 750, Management of Process Hazards, American Petroleum Institute, Washington DC.

API 581, Risk-Based Inspection, Base Resource Document, American Petroleum Institute, Washington DC.

API 598, Valve Inspection and Test, American Petroleum Institute, Washington DC.

ASME I, Power Boilers, American Society of Mechanical Engineers, New York.

ASME VIII M, ASME Boiler and Pressure Vessel Code Section VIII Division 1, Pressure Vessels, Appendix M, Installation and Operation, American Society of Mechanical Engineers, New York.

ASME VIII, ASME Boiler and Pressure Vessel Code Section VIII Division 1, Pressure Vessels, Section UG-125 to 137, Pressure Relief Devices, American Society of Mechanical Engineers, New York.

ASME B 16.34, Valves—Flanged, Threaded and Welding End, American Society of Mechanical Engineers, New York.

Crane, Flow of Fluids Through Valves, Fittings, and Pipe, CRANE Technical Paper 410, Crane Co., King of Prussia, PA.

Deshotels, R., Zimmerman, R.D., Cost Effective Risk Assessment for Process Design, McGraw Hill, NY.

Dickenson, T.C., Valves, Piping & Pipelines Handbook, Elsevier Advanced Technology.

EPRI, Electric Power Research Institute, Valve Application, Maintenance and Repair Guide, TR-105852, 2 volumes, 1996.

Fisher Controls, Control Valve Handbook, Fisher Controls.

FCI 70–2, Control Valve Seat Leakage, Flow Control Institute, Cleveland, Ohio.

FCI 91–2, Standard for Solenoid Valve Seat Leakage, Flow Control Institute, Cleveland, Ohio.

Lees, F.P., Loss Prevention in the Process Industries, Butterworth Heinemann, Oxford, UK.
MSS-SP-61, Pressure Testing of Steel Valves, Manufacturers Standardization Society, Vienna, VA.
MSS-SP-82, Valve Pressure Testing Methods, Manufacturers Standardization Society of the Valve and Fitting Industry, Vienna, VA.
Skousen, P.L., Valve Handbook, McGraw Hill.
Smith, E., Vivian, B.E., An Introductory Guide to Valve Selection, Mechanical Engineering Publications Limited, London.
Ulanski, W., Valve and Actuator Technology, McGraw-Hill, NY.
Zappe, R.W., Valve Selection Handbook, Gulf Publishing.

# Appendix

## Standard Pipe Sizes

A pipe (as opposed to a tube or "tubing") may be defined as a cylinder whose dimensions (inner and outer diameters and wall thickness) comply with dimensional standards ASME B36.10 or ASME B36.19. A line pipe is a pipe commonly used for oil, hydrocarbon products and gas pipelines, whose dimensions comply with API 5L.

A pipe is called out by its nominal pipe size (NPS) and its schedule. Up to NPS 12, the nominal pipe size is a rounded approximation of the outer diameter (for example NPS 4 is a pipe with an outside diameter of 4.5"). Above NPS 12, the NPS is the actual outer diameter of the pipe (an NPS 20 pipe has an outer diameter of 20"). There used to be three schedules of pipe: Standard (std.), extrastrong (XS), and extra- extra-strong (XXS). The current commercial schedules are 5 and 10 for stainless steel (followed by the letter S to designate stainless steel: 5S and 10S), and 40, 60, 80, 100, 120, 140 and 160. But, as indicated in the enclosed tables, not all schedules are commercially available for all sizes.

If t=nominal wall (in), D=outside diameter OD (in), d=inside diameter ID (in), then:

| | | |
|---|---|---|
| The weight of steel pipe per foot is | $W_P=10.6802\ t\ (D-t)$ | lb/ft |
| The weight of water per foot is | $W_W=0.3405\ d^2$ | lb/ft |
| The moment of inertia of the metal cross section is | $I=0.0491\ (D^4-d^4)$ | $in^4$ |
| The section modulus of the metal cross section is | $Z=0.0982\ (D^4-d^4)/D$ | $in^3$ |

**Table A-1** Standard Pipe Sizes

| NPS OD (in) | Sch. | t (in) | ID (in) | $W_P$ (lb/ft) | $W_W$ (lb/ft) | I ($in^4$) | Z ($in^3$) |
|---|---|---|---|---|---|---|---|
| 1/8 | 10S | 0.049 | 0.307 | 0.2 | 0.03 | 0.001 | 0.004 |
| 0.405 | 40-Std-S | 0.068 | 0.269 | 0.2 | 0.02 | 0.001 | 0.005 |
| | | 0.095 | 0.215 | 0.3 | 0.02 | 0.001 | 0.006 |
| ¼ | 10S | 0.065 | 0.410 | 0.330 | 0.06 | 0.003 | 0.01 |
| 0.540 | 40-Std-S | 0.088 | 0.364 | 0.425 | 0.05 | 0.003 | 0.01 |
| | 80-XS-S | 0.119 | 0.302 | 1.065 | 0.03 | 0.01 | 0.03 |
| 3/8 | 10S | 0.065 | 0.545 | 0.4 | 0.1 | 0.006 | 0.02 |
| 0.675 | 40-Std-S | 0.091 | 0.493 | 0.6 | 0.08 | 0.007 | 0.02 |
| | 80-XS-S | 0.126 | 0.423 | 0.7 | 0.06 | 0.009 | 0.03 |
| ½ | 10S | 0.083 | 0.674 | 0.7 | 0.2 | 0.01 | 0.03 |
| 0.840 | 40-Std-S | 0.109 | 0.622 | 0.9 | 0.1 | 0.02 | 0.04 |
| | 80-XS-S | 0.147 | 0.546 | 1.1 | 0.1 | 0.02 | 0.05 |
| | 160 | 0.187 | 0.466 | 1.3 | 0.1 | 0.02 | 0.05 |
| | XXS | 0.294 | 0.252 | 1.7 | 0.02 | 0.02 | 0.06 |

| ¾<br>1.050 | 5S<br>10S<br>40-Std-S<br>80-XS-S<br>160<br>XXS | 0.065<br>0.083<br>0.113<br>0.154<br>0.218<br>0.308 | 0.92<br>0.884<br>0.824<br>0.742<br>0.614<br>0.434 | 0.7<br>0.9<br>1.1<br>1.5<br>1.9<br>2.4 | 0.3<br>0.3<br>0.2<br>0.2<br>0.1<br>0.06 | 0.02<br>0.03<br>0.04<br>0.04<br>0.05<br>0.06 | 0.05<br>0.06<br>0.07<br>0.08<br>0.1<br>0.1 |
|---|---|---|---|---|---|---|---|
| 1<br>1.315 | 5S<br>10S<br>40-Std-S<br>80-XS-S<br>160<br>XXS | 0.065<br>0.109<br>0.133<br>0.179<br>0.25<br>0.358 | 1.185<br>1.097<br>1.049<br>0.957<br>0.815<br>0.599 | 0.9<br>1.4<br>1.7<br>2.2<br>2.8<br>3.7 | 0.5<br>0.4<br>0.4<br>0.3<br>0.2<br>0.1 | 0.05<br>0.08<br>0.09<br>0.1<br>0.1<br>0.1 | 0.08<br>0.1<br>0.1<br>0.2<br>0.2<br>0.2 |
| 1–1/4<br>1.660 | 5S<br>10S<br>40-Std-S<br>80-XS-S<br>160<br>XXS | 0.065<br>0.109<br>0.140<br>0.191<br>0.250<br>0.382 | 1.530<br>1.442<br>1.380<br>1.278<br>1.160<br>0.896 | 1.1<br>1.8<br>2.3<br>3.0<br>3.8<br>5.2 | 0.8<br>0.7<br>0.6<br>0.6<br>0.5<br>0.3 | 0.1<br>0.2<br>0.2<br>0.2<br>0.3<br>0.3 | 0.1<br>0.2<br>0.2<br>0.3<br>0.3<br>0.4 |
| 1–1/2<br>1.9 | 5S<br>10S<br>40-Std-S<br>80-XS-S<br>160<br>XXS | 0.065<br>0.109<br>0.145<br>0.200<br>0.281<br>0.400 | 1.770<br>1.682<br>1.610<br>1.500<br>1.338<br>0.950 | 1.3<br>2.1<br>2.7<br>3.6<br>4.9<br>5.8 | 1.1<br>1.0<br>0.9<br>0.8<br>0.6<br>0.3 | 0.2<br>0.2<br>0.3<br>0.4<br>0.5<br>0.4 | 0.2<br>0.3<br>0.3<br>0.4<br>0.5<br>0.5 |
| NPS OD (in) | Sch. | t (in) | ID (in) | $W_P$ (lb/ft) | $W_W$ (lb/ft) | I ($in^4$) | Z ($in^3$) |
| 2<br>2.375 | 5S<br>10S<br>40-Std-S<br>80-XS-S<br>160<br>XXS | 0.065<br>0.109<br>0.154<br>0.218<br>0.343<br>0.436 | 2.245<br>−2.157<br>2.067<br>1.939<br>1.689<br>1.503 | 1.6<br>2.6<br>3.7<br>5.0<br>7.4<br>9.0 | 1.7<br>1.6<br>1.5<br>1.3<br>1.0<br>0.8 | 0.3<br>0.5<br>0.7<br>0.9<br>1.2<br>1.3 | 0.3<br>0.4<br>0.6<br>0.7<br>1.0<br>1.1 |
| 2–1/2<br>2.875 | 5S<br>10S<br>40-Std-S<br>80-XS-S<br>160<br>XXS | 0.083<br>0.120<br>0.203<br>0.276<br>0.375<br>0.552 | 2.709<br>2.635<br>2.469<br>2.323<br>2.125<br>1.771 | 2.5<br>3.5<br>5.8<br>7.7<br>10.0<br>13.7 | 2.5<br>2.4<br>2.1<br>1.8<br>1.5<br>1.1 | 0.7<br>1.0<br>1.5<br>1.9<br>2.4<br>2.9 | 0.5<br>0.7<br>1.1<br>1.3<br>1.6<br>2.0 |
| 3<br>3.5 | 5S<br>10S<br>40-Std-S<br>80-XS-S<br>160<br>XXS | 0.083<br>0.120<br>0.216<br>0.300<br>0.437<br>0.60 | 3.334<br>3.260<br>3.068<br>2.900<br>2.626<br>2.300 | 3.0<br>4.3<br>7.6<br>10.2<br>14.3<br>18.6 | 3.8<br>3.6<br>3.2<br>2.9<br>2.3<br>1.8 | 1.3<br>1.8<br>3.0<br>3.9<br>5.0<br>6.0 | 0.7<br>1.0<br>1.7<br>2.2<br>2.9<br>3.4 |
| 3–1/2<br>4.0 | 5S<br>10S<br>40-Std-S | 0.083<br>0.120<br>0.226 | 3.834<br>3.760<br>3.548 | 3.5<br>5.0<br>9.1 | 5.0<br>4.8<br>4.2 | 2.0<br>2.8<br>4.8 | 1.0<br>1.4<br>2.4 |

| | | | | | | | |
|---|---|---|---|---|---|---|---|
| | 80-XS-S | 0.318 | 3.364 | 12.5 | 3.9 | 6.3 | 3.1 |
| 4<br>4.5 | 5S<br>10S<br>40-Std-S<br>80-XS-S<br>120<br>160<br>XXS | 0.083<br>0.120<br>0.237<br>0.337<br>0.437<br>0.531<br>0.674 | 4.334<br>4.260<br>4.026<br>3.826<br>3.626<br>3.438<br>3.152 | 3.9<br>5.6<br>11<br>15<br>19<br>23<br>28 | 6.3<br>6.2<br>5.5<br>4.9<br>4.5<br>4.0<br>3.4 | 2.8<br>4.0<br>7.2<br>9.6<br>12<br>13<br>15 | 1.2<br>1.8<br>3.2<br>4.3<br>5.2<br>5.9<br>6.8 |
| 5<br>5.563 | 5S<br>10S<br>40-Std-S<br>80-XS-S<br>120<br>160<br>XXS | 0.109<br>0.134<br>0.258<br>0.375<br>0.500<br>0.625<br>0.750 | 5.345<br>5.295<br>5.047<br>4.813<br>4.563<br>4.313<br>4.063 | 6.3<br>7.8<br>15<br>21<br>27<br>33<br>39 | 9.7<br>9.5<br>8.7<br>7.9<br>7.1<br>6.3<br>5.6 | 6.9<br>8.4<br>15<br>21<br>26<br>30<br>33 | 2.5<br>3.0<br>5.5<br>7.4<br>9.3<br>11<br>12 |

| NPS OD (in) | Sch. | t (in) | ID (in) | $W_P$ (lb/ft) | $W_W$ (lb/ft) | I ($in^4$) | Z ($in^3$) |
|---|---|---|---|---|---|---|---|
| 6<br>6.625 | 5S<br>10S<br>40-Std-S<br>80-XS-S<br>120<br>160<br>XXS | 0.109<br>0.134<br>0.280<br>0.432<br>0.562<br>0.718<br>0.864 | 6.407<br>6.357<br>6.065<br>5.761<br>5.501<br>5.189<br>4.897 | 7.6<br>9.3<br>19<br>29<br>36<br>45<br>53 | 14<br>14<br>13<br>11<br>10<br>9.2<br>8.2 | 12<br>14<br>28<br>41<br>50<br>59<br>66 | 3.6<br>4.3<br>8.5<br>12<br>15<br>18<br>20 |
| 8<br>8.625 | 5S<br>10S<br>20<br>30<br>40-Std-S<br>60<br>80-XS-S<br>100<br>120<br>140<br>XXS<br>160 | 0.109<br>0.148<br>0.250<br>0.277<br>0.322<br>0.406<br>0.500<br>0.593<br>0.718<br>0.812<br>0.875<br>0.906 | 8.407<br>8.329<br>8.125<br>8.071<br>7.981<br>7.813<br>7.625<br>7.439<br>7.189<br>7.001<br>6.875<br>6.813 | 9.9<br>13<br>22<br>25<br>29<br>36<br>43<br>51<br>61<br>68<br>72<br>75 | 24<br>24<br>22<br>22<br>22<br>21<br>20<br>19<br>18<br>17<br>16<br>16 | 26<br>35<br>58<br>63<br>73<br>89<br>106<br>121<br>140<br>154<br>162<br>166 | 6.1<br>8.2<br>13<br>15<br>17<br>21<br>25<br>28<br>33<br>36<br>38<br>38 |
| 10<br>10.75 | 5S<br>10S<br>20<br>-<br>30<br>40-Std-S<br>60-XS-80S<br>80<br>100<br>120<br>140<br>160 | 0.134<br>0.165<br>0.250<br>0.279<br>0.307<br>0.365<br>0.500<br>0.593<br>0.718<br>0.843<br>1.000<br>1.125 | 10.482<br>10.420<br>10.250<br>10.192<br>10.136<br>10.020<br>9.750<br>9.564<br>9.314<br>9.064<br>8.750<br>8.500 | 15<br>19<br>28<br>31<br>34<br>40<br>55<br>64<br>77<br>89<br>104<br>116 | 37<br>37<br>36<br>35<br>34<br>34<br>32<br>31<br>30<br>28<br>26<br>25 | 63<br>77<br>114<br>126<br>137<br>161<br>212<br>244<br>286<br>324<br>367<br>399 | 12<br>14<br>21<br>23<br>26<br>30<br>39<br>46<br>53<br>60<br>68<br>74 |

1 bar=$10^5$ pascals
1 kgf/cm$^2$=9.8 $10^4$ pascals
1 kgf/cm$^2$=14.22 psi
1 psi=6.9 $10^3$ pascals=6.9 kPa
1 pascal=1 N/m$^2$
1 megapascal=1 MPa=$10^6$ Pa=10 bars=147 psi
1 lbf=4.45 Newtons
1 psi=0.069 bar
1 in Hg=0.491 psi
1 bar=0.9869 atmospheres
1 atm.=76 cm Hg
1 atm.= 33.9 ft water
1 atm=29.92 in Hg

| NPS OD (in) | Sch. | t (in) | ID (in) | $W_P$ (lb/ft) | $W_W$ (lb/ft) | I (in$^4$) | Z (in$^3$) |
|---|---|---|---|---|---|---|---|
| 12 | 5S | 0.165 | 12.420 | 22 | 53 | 129 | 20 |
| 12.75 | 10S | 0.180 | 12.390 | 24 | 52 | 140 | 22 |
| | 20 | 0.250 | 12.250 | 33 | 51 | 192 | 30 |
| | 30 | 0.330 | 12.090 | 44 | 50 | 249 | 39 |
| | Std-40S | 0.375 | 12.000 | 50 | 49 | 279 | 44 |
| | 40 | 0.406 | 11.938 | 54 | 49 | 300 | 47 |
| | XS-80S | 0.500 | 11.750 | 65 | 47 | 362 | 57 |
| | 60 | 0.562 | 11.626 | 73 | 46 | 401 | 63 |
| | 80 | 0.687 | 11.376 | 89 | 44 | 475 | 75 |
| | 100 | 0.843 | 11.064 | 107 | 42 | 561 | 88 |
| | 120 | 1.000 | 10.750 | 125 | 39 | 641 | 101 |
| | 140 | 1.125 | 10.500 | 140 | 38 | 701 | 110 |
| | 160 | 1.312 | 10.126 | 160 | 35 | 781 | 123 |
| 14 | 10 | 0.250 | 13.500 | 37 | 62 | 255 | 36 |
| 14.0 | 20 | 0.312 | 13.376 | 46 | 61 | 314 | 45 |
| | 30-Std | 0.375 | 13.250 | 55 | 60 | 373 | 53 |
| | 40 | 0.437 | 13.126 | 63 | 59 | 429 | 61 |
| | XS | 0.500 | 13.000 | 72 | 57 | 484 | 69 |
| | - | 0.562 | 12.876 | 81 | 56 | 537 | 77 |
| | 60 | 0.593 | 12.814 | 85 | 56 | 562 | 80 |
| | - | 0.625 | 12.750 | 89 | 55 | 589 | 84 |
| | - | 0.687 | 12.626 | 98 | 54 | 638 | 91 |
| | 80 | 0.750 | 12.500 | 106 | 53 | 687 | 98 |
| | - | 0.875 | 12.250 | 123 | 51 | 781 | 112 |
| | 100 | 0.937 | 12.126 | 131 | 50 | 825 | 118 |
| | 120 | 1.093 | 11.814 | 151 | 48 | 930 | 133 |
| | 140 | 1.250 | 11.500 | 170 | 45 | 1027 | 147 |
| | 160 | 1.406 | 11.188 | 189 | 43 | 1116 | 160 |

1 mile=1.61 km
1 mile=5280 ft
1 mph=1.47 ft/sec
1 in=25.4 mm
T °C=(T °F−32)/1.8
T °K=(T °F+459.7)/1.8

T °R=1.8 T °K
T °R=T °F+459.7

| NPS OD (in) | Sch. | t (in) | ID (in) | $W_P$ (lb/ft) | $W_W$ (lb/ft) | I ($in^4$) | Z ($in^3$) |
|---|---|---|---|---|---|---|---|
| 18 | 10 | 0.250 | 17.500 | 47 | 104 | 549 | 61 |
| 18.0 | 20 | 0.312 | 17.376 | 59 | 103 | 678 | 75 |
| | Std | 0.375 | 17.250 | 71 | 101 | 807 | 90 |
| | 30 | 0.437 | 17.126 | 82 | 100 | 931 | 103 |
| | XS | 0.500 | 17.000 | 93 | 98 | 1053 | 117 |
| | 40 | 0.562 | 16.876 | 105 | 97 | 1172 | 130 |
| | - | 0.625 | 16.750 | 116 | 96 | 1289 | 143 |
| | - | 0.687 | 16.626 | 127 | 94 | 1402 | 156 |
| | 60 | 0.750 | 16.500 | 138 | 93 | 1515 | 168 |
| | - | 0.875 | 16.250 | 160 | 90 | 1731 | 192 |
| | 80 | 0.937 | 16.126 | 171 | 89 | 1834 | 204 |
| | 100 | 1.156 | 15.688 | 208 | 84 | 2180 | 242 |
| | 120 | 1.375 | 15.250 | 244 | 79 | 2499 | 278 |
| | 140 | 1.562 | 14.876 | 274 | 75 | 2750 | 306 |
| | 160 | 1.781 | 14.438 | 309 | 71 | 3021 | 336 |
| 20 | 10 | 0.250 | 19.500 | 53 | 129 | 757 | 76 |
| 20.00 | - | 0.312 | 19.376 | 66 | 128 | 935 | 94 |
| | 20-Std | 0.375 | 19.250 | 79 | 126 | 1113 | 111 |
| | - | 0.437 | 19.126 | 91 | 125 | 1286 | 129 |
| | 30-XS | 0.500 | 19.000 | 104 | 123 | 1457 | 146 |
| | - | 0.562 | 18.876 | 117 | 121 | 1623 | 162 |
| | 40 | 0.593 | 18.814 | 123 | 121 | 1704 | 170 |
| | - | 0.625 | 18.750 | 129 | 120 | 1787 | 179 |
| | - | 0.687 | 18.626 | 142 | 118 | 1946 | 195 |
| | - | 0.750 | 18.500 | 154 | 117 | 2105 | 210 |
| | 60 | 0.812 | 18.376 | 166 | 115 | 2257 | 226 |
| | - | 0.875 | 18.250 | 179 | 113 | 2409 | 241 |
| | 80 | 1.031 | 17.938 | 209 | 110 | 2772 | 277 |
| | 100 | 1.281 | 17.438 | 256 | 104 | 3315 | 332 |
| | 120 | 1.500 | 17.000 | 296 | 98 | 3755 | 376 |
| | 140 | 1.750 | 16.500 | 341 | 93 | 4217 | 422 |
| | 160 | 1.968 | 16.064 | 379 | 88 | 4586 | 459 |

1 gallon=231 $in^3$
1 gallon=0.134 $ft^3$
1 barrel oil=42 U.S. gallons
1 joule=0.73756 ft-lb
1 Btu=778.169 ft-lb

| NPS OD (in) | Sch. | t (in) | ID (in) | $W_P$ (lb/ft) | $W_W$ (lb/ft) | I ($in^4$) | Z ($in^3$) |
|---|---|---|---|---|---|---|---|
| 24 | 10 | 0.250 | 23.500 | 63 | 188 | 1316 | 110 |
| 24.0 | - | 0.312 | 23.376 | 79 | 186 | 1629 | 136 |
| | 20-Std | 0.375 | 23.250 | 95 | 184 | 1943 | 162 |
| | - | 0.437 | 23.126 | 110 | 182 | 2246 | 187 |
| | XS | 0.500 | 23.000 | 125 | 180 | 2550 | 213 |
| | 30 | 0.562 | 22.876 | 141 | 178 | 2844 | 237 |

| | | | | | | | |
|---|---|---|---|---|---|---|---|
| | - | 0.625 | 22.750 | 156 | 176 | 3138 | 261 |
| | 40 | 0.687 | 22.626 | 171 | 174 | 3422 | 285 |
| | - | 0.750 | 22.500 | 186 | 172 | 3706 | 309 |
| | 60 | 0.968 | 22.064 | 238 | 166 | 4654 | 388 |
| | 80 | 1.218 | 21.564 | 296 | 158 | 5673 | 473 |
| | 100 | 1.531 | 20.938 | 367 | 149 | 6853 | 571 |
| | 120 | 1.812 | 20.376 | 429 | 141 | 7827 | 652 |
| | 140 | 2.062 | 19.876 | 483 | 135 | 8627 | 719 |

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