

HVAC

Systems and Components HANDBOOK

SECOND EDITION

Niles R. Grimm
Robert C. Rosaler

HVAC SYSTEMS AND COMPONENTS HANDBOOK

Nils R. Grimm
Robert C. Rosaler

Second Edition

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Dedication to Second Edition

*This second edition is dedicated in memory of my father,
Akim O. Rosaler.*

*Born in pre-Communist Russia, he was the son of the owner
and president of the Russian-American Steamship Lines.
Educated as an electrical engineer at the University of
Karlsruhe, he was soon engaged in the development of
electric railroads. Emigrating from Germany to the United
States at the onset of World War I, he developed munitions
controls for the U.S. Army, finally settling in to a long post-
war career as a designer of power distribution systems.
During World War II, he developed advanced radar systems
at Bell Laboratories.*

*Dad loved his profession and had a great respect for the
integrity of all engineers.*

I share those feelings.

R. C. R.

Dedication to First Edition

*We fondly dedicate this volume to our dear wives, Lillian
Grimm and Shirley Rosaler, for whose patience and
understanding we are very grateful. They shared in our
problems and frustrations, and finally in our gratification
from creating this work.*

N. R. G. and R. C. R.

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PREFACE TO SECOND EDITION

The *HVAC Systems and Components Handbook* is the second edition of the *Handbook of HVAC Design*.

To keep up with, and sometimes anticipate the technological and societal changes in HVAC, this second edition contains important new information. The entire book has been rearranged to follow a logical progression in format, making it easier to use as a textbook readable from beginning to end as an overview of the industry. Each chapter of the first edition was reviewed, updated and sometimes consolidated with other chapters when appropriate, to focus the book on systems and components.

Specifically, the new subjects include condensate control, radiant panel heating, refrigerants, heat pumps, air filtration and air pollution control equipment, scroll compressors, indoor air quality.

This book is being published concurrently with the *HVAC Maintenance and Operations Handbook*. These two complementary volumes form the complete reference to HVAC technology.

Robert C. Rosaler

PREFACE TO FIRST EDITION

Heating, ventilating, and air-conditioning (HVAC)—or creating a comfortable environment—is at once one of the oldest and one of the most modern technologies. It encompasses everything from the warming radiant heat of the caveman's flames to the comfortably cooled industrial complexes in the Sahara desert and the pressurized comfort of the Challenger space module. Today it is not unusual for an inhabitant of an advanced industrial country to live almost entirely within an artificially created environment. HVAC has turned many environmentally hostile regions into useful, productive areas.

The objective of the *Handbook of HVAC Design* is to provide a practical guide and a reliable reference for designing and operating HVAC systems. It details the necessary steps for planning, design, equipment selection, operation and maintenance. Included are the relevant associated disciplines and considerations necessary for a broad understanding of this subject, including economic factors, pollution controls, and the physiology of comfort.

Each topic is addressed by a leading organization or practitioner in the field.

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*Nils R. Grimm
Robert C. Rosaler*

GENERAL REFERENCES

A project design program is essential to assure an economical, energy-efficient, maintainable, and flexible design that will not only be technically adequate but also meet the client's and/or user's needs within the allocated budget. Three good references for developing design criteria for the total project (all disciplines) are:

Architects Handbook of Professional Practice, 11th ed., Chapter 11, "Project Practices," American Institute of Architects, Washington, D.C., 1988.

Project Checklist, document D200, American Institute of Architects, Washington, D.C., 1982.

Guidelines for Development of Architect/Engineer (A/E) Quality Control Manual, National Society of Professional Engineers (NSPE), Washington, D.C., 1977.

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P · A · R · T A

SYSTEM CONSIDERATIONS

SECTION 1

SYSTEM FUNDAMENTALS

CHAPTER 1.1

CONCEPTUAL AND PRELIMINARY DESIGN

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1.1.1 INTRODUCTION

Heating, ventilating, and air-conditioning (HVAC) systems are designed to provide control of space temperature, humidity, air contaminants, differential pressurization, and air motion. Usually an upper limit is placed on the noise level that is acceptable within the occupied spaces. To be successful, the systems must satisfactorily perform the tasks intended.

Most heating, ventilating, and air-conditioning systems are designed for human comfort. Human comfort is discussed at length in Ref. 1. This reference should be studied until it is understood because it is the objective of HVAC design.

Many industrial applications have objectives other than human comfort. If human comfort can be achieved while the demands of industry are satisfied, the design will be that much better.

Heating, ventilating, and air-conditioning systems require the solution of energy-mass balance equations to define the parameters for the selection of appropriate equipment. The solution of these equations requires the understanding of that branch of thermodynamics called "psychometrics." Ref. 2 should be studied.

Automatic control of the HVAC system is required to maintain desired environmental conditions. The method of control is dictated by the requirements of the space. The selection and the arrangement of the system components are determined by the method of control. Controls are necessary because of varying weather conditions and internal loads. These variations must be understood before the system is designed. Control principles are discussed in Chap. 8.1 and in Ref. 3.

The proliferation of affordable computers has made it possible for most offices to automate their design efforts. Each office should evaluate its needs, choose from the available computer programs on the market, and then purchase a compatible computer and its peripherals.

No one office can afford the time to develop all its own programs. Time is also required to become proficient with any new program, including those developed "in-house."

Purchased programs are not always written to give the information required, thus they should be amenable to in-house modification. Documentation of purchased

programs should describe operation in detail so that modification can be achieved with a minimum of effort.

1.1.2 CONCEPT PHASE

The *conceptual phase* of the project is the feasibility stage; here the quality of the project and the amount of money to be spent are decided. This information should be gathered and summarized on a form similar to Fig. 1.1.1.

1.1.2.1 Site Location and Orientation of Structure

The considerations involved in the selection of the site for a facility are economic:

1. Nearby raw materials
2. Nearby finished-goods markets
3. Cheap transportation of materials and finished goods
4. Adequate utilities and low-cost energy sources for manufacturing
5. Available labor pool
6. Suitable land
7. Weather

These factors can be evaluated by following the analysis given in the *Handbook of Industrial Engineering and Management Bibliography*. It is prudent to carefully evaluate several alternative sites for each project.

The orientation of the structure is dictated by considering existing transportation routes, obstructions to construction, flow of materials and products through the plant, personnel accessibility and security from intrusion, and weather.

1.1.2.2 Codes, Rules, and Regulations

Laws are made to establish minimum standards, to protect the public and the environment from accidents and disasters. Federal, state, and local governments are involved in these formulations. Insurance underwriters may also impose restraints on the design and operation of a facility. It is incumbent upon the design team to understand the applicable restraints *before the design is begun*. Among the applicable documents that should be studied are

1. Occupational Safety and Health Act (OSHA)
2. Environmental Protection Agency (EPA) requirements
3. National Fire Protection Association (NFPA), Fire Code (referenced in OSHA)
4. Local building codes
5. Local energy conservation laws, which usually follow the American Society of Heating, Refrigeration, and Air-Conditioning Engineers (ASHRAE) Standard 90.1A

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1.1.2.3 Concept Design Procedures

The conceptual phase requires the preparation of a definitive scope of work. Describe the project in words. Break it down to its components. Itemize all unique requirements, what is required, why, and when. Budgeting restraints on capital costs and labor hours should be included. A convenient form is shown in Fig. 1.1.2. This form is a starting tool for gathering data. It will suffice for many projects. For a major project, a more formal written document should be prepared and approved by the client. This approval should be obtained before proceeding with the design.

The method of design is influenced by the client's imposed schedule. Fasttracking methods will identify long delivery items that might require early purchase. Multiple construction packages are not uncommon, since they appreciably reduce the length of construction time. Usually, more engineering effort is required to divide the work into separate bid packages. Points of termination of each contract must be shown on the drawings and reflected in the scope of work in the specifications. Great care in the preparation of these documents is required to prevent omission of some work from all contracts and inclusion of some work in more than one contract.

Some drawings and some sections of the specifications will be issued in more than one bid package. To prevent problems, the bid packages should be planned in the concept stage and carried through to completion of the project. All changes must be defined clearly for everyone involved in the project.

Every step of the design effort should be documented in written form. When changes are made that are beyond the scope of work, the written documents help recover costs necessitated by these changes. Also, any litigation that may be instituted will usually result in decisions favorable to those with the proper documentation.

After the scope of work has been accurately documented and approved, assemble the data necessary to accomplish the work:

1. Applicable building codes
2. Local laws and ordinances
3. Names, titles, addresses, and telephone numbers of local officials
4. Names, titles, addresses, and telephone numbers of client contacts
5. Client's standards

If the project is similar to previous designs, review what was done before and how well the previous design fulfilled its intended function.

Use check figures from this project to make an educated guess of the sizes and capacities of the present project. Use Figs. 1.1.3 and 1.1.4 to record past projects.

Every project has monetary constraints. It is incumbent upon the consultant to live within the monies committed to the facility. Use Figs. 1.1.5 and 1.1.6 to estimate the capacities and costs of the systems. Do not forget to increase the costs from the year that the dollars were taken to the year that the construction is to take place.

Justification for the selection of types of heating, ventilating, and cooling systems is usually required. Some clients require a detailed economic analysis based on life cycle costs. Others may require only a reasonable payback time. If a system cannot be justified on a reasonable payback basis, then it is unreasonable to expect the more detailed analysis of life cycle costs to reverse the negative results. A simple comparison between two payback alternatives can be made as follows:

COMPANY _____	PO NO. _____	DATE _____
LOCATION _____	SHEET NO. _____	
SUBJECT _____		

PROJECT BRIEF

COMPUTED BY _____ **CHECKED BY** _____

TYPE OF PROJECT

- ☐ HEATING _____
- ☐ VENTILATING, Comfort, Process, _____
- ☐ AIR CONDITIONING, Comfort, Process, _____
- ☐ PLUMBING, Sewage Treatment _____
- ☐ FIRE PROTECTION _____
- ☐ PROCESS PIPING _____
- ☐ ELECTRICAL, Power, Lighting, Control _____
- ☐ STRUCTURAL, Civil _____
- ☐ ARCHITECTURAL _____
- ☐ _____

DUE DATES:

Preliminaries _____ , Cost Estimates _____ , Final Documents _____ ,

SCOPE OF WORK

PROJECT ASSIGNMENTS: Proj. Mgr. _____ Proj. Engr. _____
Discipline Engrs. _____

Name & Title	Firm Name	Address	Telephone

FIGURE 1.1.2 Project brief.

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COMPANY _____ PO NO. _____ DATE _____										
LOCATION _____ SHEET NO. _____										
SUBJECT _____										
COMPUTED BY _____ CHECKED BY _____										
ESTIMATED COST										
REFRIG. TONS (KW)										
AIR QUALITY	EXHAUST CFM (CMS)									
	SUPPLY CFM (CMS)									
	ACHR									
	CFM/SQ FT (CMS/SQ M)									
ROOM VOLUME CU FT (CU M)										
FLOOR AREA SQ FT (SQ M)										
ROOM NAME & SIZE TYPE OF SYSTEM										

FIGURE 1.15 Conceptual design estimate.

COMPANY _____ PO NO. _____ DATE _____										
LOCATION _____ SHEET NO. _____										
SUBJECT _____										
COMPUTED BY _____ CHECKED BY _____										
ESTIMATED COST										
HEAT LOAD BTU/HR (KW)										
HEAT REQUIRED	BTU/SQ FT (W/CMS)									
	BTU/SQ FT (W/CU M)									
	BTU/SQ FT (W/SQ M)									
ROOM VOLUME CU FT (CU M)										
FLOOR AREA SQ FT (SQ M)										
ROOM NAME & SIZE TYPE OF SYSTEM										

FIGURE 1.1.6 Conceptual design estimate for heating.

$$\text{Payback years } N = \frac{\$ \text{ first cost}}{\$ \text{ savings, first year}} \quad (1.1.1)$$

This simple payback can be refined by considering the cost of money, interest rate i (decimal), and escalation rate e (decimal). The escalation rate is the expected rate of costs of fuel, power, or services. The actual number of years for payback n is given by

$$n = \frac{\log [1 + N(R - 1)/R]}{\log R} \quad (1.1.2)$$

where

$$R = \frac{1 + e}{1 + i}$$

and N is defined by Eq. (1.1.1). This formula is easily programmed on a hand-held computer. A nomographic solution is provided in Ref. 4.

There are many other economic models that a client or an engineering staff can use for economic analysis. Many books have been published on this subject from which the engineer may choose. Refer to Chap. 8.4.

1.1.3 PRELIMINARY DESIGN PHASE

The *preliminary design phase* is the verification phase of the project. Review the concept phase documents, especially if a time lapse has occurred between phases. Verify that the assumptions are correct and complete. If changes have been made, even minor ones, document these in writing to all individuals involved.

1.1.3.1 Calculation Book

The calculations are the heart of decision making and equipment selection. The calculation book should be organized so that the calculations for each area or system are together. Prepare a table of contents so anyone may find the appropriate calculations for a given system. Use divider sheets between sections to expedite retrieval. All calculations should be kept in one place. Whenever calculations are required elsewhere, make the necessary reproductions and promptly return the originals to their proper place in the calculation book.

1.1.3.2 Calculations

The calculations reflect on the design team. The calculations should be neat, orderly, and complete, to aid checking procedures. Most industrial clients require that the calculations be submitted for their review. Also when revisions are required, much less time will be spent making the necessary recalculations. All calculations made during this phase should be considered accurate, final calculations.

Many routine calculations can now be done more rapidly and more accurately with the aid of a computer. The computer permits rapid evaluation of alternatives

and changes. If a computer program is not available for a routine calculation, the calculation should be done and documented on a suitable form. If a form does not exist, develop one.

All calculations should be dated and signed by the designer and checker. Each sheet should be assigned an appropriate number. When a calculation sheet is revised, a revision date should be added. When a calculation sheet is superseded, the sheet should be marked "void." Do not dispose of superseded calculations until the project is built satisfactorily and functioning properly.

List all design criteria on sheets such as Fig. 1.1.7, referencing sources where applicable. List all references used in the design at appropriate points in the calculations.

When you are doing calculations, especially where forms do not exist, always follow a number with its units, such as feet per second (meters per second), British thermal units (watts, foot-pounds, newton-meters), etc. This habit will help to prevent the most common blunders committed by engineers.

To avoid loose ends and errors of omission, *always try to complete one part or section of the work before beginning the next*. If this is impossible, keep a "things to do" list, and list these open ends.

1.1.3.3 Equipment Selection

From the calculations and the method of control, the capacity and operating conditions may be determined for each component of the system. Manufacturers' catalogs give extensive tables and sometimes performance curves for their equipment. All equipment that moves or is moved vibrates and generates noise. In most HVAC systems, noise is of utmost importance to the designer. The designer should know a lot about acoustics and vibrations. Read Chapters 8.2 and 8.3 carefully. Beware of the manufacturer that is vague or ignorant about the noise and vibration of its equipment or is reluctant to produce certified test data.

Many equipment test codes have been written by ASHRAE, American Refrigeration Institute (ARI), Air Moving and Conditioning Association (AMCA), and other societies and manufacturer groups. A comprehensive list of these codes is contained in ASHRAE handbooks. Manufacturer's catalogs usually contain references to codes by which their equipment has been rated. *Designers are warned to remember that the manufacturer's representative is awarded for sales of equipment, and not for disseminating advice*. Designers should make their own selections of equipment and should write their own specifications, based on past experience.

1.1.3.4 Equipment Location

Mechanical and electrical equipment must be serviced periodically and eventually replaced when its useful life has expired. To achieve this end, every piece of equipment must be accessible and have a planned means of replacement.

The roof and ceiling spaces are not adequate equipment rooms. Placing equipment on the roof subjects the roof to heavy traffic, usually enough to void its guarantee. The roof location also subjects maintenance personnel to the vagaries of the weather. In severe weather, the roof may be too dangerous for maintenance personnel.

Ceiling spaces should not be used for locating equipment. Servicing equipment in the ceiling entails erecting a ladder at the proper point and removing a ceiling

FIGURE 1.1.7 Outside design data.

tile or opening an access door, to gain access to the equipment. Crawling over the ceiling is dangerous and probably violates OSHA regulations. No matter how careful the maintenance personnel are, eventually the ceiling will become dirty, the tiles will be broken, and if water is involved, the ceiling will be stained.

Also, the equipment will suffer from lack of proper maintenance, because no one on a ladder can work efficiently. This work in the occupied space is disruptive to the normal activities of that space.

Equipment should be located in spaces specifically designed to house them. Sufficient space should be provided so that workers can walk around pieces of equipment, swing a wrench, rig a hoist, or replace an electric motor, fan shaft, or fan belts. Do not forget to provide space for the necessary electrical conduits, piping, and air ducts associated with this equipment. Boilers and other heat exchangers require space for replacing tubes. Valves in piping should be located so that they may be operated without resorting to a ladder or crawling through a tight space. If equipment is easily reached, it will be maintained. Adequate space also provides for good housekeeping, which is a safety feature.

Provision of adequate space in the planning stage can be made only after the types and sizes of systems have been estimated. Select equipment based on the estimated loads. Lay out each piece to a suitable scale. Arrange the equipment room with cutout copies of the equipment. Allow for air ducts, piping, electrical equipment, access aisles, and maintenance workspace. Cutouts permit several arrangements to be prepared for study.

When you are locating the equipment rooms, be sure each piece of equipment can be brought into and removed from the premises at any time during the construction. A strike may delay the delivery of a piece of equipment beyond its scheduled delivery date. This delay should not force construction to be halted, as it would if the chiller or boiler had to be set in place before the roof or walls were constructed.

1.1.3.5 Distribution Systems

HVAC distribution systems are of two kinds: air ducts and piping. Air ducts are used to convey air to and from desired locations. Air ducts include supply air, return-relief air, exhaust air, and air-conveying systems. Piping is used to convey steam and condensate, heating hot water, chilled water, brine, cooling tower water, refrigerants, and other heat-transfer fluids. Energy is required to force the fluids through these systems. This energy should be considered when systems are evaluated or compared.

System Layouts. Locate the air diffusers and heat exchangers on the prints of the architectural drawings. Note the air-flow rates for diffusers and the required capacities for the heat exchangers. Draw tentative single-line air ducts from the air apparatus to the air diffusers. Mark on these lines the flow rates from the most remote device to the fan. With these air quantities, the air ducts may be sized. Use Chap. 3.2 or *ASHRAE Handbook, Fundamentals*, Chap. 32, or the *Industrial Ventilation Manual* to size these ducts. Record these sizes on a form similar to those shown there.

A similar method is used to size the piping systems; see Chap. 3.1. Remember, steam, condensate, and refrigerant piping must be pitched properly for the systems to function correctly. Water systems should also be pitched to facilitate draining and elimination of air.

Piping systems are briefly described in Chap. 3.1 of this book and in the *ASHRAE Handbook, Fundamentals*. A more substantial treatment is contained in *Piping Handbook* (see Bibliography).

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CHAPTER 1.2

HEATING AND COOLING LOAD CALCULATIONS

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1.2.1 INTRODUCTION

One of the cardinal rules for a good, economical energy-efficient design is *not* to design the total system (be it heating, ventilating, air conditioning, exhaust, humidification, dehumidification, etc.) to meet the most critical requirements of just a small (or minor) portion of the total area served. That critical area should be isolated and treated separately.

The designer today has the option of using either a manual method or a computer program to calculate heating and cooling loads, select equipment, and size piping and ductwork. For large or complex projects, computer programs are generally the most cost effective and *should be used*. On projects where life cycle costs and/or annual energy budgets are required, *computer programs should be used*.

Where one or more of the following items will probably be modified during the design phase of a project, *computer programs should be used*:

- Building orientation
- Wall or roof construction (overall U value)
- Percentage of glazing
- Building or room sizes

However, for small projects a manual method should be seriously considered before one assumes automatically that computer design is the most cost-effective for all projects.

In the next section, heating and cooling loads are treated together since the criteria and the computer programs are similar.

1.2.2 HEATING AND COOLING LOADS

The first step in calculating the heating and cooling loads is to establish the project's heating design criteria:

- Ambient dry-bulb or wet-bulb temperature (or relative humidity), wind direction and speed
- Site elevation above sea level, latitude
- Space dry-bulb or wet-bulb temperature (or relative humidity), ventilation air
- Internal or process heating or cooling and exhaust air requirements
- Hours of operation of the areas or spaces to be heated or cooled (day, night, weekday, weekends, and holidays)

Even when the owner or user has established the project design criteria, the designer should determine that they are reasonable.

The winter outdoor design temperature should be based preferably on a minimum temperature that will not be exceeded for 99 percent of the total hours in the months of December, January, and February (a total of 2160 h) in the northern hemisphere and the months of June, July, and August in the southern hemisphere (a total of 2208 h). However, for energy conservation considerations, some government agencies and the American Society of Heating, Refrigeration, and Air-Conditioning Engineers (ASHRAE) Standard 90-75, *Energy Conservation in New Building Design*, require the outdoor winter design temperature to be based on a temperature that will not be exceeded 97.5 percent of the same total heating hours.

Similarly, the summer outdoor design dry-bulb temperature should be based on the lowest dry-bulb temperature that will not be exceeded 2½ percent of the total hours in June through September (a total of 2928 h) in the northern hemisphere and in December through March in the southern hemisphere (a total of 2904 h). For energy conservation reasons, some government agencies require the outdoor summer design temperature to be based on a dry-bulb temperature that will not be exceeded 5 percent of the same total cooling hours.

More detailed or current weather data (including elevation above sea level and latitude) are sometimes required for specific site locations in this country and around the world than are included in standard design handbooks such as Refs. 1 and 2 or computer programs such as Refs. 3 and 4 or from Ref. 5.

It is generally accepted that the effect of altitude on systems installed at 2000 ft (610 m) or less is negligible and can be safely omitted. However, systems designed for installations at or above 2500 ft (760 m) must be corrected for the effects of high altitude. Appropriate correction factors and the effects of altitudes at and above 2500 ft (760 m) are discussed in App. A of this book.

To avoid overdesigning the heating, ventilating, and air-conditioning system so as to conserve energy and to minimize construction costs, each space or area should be analyzed separately to determine the minimum and maximum temperatures that can be maintained and whether humidity control is required or desirable. For a discussion of humidity control see Chap. 7.7, “Dessicant Dehumidifiers,” in this book.

The U.S. government has set 68°F (20°C) as the maximum design indoor temperature for personnel comfort during the heating season in areas where employees work. In manufacturing areas the *process requirements* govern the actual temperature. From an energy conservation point of view, if a process requires a space temperature greater than 5°F (2.8°C) above or below 68°F (20°C), the space should, if possible, be treated separately and operate independently from the general personnel comfort areas. The staff members working in such areas should be provided with supplementary spot (localized) heating, ventilating, and air conditioning systems as the conditions require, in order to maintain personnel comfort.

The space's dry-bulb temperature, relative humidity, number of people, and ventilation air requirements can be established (once the activity to be performed in

each space is known) from standard design handbook sources such as Refs. 2, 6 to 8, 10, and 22 for heating and Refs. 1, 6 to 22, 27, and 40 for cooling.

The normal internal loads generally produce a heat gain and therefore usually are not considered in the space heating load calculations but must be included in cooling load calculations. These internal loads, including process loads, are listed in standard design handbook sources such as Refs. 23 and 24.

The process engineering department or quality control group should determine the manufacturing process space temperature, humidity, and heating requirements. The manufacturer of the particular process equipment is an alternative source for the recommended space and process requirements.

The air temperature at the ceiling may exceed the comfort range and should be considered in calculating the overall heat transmission to or from the outdoors. A normal 0.75°F (0.42°C) increase in air temperature per 1 ft (0.3 m) of elevation above the breathing level [5 ft (1.5 m) above finish floor] is expected in normal applications, with approximately 75°F (24°C) temperature difference between indoors and outdoors.

There is limited information on process heating requirements in standard handbooks, such as Refs. 25 to 35, and on cooling requirements, such as Refs. 25, 27, and 29 to 35.

Usually the owner and/or user establishes the hours of operation. If the design engineer is not given the hours of operation for the basis of the design, she or he must jointly establish them with the owner and/or user.

The method of calculating the heating or cooling loads (manual or computer) should be determined next.

1.2.2.1 Manual Method

If the manual method is selected, the project heating loads should be calculated by following one of the accepted procedures found in standard design sources such as Refs. 21, 22, and 36 to 39. For cooling loads, see Refs. 21 to 24, 37 to 39, 41, and 42.

1.2.2.2 Computer Method

If the computer method has been chosen to calculate the project heating or cooling loads, one must then select a program to use among the several available. Two of the most widely used for heating and cooling are Trane's TRACE and other Customer Direct Service (CDS) Network diskettes and Carrier's E20-II programs.

Regardless of the program used, its specific input and operating instructions must be strictly followed. It is common to trace erroneous or misleading computer output data to mistakes in inputting the design data into the computer. *It cannot be overstressed that to get meaningful output results, the input data must be correctly entered and checked after entry before the program is run.* It is also a good policy, if not a mandatory one, to independently check the computer results the first time you run a new or modified computer program, to ensure the results are valid.

If the computer program used does not correct the computer output for the effects of altitude when the elevation of the project is equal to or greater than 2500 ft (760 m) above sea level, the computer output must be manually corrected by using the appropriate correction factors, listed in App. A of this book.

We outline the computer programs available with TRACE® and other CDS diskettes and E-20-II in the remainder of this chapter. However, *this is not to imply that*

these are the only available sources of programs for the HVAC fields. Space restraints and similarities to other programs are the same reasons for describing programs from only two sources. Programs are changing rapidly, and you should keep up-to-date on these continually.

1.2.3 TRANE PROGRAMS*

Software can dramatically aid the system selection process by simulating various alternatives accurately and quickly.

Programs are available that perform accurate energy and load analyses which can then be translated into dollars and cents by modeling a particular utility's rates. Still other computerized design tools predict acoustical performance and simplify HVAC equipment selection, air and water distribution, life-cycle costing, and system comparisons.

The following summary describes programs available. (Ref. 43)

1.2.3.1 Analysis Tools

TRACE® 600

Load Calculation and Energy Analysis Software. TRACE® performs life-cycle cost analyses that help the user evaluate various combinations of alternatives in building envelope construction, HVAC system design/operation, equipment choices, and control strategies.

For example, TRACE can help predict the effect of installing better window glazing on HVAC operating costs, or how changing the temperature difference across the chiller's evaporator or condenser will impact the operating costs of the pumps and cooling tower.

A partial list of the many options TRACE® 600 can model follows. (*Those marked with an asterisk can also be simulated with Trane's System Analyzer® software.*)

- Variable vs. constant air volume systems*
- Multiple air distribution systems
- Separate makeup air systems
- Supply air reset*
- Ventilation reset
- Air-side economizer*
- Water-side economizer*
- Equipment heat recovery*
- Exhaust-air heat recovery
- Desiccant dehumidification
- Gas absorption
- Hybrid chiller plants*
- Decoupled chiller systems*
- High-efficiency equipment*
- Integrated Comfort™ system (ICS) control strategies*
- Switchover controls*
- Variable-speed drives*
- Thermal storage*
- Demand limiting with prioritized shutdown*

TRACE® 600 is based entirely on ASHRAE algorithms and actual hour-by-hour

*This section courtesy of the Trane Corporation, LaCrosse, WI.

weather data. An extensive library of predefined building elements and equipment simplify data entry. Comprehensive output reports detail analysis results to aid the decision-making process.

The program is accompanied by a reference manual of “recipes” for modeling complex HVAC systems, equipment, and control strategies.

System Requirements

- IBM-compatible computer (286 or higher) with math coprocessor
- 640 KB of RAM
- 16–20 MB of available hard disk space (10 MB for the program, 6–10 MB for run-time files)
- DOS 3.1 or higher

System Analyzer™

Windows™-Based Energy and Economic Analysis Program. System Analyzer performs load calculations and allows the user to generate and present impressive energy and economic analyses in just a few minutes—with little or no HVAC training. Experienced designers can use the program as a “scoping” tool to quickly and easily examine different systems and assess the impact of control strategies such as night setback, demand limiting and optimum start/stop. If a particular combination of equipment appears promising, *TRACE® 600* can be used to conduct a more detailed analysis later.

Rather than require detailed building entries like *TRACE*, *System Analyzer* is based on simplifying assumptions that expedite the comparison of virtually any building, system, and equipment combination. The program models many of the same advanced HVAC options as *TRACE® 600* (see the preceding asterisked list), and includes a library of predefined building and equipment templates that are readily customized. *System Analyzer's* output reports include visual graphs suitable for inclusion in proposals.

System Requirements

- IBM-compatible computer with 386 (or higher) processor and math coprocessor
- 4 MB of RAM
- Windows 3.1 or higher
- 10 MB of available hard disk space

Load Express™

Light Commercial Load Calculator. *Load Express®* is a Windows™-based load design program for light commercial buildings, with a graphical interface, minimal entries and libraries of predefined building elements such as walls and roofs. Additional elements can be created as needed. Program calculations are based on ASHRAE-approved algorithms, and the results are documented in reports that detail the expected cooling load, heating load and airflow capacity. All zone information is summarized on one screen for easy review.

System Requirements

- IBM-compatible computer with 486 (or higher) processor
- 4 MB of RAM

- Windows 3.1 or higher
- 16 MB of available hard disk space

TRACE® Load 700

Load Design Tool Designed for Windows™ 3.1. Performing iterative cooling and heating load calculations is one of the most common (and time-consuming) tasks HVAC system designers face. To improve the accuracy and efficiency of this task, *TRACE® Load 700* combines the power building load and design portion of *TRACE® 600* with the simplicity of a Windows™-based operating environment. Like its predecessor, *TRACE® Load 700* uses ASHRAE-standard algorithms to assure calculation integrity. It also enables nonsequential data entry that encourages “what if” analysis. Users can edit building construction details in any order and change the building model as the design progresses.

Two distinct levels of data entry permit either quick calculation of a building’s load or modeling of complex building geometries and systems. Extensive libraries of predefined (but editable) templates of construction materials and building load information increase the speed and accuracy of the modeling process.

TRACE® Load 700 automatically creates detailed reports of entered data and calculation results. Once the load and design calculations are complete, the resulting output file can be exported to *TRACE® 600* for a detailed energy analysis.

System Requirements

- IBM-compatible computer with 486 (or higher) processor
- 8 MB of RAM
- Windows 3.1 or higher (also compatible with Windows 95)
- 10 MB of available hard disk space

Trane Acoustics Program (TAP®)

Automates ASHRAE’s “Algorithms for HVAC Acoustics.” Evaluating the total effect of sound in an enclosed space requires many complex mathematical equations. Solving those equations manually takes hours of precious design time and is prone to error. The *Trane Acoustics Program—TAP®*—streamlines this analysis task with easy-to-use menus and dialog boxes that help the user create pictorial diagrams of sound paths. As path elements are added, moved, or deleted, *TAP* dynamically recalculates the resulting sound power levels; and when multiple paths are involved, *TAP* not only determines the overall sound level at the receiver, but also how much of that sound each path contributes. Analysis results can be viewed on screen or printed either as a series of detailed tables or as plots on an NC or RC chart with *TAP*’s built-in graphing function.

System Requirements

- IBM-compatible computer with 486 (or higher) processor
- 8 MB of RAM
- VGA (or better) display
- Windows 3.1 or higher
- 10 MB of available hard disk space

VentAir 62™

Ventilation Airflow “Calculator.” VentAir 62™ helps engineers design multiple-space ventilation systems that satisfy the requirements of ASHRAE Standard 62-1989. Its user-friendly, Windows™-based interface and powerful calculation engine simplify the otherwise time-consuming, complex, and iterative computations required to accomplish that task. The program automates multiple-space Equation 6-1 of Standard 62 and accurately predicts the effect of reducing the critical zone airflow requirement. It also generates comprehensive reports that document design assumptions, calculations and equations—all of the information needed to demonstrate compliance with the Standard.

System Requirements

- IBM-compatible computer with 486 (or higher) processor
- 4 MB of RAM
- Windows 3.1 or higher
- 10 MB of available hard disk space

Distribution Design

Windows™-Based Tool for Sizing Ductwork and Water Piping. Choose the equal friction or static regain method to accurately size the ductwork needed for a new or existing air distribution. In either case, use the *Duct Design* portion *Distribution Design* to create a complete bill of air-side material, from the fan to the diffusers, that simplifies installation cost estimates. To save valuable design time, *Duct Design* interfaces with Trane’s *Trace® Load 700* design-and-analysis program and the *VariTrane®* air terminal selection program. It also contains a table of ASHRAE fittings and a computerized version of the Trane Ductulator®.

Similarly, the *Water Piping* portion of *Distribution Design* facilitates system piping design and allows the user to optimize the piping layout for cost and operating savings.

System Requirements

- IBM-compatible computer with 386 (or higher) processor and math coprocessor
- 4 MB of RAM
- 400 KB of available hard disk space

Engineering Toolbox

Useful “Calculators” for HVAC System Designers. The *Engineers Toolbox* is a selection of five small-but-powerful calculation programs that are invaluable for HVAC design professionals. This software suite includes:

- *Diskette Ductulator*, an electronic version of the Trane Ductulator®
- *PFC Correction Calculator*, an application that calculates the trigonometric relationships between inductance and capacitance for AC electric motors
- *Properties of Air*, an electronic version of the Trane psychrometric chart
- *Properties of Fluids*, an application that accurately predicts the physical properties of typical chiller mixtures (e.g., water and glycol) and nine refrigerants

- *Refrigerant Line Sizing*, an application that combines refrigerant properties and piping design fundamentals

System Requirements. DOS-based programs:

- IBM AT-compatible computer with math coprocessor
- 640 KB of RAM
- DOS 3.1 or higher
- 278 KB of available hard disk space

Windows[™]-Based Programs

- IBM-compatible computer with 386 (or higher) processor and math coprocessor
- 4 MB of RAM
- Windows 3.1 or higher
- 10 MB of available hard disk space

1.2.3.2 Economics Tools

System Speculator[™]

Comparative System Cost Estimates. *System Speculator[™]*, with its easy-to-use Windows[™] interface, helps users of all HVAC experience levels make quick, educated cost comparisons of various systems. The program estimates installation, operating and annual maintenance costs for multiple combinations of air distribution system and equipment combinations.

System Requirements

- IBM-compatible computer with 386 (or higher) processor and math coprocessor
- 4 MB of RAM
- Windows 3.1 (or higher)
- 4 MB of available hard disk space

TRACE[®] Economics

Life-Cycle Cost Analysis Software. *TRACE Economics*, a companion to *TRACE[®] 600*, accurately predicts the life-cycle cost, payback period and internal rate of return associated with a particular HVAC system. Based on energy consumption and utility rate structures (including “stepped” and “time-of-day” rates), the program’s calculations also accounts for depreciation and replacement costs.

System Requirements

- IBM AT-compatible computer (or better) with math coprocessor
- 640 KB of RAM
- 18 MB of available hard disk space (10 MB for the program, 7.5 MB for run-time files)
- DOS 3.1 or higher

Equipment Economics[®]

HVAC Economic Feasibility Program. With *Equipment Economics[®]*, the user can quickly perform an economic analysis that compares several equipment alternatives when the load profile is already known or only general building information is available. The program can model equipment and control strategies based on utility rates to calculate life-cycle costs and payback periods.

System Requirements

- IBM AT-compatible computer (or better) and math coprocessor
- 640 KB of RAM
- 15 MB of available hard disk space (10 MB for the program, 5 MB for run-time files)
- DOS 3.1 or higher

Chiller Economics

Chiller Plant “Cost Estimator.” Some users need to quickly estimate the cost of operating different chillers systems, and seldom model complex building geometries and air-side systems. *Chiller Economics* is a specific-purpose software program capable of modeling advanced chiller plant configurations and control strategies, including chiller sequencing, free cooling, thermal storage and building automation system optimization strategies.

System Requirements.

- IBM AT-compatible computer (or better) with math coprocessor
- 640 KB of RAM
- 200 KB of available hard disk space
- DOS 3.1 or higher

FANMOD

Cost Estimating Program for Fans and Air Handlers. The energy used to distribute air through ductwork is often a significant portion of a building's overall energy consumption. *FANMOD* is another specific-purpose tool that allows the user to quickly estimate the cost of operating different fan and air-handling systems. The program can model options such as frequency inverters, inlet vanes and motor sizes, and can be used to determine the optimum air modulation method for a particular application.

System Requirements.

- IBM AT-compatible computer (or better) with math coprocessor
- 640 KB of RAM
- 200 KB of available hard disk space
- DOS 3.1 or higher

1.2.3.3 Equipment Selection Tools

Equipment Selection Programs

Automated Product Selections. A number of equipment selection programs are available at *no charge* to save designers valuable time and encourage comparison of a wide variety of options. With these tools, the user can avoid countless hours spent locating the catalog data and performing the necessary calculations (and re-calculations) by hand. The programs used to select the following equipment include sound power data and allow the user to make multiple selections:

- Modular Climate Changers® air handlers
- Chilled water coils
- Hot water coils
- Refrigerant coils
- Steam coils
- Refrigerant heat-recovery coils
- Model Q® vaneaxial fans
- Centrifugal and propeller fans (“Fan B”)
- Commercial self-contained air conditioners
- Large commercial rooftop air conditioners
- Fan-coil terminal units
- VariTrane® variable-air-volume terminal units

System Requirements. DOS-based programs:

- IBM AT-compatible computer with math coprocessor
- 640 KB of RAM
- DOS 3.1 higher
- 5.1 MB of available hard disk space

Windows®-based programs:

- IBM-compatible computer with 386 (or higher) processor and math coprocessor
- 4 MB of RAM
- Windows 3.1 or higher
- 10 MB of available hard disk space

CAD Equipment Templates

Plan-View, “To Scale” Drawings of Equipment. Trane provides undimensioned AutoCAD® equipment templates that can be inserted, to scale, into system schematics. The templates are provided at no charge, and are compatible with AutoCAD DOS Releases 10, 11 and 12 and AutoCAD Release 12 for Windows®. They are also available in a 2-D drawing exchange format, .DXF, so that they can be used with other CAD programs.

The package includes 2-D and 3-D templates of a wide variety of Trane equipment and a documentation diskette with installation instructions.

System Requirements

- 3 MB of available hard disk space (2 MB for 2-D template files, 1 MB for 3-D files)

1.2.4 CARRIER PROGRAMS *

Carrier's E20-II programs are available to assist HVAC engineers in the layout and design of commercial air conditioning systems. This section summarizes the features and capabilities of each E20-II program. (Ref. 44)

Hourly Analysis Program v3.20. Advanced system-based HVAC design load program AND full 8760-hour-per-year energy analysis program. System-based design loads is a technique which considers specific HVAC system features when performing load estimating and system sizing calculations.

- System-based design loads of all common HVAC systems for sizing and selecting fans, central cooling and heating coils, air terminal equipment, space heating coils, preheat coils, and central chillers and boilers.
- Performs detailed 8760-hour-per-year simulation of airside and plant equipment.
- Uses ASHRAE-endorsed Transfer Function method and heat extraction procedure.
- Uses ASHRAE clear sky solar algorithms.
- Analyzes sloped roofs and skylights.
- Permits hourly scheduling of lights, occupancy, electrical equipment and other miscellaneous loads.
- Analyzes chiller networks.
- Analyzes thermal storage systems.
- Analyzes complex electric and gas utility rates, including demand charges.
- Contains weather library of over 500 cities worldwide.
- Provides data for common wall and roof constructions, and common windows.
- Built-in transfer function coefficient generator.
- Storage for 1200 spaces, 250 air systems, 100 plants, and 20 entire buildings.

Block Load v2.12. HVAC load estimating program suitable for commercial buildings of any size. Handles everything from simple rooftop jobs to 150-zone central air handlers.

- Load analysis uses the ASHRAE-endorsed Transfer Function method.
- Contains weather library of over 500 cities worldwide.
- Provides selection information for coils fans and terminal diffusers.
- Provides detailed breakdown of zone and system loads, and handy 'rule-of-thumb' check figures.

Duct Design v3.24. Used to design duct systems based on the latest ASHRAE & SMACNA standards.

- Static regain or equal friction sizing methods.
- Supply and return duct systems.

*This section courtesy of the Carrier Corp., Syracuse, NY.

- Up to 500 sections per duct system.
- Round, rectangular, flat oval and flex duct.

Refrigerant Piping Design v3.00. Determines the minimum pipe size required to deliver refrigerant between the compressor, condenser, and evaporator. The program will also size risers so that oil entrainment is ensured.

- Sizes suction, hot gas discharge and liquid lines.
- Sizes single and double vertical risers.
- Handles steel or copper tube.
- Sizes piping for refrigerants R-12, R-22, R-500, R-502 and R-717.

Water Piping Design v3.03. Used to design well-balanced water piping systems. It allows the designer to look at the balancing required for each piping section.

- Allows up to 200 piping sections per analysis.
- Handles closed or open systems.
- Handles steel, copper, or plastic pipe.
- Analyzes water or ethylene glycol.
- Up to 35 different pipe sizes.

Engineering Economic Analysis v2.10. Provides tools for evaluating the long-term economic performance of building and HVAC system designs. The software permits consideration of investment and operating costs, investment financing methods, and rates of cost escalation.

- Calculates payback, cash flow, and savings-to-investment ratio.
- Up to three different financed investments can be considered.
- Costs for maintenance and four types of fuel may be evaluated.

Bin Operating Cost Analysis v2.11. Calculates annual operating costs for commercial HVAC and non-HVAC energy consuming systems. The modified bin method is used to provide quick, accurate results.

- Considers costs for air system fans, cooling and heating plants, pumps, lights, miscellaneous equipment and machinery, and domestic water heating systems.
- Contains weather library of over 300 cities in North America.
- Handles interior and perimeter regions of a building.

Applied Acoustics v1.10. Engineering tool which uses ASHRAE and ARI-endorsed procedures to determine the acoustic quality of indoor and outdoor spaces. It estimates the sound pressure level at a receiver location in response to one or more sound sources.

- Computes Noise Criteria, Room Criteria and A-Weighted Sound Level (dBA) ratings.
- Ability to analyze sound levels in indoor or outdoor spaces.

1.2.5 REFERENCES

1. 1995 *ASHRAE Handbook, Fundamentals*, ASHRAE, 1791 Tullie Circle N. E. Atlanta, GA, 30329, chap. 24, "Weather Data."
2. Carrier Corporation, *Handbook of Air Conditioning System Design*, McGraw-Hill, New York, 1965, part 1, chap. 2.
3. Loads Design Weather Region diskettes from the Trane Company, La Crosse, WI.
4. E20-II diskettes from Carrier Corp., Syracuse, NY.
5. National Climatic Data Center, Nashville, NC.
6. 1993 *ASHRAE Handbook, Fundamentals*, chap. 8, "Physiological Principles and Thermal Comfort," ASHRAE, Atlanta, GA, 30329.
7. *Ibid.*, chap. 23, "Infiltration and Ventilation."
8. *Ventilation Standard*, ANSI/ASHRAE document 61-1981R, ASHRAE, 1791 Tullie Circle N. E. Atlanta, GA, 30329.
9. 1995 *ASHRAE Handbook, HVAC Applications*, ASHRAE, 1791 Tullie Circle N. E. Atlanta, GA, 30329, chap. 2, "Retail Facilities."
10. *Ibid.*, chap 3, "Commercial and Public Buildings."
11. *Ibid.*, chap 4, "Places of Assembly."
12. *Ibid.*, chap 5, "Domiciliary Facilities."
13. *Ibid.*, chap 6, "Educational Facilities."
14. *Ibid.*, chap 7, "Health Care Facilities."
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16. *Ibid.*, chap 10, "Ships."
18. *Ibid.*, chap 13, "Laboratory Systems."
19. *Ibid.*, chap 15, "Clean Spaces."
20. *Ibid.*, chap 16, "Data Processing System Areas."
21. Carrier Corp., *Handbook of Air Conditioning System Design*, part 1, chap. 1, McGraw-Hill, New York, 1965.
22. *Ibid.*, chap. 6.
23. 1993 *ASHRAE Handbook, Fundamentals*, chapter 25, "Residential Cooling and Heating Load Calculations." Chapter 26, "Non residential Cooling and Heating Load Calculations." ASHRAE, 1791 Tullie Circle N. E. Atlanta, GA, 30329.
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26. *Ibid.*, chap. 10, "Physiological Factors in Drying and Storing Farm Crops."
27. 1995 *ASHRAE Handbook, Applications*, chap 11, "Industrial Air Conditioning," ASHRAE, 1791 Tullie Circle N. E. Atlanta, GA, 30329.
28. *Ibid.*, chap 14, "Engine Test Facilities."
29. *Ibid.*, chap 17, "Printing Plants."
30. *Ibid.*, chap 18, "Textile Processing."
31. *Ibid.*, chap 19, "Photographic Materials."
32. *Ibid.*, chap 20, "Environment Control for Animals and Plants."
33. *Ibid.*, chap 22, "Air Conditioning of Wood and Paper Products Facilities."
34. *Ibid.*, chap 23, "Nuclear Facilities."

35. Ibid., chap 25, "Mine Air Conditioning and Ventilation."
36. 1993 *ASHRAE Handbook, Fundamentals*, Chapter 25, "Residential Cooling and Heating Load Calculations." Chapter 26, "Non residential Cooling and Heating Load Calculations," ASHRAE, 1791 Tullie Circle N. E. Atlanta, GA, 30329.
37. Ibid., chap 3, "Heat Transfer."
38. Ibid., chap. 27, "Fenestration."
39. Carrier Corp., *Handbook of Air Conditioning System Design*, part 1, chap. 5, McGraw-Hill, New York, 1965.
40. 1995 *ASHRAE Handbook, Fundamentals*, chap. 12, "Enclosed Vehicular Facilities," ASHRAE, 1791 Tullie Circle N. E. Atlanta, GA, 30329.
41. Carrier Corp., *Handbook of Air Conditioning Systems Design*, part 1, chap. 3, McGraw-Hill, New York, 1965.
42. Ibid., chap. 4.
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44. Carrier Software Programs for HVAC, Carrier Corp., Syracuse, NY.

SECTION 2

DESIGN CONSIDERATIONS

CHAPTER 2.1

APPLICATIONS OF HVAC SYSTEMS*

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2.1.1 GENERAL CONSIDERATIONS

As a system design develops from concept to final contract documents, the following subjects (in Secs. 2.1.1–2.1.11) should be considered throughout the HVAC design period.¹ These subjects are of a general nature inasmuch as they are applicable to all HVAC designs, and they may become specific requirements inasmuch as codes are continually updated.

2.1.1.1 Cooling Towers and Legionnaire's Disease

Since the 1976 outbreak of pneumonia in Philadelphia, cooling towers have frequently been linked with the *Legionella pneumophila* bacteria, or Legionnaires' disease. Much is yet to be learned about this bacteria, but until it is known to be eliminated, several precautions should be taken:

1. Keep basins and sumps free of mud, silt, and organic debris.
2. Use inhibitors as recommended by water-treatment specialists. Do not overfeed, because high concentrations of some inhibitors are nutrients for microbes.
3. Do not permit the water to stagnate. The water should be circulated throughout the system for at least 1 h each day regardless of the water temperature at the

*Updated for this second edition by Alfred W. Woody, Chief Mechanical Engineer, Giffels Associates, Inc., Southfield, Michigan.

¹The preliminary design, calculations, equipment, and control of heating, ventilating, and air-conditioning (HVAC) systems are discussed in other chapters.

tower. The water temperature in indoor piping will probably be 60°F (15.6°C) or warmer, and one purpose of circulating the water is to disperse active inhibitors throughout the system.

4. Minimize leaks from processes to cooling water, especially at food plants. Again, the processes may contain nutrients for microbes.

2.1.1.2 Elevator Machine Rooms

These spaces are of primary importance to the safe and reliable operation of elevators. In the United States, all ductwork or piping in these rooms must be for the sole purpose of serving equipment in these rooms unless the designer obtains permission from the authorities in charge of administering ANSI Standard 17.1, *Safety Code for Elevators and Escalators*. If architectural or structural features tend to cause an infringement of this rule, the duct or pipe must be furred in and enclosed in an approved manner.

2.1.1.3 Energy Conservation

A consequence of the 1973 increase in world oil prices is legislation governing the design of buildings and their HVAC systems. Numerous U.S. states and municipalities include an energy code or invoke a particular issue of ASHRAE Standard 90 as a part of their building code. Standard 90 establishes indoor and outdoor design conditions, limits the overall U-factor for walls and roofs, limits reheat systems, requires the economizer cycle on certain fan systems, limits fan motor power, requires minimum duct and pipe insulation, requires minimum efficiencies for heating and cooling equipment, etc. Certain occupancies, including hospitals, laboratories, and computer rooms, are exempt from portions of the standard.

In the interest of freedom of design, the energy codes permit trade-offs between specified criteria as long as the annual consumption of depletable energy does not exceed that of a system built in strict conformance with the standard. Certain municipalities require that the drawings submitted for building-permit purposes include a statement to the effect that the design complies with the municipality's energy code. Some states issue their own preprinted forms that must be completed to show compliance with the state's energy code.

2.1.1.4 Equipment Maintenance

The adage "out of sight out of mind" applies to maintenance. Equipment that a designer knows should be periodically checked and maintained may get neither when access is difficult. Maintenance instructions are available from equipment manufacturers; the system designer should be acquainted with these instructions, and the design should include reasonable access, including walk space and headroom, for ease of maintenance. Some features for ease of maintenance will increase project costs, and the client should be included in the decision to accept or reject these features.

Penthouse and rooftop equipment should be serviceable via stairs or elevators and via roof walkways (to protect the roofing). Ship's ladders are inadequate when tools, parts, chemicals, etc., are to be carried. Rooftop air handlers, especially those

used in cold climates, should have enclosed service corridors. If heavy rooftop replacement parts, filters, or equipment are expected to be skidded or rolled across a roof, the architect must be advised of the loading to permit proper roof system design.

Truss-mounted air handlers, unit heaters, valves, exhaust fans, etc., should be over aisles (for servicing from mechanized lifts and rolling platforms) when catwalks are impractical. Locate isolated valves and traps within reach of building columns and trusses to provide a degree of stability for service personnel on ladders.

It is important that access to ceiling spaces be coordinated with the architect. Lay-in ceilings provide unlimited access to the space above, except possibly at lights, speakers, sprinklers, etc. When possible, locate valves, dampers, air boxes, coils, etc., above corridors and janitor closets so as to disturb the client's operations the least.

Piping-system diagrams and valve charts are important and should be provided by the construction documents. Piping should be labeled with service and flow arrows, and valves should be numbered, especially when not within easy view of the source (such as steam piping not being within easy view of the boiler).

For piping of approximately 3 in (7.5 cm) and larger, use only flanged or lugged valves when it is intended that the item immediately adjacent to the valve will be removed for servicing. Remember that wafer valves are unsuitable inasmuch as both pipe flanges are required to hold the valve in place (see Chap. 3.6).

Pump performance and strainer clogging can be monitored by the pressure-gauge arrangement shown in Fig. 2.1.1 or by installing pressure gauges upstream and downstream of strainers, pumps, etc. Using the readings from one gauge eliminates the suspicions caused by the inherent inaccuracies among multiple gauges. Frequently remaining serviceable for a long time, $\frac{3}{8}$ -in (10-mm) globe-pattern gauge valves are preferred to gauge cocks.

The observation of steam-trap operation can be facilitated by having a $\frac{3}{8}$ -in (10-mm) test valve at the trap discharge pipe (Fig. 2.1.2). With valve V-1 closed, trap leakage and cycling may be observed at an open test valve. The test valve can be used to monitor reverse-flow leaks at check valves.

2.1.1.5 Equipment Noise and Vibration

Noise and vibration can reach unacceptable levels in manufacturing plants as well as in offices, auditoriums, etc. Once an unacceptable level is "built in," it is very

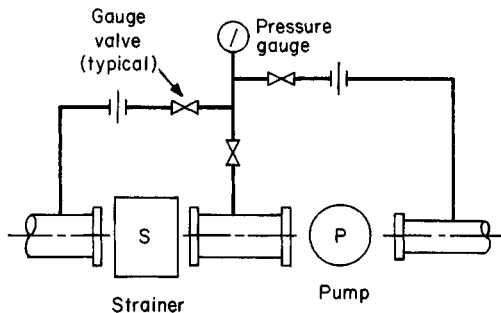


FIGURE 2.1.1 Multiple-point pressure gauge.

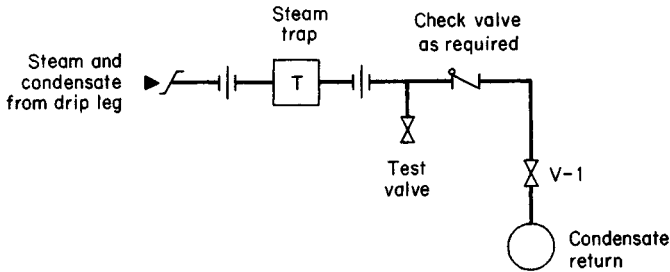


FIGURE 2.1.2 Test valve at steam trap.

costly to correct. The noise and vibration control recommendations in Chaps 8.2 and 8.3 of this book and in the *1995 ASHRAE Handbook, HVAC Applications*, should be followed. Sound and vibration specialists should be consulted for HVAC systems serving auditoriums and other sensitive areas. Fans, dampers, diffusers, pumps, valves, ducts, and pipes which have sudden size changes or interior protrusions or which are undersized can be sources of unwelcome noise.

Fans are the quietest when operating near maximum efficiency, yet even then they may require sound attenuation at the inlet and outlet. Silencers and/or a sufficient length of acoustically lined ductwork are commonly used to “protect” room air grilles nearest the fan. Noise through duct and fan sides must also be considered. In the United States, do not use acoustic duct lining in hospitals except as permitted by the U.S. Department of Health and Human Services (DHHS) Publication HRS-M-HF 84-1.

Dampers with abrupt edges and those used for balancing or throttling air flows cause turbulence in the air stream, which in turn is a potential noise source. Like dampers, diffusers (as well as registers, grilles, and slots) are potential noise sources because of their abrupt edges and integral balancing dampers. Diffuser selection, however, is more advanced in that sound criteria are readily available in the manufacturers’ catalogs. Note, however, that a background noise (or “white” noise) is preferable in office spaces because it imparts a degree of privacy to conversation. Diffusers can provide this.

Pumps are also the quietest when operating near maximum efficiency. Flexible connectors will dampen vibration transmission to the pipe wall but will not stop water- or liquid-borne noise.

Valves for water, steam, and compressed-air service can be a noise source or even a source of damaging vibration (cavitation), depending on the valve pattern and on the degree of throttling or pressure reduction. Here again, the findings of manufacturers’ research are available for the designer’s use. (See Chap. 3.6 for a discussion of cavitation in valves.)

Equipment rooms with large fans, pumps, boilers, chillers, compressors, and cooling towers should not be located adjacent to sound- or vibration-sensitive spaces. General office, commercial, and institutional occupancies usually require that this equipment be mounted on springs or vibration isolation pads (with or without inertia bases) to mitigate the transfer of vibration to the building’s structure. Spring-mounted equipment requires spring pipe hangers and flexible duct and conduit connections. Air-mixing boxes and variable-volume boxes are best located above corridors, toilet rooms, public spaces, etc. Roof fans, exhaust pipes from diesel-driven generators, louvers, etc., should be designed and located to minimize noise levels, especially when near residential areas.

2.1.1.6 Evaporative Cooling

An air stream will approach at its wet bulb temperature a 100 percent saturated condition after intimate contact with recirculated water. Evaporative cooling can provide considerable relief without the cost of refrigeration equipment for people working in otherwise unbearably hot commercial and industrial surroundings, such as laundries, boiler rooms, and foundries. Motors and transformers have been cooled (and their efficiency increased) by an evaporatively cooled air stream.

Figure 2.1.3 shows the equipment and psychrometric elements of a “direct” evaporative cooler. Its greatest application is in hot, arid climates. For example, the 100°F (38°C), 15 percent relative humidity (RH) outdoor air in Arizona could be cooled to 70°F (21°C), 82 percent RH with an 88 percent efficient unit. Efficiency is the quotient of the dry-bulb conditions shown at (2), (3), and (4) in Fig. 45.3. Note that the discharge air from a direct evaporative cooler is near 100 percent humidity and that condensation will result if the air is in contact with surfaces below its dew point. The discharge dew point in the above example is 64°F (18°C).

Figure 45.4 schematically shows an “indirect” evaporative cooler. Whereas a direct evaporative cooler increases the air stream’s moisture, an indirect evaporative cooler does not; that is, there is sensible cooling only at (1) to (2) in Fig. 2.1.4. Air is expelled externally at (5). When an indirect cooler’s discharge (2) is ducted to a direct cooler’s inlet, the final discharge (3) will be somewhat cooler and include less moisture than that of a direct cooler only. Various combinations of direct and indirect equipment have been used as stand-alone equipment or to augment refrigeration equipment for reduced overall operating costs. Refer to the *1992 ASHRAE Handbook, Systems and Equipment*, and the *1995 ASHRAE Handbook, HVAC and Applications*.

Some evaporative cooling equipment operates with an atomizing water spray only, with any overspray going to the drain. Some additional air cooling is available when the water temperature is less than the air wet-bulb temperature. Evaporative cooling involves large quantities of outdoor air, and there must be provisions to exhaust the air. Evaporative cooling has also been applied to roof cooling; a roof is wetted by fine sprays, and the water evaporation causes cooler temperatures at the roof’s upper and lower surfaces. The water supply for all applications must be analyzed for suitability and, as needed, treated to control scale, algae, bacteria, etc.

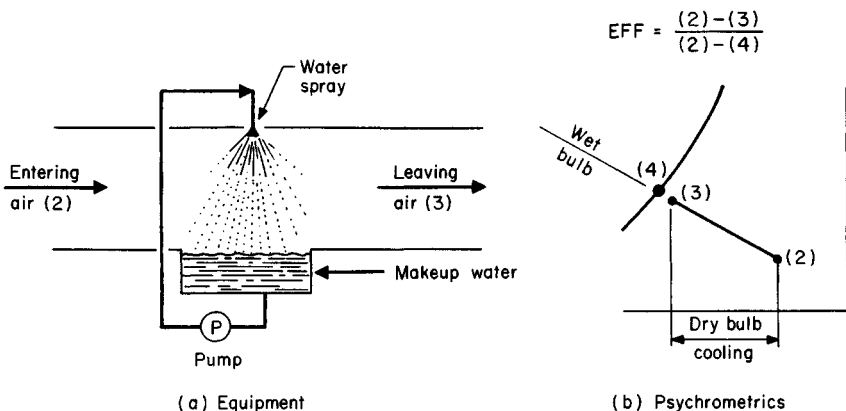


FIGURE 2.1.3 Direct evaporative cooling.

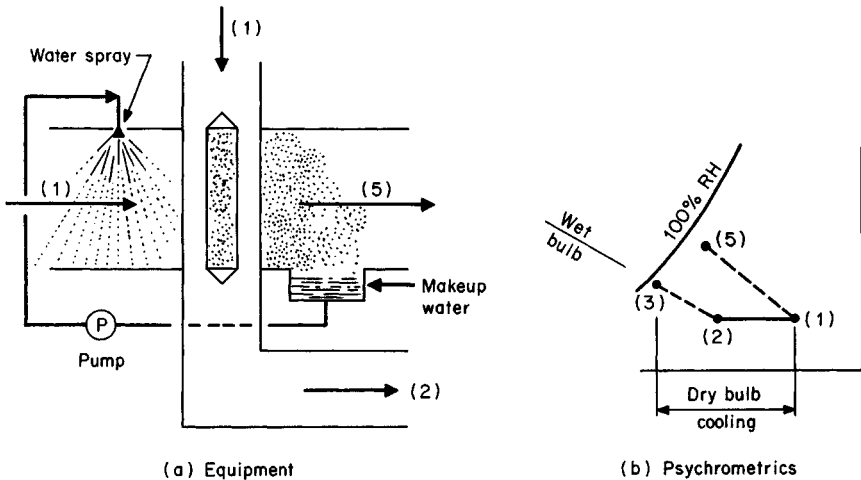


FIGURE 2.1.4 Indirect evaporative cooling.

2.1.1.7 Fire and Smoke Control Dampers

Wherever practical and/or necessary, building walls and floors are made of fire-resistant material to hinder the spread of fire. Frequently, HVAC ducts must penetrate walls and floors. In order to restore the fire resistance of a penetrated wall, fire dampers or equal protection must be provided whenever a fire-resistance-rated wall, floor, or ceiling is penetrated by ducts or grilles. Fire dampers are approved devices (approved by administrators of the building code, fire marshall and/or insurance underwriter) that automatically close in the presence of higher-than-normal temperatures to restrict the passage of air and flame. Smoke dampers are approved devices that automatically close to restrict the passage of smoke.

The following are general applications for fire or smoke dampers per the National Fire Protection Association Standard NFPA-90A, 1989 edition:

- Provide 3-h fire dampers in ducts that penetrate walls and partitions which require a 3-h or higher resistance rating, provide 1½-h dampers in ducts that penetrate those requiring a rating of 2 h or higher but less than 3 h, and provide 1½-h dampers in ducts that penetrate shaft walls requiring a rating of 1 to 2 h.
- Provide fire dampers in all nonducted air-transfer openings that penetrate partitions if they require a fire-resistance rating.
- Provide smoke dampers at air-handling equipment whose capacity exceeds 15,000 ft³/min (7080 L/s). The dampers shall isolate the equipment (including filters) from the remainder of the system except that the smoke dampers may be omitted (subject to approval by the authority having jurisdiction) when the entire air-handling system is within the space served or when rooftop air handlers serve ducts in large open spaces directly below the air handler.

Exceptions to the above are allowed when the facility design includes an engineered smoke control system. Note that schools, hospitals, nursing homes, jails, etc., may have more stringent requirements.

Dampers that “snap” closed have often incurred sufficient vacuum on the downstream side to collapse the duct (see Ref. 1). Smoke and other control dampers that close “normally” and restrict the total air flow of a rotating fan can cause pressure (or vacuum) within the duct equal to fan shutoff pressure. A fan might require a full minute after the motor is de-energized before coasting to a safe speed (pressure). Provide adequate duct construction, relief doors, or delayed damper closure (as approved by the authority having jurisdiction).

Refer to the building codes, local fire marshal rules, insurance underwriter’s rules, and NFPA-90A for criteria regarding fire and smoke dampers.

2.1.1.8 Outdoor Air

This is needed to make up for air removed by exhaust fans; to “pressurize” buildings so as to reduce the infiltration of unwanted hot, cold, moist, or dirty outdoor air; to dilute exhaled carbon dioxide, off-gassing of plastic materials, tobacco smoke, body odors, etc.; and to replenish oxygen.

A frequently used rule of thumb to provide building pressurization is to size the return fan’s air flow for 85 percent of the supply fan’s, thereby leaving 15 percent for pressurization and small toilet-exhaust makeup. This is acceptable for simple, constant-volume systems and buildings. The required outdoor air can also be established by estimating the air flow through window and door cracks, open windows and doors, curtain walls, exhaust fans, etc. Building pressurization should be less than 0.15 in water gauge [WG] (4 mm WG) on ground floors that have doors to the outside so that doors do not “hang” open from outflow of air. The building’s roof and walls must be basically airtight to attain pressurization. If there are numerous cracks, poor construction joints, and other air leaks throughout the walls, it is impractical to pressurize the building—and worse, the wind will merely blow in through the leaks on one side of the building and out through the leaks on the other side. Variable-air-volume (VAV) systems require special attention regarding outdoor air because as the supply fan’s air flow is reduced, the outdoor and return air entering this fan tend to reduce proportionately.

The National Fire Protection Association (NFPA) standards recommend minimum outdoor air quantities for hazardous occupancies. NFPA standards are a requirement insofar as building codes have adopted them by reference. Building codes frequently specify minimum outdoor air requirements for numerous hazardous and nonhazardous occupancies. ASHRAE Standard 62 recommends minimum quantities of outdoor air for numerous activities. In the interest of energy conservation, 5 ft³/min (2.4 L/s) per person had been considered acceptable for sedentary non-smoking activities, but this was later determined to be inadequate. ASHRAE 62-1989 requires at least 15 ft³/min (7.1 L/s) per person.

2.1.1.9 Perimeter Heating

The heat loss through outside walls, whether solid or with windows, must be analyzed for occupant comfort. The floor temperature should be no less than 65°F (18°C), especially for sedentary activities. In order to have comfortable floor temperatures, it is important that perimeter insulation be continuous from the wall through the floor slab and continue below per Refs. 2 and 3.

Walls with less than 250 Btu/h · lin ft (240 W/lin m) loss may generally be heated by ceiling diffusers that provide air flow down the window—unless the

occupants would be especially sensitive to cold, such as in hospitals, nursing homes, day-care centers, and swimming pools. Walls with 250 to 450 Btu/h · lin ft (240 to 433 W/lin m) can be heated by warm air flowing down from air slots in the ceiling; the air supply should be approximately 85 to 110°F (29 to 43°C). Walls with more than 450 Btu/h · lin ft (433 W/lin m) should be heated by underwindow air supply or radiation. See Ref. 4 for additional discussion. The radiant effect of cold surfaces may be determined from the procedures in ANSI/ASHRAE Standard 55.

Curtain-wall construction, custom-designed wall-to-roof closures, and architectural details at transitions between differing materials have, at times, been poorly constructed and sealed, with the result that cold winter air is admitted to the ceiling plenum and/or occupied spaces. Considering that the infiltration rates published by curtain-wall manufacturers are frequently exceeded because of poor construction practices, it is prudent to provide overcapacity in lieu of undercapacity in heating equipment. The design of finned radiation systems should provide for a *continuous* finned element along the wall requiring heat. Do not design short lengths of finned element connected by bare pipe all within a continuous enclosure. Cold downdrafts can occur in the area of bare pipe. Reduce the heating-water supply temperature and then the finned-element size as required to provide the needed heat output and water velocity.

The surface temperatures of glass, window frames, ceiling plenums, structural steel, vapor barriers, etc., should be analyzed for potential condensation, especially when humidifiers or wet processes are installed.

2.1.1.10 Process Loads

Heat release from manufacturing processes is frequently a major portion of an industrial air-conditioning load. Motors, transformers, hot tanks, ovens, etc., form the process load. If all motors, etc. in large plants are assumed to be fully loaded and to be operating continuously, then invariably the air-conditioning system will be greatly oversized. The designer and client should mutually establish diversity factors that consider actual motor loads and operating periods, large equipment with motors near the roof (here the motor heat may be directly exhausted and not affect the air-conditioned zone), amount of motor input energy carried off by coolants, etc. Diversity factors could be as much as 0.5 or even 0.3 for research and development shops containing numerous machines that are used only occasionally by the few operators assigned to the shop.

2.1.1.11 Room Air Motion

Ideally, occupied portions [or the lower 6 ft (2 m)] of air-conditioned spaces for sedentary activities would have 20- to 40-ft/min (0.1- to 0.2-m/s) velocity of air movement, with the air being within 2°F (1°C) of a set point. It is impractical to expect this velocity throughout an entire area at all times inasmuch as air would have to be supplied at approximately a 2-ft³/min · ft² (10.2-L/s · m²) rate or higher. This rate is easily incurred by the design load of perimeter offices, laboratories, computer rooms, etc., but would only occur in an inferior office when there is considerable heat-release equipment. The supply air temperature should be selected such that, at design conditions, a flow rate of at least 0.8 ft³/min · ft² (4.1 L/s · m²), but never less than 0.5 ft³/min · ft² (2.5 L/s · m²), is provided.

People doing moderate levels of work in non-air-conditioned industrial plants might require as much as a 250-ft/min (1.3-m/s) velocity of air movement in order to be able to continue working as the air temperature approaches 90°F (32°C). This would not necessarily provide a “full comfort” condition, but it would provide acceptable relief. Loose paper, hair, and other light objects may start to be blown about at air movements of 160 ft/min (0.8 m/s); see Ref. 5. Workers influenced by high ambient temperatures and radiant heat may need as much as a 4000-ft/min (20-m/s) velocity of a 90°F (32°C) air stream to increase their convective and evaporative heat loss. These high velocities would be in the form of spot cooling or of a relief station that the worker could enter and exit at will. Air movement can only compensate for, but not stop, low levels of radiant heat. Only effective shielding will stop radiant energy. Continuous air movement of approximately 300 ft/min (1.5 m/s) and higher can be disturbing to workers.

Situations involving these higher air movements and temperatures should be analyzed by the methods in Refs. 6 to 9.

2.1.2 OCCUPANCIES

2.1.2.1 Clean Rooms

For some manufacturing facilities, an interior room that is conditioned by a unitary air conditioner with 2-in- (5-cm-) thick throwaway filters might be called a “clean room”; that is, it is “clean” relative to the atmosphere of the surrounding plant. Generally, however, clean rooms are spaces associated with the micro-chip, laser optics, medical, etc., industries where airborne particles as small as 0.5 micrometer (μm) and less are removed. One micrometer equals one-millionth of a meter, or 0.000039 in (0.000001 m).

Clean rooms are identified by the maximum permissible number of 0.5- μm particles per cubic foot. For example, a class 100 clean room will have no more than 100 of these particles per cubic foot, a class 10 clean room no more than 10, etc. This degree of cleanliness can be attained by passing the air through a high-efficiency particulate air (HEPA) filter installed in the plane of the clean-room ceiling, after which the air continues in a downward vertical laminar flow (VLF) to return grilles located in the floor or in the walls at the floor. Horizontal laminar flow (HLF) rooms are also built wherein the HEPA filters are in one wall and the return grilles are in the opposite wall. A disadvantage with an HLF room is that downstream activities may receive contaminants from upstream activities.

An alternative to an entire space being ultraclean is to provide ultraclean chambers within a clean room (e.g., class 100 chambers in a class 10,000 room). This is feasible when a product requires the class 100 conditions for only a few operations along the entire assembly line.

The air-conditioning system frequently includes a three-fan configuration (primary, secondary, and makeup) similar to that shown in Fig. 2.1.5. The primary fan maintains the high air change through the room and through the final HEPA filters. The secondary fan maintains a side-stream (to the primary circuit) air flow through chilled-water or brine cooling coils, humidifiers, and heating coils. The makeup fan injects conditioned outdoor air into the secondary circuit, thereby providing clean-room pressurization and makeup for exhaust fans. Clean-room air changes are high, such that the total room air might be replaced every 7 s, and this generally results in the fan energy being the major portion of the internal heat gain. Whenever space

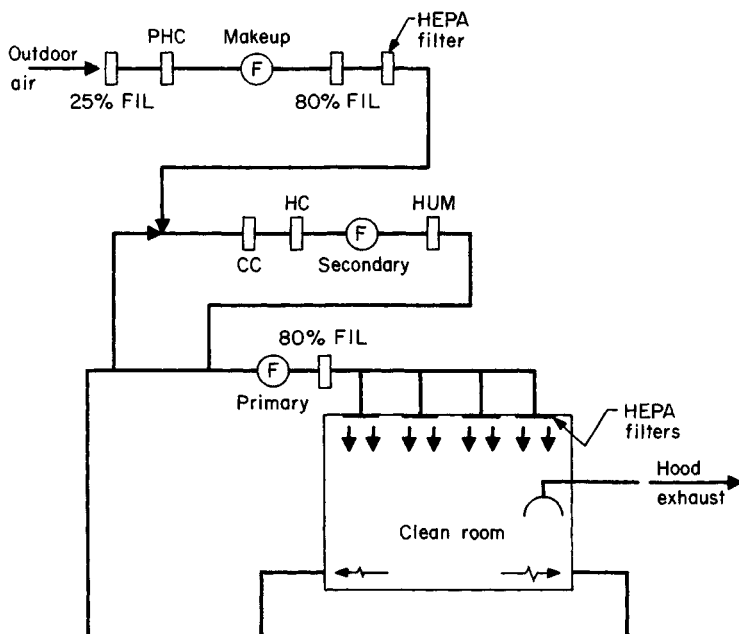


FIGURE 2.1.5 Three-fan clean-room air system.

permits, locate filters downstream of fans so as to intercept containments from the lubrication and wear of drive belts, couplings, bearings, etc.

For additional discussions, refer to the *1995 ASHRAE HVAC Handbook, Applications*, and to the latest issue of federal Standard 209, entitled *Clean Room and Work Station Requirements, Controlled Environment*.

2.1.2.2 Computer Rooms

These rooms are required to house computer equipment that is sensitive to swings in temperature and humidity. Equipment of this type normally requires controlled conditions 24 hours per day, 7 days per week. Computer equipment can be classified as (1) data processing, (2) computer-aided design and drafting (CADD), and (3) microcomputer. Microcomputers are generally similar to standard office equipment and require no special treatment. Some CADD equipment is also microcomputer-based and falls into the same category. Data processing and larger CADD systems fall into the realm of specialized computer rooms, and these are discussed below.

Data processing and large CADD systems operate on a multiple-shift basis, requiring air-conditioning during other than normal working hours. Humidity stability is of prime importance with data processing equipment and CADD plotters. The equipment is inherently sensitive to rapid changes in moisture content and temperature.

To provide for the air-conditioning requirements of computer equipment, two components are necessary: a space to house the equipment and a system to provide cooling and humidity control. Fundamental to space construction is a high-quality

vapor barrier and complete sealing of all space penetrations, such as piping, ductwork, and cables. To control moisture penetration into the space effectively, it is necessary to extend the vapor barrier up over the ceiling in the form of a plenum enclosure. Vapor-sealing the ceiling itself is not generally adequate due to the nature of its construction and to penetration from lighting and other devices.

A straightforward approach to providing conditioning to computer spaces is to use packaged, self-contained computer-room units specifically designed for the service. Controls for these units have the necessary accuracy and response to provide the required room conditions. An added advantage to packaged computer-room units is flexibility. As the needs of the computer room change and as the equipment and heat loads move around, the air-conditioning units can be relocated to suit the new configuration. The units can be purchased either with chilled-water or direct-expansion coils, as desired. Remote condensers or liquid coolers can also be provided. Large installations lend themselves quite well to heat recovery; therefore, the designer should be aware of possible potential uses for the energy.

Centrally located air-handling units external to the computer space offer benefits on large installations. More options are available with regard to introduction of ventilation air, energy recovery, and control systems. Maintenance is also more convenient where systems are centrally located. There are obvious additional benefits with noise and vibration control. Use of a centrally located system must be carefully evaluated with regard to first cost and to potential savings, as the former will carry a heavy impact.

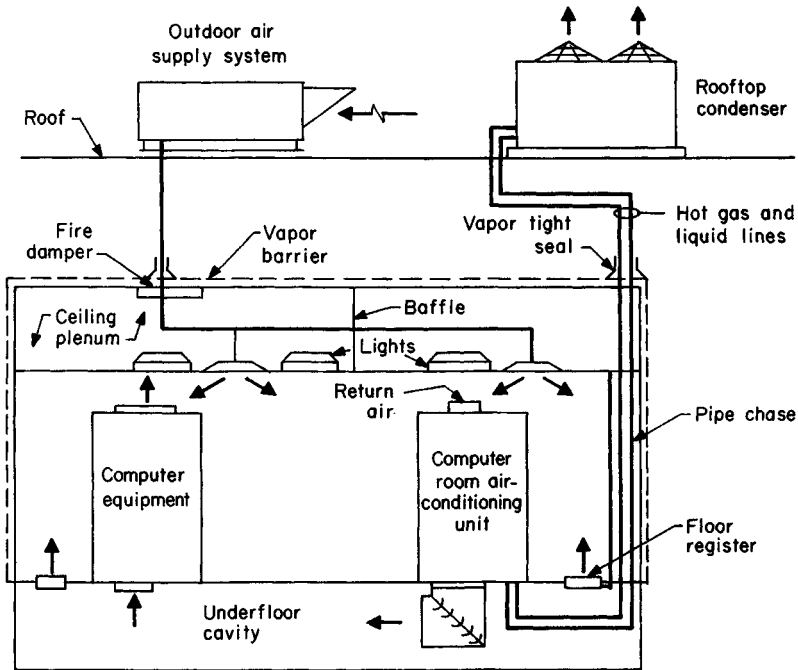
The load in the room will be primarily sensible. This will require a fairly high air-flow rate as compared to comfort applications. High air-flow rates require a high degree of care with air distribution devices in order to avoid drafts. One way to alleviate this problem is to utilize underfloor distribution where a raised floor is provided for computer cable access. A typical computer-room arrangement is shown in Fig. 2.1.6. Major obstructions to air flow below the floor must be minimized so as to avoid dead spots.

In summary, important points to remember are:

1. Completely surround the room with an effective vapor barrier.
2. Provide well-sealed wall penetrations where ductwork and piping pass into computer space.
3. Provide high-quality humidity and temperature controls capable of holding close tolerances: $\pm 1^{\circ}\text{F}$ (0.6°C) for temperature, and ± 5 percent for relative humidity.
4. Pay close attention to air distribution, avoiding major obstructions under floors where underfloor distribution is used.
5. Be alert to opportunities for energy recovery.
6. Make sure that the chosen control parameters and design temperatures and conditions satisfy the equipment manufacturer's specifications.
7. Be attentive to operating-noise levels within the computer space.
8. If chilled water or cooling water is piped to computer-room units within the computer-room space, provide a looped- or grid-type distribution system with extra valved outlets for flexibility.

2.1.2.3 Offices

Cooling and heating systems for office buildings and spaces are usually designed with an emphasis on the occupants' comfort and well-being. The designer should



1. Locate floor registers so as to be in nontraffic areas and free from obstruction
2. Ceiling plenum baffles located where and as directed by local codes and insurance underwriters

FIGURE 2.1.6 Typical computer-room layout.

remain aware that not only the mechanical systems but also the architectural features of the space affect the comfort of the occupants. And the designer will do well to remember that the mechanical system should in all respects be invisible to the casual observer.

The application of system design is divided into three parts: the method of energy transfer, the method of energy distribution, and the method of control. Controls are discussed in Chap. 8.1 and will therefore not be discussed here.

To properly apply a mechanical system to control the office environment, it is necessary to completely understand the nature of the load involved. This load will have a different character depending on the part of the office that is being served. Perimeter zones will have relatively large load swings due to solar loading and heat loss because of thermal conduction. The loading from the occupants will be relatively minor. Core zones, on the other hand, will impart more loading from building occupants and installed equipment.

For the office environment, the more common system used today is the variable-air-volume (VAV) system. This approach was originally developed as a cooling system, but with proper application of control it will serve equally well on heating. In climates where there is need for extensive heating, perimeter treatment is required

to replace the skin loss of the building structure. An old but reliable method is fin-tube radiation supplied with hot water to replace the skin loss. A system that is being seen with more regularity is in the form of perimeter air supply. Care should be taken with the application of perimeter air systems to ensure that wall U-values are at least to the level of ASHRAE Standard 90. If this is not done, interior surface temperatures will be too low and the occupants in the vicinity will feel cold.

Avoid striking the surface of exterior windows with conditioned air, as this will probably cool even double-pane glass to below the dew point of the outdoor air in the summer. The result will be fogged windows and a less-than-happy client.

In the interest of economy from a final cost and operating basis, it is best to return the bulk of the air circulated to the supply fan unit. Only enough outdoor air should be made up to the building space to provide ventilation air, replace toilet exhaust, and pressurize the building. For large office systems, it is generally more practical to return spent air to the central unit or units through a ceiling plenum. If the plenum volume is excessively large, a better approach would be to duct the return air directly back to the unit. The ceiling plenum will be warmer during the cooling season when the return air is ducted, and this will require a somewhat greater room air supply because more heat will be transmitted to the room space from the ceiling rather than directly back to the coil through the return air.

Terminal devices require special attention when applied to VAV systems. At low flow rates, the diffuser will tend to dump unless care is taken in the selection to maintain adequate throw. Slot-type diffusers tend to perform well in this application, but there are other diffuser designs, such as the perforated type, that are more economical and will have adequate performance.

The air-handling, refrigeration, and heating equipment could be located either within an enclosed mechanical-equipment room or on the building roof in the form of unitary self-contained equipment. For larger systems, of 200 tons (703 kW) of refrigeration or more, the mechanical-equipment room offers distinct advantages from the standpoint of maintenance; however, the impact on building cost must be evaluated carefully. An alternate approach to the enclosed equipment room is a custom-designed factory-fabricated equipment room. These are shipped to the job-site in preassembled, bolted-together, ready-to-run modules. For small offices and retail stores, the most appropriate approach would be roof-mounted, packaged, self-contained, unitary equipment. It will probably be found that this is the lowest in first cost, but it will not fare well in a life-cycle analysis because of increased maintenance costs after 5 to 10 years of service.

2.1.2.4 Test Cells

The cooling and heating of test cells poses many problems.

Within the automotive industry, test cells are used for:

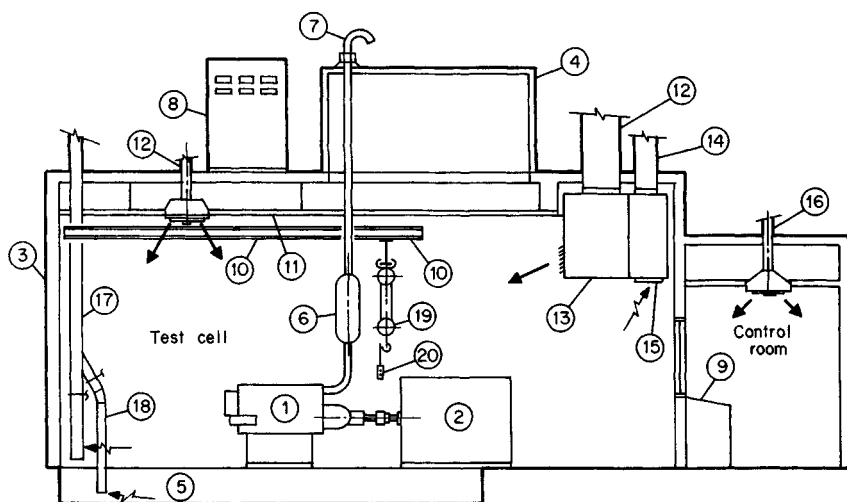
- Endurance testing of transmissions and engines
- Hot and cold testing of engines
- Barometric testing and production testing

The treatment of production test cells would be very similar to the treatment of noisy areas in other parts of an industrial environment. These areas are generally a little more open in design, with localized protection to contain the scattering of loose pieces in the event of a mechanical failure of the equipment being tested.

Hot and cold rooms and barometric cells are usually better left to a package purchase from a manufacturer engaged in that work as a specialty.

Endurance cells, on the other hand, are generally done as a part of the building package (Fig. 2.1.7). It will be found that these spaces are air-conditioned for personnel comfort during setup only. The cell would be ventilated while a test is under way. Heat gains for the nontest air-conditioned mode would be from the normal sources: ambient surroundings, lights, people, etc. Air distribution for air-conditioning would be similar to any space with a nominal loading of 200 to 400 ft²/ton (5.3 to 10.6 m²/kW) of refrigeration. It should be remembered, however, that sufficient outdoor air will be needed to make up for trench and floor exhaust while maintaining the cell at a negative-pressure condition relative to other areas. Consult local building codes to ensure compliance with regulations concerning exhaust requirements in areas of this nature.

During testing, as stated above, the cell would only be ventilated. Outdoor air would be provided at a rate of 100 percent in sufficient quantity to maintain reasonable conditions within the cell. Temperatures within the cell could often be in excess of 120°F (49°C) during a test. Internal-combustion engines are generally liquid-cooled, but even so, the frame losses are substantial and large amounts of



- | | |
|----------------------------|---|
| 1. Engine | 11. Suspended ceiling |
| 2. Dynamometer | 12. Supply air (conditioned, unconditioned) |
| 3. Blast wall | 13. Supply air plenum |
| 4. Blast cupola | 14. Cell exhaust |
| 5. Fuel and service trench | 15. Exhaust plenum |
| 6. Muffler | 16. Control room supply (conditioned) |
| 7. Engine exhaust | 17. Exhaust duct |
| 8. Dynamometer | 18. Trench exhaust duct |
| 9. Control panel | 19. Electric hoist |
| 10. Crane | 20. Hoist electric control |

FIGURE 2.1.7 Typical test-cell layout.

outdoor air will be required in order to maintain space conditions to even these high temperature limits. In cold climates, it is necessary to temper ventilation air to something above freezing; 50°F (10°C) is usually appropriate, but each situation needs to be evaluated on its own merit. The engine losses are best obtained from the manufacturer, but in the absence of this data there is information in the 1995 *ASHRAE Handbook, HVAC Applications*, that will aid in completing an adequate heat balance. The dynamometer is most often air-cooled and can be thought of as similar to an electric motor. The engine horsepower (wattage) output will be converted to electricity, which is usually fed into the building's electrical system; therefore, the dynamometer losses to the cell will be on the order of 15 to 20 percent of the engine shaft output.

The engine test cell will require a two-stage exhaust system for cooling. The first stage would be to provide low-level floor and trench exhaust to remove heavy fuel vapors and to maintain negative conditions in the cell at all times. The second stage would be interlocked with the ventilation system and would come on during testing and would exhaust at a rate about 5 to 10 percent greater than the supply rate to maintain negative-pressure conditions. The second stage would also be activated in the event of a fuel spill to purge the cell as quickly as possible. Activation of the purge should be by automatic control in the event that excessive fumes are detected. An emergency manual override for the automatic purge should be provided. Shutdown of the purge should be manual. Consult local codes for explicit requirements.

Depending on the extent of the engine exhaust system, a helper fan may be required to preclude excessive back pressure on the engine. Where more than one cell is involved, one fan would probably serve multiple cells. Controls would need to be provided to hold the back pressure constant at the engine (Fig. 2.1.8).

Air-conditioning for the test cell could be via either direct-expansion or chilled-water coils. During a test, the cell conditioning would be shut down in all areas except the control room. Depending on equipment size, it usually is an advantage to have a separate system cooling the control room. One approach to heating and cooling an endurance-type test cell is shown schematically in Fig. 2.1.9. Local building codes and the latest volumes of NFPA should be reviewed to ensure that local requirements are being met. Fuel vapors within the cell should be continually monitored. The cell should purge automatically in the event that dangerous concentrations are approached.

The following is suggested as the sequence of events for the control cycle of the test cell depicted in Fig. 2.1.9:

Setup Mode

1. AC-1 and RF-1 are running. Outdoor-air and relief-air dampers are modulated in an economizer arrangement.
2. EF-2 is controlled manually and runs at all times, maintaining negative conditions in the cell and the control room.
3. EF-1 is off and D-1 is shut.

Emergency Ventilation Mode

1. If vapors are detected, D-2 shuts and D-1 opens.
2. EF-1 starts and AC-1 changes to high-volume delivery with cooling coil shut down and outdoor-air damper open.

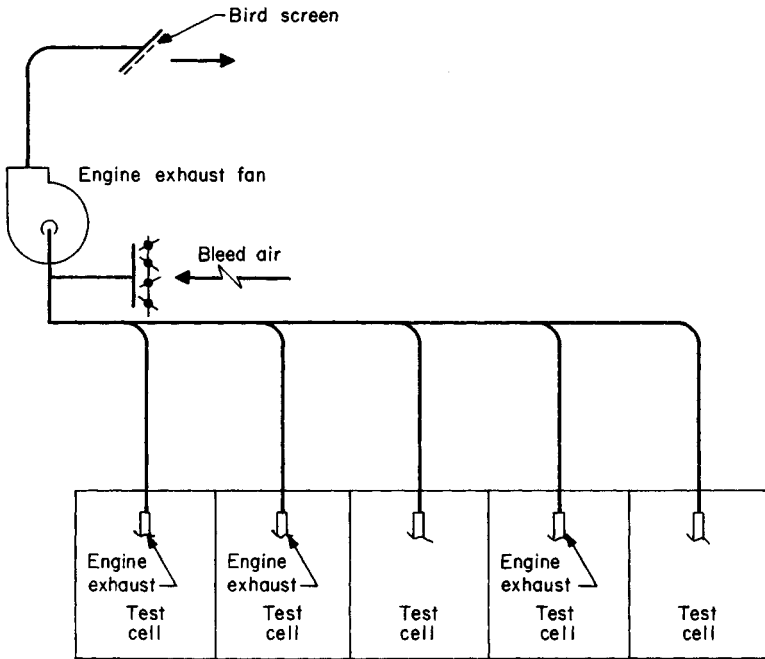


FIGURE 2.1.8 Engine exhaust helper fan.

3. HV-1 starts and its outdoor-air damper opens.
4. System should be returned to normal manually.

Test Mode

1. AC-1 cooling coil shuts down.
2. AC-1 changes to high-volume delivery with outdoor-air damper fully open. D-2 closes and D-3 opens.
3. HV-1 starts and EF-1 starts.

2.1.3 EXHAUST SYSTEMS

One of the early considerations in the design of an exhaust (or ventilation) system should be the ultimate discharge point into the atmosphere. Most of the emissions from ventilation systems are nontoxic or inert and thus will not require a permit for installation or building operation. But should the exhaust air stream contain any of the criteria pollutants—those pollutants for which emissions and ambient concentration criteria have been established, such as CO, NO_x, SO₂, lead, particulate matter (PM), and hydrocarbons (HC)—it is likely that a permit to install the system will be required.

Once it is determined that a permit will be necessary, an emissions estimate must be made to determine estimates of both uncontrolled (before a pollution con-

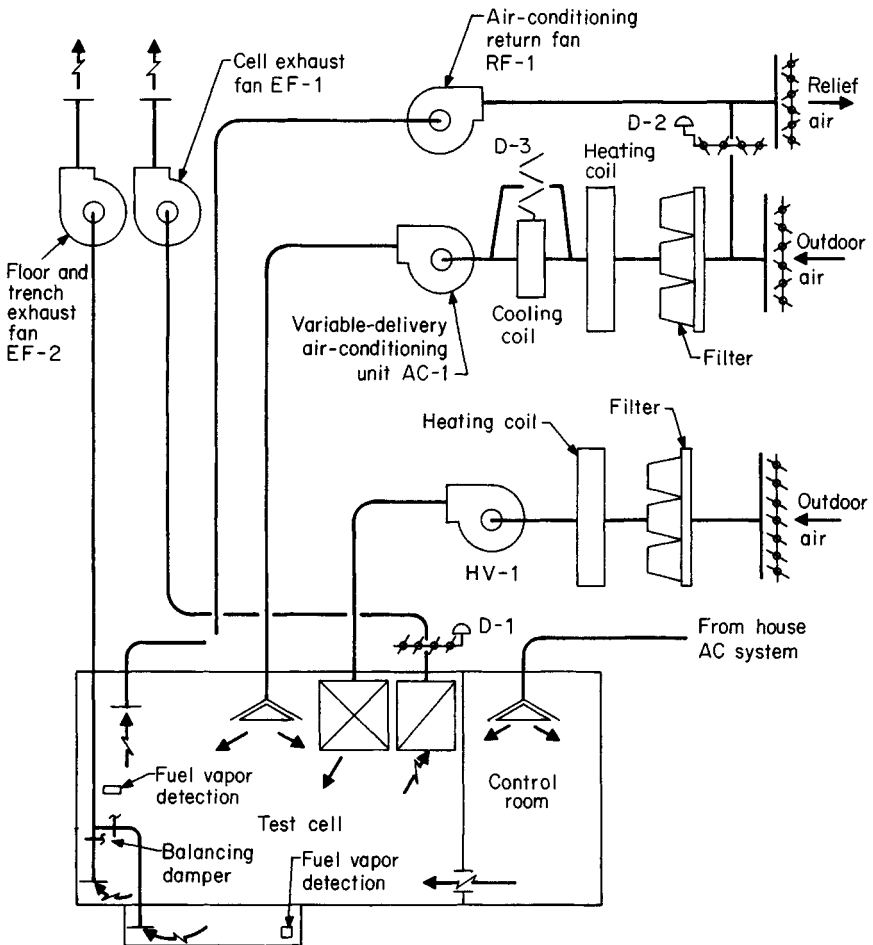


FIGURE 2.1.9 Test-cell heating, ventilating, and cooling.

trol device) and controlled emissions. The emissions estimate may be obtained from either the supplier of the equipment being contemplated for installation or from the Environmental Protection Agency (EPA) Publication AP-42, *Compilation of Air Pollutant Emission Factors*. AP-42 contains emission factors for many common industrial processes, which, when applied to process weight figures, yield emission rates in pounds (kilograms) per hour or tons per year, depending on process operating time. The permit to install an application may be obtained from the state agency responsible for enforcing the federal Clean Air Act. In most states, the Department of Environmental Protection or Department of Natural Resources will have jurisdiction. In general, the permit-to-install application requires the information and data listed in Fig. 2.1.10.

When designing an area or process exhaust system and a control system for the exhaust, it would be well to keep in mind that federal and local air-quality regulations may govern the type of emission control equipment installed and the max-

1. Applicant name and address.
2. Person to contact and telephone number.
3. Proposed facility location.
4. SIC (Standard Industrial Classification Code).
5. Amount of each air contaminant from each source in pph (pounds per hour) and tpy (tons per year) at maximum and average.
6. What federal requirement will apply to the source?
 - NESHAPS (national emission standards for hazardous air pollutants).
 - NSPS (news source performance standards).
 - PSD (prevention of significant deterioration).
 - EOP (emission offset policy).
7. Will BACT (best available control technology) be used?
8. Will the new source cause significant degradation of air quality?
9. How will the new source affect the ambient air quality standard?
10. What monitoring will be installed to monitor the process, exhaust, or control device?
11. What is the construction schedule and the estimated cost of the pollution abatement devices?

FIGURE 2.1.10 Commonly requested information for air-quality permit applications.

imum allowable emissions. The factors dictating what regulations apply include the type of process or equipment being exhausted, the type and quantity of emissions, the maximum emission rate, and the geographic location of the exhausted process. In order to determine what specific rules and regulations apply, the requirements of the U.S. Code of Federal Regulations Title 40 (40 CFR) should be understood early in the project stages so that all applicable rules may be accommodated. Should the design office lack the necessary expertise in this area, a qualified consultant should be engaged. The federal government has issued a list entitled "Major Stationary Sources." The exhaust system's designer should be acquainted with this list, for it identifies the pollutant sources governed by special requirements. Several of the more common sources are listed in Fig. 2.1.11, and 40 CFR should be consulted for the complete listing. One of the major sets of rules included in 40 CFR are the Prevention of Significant Deterioration (PSD) rules, which establish the extent of pollution control necessary for the major stationary sources.

If a source is determined to be "major" for any pollutant, the PSD rules may require that the installation include the best available control technology (BACT). The BACT is dependent on the energy impact, environmental impact, economic impact, and other incidental costs associated with the equipment. In addition, the following items are prerequisites to the issue of a permit for pollutants from a major source:

1. Review and compliance of control technology with the:
 - a. State Air Quality Implementation Plan (SIP).
 - b. New Source Performance Standards (NSPS) (see Fig. 2.1.12).
 - c. National Emissions Standards for Hazardous Air Pollutants (NESHAPs).
 - d. BACT.

1. Fossil fuel-fired generating plants greater than 250 million Btu/h (73 MW) input
2. Kraft pulp mills
3. Portland cement plants
4. Iron and steel mill plants
5. Municipal incinerators greater than 250 tons/day charging
6. Petroleum refineries
7. Fuel conversion plants
8. Chemical process plants
9. Fossil fuel boilers, or combination thereof totaling more than 250 million Btu/h (73 MW) input
10. Petroleum storage and transfer units exceeding 300,000 barrel storage
11. Glass fiber processing plants

FIGURE 2.1.11 Major stationary sources—partial list.

1. Fossil-fuel-fired steam generators with construction commencing after 8-17-71
2. Electric utility steam generators with construction commencing after 9-18-78
3. Incinerators
4. Portland cement plants
5. Sulfuric acid plants
6. Asphalt concrete plants
7. Petroleum refineries
8. Petroleum liquid storage vessels constructed after 6-11-73 and prior to 5-19-78
9. Petroleum liquid storage vessels constructed after 5-18-78 and prior to 7-23-84
10. Sewage treatment plants
11. Phosphate fertilizer industry — wet process phosphoric acid plants
12. Steel plants — electric arc furnaces
13. Steel plants — electric arc furnaces and argon decarburization vessels constructed after 8-17-83
14. Kraft pulp mills
15. Grain elevators
16. Surface coating of metal furniture
17. Stationary gas turbines
18. Automobile and light-duty truck painting
19. Graphic arts industry — rotogravure printing
20. Pressure-sensitive tape and label surface coating operations
21. Industrial surface coating: large appliance
22. Asphalt processing and asphalt roofing manufacture
23. Bulk gasoline terminals
24. Petroleum dry cleaners

FIGURE 2.1.12 New Source Performance Standards—partial list.

2. Evidence that the source's allowable emissions will not cause or contribute to the deterioration of the National Ambient Air Quality Standard (NAAQS) or the increment over baseline, which is the amount the source is allowed to increase the background concentration of the particular pollutant.
3. The results of an approved computerized air-quality model that demonstrates the acceptability of emissions in terms of health-related criteria.
4. The monitoring of any existing NAAQS pollutant for up to 1 year or for such time as is approved.

5. Documentation of the existing (if any) source's impact and growth since August 7, 1977, in the affected area.
6. A report of the projected impact on visibility, soils, and vegetation.
7. A report of the projected impact on residential, industrial, commercial, and other growth associated with the area.
8. Promulgation of the proposed major source to allow for public comment. Normally, the agency processing the permit application will provide for public notice.

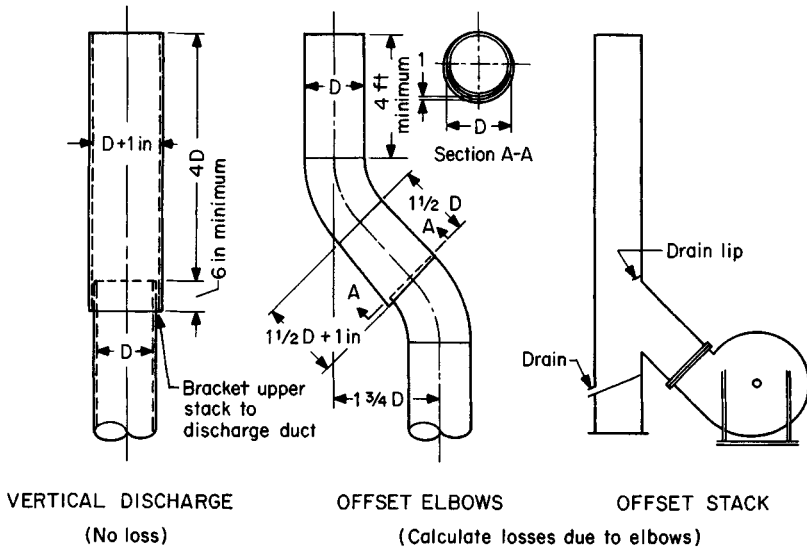
One of the first steps regarding potential pollutant sources is to determine the applicable regulations. For this, an emissions estimate must be made, and the "in-attainment" or "non-attainment" classification of the area in which the source is to be located must be determined. The EPA has classified all areas throughout the United States, including all U.S. possessions and territories. The area is classified as either "in-attainment" (air quality is better than federal standards) or "non-attainment" (air quality is worse than federal standards).

If the source is to be located in a non-attainment area, the PSD rules and regulations do not apply, but all sources that contribute to the violation of the NAAQS are subject to the Emissions Offset Policy (EOP). The following items must be considered when reviewing a source that is to be located in a nonattainment area:

1. The lowest achievable emission rate (LAER), which is defined as the most stringent emission limit that can be achieved in practice
2. The emission limitation compliance with the SIP, NSPS, and NESHAPS
3. The contribution of the source to the violation of the NAAQS
4. The impact on the non-attainment area of the fugitive dust sources accompanying the major source

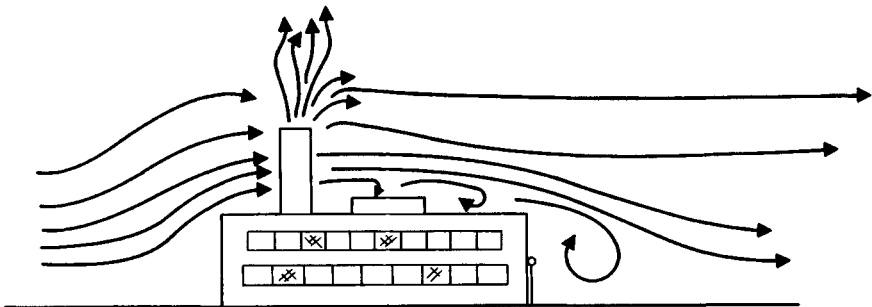
In general, the EOP requires that for a source locating in a non-attainment area, more than equivalent offsetting emission reductions must be obtained from existing emissions prior to approval of the new major source or major modification. The "bubble" concept, wherein the total emissions from the entire facility with the new source does not exceed the emissions prior to addition of the new source, may be used to determine the emission rate. If there were emission reductions at "existing" sources, they would offset the contributions from the new source, or "offset" the new emissions. This same bubble concept may be used for sources that qualify for in-attainment or PSD review.

In the design of a polluting or pollution control facility, stack design should be considered. A stackhead rain-protection device (Figs. 2.1.13 and 2.1.17) should be used in lieu of the weather cap found on many older installations, since this cap does not allow for adequate dispersion of the exhaust gas. When specifying or designing stack heights, it should be noted that the EPA has promulgated rules governing the minimum stack height; these rules are known as "good engineering practice" (GEP). A GEP stack has sufficient height to ensure that emissions from the stack do not result in excessive concentrations of any air pollutant in the vicinity of the source as a result of atmospheric downwash, eddy currents, or wakes caused by the building itself or by nearby structures (Figs. 2.1.14 and 2.1.15). For uninfluenced stacks, the GEP height is 98 ft (30 m). For stacks on or near structures, the GEP height is (1) 1.5 times the lesser of the height or width of the structure, plus the height of the structure, or (2) such height that the owner of the building



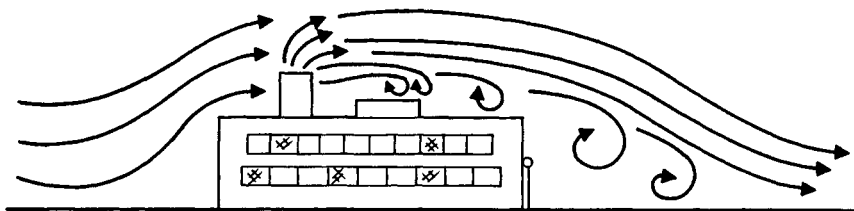
1. Rain protection characteristics of these caps are superior to a deflecting cap located 0.75D from top of stack.
2. The length of upper stack is related to rain protection. Excessive additional distance may cause "blowout" of effluent at the gap between upper and lower sections.

FIGURE 2.1.13 Typical rain-protection devices. (From *Industrial Ventilation—A Manual of Recommended Practice*, 21st ed., Committee on Industrial Ventilation, American Conference of Governmental Industrial Hygienists, copyright 1992, p. 5–53.).



GEP stack height minimizes re-entrainment of exhaust gasses into air which might enter building ventilation system.

FIGURE 2.1.14 GEP stack.



Non-GEP stack allows exhaust gasses to be entrained in building wakes and eddy currents.

FIGURE 2.1.15 Non-GEP stack.

can show is necessary for proper dispersion. In addition to GEP stack height, stack exit velocity must be maintained for proper dispersion characteristics.

Figures 2.1.16 and 2.1.17 illustrate the relationship between velocity at discharge and the velocity at various distances for the weather-cap and stackhead-type rain hoods, respectively. Maintaining an adequate exit velocity ensures that the exhaust gases will not reenter the building through open windows, doors, or mechanical ventilation equipment. Depending on normal ambient atmospheric conditions, the exit velocities may range from 2700 to 5400 ft/min (14 to 28 m/s). In practice, it has been found that 3500 ft/min (18 m/s) is a good average figure for stack exit

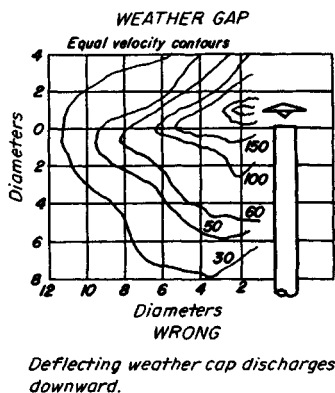


FIGURE 2.1.16 Weather-cap dispersion characteristics. (From *Industrial Ventilation—A Manual of Recommended Practice*, 21st ed., Committee on Industrial Ventilation, American Conference of Governmental Industrial Hygienists, copyright 1992, p. 5.62.)

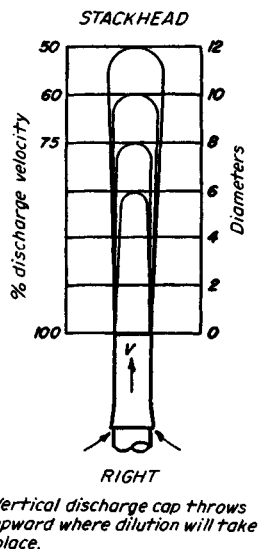


FIGURE 2.1.17 Stackhead dispersion characteristics. (From *Industrial Ventilation—A Manual of Recommended Practice*, 21st ed., Committee on Industrial Ventilation, American Conference of Governmental Industrial Hygienists, copyright 1992, p. 5.62.)

velocity, giving adequate plume rise yet maintaining an acceptable noise level within the vicinity of the stack.

Care must be taken when designing exhaust systems handling pollutants for which no specific federal emission limit exists (noncriteria pollutants). All pollutant not included in the criteria pollutant category or the NESHAPS category are considered noncriteria pollutants. When establishing or attempting to determine acceptable concentration levels for noncriteria pollutants, the local authority responsible for regulating air pollution should be consulted since policy varies from district to district. In general, however, noncriteria pollutants' allowable emission rates are based on the American Conference of Governmental Industrial Hygienists (ACGIH) time-weighted average acceptable exposure levels.

A hazardous air pollutant is one for which no ambient air-quality standard is applicable, but which may cause or contribute to increased mortality or illness in the general population. Emission standards for such pollutants are required to be set at levels that protect the public health. These allowable pollutants' emission levels are known as NESHAPS and include levels for radon-222, beryllium, mercury, vinyl chloride, radionuclides, benzene, asbestos, arsenic, and fugitive organic leaks from equipment.

An exhaust stream that includes numerous pollutants, with some being noncriteria pollutants, can be quickly reviewed by assuming that all the exhaust consists of the most toxic pollutant compound. If the emission levels are acceptable for that review, they will be acceptable for all other compounds.

2.1.4 REFERENCES

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2. *Energy Conservation in New Building Design*, ASHRAE Standard 90A-1980, ASHRAE, Atlanta, GA, p. 18, para. 4.4.2.4.
3. *1993 ASHRAE Handbook, Fundamentals*, ASHRAE, Atlanta, GA, 1993, p. 25-13, fig. 8.
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CHAPTER 2.2

HVAC APPLICATIONS FOR COGENERATION SYSTEMS

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2.2.1 INTRODUCTION

A *cogeneration facility* consists of equipment that uses energy to produce both electric energy and forms of useful thermal energy (such as heat or steam) for industrial, commercial, heating, or cooling purposes. Cogeneration facilities are designed as either *topping-cycle* or *bottoming-cycle* facilities. Topping-cycle facilities first transform fuel into useful electric power output; the reject heat from power production is then used to provide useful thermal energy. In contrast, bottoming-cycle facilities first apply input energy to a useful thermal process, and the reject heat emerging from the process is then used for power production. Either of these cycles can efficiently apply thermal energy to meet process or comfort heating, ventilating, and air-conditioning (HVAC) by generating steam, hot water or chilled water.

This chapter describes the various methods of applying thermal energy from a cogeneration system to HVAC systems.

2.2.2 HVAC APPLICATIONS FOR THERMAL ENERGY

Feasible methods for applying thermal energy to meet process, HVAC or comfort (hereafter referred to as “utility”) requirements are:

2.2.2.1 Steam or Hot-Water Absorption Chiller Units

Steam generation and mechanical drive and/or absorption chillers (Fig. 2.2.1) are cost-effective in cases where the additional steam supplements a facility’s existing steam requirements. A hot-water system rather than a steam system should be considered for facilities with requirements for hot water, as equipment required to transfer energy from steam to hot water is not required. The steam or hot-water

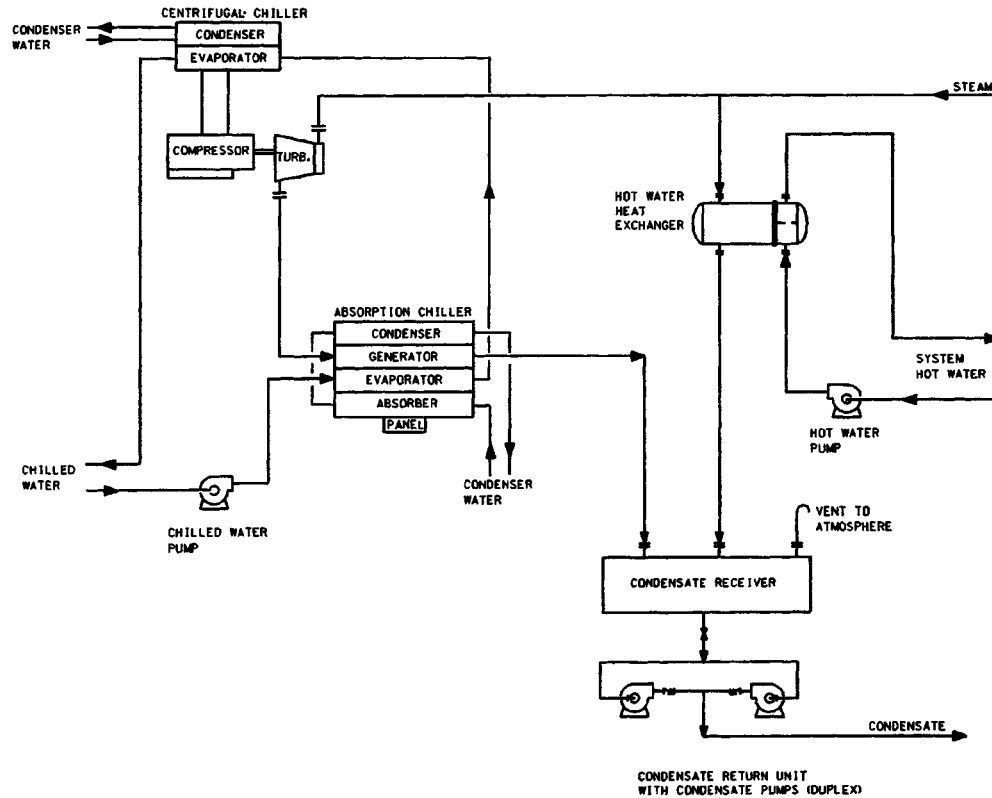


FIGURE 2.2.1 Cogeneration system utilizing steam for HVAC processes

generation system design should include a standby energy source to ensure that utility requirements are met if the cogeneration system operates at a reduced electrical generation level or suffers an unplanned outage. For maintenance purposes, the exhaust system design for the heat-recovery steam generator (HRSG) units should include guillotine seal plates and a seal air blower to isolate the HRSG, if the HRSG is required to operate with fresh air firing while maintenance is being conducted on the system's prime mover.

The HRSG typically has a minimum exhaust temperature of 250°F (121°C) when natural gas fuel is used to fire the prime mover. Maintaining this temperature protects the HRSG from water-vapor condensation and acid formation that occurs when the exhaust temperature drops below the dew point. Although an HRSG can be designed for lower exhaust temperatures, the corrosion-resistant design is not economically feasible.

The HRSG normally imposes a back pressure of 8 to 12 in water gauge (in WG) (1990 to 2985 Pa) on the prime mover exhaust. This back pressure results in a horsepower penalty of approximately 17.5 hp/in WG (1.89 W/Pa) for a combustion gas turbine rated at 4900 BHP (3653.93 kW). Within the limitations specified by internal-combustion-engine vendors, exhaust-gas back pressure does not appreciably reduce the mechanical power output of the engine.

Steam or Hot-Water Generation Control. In facilities that can use a limited amount of thermal energy from a cogeneration system, the system or hot-water production rate can be controlled by regulating the throttle of the prime mover or by bypassing the exhaust-gas heat around the heat-recovery unit and sending it up the stack. Excess steam or hot water can also be diverted to the condensers.

Using steam turbines, combined-cycle cogeneration systems use thermal energy not required for utility service to generate additional electricity.¹ One patented cycle varies its steam production rate by reinjecting high-pressure steam into the power turbine section of a combustion gas turbine. This procedure reduces the amount of steam that must be used by a facility and increases the electric power output of the unit.

Factors that affect the selection of an absorption chiller in a cogeneration system are:

- available steam or hot-water pressure and temperature
- steam consumption rate
- physical size
- machine performance under partial-load conditions

Steam and hot-water requirements for typical units are summarized in Table 2.2.1. The steam consumption rate of the two-stage machine is approximately 40 percent less than that of the single-stage machines. Condenser water requirements are also reduced more than 20 percent compared to the requirements of similar amounts of single-stage absorption chillers. These rates will even get lower when compared with the combination centrifugal-absorption chiller units, often called "piggy-back system", as described below.

Two-stage absorption machines are designed with absorbent streams using parallel or series flow. The configuration of the parallel-flow machine results in reduced

¹Combined-cycle systems simultaneously produce power using a fossil-fueled prime mover and a steam turbine generator unit.

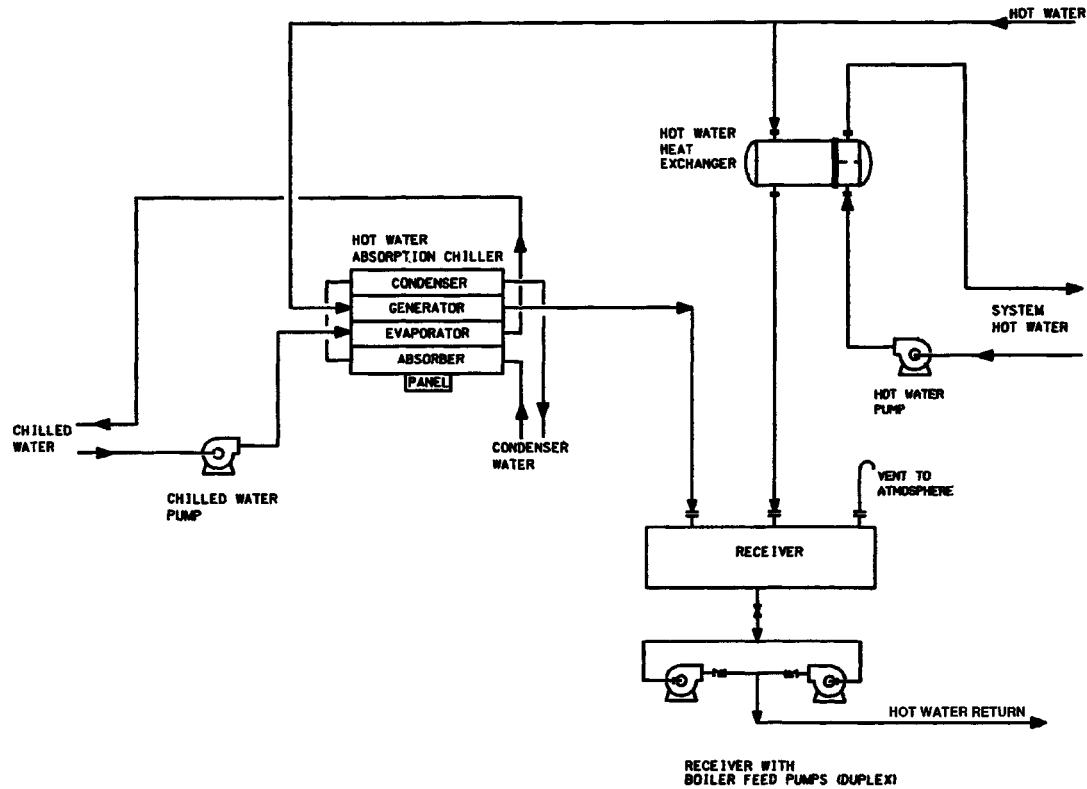


FIGURE 2.2.2 Cogeneration system utilizing hot water for HVAC processes

TABLE 2.2.1 Thermal Energy Requirements for Chillers

Chiller Type	Steam Supply Conditions		Hot-Water Supply Conditions		Nominal Steam Consumption Rate	
	psig	kPa	°F	°C	lbm/(h · ton)	kg/(s · W)
Single-stage, absorption small	—	—	160	71.1	—	—
Single-stage absorption	8–15	55.16–103.42	270	132.2	17.5–20	8.1–9.29
Two-stage absorption	100–120	689.50–827.40	300–400	148.9–204.4	9.9–12	4.6–5.5
Combination Centrifugal—absorption	600 & higher	4136.84 & higher	—	—	8.0 & lower	3.7 & lower
Ammonia Absorption, single-stage*	40–160	275.8–1103.2	—	—	30.6–57.8	14.2–26.8
Ammonia Absorption, two-stage*	1–24	6.9–165.5	—	—	47.5–67.5	22.1–31.3

*Used primarily in low temperature applications

height in all machine sizes and reduced width in larger machine sizes. Either type of machine can be installed assembled in capacities up to 750 tons (2635.7 kW). Above 750 tons (2635.7 kW), the series-flow machine must be partially assembled at the installation site, while the parallel-flow machine can be transported and installed as a single unit.

The steam utilization characteristics of absorption chillers affect their sizing in cogeneration systems. The single-stage absorption machine's electricity and steam consumption rate per ton (kW) of chilled-water production decreases with reduced load to approximately 30 percent of design capacity. At this point, consumption rises unless other cycle enhancement is added. Steam consumption curves decrease slightly at reduced-load conditions for series-flow two-stage machines. Two-stage machines using parallel flow maintain flat steam consumption curves over the entire load range. Occasionally ammonia absorption machines are used in low temperature applications for cold storage or freezer storage warehouse use.

Combination Centrifugal-Absorption Chiller Units. Noncondensing (back-pressure) steam turbines driving mechanical chillers can be used in series with conventional single stage absorption chillers by matching steam flow rates and exhaust pressure from the steam turbine (Fig. 2.2.1). This type of system (piggy-back system) must always run as a pair. The traditional distribution of chiller capacity is one-third of the tonnage for the mechanical-drive chiller and two-thirds of the tonnage for absorption chiller. At higher steam pressures the capacity distribution may approach 50% tonnage for each type of system.

Typical steam inlet pressure for non-condensing steam turbines is at least 400 lb/in² (2757.9×10^3 Pa), with exhaust steam pressure approximately 8 lb/in² (55.1×10^3 Pa). Figure 2.2.4 illustrates the range of inlet steam pressures and flows commonly used with non-condensing steam turbines. The typical steam consumption rate for steam turbines which power mechanical-drive centrifugal chillers ap-

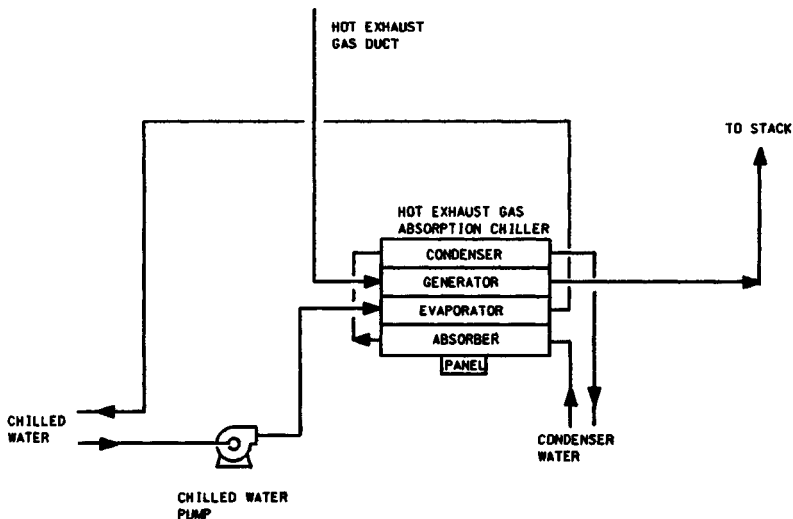


FIGURE 2.2.3 Cogeneration system utilizing hot exhaust-gas for HVAC processes

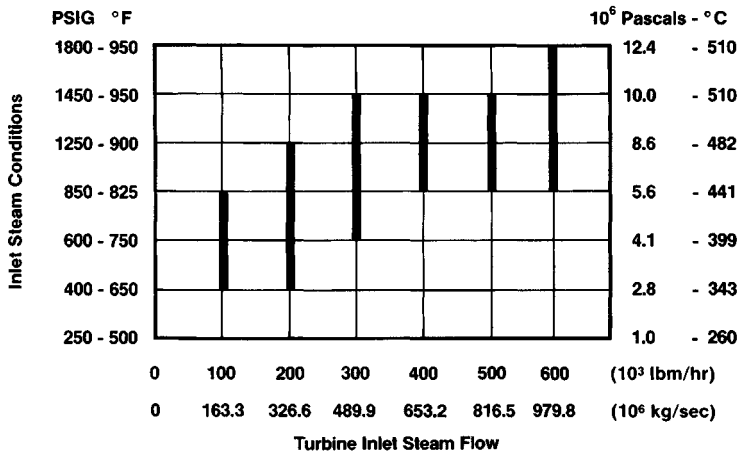


FIGURE 2.2.4 Range of initial steam conditions normally selected for industrial steam turbines

proach the two-stage absorption machines. Non-condensing steam turbines enhance the energy efficiency of a cogeneration cycle, because exhaust steam can be used for other heating or absorbing processes. For example, the exhaust steam can be used for a steam absorption chiller rather than being exhausted to the facility's condenser (Fig. 2.2.1). A typical chilled water piping system in a combination centrifugal-absorption chiller system (piggy-back system) connects the pair in series, allowing chilled water to flow first through the absorption chiller and then the centrifugal chiller. This arrangement allows the absorption chiller to operate at a higher chilled water supply temperature, thus causing less operational problems associated with lower evaporator temperatures.

Exhaust-Gas-Driven Chiller-Heater Units

A modification of the two-stage parallel-flow absorption chiller permits driving the chiller with high-temperature exhaust gas from a combustion gas turbine or an internal-combustion engine (Fig. 2.2.3). Moreover, the chillers can be purchased with an additional second-stage heat exchanger that converts the thermal energy contained in an internal-combustion engine's jacket cooling water into additional chilled-water capacity.

Exhaust-gas chillers simultaneously produce chilled water and hot water. The units can be equipped with supplemental firing (90 percent efficiency) to add energy to the exhaust gas as well as maintain utility service if the prime mover fails.

The use of exhaust-gas chillers eliminates the need for a steam or hot-water generation system and its associated condensate feedwater system. Consequently, layout space and maintenance requirements are substantially reduced, compared to the conventional steam or hot-water systems.

The exhaust-gas system design between the prime mover and the exhaust-gas chiller should include an effective bypass damper or guillotine seal plates combined with a seal air blower. The guillotine seal plates isolate the chiller from prime mover exhaust gas during chiller maintenance operations. If a bypass damper is used with-

TABLE 2.2.2 Operating Parameters for Exhaust-Gas Chiller

Parameter	Exhaust Gas	Jacket Water
Coefficient of performance	1.14	0.6–0.7
Interconnection efficiency	0.95	0.95
Minimum temperature	550°F (287.8°C)	180°F (82.2°C)
Stack temperature	375°F (max.) (190°C)	—
Jacket-water temperature difference	—	10–20°F (5.6–11.1°C)

Source: Courtesy of York International.

out seal plates, the user should verify that the damper has performed successfully in similar service. The position of the bypass damper should also be indicated directly, to aid operation by confirming the exhaust-gas flow path.

Typical heat-recovery parameters for the exhaust-gas chillers are summarized in Table 2.2.2. The thermal energy used by the exhaust-gas chiller and its resulting cooling capacity are then:

$$Q = MC_p(T_1 - T_2) \quad (2.2.1)$$

$$Q_{hx} = MjC_{pj}(T_1 - T_2) \quad (2.2.2)$$

where Q = heat removed from exhaust gas

Q_{hx} = heat removed from jacket water²

M = exhaust-gas flow rate

Mj = jacket-water mass flow rate²

C_p = exhaust-gas specific heat

C_{pj} = jacket-water specific heat²

T_1 = entering temperature

T_2 = exiting temperature

$$\begin{aligned} \text{Cooling capacity} &= \text{exhaust-gas cooling capacity} + \text{jacket-water cooling capacity} \\ &= (1.14 \times 0.95 \times Q) + (0.6 \times 0.95 \times Q_{hx}) \end{aligned} \quad (2.2.3)$$

2.2.3 OPERATIONAL CRITERIA

Electricity demand and process energy demand (chilled water, hot water, and steam) establish sizing and operating criteria for a cogeneration system. These data must be examined over specific periods of time (seasonally, weekly, daily, and even hourly in some cases) to establish a specific cyclic pattern for the energy.

The specific components and sources of the demand must be known. Careful consideration should be given to the decrease in a facility's electricity requirements if electric-driven centrifugal chillers are to be replaced by steam absorption units as part of the cogeneration system.

²Jacket-water heat recovery is associated with internal-combustion engines.

Typical operational criteria that could result from process data are:

- The facility will be able to efficiently use thermal energy produced by the prime movers.
- The cogeneration facility will supply the base electric load.
- The cogeneration facility will engage in interchange sales with the local utility.

The decision to engage in interchange sales of electricity to the interconnected utility should be studied. The capital cost associated with compliance with utility interconnection standards may exceed the revenue obtained from selling a small amount of power to the interconnecting utility.

Typical ranges for the electric power generation capacity of industrial, institutional, residential, and commercial cogeneration systems are summarized in Table 2.2.3. Industrial and institutional facilities can achieve significant economic benefit from cogeneration systems due to their balanced requirements for electric and thermal energy.

TABLE 2.2.3 Typical Cogeneration System Electric Power Generation Capacities

Application	Electrical Output, kW
One and two family homes	5–15
Multifamily dwellings	20–5,000
Office-buildings	2000–10,000
Local shopping centers	100–250
Distribution centers	250–2,500
Regional shopping centers	5,000–15,000
Industrial institutional facilities	Site dependent

Source: Richard Stone, "Stand Alone Cogeneration By Large Building Complexes,"

Energy Economics, Policy and Management (Fairmont Press, Atlanta), vol. 62, Summer 1982.

2.2.4 FUEL

The selection of a cogeneration system's fuel supply and an assessment of the system's economic viability are affected by fuel supply reliability and by projections of future fuel prices. Fuel choice also affects the heat-recovery equipment design downstream of the prime mover. The HVAC unit or the HRSG heat transfer surface design must be compatible with constituents contained in the prime mover exhaust gases.

2.2.4.1 Fuel Supply Reliability

Factors useful in assessing fuel supply reliability include:

- assurances from the supplier that fuel supplies are adequate
- identification of alternative fuel sources, including provisions to use them in the system design (No. 2 fuel oil or natural gas)
- identification of alternative means of providing utility services (a standby electric-motor-driven chiller or steam generation from another source)

2.2.4.2 Fuel Price Forecasts

The economic benefit of a cogeneration system may be determined through comparison of the total cost associated with a cogeneration system and the cost of providing similar services using electricity purchased from the existing utility. Electricity cost projections are required in order to make this comparison. Rate structure information required for this task can be obtained from both the electric service contract between the facility and the utility and form 10K that the utility files with the Securities and Exchange Commission.

Form 10K can supply data useful in establishing a demand component and a fuel component in the rate structure, such as:

- present and future fuel mixture
- historical fuel cost
- projected capital requirements

Industry trade groups and government organizations are also valuable sources for obtaining fossil-fuel cost, availability, and demand data. Publications prepared by the U.S. Department of Energy provide sample methodology for making these projections.

Additionally, federal regulations regarding fuel pricing can materially affect the fuel selection process. For example, the natural-gas pricing structure has changed as a result of the 1981 Federal Energy Regulatory Commission (FERC) Order 319, which authorized transportation services for up to five (5) years of natural gas purchased from sources other than pipeline companies. Using this program, “high-priority users”—schools and hospitals—have achieved energy cost savings ranging from 20 to 45 percent, depending on wellhead prices and transportation costs.

2.2.4.3 Heat-Recovery Equipment

Fuels having large amounts of particulate or corrosive substances may require special handling, such as a washing system. This will ensure proper heat transfer across surfaces inside the recovery equipment by avoiding excessive fouling.

2.2.5 PRIME MOVERS

Combustion gas turbines and internal-combustion engines are the prime movers used in topping cycles. Typical thermal energy temperatures are summarized in Table 2.2.4.

TABLE 2.2.4 Typical Waste-Heat Temperatures

Thermal Energy Source	Gas Turbine		Internal-combustion Engine	
	°F	°C	°F	°C
Exhaust gas	900–1000	482–537.8	1000–1200	537.8–648.9
Lube oil	165 (max.)	73.8	160–200	71.1–93.3
Jacket water	—	—	180–250	82.2–121.1

2.2.5.1 Combustion Gas Turbine Generators

Combustion gas turbine generator (CGTG) units exhibit the following characteristics in a cogeneration system:

- High temperature of exhaust gas
- High quantity of exhaust gas

With thermal energy recovery, the overall cycle energy efficiency of a CGTG unit typically exceeds 60 percent. Common types of heat-recovery equipment used in CGTG cogeneration systems are:

- Heat-recovery steam generator (HRSG) or hot-water heater
- Exhaust-gas chillers

Combustion turbines typically generate up to 10 lb/h (16,330 kg/s) of 15 to 150-psig (103.42 to 1034.2-kPa) steam per horsepower (0.7457 kW) of output. Because of the volume of excess air contained in the CGTG exhaust, it is possible to supplement the heat contained in the turbine exhaust to gain additional steam-generating capacity or cooling capacity by burning additional fuel. This supplemental gas firing typically has an efficiency of 90 percent.

Heat Balance. Mechanical energy makes up approximately 30 percent of a CGTG unit's heat balance under full-load conditions. Exhaust gas contains essentially the remainder of the energy, with small portions allocated to lube oil and radiation. This exhaust-gas thermal energy can be directly applied to driving an HRSG or an exhaust-gas chiller-heater.

The lube oil temperature is low and the quantity of heat is small, and thus, in most cases, it is not economical to recover heat from this source.

Load Control. Single- and multi-shaft combustion turbines are available. The multi-shaft units are designed with separate shafts for the compressor section and the power turbine section. Separate shafts permit the rotating speed of the compressor section to be controlled by the requirements of the power turbine, rather than by the rotating speed of the generator.

Partial-load operating efficiencies between the single- and two-shaft types of combustion gas turbines are illustrated in Fig. 2.2.5. The two-shaft units are able to maintain higher exhaust temperatures, and therefore greater operating efficiency, under partial-load conditions. The two-shaft units, however, will have higher heat rates at full-load conditions. If partial-load operation of a combustion turbine is

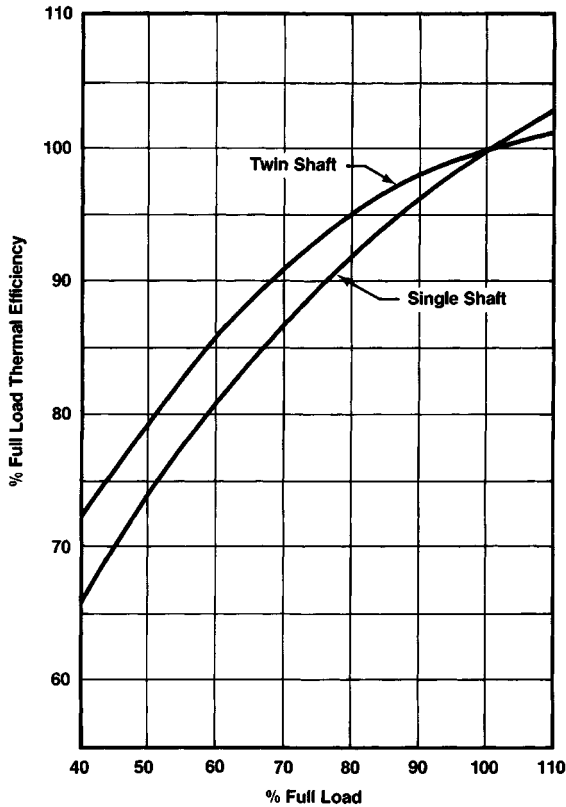


FIGURE 2.2.5 Partial-load cycle efficiency- single- and twin-shaft turbines

required because of cogeneration system operating criteria, consideration should be given to a two-shaft combustion turbine.

2.2.5.2 Internal-Combustion Engines

Internal-combustion engines exhibit the following characteristics in cogeneration systems:

- High mechanical efficiency
- More efficient operation at partial loads (Fig. 2.3.6)
- High-temperature exhaust gases
- Readily available maintenance services

Heat-recovery equipment used in cogeneration systems using internal-combustion engines includes:

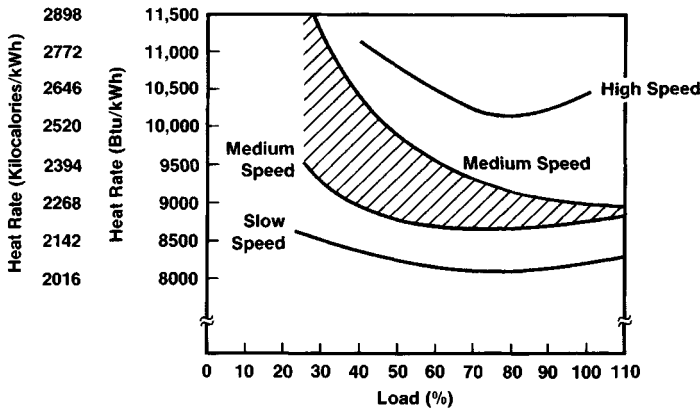


FIGURE 2.2.6 Typical variation of internal-combustion engine heat rate with load

- Water tube boilers with steam separators
- Coil-type hot-water heaters
- Steam separators for use with high-temperature cooling of engine jackets
- Exhaust-gas-driven chillers

Internal-combustion engines typically generate 3 lb/h (4899 kg/s) of 15- to 150-psig (103.42- to 1034.32-kPa) steam per horsepower output. Due to the lack of oxygen in the exhaust gas, electric heaters are required to supplement the exhaust-gas thermal energy.

Jacket-Water Heat Recovery. Cogeneration heat-recovery systems that use engine jacket-water thermal energy take four forms:

1. The heated jacket water may be routed to process needs. Engine cooling is dependent on the leak-tight integrity of this system.
2. The jacket cooling-water circuit for each engine transfers heat to an overall utilization circuit serving facility process needs. The overall utilization circuit may also be heated by the engine exhaust. This configuration minimizes connections to the jacket cooling-water system.
3. The recovered heat in the jacket cooling-water system is flashed to steam in an attached steam flash chamber. Water enters the engine at 235°F (112.7°C) and exits at 250°F (121.1°C). Steam is produced 235°F (112.7°C), 8 psig (55.168 kPa). Flow must be restricted at the entrance to the steam flash chamber to maintain sufficient back pressure on the liquid coolant in the engine chambers.
4. Some engines use natural-convection ebullient cooling. A steam-and-water mixture rises through the engine to a separating tank, where the steam is released and the water is recirculated. A rapid coolant flow is required through the engine due to a small rise in the temperature of the fluid. Moreover, back pressure must be controlled, for the steam bubbles in the engine could rapidly expand, causing the engine to overheat. This system produces 15-psig (103.42-kPa) steam at 250°F (121.1°C).

The temperature and pressure of these jacket-water heat-recovery systems make them suitable for single-stage absorption chiller application (Table 2.2.1).

Heat Balance. A typical heat balance for an internal-combustion engine is illustrated in Fig. 2.2.7. The exhaust heat makes up the largest portion of the energy.

The jacket cooling-water component of thermal energy from an internal-combustion engine contains 30 percent of the heat input (Fig. 2.2.7). Jacket cooling-water temperatures are summarized in Table 2.2.4. Some internal-combustion-engine manufacturers discourage operating with high jacket-water temperatures, for special gasket and seal designs are required.

The lubricating oil system also contains usable heat (Fig. 2.2.7). The normal operating temperature for the system is 165°F (73.9°C). The lube oil cooling fluid may also be routed through the exhaust heat-recovery unit if process requirements specify heat at a higher temperature. By elevating the lube oil coolant temperature above 180°F (82.2°C) toward 200°F (93.3°C), special lubricants may be required to ensure an adequate useful life of the oil.

Load Control. The heat rate of an internal-combustion engine remains almost constant above approximately 50 percent load, as illustrated in Fig. 2.2.6. From the engine heat balance, energy normally being converted to mechanical energy is transferred to thermal energy below 50 percent power. Cogeneration systems are suited to using a large portion of this thermal energy.

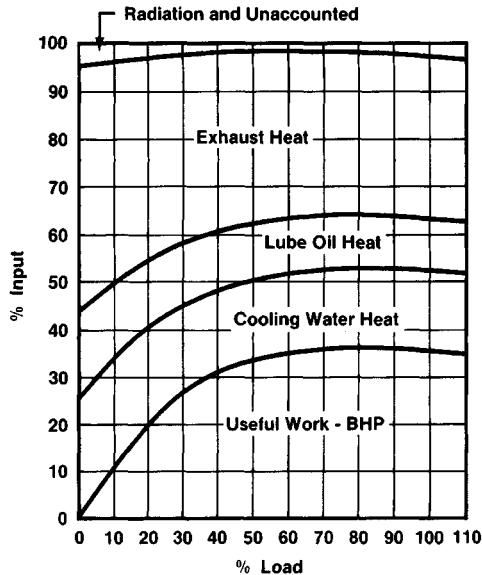


FIGURE 2.2.7 Heat balance for eight-cylinder diesel engine

P · A · R · T B

SYSTEMS AND COMPONENTS

SECTION 3

COMPONENTS FOR HEATING AND COOLING

CHAPTER 3.1

PIPING

PART 1: WATER AND STEAM PIPING*

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3.1.1 INTRODUCTION

Once the designer has calculated the required flows in gallons per minute (cubic meters per second or liters per second) for chilled-water, condenser water, process water, and hot-water systems or pounds per hour (kilograms per hour) for steam systems and tons or Btu per hour (watts per hour) for refrigeration, calculation of the size of each piping system can proceed.

3.1.2 HYDRONIC SYSTEMS

With respect to hydronic systems (chilled water, condenser water, process water, hot water, etc.), the designer has the option of using the manual method or one of the computer programs.

Whether the piping system is designed manually or by the computer, the effects of high altitude must be accounted for in the design if the system will be installed at elevations of 2500 ft (760 m) or higher. Appropriate correction factors and the effects of altitudes 2500 ft (760 m) and higher are discussed in App. A of this book.

The following is a guide for design water velocity ranges in piping systems that will not result in excessive pumping heads or noise:

Boiler feed	8 to 15 ft/s (2.44 to 4.57 m/s)
Chilled water, condenser water, hot water, process water, makeup water, etc.	4 to 10 ft/s (1.22 to 3.05 m/s)
Drain lines	4 to 7 ft/s (1.22 to 2.13 m/s)

*Edited for 2nd Edition by Robert O. Couch, Perma-Pipe Corp., Niles, IL.

Pump suction	4 to 6 ft/s (1.22 to 1.83 m/s)
Pump discharge	8 to 12 ft/s (2.44 to 3.66 m/s)

Where noise is a concern, such as in pipes located within a pipe shaft adjacent to a private office or other quiet areas, velocities within the pipe should not exceed 4 ft/s (1.22 m/s) unless acoustical treatment is provided. (Noise control and vibration are discussed in Chapters 8.2 and 8.3 of this book.)

Flow velocities in PVC pipe should be limited to 5 ft. (1.5 m)/sec unless special care is taken in the design and operation of valves and pumps. This is necessary to prevent pressure surges (water hammer) that could be damaging to pipe.

Erosion should also be considered in the design of hydronic piping systems, especially when soft material such as copper and plastic is used. Erosion can result from particles suspended in the water combined with high velocity. To assist the designer, Table 3.1 shows maximum water velocities that are suggested to minimize erosion, especially in soft piping materials.

Pipe size depends on the required amount of flow, the permissible pressure drop and the desired velocity of the fluid. This may be manually calculated by various methods given in Refs. 1 to 5. An acceptable method of evaluating water flow is the Hazen-Williams formula:

$$f = 0.2083 \times \left(\frac{100}{C} \right)^{1.852} \times \frac{Q^{1.852}}{id^{4.8655}} \quad (3.1.1)$$

where f = friction head loss in ft of water per 100 ft of pipe (Divide by 2.31 to obtain pounds per square inch)

C = constant for inside pipe roughness (See Table 3.1.2 below)

Q = flow in U.S. gal/m

id = inside diameter of pipe, in.

Water velocity in f/s may be calculated as follows:

$$V = 0.408709 \times \frac{Q}{id^2} \quad (3.1.2)$$

where V = velocity in f/s

Q = flow in U.S. gal/m

id = inside diameter of pipe

TABLE 3.1.1 Maximum Water Velocities to Minimize Erosion

Annual operating hours	Maximum water	velocity
	ft/s	m/s
1500	11	3.35
2000	10.5	3.20
3000	10	3.05
4000	9	2.74
6000	8	2.44
8000	7	2.13

TABLE 3.1.2 Typical Values to Use for the Hazen-Williams Coefficient

Pipe material	C
PVC, FRP, PE	150
Very to extremely smooth metal pipes	130–140
Smooth wooden or masonry pipe	120
Vitrified clay	110
Old cast iron or old steel pipe	100
Brick	90
Corrugated metal	60

If the computer method is chosen to size the hydraulic piping systems, the designer must select a software program from the several that are available. Two of the most widely used are Trane's CDS Water Piping Design program and Carrier's E20-II Piping Data program. In addition to determining the pipe sizes, both programs print a complete bill of materials (quantity takeoff by pipe size, length, fittings, and insulation).

Whichever program is used, the specific program input and operating instructions must be strictly followed. It is common to trace erroneous or misleading computer output data to mistakes in inputting design data. *It cannot be overstressed that in order to get meaningful output data, input data must be correctly entered and checked after entry before the program is run.* It is also a good, if not mandatory, policy to independently check the computer results the first time you run a new or modified program, to ensure that the results are valid.

If the computer program used does not correct the computer output for the effects of altitude when the elevation of the project is equal to or greater than 2500 ft (760 m) above sea level, the computer output must be manually corrected by using the appropriate correction factors listed in App. A of this book.

The following describe the programs available to the designer using Trane's CDS Water Piping Design program for sizing hydronic systems.

Water Piping Design (DSC-IBM-123). This pipe-sizing program is for open and closed systems, new and existing systems, and any fluid by inputting the viscosity and specific gravity. The user inputs the piping layout in simple line-segment form with the gallons per minute of the coil and pressure drops or with the gallons per minute for every section of pipe. The program sizes the piping and identifies the critical path, and then it can be used to balance the piping so that the loops have equal pressure drops.

The output includes

- Complete bill of materials (including pipe sizes and linear length required, fittings, insulation, and tees)
- Piping system costs for material only or for material and labor
- Total gallons of fluid required

The following summary describes the program available to the designer using Carrier's E20-II Water Piping Design for sizing hydronic systems.

Water Piping Design (Version 1.0). This program provides the following:

- Enables the designer to look at the balancing required for each piping section, thereby permitting selective reduction of piping sizes or addition of balancing valves
- Calculates pressure drop and material takeoff for copper, steel, or plastic pipe
- Sizes all sections and displays balancing required for all circuits
- Sizes closed or open systems
- Corrects pressure drop for water temperature and/or ethylene glycol
- Calculates gallons per minute of total system
- Calculates total material required, including fittings
- Ability to store for record or later changes up to 200 piping sections
- Ability to change any item and immediately rerun
- Allows sizing of all normally used piping materials
- Allows balancing of system in a minimum amount of time
- Allows easy sizing of expansion tanks and determination of necessary gallons per minute of glycol for brine applications
- Estimates piping takeoff fitting by pipe size, quantities (linear feet, fittings, valves, etc.).

3.1.3 STEAM SYSTEMS

There are few computer programs available for sizing complex networks of steam piping. Most design is done manually although simple computer programming of the various formulas such as the Fritzsche and Unwin formulas will save a considerable amount of time. Unwin's formula which appears to be the preferred method of district heating engineers is as follows:

$$P = \frac{0.0001306 \times W^2 \times L \left(1 + \frac{3.6}{d}\right)}{y d^5} \quad (3.1.3)$$

where P = pressure drop—psi
 W = pounds of steam—lb/m
 L = length of pipe—ft
 d = inside diameter of pipe—in.
 y = Average density of steam lb/ft³

It is advisable to use values for the specific volume corresponding to the average pressure if the drop exceeds 10 percent to 15 percent of the initial absolute pressure.

Figure 3.1.1 gives a graphical solution to Unwin's formula.

The effects of high altitude must be accounted for in the design when the system will be installed at elevations of 2500 ft (760 m) or higher. Appropriate correction factors and the effects of altitudes 2500 ft (760 m) and higher are discussed in App. A.

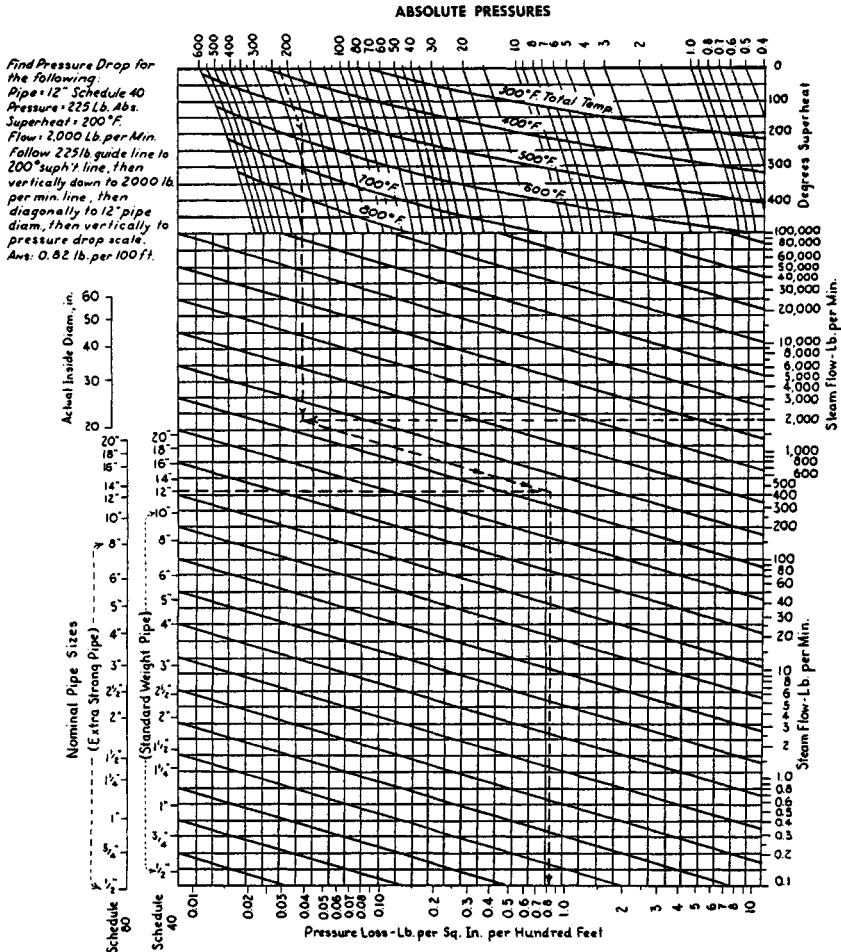


FIGURE 3.1.1 Courtesy Perma-Pipe, Inc.

Table 3.1.3 gives reasonable velocities for stem lines based on average practice. The lower velocities should be used for smaller pipes and the higher velocities for pipes larger than 12 in (30 cm).

Steam piping systems may also be sized by following one of the accepted procedures found in standard design handbook sources such as Refs. 2, 3, 5.

TABLE 3.1.3

Condition of steam	Psi	Bar	Ft/min	m/s
Saturated	0–15	0–1.03	4000–6000	20.32–30.48
Saturated	50 and up	3.43 and up	6000–10000	30.48–50.08
Superheated	200 and up	13.73 and up	7000–20000	35.56–101.60

3.1.4 REFRIGERANT SYSTEMS

Here the designer has the option of using the annual method or at least one computer program.

Whether the piping system is designed manually or by computer, the effects of high altitude must be accounted for in the design when the system will be installed at elevations of 2500 ft (760 m) or higher. Appropriate correction factors and the effects of altitudes 2500 ft (760 m) or higher. Appropriate correction factors and the effects of altitudes 2500 ft (760 m) and higher are discussed in App. A.

Liquid line sizing is considerably less critical than the sizing of suction or hot gas lines, since liquid refrigerant and oil mix readily. There is no oil movement (separation) problem in designing liquid lines. It is good practice to limit the pressure drop in liquid lines to an equivalent 2°F (1°C). It is also good practice to limit the liquid velocity to 360 ft/min (1.83 m/s).

The suction line is the most critical line to size. The gas velocity within this line must be sufficiently high to move oil to the compressor in horizontal runs and vertical risers with upward gas flow. At the same time, the pressure drop must be minimum to prevent penalizing the compressor capacity and increasing the required horsepower. It is good practice, where possible, to limit the pressure drop in the suction line to an equivalent temperature penalty of approximately 2°F (1°C). In addition to the temperature (pressure drop) constraints, the following minimum gas velocities are required to move the refrigerant oil:

Horizontal suction lines	500 ft/min (2.54 m/s) minimum
Vertical upflow suction lines	1000 ft/min (5.08 m/s) minimum

The velocity in upflow risers must be checked at minimum load; if it falls below 1000 ft/min (5.08 m/s), double risers are required. To avoid excess noise, the suction line velocity should be below 4000 ft/min (20.32 m/s).

The discharge (hot-gas) line has the same minimum and maximum velocity criteria as suction lines; however, the pressure drop is not as critical. It is good practice to limit the pressure drop in the discharge (hot-gas) line to an equivalent temperature penalty of approximately 2 to 4°F (1 to 2°C).

If the manual method is used to size the project, refrigerant piping systems should be calculated by following one of the accepted procedures found in standard design handbook sources such as Refs. 3, 6, and 7.

If the computer method is used to size the project hydraulic piping systems, the designer must choose a program among the several available. Two of the most widely used are Trane's CDS Water Piping Design program and Carrier's E20-II Piping Data program. In addition to determining the pipe sizes, both programs print a complete bill of materials (Quantity takeoff by pipe size, length, fittings, and insulation). Whichever program is used, it is mandatory that the specific program's input and operating instructions be strictly followed. It is common to trace erroneous or misleading computer output data to mistakes in inputting design data into the computer. *In order to get meaningful output data, input data must be correctly entered and checked after entry before the program is run.* It is also a good, if not mandatory, policy to independently check the computer results the first time you run a new or modified program, to ensure that the results are valid.

If the computer program used does not correct the computer output for the effects of altitude when the elevation of the project is equal to or greater than 2500 ft

(760 m) above sea level, the computer output must be manually corrected by using the appropriate correction factors, listed in App. A.

DX Piping Design (Version 1.0). Described in the following summary, this program is available to the designer using Carrier's E20-II DX Piping Design to size the refrigerant systems.

- This program will determine the minimum piping size to deliver the refrigerant between compressor, condenser, and evaporators while ensuring return at maximum unloading.
- This program is able to size piping systems using ammonia and Refrigerants 12, 22, 500, 503, 717.
- This program is capable of calculating low-temperature as well as comfort cooling applications.
- This program determines when double risers are needed, sizes the riser, and calculates the pressure drop.
- This program will include accessories in the liquid line and automatically calculates the subcooling required.
- This program permits entering, for all fittings and accessories, pressure drops in degrees Fahrenheit or pounds per square inch.
- This program will size copper or steel piping.
- This program can select pipe size based on the specific pressure drop.
- This program will calculate the actual pressure drop in degrees Fahrenheit and pounds per square inch for selected size.
- This program will estimate piping takeoff, listing by pipe size the quantities of linear feet, fittings, valves, etc.

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6. *Ibid.*, part 3, chaps. 1 and 3.
7. *Trane Reciprocating Refrigeration Manual*, Trane Company, La Crosse, WI, 1989.

PIPING

PART 2: OIL AND GAS PIPING

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3.1.5 INTRODUCTION

The fuel oil piping system consists of two lines. The suction line is from the storage tank to the fuel oil pump inlet. On small burners the fuel oil pump is an integral part of the burner. The discharge line is from the fuel oil pump outlet to the burner. On systems that have a return line from the burner to the storage tank, this return line is considered part of the discharge piping when the piping losses are calculated.

3.1.6 OIL PIPING

Suction

Suction requirements are a function of

1. Vertical lift from tank to pump
2. Pressure drop through valves, fittings, and strainers
3. Friction loss due to oil flow through the suction pipe. This loss varies with:
 - a. Pumping temperature of the oil, which determines viscosity
 - b. Total quantity of oil being pumped
 - c. Total length of suction line
 - d. Diameter of suction line

To determine the actual suction requirements, two assumptions must be made, based on the oil being pumped. First, the maximum suction pressure on the system should be as follows:

No. 2 oil	12 inHg (305 mmHg)
No. 4 oil	12 inHg (305 mmHg)
Nos. 5 and 6 oil	17 inHg (432 mmHg)

Second, the lowest temperature likely to be encountered with a buried tank is 40°F (5°C). At this temperature the viscosity of the oil would be:

No. 2 oil	68 SSU* (12.5 cSt)
No. 4 oil	1000 SSU (21.6 cSt)

In the case of Nos. 5 and 6 oil, the supply temperature of the oil should correspond to a maximum allowable viscosity of 4000 SSU (863 cSt). This viscosity corresponds to a supply temperature of 110 to 225°F (43 to 105°C) for commercial grades of Nos. 5 and 6 oils. Then, using Fig. 20.1 and entering at 4000 SSU and going horizontally to the No. 5 fuel range, the maximum corresponding temperature is about 70°F (21°C). Likewise, the maximum corresponding temperature for No. 6 fuel is about 115°F (46°C).

The suction pressure limits noted above also allow for the following:

1. The possibility of encountering lower supply temperatures than indicated above, which would result in higher viscosities
2. Some fouling of suction strainers
3. In the case of heavy oil (Nos. 5 and 6), pump wear, which must be considered with heavy oils (See Figs. 20.3 to 20.6 for suction pressure curves.)

Strainers. It is a good practice to install suction-side strainers on all oil systems to remove foreign material that could damage the pump. The pressure drop associated with the strainer must be included in the overall suction pressure requirements.

Strainers are available as simplex or duplex units. Duplex strainers allow the ability to inspect and clean one side of the strainer without shutting down the flow of oil.

Discharge

Pumps. Pumps for fuel oil must be chosen based on several design criteria; viscosity of fuel oil, flow requirements, discharge pressure required, and fluid pumping temperature.

Viscosity. Charts for commercial grades of fuel oil are shown in Fig. 3.1.2. The pump must be designed for the viscosity associated with the lowest expected pumping temperatures.

Flow. Fuel oil pumps should be selected for approximately twice the required flow at the burner. The additional flow will allow for pressure regulation, so that constant pressure can be supplied at the burner.

Pressure. The supply pressure of the pump is based on the required regulated pressure at the burner.

A system utilizing a variable orifice for flow control typically requires from 30 to 60 psig (207 to 414 kN/m²). The metering orifice type of system can be used on all grades of fuel oil. Burners utilizing an oil metering pump usually limit the supply pressure to prevent seal failure. As with metering orifices, there is no limitation on the grade of fuel oil used.

Temperature. The temperature of the oil must be considered, to ensure that the seals and gaskets supplied can withstand the fluid temperature.

*SSU is the abbreviation for standard Saybolt unit.

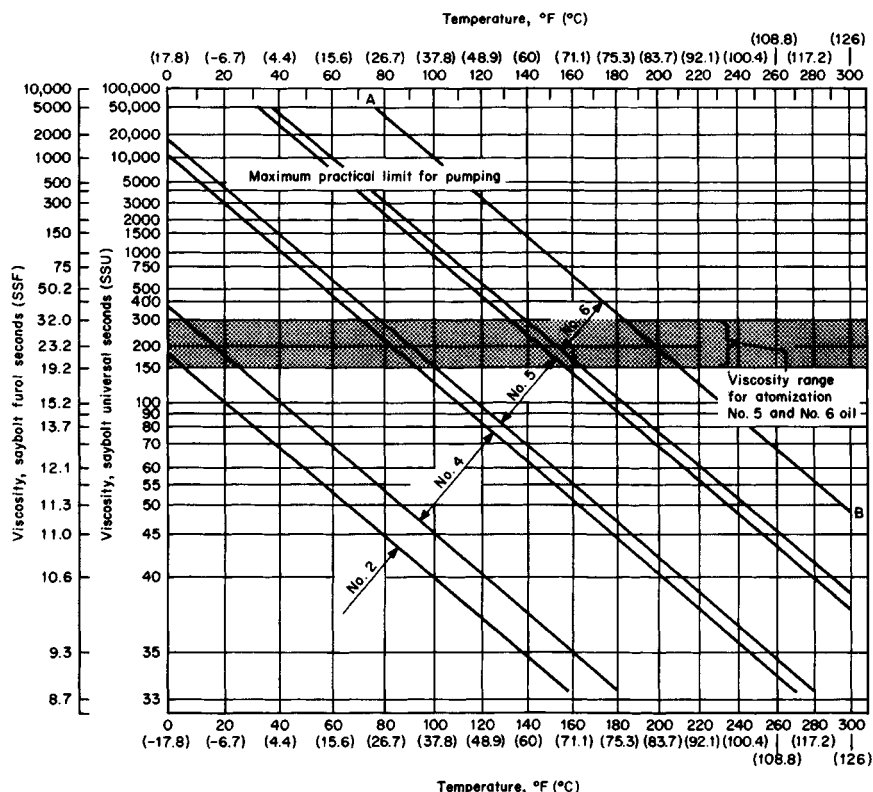


FIGURE 3.1.2 Viscosity-temperature curves for fuel oil Nos. 2, 4, 5, and 6. Based on U.S. Department of Commerce's Commercial Standard CS12-48. (Courtesy of Cleaver-Brooks.)

Pumping. The major difference between calculating hydronic and fuel oil piping systems is that the actual specific gravity of the oil being pumped must be accounted for.

The design pump head is equal to the suction lift, dynamic piping loss (including fittings, valving, etc.), and required supply pressure at the burner (if applicable).

Figure 3.1.3 should be used to determine the equivalent length of straight pipe that results in the same pressure drop as the corresponding pipe fitting or valve.

Figures 3.1.4 to 3.1.9 should be used to determine the appropriate dynamic piping losses with respect to type of oil being pumped, flow rate, and pipe size. The total equivalent length of straight pipe for fittings and valving, from Fig. 3.1.9, must be added to the total length of horizontal and vertical piping before multiplying by the appropriate piping loss factor.

The pressure loss for each strainer generally must be calculated separately and added to the total.

To obtain the suction lift in inches (millimeters) of mercury (Hg) from the bottom of the suction pipe (in the tank) to the boiler connection or pump suction centerline, multiply this vertical distance in feet (meters) by 0.88155 inHg/ft of water (73.428 mmHg/m of water) by the specific gravity of the oil being pumped.

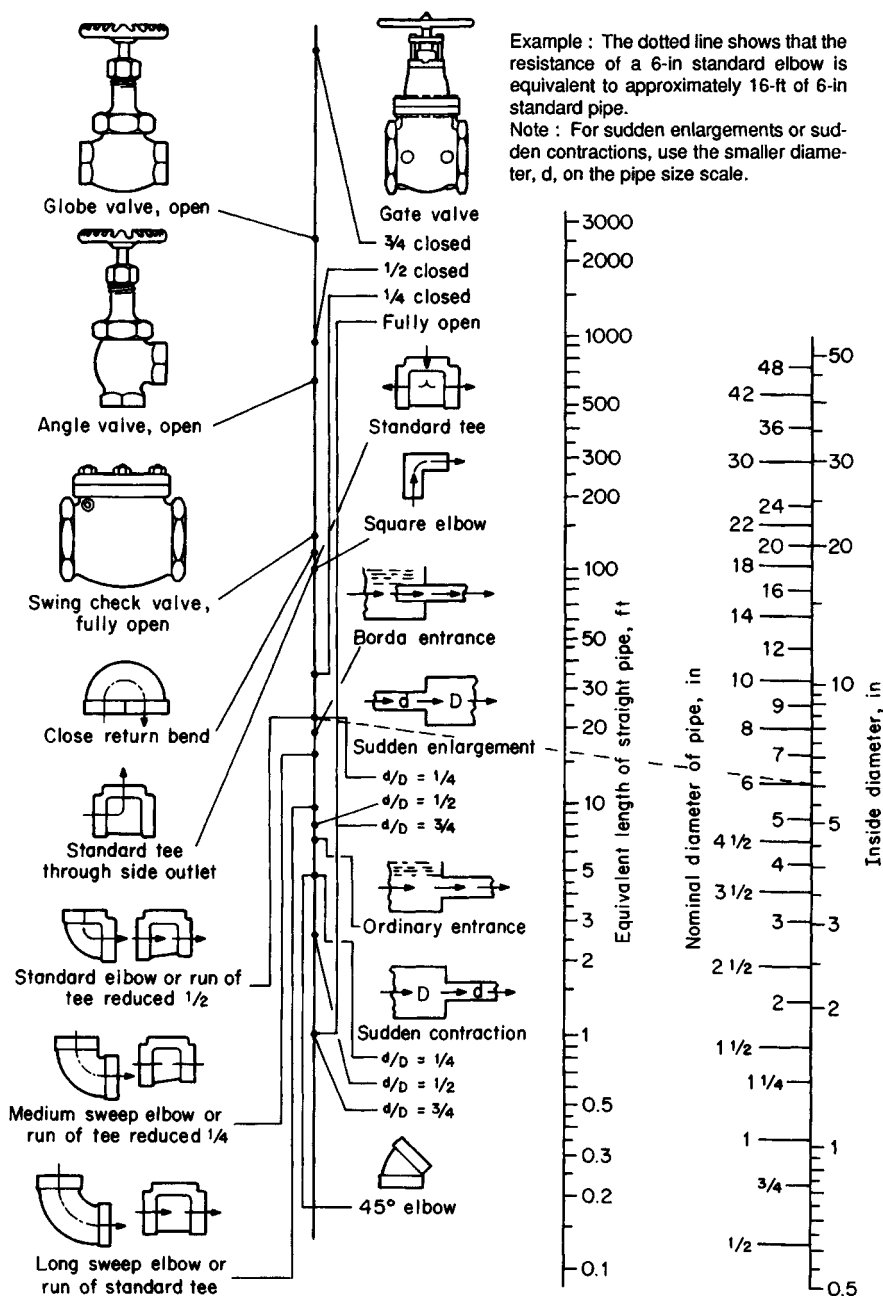


FIGURE 3.1.3 Friction losses in pipe fittings. The chart may be used for any liquid or gas. (Courtesy of Cleaver-Brooks.)

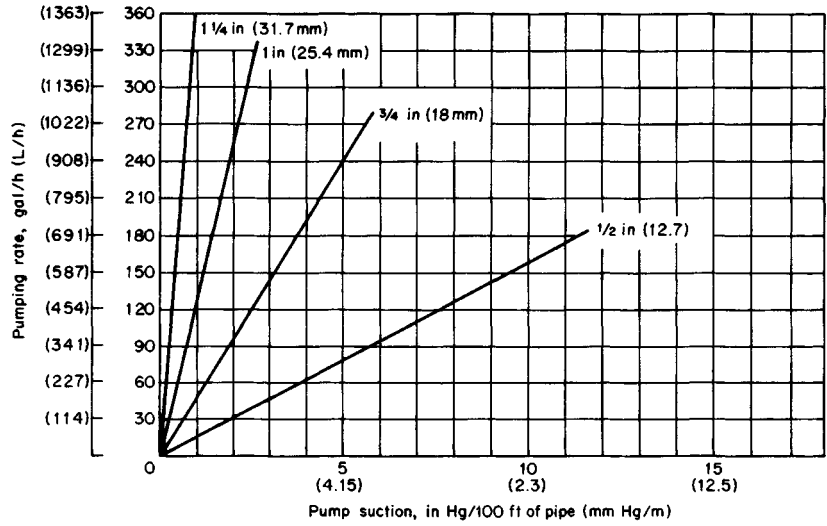


FIGURE 3.1.4 Pump suction curves for No. 2 fuel oil. Curves are based on a pumping temperature of 40°F (4.4°C), or 68 SSU. (Courtesy of Cleaver-Brooks.)

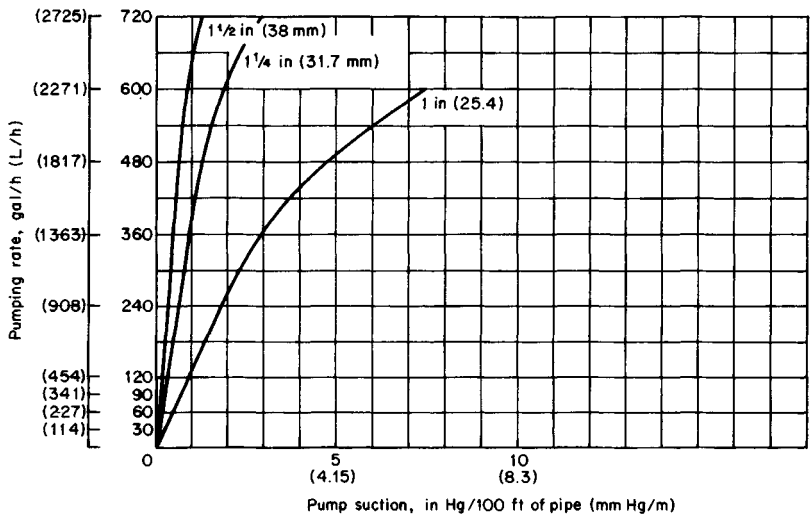


FIGURE 3.1.5 Pump suction curves for No. 2 fuel oil. Curves are based on a pumping temperature of 40°F (4.4°C), or 68 SSU. (Courtesy of Cleaver-Brooks.)

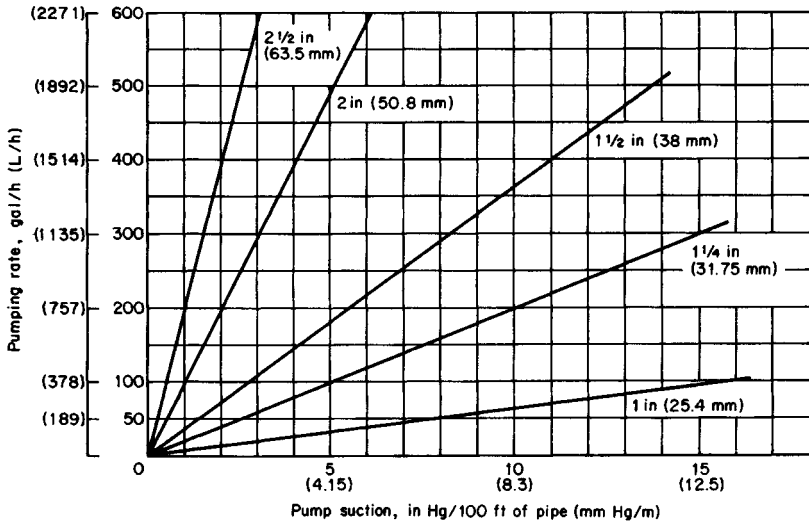


FIGURE 3.1.6 Pump suction curves for No. 4 fuel oil. Curves are based on a pumping temperature of 40°F (4.4°C), or 1000 SSU. (Courtesy of Cleaver-Brooks.)

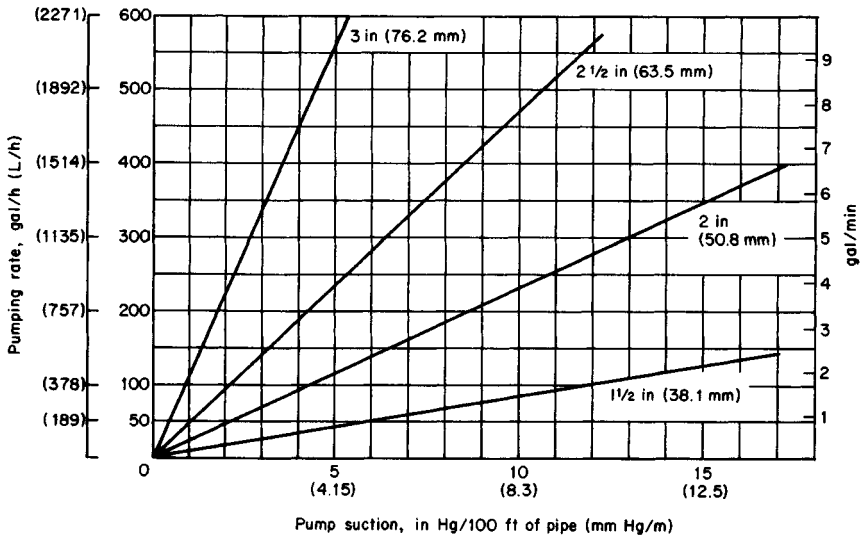
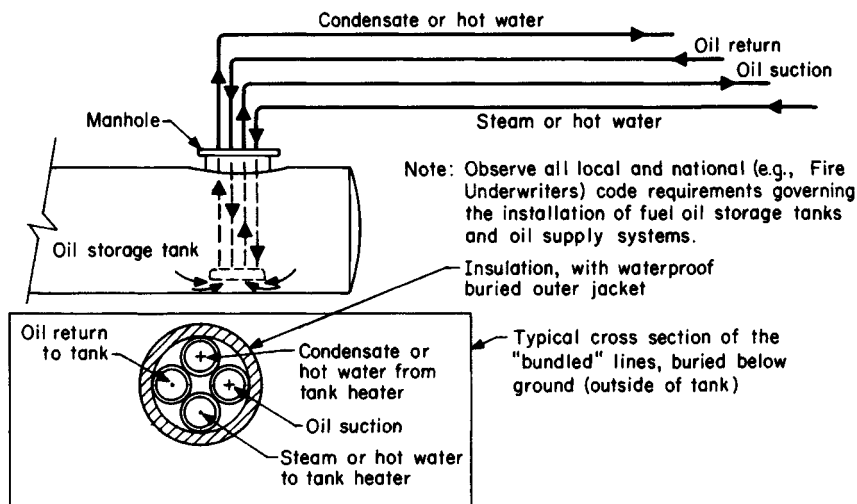


FIGURE 3.1.7 Pump suction curves for Nos. 5 and 6 fuel oils. Curves are based on a pumping limit of 4000 SSU. (Courtesy of Cleaver-Brooks.)



Note: The temperature of the oil suction line should not exceed 130°F (54.4°C). Higher temperatures could cause vapor binding of the oil pump, which would decrease oil flow.

FIGURE 3.1.8 Tank heaters. (Courtesy of Cleaver-Brooks.)

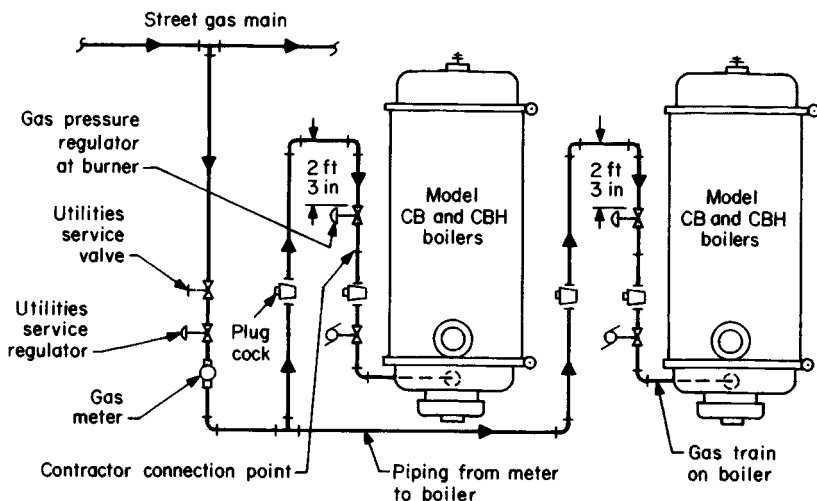


FIGURE 3.1.9 Gas piping to boiler. The figure illustrates the basic gas valve arrangement on boilers and shows the contractor's connection point for a typical installation. Actual requirements may vary depending on local codes and local gas company requirements, which should be investigated prior to both the preparation of specifications and construction. (Courtesy of Cleaver-Brooks.)

For No. 2 oil with a specific gravity of 0.85 at maximum 40 SSU and 100°F (37.8°C):

$$\text{Suction lift} = Cd \quad (3.1.4)$$

Where the suction lift is inHg (J), C is in inHg/ft (mmHg/m), and d is in ft (m).

Heaters. Heaters are used to increase fuel oil temperatures, to provide the viscosity to atomize properly. Oil temperatures corresponding to a viscosity of 100 SSU [2×1.6 centistokes (cSt)] or less are recommended.

Heating can be accomplished by using hot water, steam, electricity, or a combination of these. Most packaged boilers have heaters that utilize electric elements for initial warmup and then transfer to either hot water or steam when the boiler has reached sufficient temperature and pressure. The heater sizing should be based on the supply pump design flow rate and temperature.

Electric heaters are commonly used to preheat heavy fuel oils on low-temperature hot-water boilers or on startup of a high-temperature hot-water or steam boiler.

The watt density of an electric heater should not exceed 5 W/in² (0.007 W/mm²) because of dangers with vapor lock and coking on the heater surface. When steam is used as the heating medium for heavy oils, the steam pressure used should have a saturation temperature at least equal to the desired oil outlet temperature.

The flow of steam is controlled by using a solenoid valve that responds to a signal from the oil heater thermostat.

Some steam heaters include electric heating elements to allow firing of oil on a cold startup. When sufficient steam pressure is available, the electric heater is automatically de-energized.

Steam from the boiler is regulated to the desired pressure for sufficient heating. If the boiler pressure exceeds the steam heater pressure by 15 lb/in² (1 bar) or more, superheated steam will be produced by the throttling process. Steam heater-lines should be left uninsulated to allow the steam to desuperheat prior to entering the heater. It is common practice to discharge the steam condensate leaving the oil heater to the sewer, to eliminate the possibility of contaminating the steam system in the event of an oil leak. The heat from the condensate is usually reclaimed prior to dumping it.

Excessive steam temperatures can also cause coking in the heater.

Hot-water oil heaters are essentially water-to-oil heat exchangers used to preheat oil. However, since the source of heat energy is boiled water circulated by the pump through the heater, any system leak could cause boiler water contamination. Therefore, safety-type heater systems are recommended for this service. Such an exchanger is frequently a double-exchange device using an intermediate fluid.

In cases where the oil must be heated to a temperature in excess of the hot-water supply temperature, supplemental heat must be provided by an electric heater. Tank heaters are commonly an insulated bundle of four pipes submerged in the oil tank. See Fig. 3.1.10. Tank preheating is required anytime the viscosity of the oil to be pumped equals 4000 SSU or greater.

Valves

Pressure Relief Valves. These are installed in the discharge line from the supply pump, to protect the pump and system from over pressure. Pressure relief valves are also commonly installed on oil heaters to relieve pressure so that oil may circulate even though the burner does not call for oil.

Pressure Regulators. These reduce system pressure and maintain a desired pressure at the burner.

Oil Shutoff. There are two commonly used styles of oil shutoff valves for burner service: electric coil and motorized. Electric coil solenoid valves are used on most small industrial and commercial burners. These valves are normally closed valves, and they control the flow of oil fuel to the burner. Two such valves for fuel shutoff are used on commercial and industrial boilers.

The second type of oil shutoff valve is a motorized valve that has a spring return to close. Motorized valves can be equipped with a proof-of-closure switch which ensures that the valve is in the closed position or prevents the burner from igniting if it is not. This type of switch is necessary to meet certain insurance requirements.

Manual Gas Shutoff Valves. Manual gas shutoff valves are typically a lubricated plug type of valve with a 90° rotation to open or close. The valve and handle should be situated such that when the valve is open, the handle points in the direction of flow.

The number of valves and their locations are based on insurance requirements. Typically, manual valves are installed upstream of the gas pressure regulator, directly downstream of the gas pressure regulator, and downstream of the last automatic shutoff valve.

Automatic Gas Shutoff Valves. Three types of automatic gas shutoff valves are used on burners: solenoid valves, diaphragm valves, and motorized valves.

Of the three automatic valves, the solenoid is the simplest and generally the least expensive. A controller opens the valve by running an electric current through a magnetic coil. The coil, acting as a magnet, pulls up the valve disk and allows the gas or oil to flow. Solenoid action provides fast opening and closing times, usually less than 1 s.

Diaphragm valves are frequently used on small to medium boilers. These valves have a slow opening and fast closing time. They are simple, dependable, and inexpensive. They are full-port valves and operate with little pressure loss.

Motorized shutoff valves are used for large gas burners that require large quantities of gas and relatively high gas pressures. There are two parts to a motorized valve: the valve and a fluid power actuator. A limit switch stops the pump motor when the valve is fully open. The valve is closed by spring pressure. The valve position (open or closed) is visible through windows on the front and side of the actuator. Motorized valves often contain an override switch which is actuated when the valve reaches the fully closed position. This proof-of-closure switch is needed to meet several different insurance company requirements.

Vent Valves. Vent valves are normally open solenoid valves that are wired in series and are located between two automatic shutoff valves in the main gas line or, in some cases, the pilot line. The vent valve vents to the atmosphere all gas contained in the line between the two valves.

Flow Control Valves

1. Butterfly valves are the most commonly used device for controlling the quantity of fuel gas flow to the burner. The pressure drop associated with a fully open butterfly valve is very low. Butterfly valves can be used for control of air flow and with special shaft seals can be used for all grades of fuel gas. Linkage arms are connected to the shaft of the valve and driven directly from the burner-modulating motor.

2. Modulating gas shutoff valves can be supplied with positioning motors that can operate on the on/off principle or high/low/off. In the case of the high/low/

off shutoff valves, the air damper is controlled by the valve-modulating motor. This allows the valve position to dictate the amount of combustion air necessary for the gas input rate.

3. Pneumatic control valves are often butterfly valves that are driven by a pneumatic actuator. The signal to the pneumatic actuator is proportional to the combustion air flow and positions the valve to deliver the appropriate amount of gas. Often additional signals such as steam flow and combustion air flow are used to determine the signal to the valve and its corresponding position.

Gas Strainer. It may be advisable to use a strainer to protect the regulators and other control equipment against any dirt or chips that might come through with the gas.

Gas Compressors or Boosters. If the local gas utility cannot provide sufficient gas pressure to meet the requirements of the boiler, a gas compressor or booster should be used. *Caution:* The use of a gas compressor or booster must be cleared with the local gas utility prior to installation.

3.1.7 GAS PIPING

Figure 3.1.11 illustrates the basic arrangement for piping gas to boilers from street gas mains for a typical installation.

Line-Sizing Criteria

The first step in designing a gas piping system is to properly size components and piping to ensure that sufficient pressure is available to meet the demand at the burner. The boiler manufacturer should be consulted to determine the pressure required.

The gas service piping installed in the building must be designed, and components selected, to provide the required fuel gas flow to the boiler at the manufacturer's recommended pressure. The utility supplying gas to the facility will provide the designer with information on the maximum available gas pressure for the site. The gas piping design must be appropriate for the specific site conditions.

The gas train pressure requirements can be expressed as

$$P_s = P_R + P_C + P_P + P_F + P_B + P_{fp} \quad (3.1.5)$$

where P_s = supply pressure available

P_R = pressure drop across gas pressure regulator

P_C = pressure drop across gas train components

P_P = pressure drop associated with straight runs of pipe

P_F = pressure drop associated with elbows, tees, or other fittings

P_B = pressure drop across burner orifice or annulus

P_{fp} = boiler furnace pressure

Pressure drop calculations for regulators and valves are normally based on the C_v factor or coefficient of valve capacity of air or in equivalent feet or diameters of pipe length.

The resistance coefficient k can be used to express the pressure drop as a number of lost velocity heads

$$k = \frac{PV^2}{2g} \quad (3.1.6)$$

Depending on the information available, the following equations can be used to determine the pressure drop through valves or across regulators:

$$k = f \cdot \frac{L}{D} \quad (3.1.7)$$

$$P = f \cdot \frac{L}{D} \cdot \frac{PV^2}{2g} \quad (3.1.8)$$

$$k = \frac{891d^4}{C_v^2} \quad (3.1.9)$$

$$H_v = 0.000228V^2 \text{ in WG} \quad \text{for air} \quad (3.1.10)$$

$$P = \frac{k}{144} H_v \quad (3.1.11)$$

$$C_v = 0.0223(\text{ft}^3/\text{h}) @ 1\text{-inWG drop})G \quad \text{for 0- to 2-psig gases} \quad (3.1.12)$$

where k = resistance coefficient

f = Darcy friction factor

L = length of pipe or equivalent length of pipe for fitting, ft

D = diameter of pipe, ft

P = pressure drop or differential, lb/in²

V = velocity, ft/s

C_v = valve conductance based on H₂O @ 1 lb/in² drop

g = acceleration of gravity

H_v = velocity head

G = gas gravity relative to air = $P/0.0765$

p = density of flowing fluid, lb/ft³

Note: Metric units must be converted to English units before Eqs. (3.1.5) to (3.1.12) can be applied.

To determine the losses associated with straight runs of pipe (P_p) and pipe fittings (P_f), Eq. (3.1.5) can be used. Values for equivalent length of pipe or equivalent pipe diameter are listed in Fig. 3.1.5. The pressure drop for the burner orifice or annulus (P_B) can be calculated by using Eq. (3.1.8) and making the appropriate gas density corrections. The furnace pressure P_{fp} is a function of the furnace geometry, size, and firing rate. This pressure is often zero or slightly negative, but for some types of boilers and furnaces it can run as high as 15 in water column (in WC) (381 mm) positive.

Gas Train Components

Pressure Regulators. Pressure regulators or pressure-reducing regulators are used to reduce the supply pressure to the level required for proper burner operation. The regulated, or downstream, pressure should be sufficient to overcome line losses and deliver the proper pressure at the burner. Pressure regulators commonly used on burners come in two types: self-operated and pilot-operated.

In a self-operated regulator, the downstream, or regulated, pressure acts on one side of a diaphragm, while a preset spring is balanced against the backside of the diaphragm. The valve will remain open until the downstream pressure is sufficient to act against the spring.

Regulators for larger pipe sizes are normally the pilot-operated type. This class of equipment provides accurate pressure control over a wide range of flows and is sometimes selected even in smaller sizes where improved flow control is desired.

A gas pressure regulator must be installed in the gas piping to each boiler. The following items should be considered when a regulator is chosen:

1. *Pressure rating:* The regulator must have a pressure rating at least equivalent to that in the distribution system.
2. *Capacity:* The capacity required can be determined by multiplying the maximum burning rate by 1.15. This 15 percent over-capacity rating of the regulator provides for proper regulation.
3. *Spring adjustment:* The spring should be suitable for a range of adjustment from 50 percent under the desired regulated pressure to 50 percent over.
4. *Sharp lockup:* The regulator should include this feature because it keeps the downstream pressure (between the regulator and the boiler) from climbing when there is no gas flow.
5. *Regulators in parallel:* This type of installation would be used if the required gas volume were very large and if the pressure drop had to be kept to a minimum.
6. *Regulators in series:* This type of installation would be used if the available gas pressure were over 5, 10, or 20 psig (34.5, 68.0, or 137.9 kPa), depending on the regulator characteristics. One regulator would reduce the pressure to 2 to 3 psig (17.8 to 20.7 kPa), and a second regulator would reduce the pressure to the burner requirements.
7. *Regulator location:* A straight run of gasline piping should be used on both sides of the regulator to ensure proper regulator operation. This is particularly important when pilot-operated regulators are used. The regulator can be located close to the gas train connection, but 2 to 3 ft (0.6 to 0.9 m) of straight-run piping should be used on the upstream side of the regulator. *Note:* Consult your local gas pressure regulator representative. She or he will study your application and recommend the proper equipment for your job.

CHAPTER 3.2

DUCT SIZING*

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3.2.1 INTRODUCTION

The function of a duct system is to provide a means to transmit air from the air-handling equipment (heating, ventilating, or air conditioning). In an exhaust system the duct system provides the means to transmit air from the space or areas to the exhaust fan to the atmosphere.

The primary task of the duct designer is to design duct systems that will fulfill this function in a practical, economical, and energy-conserving manner within the prescribed limits of available space, friction loss, velocity, sound levels, and heat and leakage losses and/or gains.

With the required air volumes in cubic feet per minute (cubic meters per second) determined for each system, the zone and space requirements known from the design load calculation, and the type of air distribution system [such as low-velocity single-zone, variable-air-volume (VAV) or multizone or high-velocity VAV or dual duct] decided upon, the designer can proceed to size the air ducts.

The designer must also choose one of three methods to size the duct systems: the equal-friction, equal-velocity, or static regain method. Of the three, the equal-friction and static regain methods are used most often. The equal-velocity method is used primarily for industrial exhaust systems where a minimum velocity must be maintained to transport particles suspended in the exhaust gases.

Static regain is the most accurate method, minimizes balancing problems, and results in the most economical duct sizes and lowest fan horsepower. It is also the only method that should be used for high-velocity comfort air-conditioning systems.

The equal-friction method is used primarily on small and/or simple projects. If manual calculations are made, this method is simpler and easier than static regain; however, if a computer is used, this advantage disappears.

Typical duct velocities for low-velocity duct systems are shown in Table 3.2.1. For high-velocity systems, typical duct velocities are shown in Table 3.2.2. The velocities suggested in Tables 3.2.1 and 3.2.2 may have to be adjusted downward to meet the required noise criteria. See Chap. 8.2 of this book for a discussion on noise and sound attenuation.

*Updated for this Second Edition by the Editor.

TABLE 3.2.1 Suggested Duct Velocities for Low-Velocity Duct System, ft/min (m/s)

Application	Main ducts		Branch ducts	
	Supply	Return	Supply	Return
Residences	1000 (5.1)	800 (4.1)	600 (3)	600 (3)
Apartments	1500 (7.6)	1300 (6.6)	1200 (6.1)	1000 (5.1)
Hotel bedrooms				
Hospital bedrooms				
Private offices	1800 (9.1)	1400 (7.1)	1400 (7.1)	1200 (6.1)
Director's rooms				
Libraries				
Theaters	1300 (6.6)	1100 (5.6)	1000 (5.1)	800 (4.1)
Auditoriums				
General offices	2000 (10.2)	1500 (7.6)	1600 (8.1)	1200 (6.1)
Expensive restaurants				
Expensive stores				
Banks	2000 (10.2)	1500 (7.6)	1600 (8.1)	1200 (6.1)
Average stores				
Cafeterias				
Industrial	2500 (12.7)	1800 (9.1)	2200 (11.2)	1600 (8.1)

TABLE 3.2.2 Suggested Duct Velocities for High-Velocity Duct System, ft/min (m/s)

Application	Main duct		Branch duct	
	Supply	Return	Supply	Return
Commercial institutions	2500–3800	1400–1800	2000–3000	1200–1600
Public buildings	(12.7–19.3)	(7.1–9.1)	(10.2–15.2)	(6.1–8.0)
Industrial	2500–4000	1800–2200	2200–3200	1500–1800
	(12.7–20.3)	(9.1–11.2)	(11.2–16.3)	(7.6–9.1)

Whether the duct system is designed manually or by computer, the effects of high altitude must be accounted for in the design if the system will be installed at elevations of 2500 ft (760 m) or higher. Appropriate correction factors and the effects of altitudes of 2500 ft (760 m) and more are discussed in App. A.

3.2.2 MANUAL METHOD

If the manual method is used to size the project duct systems, they should be calculated by following one of the accepted procedures found in standard design handbooks such as Refs. 1 and 2. A detailed discussion on air-handling system

design is shown in Ref. 3. For industrial dilution, ventilation, and exhaust duct systems, they should be calculated and sized by the procedures set forth in Ref. 4.

When the equal-friction or equal-velocity method is used manually, the time to calculate duct sizes can be shortened by using Carrier's Ductronic Calculator or Trane's Ductulator. Both will size round or rectangular ducts in U.S. Customary System (USCS) or metric units.

3.2.3 COMPUTER METHOD

If the computer method is used to size the project's duct systems, one must select a program among the several available. Two of the most widely used are Trane's CDS Duct Design program and Carrier's E20-II Duct Layout program. In addition to determining the duct sizes, both programs print a complete bill of materials (quantity takeoff by pipe size, length, fittings, and insulation).

Whichever program is used, the specific program's input and operating instructions must be strictly followed. It is common to trace erroneous or misleading computer output data to mistakes in inputting design data. It cannot be overstressed that in order to get meaningful output data, the input data must be correctly entered and checked after entry before the program is run. It is also a good, if not mandatory, policy to independently check the computer results the first time you run a new or modified program to ensure that the results are valid.

If the computer program used does not correct the output for the effects of altitude when the elevation of the project is equal to or greater than 2500 ft (760 m) above sea level, then the output must be manually corrected by using the appropriate correction factors, listed in App. A.

3.2.3.1 Trane Programs

The following summary describes programs available to the designer using Trane's CDS Duct Design program to size the duct systems.

Varatrain (Static Regain) Duct Design (DSC-IBM-113). With this duct-sizing program, the user inputs the duct layout in simple line-segment form with the cubic feet per minute for the zone, the supply fan value of cubic feet per minute, and the desired noise criteria (NC) level.

The program sizes all the ductwork based on an iterative static regain procedure and selects all the VAV boxes when desired. It identifies the critical path and down-sizes the entire ductwork system to match the critical-path pressure drop without permitting zone NC levels to exceed design limits.

The output of this program is an efficient, self-balancing duct design. It gives the designer a printout of the static pressure at every duct node, making troubleshooting on the jobsite a snap. The program will estimate the duct system and print a complete bill of materials, including schedule.

Equal-Friction Duct Design (DSC-IBM-108). This program outputs the total pressure as well as the pressure drop for each trunk section. The output also includes duct sizes, air velocity, and friction losses. The program can be used for fiber-glass selection.

The program will calculate the metal gauges, sheet-metal requirements, and total poundage and provide a complete bill of materials.

3.2.3.2 Carrier Program

The following summary describes the program available to the designer using Carrier's E20-II Duct Design to size the duct system.

Duct Design. This program:

- Uses the static regain and equal-friction methods simultaneously
- Calculates round and rectangular ducts
- Allows for sound attenuation and internally insulated ducts
- Permits material changes in duct system for different sections
- Shows balancing requirements between circuits in same duct system
- Is capable of handling up to 200 sections of ductwork in one system
- Calculates sheet-metal poundage and material quantities and shows them in the summary

3.2.4 REFERENCES

1. 1993 *ASHRAE Handbook, Fundamentals*, ASHRAE, Atlanta, GA, 1985, chap. 33, "Duct Design."
2. Carrier Corp., *Air Conditioning System Design*, McGraw-Hill, New York, 1965, part 2, chaps. 1–3.
3. Engineering Design Reference Manual for Supply Air Handling Systems, United McGill Corp., 1996.
4. Committee on Industrial Ventilation, *Industrial Ventilation—A Manual of Recommended Practice*, American Conference of Governmental Industrial Hygienists, Lansing, MI, 1989.

3.2.5 BIBLIOGRAPHY

Publications of the Air Diffusion Council, Cincinnati, OH.

SECTION 4

HEAT GENERATION EQUIPMENT

CHAPTER 4.1

BOILERS*

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4.1.1 INTRODUCTION

The term boiler applies to a device which (1) generates steam for power, processing or space heating or (2) heats water for processing, space heating or hot water supply. Generally, a boiler is considered a steam producer; however, most boilers used currently for space heating purposes are specially designed to produce hot water.

Boilers are designed to transmit heat from a high temperature source (usually fuel combustion) to a fluid contained within the boiler vessel. In some cases, the heat source may be a bank of electric resistance elements, or a bundle of heat transfer tubes. If the heat source is a high temperature fluid or electricity, the unit is said to be an "unfired" boiler. If the fluid heated is other than water, e.g., Dow-therm®, the unit is classified as a thermal liquid heater or vaporizer.

To ensure safe control over construction features, stationary boilers installed in the United States must be constructed in accordance with applicable sections of the ASME Boiler and Pressure Vessel Code. Known as the ASME Boiler Code, this group of publications contains rules governing the design, construction, manufacturing quality control, testing, installation and operation of boilers. Most states have adopted the ASME Boiler Code, in most cases in its entirety, providing governmental enforcement of the Code throughout the United States.

In addition, the National Board of Boiler and Pressure Vessel Inspectors, a group which comprises all of the Chief Boiler Inspectors of the States and other "jurisdictions" (some cities are separate jurisdictions within the States) provides rules for uniform boiler inspection procedures, both during manufacture and subsequently in field installation and operation.

Further evidence of compliance with good design practice and quality control is found in the product listing programs of "third-party" testing laboratories such as Underwriters Laboratories Inc. (UL) and the American Gas Association (AGA).

*Section 4.1.17, Electric Boilers, is based on Chapter 29 of the 1st edition, written by Robert G. Reid, CAM Industries, Kent, WA, as revised by Curt Diedrick, Precision Parts Corp., Morristown, TN. This chapter is a revision of the 1st edition chapter by Cleaver Brooks, Inc.

4.1.2 BOILER TYPES

Today's boiler industry manufactures a broad range of types and sizes of boiler encompassing tiny packaged residential hot water boilers through huge field-erected utility power generating boilers which might stand in excess of 200 feet (60 m) high.

Boilers are classified by the output form of the water being heated.

Steam boilers are classified for HVAC purposes as (1) low-pressure boilers with maximum allowable working pressure (MAWP) of 15 lb/in² (1.03 bar), constructed to ASME Section IV, or (2) high-pressure boilers, generally 150 lb/in² (10.3 bar) MAWP, constructed to ASME Section I.

Water boilers are generally constructed to ASME Section IV with maximum allowable working pressure to 160 lb/in² (11 bar) and maximum temperature 250°F (121°C).

Water boilers exceeding these Section IV limits are classified as medium or high-temperature hot water (MTHW or HTHW) boilers.

For HVAC purposes, most boilers are constructed as "packaged boilers." They are completely shop assembled with fuel burner, draft system, insulation and jacket and all controls. The advantages of the "packaged boiler" are:

1. Minimum installation work is required at the job site.

The boiler is mounted on an integral base ready to be moved into place on a simple foundation pad.

The connections required are (1) sources of water, fuel and electricity, (2) steam and condensate return piping (or hot water supply and return), (3) a stack for vent gases and (4) foundation anchor bolts.

2. The boiler is completely constructed in the boiler manufacturer's plant—standard models give minimum costs, fast lead times and optimum quality.
3. Responsibility for design and performance is assigned to a single source, the manufacturer. The boiler is test fired prior to shipping. A third-party (UL or AGA) label is further evidence of design approval and proper quality control.
4. The input-to-output efficiency of packaged boilers is relatively constant over the firing range which, depending on boiler size, varies from 60% to 100% to 25% to 100% capacity. The ratio between maximum and minimum firing rates is known as "turndown ratio." A boiler with a 50% minimum firing rate is said to have a 2:1 turndown ratio.
5. Packaged boilers save space and are adaptable to a wide variety of locations from subbasements to penthouses. Some manufacturers provide boilers equipped for outdoor operation.

4.1.3 OPERATING PRESSURE

Low-pressure heating boilers in the United States are fabricated in accordance with Section IV of the ASME Code, which limits the maximum allowable working pressure of low-pressure steam boilers to 15 psig (1.03 bar) and low-pressure hot water boilers to 160 psig (11 bar) at temperatures not exceeding 250°F (121°C).

In practice, while the above limits are labeled maxima, the practical *operating* limits are lower to allow for operation of pressure and temperature controls and relief valves. Realistic maximum operating values are:

- Low-pressure steam boilers 13.5 psig (0.93 bar)
- Low-pressure hot water boilers 140 psig (9.6 bar) at 230°F (110°C).
- For operating pressures or temperatures above these values, the boiler must be constructed to ASME Code Section I.

4.1.4 BOILER DESIGN CLASSIFICATIONS

Boiler designs can be broadly separated into three classifications, *water-tube*, *fire-tube* and *cast-iron sectional*.

- Water-tube boilers are constructed to contain water inside the tubes and other vessel members with hot combustion gases passing across the outside tube surfaces. See Fig. 4.1.1.
- Fire-tube boilers are built to channel hot combustion gases through the inside tube passages. See Fig. 4.1.2.
- Cast-iron sectional boilers are patterned after the fire-tube concept; however, the hot gas passages are formed into the multiple cast-iron sections which are bolted together.

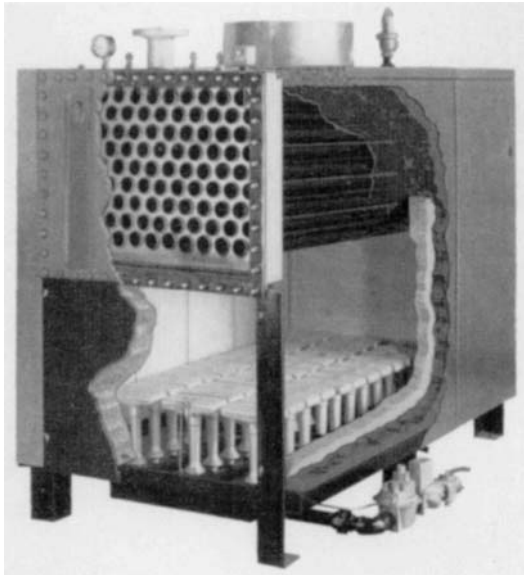


FIGURE 4.1.1 Atmospheric watertube boiler. (Courtesy of Ajax Boiler Inc.)

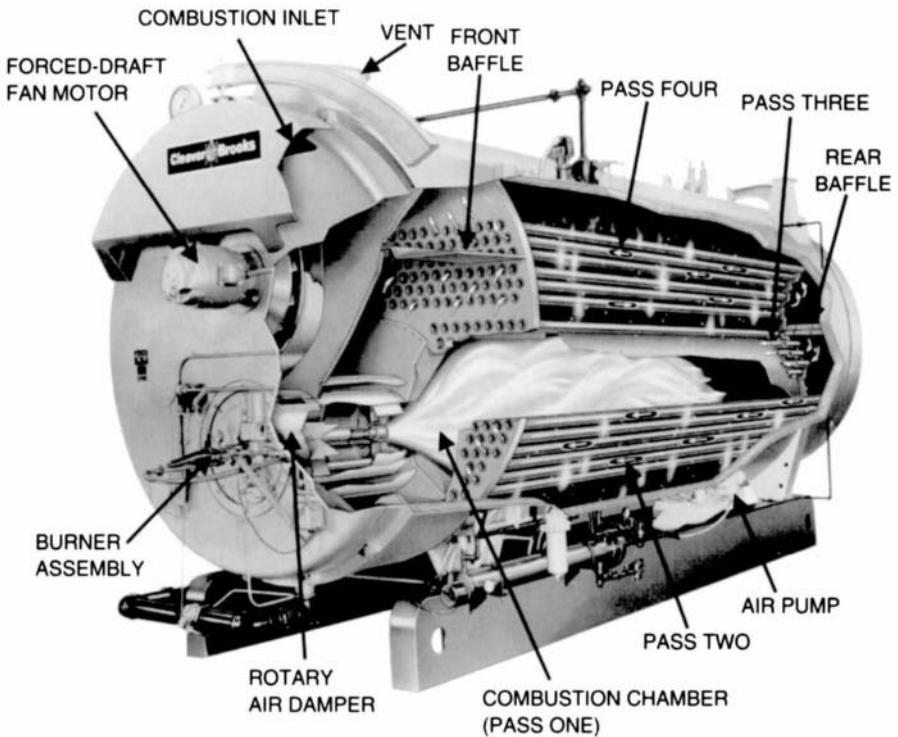


FIGURE 4.1.2 Firetube boiler. (Courtesy of Cleaver-Brooks.)

Further subgroups are, for water-tube boilers:

- straight tube (See Fig. 4.1.1)
- bent tube (See Fig. 4.1.3)
- coiled tube (See Fig. 4.1.4)

In all of these subgroups, tubes may be *plain* or *finned* and, while in most cases tube material is carbon steel, finned tubes tend to be copper or composite steel/copper construction.

Further subgroups for fire-tube boilers are:

- Scotch, in which the horizontal tube banks are housed within a horizontal cylindrical pressure vessel or “shell” (shown in Fig. 4.1.2)
- Firebox, where the horizontal tube bank and box-shaped shell are mounted above a refractory-lined “firebox” or combustion chamber
- Vertical fire-tube boilers, generally smaller in size where the fire-tubes are mounted vertically in a vertical, cylindrical shell.

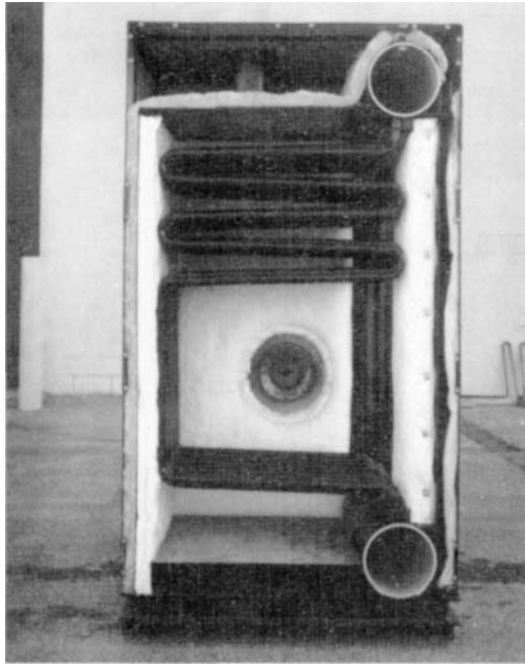


FIGURE 4.1.3 Bent tube watertube boiler. (Courtesy of Bryan Steam Corp.)

4.1.5 SELECTING A PACKAGED BOILER

There are several criteria involved in selecting a packaged boiler. These include:

1. The fluid to be produced (low pressure steam, high pressure steam, hot water, high temperature hot water).
2. The size of the unit (the rate of heat transfer).
3. The service—space heating, humidification air reheat, laundry, kitchen or domestic water system use.
4. The level of availability required and the need for redundant capacity. Generally, it is preferable to provide redundancy by having multiple boilers with a total capacity exceeding design load. For example, two boilers each capable of providing 75% of the required energy output would provide complete redundancy (100% backup) for a large part of the heating season.
5. Type of fuel, primarily natural gas or No. 2 fuel oil and, to a lesser degree, heavy fuel oil, grades 4 through 6 and, in remote locations, propane. Other types of fuel are available, e.g., coal, wood, biomass, but these are seldom used in conventional applications.
6. Type of combustion air system. For all fuel types, gas and oil, forced draft systems are available wherein combustion air is provided by a blower mounted on the inlet to the combustion chamber, generally part of the burner assembly.



FIGURE 4.1.4 Coiled tube copper high-fin boiler.
(Courtesy Ace Boiler Inc.)

Also available, for gas fuels only, are “atmospheric” boilers where combustion air is induced into the bottom of the combustion chamber by the action of the stack effect (the buoyancy of the hot gases rising up the stack or chimney.)

Atmospheric boilers are simpler and less expensive to buy and maintain than forced draft units, but generally are less efficient. Most smaller gas-fired boilers sold in the United States are atmospheric units. Larger gas-fired units, where improved operating efficiency outweighs increased first cost and maintenance costs, tend to be forced draft units.

A third option is the induced draft system, wherein a blower mounted in the boiler flue gas outlet draws gas through the boiler. In this case, the blower is handling flue gas and must be constructed for high temperature operation and corrosion resistance. The required volumetric flow from a draft inducer is approximately double that of the equivalent forced draft blower.

7. Controls system complexity
8. Emissions control requirements
9. Location, available space, and access limitations

10. Noise levels
11. Life cycle costing, including warranty coverage

4.1.6 GENERAL DESIGN CRITERIA

There are several design criteria which apply to all types of boiler.

1. The combustion system must operate effectively to provide complete combustion within the area designated as the combustion chamber. The definition of "complete" here depends upon local air quality regulations. In the absence of specific regulations, maximum limitations of 100 ppm (parts per million) carbon monoxide (CO) and 250 ppm Nitrogen oxides (NO_x) are generally deemed acceptable. Refer to Section 4.1.10 of this chapter for more information on emissions.
2. The combustion chamber must contain sufficient water-backed surface, referred to as "radiant heating surface," to absorb radiant heat from the flame zone without "steam-packing." Steam-packing occurs when all the water in the tube turns to steam at which point the steam becomes superheated and tube metal temperatures rapidly rise to unacceptable, even damaging, levels.
3. Hot gases leaving the combustion chamber must pass across the water-backed heat transfer surfaces, referred to as convective heating surface, with sufficient velocity to effectively transfer heat through the hot surface film. Each unit area of tube surface will transfer an increasing amount of heat as scrubbing velocity increases.
4. There must be adequate heat transfer surface to absorb an appropriate amount of heat from the gases leaving the combustion chamber. The generally accepted criterion for "adequate" here is 5 sq. ft. of heat transfer surface per boiler horsepower (0.0474 m²/kW) although successful and efficient boiler designs exist with between 4 and 9 sq. ft. per boiler horsepower (0.0379–0.0853 m²/kW).

The definition of boiler heating surface is often a subject of controversy. Heating surface continues to be defined in the appropriate sections (I and IV) of the ASME BPV Code and reference should be made to the current version of these publications in the event a dispute arises.

5. Furnace Heat Release. The furnace heat release rate per unit of furnace volume has, for many years, been a governing factor in the selection of boilers. Current packaged boiler designs utilize furnace heat release rates as high as 150,000 Btu/hr/ft³ (1550 kW/m³). While it is clear that the permissible furnace heat release rate depends upon the design and relative placement of water-backed and refractory surfaces, optimum emissions (NO_x, CO) levels are obtained in these boilers with low furnace heat release rates, generally not exceeding 70,000 Btu/hr/ft³ (725 kW/m³).
6. The boiler must function with minimum excess air. "Excess air" is the term used to describe the air entering the combustion process whose oxygen content is not consumed in burning the fuel. This air appears at the boiler stack and can be measured in terms of the oxygen content of the stack gases. Excess air is usually expressed as a percentage of the stoichiometric requirement. Air which passes through but does not impact the combustion process wastes en-

ergy because fuel has been consumed to heat the excess air to the boiler exit temperature and, in the case of forced draft units, electrical power has been wasted in blowing the excess air through the boiler.

Some degree of excess air is inevitable, however, since (1) none of the available combustion processes provides completely homogeneous fuel-air mixing and (2) allowance must be made for the effects of wear on the burner air fuel ratio controls. Further, in some recent low emissions designs, a high level of excess air is used to lower combustion chamber temperatures and thus reduce formation of nitrogen oxides.

7. Water circulation within the boiler must be adequate to carry heat away from localized high temperature areas (hot spots) and thus prevent damage from overheating. In a water boiler this is particularly important since hot spots may result in the localized generation of steam bubbles which, on moving to lower temperature areas, collapse, resulting in noise and vibration.

In steam boilers, circulation is further complicated by the need to provide proper "disengaging" space for the steam bubbles to break free of the water surface and adequate internal circulation to allow continuous delivery of water and steam-water mix to the surfaces receiving heat from the combustion process and prevent "steam-packing." In most steam boilers, this circulation is generated through a designated flow path of heated "riser" passages and unheated "downcomer" passages.

8. In steam boilers, boiler size must be adjusted to take account of "factor of evaporation." Steam boilers in lower pressure ranges [up to 150 psi (10.3 bar)] are generally rated on a "from and at 212°F (100°C) basis. This identifies the performance as though available heat is used only to boil the water at a temperature of 212°F (100°C) at atmospheric pressure. In fact, water in steam boilers must first be heated from entering temperature to boiling temperature and then boiled and then, where applicable, heated to superheat temperatures. Table 4.1.1 gives factors of evaporation in lb/bhp.
9. Water level controls must be properly applied, installed and maintained. Failure to maintain a high enough water line in the boiler will inevitably result in damage to the pressure vessel with possible failure. Too high a water level in steam boilers will result in abnormally wet steam and carryover of water into the steam piping system degrading the heat transfer system and overworking condensate traps.
10. In hot water heating applications, the boiler must be selected appropriately and the system designed to avoid "thermal shock." Thermal shock occurs when a rapid reduction in inlet water temperature results in changes in temperature-induced stresses in boiler pressure vessel components. In extreme cases, conflicting expansion-contraction loads can result in failure of the pressure vessel requiring substantial repairs or even complete vessel replacement. Water-tube boilers are generally more resistant to thermal shock; however, good design practice dictates selection of hot water boilers with long-term (20 years or longer) warranties against thermal shock damage.
11. In hot water heating applications, operation with boiler inlet water temperatures below condensing should be minimized. The temperature at which water vapor in combustion products gases will condense is approximately 135°F (57°C). Condensation will occur anytime combustion products come into contact with boiler metal surfaces at or below this temperature. While some boilers are designed to accept condensing in order to obtain ultra high efficiencies, con-

TABLE 4.1.1. Factor of Evaporation, lb/bhp Dry Saturated Steam

Feed-water temp., °F	Gauge pressure, psig																	
	0	2	10	15	20	40	50	60	80	100	120	140	150	160	180	200	220	240
30	29.0	29.0	28.8	28.7	28.6	28.4	28.3	28.2	28.2	28.1	28.0	28.0	27.9	27.9	27.9	27.9	27.9	27.8
40	29.3	29.2	29.1	29.0	28.9	28.7	28.6	28.5	28.4	28.3	28.2	28.2	28.2	28.2	28.2	28.1	28.1	28.1
50	29.6	29.5	29.3	29.2	29.1	28.9	28.8	28.8	28.7	28.6	28.5	28.5	28.4	28.4	28.4	28.3	28.3	28.3
60	29.8	29.8	29.6	29.5	29.4	29.2	29.1	29.0	28.9	28.8	28.8	28.7	28.7	28.6	28.6	28.6	28.6	28.5
70	30.1	30.0	29.9	29.8	29.7	29.5	29.4	29.3	29.2	29.1	29.0	29.0	28.9	28.9	28.9	28.3	28.8	28.8
80	30.4	30.3	30.1	30.0	30.0	29.8	29.6	29.6	29.5	29.3	29.2	29.2	29.2	29.2	29.1	29.1	29.1	29.0
90	30.6	30.6	30.4	30.3	30.2	30.0	29.9	29.8	29.7	29.6	29.5	29.5	29.4	29.4	29.4	29.3	29.3	29.3
100	30.9	30.8	30.6	30.6	30.5	30.3	30.2	30.1	30.0	29.8	29.8	29.8	29.7	29.7	29.7	29.6	29.6	29.6
110	31.2	31.2	30.9	30.8	30.8	30.6	30.4	30.3	30.2	30.0	30.0	30.0	30.0	30.0	29.9	29.9	29.8	29.8
120	31.5	31.4	31.2	31.2	31.1	30.8	30.7	30.6	30.5	30.4	30.3	30.3	30.2	30.2	30.2	30.1	30.1	30.1
130	31.8	31.7	31.5	31.4	31.4	31.1	31.0	30.9	30.8	30.7	30.6	30.6	30.5	30.5	30.4	30.4	30.4	30.4
140	32.1	32.0	31.8	31.7	31.6	31.4	31.3	31.2	31.1	31.0	30.9	30.8	30.8	30.8	30.8	30.7	30.7	30.6
150	32.4	32.4	32.1	32.0	31.9	31.7	31.6	31.5	31.4	31.2	31.2	31.2	31.1	31.1	31.0	31.0	30.9	30.9
160	32.7	32.7	32.4	32.4	32.3	32.0	31.9	31.8	31.7	31.5	31.4	31.4	31.4	31.4	31.3	31.3	31.2	31.2
170	33.0	33.0	32.7	32.6	32.6	32.3	32.2	32.1	32.0	31.8	31.7	31.7	31.7	31.6	31.6	31.6	31.5	31.5
180	33.4	33.3	33.0	33.0	32.9	32.6	32.5	32.4	32.3	32.2	32.1	32.0	32.0	32.0	31.9	31.9	31.8	31.8
190	33.8	33.7	33.4	33.3	33.2	32.9	32.8	32.7	32.6	32.5	32.4	32.4	32.3	32.3	32.2	32.2	32.1	32.1
200	34.1	34.0	33.7	33.6	33.5	33.2	33.1	33.0	32.9	32.8	32.7	32.6	32.6	32.6	32.6	32.5	32.4	32.4
212	34.5	34.4	34.2	34.1	33.9	33.6	33.5	33.4	33.3	33.2	33.1	33.0	33.0	33.0	32.9	32.9	32.8	32.8

Note: These metric conversion factors can be used: 1 psig = .069 bar, 1 lb = 0.45 kg, and °C = $\frac{1}{2}(\text{°F} - 32)$, 1 bhp = 9.81 kW.

ventional boilers of steel construction will suffer corrosion damage if operated in this mode, as will conventional steel boiler stacks.

12. The boiler must, when completely assembled, be capable of being installed in the space available, including allowance for access areas and periodic maintenance functions such as inspection and tube replacement.
13. The following installation features must be properly designed: Foundations, electrical supply, water supply, relief valve venting, combustion air supply, noise parameters, and alarm systems.

4.1.7 WATER-TUBE BOILERS

4.1.7.1 Operating Pressure

Water-tube boilers are available for all operating pressures from 15 psi (103 kPa) through the ultra-high pressures used in utility boilers which often exceed 3500 psi (241 bar).

The most common design pressures are 15, 150, 200, 250 and 300 lb/in² (1.03, 10.3, 13.8, 17.2, 20.7 bar) for steam boilers, 30, 60, 125 and 160 lb/in² (2.1, 4.1, 8.6, 11.0 bar) for water boilers and 300, 400 and 500 lb/in² (20.7, 27.6, 34.5) for HTHW boilers.

4.1.7.2 Size Range

Water-tube boilers are available in all sizes from residential through large utility power generation boilers. Above 800 bhp (7849 kW), water-tube boilers are used almost exclusively since the large rolled shell of the scotch boiler becomes prohibitively expensive, both to manufacture and to transport.

In recent years, small packaged water-tube boilers, ranging to 800 bhp (7849 kW) have become the preferred design for hot water space heating applications. This preference has developed because, unlike fire-tube boilers, water-tube boilers are largely impervious to and invariably guaranteed against damage caused by "thermal shock." Thermal shock usually occurs when a hot boiler is subjected to a surge of cold water. However, with some fire-tube designs, continuous operation outside a limited temperature differential band (outlet temperature minus inlet temperature) has the same effect. In most fire-tube designs, thermal shock causes large differential expansion forces which often loosen rolled tube joints and, in extreme cases, result in rupture of the boiler vessel.

4.1.7.3 Types of Water-Tube Boiler

1. Straight tube: This type consists of parallel tubes joined at each end to a heater box which may be rectangular or cylindrical. Straight tube boilers are generally of the horizontal inclined tube pattern (see Fig. 4.1.5) but may have vertical tubes with headers at top and bottom.
2. Bent tube: This type has a number of variants.
 - (a) Serpentine tube: This variant incorporates tubes bent into a multiple pass arrangement connected top and bottom to one or more drums (see Fig.

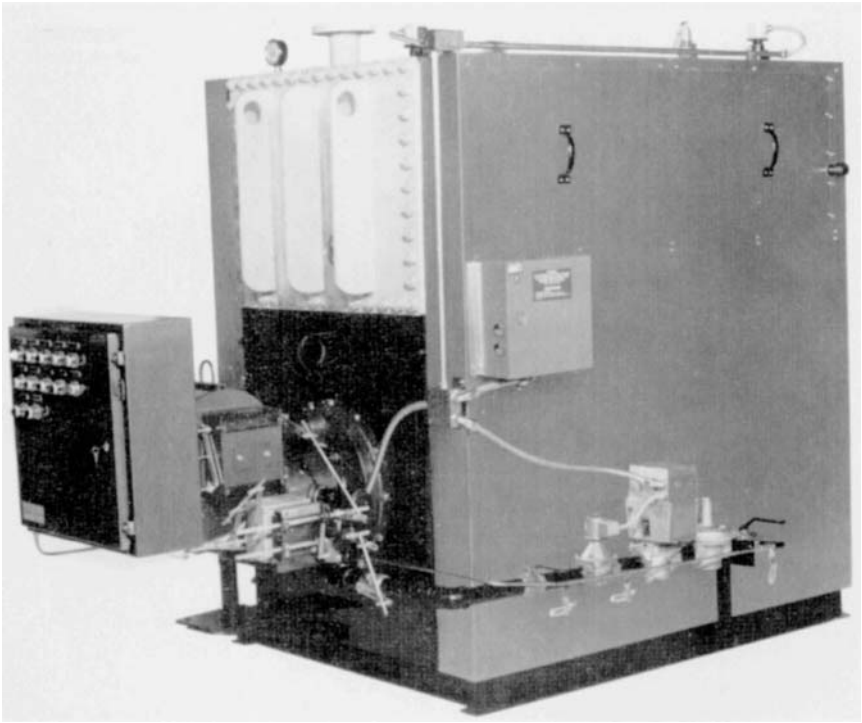


FIGURE 4.1.5 Straight tube watertube boiler. (Courtesy Ajax Boiler Inc.)

4.1.3). In some designs, the tubes are individually connected to the drums using mechanical taper joints.

- (b) D-style: This unit consists of an upper drum and a lower drum connected by tubes (see Fig. 4.1.6).
 - (c) A-style: Fig. 4.1.7 shows a typical A-style boiler comprising a single upper drum and the lower drums in symmetrical pattern.
 - (d) O-style: Similar to A-style, but with one lower drum (see Fig. 4.1.8).
3. Coiled tube: This type of boiler is used generally up to around 350 bhp (3334 kW) and has a vertical cylindrical coil comprising one or more tube flow paths (see Fig. 4.1.4).

4.1.7.4 Watertube Boiler Design

1. Pressure Vessel: Watertube boilers use drums fabricated from steel pipe or rolled steel plate. Small drums are equipped with inspection openings at each end. Large drums requiring entry for internal inspection and maintenance are equipped with manways. In smaller watertube boilers, upper and lower drums are connected using downcomer tubes located in the coolest section of the boiler to enhance downward flow. In larger boilers, the upper drum is generally connected to the lower drum only by the boiler tubes. In steam boilers, the upper

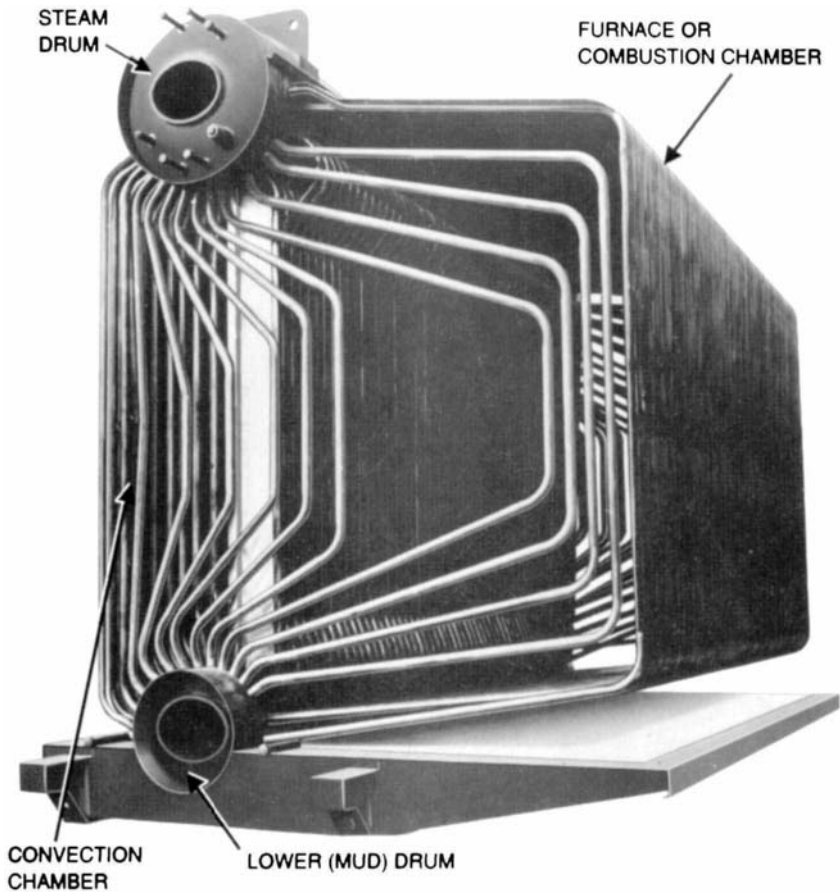


FIGURE 4.1.6 D style watertube boiler. (Courtesy of Cleaver-Brooks.)

drum will contain baffling to direct and dry the steam before it exits the boiler (see Fig. 4.1.9).

2. **Tubes and Tube Attachments:** The most commonly used watertube material is SA-178 steel and tube sizes vary between 1" (25.4 mm) and 2" (50.8 mm) outside diameter. Tubes may be straight or bent. On smaller units, straight tubes facilitate inspection and mechanical cleaning of inside surfaces. For bent tubes, good design practice requires that tubes maintain their round cross-section in the bends.

Tubes are generally expanded into drums and tube sheets. However, some smaller boilers are provided with mechanical tube fittings to allow for replacement without tube rolling and some boilers may have tube joints which are welded in addition to being rolled. In most instances, straight tubes with rolled joints provide the most economical replacement potential.

Tubes which are not vertical must be sloped to encourage convection flow. The exact amount of slope depends on the location of the tubes in the boiler. Low pressure boilers with large (2"/50.8 mm) tubes need relatively little pitch

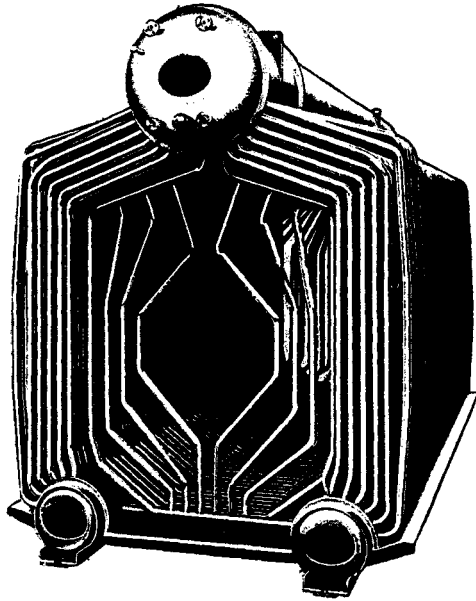


FIGURE 4.1.7 A-style boiler. (Courtesy of Cleaver-Brooks.)

or slope but higher pressure boilers, or those with smaller ($1''/25.4$ mm or less) diameter tubes, should be pitched with minimum slope from horizontal as follows. All furnace floor tubes must have a minimum slope of 6.5° to the horizon to achieve good circulation and drainage. All furnace roof tubes must have a minimum slope of 7.5° to the horizon to permit good circulation and maximum steam-relieving capacity.

3. Furnace Design (Six Wall Cooling): Furnace design is important because as much as 50% of the total heat transfer can occur within the furnace. Several surfaces are used to contain the heat of the combustion process and channel it to the heat-absorbing surfaces (see Fig. 4.1.10).
 - (a) Tangent tube walls provide a single row of tubes placed adjacent to one another.
 - (b) Multiple-row tube walls provide more water flow per square foot of radiant heating surface. A double-row configuration maximizes radiant heating surface and extends boiler life.
 - (c) Finned Tube walls. Fins are welded to the tubes to extend external heating surface. The tube wall temperature is higher with this type of wall because less cooling water is available per unit of heat-absorbing surface.
 - (d) Membrane Tube Walls. Solid fins are welded between tubes in this construction. The tube wall temperature is higher than with plain tube construction, as with finned tubes.
 - (e) Refractory walls. Many boilers are constructed with no water-backed surface in one or more of the furnace walls and/or the furnace floor. In this case, the material of construction is generally refractory cement backed with high

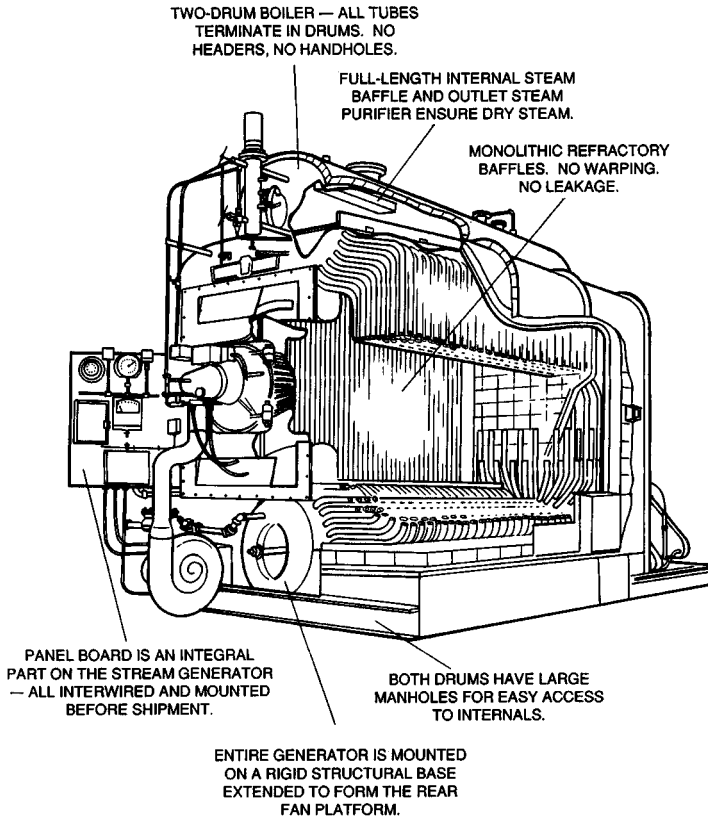
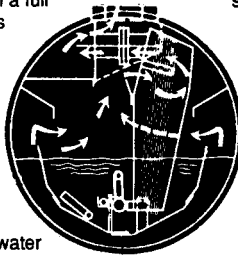


FIGURE 4.1.8 O-style boiler. (Courtesy of Cleaver-Brooks.)

Heater control over varying loads. This big 42" O. D. steam drum comes with a full complement of steam dryers, plus Cleaver-Brooks' patented water level control baffles. This combination results in a dry steam product even when load swings far beyond the ordinary. The baffles prevent diluting of the entering steam/water mixture through reservoir water. This results in more effective steam separation and greatly improves water level control in the drum.

Cleaver-Brooks' exclusive patented steam



purifiers are also available to meet the solids concentration requirements of central station installations.

Extra storage capacity, easier access. Two 24", I. D. lower drums mean that CA steam generators keep more water on reserve to meet sudden load demands. The steam drum and the lower water drums have 12" x 16" manways at each end — providing access for servicing and eliminating troublesome leaking handhole plates normally required with header-type drums.

FIGURE 4.1.9 Steam separator-drum internals. (Courtesy of Cleaver-Brooks.)

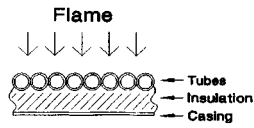
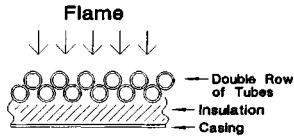
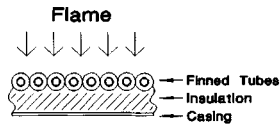
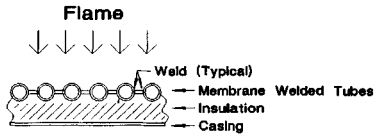
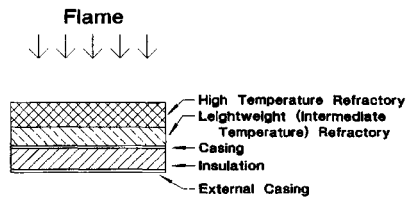
A) Tangent Tube Walls**B) Multiple-row Tube Walls****C) Finned Tube Walls****D) Membrane Tube Walls****E) Refractory Walls**

FIGURE 4.1.10 Furnace wall construction. (Courtesy Ajax Boiler Inc.)

temperature insulation. The hot surface material may be formed of refractory clay material or ceramic fiber product. In either case, the material may be applied by spreading or may take the form of preformed panels.

4. Convection Heating Surface: Convection heating surface is designed to incorporate the maximum number of tubes in the smallest possible space consistent with flue gas pressure drop limitations and adequate accessibility to clean and, if necessary, replace tubes.

Sootblowers are sometimes provided in convection sections when heating oil or solid fuels are fired.

5. Boiler Casing and Insulation: Modern watertube boilers with forced draft combustion systems use pressurized furnaces to maximize flue gas pressure drop across the convection tube banks. Two types of casing are used; membrane and double-wall.

(a) Membrane construction. Membranes between the tubes in the outermost tube rows or a continuous membrane casing outside the tubes provide a means of containing the hot combustion gases. The membrane is backed by insulation or an insulation/air gap combination (see Fig. 4.1.10d).

(b) Double-Wall construction (Fig. 4.1.11). Double-wall constructions consist of an inner and outer casing with either insulation or circulated combustion air between the casings. The inner casing is welded or otherwise sealed to provide a leakproof containment for the two combustion gases.

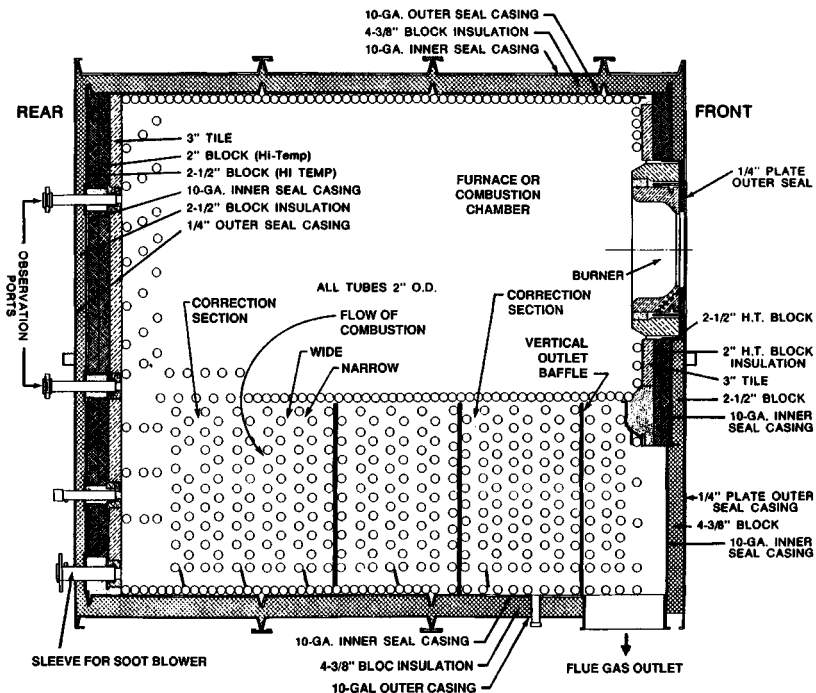


FIGURE 4.1.11 Double-wall construction. *Note:* This is the plan of a D-type boiler. (Courtesy of Cleaver-Brooks.)

Insulation is laid over the inner casing to reduce heat losses or, in some cases, the gap between the inner and outer casings is arranged to form a channel for combustion air flow. By this means, the heat energy which would have been lost to the boiler room is captured by the combustion air and returned to the furnace.

The outer casing provides additional strength, a cover for the insulation and an aesthetic appearance.

4.1.8 FIRE-TUBE BOILERS

Fire-tube boiler designs originated many years ago and form the basis for many of the modern boiler pressure vessel/combustion chamber concepts. The needs for conservation of space and improved energy conversion efficiencies have resulted in modification to the early designs, but the basic functional principle remains unchanged.

4.1.8.1 Operating Pressure

Fire-tube boilers are commonly available for maximum allowable working pressures up to 150 psi (10.3 bar). Some manufacturers build custom scotch units to 300 psi (20.6 bar); however these are generally limited in size to 250 boiler horsepower (2453 kw) because of the high cost of producing the rolled cylindrical outer shell.

4.1.8.2 Size Ranges

Fire-tube boilers are generally available in the range 20 through 800 bhp (196–7848 kW) and in pressure up to 150 psi (10.3 bar).

The larger units, 150 hp (1471 kW) and above tend to use the scotch design.

The scotch boiler, used for many years as the mainstay of marine propulsion boilers, is rugged and dependable; however, its application to water heating is limited (see “Thermal Shock” section 4.1.6.10 of this chapter).

4.1.8.3 Types of Fire-Tube Boilers

1. The modified scotch boiler (see Fig. 4.1.2) is the most readily recognizable type of fire-tube boiler though not, in fact, the most prolific. In this type, the burner fires into a cylindrical steel combustion chamber after which the hot gases pass through one, two or three tube passes before leaving the boiler. Two, three and four pass boiler gas flows are identified in Fig. 4.1.12. The combustion chamber and all of the tubes are immersed in boiler water inside a larger cylindrical pressure vessel, or shell.

Scotch boilers are further classified into “dryback” and “wetback” types. In the dryback boiler, the “turnaround space” in which combustion gases are directed from combustion chamber to tube-pass and from tube-pass to tube-pass is an insulated steel casing. In the wetback design, the same enclosure is water cooled.

2. The firebox boiler (see Fig. 4.1.13) comprises a bank of fire tubes immersed in boiler water mounted adjacent to, generally above, a combustion chamber fire-

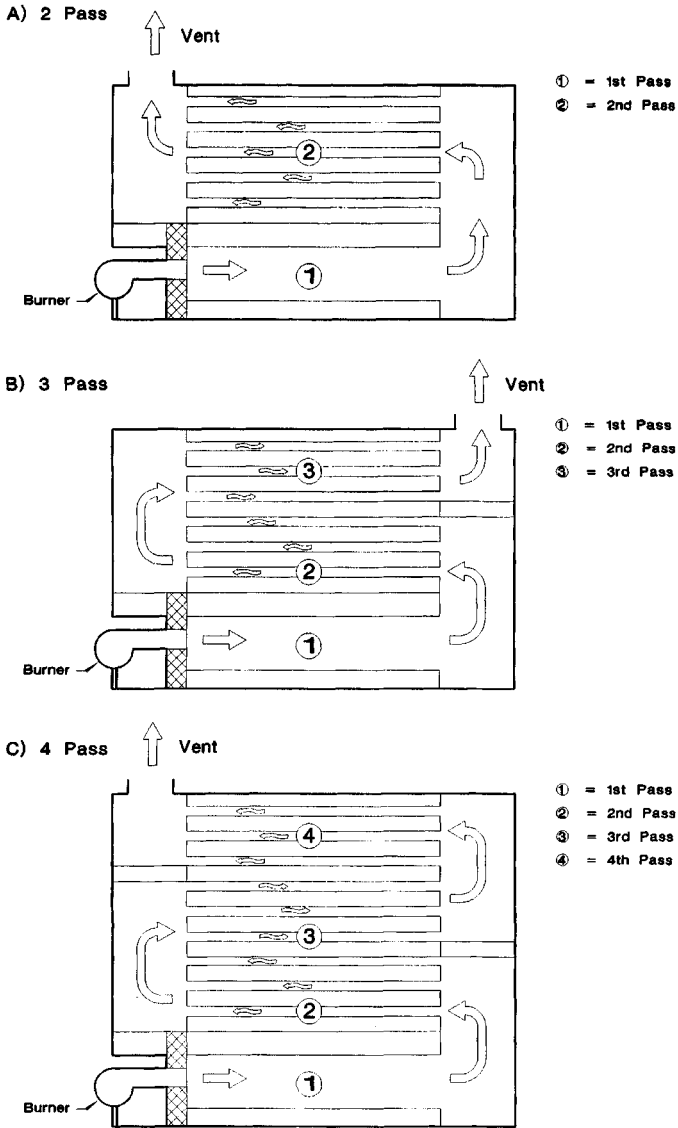


FIGURE 4.1.12 Firetube boiler pass arrangements. (Courtesy Ajax Boiler Inc.)

box) lined with high temperature refractory materials. Earlier firebox boilers were often factory-built pressure vessels mounted on field erected fireboxes, but current types are almost universally totally factory packaged with combustion chamber and shipped complete to the jobsite.

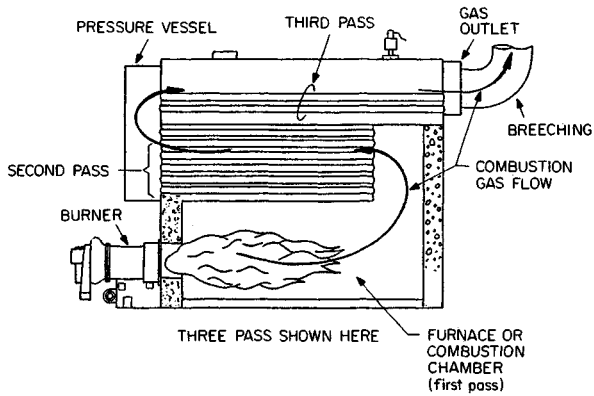


FIGURE 4.1.13 Firebox boiler. (Courtesy of Cleaver-Brooks.)

Packaged firebox boilers are produced by a large number of manufacturers in widely varying formats and in sizes up to around 300 bhp (2943 kW). In the typical current design, the pressure vessel is rectangular in shape (see Fig. 4.1.14) with water legs framing the sides and rear of the combustion chamber.

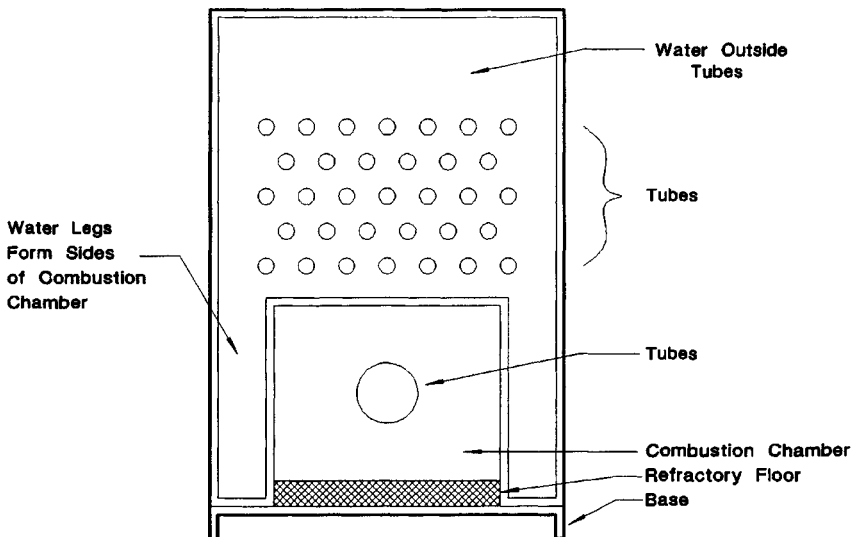


FIGURE 4.1.14 Cross section through typical packaged firebox boiler. (Courtesy Ajax Boiler Inc.)

4.1.8.4 Firetube Boiler Design

1. **Pressure Vessel:** Firetube boilers use round drums rolled from steel plate or rectangular drums and headers fabricated and welded from steel plate. Rectangular boilers are fabricated in sizes through 400 bhp (3924 kW) and cylindrical, or scotch, boilers are fabricated in sizes up to 1000 bhp (9810 kW). In all cases, the relatively large expanses of flat pressure vessel surfaces often require stay rods for support against internal pressure.

2. **Tubes and Tube Attachments:** The most commonly used tube material is SA178 steel and tube sizes vary between 1½" (38 mm) and 3" (76 mm) outside diameter.

Tubes are generally straight but are occasionally used in a bent pattern. Straight tubes facilitate inspection and mechanical cleaning.

Tubes are normally expanded into tube sheets and, in some cases, seal welded, especially for those joints exposed to combustion gases leaving the combustion chamber.

3. **Furnace Designs:** Firetube boilers utilize widely varying furnace designs. The simplest design locates the boiler on a five-sided refractory-lined box (four walls and a floor). When this box is constructed from bricks, the boiler is said to be *brickset*. In an adaptation of the design for packaging purposes, the firebox is mounted inside a steel casing to which the boiler pressure vessel is bolted or welded.

Further adaptations have water legs descending from the boiler pressure vessel to the floor of the firebox to form the water-backed walls for the combustion chamber. In the scotch boiler, the cylindrical combustion chamber is completely water-backed (see Fig. 4.1.2).

4.1.9 CAST-IRON BOILERS

4.1.9.1 Operating Pressure

Cast-iron boilers are used to provide low pressure steam 15 lb/in² (1.03 bar) maximum and hot water up to maxima of 30 lb/in² (2.06 bar) and 250°F (121°C).

4.1.9.2 Size Ranges

The primary use for cast-iron boilers lies in the residential market; however, small commercial sizes are available up through 200 bhp (150 kw).

4.1.9.3 Types of Cast-Iron Boilers

Cast-iron boilers can be classified as sectional with internal push-nipples or sectional with external headers (drums).

Sectional boilers are made up of vertical sections and look much like a loaf of sliced bread. The number of sections determines the boiler capacity since each inside section of a particular model are identical. The end sections are, of course, different from the inside sections. Because of the sectional construction mode, cast-iron boilers have a distinct advantage over other types since individual sections can

be moved into the boiler room and then assembled. Even so, individual sections are heavy and may weigh as much as 2000 lb (910 kg).

Most modern cast-iron boilers are built with water-filled spaces that completely surround the combustion chamber. The type of boiler is called a wet-base boiler and its design permits unrestricted circulation of water within the base and sides of each section. This design increases heat absorption surface and reduces boiler heat loss to the mounting pad.

Internal Push-Nipple Boilers: The cast sections of internal push-nipple boilers are joined with tapered nipples and held tightly together with tie rods or bolts. The gas joints between sections are sealed with gasket material. The nipples are sized to provide the required free circulation of water and/or steam from section to section.

External Header Boilers: Each cast section of the external header boiler is connected to the headers (drums) with screwed nipples. There are three headers, a supply manifold centered at the top and two at the bottom, one at each side.

With this type of boiler, it is possible to temporarily isolate a damaged section and put the boiler back into service in the event that one section becomes cracked.

4.1.9.4 Cast-Iron Boiler Design

Because of their sectionalized design, cast-iron boilers can be field assembled or factory assembled complete with burner, valving trim and prewired panel-mounted controls.

Sectionalized cast-iron boilers are generally designed for low draft loss. This feature is particularly beneficial to boilers with atmospheric burners.

Cast-iron boilers, due to their relatively heavy (thick) sections, are intrinsically slow in heating up. Because of the mass of iron, these boilers can retain heat and dampen temperature swings. The system designer should, however, take steps to prevent thermal shock-producing conditions since hot cast-iron sections may crack if “slugged” with cold water.

More recent cast-iron boilers follow the wet-base principle, where the flame is surrounded by water passages.

4.1.10 SPECIFIC DESIGN CRITERIA

Several additional design criteria should be examined during the selection process. These are efficiency, emissions control equipments, control system complexity and life cycle costing.

4.1.10.1 Efficiency

In HVAC terms, efficiency is generally defined as the ratio between heat output in the fluid being heated and heat input in the form of fuel being consumed. Typical efficiencies for contemporary boiler packages range from 75% to 95% based on this definition, with the majority in the 80% to 85% range. There are several caveats related to efficiency-based selection of boiler products.

- (a) Higher efficiencies, those over 85%, require partial or complete condensation of the water vapor contained in the flue gas. This requires inlet water temper-

atures below 135°F (57°C), a condition generally not available in cold climate hydronic heating systems.

- (b) Boilers which will operate in a partial or fully-condensing mode require corrosion-resistant stack or chimney and provisions for safe drainage of the acidic condensate.
- (c) In general, the higher the claimed operating efficiency, the higher the first cost of the boiler. In some cases, first cost increases by a factor of two to four times. Examine the fuel savings payback before selecting a high efficiency boiler.
- (d) High efficiency boilers are invariably more complex than standard efficiency units and so expect maintenance costs to be higher.

4.1.10.2 Emissions Control Requirements

Specific emissions control requirements are mandated nationally and locally and may also be specified by the owner or design engineer.

Emissions controls fall into the following categories: Smoke, carbon monoxide, nitrogen oxides, and sulphur oxides.

- (a) Smoke is primarily a fuel-based issue. Natural gas and propane are inherently smoke free. Fuel oil is inherently more smoky as fuel grade number increases. The normal benchmark for smoke measurement is the Bacharach smoke scale where 0 represents a clean stack and 9 represents a smoke level approximately equivalent to 10% obscuration through the stack plume. Note that the Bacharach scale covers the lower 10% of the older Ringelmann scale and thus provides a far more sensitive means of measurement. The pump/filter apparatus used in the Bacharach test also provides an easy means of measurement at the boiler outlet as opposed to visual observations at the stack.
- (b) Carbon Monoxide (CO) is a poisonous pollutant emitted from all carbon-based combustion processes. Stationary boilers are required to emit no more than 300 ppm by volume of carbon monoxide in the flue gas but most properly designed boilers will, in good time, provide CO levels of 50 ppm or less.
- (c) Nitrogen oxides (NOx) are a constituent of acid rain and photochemical smog and are, in some areas of the US, severely restricted. In Southern California, for instance, NOx levels are limited to 30 ppm by volume, corrected to 3% oxygen. The correction factor is:

$$\text{Corrected NOx at 3\% O}_2 = \text{Measured NOx (ppm)} \frac{18}{21 - \text{measured O}_2} \quad (1)$$

NOx generation is dependent on both fuel and combustion system type. Natural gas premix-type combustion systems provide the lowest NOx levels (10–20 ppm) followed by forced draft natural gas boilers with flue gas recirculation (FGR) (20–30 ppm). Forced draft No. 2 oil systems typically run 60–80 ppm with flue gas recirculation and 80–120 ppm without FGR. Low nitrogen No. 2 fuel oils are available which can give NOx levels in the 35–50 ppm range with FGR.

- (d) Sulphur Oxides (SOx) are emitted when fuels containing sulphur are burned. Fuel oils and coal contain sulphur in varying degrees and, in HVAC systems, the primary form of SOx control is by fuel selection. Clean fuels, like natural

gas, contain no sulphur and therefore produce no SO_x emissions. No. 2 fuel oil contains only limited amounts of sulphur and is generally considered "clean" in terms of SO_x emissions. Coal and heavy oil burning systems generally require "back end cleanup," usually in the form of wet scrubbers to meet SO_x requirements.

4.1.10.3 Control System Complexity

The level of controls complexity is primarily mandated by insurance code and/or by local, state or national codes. The owner's insurer will require that safety controls on the boiler (fuel shut-off valves, water level controls, limit controls etc.) conform to one of the available safety codes (FM, IRI, etc.).

In addition, many states have adopted the ASME Controls and Safety Devices Code CSD-1 and some states and local jurisdictions have their own specific safety requirements.

Once safety requirements have been satisfied, controls complexity may be limited to simple, stand-alone on-off automatic operation or may expand to include staged or fully-modulated firing, outdoor reset temperature control of water boilers, lead-lag control of multiple boilers, microprocessor burner management/flame safeguard systems which provide diagnostics and status and fault communication and communication with building energy management systems.

4.1.11 SYSTEMS AND SELECTIONS

4.1.11.1 Load Analysis

Once the heating and/or process loads have been calculated, a decision must be made to use a steam, water or high temperature hot water system. A comparative study of capital, operational and maintenance costs should be made, including electrical requirements.

The frequently cited advantages attributed to hot-water systems are:

1. Lower operational temperatures reduce system heat losses and improve boiler efficiency.
2. No condensate return systems required.
3. Need for steam traps (and associated maintenance costs) eliminated.
4. Smaller pipe size for a given load.
5. Quieter operation than steam.
6. More uniform space heating environment.
7. Minimizes water treatment and make up water requirements.
8. Lower levels of piping expansion and contraction.

Advantages cited for steam systems are:

1. Higher temperature available at user.
2. Constant temperature at user at condensing condition.
3. Improved heat transfer performance allows smaller user equipment.

4. Fast response to load change.
5. Reduced pumping effort required (less electrical power).
6. Eliminates generator corrosion caused by operation at condensing temperatures.

4.1.11.2 Number of Boilers Required

Use of a single boiler to provide 100 percent of the required system heat is permissible provided that loss of the boiler for a time will not seriously curtail operation of the facility. In most cases however, a complete loss of heat and/or process steam cannot be tolerated and multiple boilers with redundant capacity must be used.

Sizing options include selecting two boilers each sized for two-thirds of the required capacity. For most of the plant operating load range, one boiler will carry the load. In the event of a breakdown, the other boiler will carry nearly full plant load.

Variations include larger numbers of boilers and uneven sizing to cover widely varying plant load conditions, including lower loads during summer periods.

Heating Boilers: The total or "gross" load, in Btu/hr or W, for the building comprises:

1. Conventional heating load including building heat loss and process requirements.
2. Domestic water heating requirements, if boiler water is to be used to generate hot potable water. Design guidelines for estimating hot water usage are given in Table 4.1.2.
3. Air-conditioning load which may, in some locations, give a summer boiler water demand which exceeds the winter heating demand.
4. Pickup allowance, which covers the need to heat the building from cold within a limited period of time. This is typically a factor in mild climates where the

TABLE 4.1.2 Hot-Water Demands and Use for Various Types of Buildings

Fuel	Nominal capacity range in $I = B = R$ net rating			
	Push-nipple boiler		Header (drum) boiler	
	Steam, lb/h (kg/h)	Hot water, MBtu/h (kW)	Steam, lb/h (kg/h)	Hot water, MBtu/h (kW)
Gas	25.2–4,035 (11.4–1,836.3)	26–4,160 (7.6–1,220)	1,008.8–11,349 (457.6–5,147.9)	1,040–11,700 (304.8–3,429.3)
Oil	76.5–407 (34.7–184.6)	79–420 (23–123)	1,008.8–11,349 (457.6–5,147.9)	1,040–11,700 (304.8–3,429.3)
Dual fuel, gas or oil	412–3,900 (186.9–1,769)	425–4,020 (125–1,180)	1,008.8–11,349 (457.6–5,147.9)	1,040–11,700 (304.8–3,429.3)
Anthracite coal	97–254 (44–115.2)	100–262 (29.3–76.8)	—	—
Bituminous coal	142.6–292 (64.7–132.5)	147–301 (43–88.2)	—	—

Note: 1 gal = 3.7 L.

*Per day of operation.

boilers are often shut off for 12 or more hours each day. In the absence of unusual factors, a maximum of 10% of known loads may be added.

5. Piping Losses. Allowance should be made to cover unusable heat loss from piping. When this amount is not specifically calculated, 15% of known loads may be added.
6. Maximum instantaneous demand includes sudden short duration peak load requirements, other than pickup allowance. Steam boilers, in particular, should be protected to keep steam flow within the boiler's rated capacity by oversizing or by control orifices, valves or other means. Water boilers must be protected from low temperature operation during peak demand and pickup periods to prevent condensation damage and, in some cases, thermal shock.

4.1.11.3 Operating Pressure

Boilers cannot be operated at their design pressure (MAWP) because the safety valves and controls must be set to operate at or below the design pressure by code and law. Boiler design pressures must therefore be selected in excess of normal system operating pressure.

For 15 lb/in² (1.03 bar) steam boilers, use 12 lb/in² (0.83 bar) maximum system pressure.

For hot water boilers, the total operating pressure, which includes static pressure (due to height of system) thermal expansion pressure and pressure due to pump head working against pipe and system resistance must not exceed the boiler design pressure minus an allowance for operation of safety valves (generally 5 to 8 lb/in² (0.35 to 0.55 bar) for 30 lb/in² (2.06 bar) design boilers and 10 to 15% for boilers with design pressures of 45 lb/in² (3.09 bar) and over). Allowance for pump head is not usually made when circulator pumps are mounted between the boiler and the system resistance.

In steam systems, the boiler generates the pressure to distribute steam through the piping system to the user equipment. The boiler selected must be capable of operating at the pressure required by the system heat user.

If the system is designed for hot water, sufficient pressure must be applied throughout the system to prevent the hot water from flashing to steam. Steam flashing results in noisy operation and mechanical damage. The recommended safety margin in system pressure is to use the saturation pressure for 40°F (22°C) above the maximum expected system water temperature.

System pressure is created by cold fill pressure plus the thermal expansion resistance provided by the expansion tank. The expansion tank uses captive air or another gas to provide a cushion of gas to absorb the increased volume of the water in the system after heating from cold. Expansion tubes are available which contain an elastomeric diaphragm which separates the air cushion from the system water, thus reducing the opportunity for corrosion due to air dissolving in the water. On high temperature hot water systems, nitrogen is generally used as the cushion gas as a means of preventing corrosion in the boiler.

4.1.11.4 Operating Temperatures

Operating temperatures for water heating systems fall into three major categories.

1. Normal cold climate heating systems in which the generally accepted inlet and outlet temperatures are 160°F (71°C) and 180°F (82°C), respectively. In cold

U.S. climates, the boiler generally runs continuously and condensation problems do not occur since boiler inlet water temperature is always above the condensing range. Water circulation rates through hot water generators are given in Table 4.1.3.

2. Warm climate heating systems where afternoon sun requires that the boiler be shut down for 10 to 15 hours per day. In such cases, system operating temperatures are often allowed to fall to the extent that boilers are run in a continuous condensing mode, causing corrosion damage and shortening the life of the boiler. To protect the boiler, provisions must be made to allow it to run at non-condensing temperatures and blend system water to produce the desired system operating temperature.
3. Heat pump loops run at very low temperatures, generally around 65°F (18.3°C) and in this kind of system, special measures must be taken to prevent damage to the boiler due to condensation-based corrosion. For such systems, positive recirculation systems with blending control valves must be used or indirect water heating boilers should be employed.

TABLE 4.1.3 Water Circulation Rates through Hot-Water Generators

		System temperature drop, °F (°C)					
		10 (5.5)	20 (11.1)	30 (16.7)	40 (22.2)	50 (27.7)	60 (33.3)
bhp	Gross boiler output at nozzle, MBtu/h	Circulation rate gal/min					
20	670	134	67	45	33	27	22
30	1,005	200	100	67	50	40	33
40	1,340	268	134	89	67	54	45
50	1,675	335	168	112	84	67	56
60	2,010	400	200	134	100	80	67
70	2,345	470	235	157	118	94	78
80	2,680	536	268	179	134	107	90
100	3,350	670	335	223	168	134	112
125	4,185	836	418	279	209	168	140
150	5,025	1005	503	335	251	201	168
200	6,695	1340	670	447	335	268	224
250	8,370	1675	838	558	419	335	280
300	10,045	2010	1005	670	503	402	335
350	11,720	2350	1175	784	587	470	392
400	13,400	2680	1340	895	670	535	447
500	16,740	3350	1675	1120	838	670	558
600	20,080	4020	2010	1340	1005	805	670
750	25,100	5025	2512	1675	1255	1006	838
800	26,800	5360	2680	1790	1340	1070	900
1000	33,500	6700	3350	2230	1680	1340	1120

Note: Conversion factors: 1 hp = 746 W and 1 gal = 3.8 L.

4.1.11.5 Boiler Capacity

For water heating boilers, the boiler size may be selected from the manufacturer's gross output ratings. For steam boilers, the calculated system load must be adjusted to the "from and at 212°F (100°C)" basis prior to selection from the manufacturer's gross output settings, using the "factor of evaporation" information given in Table 4.1.1. Use the following equation:

$$\begin{aligned} &\text{Required boiler capacity, from and at 212°F, bhp} \\ &= \frac{\text{design load at design temperature, lb/h}}{\text{factor of evaporation from Table 4.1.1 lb/bhp}} \end{aligned} \quad (2)$$

Select factor of evaporation from Table 4.1.1 using design feedwater temperature and design *operating* pressure.

4.1.12 HIGH TEMPERATURE WATER SYSTEMS

High temperature water (HTW) systems are generally operated between 350 and 430°F (177–221°C) with corresponding generator operating pressures between 200 and 525 psi (13.8 and 36.2 bar).

Medium temperature water (MTW) systems generally operate between 250 and 325°F (121 to 163°C) with corresponding operating pressures between 50 and 180 psi.

All boilers operated as hot water generators must be pressurized to prevent the formation of steam. Means must also be provided to allow for expansion of the water as it is heated to operating temperature. The appropriate operating pressure levels required for various operating temperatures are shown in Fig. 4.1.15.

System pressurization, which is crucial to proper system operation, is accomplished through the use of a properly designed expansion tank. The design of this tank depends on system size and operating temperatures and pressures.

4.1.12.1 Circulation in HTW Boilers

Because of the elevated temperatures in HTW boilers, circulation of water in HTW boilers is crucial to boiler longevity. Circulation types are:

- (a) Pumped jet-induced circulation (see Fig. 4.1.16)
- (b) Forced circulation with or without separation drum
- (c) Once through orifice controlled (see Fig. 4.1.17)

1. Jet-Induced Circulation: The D-type two drum watertube HTW generators have proved their dependability by operating trouble-free more than thirty years. They have low first, operating and maintenance costs. Output capacities range to 65 million Btu/hr (19 MW) for pressures to suit water temperatures up to 430°F (221°C) with supply-to-return temperature differences up to 200°F (93°C).

Fig. 4.1.17 shows the water circulation through the HTW unit. The return water flows from the circulation pump through the internal distribution pipe into the plenum chamber and then rises through all furnace tubes to the top drum. Here

SECTION 5

HEAT DISTRIBUTION SYSTEMS

CHAPTER 5.1

STEAM

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5.1.1 INTRODUCTION TO STEAM

Nearly any material, at a given temperature and pressure, has a set amount of energy within it. When materials change their physical state, i.e., go from a liquid to a gas, that energy content changes. Such a change occurs when water is heated to a gaseous state—steam. When steam is used for heating, a cycle of different energy states occurs. First, water is heated in a boiler to its vaporization point, when it boils off as steam. The vapor is carried to the desired estimation where it is allowed to cool, giving off heat. Usually, the water, now cooled back to a liquid, is returned to the boiler to be revaporized.

The heat content of water is usually measured in British thermal units (Btu's) or calories. Knowing the temperature is not sufficient to determine the energy content of steam—the pressure must also be known as well as the amount of actual vapor or condensate (moisture). “Steam” can exist as saturated (containing all the vapor it can), dry (at the saturation point or above), wet (below the saturation point), and superheated (capable of holding even more vapor). Wet steam—containing condensate—has less energy than dry steam.

These conditions are specified for water in a chart called Mollier diagram (see Fig. 5.1.1). The Mollier diagram specifies the energy content for steam at various vaporization levels. On the two axes of the diagram are enthalpy (a measure of the heat content of a volume of steam) and entropy (a measure of the energy available for work). Rigorous analysis of the thermodynamics of a heating system involves measurements of the specific volume of steam available; its pressure, temperature, and moisture values; and the efficiencies of heat transfer of the elements of the heating system. Usually vendors of steam equipment provide details of their systems based on saturated-steam conditions, which simplifies their sizing and use. Saturated-steam tables (see Table 5.1) give the values that are necessary to determine the amount of energy the steam has available for heating.

To calculate the steam consumption of a heating device, the following equation should be employed:

$$Q = \frac{H}{\text{SP}_{\text{wv}}(T_e - T_v) + h_{fg} + \text{SP}_{\text{w}}(T_v - T_c)} \quad (5.1.1)$$

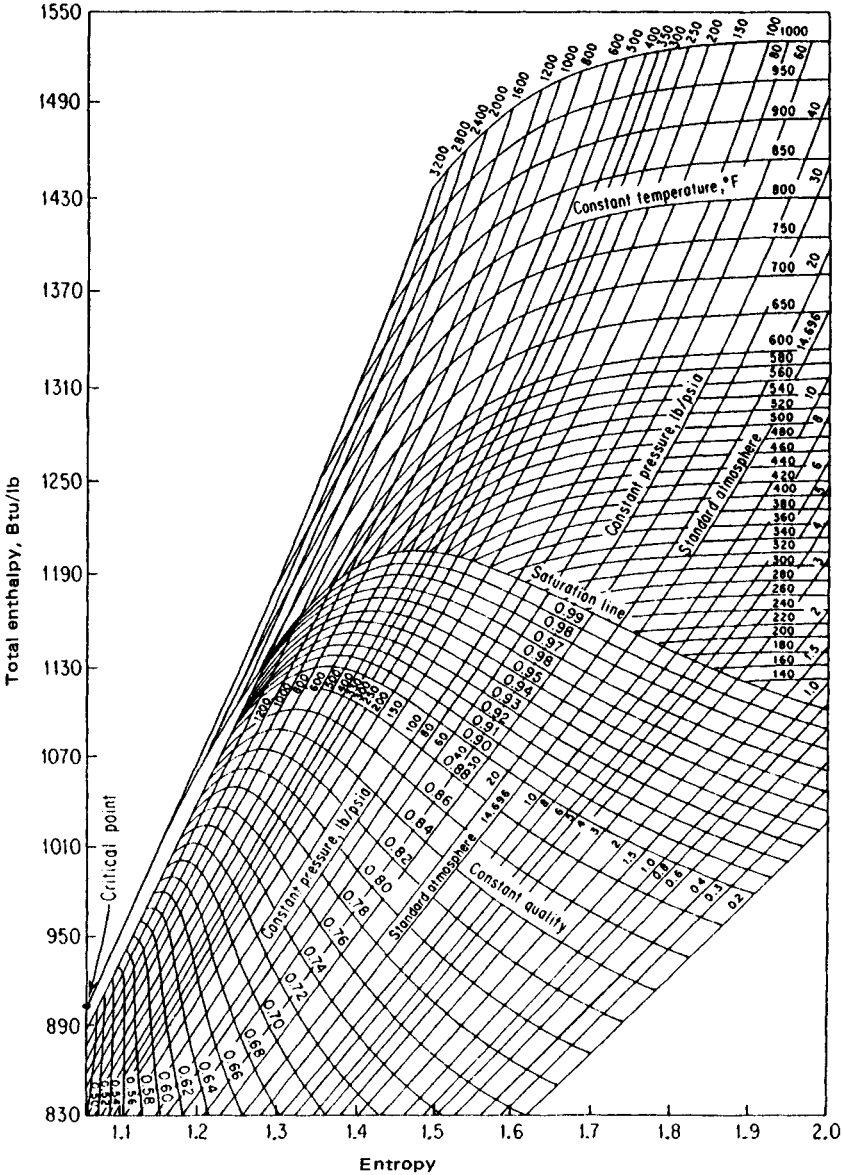


FIGURE 5.1.1 Mollier diagram.

where H = heating load, Btu/h (W)

h_{fg} = latent heat of vaporization, Btu/lb (kJ/kg)

T_e = entering steam temperature, °F (°C)

T_v = steam temperature at vaporization, °F (°C)

SP_{wv} = specific heat of water vapor, Btu/(lb · °F) [cal/(g · °C)]

SP_w = specific heat of water, Btu/(lb · °F) [cal/(g · °C)]

T_c = leaving temperature of condensate, °F (°C)

$\dot{Q}$ = steam rate, lb/h (kg/h)

TABLE 5.1.1 Saturated-Steam Tables

Gauge pressure		Absolute pressure, psig	Temperature, °F	Heat content			Specific volume of steam V_g , ft ³ /lb
in Hg vacuum	psig			Sensible (h_f), Btu/lb	Latent (h_{fg}), Btu/lb	Total (h_g), Btu/lb	
27.9		1	101.7	69.5	1032.9	1102.4	330.0
25.9		2	126.1	93.9	1019.7	1113.6	173.5
23.9		3	141.5	109.3	1011.3	1120.6	118.6
21.8		4	153.0	120.8	1004.9	1125.7	90.5
19.8		5	162.3	130.1	999.7	1129.8	73.4
17.8		6	170.1	137.8	995.4	1133.2	61.9
15.7		7	176.9	144.6	991.5	1136.1	53.6
13.7		8	182.9	150.7	987.9	1138.6	47.3
11.6		9	188.3	156.2	984.7	1140.9	42.3
9.6		10	193.2	161.1	981.9	1143.0	38.4
7.5		11	197.8	165.7	979.2	1144.9	35.1
5.5		12	202.0	169.9	976.7	1146.6	32.4
3.5		13	205.9	173.9	974.3	1148.2	30.0
1.4		14	209.6	177.6	972.2	1149.8	28.0
	0	14.7	212.0	180.2	970.6	1150.8	26.8
	1	15.7	215.4	183.6	968.4	1152.0	25.2
	2	16.7	218.5	186.8	966.4	1153.2	23.80
	5	19.7	227.4	195.5	960.8	1156.3	20.4
	10	24.7	239.4	207.9	952.9	1160.8	16.5
	15	29.7	249.8	218.4	946.0	1164.4	13.9
	20	34.7	258.8	227.5	940.1	1167.6	12.0
	25	39.7	266.8	235.8	934.6	1170.4	10.6
	30	44.7	274.0	243.0	929.7	1172.7	9.5
	40	54.7	286.7	256.1	920.4	1176.5	7.8
	50	64.7	297.7	267.4	912.2	1179.6	6.7
	60	74.7	307.4	277.1	905.3	1182.4	5.8
	70	84.7	316.0	286.2	898.8	1185.0	5.2
	80	94.7	323.9	294.5	892.7	1187.2	4.7
	90	104.7	331.2	302.1	887.0	1189.1	4.3
	100	114.7	337.9	309.0	881.6	1190.6	3.9
	125	139.7	352.8	324.7	869.3	1194.0	3.2
	150	164.7	365.9	338.6	858.0	1196.6	2.8
	175	189.7	377.5	350.9	847.9	1198.8	2.4
	200	214.7	387.7	362.0	838.4	1200.4	2.1

Note: Metric conversion factors are: 1 in Hg = 25.4 mm Hg; 1 lb/in² = 0.07 bar; °F = 1.8 × °C + 32; 1 Btu/lb = 554 cal/kg; 1 ft³/lb = 0.06 m³/kg.

$$Q = \frac{H}{0.45(T_e - T_v) + h_{fg} + T_v - T_c} \quad (5.1.2)$$

or in International System (SI) units,

$$Q = \frac{H}{0.52(T_e - T_v) + 0.28h_{fg} + 1.16(T_v - T_c)} \quad (5.1.3)$$

When saturated steam is supplied to the heating unit, $T_e = T_v$, so $T_e - T_v = 0$. Normally T_c is maintained at or near T_v so that the factor $T_v - T_c$ can be omitted from the calculation without significantly affecting the outcome.

For a system supplying saturated steam we can simplify the calculation to

$$Q = \frac{H}{h_{fg}} \quad \text{or} \quad Q = \frac{H}{0.28h_{fg}} \quad (\text{SI units}) \quad (5.1.4)$$

The following formula converts the steam rate Q into gallons per minute (liters per second) so that the condensate will be in units normally associated with the flow of liquids:

$$\frac{Q}{500} = \text{gal/min} \quad \text{or} \quad \frac{Q}{3600} = \text{L/s (SI units)} \quad (5.1.5)$$

5.1.2 INTRODUCTION TO STEAM HEATING SYSTEMS

Steam systems are used to heat industrial, commercial, and residential buildings. These systems are categorized according to the piping layout and the operating steam pressure. This section discusses steam systems which operate at or below 200 psig (14 bar).

5.1.3 GENERAL SYSTEM DESIGN

The mass flow rate of steam through the piping system is a function of the initial steam pressure, pressure drop through the pipe, equivalent length of piping, and size of piping. The roughness of the inner pipe wall is a variable in determining the steam's pressure drop. All the charts and tables in this section that outline the performance of the steam transmitted through the piping assume that the roughness of the piping is equal to that of new, commercial-grade steel pipe.

5.1.4 PRESSURE CONDITIONS

Steam piping systems are usually categorized by the working pressure of the steam they supply. The five classes of steam systems are high-pressure, medium-pressure, low-pressure, vapor, and vacuum systems. A high-pressure system has an initial pressure in excess of 100 psig (6.9 bar). The medium-pressure system operates with

pressures between 100 psig (6.9 bar) and 15 psig (1 bar). Systems that operate from 15 psig (1 bar) to 0 psig (0 bar) are classified as low-pressure. Vapor and vacuum systems operate from 15 psig (1 bar) to vacuum. Vapor systems attain subatmospheric pressures through the condensing process, while vacuum systems require a mechanically operated vacuum pump to attain subatmospheric pressures.

5.1.5. PIPING ARRANGEMENTS

The general piping scheme of a steam system can be distinguished by three different characteristics. First, the number of connections required at the heating device describes the system. A one-pipe system has only one piping connection which supplies steam and allows condensate to return to the boiler by flowing counter to the steam in the same pipe. The more common design is to have two piping connections, one for the supply steam and one for the condensate. This arrangement is known as a two-pipe system.

Second, the direction of the supply steam in the risers characterizes the piping design. An up-feed system has the steam flowing up the riser; conversely, a down-feed system supplies steam down the riser.

Third, the final characteristic of the piping design is the location of the condensate return to the boiler. A dry return has its condensate connection above the boiler's waterline, while a wet-return connection is below the waterline.

5.1.6 CONDENSATE RETURN

By analyzing how the condensate formed in the heating system is returned to the boiler, an understanding of how the system should operate is achieved. There are two commonly used return categories: mechanical and gravity.

If devices such as pumps are used to aid in the return of condensate, the system is known as a *mechanical return*. When no mechanical device is used to return the condensate, the system is classified as a *gravity return*. The only forces pushing the condensate back to the boiler or condensate receiver are gravity and the pressure of the steam itself. This type of system usually requires that all steam-consuming components be located at a higher elevation than the boiler or the condensate receiver.

With either mechanical or gravity return systems, the mains are normally pitched $\frac{1}{4}$ in (6.3 mm) for every 10 ft (3 m) of length, to ensure the proper flow of condensate. The supply mains are sloped up away from the boiler, and the return mains are pitched down toward the boiler. This allows condensate to flow back to the boiler.

5.1.7 PIPE-SIZING CRITERIA

Once the heating loads are known, the steam flow rates can be determined; then the required size of the steam piping can be specified for proper operation. The following factors must be analyzed in sizing the steam piping:

- Initial steam pressure
- Total allowable pressure drop
- Maximum steam velocity
- Direction of condensate flow
- Equivalent length of system

For different initial pressures, the allowable pressure drop in the piping varies. Table 5.2 gives typical values in selecting pressure-drop limits. To ensure that the parameters from the table are suitable for an application, check that the total system pressure drop does not exceed 50 percent of the initial pressure, that the condensate has enough steam pressure to return to the boiler, and that the steam velocity is within specified limits to ensure quiet and long-lasting operation.

When steam piping is sized, there is a trade-off between quiet, efficient operation and first-cost considerations. A good compromise point exists when the steam supply pipe is sized for velocities between 6000 and 12,000 ft/min (30.5 and 61 m/s). This allows quiet operation while offering a reasonable installed cost. If the piping is downsized so that the velocity exceeds 20,000 ft/min (101 m/s), the system may produce objectional hammering noise or restrict the flow of condensate when it is counter to the steam's direction. It is recommended that the piping be sized so that the velocity will never approach 20,000 ft/min (101 m/s) in any leg.

As condensate flows into the return line, a portion of it will flash into steam. The volume of the steam-condensate mixture is much greater than the volume of pure condensate. To avoid undersizing the return lines, the return piping should be sized at some reasonable proportion of dry steam. A maximum size would be to assume that the return is 100 percent saturated steam. An acceptable velocity for the design of the return lines is 5000 ft/min (25.4 m/s).

5.1.8 DETERMINING EQUIVALENT LENGTH

The “equivalent length” of pipe is equal to the actual length of pipe plus the friction losses associated with fittings and valves. For simplicity's sake, the fitting and valve losses are stated as the equivalent length of straight pipe needed to produce the same friction loss. Values for common fittings and valves are stated in Table 5.1.3.

The equivalent length—*not the actual length*—is the value used in all the figures and charts for pipe sizing. Common practice is to assume that the equivalent length is 1.5 times the actual length when a design is first being sized. After the initial sizing and layout are completed, the exact equivalent length should be calculated and all the pipe sizes checked.

5.1.9 BASIC TABLES FOR STEAM PIPE SIZING

Figure 5.1.2 is used to determine the flow and velocity of steam in Schedule 40 pipe at various values of pressure drop per 100 ft (30.5 m), based on 0 psig (1-bar) saturated steam. By using the multiplier tables, it may also be used at all saturated pressures between 0 and 200 psig (1 and 14 bar). Figure 5.1.2 is valid only when steam and condensate flow in the same direction.

TABLE 5.1.2 Pressure Drops for Steam Pipe Sizing

Initial steam pressure		Total pressure drop in supply piping		Pressure drop for mains and risers		Total pressure drop in return piping	
psig	bar	lb/in ²	bar	(lb/in ²)/ 100 ft	bar/100 m	lb/in ²	bar
Vacuum		1–2	0.069–0.138	1/8–1/4	0.028–0.057	1	0.069
0	0	1/6–1/4	0.004–0.017	1/32	0.007	1/16	0.004
2	0.138	1/4–3/4	0.017–0.052	1/8	0.028	1/4	0.017
5	0.345	1–2	0.069–1.38	1/4	0.057	1	0.069
15	1.03	4–6	0.276–0.414	1	0.228	4	0.276
30	2.07	5–10	0.345–0.069	2	0.455	5	0.345
50	3.45	10–15	0.069–1.03	2–5	0.455–1.14	10	0.69
100	6.90	15–25	1.03–1.72	2–5	0.455–1.14	15	1.03
150	10.3	25–30	1.72–2.07	2–10	0.455–2.28	20	1.37

TABLE 5.1.3 Length of Pipe to Be Added to Actual Length of Run—Owing to Fittings—to Obtain Equivalent Length

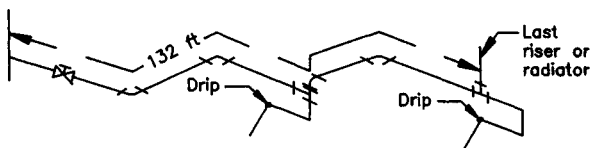
Size of pipe, in	Length to be added to run, ft*				
	Standard elbow	Side outlet tee†	Gate valve‡	Globe valve‡	Angle valve‡
½	1.3	3	0.3	14	7
¾	1.8	4	0.4	18	10
1	2.2	5	0.5	23	12
1¼	3.0	6	0.6	29	15
1½	3.5	7	0.8	34	18
2	4.3	8	1.0	46	22
2½	5.0	11	1.1	54	27
3	6.5	13	1.4	66	34
3½	8	15	1.6	80	40
4	9	18	1.9	92	45
5	11	22	2.2	112	56
6	13	27	2.8	136	67
8	17	35	3.7	180	92
10	21	45	4.6	230	112
12	27	53	5.5	270	132
14	30	63	6.4	310	152

*Metric conversion: 1 in = 2.54 cm and 1 ft = 0.31 m.

†Values given apply only to a tee used to divert the flow in the main to the last riser.

‡Valve in full-open position.

Example: Determine the length in feet of pipe to be added to actual length of run illustrated.



Measured length	132.0 ft
4-in gate valve	1.9 ft
4-4 in elbows	36.0 ft
2-4 in tees	36.0 ft
Equivalent	205.9 ft

Source: Reprinted by permission from ASHRAE Handbook—1989 Fundamentals.

5.1.10 TABLES FOR LOW-PRESSURE STEAM PIPE SIZING

Table 5.1.4, derived from Fig. 5.1.2, gives the values needed to select pipe sizes at various pressure drops for systems operating at 3.5 and 12 psig (0.24 and 0.84 bar). The flow rates given for 3.5 psig (0.24 bar) can be used for saturated-steam pressures from 1 to 5 psig (0.07 to 0.34 bar), and those for 12 psig (0.84 bar) can be used for saturated pressures from 8 to 16 psig (0.55 to 1.1 bar) with an error not to exceed 8 percent.

Table 5.1.5 is used for systems where the condensate flows counter to the supply steam.

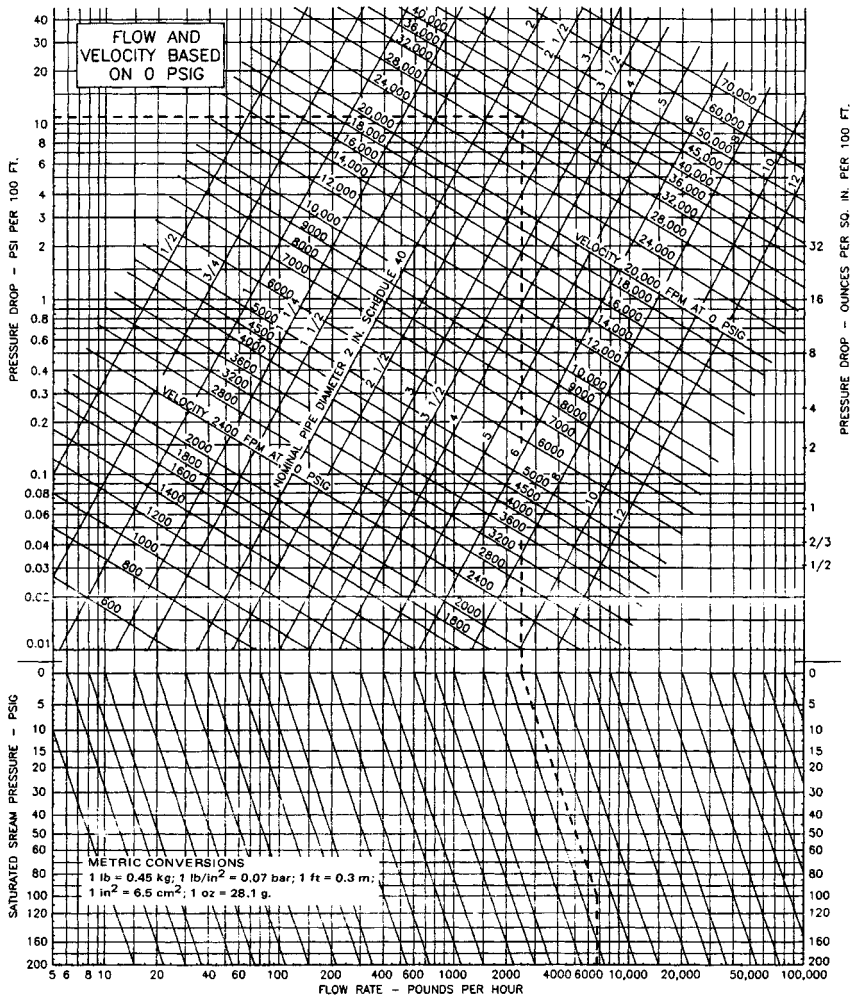


FIGURE 5.1.2 Basic chart for flow rate and velocity of steam in Schedule 40 pipe, based on saturation pressure of 0 psig (0 bar). (Reprinted by permission from ASHRAE Handbook—1989 Fundamentals.)

To size return piping, Table 5.1.6 is used. This table gives guidelines for return piping for wet, dry, and vacuum return systems.

5.1.11 TABLES FOR SIZING MEDIUM- AND HIGH-PRESSURE PIPE SYSTEMS

Larger, industrial-type space-heating systems are designed to use either medium- or high-pressure steam at 15 to 200 psig (1.03 to 14 bar). These systems often

TABLE 5.1.4 Flow Rate of Steam, lb/h, in Schedule 40 Pipe* at Initial Saturation Pressure of 3.5 and 12 psig†

Nom. pipe size, in	Pressure drop, lb/in ² per 100-ft length‡													
	1/16 lb/in ² (1 oz) Sat. press., psig		1/8 lb/in ² (2 oz) Sat. press., psig		1/4 lb/in ² (4 oz) Sat. press., psig		1/2 lb/in ² (8 oz) Sat. press., psig		3/4 lb/in ² (12 oz) Sat. press., psig		1 lb/in ² Sat. press., psig		2 lb/in ² Sat. press., psig	
	3.5	12	3.5	12	3.5	12	3.5	12	3.5	12	3.5	12	3.5	12
1/4	9	11	14	16	20	24	29	35	36	43	42	50	60	73
1	17	21	26	31	37	46	54	66	68	82	81	95	114	137
1 1/4	36	45	53	66	78	96	111	138	140	170	162	200	232	280
1 1/2	56	70	84	100	120	147	174	210	218	260	246	304	360	430
2	108	134	162	194	234	285	336	410	420	510	480	590	710	850
2 1/2	174	215	258	310	378	460	540	660	680	820	780	950	1,150	1,370
3	318	380	465	550	660	810	960	1,160	1,190	1,430	1,380	1,670	1,950	2,400
3 1/2	462	550	670	800	990	1,218	1,410	1,700	1,740	2,100	2,000	2,420	2,950	3,450
4	640	800	950	1,160	1,410	1,690	1,980	2,400	2,450	3,000	2,880	3,460	4,200	4,900
5	1,200	1,430	1,680	2,100	2,440	3,000	3,570	4,250	4,380	5,250	5,100	6,100	7,500	8,600
6	1,920	2,300	2,820	3,350	3,960	4,850	5,700	7,000	7,200	8,600	8,400	10,000	11,900	14,200
8	3,900	4,800	5,570	7,000	8,100	10,000	11,400	14,300	14,500	17,700	16,500	20,500	24,000	29,500
10	7,200	8,800	10,200	12,600	15,000	18,200	21,000	26,000	26,200	32,000	30,000	37,000	42,700	52,000
12	11,400	13,700	16,500	19,500	23,400	28,400	33,000	40,000	41,000	49,500	48,000	57,500	67,800	81,000

*R.

*Based on Moody friction factor, where flow of condensate does not inhibit the flow of steam.

†The flow rates of 3.5 psig can be used to cover saturated pressure from 1 to 6 psig, and the rates at 12 psig can be used to cover saturated pressure from 8 to 16 psig with an error not exceeding 8 percent. The steam velocities corresponding to the flow rates given in this table can be found from the basic chart and velocity multiplier chart, Fig. 5.2.

‡Metric conversions: 1 in = 2.54 cm, 1 lb/in² = 0.07 bar, and 1 lb = 0.46 kg.

Source: Reprinted by permission from ASHRAE Handbook—1989 Fundamentals.

TABLE 5.1.5 Steam Pipe Capacities for Low-Pressure Systems, lb/h

For use on one-pipe systems or two-pipe systems in which condensate flows against the steam flow

Nominal pipe size, in	Two-pipe systems		One-pipe systems		
	Condensate flowing against steam		Supply risers up-feed	Radiator valves and vertical connections	Radiator and riser runouts
	Vertical	Horizontal			
A	B*	C†	D‡	E	F†
¾	8	7	6	...	7
1	14	14	11	7	7
1¼	31	27	20	16	16
1½	48	42	38	23	16
2	97	93	72	42	23
2½	159	132	116	...	42
3	282	200	200	...	65
3½	387	288	286	...	119
4	511	425	380	...	186
5	1,050	788	...	...	278
6	1,800	1,400	...	...	545
8	3,750	3,000			
10	7,000	5,700			
12	11,500	9,500			
16	22,000	19,000			

*Do not use column B for pressure drops of less than 1/16 lb/in² per 100 ft of equivalent run. Use Fig. 5.2 or Table 5.4 instead.

†Pitch of horizontal runouts to risers and radiators should be not less than 1/2 in/ft. Where this pitch cannot be obtained, runouts over 8 ft in length should be one pipe size larger than called for in this table.

‡Do not use column D for pressure drops of less than 1/24 lb/in² per 100 ft of equivalent run except on sizes 3 in and over. Use Fig. 5.2 or Table 5.4 instead.

Note: Steam at an average pressure of 1 psig is used as a basis of calculating capacities. Metric conversion factors of 1 in = 2.54 cm and 1 lb = 0.46 kg can be used.

Source: Reprinted from *ASHRAE Handbook—1989 Fundamentals*.

TABLE 5.1.6 Return Main and Riser Capacities for Low-Pressure Systems, lb/h

Pipe size, in	$\frac{1}{2}$ lb/in ² or $\frac{1}{2}$ -oz drop per 100 ft			$\frac{1}{4}$ lb/in ² or $\frac{3}{4}$ -oz drop per 100 ft			$\frac{1}{16}$ lb/in ² or 1-oz drop per 100 ft			$\frac{1}{8}$ lb/in ² or 2-oz drop per 100 ft			$\frac{1}{4}$ lb/in ² or 4-oz drop per 100 ft			$\frac{1}{2}$ lb/in ² or 8-oz drop per 100 ft		
	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.
G	H	I	J	K	L	M	N	O	P	Q	R	S	T	U	V	W	X	Y
Return Main																		
$\frac{3}{4}$	...	...	...	...	...	42	...	...	100	...	...	142	...	...	200	...	...	283
1	125	62	...	145	71	143	175	80	175	250	103	249	350	115	350	...	...	494
1 $\frac{1}{4}$	213	130	...	248	149	244	300	168	300	425	217	426	600	241	600	...	...	848
1 $\frac{1}{2}$	338	206	...	393	236	388	475	265	475	675	340	674	950	378	950	...	...	1,340
2	700	470	...	810	535	815	1000	575	1,000	1400	740	1,420	2,000	825	2,000	...	...	2,830
2 $\frac{1}{2}$	1180	760	...	1580	868	1,360	1680	950	1,680	2350	1230	2,380	3,350	1360	3,350	...	...	4,730
3	1880	1460	...	2130	1560	2,180	2680	1750	2,680	3750	2250	3,800	5,350	2500	5,350	...	...	7,560
3 $\frac{1}{2}$	2750	1970	...	3300	2200	3,250	4000	2500	4,000	5500	3230	5,680	8,000	3580	8,000	...	...	11,300
4	3880	2930	...	4580	3350	4,500	5500	3750	5,500	7750	4830	7,810	11,000	5380	11,000	...	...	15,500
5	...	...	...	...	...	7,880	...	...	9,680	...	...	13,700	...	...	19,400	...	...	27,300
6	...	...	...	...	...	12,600	...	...	15,500	...	...	22,000	...	...	31,000	...	...	43,800
Riser																		
$\frac{3}{4}$	...	48	...	...	48	143	...	48	175	...	48	249	...	48	350	...	...	494
1	...	113	...	...	113	244	...	113	300	...	113	426	...	113	600	...	...	848
1 $\frac{1}{4}$	...	248	...	...	248	388	...	248	475	...	248	674	...	248	950	...	...	1,340
1 $\frac{1}{2}$	...	375	...	...	375	815	...	375	1,000	...	375	1,420	...	375	2,000	...	...	2,830
2	...	750	...	...	750	1,360	...	750	1,680	...	750	2,380	...	750	3,350	...	...	4,730
2 $\frac{1}{2}$	...	...	...	...	...	2,180	...	...	2,680	...	...	3,800	...	...	5,350	...	...	7,560
3	...	...	...	...	...	3,250	...	...	4,000	...	...	5,680	...	...	8,000	...	...	11,300
3 $\frac{1}{2}$	...	...	...	...	...	4,480	...	...	5,500	...	...	7,810	...	...	11,000	...	...	15,500
4	...	...	...	...	...	7,880	...	...	9,680	...	...	13,700	...	...	19,400	...	...	27,300
5	...	...	...	...	...	12,600	...	...	15,500	...	...	22,000	...	...	31,000	...	...	43,800

Note: This table is based on pipe size data developed through the research investigations of The American Society of Heating, Refrigerating and Air-Conditioning Engineers. Metric conversion factors of 1 in = 2.54 cm, 1 lb/in² = 0.07 bar, and 1 ft = 0.31 m can be used.

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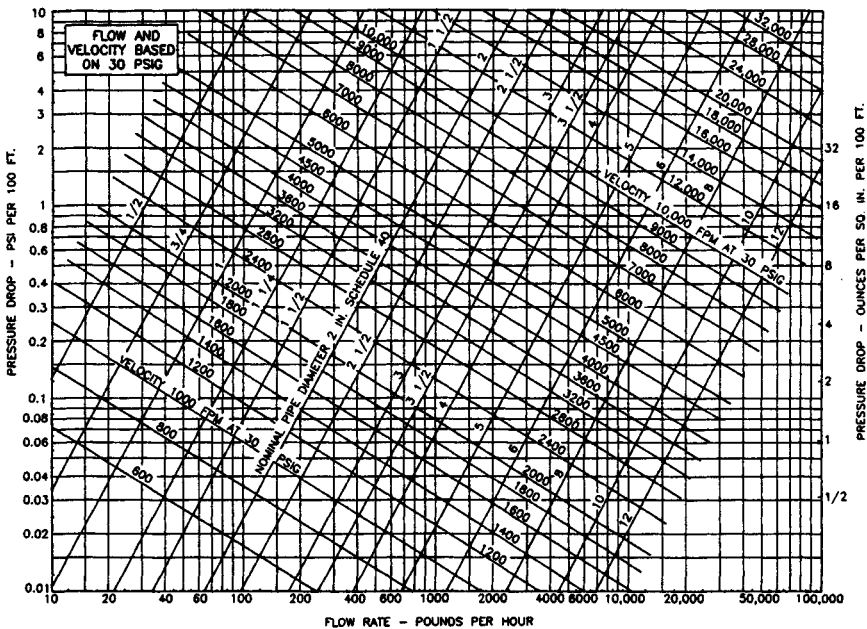
involve unit heaters and/or air-handling units. Figures 5.1.3 to 5.1.6 provide tables for sizing steam piping for systems of 30, 50, 100, and 150 psig (2, 3.5, 6.9, and 10.5 bar).

5.1.12 AIR VENTS

The presence of air in the steam supply line impedes the heat-transfer ability of the system due to the high insulating value of air. Air also interferes with the flow of steam by forming pockets at the ends of runs that prevent the steam from reaching the system's extremities.

A valve that releases air from the system while restricting the flow of all other fluids is known as an "air vent." Air vents should be located at all system high points and where air pockets are likely to form. Venting should be done continually to prevent the buildup of air in the system.

Air enters the system by two means. First, when cold makeup feed water is supplied to the boiler, air is present in the water. As the water is heated, the air



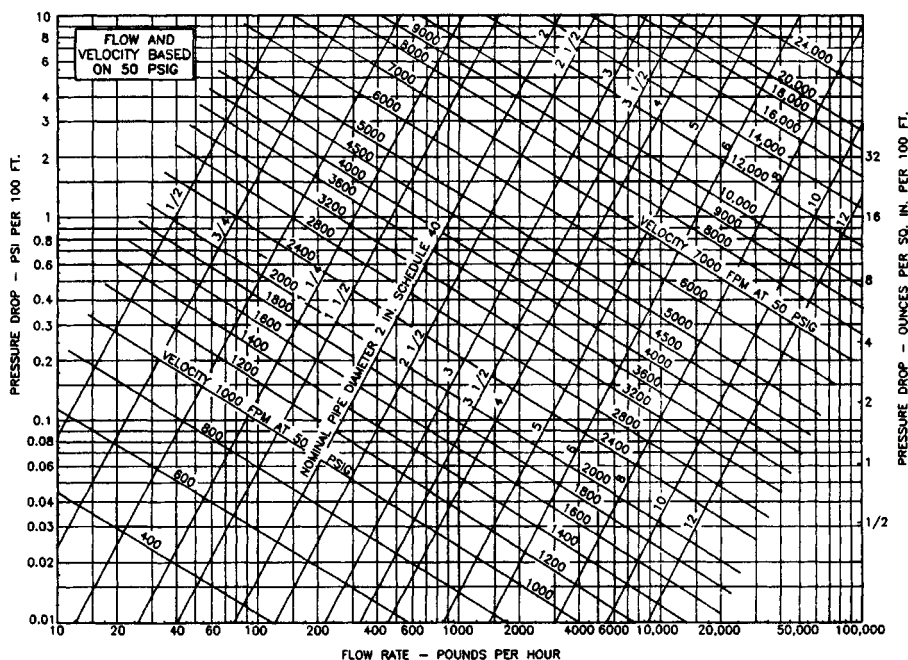
BASED ON MOODY FRICTION FACTOR WHERE FLOW OF CONDENSATE DOES NOT INHIBIT THE FLOW OF STEAM
(MAY BE USED FOR STEAM PRESSURE FROM 23 TO 37 PSIG WITH AN ERROR NOT EXCEEDING 9%)

METRIC CONVERSIONS:

1 lb = 0.45 kg; 1 lb/in² = 0.07 bar; 1 ft = 0.3 m;

1 in² = 6.5 cm²; 1 oz = 28.1 g.

FIGURE 5.1.3 Chart for flow rate and velocity of steam in Schedule 40 pipe, based on saturation pressure of 30 psig (2.1 bar). (Reprinted by permission from ASHRAE Handbook—1989 Fundamentals.)



BASED ON MOODY FRICTION FACTOR WHERE FLOW OF CONDENSATE DOES NOT INHIBIT THE FLOW OF STEAM
(MAY BE USED FOR STEAM PRESSURE FROM 40 TO 60 PSIG WITH AN ERROR NOT EXCEEDING 8%)

METRIC CONVERSIONS:

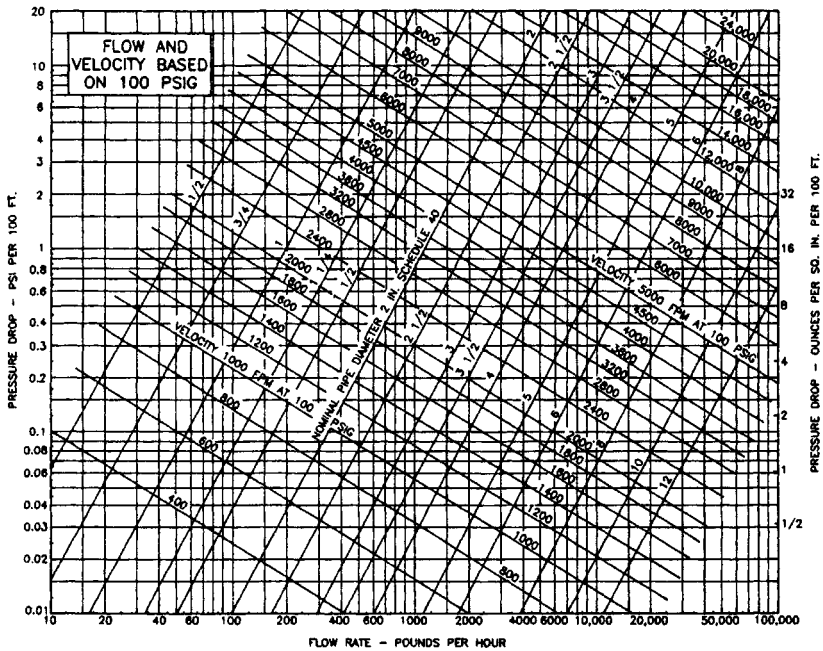
1 lb = 0.45 kg; 1 lb/in² = 0.07 bar; 1 ft = 0.3 m;
1 in² = 6.5 cm²; 1 oz = 28.1 g.

FIGURE 5.1.4 Chart for flow rate and velocity of steam in Schedule 40 pipe, based on saturation pressure of 50 psig (3.5 bar). (Reprinted by permission from *ASHRAE Handbook—1989 Fundamentals*.)

tends to separate from the water. Second, when the system is turned off, steam is trapped in the pipes. Eventually the steam cools and condenses. Since the volume of the condensate is negligible compared to the initial volume of the steam, a vacuum is formed in the piping. Air leaks into the system through openings in the joints until the internal pressure equalizes. Upon restarting the system, the air is swept along with the steam and becomes entrained in the system.

5.1.13 STEAM TRAPS

When steam is transmitted through the piping or the end-user equipment, it loses part of its heat energy. As heat is removed from saturated steam, a vapor-liquid mixture forms in the pipe. The presence of liquid condensate in the steam lines interferes with the proper operation of the system. Liquid condensate derates the system's heating capacity because water has a much smaller amount of available



BASED ON MOODY FRICTION FACTOR WHERE FLOW OF CONDENSATE DOES NOT INHIBIT THE FLOW OF STEAM
(MAY BE USED FOR STEAM PRESSURE FROM 85 TO 120 PSIG WITH AN ERROR NOT EXCEEDING 8%)

METRIC CONVERSIONS:

1 lb = 0.45 kg; 1 lb/in² = 0.07 bar; 1 ft = 0.3 m;
1 in² = 6.5 cm²; 1 oz = 28.1 g.

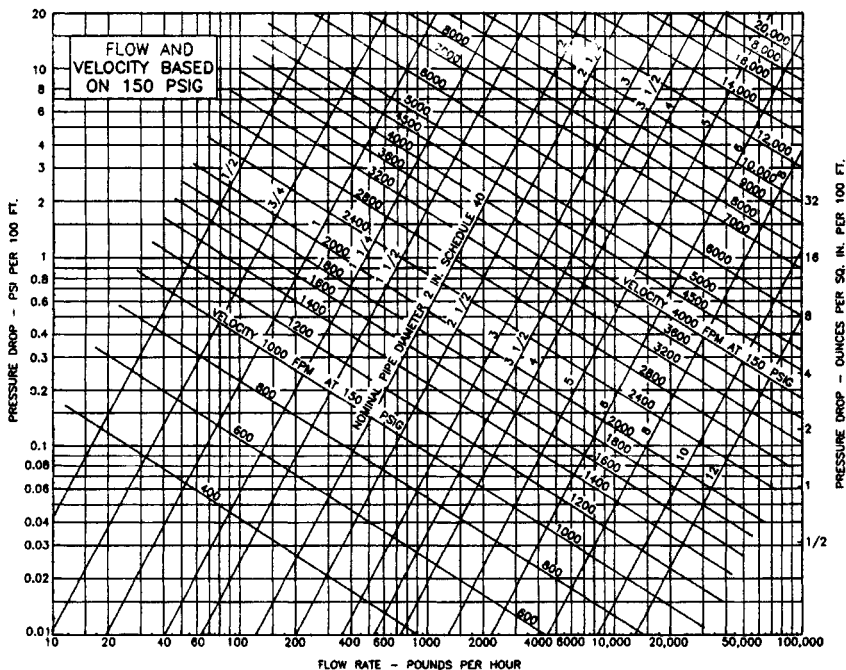
FIGURE 5.1.5 Chart for flow rate and velocity of steam in Schedule 40 pipe, based on saturation pressure of 100 psig (7 bar). (Reprinted by permission from ASHRAE Handbook—1989 Fundamentals.)

energy than steam does. Furthermore, the accumulation of water in the supply steam piping can obstruct the flow of the steam through the system.

A valve that permits condensate to flow from the supply line without allowing steam to escape is known as a “steam trap.” All steam traps should be located such that condensate can flow via gravity through them. *Through mechanical means*, the steam trap recognizes when steam is present by sensing the density, kinetic energy, or temperature of the fluid at the trap. When conditions indicate that steam is absent, the trap opens and allows the condensate to drop to the return line. As soon as the trap senses the presence of steam, it slams shut.

5.1.14 STEAM TRAP TYPES

There are six types of steam traps normally employed in the heating, ventilating, and air-conditioning (HVAC) industry. Since traps differ in their operational char-



BASED ON MOODY FRICTION FACTOR WHERE FLOW OF CONDENSATE DOES NOT INHIBIT THE FLOW OF STEAM
(MAY BE USED FOR STEAM PRESSURE FROM 127 TO 180 PSIG WITH AN ERROR NOT EXCEEDING 8%)

METRIC CONVERSIONS:

1 lb = 0.45 kg; 1 lb/in² = 0.07 bar; 1 ft = 0.3 m;

1 in² = 6.5 cm²; 1 oz = 28.1 g.

FIGURE 5.1.6 Chart for flow rate and velocity of steam in Schedule 40 pipe, based on saturation pressure of 150 psig (10.5 bar). (Reprinted by permission from ASHRAE Handbook—1989 Fundamentals.)

acteristics, selection of the proper trap is critical to efficient operation of the system. Different applications require specific types of traps, and no one type of trap will perform satisfactorily in all situations.

Three of the six basic types of traps operate thermostatically by sensing a temperature difference between subcooled condensate and steam: liquid-expansion, balanced-pressure thermostatic, and bimetallic thermostatic traps. Two other types—the bucket trap and the float-and-thermostatic trap—are activated by differences in density between steam and condensate. These are also known as blast type traps. Finally, the thermodynamic steam trap operates on the differences in the velocity at which steam passes through the trap. This velocity difference can also be considered as a change in kinetic energy.

5.1.15 BALANCED-PRESSURE STEAM TRAPS

The balanced-pressure steam trap (Fig. 5.1.7) employs a bellows filled with a fluid mixture that boils below the steam temperature. When steam is present at the trap

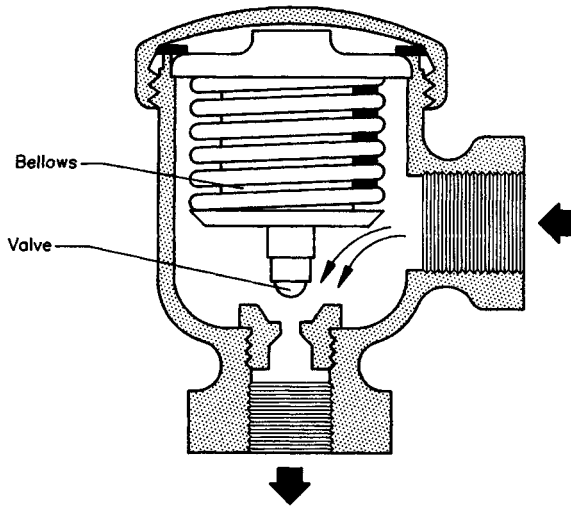


FIGURE 5.1.7 Balanced-pressure steam trap.

inlet, the liquid in the bellows is vaporized and expands to seal the trap. Condensate accumulates at the trap and starts to subcool. When the condensate cools enough to condense the fluid in the bellows, the trap opens and the condensate flows through the trap.

This type of trap has two possible drawbacks. First, it must allow condensate to subcool 5 to 30°F (2.8 to 16.7°C) below the steam temperature to operate. Second, it discharges condensate intermittently.

Advantages of the balanced-pressure trap are that it is freeze-proof, can handle a large condensate load, does a good job of air venting, and is self-adjusting throughout its operating range. These traps are typically used in conjunction with steam radiators and sterilizers.

5.1.16 BIMETALLIC THERMOSTATIC STEAM TRAPS

These traps operate on the same principle as the balanced-pressure steam trap. The bellows mechanism is replaced by a bimetallic strip formed from two dissimilar metals that have very different coefficients of expansion. As the bimetallic strip is heated, the difference in the expansion rate of the metals causes the strip to bend. The trap is fabricated so that when the strip is heated to the steam's temperature, there is enough movement to close off the valve. The bimetallic thermostatic trap (Fig. 5.1.8) has a slow response to load conditions, requiring as much as 100°F (55.5°C) of subcooling, and is not self-adjusting to changes in inlet pressure.

These traps are suited for superheated steam applications and situations where a great deal of condensate subcooling is required to prevent flashing in the return line. Normally these traps are applied to steam-tracing lines that can tolerate partial flooding.

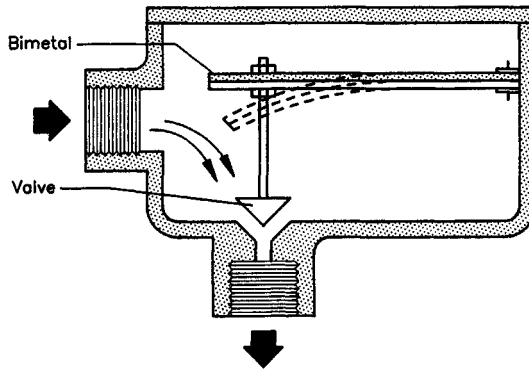


FIGURE 5.1.8 Bimetallic steam trap.

5.1.17 LIQUID-EXPANSION STEAM TRAPS

The liquid-expansion steam trap (Fig. 5.1.9) is designed with an oil-filled cylinder which drives a piston. When steam is present, the oil expands, thrusting the piston out. The end of the piston acts as the valve and seals the port to the return line. As condensate collects in the trap and cools, the oil starts to contract. The contraction of the oil causes the piston to move away from the port and permits the flow of condensate from the trap.

These traps are freeze-proof and are used for freeze protection of system low points and heating coils. Their limitations are that they are not self-adjusting to changes of inlet pressure and that they require condensate subcooling by 2 to 30°F (1.1 to 16.7°C).

5.1.18 BUCKET STEAM TRAPS

Bucket traps operate by gravity, utilizing the density difference between liquid and vapor. When the body of the trap is filled with liquid and a vapor enters the bucket, the bucket will float. As the bucket fills with liquid, the bucket sinks. The bucket's

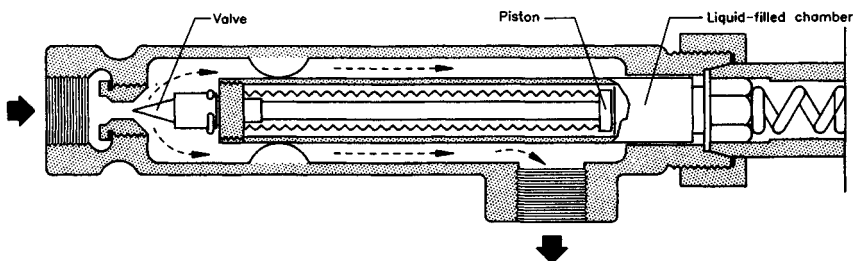


FIGURE 5.1.9 Liquid-expansion steam trap.

movement activates a valve. If the bucket rises due to the vapor pressure, the valve closes; and when the bucket sinks, the valve opens, permitting condensate to flow from the trap. The most common type of bucket trap is the inverted bucket (Fig. 5.1.10), so named because the bucket has its open side facing down.

Bucket traps are capable of working at very high pressures, can discharge condensate at the saturated-steam temperature, and are resistant to water hammer. Unfortunately, if the water seal is lost, the bucket trap will continuously allow steam to pass through. Other disadvantages of these traps are their susceptibility to freeze-up, their lack of good air-venting capability, and their intermittent discharge.

Inverted-bucket traps are usually installed on high-pressure indoor steam main drips.

5.1.19 FLOAT-AND-THERMOSTATIC STEAM TRAPS

A float-and-thermostatic steam trap (Fig. 5.1.11) is actually two distinct traps in one unit. The balanced-pressure steam trap, outlined previously, is located at the top of the trap body and acts as an air vent. The rest of the unit consists of a float that rises and falls based on the level of condensate in the trap. The trap inlet is located above the outlet. The float position operates a valve that controls flow to the return line. As the condensate level rises above the outlet, the float causes the valve to open. If the condensate level drops enough, the float causes the valve to close. Since the float allows the valve to open only when the condensate level is

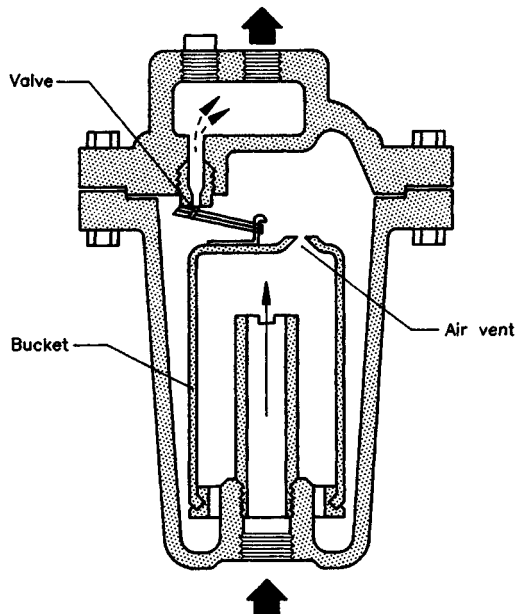


FIGURE 5.1.10 Inverted-bucket steam trap.

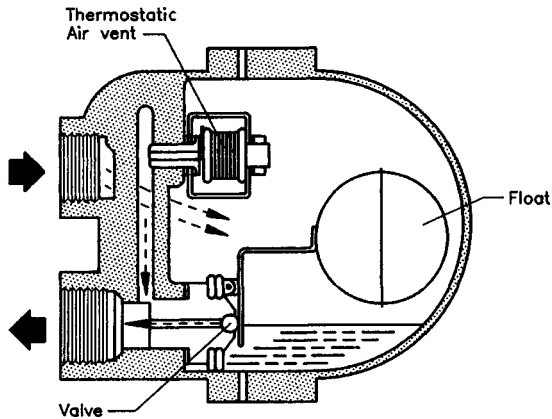


FIGURE 5.1.11 Float-and-thermostat steam trap.

above the outlet, a water seal is maintained to prevent steam from passing through the outlet when the valve is open.

The float-and-thermostatic steam traps cannot be used on a superheated-steam system unless they are modified and are usually not installed outdoors because they are subject to freeze-up. These types of traps will continuously vent air. They do not require subcooling of condensate and are unaffected by changes in system pressure. Typically float-and-thermostatic traps are used in conjunction with heating devices, such as unit heaters, water heaters, and converters.

5.1.20 THERMODYNAMIC STEAM TRAPS

The design of the thermodynamic steam trap (Fig. 5.1.12) is based on the theory that the total pressure of fluid passing through the trap will remain constant. Since the total pressure equals the sum of the static and dynamic pressures, any increase

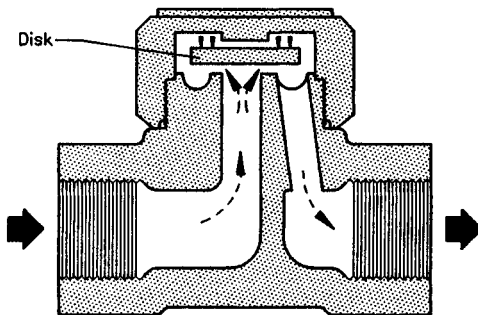


FIGURE 5.1.12 Thermodynamic steam trap.

in dynamic pressure will cause a decrease in the static pressure, and vice versa. These traps have only one moving part, a disk that can seal off both the inlet and the outlet of the trap. Steam entering the trap accelerates radially over the disk, causing a reduction in static pressure under the disk. As the steam dead-ends above the disk, the static pressure above the disk increases. This difference in pressure induces the disk to seal off the trap's openings. The trap will remain closed until the steam in the trap condenses sufficiently to reduce the pressure above the disk to an amount less than the inlet steam pressure. At that point, the disk moves away from the inlet port.

Thermodynamic steam traps should not be used on systems operating below 5 psig (0.34 bar) or on those that have back pressures equal to or greater than 80 percent of their supply pressure.

These traps are compact and have a long life due to the simplicity of their design. They can operate under high pressures, responding quickly to load and pressure variations while discharging condensate without requiring subcooling. Thermodynamic traps are usually installed in main drips and steam tracer lines.

5.1.21 STEAM TRAP LOCATION

Steam traps are located either in the return line or in drip legs. A "drip leg" (shown in Fig. 5.1.13) is a piping assembly that hangs below the supply main; its purpose is to remove condensate and sediment from the main. Gravity allows condensate and sediment to leave the main and accumulate in the drip leg. When the condensate in the leg rises to the level of the trap intake, the trap fills and then discharges the condensate to the return line. The drip leg pipe should be of sufficient size to permit condensate to drain freely from the main. For mains of 4 in (102 mm) or less in diameter, the drip leg should be the same size as the main pipe. For mains larger than 4 in (102 mm), the pipe diameter of the drip leg should be half of the main's size, but not less than 4 in (102 mm). Where possible, all drip legs should be at least 18 in (45.7 cm) long. A trap should be installed in the return line after every steam-consuming device. Each device should have its own trap to prevent possible "short-circuiting" that could occur if multiple devices share a common trap. A drip leg should be located before risers, expansion joints, bends, valves, and regulators. System low points, end of mains, and untrapped supply runs of over 300 ft (100 m) are additional locations where drip legs should be installed.

5.1.22 STEAM TRAP SIZING

A steam trap must be properly sized to handle the full load of condensate. For heating devices, the method of determining the amount of condensate was discussed in Sec. 5.1. (See also Sec. 5.24. See also "determining condensate load for a system," next page.) Mains have their largest condensate loads during startup. Table 5.7 gives values for the condensate load of mains at startup.

The performance of a steam trap is affected by the inlet pressure and back pressure of the system. Therefore, when a trap is chosen, it is prudent to oversize the trap by a reasonable amount. Table 5.1.8 gives a guideline on how large to size traps. Grossly oversizing a trap will cause the system to operate improperly.

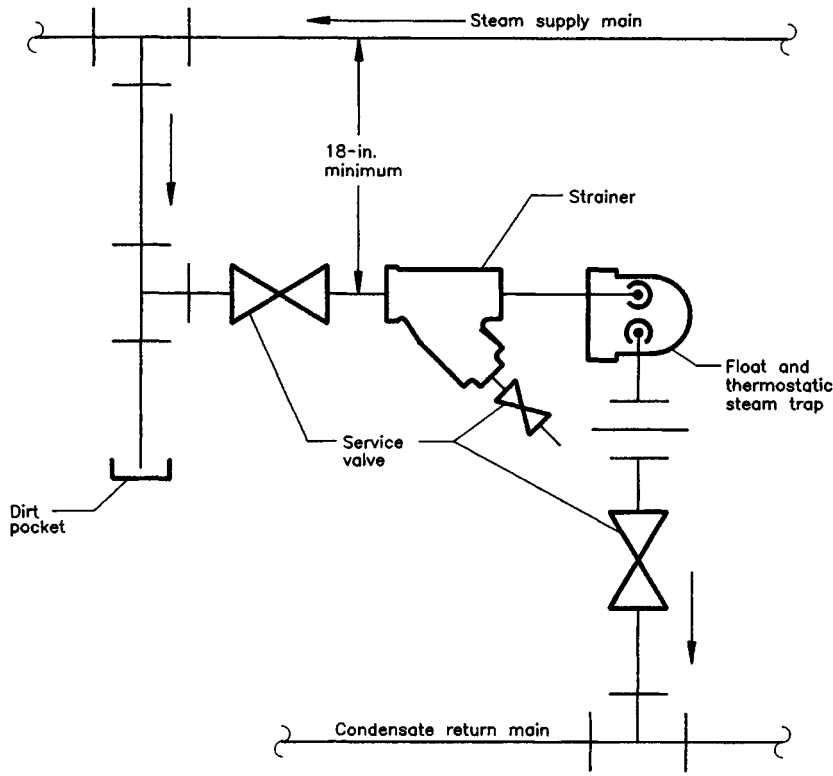


FIGURE 5.1.13 Typical drip-leg piping assembly.

TABLE 5.1.7 Startup Condensate Loads in Steam Mains, lb/h per 100-ft Length

Pipe size, in	Steam pressure, psig*†							
	0	5	15	30	50	100	150	200
2	6	7	8	9	10	13	15	16
2½	10	11	12	14	16	20	23	25
3	13	14	17	19	22	27	30	33
4	18	20	23	27	31	38	43	47
5	25	28	32	37	42	51	58	64
6	32	36	41	48	55	67	75	83
8	48	54	62	72	82	100	113	125
10	68	77	88	102	116	142	160	177
12	90	101	116	134	153	188	212	234

*Based on 70°F (21°C) ambient air. Schedule 40 pipe uninsulated.

†For metric equivalents, use the following conversion factors: 1 in = 2.54 cm = 25.4 mm; 1 lb/in² = 0.07 bar.

TABLE 5.1.8 Steam Trap Selection: Safety Factor

Trap type	Safety factor multiplier
Balanced-pressure thermostatic trap	3
Bimetallic trap	2.5
Liquid-expansion trap	3
Inverted-bucket trap	2.5
Float-and-thermostatic trap	2
Thermodynamic trap	1.5

5.1.23 STEAM TRAP SELECTION

Once the size of the steam trap is known, the type of trap which will provide the best performance must be selected. When a trap is chosen, care must be taken to select a type that will operate over the full range of pressures that the system will exert.

The best operating economy based on trap life and minimization of waste steam must be considered. If the trap will be subjected to low ambient temperatures, it should be of a freeze-proof design. For traps serving heating devices, continuous gas-venting capability is desirable. When the application is examined, the need for steam trap construction which is resistant to corrosion and water hammering should be considered.

5.1.24 DETERMINING CONDENSATE LOAD FOR A SYSTEM

The steam consumption of a system over time is equal to the amount of condensate formed during that period. Unfortunately, only when traps of the modulating type (such as float-and-thermostatic traps) are employed does the condensate return simultaneously equal the steam consumption.

If a blast type, say a bucket trap, is installed, the flow of condensate will be intermittent and equal to the trap's discharge rate, not the steam consumption rate. Since blast-type traps discharge intermittently, you can safely assume that not all the traps will discharge at once. For sizing purposes, the rule of thumb is that no more than two-thirds of the blast-type traps will discharge at any given time. This condensate load and the design steam consumption for the equipment utilizing modulating-type traps should be combined to determine the peak condensate load of the entire system. When the piping is sized, consider oversizing the condensate return main by one pipe size. This can be beneficial when future increases in the system's steam consumption are anticipated.

5.1.25 WATER DAMAGE

Water hammering is a phenomenon that occurs when condensate remaining in a pipe flashes into steam. The sudden expansion of the condensate causes a vibration

in the pipe which can lead to premature failure of joints and can cause an objectional noise throughout the structure the pipe is serving. A more dangerous situation can develop if enough condensate accumulates in the pipe to block the passage of steam. The steam pressure behind the blockage will build up. Eventually the blockage may be transmitted through the pipe at a speed approaching the design velocity of the steam. When water travels at such a high velocity, it can damage the first obstruction it comes to, such as a valve or elbow. Both water hammering and damage from blockages can be prevented by proper trapping and pitching of the steam lines.

When certain gases, such as carbon dioxide (CO₂), are trapped in steam lines, the gases tend to mix with the condensate and form unwanted by-products, such as mild acids. These by-products will accelerate the rate of erosion in the system and cause premature failure in the system's components. Proper air venting will reduce the amount of gas in the system and increases its operating life.

5.1.26 WATER CONDITIONING

The formation of scale and sludge deposits on boiler heating surfaces creates a problem in generating steam. Water conditioning in a steam generating system should be under the supervision of a specialist. Refer to Chap. 8.5 of this handbook for a discussion of water treatment.

5.1.27 FREEZE PROTECTION

Whenever a steam system is servicing an area whose outdoor temperature will drop below 35°F (1.7°C), the designer must make provisions to prevent freezing. An alarm should be installed to alert the building operator of a loss of steam pressure or exceptionally low condensate temperatures. If air-handling units are used, the alarm should also terminate the supply fan's operation. The following recommendations will help to minimize freezing problems in steam systems:

1. Select traps of nonfreezing design if they are located in potentially cold areas.
2. Install a strainer before all heating units.
3. Do not oversize traps.
4. Make sure that condensate lines are properly pitched.
5. Keep condensate lines as short as possible.
6. Where possible, do not use overhead return.
7. If heating coils are used, allow only the interdistributing tube type.
8. Limit the maximum tube length of heating coils to 10 ft (3 m).
9. All coils and lines should be vented and drainable.

5.1.28 PIPING SUPPORTS

All steam piping is pitched to facilitate the flow of condensate. Table 5.1.9 contains the recommended support spacing for piping. The data are based on Schedule 40 pipe filled with water and an average amount of valves and fittings.

TABLE 5.1.9
Recommended Hangar Spacing

Pipe size, in	Distance between supports, ft
	Length
$\frac{3}{4}$	4
1	7
$1\frac{1}{4}$	7
$1\frac{1}{2}$	9
2	10
$2\frac{1}{2}$	11
3	12
$3\frac{1}{2}$	12
4	14
5	15
6	17
8	19
10	20
12	23
14	25

Note: Figures are based on Schedule 40 steel pipe filled with water including a normal amount of valving and fittings. These conversion factors can be used: 1 in = 2.54 cm and 1 ft = 0.3 m.

5.1.29 STRAINERS

Strainers (Fig. 5.1.24) should be located in the supply main before all steam-consuming devices and as part of the drip-leg assembly to collect particles and sediment carried in the system. Strainers located in areas not susceptible to freeze-

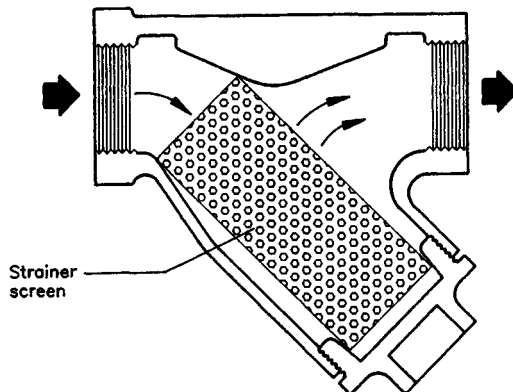


FIGURE 5.1.14 Typical strainer.

up should extend down directly under the steam lines to allow sediment and particles to collect at the bottom of the strainer. In areas where freezing is possible, strainers should be installed at about a 20° angle below the horizontal plane. This will form an air pocket which will allow for expansion if the water in the strainer freezes.

The strainers should be cleared regularly as part of a routine maintenance schedule.

5.1.30 PRESSURE-REDUCING VALVES

As steam pressure increases, the specific volume of the steam decreases as well as the heat of vaporization.

Many times the boiler is designed to operate at a higher steam pressure than the heating components. The higher boiler pressure allows the supply-main size to be reduced because of the smaller specific volume of the steam. At a convenient point in the main near the heating devices, a “pressure-reducing valve” is installed. This valve reduces the pressure and allows the steam to expand. As the steam expands, its heat of vaporization increases, allowing for greater system efficiency. The pipe size directly downstream of the pressure-reducing valve should be increased to accommodate the steam’s expansion. This should be done even if the reducing-valve connections for the inlet and outlet are the same size.

5.1.31 FLASH TANKS

A reservoir where condensate accumulates at low pressure before it returns to the boiler is normally provided. Another name for this reservoir is the *flash tank*. As the hot condensate reaches a low-pressure area, some of the liquid will flash into steam.

At the top of the flash tank, a steam line routes the steam that has just formed back into the system to be utilized. The flash tank improves the efficiency of the system and guarantees that only liquid condensate is returned to the boiler.

5.1.32 STEAM SEPARATORS

The need for pure steam without the presence of water droplets is imperative to permit control devices to operate properly. A device that allows vapor to pass while knocking water droplets from the stream is known as a *steam separator*.

Steam separators should be installed before all control devices and anywhere else in the system where small water droplets cannot be tolerated. Obviously, steam separators are not required on superheated-steam installations.

CHAPTER 5.2

HOT-WATER SYSTEMS

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5.2.1 INTRODUCTION

The predominant method of heating today's buildings, whether single-family dwellings or large structures, uses hot water to convey heat from a central generating source throughout the building. In nearly all new construction, the water is circulated through a piping distribution network by an electrically driven pump; this type of system is classified as a forced-circulation system. Heat from the circulating water is transferred to radiators, finned tubes, cabinet heaters, or other types of terminal units (see Chaps. 5.9 to 5.13) distributed strategically throughout the structure.

Older systems used gravity to circulate the hot water, by utilizing the difference in density between supply and return columns of the piping network. Since this type of system is rarely installed today, this chapter confines itself to forced-circulation systems. As a matter of fact, the latest American Society of Heating, Refrigeration, and Air-Conditioning Engineers (ASHRAE) guide refers readers to editions published before 1957 for details on designing gravity hot-water systems.

All hot-water heating systems rely on some form of central generating facility as the source of heat. This facility can be in the form of a boiler that consumes oil, gas, or electricity as the prime energy source or steam-to-water and water-to-water heat exchangers that derive heat from a utility or district-heating network.

This chapter gives details on the basic types of hot-water systems, as characterized by their temperature rating, general principles of system design, and special considerations of the equipment that comprises hot-water systems.

5.2.2 CLASSES OF HOT-WATER SYSTEMS

Hot-water systems are classified by operating temperature into three groups: low, medium, and high temperature. The *1987 ASHRAE Handbook* provides the following distinctions among these systems:

1. *Low-temperature water (LTW) system:* A low-temperature hot-water system operates within the pressure and temperature limits of the American Society of Mechanical Engineers' (ASME) *Boiler Construction Code* for low-pressure heating boilers. The maximum allowable working pressure for such boilers is 160

lb/in² (11 bar) with a maximum temperature of 250°F (121°C). The usual maximum working pressure for LTW systems is 30 lb/in² (2 bar), although boilers specifically designed, tested, and stamped for higher pressures frequently may be used with working pressures to 160 lb/in² (11 bar). Steam-to-water or water-to-water heat exchangers are often used, too.

2. *Medium-temperature water (MTW) system:* MTW hot-water systems operate at temperatures of 350°F (177°C) or less, with pressures not exceeding 150 psia (10.5 bar). The usual design supply temperature is approximately 250 to 325°F (121 to 163°C), with a usual pressure rating for boilers and equipment of 150 lb/in² (10.5 bar).
3. *High-temperature water (HTW) system:* When operating temperatures exceed 350°F (177°C) and the operating pressure is in the range of 300 lb/in² (20.7 bar), the system is an HTW type. The maximum design supply water temperature is 400 to 450°F (205 to 232°C). Boilers and related equipment are rated for 300 lb/in² service (21 bar). The pressure and temperature rating of each component must be checked against the system's design characteristics.

LTW systems are generally used for space heating in single homes, residential buildings, and most commercial- and institutional-type buildings such as office structures, hotels, hospitals, and the like. With a heat-transfer coil or similar device inside or near the boiler, LTW systems can supply hot water for domestic water supplies. Terminal units vary widely and include radiators, finned-tube fan-coil units, unit heaters, and others. Typically overall heat loads do not exceed 5000 to 10,000 MBtu/h (1.5 to 3 MW).

MTW systems show up in many industrial applications for space heating and process-water requirements. Overall loads range up to 20,000 MBtu/h (6 MW). Generally HTW systems are limited to campus-type district heating installations or to applications requiring process heat in the HTW range. System loads are generally greater than 20,000 MBtu/h (6 MW).

The designs of MTW and HTW systems resemble each other closely. The systems are completely closed, with no losses from flashing. Piping can run in practically any direction, since supply and return mains are kept at substantial pressures. Higher temperature drops occur in MTW and HTW systems, relative to LTW systems, while a lesser volume of water is circulated (depending on the heat load of the system). LTW systems lend themselves better to combined hot-water/chilled-water heating/cooling systems. Extra care and expense must be devoted to fittings, terminal equipment, and mechanical components, especially for HTW systems.

Finally, often a combined system is desirable: an MTW or HTW circuit for process heat and an LTW circuit for space heating. The hot water for the LTW system can be obtained via a heat exchanger with the main heating system.

5.2.3 DESIGN OF HOT-WATER SYSTEMS

Design hot-water systems involves a complex interplay of heat loads and the type of generating system. A traditional starting point, primarily for residential LTW systems, was the assumption of a 20°F (11°C) temperature drop through the circuit, from which the overall flow rate could be determined. A more recent practice is to perform a rigorous analysis, because the 20°F (11°C) assumption can lead to oversized pipes and flow rates.

TABLE 5.2.1 Typical Ratings of Wall Fin Elements

Element type	Rows	Hot-water capacity, Btu/(h · ft),* at 65°F (17.4°C), entering air with average water temperature of:					
		220°F 104.4°C	210°F 98.9°C	200°F 93.3°C	190°F 87.8°C	180°F 82.2°C	170°F 76.7°C
Steel, 1¼ in (32 mm)}	1	1260	1140	1030	940	830	730
	2†	2050	1850	1680	1520	1350	1190
Copper-aluminum, 1 in (25.4 mm)}	1	1000	900	820	740	660	580
	2†	1480	1340	1210	1100	970	860
Steel, grilled enclosure, 1 in (25.4 mm)}	1	1310	1190	1080	980	860	760
	2†	2080	1880	1700	1540	1370	1210

*1 Btu/(h · ft) = 0.0768 kcal/(h · m).

†4-in (10.2-mm) center-to-center gap.

System design can be broken down into five elements:

1. Determining the heat load
2. Selecting terminal units or convectors based on the average water temperature and temperature drop and locating them on the architectural plan
3. Developing a piping layout, including the choice of return system
4. Locating mains, side branches, and other piping elements
5. Specifying mechanical components, the expansion tank, and the boiler

A good initial point is to run the flow main from the boiler to the terminal unit or units with the largest heat load and then to select branch runs to connect other terminal units. Common space-heating terminal elements are convectors of wall fins, both of which contain a length of finned tube over which air can be fanned if desired. The air entering temperature is usually assumed to be 65°F (18°C). Most manufacturers supply tables showing heat ratings of the convectors, based on the assumed temperature drop, and the average entering water temperature (AWT). See Table 5.2.1 for an example for finned-tube convectors.

An alternative approach is to assume a constant-temperature water flow (based on the leaving temperature of each class of terminal equipment) and to compute the required flow rate.

Both daily and annual variations in heat loads should be evaluated in order to arrive at a suitable design. This is especially true when LTW systems combining hot-water heating and cool-water cooling are envisioned. Figure 5.2.1 shows the seasonal effects of outside temperature on one type of piping design, the two-pipe system.

5.2.4 PIPING LAYOUT

Once a preliminary evaluation of heat load and terminal units has been performed, a piping layout can be undertaken. The usual starting-point options—running the flow main by the shortest and most accessible route to the larger heat loads—can be explored for the type of overall piping arrangement desired.

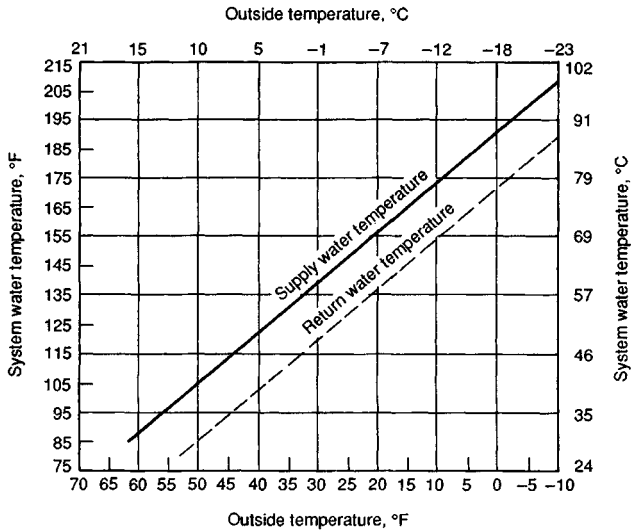


FIGURE 5.2.1 Seasonal operating characteristics of a two-pipe forced hot-water system. (Courtesy of The Industrial Press.)

Pipe circuits generally are organized into one- or two-pipe arrangements. One-pipe systems with radiators or similar terminal units often have a feed and return pipe that diverts water from the flow main to the radiator and back to the flow main; even though two pipes are present, the system is still considered a one-pipe arrangement (see Fig. 5.2.2). Finned-tube heating elements running along the outer walls of small residences—a common arrangement—are true one-pipe systems, as shown in Fig. 5.2.3. Each terminal unit in the circuit receives progressively lower

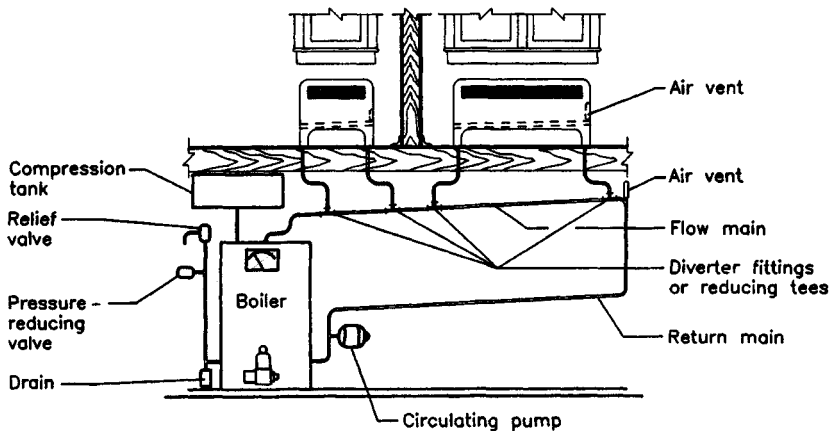


FIGURE 5.2.2 Arrangement of piping for a one-pipe forced hot-water system with closed expansion tank. (Courtesy of The Industrial Press.)

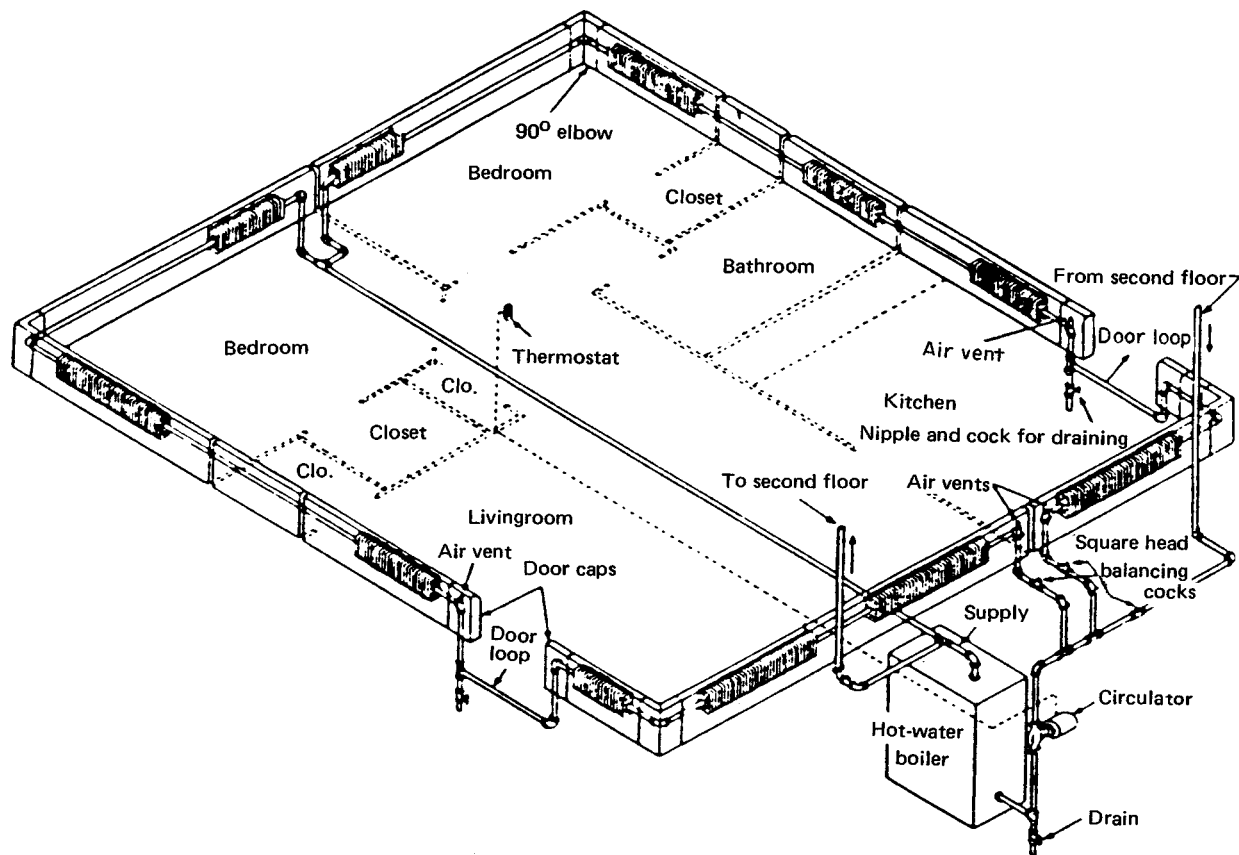


FIGURE 5.2.3 Typical installation of one-pipe forced-circulation “loop” hot-water system using baseboard radiators. (Courtesy of The Industrial Press.)

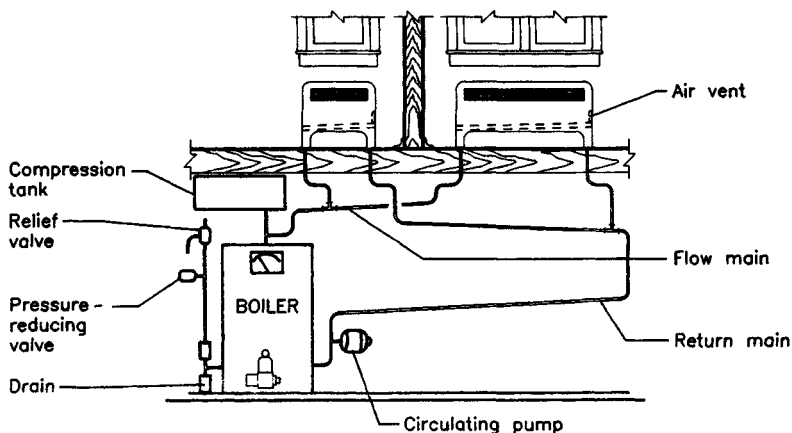


FIGURE 5.2.4 A two-pipe forced hot-water heating system with reverse-return piping.
(Courtesy of The Industrial Press.)

water temperature; thus the units are sized larger as they are located farther from the heat source.

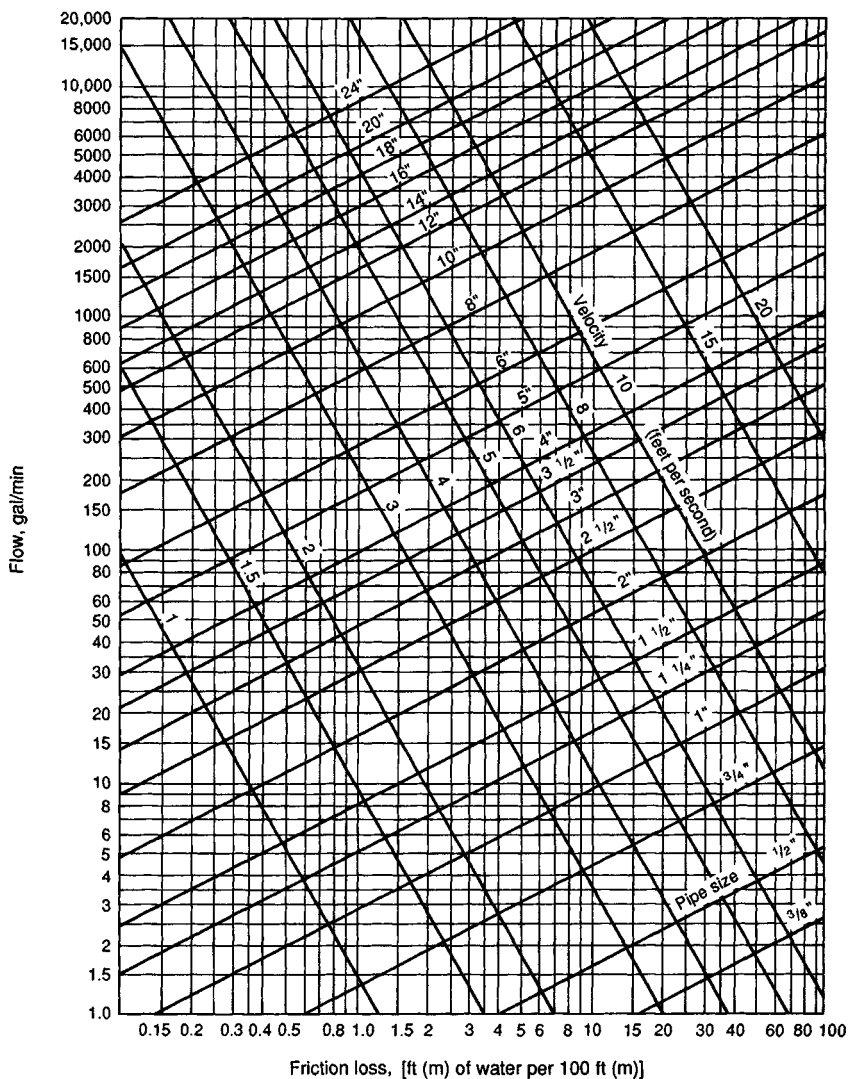
Two-pipe systems allow for parallel heating arrangements, whereby terminal units can receive hot water at roughly similar inlet temperatures. The cooled water returns via a second pipe. The flow of this pipe can be specified to run in direct or reverse fashion back to the heat generator. Choosing between these options allows for better balancing of heat supplies among various terminal units and for some variation in overall system capital cost. Reverse-return systems specify that the distance that the water travels to a particular unit is the same as the return distance from that unit (Fig. 5.2.4).

5.2.5 PRESSURE DROP AND PUMPING REQUIREMENTS

All hot-water systems require some type of pumping to overcome friction losses of the flowing water, because whatever head is developed by the height of the water system (static pressure) is offset by the return pressure. Some more complex systems are better served economically by two or more pumps strategically located, rather than one large pump.

Standard charts provide data on friction loss for runs of common types of piping (Fig. 5.2.5). To this should be added pressure losses from elbows, fittings, and other elements (Table 5.2.2). Similarly, manufacturers of radiators and other terminal units provide data on friction losses through their equipment.

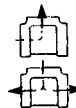
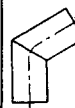
Pump specifications are arrived at by first computing the overall pressure drop and the amount of desired water flow. "Pump curves"—charts which show the pressure developed by pumps as a function of the flow rate—can be used to arrive at the correct sizing. Many designers prefer to work with mass flow rate [pounds (kilograms) per hour] rather than gallons per minute (liters per second), units com-



Metric conversion: gal/min to L/min = 3.78

FIGURE 5.2.5 Friction loss for open-system piping. (From Carrier Air Conditioning Company, *Handbook of Air Conditioning System Design*, McGraw-Hill, New York, © 1965. Used with permission.)

TABLE 5.2.2 Fitting Losses in Equivalent Feet* of Pipe
Screwed, welded, flanged, flared, and brazed connections

Nom- inal pipe or tube size, in*	Smooth bend elbows						Smooth bend tees				Mitre elbows			
	90° Std.†	90° Long Rad.‡	90° Street†	45° Std.†	45° Street†	180° Std.†	Flow- through branch	Straight-through flow			90° El	60° El	45° El	30° El
								No re- duction 	Re- duced ¼ 	Re- duced ½ 				
3/8	1.4	0.9	2.3	0.7	1.1	2.3	2.7	0.9	1.2	1.4	2.7	1.1	0.6	0.3
1/2	1.6	1.0	2.5	0.8	1.3	2.5	3.0	1.0	1.4	1.6	3.0	1.3	0.7	0.4
3/4	2.0	1.4	3.2	0.9	1.6	3.2	4.0	1.4	1.9	2.0	4.0	1.6	0.9	0.5
1	2.6	1.7	4.1	1.3	2.1	4.1	5.0	1.7	2.3	2.6	5.0	2.1	1.0	0.7
1 1/4	3.3	2.3	5.6	1.7	3.0	5.6	7.0	2.3	3.1	3.3	7.0	3.0	1.5	0.9
1 1/2	4.0	2.6	6.3	2.1	3.4	6.3	8.0	2.6	3.7	4.0	8.0	3.4	1.8	1.1
2	5.0	3.3	8.2	2.6	4.5	8.2	10	3.3	4.7	5.0	10	4.5	2.3	1.3
2 1/2	6.0	4.1	10	3.2	5.2	10	12	4.1	5.6	6.0	12	5.2	2.8	1.7
3	7.5	5.0	12	4.0	6.4	12	15	5.0	7.0	7.5	15	6.4	3.2	2.0
3 1/2	9.0	5.9	15	4.7	7.3	15	18	5.9	8.0	9.0	18	7.3	4.0	2.4
4	10	6.7	17	5.2	8.5	17	21	6.7	9.0	10	21	8.5	4.5	2.7
5	13	8.2	21	6.5	11	21	25	8.2	12	13	25	11	6.0	3.2
6	16	10	25	7.9	13	25	30	10	14	16	30	13	7.0	4.0
8	20	13	...	10	...	33	40	13	18	20	40	17	9.0	5.1
10	25	16	...	13	...	42	50	16	23	25	50	21	12	7.2
12	30	19	...	16	...	50	60	19	26	30	60	25	13	8.0
14	34	23	...	18	...	55	68	23	30	34	68	29	15	9.0
16	38	26	...	20	...	62	78	26	35	38	78	31	17	10
18	42	29	...	23	...	70	85	29	40	42	85	37	19	11
20	50	33	...	26	...	81	100	33	44	50	100	41	22	13
24	60	40	...	30	...	94	115	40	50	60	115	49	25	16

*Conversion factors: 1 ft = 0.31 in; 1 in = 25.4 mm.

†R/D approximately equal to 1.

‡R/D approximately equal to 1.5.

Source: Carrier Air Conditioning Company, *Handbook of Air Conditioning System Design*, McGraw-Hill, New York, © 1965. Used with permission.

mon to pump curves. The conversion between the two is temperature-dependent; two quick conversions commonly used are

Water at 40°F (4.4°C): 1 lb/h = 0.002 gal/min (1 kg/h = 1.26 E - 4 L/s)

Water at 400°F (204.5°C): 1 lb/h = 0.0023 gal/min (1 kg/h = 1.45 E - 4 L/s)

The next step is to determine the system curve for the hot-water circuit. The following formula is employed:

$$\frac{H_1^{0.5}}{W_1} = \frac{H_2^{0.5}}{W_2} \quad (5.2.1)$$

where H_1 = known or calculated head, ft (m)

W_1 = design flow rate, gal/min (L/s)

H_2 = system curve head point, ft (m)

W_2 = system curve flow-rate point, gal/min (L/s)

With this equation, various system curve points can be plotted on the pump curve. The point where the system curve and the pump curve intersect is the operating point of the pump. Pump manufacturers specify optimum operating conditions (in terms of energy consumption, efficiency, and capacity of the pump) for their equipment.

5.2.6 PIPE SIZING

Hot-water system piping must be sized to carry the maximum desired amount of heating water throughout the system, while accounting for the static head of the main to account for fittings. More accurate determinations are made by adding the equivalent pipe lengths of the fittings on the main to the length of the longest main. The manufacturer's literature usually includes charts similar to Table 5.2.2 showing equivalent lengths of common fittings.

Once the overall heating demand and the operating temperature of the heating system are known, calculations can be made for pipe sizes. The relationship between Btu demand and water flow rate is

$$\text{Btu/h} = \text{gal/min} (500 \Delta T^\circ\text{F}) \quad (5.2.2)$$

A rough calculation of the overall friction head for the main can be done by measuring the longest main circuit and adding an equivalent length of 50 percent of the main to account for fittings. More accurate determinations are made by adding the equivalent pipe lengths of the fittings on the main to the length of the longest main. The manufacturer's literature usually includes charts similar to Table 5.2.2 showing equivalent lengths of common fittings.

Various methods have been worked out to determine the suitable pipe diameters to provide a sufficient flow rate. Usually the procedure must be iterated several times to select the best combination of flow rate, fluid velocity, and pressure drop. Table 5.2.3 shows these relationships for various pipe sizes if one assumes a maximum pressure drop of 4 ft per 100 ft (1.2 m per 30.5 m) and a maximum velocity of 10 ft/s (3 m/s). Once the pipe sizes have been determined, the system's pressure head should be compared to the head developed by the circulation pump. The pump may have to be resized, necessitating another iteration of the pipe sizing.

TABLE 5.2.3 Allowable Flow Rates for Closed System Piping, Standard-Weight Steel Pipe

Pipe size, in	Flow range, gal/min	Pressure drop range, ft per 100 ft
½	0–2	0–4
¾	3–4	2.5–4
1	5–7.5	2.0–4
1¼	8–16	1.25–4
1½	17–24	2–4
2	25–48	1.25–4
2½	49–77	2–4
3	78–140	1.5–4
4	141–280	1.25–4
5	281–500	1.5–4
6	501–800	1.75–4
8	801–1700	1.0–4
10	1701–2500	1.25–2.75
12	2501–3600	1.25–2.25
14	3601–4200	1.25–2.0
16	4201–5500	1.0–1.75
18	5501–7000	0.9–1.50
20	7001–9000	0.8–1.25
24	9001–13,000	0.6–1.00

Note: The above capacities are based on a maximum pressure drop of 4 ft per 100 ft and a maximum velocity of 10 ft/s. Conversions: 1 in = 25.4 mm, 1 gal = 3.8 L, and 1 ft = 0.31 m.

5.2.7 VENTING AND EXPANSION TANKS

Hot-water systems require pressures greater than atmospheric at all times to prevent air infiltration. Flashing or boiling of water is also minimized by maintaining the system above the water vapor pressure—preventing this also minimizes water hammer.

Maintaining this pressure, as well as allowing for the expansion and contraction of water as it is heated and cools, is most frequently carried out by means of an expansion tank. The expansion of medium- or high-temperature water systems can be calculated by consulting steam tables. The specific volume of water at its initial conditions is subtracted from its volume at the highest temperature, to calculate the volume change. To a certain limited extent, the water's expansion and contraction are offset by the similar changes that system piping and heating units undergo. These changes can be calculated from coefficients of expansion of the materials of the piping.

The simplest type of expansion tank is open to atmosphere at an elevation that provides the pressurization (head) the system requires. Open tanks have the disadvantage of allowing air to enter the system via absorption in the water. Closed tanks are more common now, especially with larger systems. Three common types of expansion/pressurization tanks are in use today:

1. *Adjustable expansion tank.* This tank employs an automatic valve along with a closed tank that has water and air feeds. As the temperature in the system rises,

the pressure rises. A control valve releases air in the tank to the atmosphere. When the pressure and the water level drop, high-pressure air is injected into the tank. High-temperature systems should use nitrogen rather than plain air to reduce corrosive effects.

2. *Pump-pressurized cushion tank.* This design involves a makeup tank which is fed by a pump and a back-pressure control valve. For small systems (depending on local codes and on the water pressure available) the pump is skipped and city water pressure is used to feed a makeup tank that pressurizes the heating circuit. In principle, either type of pressurized tank can be roughly sized by assuming the expansion and contraction rates of the water to be equal.
3. *Compression tank.* A compression tank employs a specified volume of gas within an enclosure. As the water temperature and volume increase, the pressure on the gas volume rises, causing that gas volume to decrease. In this manner, the tank accommodates changing water volumes while keeping the system within a specified range of upper and lower pressures.

In low-temperature systems, the compression tank is usually connected to the system through an air separator situated between the boiler exit and the suction inlet of the circulating pump. Air separated from the water will rise into the compression tank. When the compression tank is located at a system's high point, it can be smaller in volume since the pressure is at its lowest. Tank sizing is also dependent on the location of the circulation pump relative to the tank.

One commonly used formula for sizing the compression tank, when operating temperatures are below 160°F (71.1°C), is

$$V_t = \frac{EV_3}{P_0/P_1 - P_0/P_2} \quad (5.2.3)$$

where V_t = compression tank volume

V_s = volume of circulating system, exclusive of compression tank

E = coefficient of expansion from initial to operating temperature

P_0 = absolute pressure in compression tank prior to filling

P_1 = absolute static pressure after filling

P_2 = absolute pressure at system operating temperature

For operating temperatures between 160 and 280°F (71.1 to 137.8°C), this formula is used:

$$V_t = \frac{(0.00041t - 0.0466)V_s}{P_0/P_1 - P_0/P_2} \quad (\text{USCS units}) \quad (5.2.4)$$

$$V_t = \frac{(0.000738t - 0.03348)V_s}{P_0/P_1 - P_0/P_2} \quad (\text{metric units}) \quad (5.2.5)$$

where t = maximum operating temperature.

Compression tanks can be supplied with an impermeable membrane (diaphragm) to prevent air from being drawn into the circulating water when the system temperature drops. The diaphragm also allows the compression tank to be smaller in volume.

Diaphragm compression tanks are equipped with sight glasses or similar devices to monitor the water level. Too low a water level prevents the air behind the diaphragm from affecting the system's pressure.

5.2.8 MECHANICAL AND CONTROL EQUIPMENT

Mechanical components for low-temperature systems are under less severe service than those for medium- and high-temperature units; correspondingly, the care with which components are specified should increase with the higher-temperature systems.

ASME and ASHRAE rules should be observed for dealing with pressure vessels. Specifically, the chemical condition of the circulating water in high-temperature systems should be checked periodically by an expert. Pressure gauges should be located at both ends of the circulation pump. Modulating combustion controls, rather than straight on/off controls, are necessary to minimize pressure swings that lead to flashing. Where compression-type expansion tanks are used, an interlock with the system's heat generator should be installed to prevent operation when the water level in the tank is too low or insufficient air is present to maintain the tank compression. Valves and fittings for high-temperature systems should be specified with materials that resist corrosion and erosion, such as stainless steel.

The primary control factor for a hot-water system is the operating temperature range, which in turn is based on outside air temperatures. Electronic thermosensors and thermostats function to keep the room air temperature within the desired range. The system should also be equipped with a manual on/off control.

The electronic control for moderating the room air temperature can be of several types. Most are based on a solenoid device, which sends a signal current on the basis of a temperature reading. The control can be a simple on/off device or can have various modulating schemes to minimize large temperature swings. Temperature controls can also be set for zone heating of certain rooms or areas within a large room, depending on the piping layout. In this case, the electronic control is connected with various flow control valves that will reduce or expand water flow to the heating units.

SECTION 6

REFRIGERATION SYSTEMS FOR HVAC

CHAPTER 6.1

REFRIGERANTS

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6.1.1 INTRODUCTION

Refrigerants are the working fluids for refrigeration cycles. They absorb heat from the medium to be cooled; air in the case of a direct expansion air conditioner or water for a chiller. The absorbed heat is then carried by the refrigerant to a heat rejector, e.g. condenser, where the heat can be given up. The refrigerant is then recycled in the system to absorb more heat. In most refrigeration systems, this is a continuous process, so heat is continually being absorbed and rejected as the refrigerant is moved around the cycle.

The most common type of refrigeration cycle is the *vapor compression cycle*. This is the type of refrigeration cycle used in household refrigerator/freezers, automobile air conditioning, most residential, commercial and institutional air conditioning and commercial (supermarket) refrigeration. Other types of refrigeration cycles include *absorption* which is used in some large water chillers and a very small percentage of residential systems. Commercial aircraft use the *Brayton cycle* which is an all gas cycle using air as the refrigerant.

6.1.2 SELECTION CRITERIA

Almost any fluid can be made to function as a refrigerant in a variety of cycles. Many fluids, however, exhibit undesirable properties which limit their utility in refrigeration cycles. Traditionally, refrigeration system designers have based the selection of the refrigerant on three major criteria; *safety, reliability and performance*. More recently, a fourth criterion has emerged; *environmental acceptability*.

6.1.2.1 Safety

Safety is generally broken down into two areas, *flammability* and *toxicity*. Both are complex issues, the details of which are beyond the scope of this text.

In general, toxicity addresses acute, subchronic and chronic effects. Within these broad categories, the effect on future generations (mutations, birth defects, etc.) as well as exposed individuals are evaluated before products are introduced into commerce. ASHRAE Standard 34-1992¹ broadly classifies refrigerants on the basis of chronic exposure limits as defined by TLV-TWA (Threshold Limit Value-Time Weighted Average) or equivalent indices. Class A refrigerants are those deemed to be of low toxicity with allowable exposure limits of 400 ppm or greater for a 40 hour work week. Class B refrigerants are those which have a greater toxicity with exposure limits of less than 400 ppm. This does not mean that refrigerants with a B classification cannot be used safely. In fact, several refrigerants that have been assigned a B classification have been successfully used for many years. ASHRAE Standard 15-1992², addresses acute toxicity by limiting the quantity of refrigerant permitted in occupied space. It is likely that future revisions of the above ASHRAE standards will address toxicity in greater detail.

Flammability is also classified by ASHRAE Standard 34-1992. This standard utilizes three classifications—1, 2 or 3 based on flammability. Class 1 refrigerants are non-flammable, Class 2 are moderately flammable and Class 3 are highly flammable. Class 3 would include chemicals that are used as fuels, such as hydrocarbons. Most refrigerants in use for air conditioning applications are Class 1 fluids. There are other organizations that classify refrigerants based on flammability. Underwriters Laboratories is one of the most widely recognized in the U.S. because of its safety listing service for air conditioning and refrigeration equipment. For transportation purposes in the U.S., there is a third basis of classification promulgated by the Department of Transportation. Each of these classification schemes can yield differing results based on the criteria used, e.g. ammonia, which is classified as *moderately flammable* by ASHRAE (Class 2), classified as *flammable* by UL and *non-flammable* by the DOT. In the past, even the test methods used to evaluate flammability differed among various organizations. Recently, there has been an effort by ASHRAE and UL to harmonize test methods and conditions.

Until recently, most refrigerants were single-component fluids plus a limited number of azeotropes. Refrigerant blends which can fractionate substantially are now being used commercially. Refrigerant blends receive dual safety classifications under the ASHRAE standards. The classifications are based on “as formulated” and “worse case of fractionation.” Blends can shift composition under various leakage scenarios and inherently have differing compositions in the liquid and vapor phases. From a safety code standpoint, *the rating which indicates the greatest hazard is the one that applies.*

6.1.2.2 Reliability

Reliability of a refrigeration system is largely dependent on the hardware design, installation and application. The refrigerant can, however, affect the reliability of the system and its properties are a part of the selection process.

Chemical stability is a very important property for a refrigerant. If the refrigerant decomposes, due to the temperatures or pressures that it is exposed to in the manufacture of the refrigeration system, shipping, storage or operation, it is unlikely that continued operation of the system will be satisfactory. The decomposition products will have property differences which can severely impact capacity, efficiency or other operating characteristics.

Material compatibility is equally vital. If the refrigerant is corrosive to metals in the system or if it dissolves or embrittles plastics and elastomers, unsatisfactory

performance and/or life can be expected. In many cases, materials can be selected in the design process for which compatibility has been determined.

Lubricant miscibility/solubility is generally considered desirable because it is the primary mechanism for oil return to the compressor. It is possible to design systems which operate with lubricants which are immiscible and insoluble with the refrigerant, but the complexity of the refrigerant piping is increased.

There are other properties which are important, including dielectric strength for systems using hermetic compressors, and a freezing point well below the expected operating range (and also below the unit storage range).

6.1.2.3 Performance

The performance of a refrigeration system is characterized by its capacity and efficiency. The choice of the refrigerant can dictate the type of system as well as the size and configuration of most components including the compressor, condenser, evaporator, expansion device and connecting lines. There are two types of properties which dictate performance; *thermodynamic and transport*. Together, these properties are considered to be thermophysical properties.

If one were to select a single property with which to characterize the performance of a refrigerant, it would be the *boiling point* (understood to be at atmospheric pressure). There is a very strong relationship between boiling point and the theoretical capacity of a refrigeration system as illustrated by Fig. 6.1.1. This correlation is useful in selecting a replacement refrigerant for the same or similar equipment. There is an inverse relationship between boiling point and vapor pressure. A refrigerant with a low boiling point will have a high vapor pressure and vice versa.

In general, refrigerants with a high boiling point have more favorable thermodynamic properties. If one analyzes two different refrigerants on Mollier (pressure-enthalpy) diagrams, the lower pressure (higher b.p.) refrigerant will generally ex-

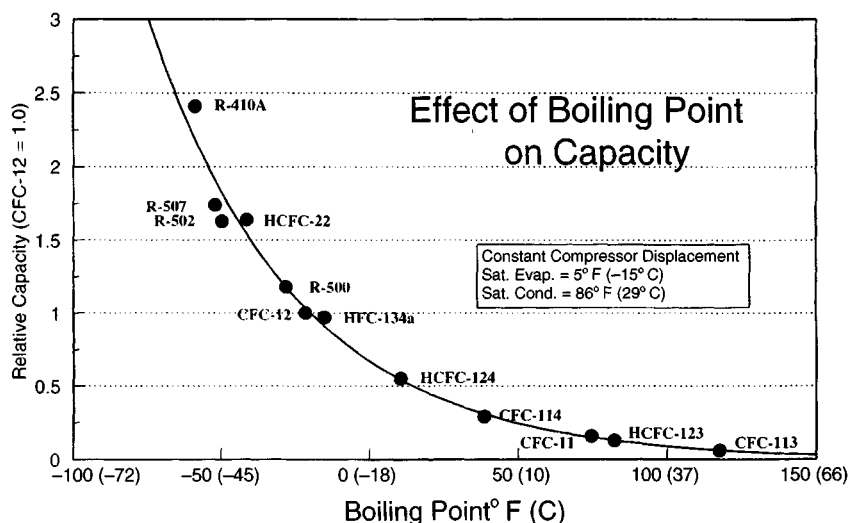


FIGURE 6.1.1 Effect of boiling point on capacity.

hibit a higher efficiency. Real systems, however have losses associated with pressure drops and heat transfer resistances. These losses are associated with the transport properties such as viscosity and thermal conductivity. Higher pressure (low b.p.) refrigerants tend to have more favorable transport properties and therefore the losses from the ideal cycle are less. This means that there is a tradeoff between the thermodynamic and transport properties of alternative refrigerants. As a result, there is seldom an obvious choice of refrigerant and the selection of the "best" refrigerant usually comes down to economics. Table 6.1.1 shows selected refrigerants arranged by boiling point.

Additional properties that can affect performance include *latent heat and vapor heat capacity*. Latent heat capacity is simply a measure of the amount of heat per lb. (kg) absorbed or rejected during phase change. Obviously a high latent heat capacity will reduce the required mass flow rate which will tend to reduce pressure drop losses. Either too high or too low vapor heat capacity can have a negative impact on both efficiency and reliability of the compressor. If the vapor heat capacity is too high, so-called *wet compression* can occur. This means that liquid is formed in the compression process which can result in physical damage to the compressor. If too low, excessive superheating will occur during compression with high discharge temperatures. Hermetic compressors usually rely on moderate vapor heat capacity to cool the motors and overheating can result if the vapor heat capacity is too low.

6.1.2.4 Environmental Acceptability

The generally accepted theory that the chlorine in fluorocarbon refrigerants can be a major contributor to stratospheric ozone depletion has resulted in international regulations to phase out the chlorine containing species. *It has also resulted in a heightened awareness of other potential environmental concerns. As a result, future*

TABLE 6.1.1 Selected Refrigerants Arranged by Boiling Point

ASHRAE Number	Type of Refrigerant	Class of Refrigerant	Boiling Point °F/(C)
123	Single Component	HCFC	82.2 (27.9)
11	Single Component	CFC	74.9 (23.8)
134a	Single Component	HFC	-15.1 (-26.2)
12	Single Component	CFC	-21.6 (-29.8)
401A	Zeotrope	HCFC	-27.7 (-33.2)
500	Azeotrope	CFC	-28.3 (-33.5)
22	Single Component	HCFC	-41.5 (-40.8)
407C	Zeotrope	HFC	-46.4 (-43.6)
502	Azeotrope	CFC	-49.8 (-45.4)
404A	Zeotrope	HFC	-51.0 (-46.1)
507	Azeotrope	HFC	-52.1 (-46.7)
402A	Zeotrope	HCFC	-54.8 (-48.2)
410A	Zeotrope	HFC	-62.9 (-52.7)
13	Single Component	CFC	-114.6 (-81.4)
23	Single Component	HFC	-115.7 (-82.1)
503	Azeotrope	CFC	-126.1 (-87.8)

fluorocarbon based refrigerants will be restricted to hydrofluorocarbons (HFCs) with no chlorine.

Another environmental issue which has been raised is the effect of refrigerants on global warming; the so-called "greenhouse effect". This effect is related to the infrared absorption characteristics of various gases. On a global basis, both carbon dioxide and water vapor are by far the major contributors to this effect, but other gases including fluorocarbons can have an impact. On a per molecule basis, fluorocarbons tend to have a relatively high potential for contribution to atmospheric warming. The term, global warming potential (GWP), has been introduced as a convenient way of indicating the direct contribution of various gases if emitted to the atmosphere, e.g. leakage or during servicing. *For most air conditioning equipment, this direct effect is overshadowed by the indirect effect created by the introduction of carbon dioxide into the atmosphere at the electric power plant.* The CO₂ is generated in the production of electric power to operate the air conditioners. A method to incorporate both of these effects into a single index number has been developed. This single number has been designated as TEWI³ (Total Equivalent Warming Impact). The TEWI value captures the total warming contribution over the life of a piece of refrigeration equipment. *It should be noted, that there is considerable controversy within the scientific community as to whether global warming is a real threat and if so, how serious.*

6.1.3 REFRIGERANT TYPES

There are several ways in which one can classify refrigerants. One method is by their *chemical/molecular composition*. Another useful way of distinguishing refrigerants is by their *physical composition*, i.e. whether they are single component fluids or mixtures. They can also be classified according to the type of refrigeration system that they are used in.

6.1.3.1 Chemical Composition

Chemicals suitable for use as refrigerants can be broadly broken down into two major classifications; *organic* and *inorganic*. The organic group can be subdivided into hydrocarbons and halocarbons. Halocarbons can be further subdivided into chlorofluorocarbons (CFCs), hydrochlorofluorocarbons (HCFCs) and hydrofluorocarbons (HFCs).

Inorganic refrigerants would include ammonia, water, and air as well as other more obscure refrigerants. ASHRAE also classifies carbon dioxide as inorganic although most chemists would disagree.

Hydrocarbons would include propane, ethane and isobutane. There are other hydrocarbons, of course, but use as refrigerants has been rare. In general, hydrocarbons have not found favor for use in air conditioning systems because of their extremely flammable characteristics. They have been successfully applied as refrigerants in industrial systems.

The halocarbons have found widespread use as refrigerants because many members of this family of compounds have exhibited desirable characteristics in terms of stability, low toxicity and non-flammability. They have also demonstrated excellent thermodynamic and transport properties which yield high efficiencies and

reliable systems in practice. The halocarbons are hydrocarbon based molecules where some or all of the hydrogens have been replaced by halogens. In general, the halogens are restricted to chlorine and fluorine although a few bromine containing compounds have seen limited usage as refrigerants, but not for air conditioning applications. Bromine is considered to be a greater depleter of stratospheric ozone than chlorine and therefore the few bromine containing refrigerants were phased out even before the CFCs.

The CFCs are fully halogenated, i.e. *all hydrogens on the base hydrocarbon have been replaced by halogens*. The HCFCs contain chlorine but still retain one or more hydrogen atoms on the molecule. The effect of the hydrogen is to reduce the atmospheric stability which shortens the lifetime of the molecule resulting in substantial reductions in the potential to deplete stratospheric ozone. As an example, HCFC-22 is almost identical in structure to CFC-12. Both molecules have a single carbon atom surrounded by four atoms of which two are fluorine. The difference is that for CFC-12, the two remaining atoms are both chlorine and for HCFC-22 only one is chlorine and the other is hydrogen. The result is that HCFC-22 has an ozone depletion potential (ODP) that is only five percent of that of CFC-12. The HFCs contain only hydrogen and fluorine atoms attached to the carbon backbone. Fig. 6.1.2 illustrates typical molecular structures for the three major classes of halocarbons.

A useful way to look at the various molecules which can be created from the substitution of chlorine and fluorine atoms on basic simple hydrocarbons is to arrange the compounds in a triangular pattern according to their molecular structure. Figures 6.1.3 and 6.1.4 show the methane and ethane based refrigerants respectively. Each dot represents a possible combination of atoms, where the top molecule is the base hydrocarbon and each successive row down represents an additional substitution of either chlorine or fluorine atoms. This particular arrangement provides one of the easiest ways of understanding the refrigerant numbering system which ASHRAE has adopted.

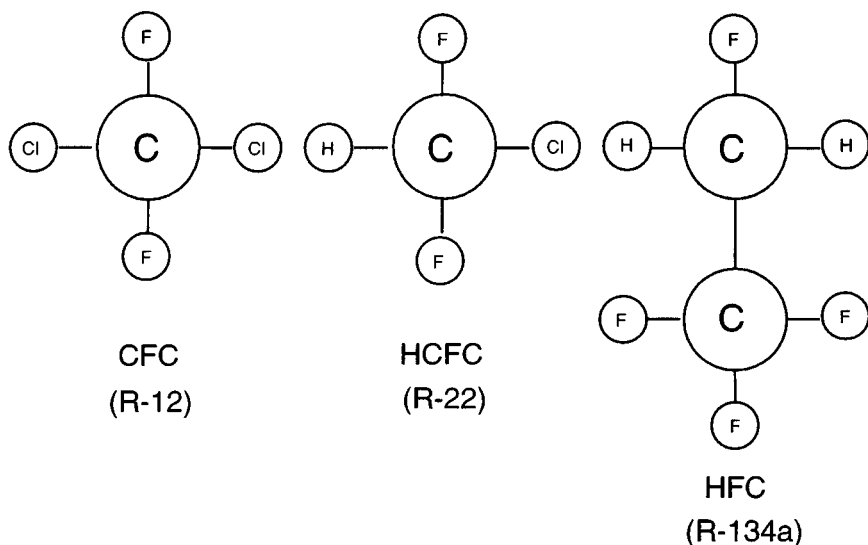


FIGURE 6.1.2 Typical molecular structures for the three major halocarbons.

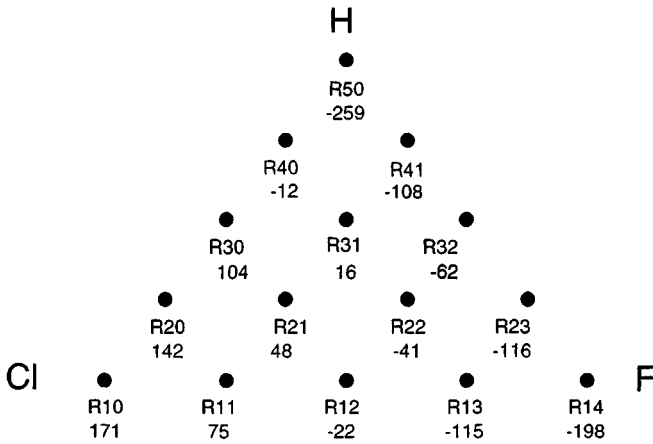


FIGURE 6.1.3 Methane-based refrigerants—molecular structure.

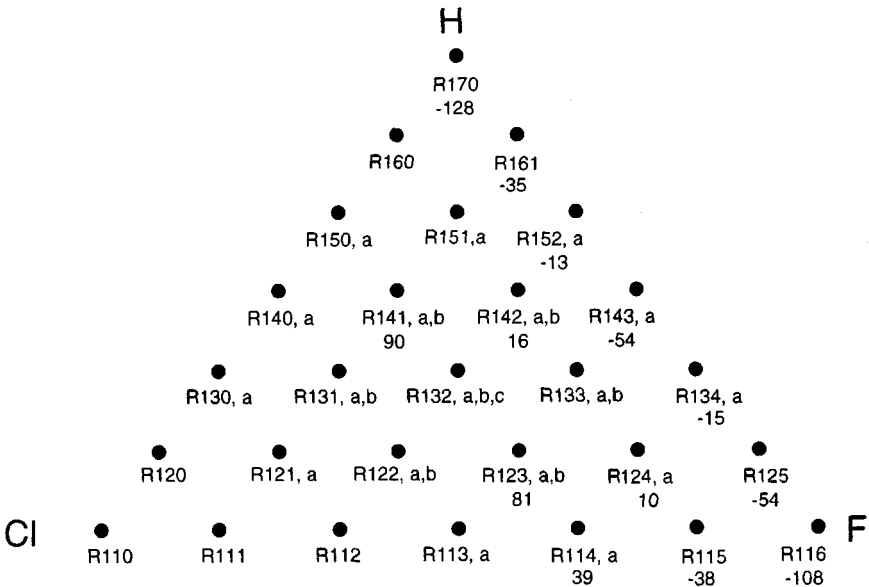


FIGURE 6.1.4 Ethane-based refrigerants—molecular structure.

The triangular representation also allows some generalizations about properties. Those molecules from the midpoint on up are flammable at atmospheric pressure, i.e. where half or fewer of the hydrogens have been replaced, the molecule will be flammable. The bottom row of molecules is fully halogenated and these molecules tend to have long atmospheric lifetimes which raises concerns from an environmental aspect. In fact, all the CFCs come from the bottom row. An observable impact of chlorine is that it raises the boiling point of the molecule. As one moves toward the lower left point of the triangles, the boiling point increases. Another property of interest is the toxicity of the chemicals. There is a less well defined, but observable trend toward greater toxicity as one moved to the left (higher in chlorine).

In the ethane series, for some combinations, it is possible to arrange the same atoms in different ways. These variations are isomers and are designated by a lower case suffix of a, b or c. R-134a is an isomer of R-134 and exhibits different properties. It is, therefore important to include the full designation when referring to the refrigerant by its number. For further reading on this area, the reader is directed to an excellent article "Quest for Alternatives"⁴.

6.1.3.2 Physical Composition

Most refrigerants are single component fluids, i.e. they consist of a single molecular species. It is possible, however, to mix two or more refrigerants. There are several reasons to consider blends of refrigerants. It is possible to modify the thermodynamic properties, e.g. change the boiling point of the refrigerant which affects capacity or change the discharge temperature to enhance reliability. In some cases, the flammability of an otherwise desirable refrigerant can be suppressed by the addition of a non-flammable component. Unfortunately it isn't possible to render a highly flammable refrigerant like propane nonflammable by the use of a minor additive. It is in fact, difficult to maintain non-flammability of any mixture which contains more than a minor amount of propane. Other reasons to consider mixtures would include improvement of lubricant miscibility and solubility or to improve leak detectability.

There are two types of mixtures which can be formed with refrigerants. The most likely type of mixture, if one were to arbitrarily select two or more refrigerants, would be a simple mixture or zeotrope. The characteristics of a zeotrope are as expected, i.e. the blend would tend to average the properties of the components. This characteristic allows great flexibility in tailoring the properties to achieve a desired result. The composition of the liquid and vapor phases would be different because the more volatile component(s) would more readily evaporate, enriching the vapor phase in the lower boiling (higher pressure) constituents. Another characteristic of zeotropic mixtures is commonly described as temperature glide. Glide occurs as a result of the segregation or fractionation which zeotropes exhibit.

To better understand temperature glide, consider a binary mixture of two arbitrary refrigerants "A" and "B" where "A" is the more volatile (lower boiling point). We will also assume that equal amounts of "A" and "B" are used. As evaporation commences, the initial vapor which boils off will have a higher percentage of "A" and a less of "B". It would not be unusual if the ratio were 70/30 rather than the 50/50 as originally formulated. The liquid which remained would then become enriched in "B" as the evaporation process continued. This shifting of the composition would also change the boiling point as the process continues. In evaporation the temperature increases as the mixture evaporates. In condensation, the

effect is reversed, i.e. the temperature decreases continually as condensation progresses. The generally accepted definition of temperature glide is the difference between the dew and the bubble points of the refrigerant at any given pressure. The dew point of a refrigerant is saturated vapor where the first drop of liquid forms in condensation (or conversely where the last drop of liquid evaporates). The bubble point is saturated liquid where the first bubble of vapor forms in boiling (evaporation). For single-component refrigerants, these points are at the same temperature at any given pressure.

There has been considerable interest in attempting to take advantage of the temperature glide by matching the refrigerant glide to the temperature differential of the source and sink fluids. This should in theory reduce the log mean temperature difference in a heat exchanger. Unfortunately, there is also a mass transfer resistance, with blends which offsets the benefits due to the temperature glide. In general, heat transfer is degraded with refrigerant blends and as a result, energy efficiency is lower than anticipated.

As mentioned above, there is another type of mixture for refrigerants. These mixtures are *azeotropes*. Azeotropes are unique and do not behave as mixtures but rather function almost exactly like a single-component refrigerant. The mixtures do not segregate and in fact are very difficult to separate by distillation. Another characteristic is that they always have a boiling point either higher or lower than either of the constituents. It is possible to form azeotropes with more than two components but these are rare. Azeotropes do not exhibit the temperature glides of zeotropic mixtures and the heat transfer characteristics are not degraded due to the mass transfer resistance. As a result azeotropes are favored by equipment system designers. In the past, CFC based azeotropes which have seen widespread acceptance in commercial use include R-500, R-502 and R-503. *New HFC azeotropes are now starting to be accepted in commercial use.* R-507 which was developed as a replacement for R-502 for frozen food applications is widely accepted in supermarket applications. It has also seen limited use as a replacement for HCFC-22 in chiller applications.⁴

An azeotropic refrigerant is only a true azeotrope at one temperature for a given composition. At other temperatures, the composition of the azeotrope will differ. From an engineering standpoint, these minor deviations from true azeotropy can be ignored, since the deviations are so small that they cannot be measured in systems. There are some zeotropes that are extremely close to azeotropes and in fact can be treated as an azeotrope for engineering purposes. An example of such a zeotrope is a 50/50 mixture of HFC-32 and HFC-125. These two compounds form an azeotrope at a 80/20 weight percent ratio. They also form azeotropes at other ratios slightly different from the 80/20. A mixture designated as R-410A consists of a 50/50 mixture. Although not a true azeotrope, this mixture is applied and handled like an azeotrope.

One way to illustrate the differences between azeotropes and zeotropic mixtures is by a phase diagram which plots either temperature or pressure as a function of the composition of a mixture. Fig. 6.1.5 and 6.1.6 illustrate typical phase diagrams.

6.1.4 REFRIGERATION SYSTEMS

For air conditioning applications of buildings, there are two types of refrigeration systems employed; *vapor compression systems* and *absorption systems*. By far, the greater majority are vapor compression, but for large water chillers, absorption

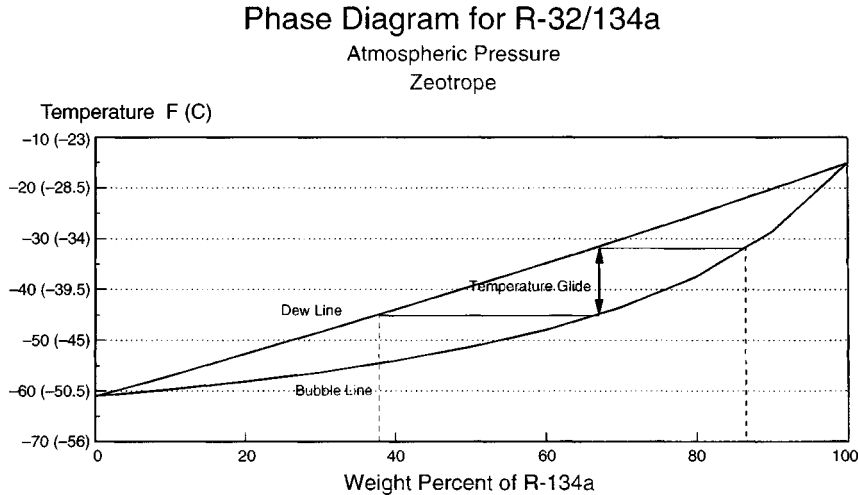


FIGURE 6.1.5 Phase diagram for R32/134a.

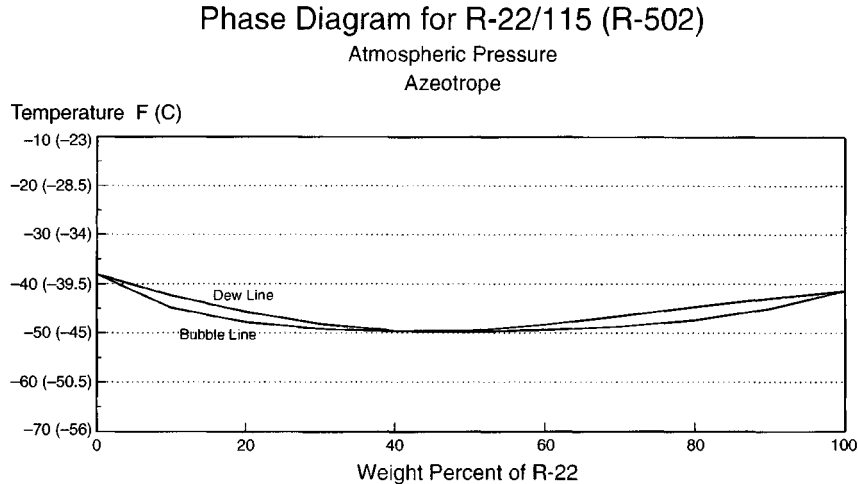


FIGURE 6.1.6 Phase diagram for R-22/115 (R502).

represents a significant portion. This is especially true outside the U.S. In Japan, absorption systems dominate the chiller market, however smaller unitary systems employ vapor compression systems almost exclusively. Almost all absorption chiller systems utilize water as the refrigerant with lithium bromide used as the absorbent. Smaller systems have used ammonia as the refrigerant with water being used as the absorbent. Both water and ammonia share some common advantages as refrigerants. These include availability, low cost and excellent heat transfer characteristics. Water has the disadvantage of high freezing and boiling points. The high boiling point results in a very low vapor pressure and therefore systems which

employ water as the refrigerant have to operate in a vacuum. Ammonia has the disadvantages of moderate flammability and a relatively high level of toxicity. It is also incompatible with copper which tends to offset the outstanding heat transfer properties of the fluid.

Vapor compression systems can employ a wide variety of refrigerants, although air conditioning systems have concentrated around a very limited number of refrigerants. Building systems have utilized both low and high pressure refrigerants. Low pressure refrigerants have been the exclusive province of centrifugal chillers. Until recently CFC-11 was the low pressure refrigerant of choice although a very limited quantity of systems used CFC-113 and CFC-114. The current low pressure refrigerant is HCFC-123. It offers performance comparable to CFC-11 but with greatly improved environmental properties. It does, however, contain chlorine and therefore is scheduled for phaseout eventually. HCFC-123 has a very short atmospheric lifetime and therefore has minimal environmental impact. As a result, its availability has been extended beyond that of other HCFCs such as R-22. Work is underway to identify a suitable low pressure HFC.

Higher pressure refrigerants can also be effectively utilized in centrifugal chillers. CFC-12 was widely used in the past but has been more recently supplanted by HFC-134a and on an interim basis HCFC-22. Positive displacement compressor systems have primarily used HCFC-22. These types of systems encompass a wide variety of equipment ranging from PTACs, using small rotary compressors with capacities down to 6000 Btu/h (1758 W), to large screw chillers with capacities exceeding 6,000,000 Btu/h (1,758,000 W). In between is a large group of unitary equipment and chillers using both reciprocating and scroll type compressors. The majority of this equipment continues to use HCFC-22, but it is recognized that ultimately HFC replacements will be required. Three major HFC alternatives have emerged to cover the range of applications. HFC-134a appears to have utility for some large screw and reciprocating chiller applications. R-407C is expected to see duty primarily as a service fluid to replace HCFC-22 in existing equipment and for a very limited number of new systems such as large rooftop units. The majority of smaller unitary systems but also some large chillers including screw compressor systems will utilize R-410A. Even though R-410A is a higher pressure refrigerant than R-22, it has been selected for the majority of applications because of its superior heat transfer characteristics. It is also more tolerant of pressure drop which minimizes these losses especially in split systems. Extensive testing has demonstrated higher energy efficiencies than R-22 with similar sized heat exchangers.

6.1.5 MATERIALS COMPATIBILITY

A large body of data has been developed over the years on CFCs and R-22. With the advent of many new refrigerants, much of the conventional wisdom is no longer relevant or correct. There has been a concentrated effort by many researchers to develop equivalent data for the newer alternatives such as the long term HFC and interim HCFC based refrigerants. The most comprehensive source of data is the ARTI Refrigerant Database.⁶

6.1.5.1 Lubricants

Virtually all vapor compression refrigeration systems require lubricants to permit reliable compressor operation. Historically, fluorocarbon based refrigerants, such

as CFCs used refined mineral oils which exhibited a high degree of miscibility and solubility with the refrigerant. Ammonia has historically also used mineral oils but was almost totally immiscible with the oils. The systems using CFCs took advantage of the miscibility/solubility characteristics to return the oil that was pumped into the system by the compressor. Most HCFC systems could also use mineral oil although miscibility was incomplete. Synthetic oils (alkylbenzenes) were developed for use where greater miscibility and solubility was required.

The advent of HFC refrigerants required a new class of lubricants since these refrigerants are almost totally immiscible with both mineral and alkylbenzene lubricants. Automobile air conditioning systems have gravitated to polyalkylene glycol lubricants (PAGs) while stationary systems generally use polyol esters (POEs) because of compatibility problems between hermetic motor materials and PAGs.

Lubricants for refrigeration systems are offered in several viscosity grades ranging from about 20 centistokes (cS) to 100 cS. In general, the grade is specified by the compressor manufacturer. With mineral oils, various brands were usually considered to be interchangeable (but not different viscosities). With the new synthetic PAG and POE lubricants, the viscosity grades may be different than in the past and different brands can no longer be considered interchangeable. Compressor manufacturers have expended considerable effort to evaluate the new lubricants and their recommendations should always be followed.

6.1.5.2 Plastics and Elastomers

The inherent chemical stability of fluorocarbons means that most plastics and elastomers are usable with them. One notable exception is the fluoropolymer class of elastomers. Many of these were compatible with CFC-12 but are not compatible with HFCs and also are not compatible with HCFCs such as R-22. It is important to note that within a particular class of materials, e.g. neoprene, there is a wide variation in formulations and it should not be assumed that all members of that class are suitable simply because most are.

HCFC-123 exhibits more aggressive solvency than most other refrigerants and therefore caution must be used before retrofitting an existing CFC-11 chiller. New materials have been developed which permit satisfactory use with HCFC-123.

The new lubricants, and particularly the POEs, are much better solvents than the mineral oils that they replace. They are also much better solvents than the refrigerants in most cases. *If compatibility problems develop, it is usually the lubricant rather than the refrigerant that is the culprit for the new alternative refrigerants.*

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CHAPTER 6.2

POSITIVE DISPLACEMENT COMPRESSORS/CHILLERS & CONDENSERS

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6.2.1 INTRODUCTION

Positive displacement compressors include reciprocating, screw and scroll compressors.

6.2.2 RECIPROCATING COMPRESSORS

A *reciprocating compressor* is a single-acting piston machine driven directly by a pin and connecting rod from its crankshaft. It is a positive-displacement compressor in which an increase in the pressure of the refrigerant gas is achieved by reducing the volume of the compression chamber through work applied to the mechanism.

Various combinations of piston size (bore), length of piston travel (stroke), number of cylinders, and shaft speed result in various compressor sizes, ranging from $\frac{1}{42}$ to 200 HP (0.06 to 149 kW).

The most commonly used reciprocating compressor is for refrigerants R-22 and R-134a. For heating, ventilating, and air conditioning (HVAC) and process cooling, the most practical refrigerant is R-134a; R-22 is the most practical refrigerant today. However, R-134a is gaining acceptance in view of the CFC regulations worldwide. As a matter of fact, the same countries only accept R-134a today. Other environmentally acceptable refrigerants are R-404A and R-507 for low and medium temperature applications; R-407C for medium temperatures and air conditioning applications. Recently, R-410A is gaining acceptance as an environmentally acceptable substitute for R-22 only for residential and small equipment. R-410A is not a drop-in refrigerant for R-22.

Reciprocating compressors are of three types: *open drive*, *hermetic*, *semi-hermetic*.

6.2.2.1 Open-Type Compressor Units

Open-type compressors are designed to have the drive shaft extend outside the crankcase through a mechanical seal. This seal prevents outward leakage of refrigerant and oil and inward leakage of air and moisture. Figure 6.2.1 shows a cut-away of a typical open-drive compressor.

The drive shaft is adaptable to an electric-motor or gas-engine drive. Electric-motor drives are either *belt-driven* or *directly coupled* to the compressor by means of a flexible coupling. Gas-engine drives are usually directly coupled.

Figure 6.2.2 shows a typical belt-driven unit. The drive consists of (1) a flywheel mounted on the compressor shaft and (2) a small pulley mounted on the motor shaft. These are interconnected by one or more V-belts. The size of the flywheel is usually fixed by the manufacturer, and the size of the motor pulley can be varied to achieve the desired speed. Speed variations are also obtained by ranging the size of the compressor flywheel and motor pulley.

The compressor is rigidly mounted on a steel base. The motor is mounted on an adjustable rail; this allows alignment of the motor pulley and tightening of the

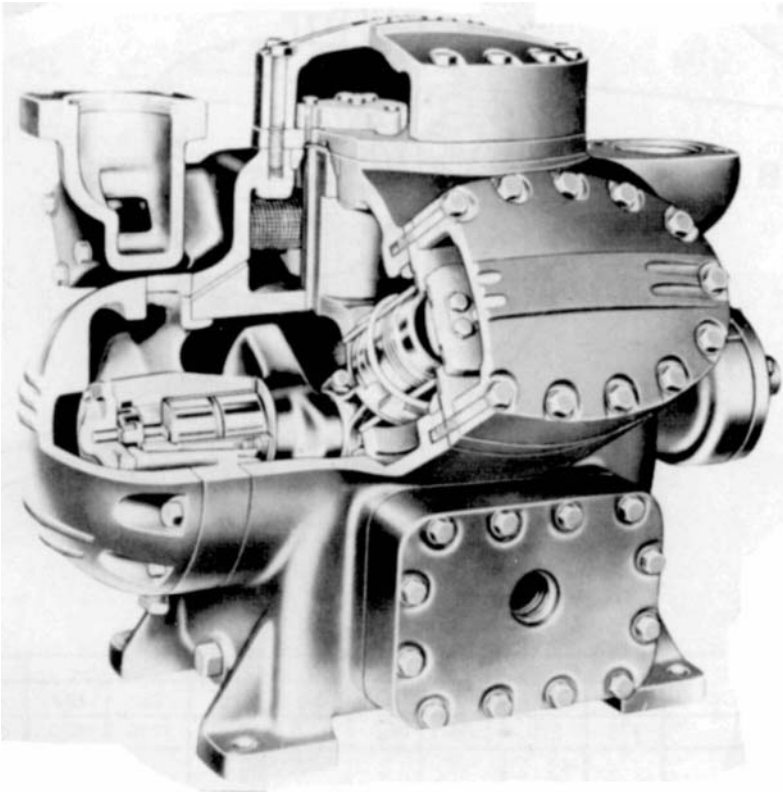


FIGURE 6.2.1 Open drive compressor. (Courtesy of Carrier Corporation.)

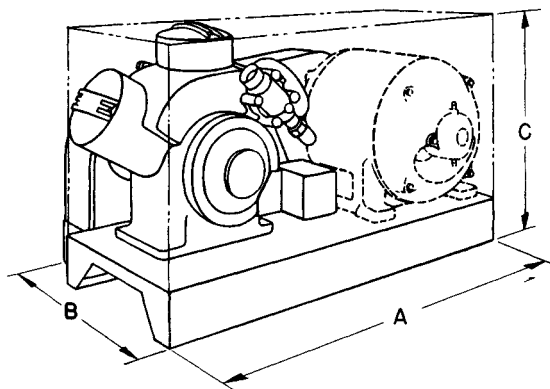


FIGURE 6.2.2 Belt driven compressor. (Courtesy of Carrier Corporation.)

belts. Proper belt alignment and belt tension are most important for efficient compressor operation.

Belt Alignment. Belt alignment (Fig. 6.2.3) can be checked as follows:

1. Line up the compressor flywheel and motor pulley with a straightedge. Slide the motor pulley on the shaft to correct any parallel misalignment. For correct angular alignment, loosen the motor hold-down bolts and turn the motor frame as required.

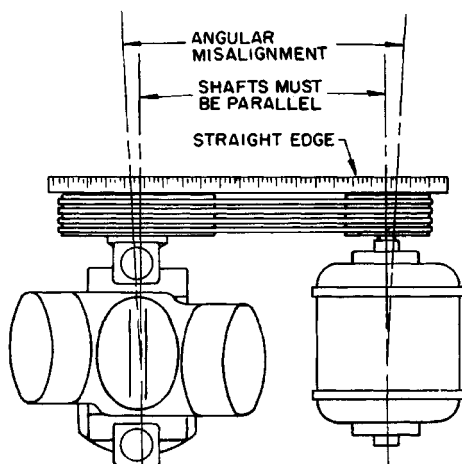


FIGURE 6.2.3 Correct belt alignment. (Courtesy of Carrier Corporation.)

2. When the alignment is completed, move the motor away from the compressor with the adjusting screws to tighten the belts. Tighten the belts just enough to prevent slippage.

Belt Tension. Belt tension can be checked by checking the amount of deflection as the belt is depressed at the center of the span. The rule of thumb is that belts deflect approximately 1 in for every 24-in span (1 cm for every 24-cm span). A longer span will deflect proportionately more.

Figure 6.2.4 shows a typical *direct drive* unit. The compressor shaft is connected directly to an electric motor through a flexible coupling and is designed to run at motor speed; this speed is 1750 rpm for a 60-Hz power supply and 1450 rpm for a 50-Hz power supply. Two-speed or variable-speed motors are sometimes used for closer capacity control, but for HVAC applications this is cost-prohibitive and not commonly used.

The compressor and motor are rigidly mounted on a steel base. Proper coupling alignment is essential for trouble-free operation. The maximum permissible angular or parallel misalignment for all couplings is 0.010 in (0.25 mm). The manufacturer's recommendations are necessary for alignment. Basically, there are two alignment methods employed:

1. The dial-indicator method (Fig. 6.2.5).
2. The straightedge-and-caliper method (Fig. 6.2.6).

6.2.2.2 Hermetic Compressors

Hermetic compressors are also known as “sealed” or “welded” compressors or as “cans,” since the motor and compressor are mounted within a common pressure vessel, sealed by welding. Figure 6.2.7 shows a cutaway of a typical hermetic compressor.

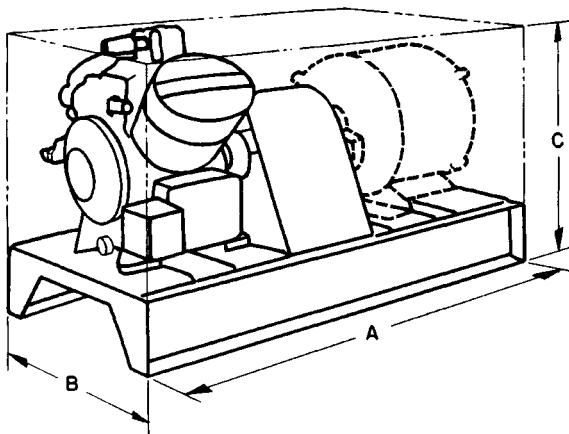


FIGURE 6.2.4 Direct-drive unit. (Courtesy of Carrier Corporation.)

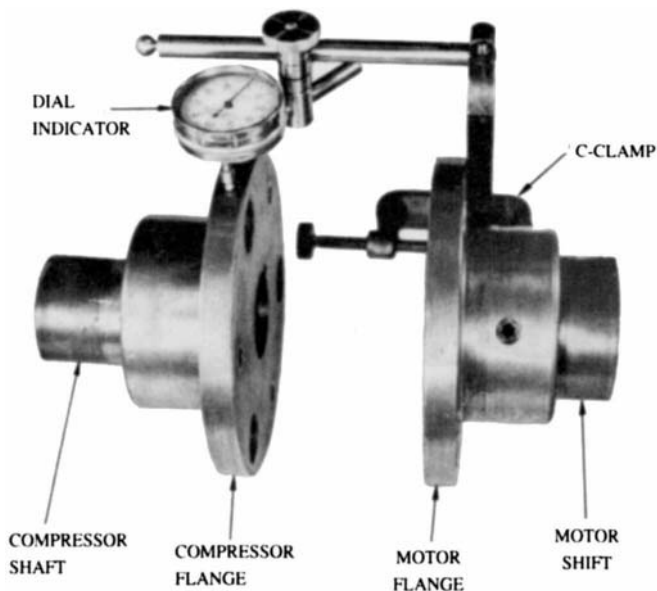


FIGURE 6.2.5 Alignment with a dial indicator. (Courtesy of Carrier Corporation.)

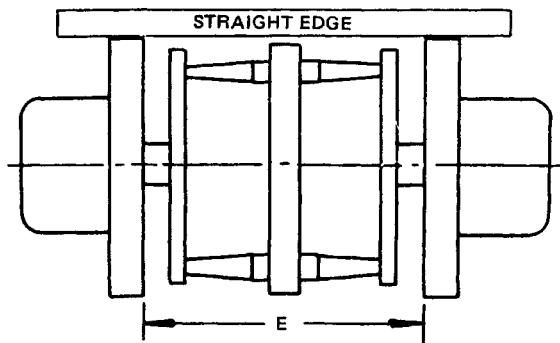


FIGURE 6.2.6 Alignment with a straight edge and calipers (Courtesy of Carrier Corporation.)

The compressor consists of pistons, connecting rods, bearings, and a lubrication system and is driven by a crankshaft that is common to both the compressor and the motor. The motor-starter windings are of refrigerant gas-cooled design. Most hermetic compressors are internally spring-mounted. Some have built-in suction accumulators for protection against liquid floodback. Lubrication of the compressor is usually achieved by a careful design that allows lubrication of the bearing and motor surfaces.

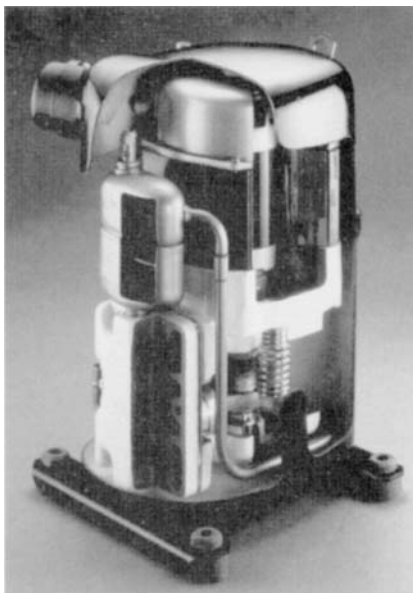


FIGURE 6.2.7 Hermetic compressor. (*Courtesy of Copeland Corporation.*)

Hermetic compressors are commercially available in sizes from $\frac{1}{12}$ to 24 hp (0.06 to 18 kW). Larger sizes are proprietary and are made by some manufacturers for use on their own packages.

Hermetic Motors and Motor Protection. Hermetic motors are specially designed by various manufacturers to keep the compressor losses to a minimum. This allows the compressor to operate effectively at maximum compression ratios.

Furthermore, these motors must have high dielectric strength, be resistant to abrasives, and be compatible with the refrigeration lubrication oil and refrigerant. Such other factors as insulation, efficiency, performance starting current, starting and breakdown torques, cost and availability are also important.

Although the suction-gas-cooled design feature allows hermetic motors to be of considerably small sizes, it also poses a problem in protecting the motors from quick overheating or from drawing excessive current (amps). The most common method of motor protection is to have thermal overload devices embedded in the windings. These mechanisms trip when overloading, overheating, or any other abnormal condition occurs.

6.2.2.3 Semihermetic Compressors

Semihermetic compressors are also known as “accessible” hermetic compressors. The motor and compressor are mounted within a common pressure vessel and sealed by bolted plates so that the motor and compressor parts are accessible. In

case of a compressor failure, these parts can easily be repaired or replaced, and the rebuilt compressor can be bolted back together. Except for this accessibility, all other features in the semihermetic design remain similar to the hermetic design: the compressor consists of pistons, connecting rods, and lubrication-system bearings, driven by a crankshaft common to both compressor and motor.

Semihermetic compressors are available in the following designs for compressor-motor cooling:

- **Air-cooled design.** This design uses air circulation for proper cooling of the compressor motor. A constant air flow is required across the compressor housing, and this is accomplished by direct impingement of air from fan discharge. Typically, air-cooled compressors are limited in size to 1/4 to 3 hp (0.19 to 2.24 kW).
- **Water-cooled design.** In this design, water coils are wrapped around the compressor housing; compressor-motor cooling is provided by circulating water. Typically, this design is limited to water-cooled condensing units and is not practical in today's HVAC market.
- **Refrigerant-cooled design.** This design is the one most commonly used in HVAC applications. As in hermetic compressors, the motor is cooled by return suction gas. Refrigerant-cooled compressors are available in sizes from 2 to 10 hp (1.5 to 75 kW).

6.2.3 SCREW COMPRESSORS

A complete discussion of screw compressors is included in Chapter 6.4 "Screw Compressors."

6.2.3.1 Screw Vs. Reciprocal & Centrifugal Compressors

Relative advantages of each:

- **Screw vs. Reciprocal:**
 1. Screw compressors operate more or less like pumps and have continuous flow of refrigerant compared to reciprocals, which have pulsations. This results in smooth compression with little vibration. Reciprocals, on the other hand, make pulsating sounds and vibrate.
 2. Screw compressors have almost linear capacity-control mechanisms resulting in excellent part-load performance.
 3. Due to its smooth operation, low vibration screw compressors tend to have a longer life than reciprocals.
- **Screw vs. Centrifugal:**
 1. Centrifugals are constant-speed machines. These machines "surge" at certain operating conditions, resulting in poor performance and high power consumption at part-load.
 2. Screw compressors have proven themselves in tough refrigeration applications, including on-board ships. Today, screw compressors practically dominate refrigerated ships, transporting fruits, vegetables and meats and

other frozen foods across the ocean with good reliability. These compressors have replaced the traditional shipboard centrifugals.

6.2.4 SCROLL COMPRESSORS

In 1886 an Italian patent was issued for the basic scroll concept. The first American patent was issued in 1905 and very little was done with the idea until the 1960's and the 1970's when scroll development work was undertaken in Germany and France. The scroll idea was tried for various applications such as vacuum pumps and expansion engines.

Although the scroll concept is rather old, it has only been in the last few years that machine tools have been developed to the point where components could be machined to the minimum clearances that are noted to produce high efficiency scrolls.

Scroll technology is based on two scrolls; the first scroll is *fixed*, while the second scroll *orbits* around the fixed scroll. These scrolls are intermeshed and form a crescent-shaped space between them. When the second scroll orbits around the fixed spiral, the suction gas in the space is compressed until the gas reaches the maximum pressure in the center of the scrolls. This compressed gas is then discharged through a port in the center of the compression chamber.

Due to the intricate design of the scroll compressor, the gas is discharged smoothly, almost like a pump. This smooth compression reduces vibration (compared to reciprocating action) which is of a pulsating nature. Other features are as follows:

- fewer moving parts than reciprocating compressors.
- less rotating mass and less internal friction.
- smooth compression cycle with low torque.
- low noise levels.
- low vibrations.

Motors for scroll compressors are suction gas-cooled where the suction gas cools the motor, achieving high efficiency and long life. The motor is protected by an external protector which senses excessive current, disconnecting it before overheating.

6.2.4.1 Lubrication

Lubrication to the scroll journal and shaft bearings is achieved by a centrifugal pump submerged in the oil sump. Oil is moved upwards through the passage to lubricate the upper and lower shaft bearing through parts in the shaft wall; it then leaves the upper end of the passage to lubricate the orbiting scroll journal bearing.

Lubrication for the scroll contacting surfaces is provided by a small amount of oil entrained within the suction gas stream.

6.2.4.2 Design Types—Compliant or Non-compliant

In the *compliant* design, the surfaces of the scroll plates are allowed to touch lightly, thereby providing high efficiency and reliability. In the *non-compliant* design, the scrolls do not touch, maintaining a slight clearance.

6.2.5 POSITIVE DISPLACEMENT LIQUID CHILLER SYSTEMS

A liquid chiller system cools water, glycol, brine, alcohol, acids, chemicals, or other fluids. The most common use of a chiller system is as a water chiller for human-comfort cooling application. The chilled water generated by the chiller system is circulated through the cooling coil of a fan coil (or air-handling unit), as shown in Fig. 6.2.8.

The fan coil circulates air within the conditioned space. Air from the room moves over the chilled-water cooling coil of the fan coil, is cooled and dehumidified, and returns to the room. In this process the chilled water in the cooling coil picks up the heat and is returned to the chiller system for cooling. As the cycle is repeated, the chiller system maintains the conditioned space at comfort level.

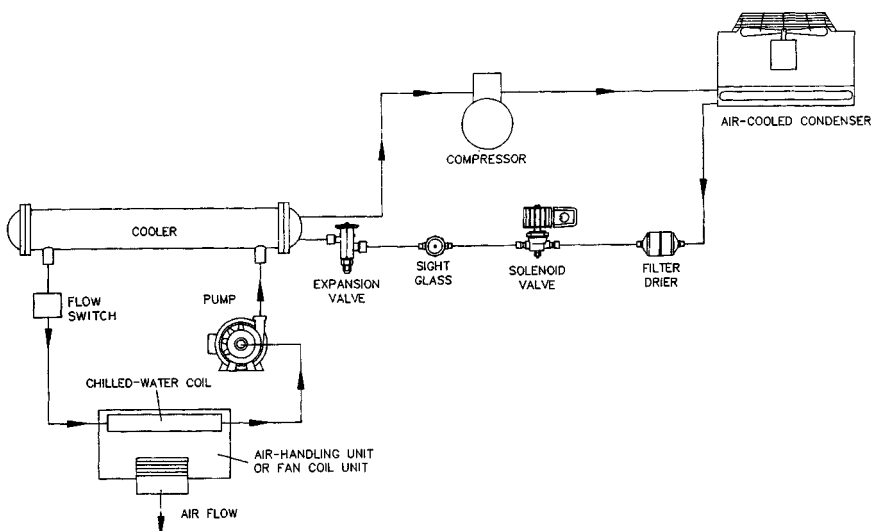


FIGURE 6.2.8 Typical liquid chiller system. (Courtesy of Continental Products, Inc.)

Field-Assembled Liquid Chillers. Originally, liquid chillers were “field-assembled,” with the components “field-matched” to develop a field-erected system. These systems were custom-built to perform a specific application. As a result the design depended on the application, the availability of parts, the labor, and the field engineer. Some systems were well thought out, were carried out with detail and care, and performed very well for the particular application. Others were ill-conceived, had a poor choice of components, and resulted in a bad experience for the owners.

Field-assembled systems could not be pretested to check if they would perform properly. They depended entirely on the design concepts, the availability of matched parts, and the field experience of the labor force.

As the cost of field labor became prohibitive and as owners had poor experiences from field-erected systems, the concept of *factory packaging* became popular.

Factory-Packaged Liquid Chillers. The idea of a completely pre-engineered system is to assemble all the components on a common steel skid and to pipe, wire, pressure-test, evacuate, and charge the system with refrigerant (usually R-22). In this manner, all the system’s components are preselected, heat-transfer balanced with each other, prepped, prewired, and factory run-tested before actually being installed on the job. Factory-packaged systems, if manufactured to good engineering standards and correctly capacity-rated, are very cost-effective, resulting in years of good service to the owners.

There have been many improvements from early factory packages to today’s systems. Today it is reasonable to expect a reliable, fully factory-tested packaged liquid chiller system that has the following: compressor(s); condenser(s); a liquid chiller; refrigeration specialties, such as expansion valve(s), filter dryer(s), sight glass(es), and solenoid valve(s); and electrical components with power and safety controls. Chiller and condenser pumps (for water-cooled systems) and factory-mounted and -wired flow switches are also available. Most manufacturers test-run the system before giving it its final, preshipment paint finish or cleanup. Field piping of water and additional electrical components are all that are needed before the system is ready for startup.

6.2.5.1 Packaged Liquid Chiller Systems

Packaged chillers are available with the following choices:

1. Scroll compressor chillers from 3 to 240 HP (2.2 to 180 kw) with multiple scrolls.
2. Screw compressor chillers from 30 through 440 HP (22 to 328 kw).
3. Hermetic/semi-hermetic reciprocal chillers from 3 through 440 HP (2.2 to 328 kw).
4. Open drive reciprocal or screw compressors for industrial applications.

Major Components. A typical liquid chiller system essentially consists of four components:

compressor(s).

condenser(s)—air cooled, water cooled, or evaporative cooled.

expansion valve(s).

evaporator cooler(s)—direct expansion or flooded type.

Other essential parts of the total system are the refrigerant charging valve, filter dryer, liquid solenoid valve, sight glass and moisture indicator, expansion valve, and electrical control center.

Electric Control Center. The control center is an essential part of the total system, it includes power, operating, and safety controls, usually mounted in a common control panel.

The power controls are separated from the operating and safety controls by a divider plate or other means. The power controls include a starting contractor (in the case of hermetic and semi-hermetic compressors, which have internally protected compressor motors) or NEMA-rated starters with overload protection (in the case of open-drive compressors, which use NEMA-rated electric motors).

The operating control includes a chiller thermostat, which senses the incoming water temperature to the cooler.

The safety controls consist of the following:

1. *High-condenser-pressure switch.* This opens if the compressor discharge pressure reaches a preset value. It is usually of a manual-reset type; i.e., the operator will have to reset the control manually to restart the system. Manual reset control is used to give the operator a chance to discover the cause of high condenser pressure.
2. *Low-refrigerant-pressure switch.* This opens when the evaporator cooler's pressure reaches a minimum safe limit preset by the manufacturer. This switch may be of a manual- or automatic-reset type; since it is also used for the pumpdown cycle, is usually of an automatic-reset type.
3. *Oil-pressure control.* This switch, usually a manual-reset type, is provided to shut down the compressor when the compressor's lubrication oil pressure drops below a minimum safe value as determined by the compressor manufacturer or when in sufficient oil pressure is developed after compressor startup. This switch is used only on semi-hermetic and open-drive compressors, not on hermetic types. (Hermetic compressors usually have no means to sense the oil pressure, but rely on internal means of lubrication.)
4. *Freeze-protection switch.* This switch (the so-called "low chilled-liquid temperature control") operates similarly to a thermostat, sensing the temperature of the chilled liquid leaving the cooler. In case of a freeze-up condition and of liquid leaving the cooler, this controller opens the circuit to stop the total system. The minimum value is preset at the factory to prevent cooler freeze-up in case other safety controls malfunction.
5. *Low-pressure freezestat.* This control is usually of a manual-reset type and has a 60 s built-in time delay. It senses the evaporator cooler's pressure, and if this pressure continues to drop below the preset value for a period of 60 s, the switch opens to shut down the total system. This acts as a protection against freeze-up as well as against a loss of refrigerant.
6. *Flow switch.* This control, which can be either factory-mounted or field-furnished, is needed in the chilled-water piping to protect against a cooler freeze-up in the event of no liquid flow through the system. This device is an

interlock type and provides essential protection against pump failure or other malfunctioning of the total system.

7. *Motor overload protection.* This is provided with hermetic and semi-hermetic compressors as a built-in feature, whereas for open-drive compressors the overload heaters are sized to protect the motor.
8. *Power-factor corrector capacitors.* These are for improved motor performance, reduced line losses, and lower utility costs.
9. *Indicator lights.* Various indicator lights show the system's operation and, in case of a system failure, provide diagnostics for ease of service.
10. *Pressure gauges.* High- and low-pressure and oil-pressure gauges are factory-installed and piped to the compressor.
11. *Compressor cycle meter, ammeter, and unit disconnect switches.*

Several other safety controls are available, as follows:

1. *Lock-out timer.* This control prevents the compressor from short-cycling on power interruptions to safety controls.
2. *Phase failure.* This control relay monitors the sequential loss and reverse of a three-phase power supply.
3. *Alarm-bell contacts.* These allow alarm-bell connections to the high-pressure control or to other safety controls for signaling if the unit fails on manual-reset safety controls.
4. *Low ambient controls.* These controls are used specially for air-cooled condenser chillers or evaporative-cooled chillers. The low ambient controls may consist of fan-cycling pressure controls or fan speed controls. The fan speed control usually has a solid-state controller and a single-phase condenser fan motor (which modulates the fan speed in response to condenser pressure). The low ambient controls allow operation of the chiller system on days of low ambient temperature.
5. *Relieve valve.* The pressure-relief valve is set at a pressure above the high-pressure cutout to relieve the system before the system reaches its maximum design working pressure. These valves should be piped and vented outside. Fusible plugs or rupture disks can be used in some instances.
6. *High-motor-temperature protection.* This control consists of a high-temperature thermostat or thermistor. It is located in the discharge gas stream of the compressor.
7. *High oil temperature.* This controller protects the compressor when there is a loss of oil cooling or when a bearing failure results in excessive heat generation.

Other Components. Factory-mounted, -piped, and -wired pumps for chilled water and condenser water are becoming available as a part of the packaged chiller system. This has eliminated the need for field labor for plumbing, wiring, and interlocking the pumps with the chiller control panels. Factory-mounted pumps are checked for pump rotation, which is phased in with an air-cooled condenser motor to ensure that the condenser fans operate vertically *up* and that the pumps operate in the correct direction.

Other accessories, such as filters, air eliminators, and storage tanks, can also be factory-mounted and -piped, eliminating the need for separate mechanical areas and the chance of incompatible field components.

6.2.5.2 Typical Chiller Refrigeration Cycle

In a typical chiller system (Fig. 6.2.8), as the water or other liquid flows through the system, the flow switch contact is made, and if the thermostat calls for cooling and all safety devices are closed, the compressor will start. The hot gas from the compressor is discharged into the condenser (air-cooled, water-cooled, or evaporative-cooled). As it travels through the condenser, this high-pressure refrigerant cools and changes its phase to high-pressure liquid. In the case of an air-cooled condenser, the condenser rejects the heat to the air; in the case of a water-cooled or evaporative-cooled condenser, the condenser rejects the heat to the water. The high-pressure liquid refrigerant now goes through a filter dryer. Then it goes through the liquid solenoid valve (which should be open now), sight glass, and moisture indicator and into the expansion valve. The expansion valve meters the liquid refrigerant through the evaporator cooler. The cooler allows the water (or other liquid) to be cooled by the action of the evaporating liquid refrigerant. The refrigerant picks up the heat from the flowing liquid, is returned back to the suction side of the compressor as a low-pressure gas, and is then ready to be recycled again through the compressor.

6.2.5.3 Chiller-System Freeze Protection

If there is any danger of freezing in a closed-loop chilled-water system, it is recommended that the system be charged with a premixed industrial-grade heat-transfer fluid. *Automotive antifreeze or other commercial glycols are not recommended*; these may include silicates that can coat the cooler tubes, fouling the system prematurely and shortening the life of the pump seals.

For more details about heat-transfer fluids, consult your local industrial chemical supplier.

Fouling Factor. Fouling results from scaling, corrosion, sediment, and biological growth (slime, algae, etc.); most water supplies contain dissolved or suspended materials that cause these problems. Such fouling causes thermal heat transfer to the water side of chiller systems, the measure for resistance to this heat transfer is called the *fouling factor*.

A general practice is to allow a fouling factor of $0.00025 \text{ (h} \cdot \text{ft}^2 \cdot ^\circ\text{F)/Btu [(m}^2 \cdot ^\circ\text{C)/W]}$ for coolers and water-cooled condensers. For seawater, or where the cooling water is untreated, a fouling factor of $0.001 \text{ (h} \cdot \text{ft}^2 \cdot ^\circ\text{F)/Btu [(m}^2 \cdot ^\circ\text{C)/W]}$ is recommended. In this case, the use of 304 or 316 stainless steel, 90/10 cupronickel, 70/30 cupronickel, or admiralty brass tubes may be considered; the condenser heads can be made of brass or can be treated with epoxy or other protective coating.

6.2.5.4 Types of Refrigerant

R-22 and R-134a are the most popular refrigerants for reciprocating liquid chillers.

6.2.5.5 Chiller Ratings

Capacity Rating Standards. Most manufacturers rate their packaged chillers according to Air Conditioning and Refrigeration Institute (ARI) Standard 590 (see Table 6.2.1). ARI Standard 590 is based on the following:

TABLE 6.2.1 Typical Air-Cooled Chiller Ratings

Models MBA use a single compressor, models DBA are dual-circuit with a dual compressor, and models FBA have four compressors with four independent circuits.

Model	Tons	kW	EER	Model	Tons	kW	EER
MBA 3	2.6	3.6	8.8	DBA 52	42.4	43.8	11.6
4	3.9	5.5	9.3	62	52.4	55.0	11.4
5	4.6	6.0	9.3	70	62.3	66.2	11.3
7	6.6	8.0	9.8	75	67.7	73.0	11.1
9	8.6	9.2	11.2	80	73.2	79.8	11.0
10	10.1	10.8	11.3	90	79.7	85.4	11.2
15	13.6	14.9	11.0	100	86.3	91.0	11.4
20	16.5	16.6	11.9	110	92.8	102.5	10.9
25	21.2	21.9	11.6	120	101.0	115.8	10.5
30	24.2	26.1	11.1	FBA 130	114.1	121.9	11.2
35	31.2	33.1	11.3	140	128.3	136.4	11.3
40	36.6	39.9	11.0	160	146.3	159.6	11.0
50	43.1	45.5	11.4	180	159.4	170.8	11.2
60	49.7	57.0	10.5	200	177.8	187.5	11.4
				240	198.8	228.0	10.5

- air-cooled package at an ambient-air temperature of 95°F (34°C).
- water-cooled package at a condenser-entering water temperature of 85°F (30°C) and at a condenser-leaving water temperature of 95°F (35°C).
- evaporative-cooled package at a dry-bulb temperature of 95°F (35°C) or at a wet-bulb temperature of 75°F (24°C).
- cooler water for all types at an entering temperature of 54°F (12°C) and at a leaving temperature of 44°F (7°C).
- fouling factor for both the cooler and the condenser = 0.00025 (h · ft² · °F)/Btu [(m² · °C)/W].

ASHRAE Standard 30-77 is used for testing reciprocating liquid chillers for rating verification.

Energy Efficiency Ratio (EER)

$$\text{EER} = \frac{\text{Btu/h output}}{\text{watts input}}$$

Typically:

- The EER for air-cooled packages ranges from 8 to 12.
- The EER for water-cooled packages ranges from 9 to 13.
- The EER for evaporative-cooled packages ranges from 10 to 16.

6.2.5.6 Chiller Selection Guidelines

To select a packaged chiller from the manufacturer's rating table, it is necessary to know at least four of the following five items:

1. capacity in tons or Btu/h (kW).
2. fluid flow rate in gal/min (L/min).
3. entering fluid temperature in °F (°C).
4. leaving fluid temperature in °F (°C).
5. type of fluid (water or other).

Use the following formula to calculate the fifth variable if only four are known:

$$\text{Tons} = \frac{\text{gal/min} \times \Delta \times \text{cp} \times \text{SG}}{24}$$

where Δ = (entering fluid temperature, °F) – (leaving fluid temperature, °F)

cp = specific heat of fluid, Btu/lb

SG = specific gravity of fluid

6.2.5.7 Types of Chillers

Heat-Recovery Chillers. Any HVAC or process application that has a simultaneous use for chilled and hot water is a potential heat-recovery installation. Typical applications are: buildings that need cooling on one side and heating on another; computer room cooling and reheating; restaurants; hotels; and hospitals.

Heat-recovery chillers extract heat from superheated gas vapor before it condenses in the condenser. Thus heat recovery offers “free heat” and eliminates, in certain instances, the need for separate heating equipment.

It is important that a heat-recovery heat exchanger not be oversized; otherwise, the advantage of high-temperature heat recovery is lost. A pressure-enthalpy diagram, as shown in Fig. 6.2.9, demonstrates the potential heat recovery for a typical chiller system.

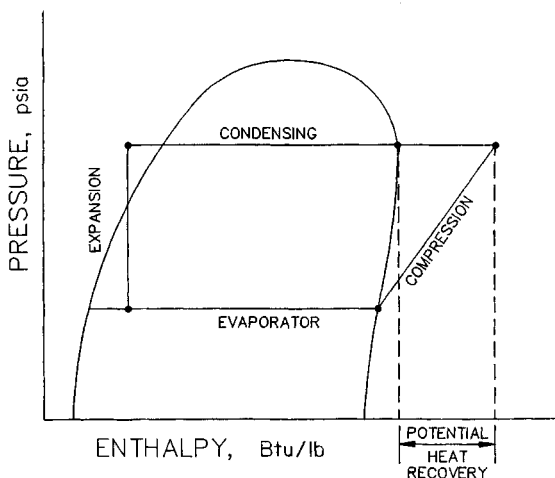


FIGURE 6.2.9 Pressure/enthalpy diagram (Courtesy of Continental Products, Inc.)

Figure 6.2.10 shows a schematic of a typical heat-recovery chiller. For air-cooled chillers, a heat-recovery heat exchanger can be piped in series, as shown. For water-cooled systems, heat recovery can be in series or in parallel.

Factory-packaged heat-recovery chillers are available in sizes from 3 to 200 hp (2.24 to 150 kW).

Heat-Pump Chillers. Heat-pump chillers are becoming more and more popular because of the following advantages:

1. They eliminate the use of a separate boiler or heating system.
2. They eliminate redundant piping for heating and cooling.
3. They use a two-pipe system but provide the comfort of a three-pipe system.
4. They use the same air handlers (or fan coils) for cooling and heating.
5. They use the same chilled-water pump for cooling and heating.

Heat-pump chillers utilize the same heat exchanger for cooling water as they do for heating water. The principle of operation is that, during heating, a reversing valve directs the flow of the hot-gas refrigerant from the compressor to the water heat exchanger instead of to the condenser. The heat exchanger is now being used as a condenser, and the condenser is being used as an evaporator. The gas is returned back to the compressor, through the reversing valve, with a common suction connection.

During heating, the same pump is used that circulates the water during cooling. This eliminates the need for separate hot-water and chilled-water pumps and piping. For the summer season, the same valve is reversed back to normal cooling.

Figure 6.2.11 shows a heat-pump chiller schematic. *Air-cooled* heat pumps use outside ambient air as the medium; therefore, they need wider fin spacing as well as hot-gas defrosting. *Water-cooled* heat pumps can use groundwater, river water, or wastewater as the medium.

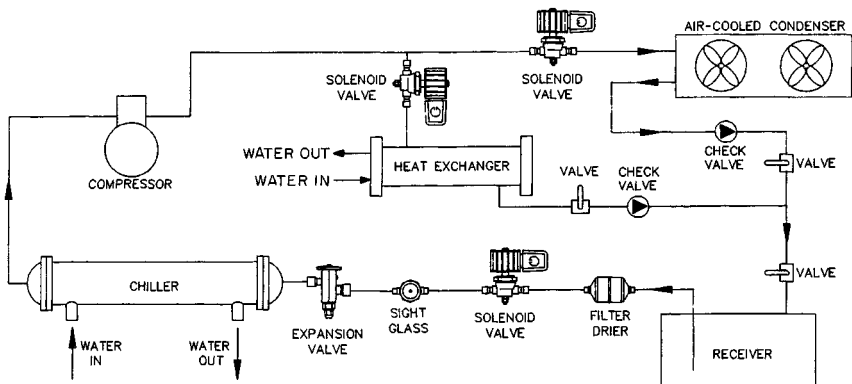


FIGURE 6.2.10 Typical heat recovery chiller (Courtesy of Continental Products, Inc.)

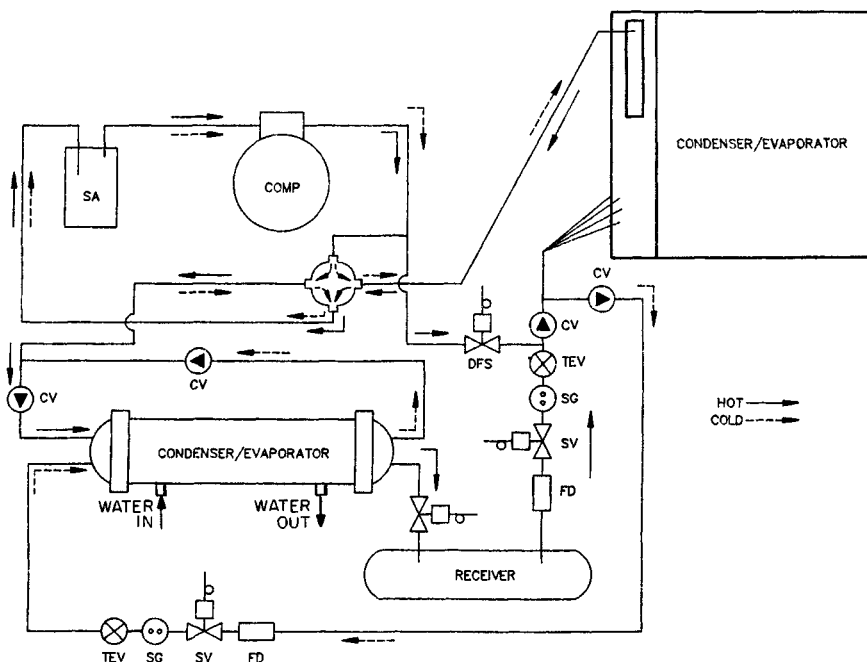


FIGURE 6.2.11 Heat pump chiller-schematic (Courtesy of Copeland Corporation)

Available in sizes of 5 to 200 hp (3.7 to 150 kW), heat-pump chillers are suitable for most locations with a winter-design dry-bulb temperature of 20°F (−7°C). They are also available with auxiliary electric heaters, which are useful for unexpected cold spells or as a backup.

Low-Temperature Glycol, Brine, Alcohol, and Gas Chillers. For low-temperature cooling with glycol, brine, alcohol, gases, or other fluids, several special features are necessary. Factory-packaged chillers for these applications are available. Field modifications of an HVAC chiller do not always produce the desired results. Some of the considerations for process chillers are:

1. Type of refrigerant. R-22 is recommended.
2. Correct sizing of expansion valve.
3. Temperature controller for low temperature.
4. Low-pressure switch.
5. Low-temperature cutout.
6. Oil separator(s).
7. Suction accumulators.
8. Dual-compressor system with common dual-circuit chiller for 50 percent redundancy.
9. Dual compressor, dual condenser, dual cooler, and dual electrical components for 100 percent redundancy.

10. Primary chiller, with secondary heat exchanger, for corrosives, chemicals, food products (e.g., wines and fruit juices), gas cooling and condensation, incineration, and environmental protection.

Packaged Process Chillers. Packaged chillers used for HVAC are designed primarily for human-comfort conditions. A typical system is designed for chilled-water flow to produce a temperature difference of 10°F (5°C), cooling from 54 to 44°F (12 to 7°C) or from 52 to 42°F (11 to 6°C). For process cooling, it is not always possible.

1. To maintain a 10°F (5°C) temperature difference.
2. To work between a 40 and 50°F (4 and 10°C) temperature range.
3. To keep a steady load.
4. To use ambient-related controls. The load may be constant year-round.
5. To use single-compressor systems.
6. To have a high return-water temperature.
7. To use standard electrical components, such as in explosion-proof atmospheres.
8. To use standard construction (steel or copper coolers or condenser) or copper or aluminum air-cooled condensers.

Typical applications for process chillers include the following:

Acid cooling	Machine-oil cooling
Bakeries	Marine systems
Breweries	Milk cooling
Candy and fruit glazing	Mushroom cooling
Chemicals and petrochemicals	Pharmaceuticals
Chicken and fish hatcheries	Photo labs
Computer and clean-room cooling	Plastics, injection and blow molding
Dough mixers	Plating and meal finishing
Electronic-cabinet cooling	Printing plants
Environmental test chambers	Pulp and paper
Explosion-proof chillers	Shrimp freezing
Flight simulators	Soil freezing
Foundries	Solvent recovery
Fruit-juice cooling	Steel mills
Ice rinks	Textile plants
Laser cooling	Welding
Lobster tanks	Wineries

6.2.6 CONDENSERS

Condensers are heat exchangers designed to condense the high-pressure, high-temperature refrigerant discharged by the compressor. In this process, the condensers reject the heat that was picked up by the evaporator cooler or chiller. At the same time, the condensers convert the high-pressure, high-temperature gas into high-pressure, high-temperature liquid refrigerant, ready to be recycled through the expansion valve, evaporator, and back to the compressor.

There are three types of condensers: air-cooled, water-cooled, and evaporative-cooled. They are discussed in Secs. 40.2.5, 40.2.6, and 40.2.7, respectively.

Condenser-Coil Circuiting. For multiple compressors, multiple-circuit condensers are used. For example, a two-, three-, or four-compressor unit has a two-, three-, or four-circuit condenser, respectively. In this manner, each circuit is independent of the other, thus providing redundancy in case of failure. Independent refrigerant circuits also ensure that refrigerant and lubrication oil for each section of the compressor system are not mixed. Typical two- and four-circuit condenser headers are shown in Fig. 6.2.12.

Parallel piping of two, three, or four compressors into a common hot-gas inlet connection, with a common liquid outlet connection, is also being utilized.

Compressors are cycled on and off for capacity reduction, but the condenser-coil surface remains the same. For example, if the condenser coil is sized for three compressors and one out of three compressors is shut down, the refrigerant gas from the other two compressors will continue to pump into the same coil. Now this coil will be oversized for the amount of hot gas being pumped into it. Although the condenser fans will also be cycled to reduce the air flow, the net effect is still a larger condenser surface, resulting in subcooled liquid. A more efficient system is to feed the subcooled liquid through the expansion valve.

For parallel piping, the following need and drawback should be considered:

1. It is essential that there be some means to interconnect multiple compressors in order to maintain lubrication oil in each compressor; otherwise, one compressor may be starved of lubrication and thus become damaged.
2. Since multiple compressors have a common refrigerant circuit, even if one (hermetic or semihermetic) compressor fails or “burns out,” it contaminates the complete system. To replace one compressor, the complete system has to be “cleaned,” evacuated before refrigerant is charged, and put back into service. Condensers with independent refrigerant circuits do not have this problem.

Condenser Components

1. **Fans.** Condenser fans are of a propeller type and are statically and dynamically balanced for low-vibration operation. Propeller fans are made of aluminum, galvanized steel, stainless steel, or plastic materials and range in diameter from 18 to 30 in (46 to 76 cm). Direct-driven fans are mounted on the fan motor shaft, and belt-driven fans have belt-and-pulley combinations.

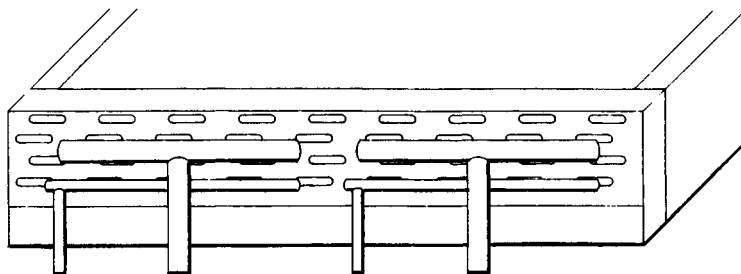


FIGURE 6.2.12 Two- and four-circuit condenser headers (typical).

For belt-driven condensers, larger-diameter fans are utilized and have the advantage of fewer fans, compared to several direct-driven smaller fans. Lower speeds (400 to 700 rpm) are achieved by belt drives. The motors are typically standard open drip-proof or totally enclosed NEMA-rated, four-pole, 1800 rpm. These are readily available.

Belt-driven chiller packages are more suitable for high to medium ambient conditions where an on-off cycling of fans is not needed. Other means of low ambient control are utilized for medium to low ambient control conditions.

If belt drives are used and the fans cycle often, the belt tension needs to be checked more often than usual. Typically, an access door is provided to access the motor bearings and belts. For additional convenience, extended lubrication lines are installed with external grease fittings.

2. Motors. Typical direct-driven fan motors are six-pole and operate at 1100-rpm speeds. These motors range in size from $\frac{1}{3}$ to 2 hp (0.25 to 1.5 kW) and have been specially designed for air-cooled condenser applications by various motor manufacturers. A typical condenser fan motor is of a 56-frame "totally enclosed air over" design with a built-in overload protector.

3. Motor speed control for low ambient operation. For multiple fans on a medium- to large-size chiller package, the fan motor can be cycled on and off by sensing condenser pressure or ambient temperature. This is adequate for medium ambient temperature operation. For lower ambients or a single-fan chiller package [3 to 9 hp (2.24 to 6.7 kW)], another choice is to modulate the fan motor speed. Typical fan-speed controllers sense condenser pressure or liquid temperature and modulate the motor in response to a rise or fall of pressure or temperature. At higher pressure or temperature, fans operate at higher speed; at lower pressure or temperature, fans modulate at lower speeds. All condenser fan motors are not suitable for fan speed control. Typically, a ball-bearing-type motor is needed to allow operation at lower speeds. Some single-phase motors, specially designed for speed control, have proved successful. Three-phase motors and controllers are being developed.

For a three-phase packaged chiller with multiple fans requiring fan speed control, a combination of three-phase and single-phase motors is used. For 208–230/3/60 power, a single-phase motor presents no problem, since the motor can be wired to two of the three power legs. For 460/3/60 power, single-phase motors require a step-down transformer or a separate single-phase power source.

4. Fan venturi. Its design is critical for optimum air flow with minimum air losses as well as for low outlet noise. See Fig. 6.2.13.

5. Fan guards. Fan guards are mounted around a fan venturi and are designed to meet the standards of the Occupational Safety and Health Administration (OSHA) so as to protect against accidents as well as to allow free air circulation.



FIGURE 6.2.13 Typical fan venturi. (Courtesy Continental Products Co.)

6.2.6.1 Air-Cooled Condensers

Air is used as the medium to cool and condense the hot refrigerant vapor. Generally, an air-cooled condenser consists of copper tubes and of copper or aluminum fins, which are expanded on the tubes for maximum heat transfer. The tubes are arranged in parallel or staggered rows and circuited for low refrigerant and air-pressure drop. The complete tube-and-fin condenser coil has a hot-gas inlet and refrigerant-liquid outlet connections. Condenser fan(s) are either directly driven or belt-driven to allow ambient air to circulate over the condenser coil.

Air is drawn from the bottom, goes over the condenser coil and condenser fan motor, and discharges upward. High-pressure, high-temperature refrigerant is being pumped through the hot-gas inlet of the condenser coil and is distributed according to the coil circuit, moving in the opposite direction of the air movement. In this process, heat transfer takes place. The cooler ambient air is circulated over the hot refrigerant. The ambient air picks up the heat, gets warm, and is discharged to the atmosphere. The hot refrigerant gas gets cooled and condenses into liquid.

Medium to Low Ambient Controls. The capacity of an air-cooled condenser is based on the temperature difference between the summer ambient-air temperature and the condensing temperature. When the packaged chiller is operated at temperature conditions lower than the design ambient-air temperature, the temperature difference between the condensing temperature and the ambient-air temperature is reduced, resulting in increased condenser capacity and lower condensing pressure.

If the ambient-air temperature falls below 60°F (15°C), the condensing pressure falls below a point where the expansion valve can no longer feed the cooler (evaporator) properly. Therefore, for 60°F (15°C) and below, it is necessary to use one or more of the following means to control the condensing pressure:

1. *Fan cycling.* By cycling one fan of a two-fan system, two fans of a three-fan system, and so on, a reasonable condensing pressure can be maintained. The cycling of fans can be in response to ambient-actuated thermostats, sensing the ambient-air temperature entering the condenser coil, or in response to the actual condensing pressure. Fan cycling is reasonable and simple and is recommended for medium-temperature applications.

2. *Fan speed control.* By fan-speed modulation, single-fan chiller packages or multiple-fan systems can operate at medium to low ambient conditions.

3. *Flooded-head pressure control.* Flooded-head pressure control holds back enough refrigerant in the condenser coils to render some of the coil surface inactive. This reduction of the effective condensing surface results in a higher condensing pressure, thus allowing enough liquid-line pressure for normal expansion-valve operation.

Typical head pressure-valve piping is shown in Figs. 6.2.14 (a nonadjustable-valve combination) and 6.2.15 (an adjustable-valve combination). Both valves require the use of a liquid-refrigerant receiver, as shown. The capacity of the receiver is critical in that it must be large enough to hold all the refrigerant during high ambient conditions. If the receiver is too small, liquid refrigerant will be held back in the condenser during high ambients, resulting in high discharge pressures. During low ambients the receiver pressure falls until it approaches the setting of the control valve orifice. This orifice throttles, restricting the flow of liquid from the condenser.

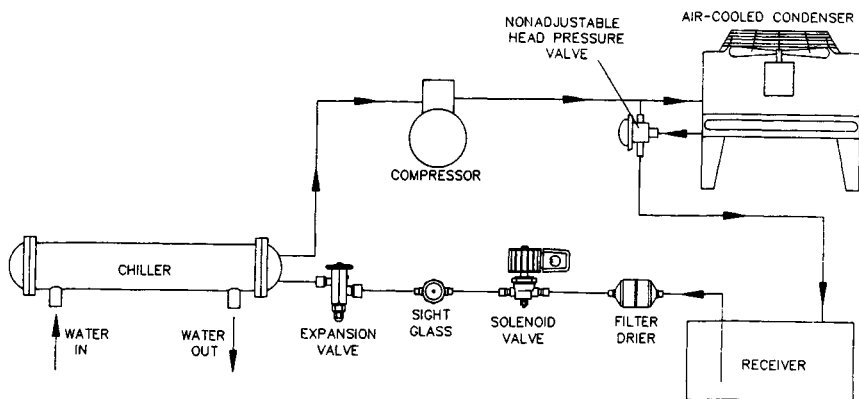


FIGURE 6.2.14 Typical head pressure-valve piping—nonadjustable. (Courtesy Continental Products Co.)

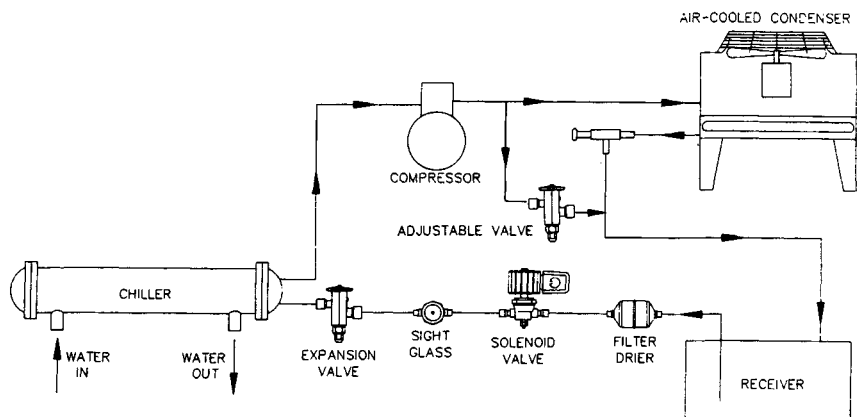


FIGURE 6.2.15 Typical head pressure-valve piping—adjustable. (Courtesy of Continental Products Co.)

Thus the liquid refrigerant is backed up in the condensing coil, reducing the surface area.

Flooded controls can maintain operation down to -40°F (-40°C) ambient or below. Under normal summer conditions the liquid side of the valve remains open, and the hot-gas side is fully closed. Under low ambient conditions, the liquid side remains closed on startup, causing the condenser to “flood.” This flooding continues until the condensing pressure reaches the valve setting [typically 180 psig (1241 kPa) for R-22, or 100 psig (689 kPa) for R-12]. Meanwhile the gas-side valve is open, allowing a portion of the hot gas to flow directly to the receiver, maintaining high pressure of the liquid for proper expansion-valve operation. Once the preset pressure is achieved, the valve modulates to maintain high head pressure, regardless of the ambients.

4. *Inlet-fan damper control.* This control modulates the air flow through the coil by the movement of dampers, in response to the condensing pressure. Usually a combination of fan cycling and damper control is utilized. Inlet dampers are mounted on the inlet of the fan and are actuated by a damper motor. Outlet dampers have also been used, but not very successfully. Experience has shown that damper control is generally not as effective as other means to achieve same results, so its usage is limited.

6.2.6.2 Water-Cooled Condensers

Water-cooled condensers are of the following three types, all of which use water as the cooling medium:

1. *Shell-and-tube condensers.* These condensers, for chillers, are generally built with a steel shell and finned copper tubes. The cooling water circulates through the tubes, and the hot-gas refrigerant is on the outside of the tubes on the shell side. The condensation of refrigerant vapor takes place as the high-temperature hot gas comes in contact with the cool tube surfaces. The condenser water thus picks up the heat rejected by the compressor. Water circuiting is baffled so as to have two, three, four, or six passes.
2. *Shell-and-coil condensers.* In this arrangement, a copper or cupronickel coil is contained within a shell. This type of condenser is limited to smaller sizes and is not generally used for chiller packages.
3. *Tube-and-tube condensers.* This type (which cannot be cleaned easily) consists of two tubes, one contained within the other. The annular space is used for water flow, and the inner tube is used for refrigerant condensing. Because the refrigerant undergoes a considerable pressure drop in the single tube, the use of tube-and-tube condensers is limited to smaller chiller packages up to 10 tons.

Condenser tubes can be cleaned mechanically or chemically. In any case, it is important to have cooling-water treatment for an efficient overall chiller system. Cooling water for condensers can be obtained from a cooling tower or from city water, but because it is uneconomical to use city water, a cooling tower is commonly used. A cooling tower's water temperature can vary according to the wet-bulb temperature of the ambient air. To keep the chiller system operating at a low water temperature, a water-regulating valve is installed on the condenser's water inlet. This valve modulates the flow of water in response to the condensing temperature or pressure.

6.2.2.7 Evaporative-Cooled Condensers

Evaporative-cooled condensers employ a copper, stainless-steel, or steel tube condensing coil that is kept continuously wet on the outside by a water-recirculating system. Simultaneously, centrifugal or propeller fan(s) move atmospheric air over the coil. A portion of the recirculated water evaporates, reusing heat from the condensing coil and thus cooling the refrigerant to its liquid state.

A typical evaporative-cooled condenser is shown in Fig. 6.2.16. The complete evaporative-cooled condenser consists of the following:

- Condensing coil (usually prime surface without fins)

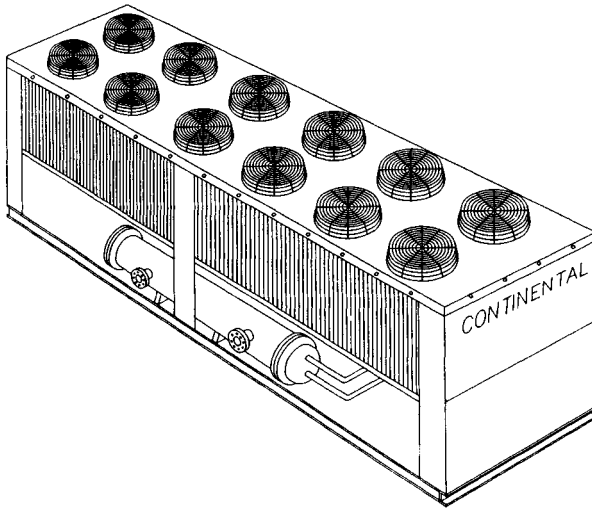


FIGURE 6.2.16 Typical evaporative-cooled condenser. (*Courtesy of Continental Products, Inc.*)

- Centrifugal or propeller fan(s)
- Water distribution system
- Drift eliminator
- Water makeup and drains

Since it combines principles of both air-cooled and water-cooled systems, an evaporative-cooled condenser can be considered a combination of these. The driving force is the ambient wet-bulb temperature, which is usually 15 to 25°F (8 to 14°C) lower than the ambient dry-bulb temperature. The overall effect is that an evaporative-cooled condenser operates at a much lower condensing temperature than an air- or water-cooled system. This results in the lowest compressor energy input and hence in the most efficient packaged chiller system.

SECTION 7

COOLING DISTRIBUTION SYSTEMS AND EQUIPMENT

CHAPTER 7.1*

CHILLED WATER AND BRINE

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7.1.1 INTRODUCTION

Water systems are used in air-conditioning applications for heat removal and dehumidification. The two most common systems use *chilled water* and *brine*. Chilled water is plain water at a temperature from 40 to 55°F (4 to 13°C). Brine is a water/antifreeze solution at a temperature below 40°F (4°C). Here we describe the basic principles and considerations for chilled water. Additional considerations for brine follow.

7.1.2 SYSTEM DESCRIPTION

A chilled-water system works in conjunction with air-handling units or process equipment to remove the heat generated within a conditioned space or process. The terminal unit cooling coil(s) collect(s) the heat and then transfers it by conduction and convection to the water, which is conveyed through connecting piping to the evaporator side of a chiller. The chiller, which is a packaged refrigeration machine, internally transfers the heat from the evaporator to the condenser, where heat is discharged to the atmosphere by the condenser system. The chilled water leaving the evaporator is circulated back to the coils, where the heat-removal process repeats again. Figure 7.1.1 is a schematic diagram of the chilled-water system. In this chapter we discuss the basic principles and details regarding the chilled-water or evaporator side of the system. The designer should refer to Chaps. 6.2 to 6.5 for types and selection of the chiller and condenser system.

*Revised for 2nd edition by Al Woody, Chief Mechanical Engineer, Giffels® Associates, Inc., Southfield, MI

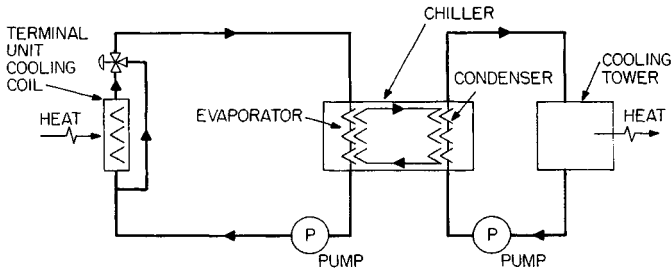


FIGURE 7.1.1 Schematic diagram of chilled-water system.

7.1.3 WHERE USED

Chilled-water systems are applicable when:

- The design considerations of a proposed air-conditioned facility require numerous separated cooling coils plus the restriction that the refrigeration system(s) of the facility be located in a single area.
- There is a need for close control of coil leaving air temperature or humidity control. Control can be very smooth and exact because of the infinite modulating capability of the chilled-water valve.
- Future expansion will require additional cooling capacity. The additional capacity might be merely a matter of new terminal units and branch piping from the chilled-water mains to the coils, although this is limited by the unused capacity of the chiller and water distribution system.
- The coil leaving air temperature desired is 45°F (7°C) or higher. The leaving air will be at least 5°F (3°C) warmer than the coil entering water temperature.

7.1.4 SYSTEM ARRANGEMENT

The system designer must consider the cooling loads involved and the type and arrangement of the facility during the conceptual phase of a chilled-water system design. During initial design development, the designer should consider the impact of future system loads. *System expansion costs can be reduced if space for additional equipment and the flow rates are planned for during the initial design.* The module design concept adapts well for planning for future expansion.

Large facilities commonly consist of terminal units located near the area they serve. The total combined loads of the facility result in a large peak demand with a wide operating range that is beyond the capability of a single chiller. A chilled-water arrangement for a large installation would commonly consist of multiple chillers centrally located with multiple cooling towers of the condenser system situated nearby outside. Figure 7.1.2 shows the evaporator side of a multiple-chiller arrangement. Installations of this type are typically arranged in modules with a chiller, cooling tower, and associated pumps dedicated to part of the peak load. A single distribution system transports the chilled water to the various areas and terminal units.

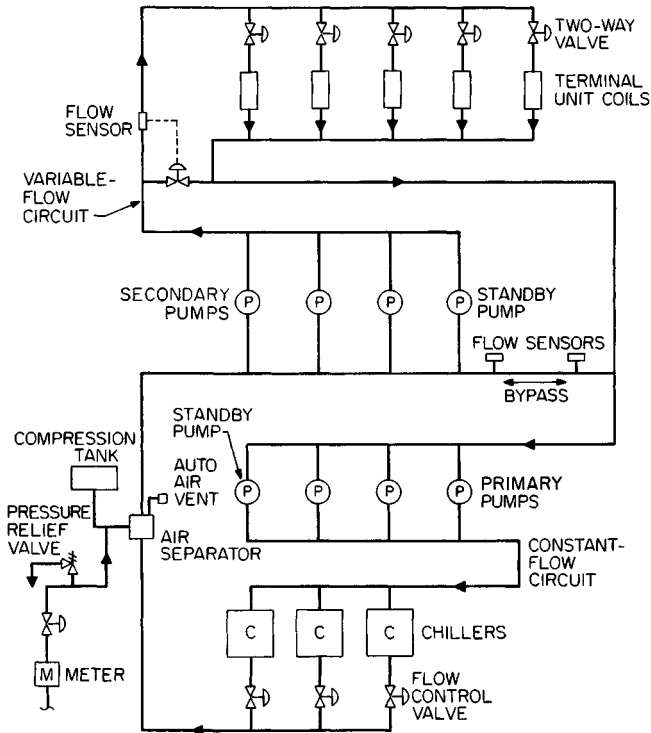


FIGURE 7.1.2 Multiple-chiller arrangement.

Two-way modulating control valves are used to vary the flow to the terminal units based on a signal from the conditioned space room thermostats. Two-way valves are preferred to three-way valves because the total system pumping cost is reduced during part-load conditions. Chiller manufacturers, however, demand a constant flow through the chillers for stable refrigeration control. In this situation a pump arrangement is needed that allows variable flow to the terminals while maintaining a constant flow through the chillers. This problem is solved by installing two sets of pumps in a primary/secondary arrangement. The secondary pumps can be controlled to match the demand of the terminals while the primary pumps maintain a constant flow through the chillers. The system bypass decouples the two pump sets which allows them to operate pressure-independent of each other. Reference 1 provides further explanation of primary/secondary pumping.

A small installation for an individual building or process may consist of a single chiller, cooling tower, pump, and small distribution system connected to nearby terminal units. The condenser system cooling tower would typically be located on the building roof or nearby, outside the building.

A three-way mixing valve modulates the terminal unit flow based on the cooling load demand while maintaining a constant flow through the chiller and pump. For small installations, the increased pumping cost is offset by the savings realized from fewer pumps and less complicated controls. Figure 7.1.3 shows the evaporator side of a single-chiller arrangement.

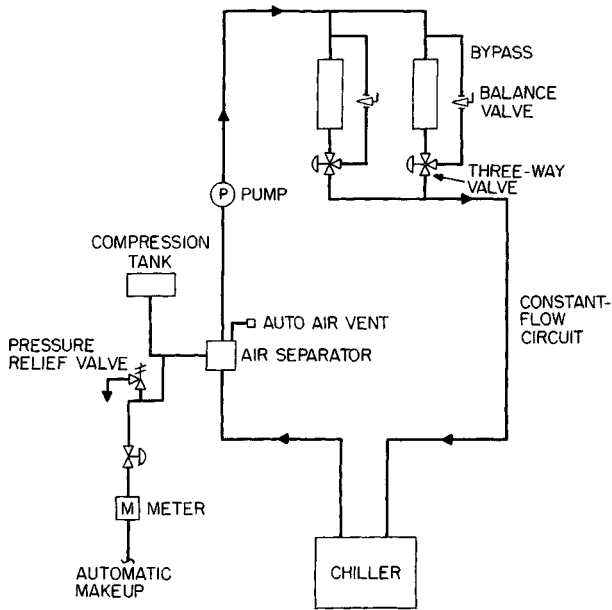


FIGURE 7.1.3 Single-chiller arrangement.

7.1.5 DISTRIBUTION SYSTEMS

There are two basic distribution systems for chilled water: the two-pipe reverse-return and the two-pipe direct-return arrangements. Figure 7.1.4 illustrates the direct-return and reverse-return configurations.

The *reverse-return* system is preferable from a control and balancing point of view, since it provides very close equivalent lengths to all terminals, resulting in closely balanced flow rates. In large installations, however, the additional piping for a reverse-return system is usually not economical.

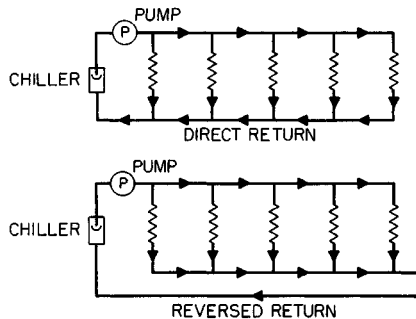


FIGURE 7.1.4 Piping distribution systems.

The *direct-return* system is more commonly used. The system must be carefully analyzed to avoid flow-balancing problems. Balancing valves and flow meters should be provided at each branch takeoff and terminal unit. Control valves with high head loss are recommended and must be analyzed for varying “shutoff” heads through the system. In large systems, it is sometimes desirable to use a combination of a direct-return system for the mains with a reverse-return system for branch piping to sets of terminal units. This combination provides an economical main distribution segment with easier balancing within the branches. The overall system should be analyzed to determine the most economical distribution system for the application.

7.1.6 DESIGN CONSIDERATIONS

7.1.6.1 Design Temperatures

The chilled-water design temperatures must be established before the terminal unit flow rates can be ascertained. Chilled-water supply temperatures range from 40 to 55°F (4 to 13°C), but temperatures from 42 to 46°F (6 to 8°C) are most common. Temperature differentials between supply and return are in the 7 to 11°F (4 to 6°C) range for small buildings and the 12 to 16°F (7 to 9°C) range for conventional systems. Higher temperature differentials are preferred since they reduce the system flow rates, resulting in smaller piping and pumps, less pumping energy, and increased chiller efficiency. Before the design temperatures are finalized, the designer should ensure that the design temperature selected will result in terminal devices properly sized for their applications. The designer should refer to Chaps. 6.2 to 6.4, and 7.3 for design considerations and selection of chilled-water coils.

For large distribution systems, a terminal supply temperature approximately 1°F (0.5°C) higher than the leaving chiller temperature is sometimes assumed. The 1°F (0.5°C) increase accounts for pump and pipe heat gains between the chiller and terminal units. The additional load from these sources must be included during the sizing of the chiller(s).

If the system is subject to freezing, a water/glycol solution (brine) may be required. Refer to the discussion of brine for additional design considerations.

After the terminal units have been located and the flow rates established, a system flow diagram should be created. Several chiller manufacturers should be consulted for sizes, types, and operating ranges available. The designer must analyze the facility for an appropriate distribution system, pump arrangement, and control sequence for all components.

If continuous system operation is required, standby pumps are recommended to ensure system operation in the event of a pump failure. Standby chillers are not usually included because of their high initial cost and rare failure.

7.1.6.2 Piping

Figure 7.1.5 is provided as a general guide for selecting pipe sizes once the system flow diagram has been established. The shaded area provides economical pipe sizes as a function of flow, velocity, and friction loss. In situations where two pipe sizes are capable of handling the design flow, the larger of the two should be selected, in case of an unexpected increase in the flow rate. The system designer should

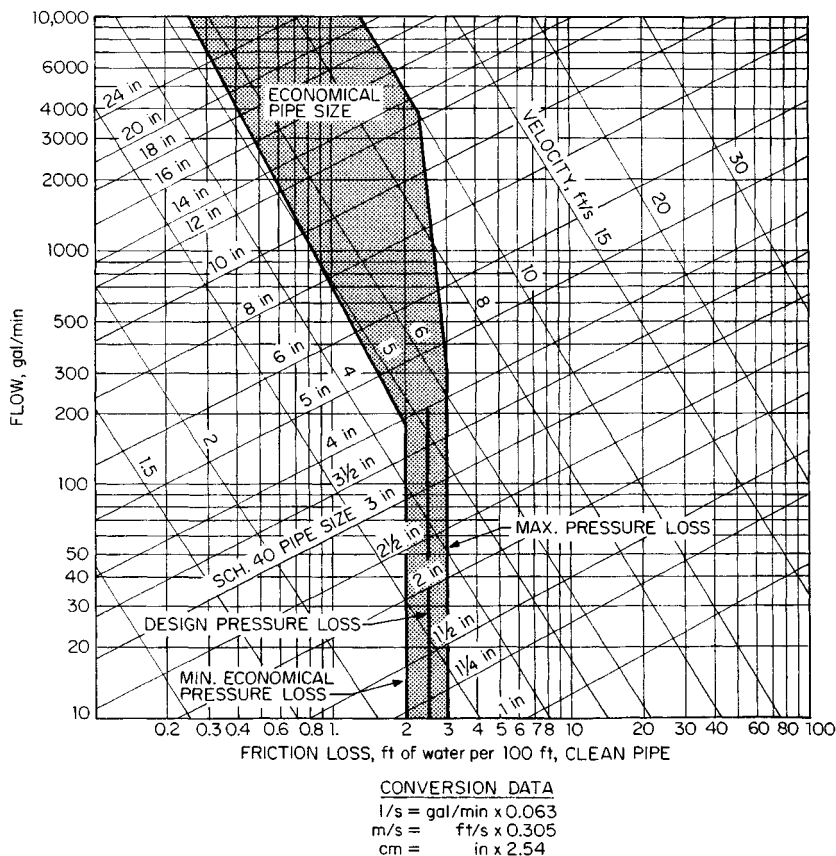


FIGURE 7.1.5 Pipe-sizing graph.

carefully size the inlet and outlet connections to terminal units. If they were left unsized, these branches could be installed, the same size as the terminal unit connection, which might result in abnormally high pressure losses.

Corrosion inhibitors are commonly added to chilled-water systems to reduce corrosion and scale. Refer to Chap. 12 for recommended water conditioning. With proper water treatment and a closed system, the pipe interior should remain relatively free of scale and corrosion. The calculated pump head can be based on relatively clean pipe, although it is prudent to assume a minimum fouling factor. A 25 percent fouling factor is equivalent to using $C = 130$ in the Williams and Hazen formula for steel pipe.

7.1.7 INSTALLATION CONSIDERATIONS

7.1.7.1 System Volume

Automatic makeup water with a positive-displacement meter is recommended for filling, monitoring volume use, and maintaining system volume. The systems should

also be attached to an adequately sized compression or expansion tank and to a pressure relief valve. If the tank is not provided and the system has automatic makeup water, the relief valve will discharge with each rise in average water temperature, wasting water and chemicals.

7.1.7.2 Air Control

Manual or automatic air vents must be installed at system high points to vent air when the system is filled. Automatic vents should be provided with an isolation valve to enable replacement. Vent blowoff lines should be piped down to the closest waste drain. Air, in horizontal mains, can generally be kept out of the branch piping when the branch connections are in the bottom 90° arc of the main. A branch pipe with vertical downflow that connects to the bottom of a main can accumulate pipe scale and similar debris. Dirt legs, such as for steam drips, or strainers may be useful in these branches, especially for 2-in (51-mm) and smaller piping. Air in vertical piping will flow *down* with the water at 2 ft/s (0.6 m/s) or greater water velocity.

7.1.7.3 System Isolation

All equipment requiring maintenance and branch piping should be provided with manual isolation valves. Chain-wheel operators are recommended for frequently used valves located out of the operator's reach. Drain connections should be provided at low points to allow partial system drainage of isolated sections.

7.1.7.4 Coil Control

The coil capacity or degree to which an airstream is cooled or dehumidified as it passes through a chilled-water coil is generally controlled by varying the quantity of water flowing through the coil. (Capacity can also be controlled by varying the water temperature or by varying a portion of air which bypasses the coil and re-mixing it with the portion passed through the coil.) Ideally the control valve will vary the water flow continuously and uniformly as the valve strokes from full open to full closed; that is, 10 percent of stroke should cause a 10 percent change in flow. However, if a line-size butterfly valve is installed, it might pass 80 percent of full flow when only 20 percent open, which means the valve is trying to control 0 to 100 percent flow while 0 to 25 percent "open." The flow tends to be unstable in these situations, especially at low flow where the valve is apt to wire-draw, chatter, and hunt. Valve size is very important. In the interest of satisfactory control stability, the valves are frequently less than line size because the pressure loss through the fully open valve should be at least 33 percent of the total pressure loss of the circuit being controlled. For example, if the total coil circuit loss is 7 lb/in² (48 kPa), a valve sized for a drop of 3 to 5 lb/in² (21 to 34 kPa) would be satisfactory. Strainers are recommended upstream of control valves and any other piping elements which require protection against pipe scale and debris within the system.

7.1.8 SYSTEM MONITORING

Pressure gauges permanently installed in a system deteriorate over time from constant vibration. Gauges should be installed only at points requiring periodic monitoring. At points where infrequent indication is required, gauge cocks should be installed with a set of spare gauges provided to the operator. Thermometers are recommended at all terminal units and chillers.

For additional explanation and considerations for the design of a chilled-water system, see Ref. 2.

7.1.9 BRINE

The term “water” is used throughout this chapter for convenience, whereas it could be plain water or a brine. The term “brine” includes a water/glycol solution, a proprietary heat-transfer liquid, water and calcium or sodium chloride solution, or a refrigerant. The best choice of brine will depend on the parameters of the system, but plain water or a water/glycol solution is the overwhelming choice for comfort air-conditioning chilled-water systems. Propylene glycol is the least toxic of the glycols and should be used if there is any possibility (e.g., piping leaks) of contact with a food or beverages.

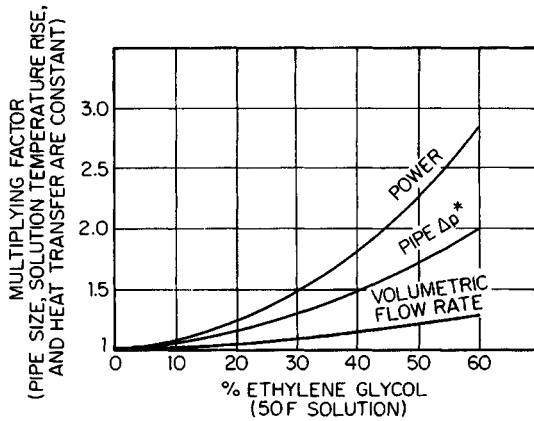
7.1.9.2 Where Used

An antifreeze must be added to the water, or a nonfreezing liquid must be used, whenever any portion of the water is subject to less than 33°F (0.6°C). Be sure to check the temperature of the refrigerant, or cooling medium, in the chiller. If it operates below 33°F (0.6°C), freeze protection is needed. In systems where the refrigerant is only a few degrees below 33°F (0.6°C), chiller freezeup can be precluded by maintaining chilled-water flow through the chiller for some period after the chiller has been shut off. This water flow will “boil” the refrigerant remaining in the evaporator.

7.1.9.2 Design Considerations

Adding an antifreeze to water will generally reduce the specific heat and conductivity and increase the viscosity of the solution. These, in turn, generally necessitate increased heat-transfer surface in the chiller and cooling coils, increased chilled-water flow, and increased pump head. See Fig. 7.1.6. For example suppose a plain water system involves 8 inch (20.3 cm) pipe, 1000 gal/min (63.1 L/s), 50 lb/in² (345-kPa) pressure drop for pipe friction, and 29 hp (21.6 kW) to overcome pipe friction. Then

- If 10% glycol is added, the parameters become 1010 gal/min (63.7 L/s), 53 lb/in² (365 kPa), and 31 hp (23.1 kW).
- If 40% glycol is added the parameters become 1150 gal/min (72.6 L/s), 75 lb/in² 516 kPa), and 53 hp (39.5 kW).



*CAUTION: FACTOR FOR Δp CURVE IS NOT FOR EQUAL VOL. FLOW RATES OF PLAIN WATER & E.G. SOLUTION

FIGURE 7.1.6 Flow, pressure drop, and power consumption factors for ethylene glycol solutions versus plain water. See para. 7.1.9 "design considerations"

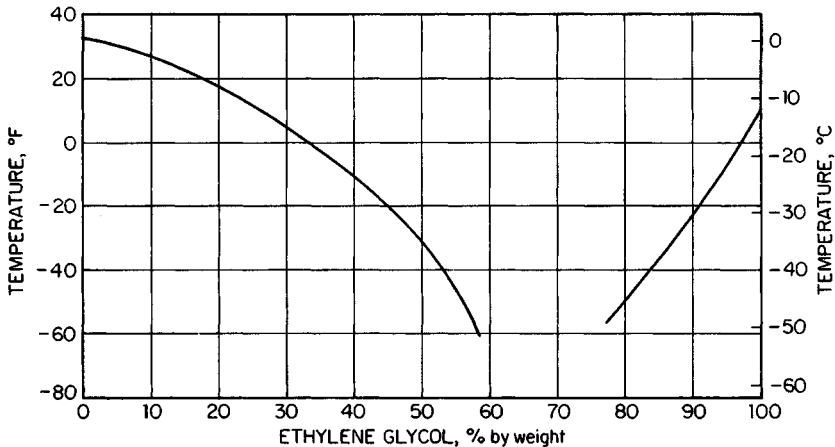


FIGURE 7.1.7 Freezing points of aqueous ethylene glycol solutions. (From Union Carbide Corp., Ethylene Glycol Product Information Bulletin, Document F-49193 10/83-2M.)

The piping size and solution temperature rise are assumed the same in the plain water and glycol systems.

The curves drawn in Fig. 7.1.8 show how the pressure loss caused by pipe friction is affected by the solution temperature and ethylene glycol concentration for various pipe inside diameters. Note that the curves are specific for a solution velocity of 6 pfs (1.83 m/c) because this was used in the formula to determine Reynold's Number which in turn was used to establish the curves.

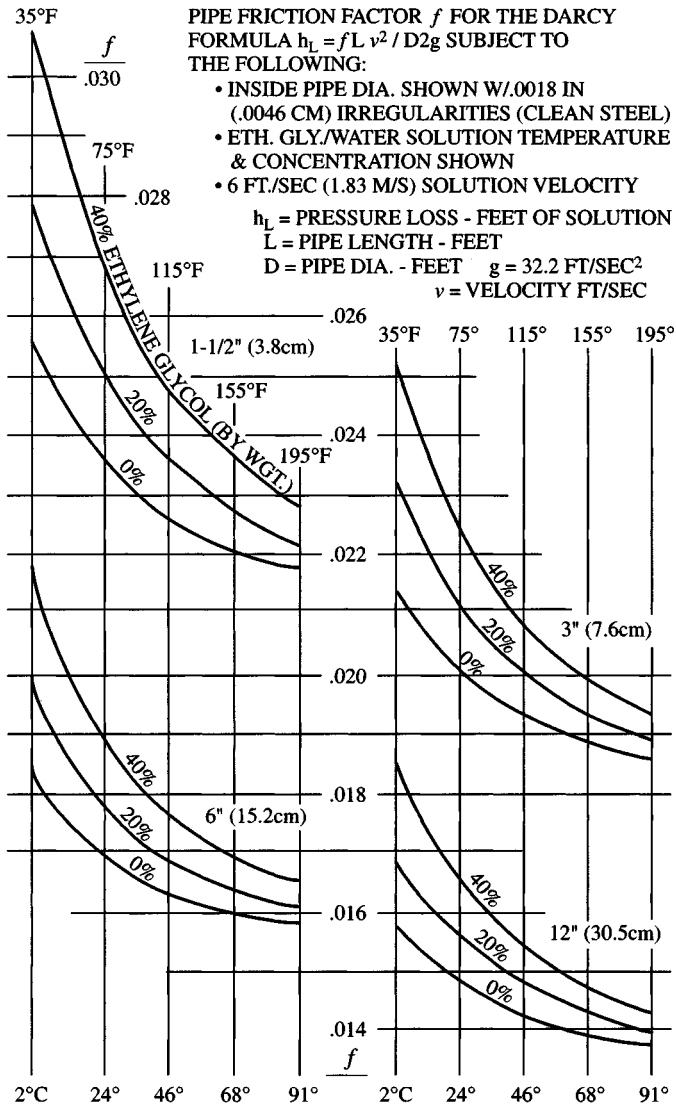


FIGURE 7.1.8 Pressure losses vs solution temperature and concentration.

The freezing point of aqueous glycol solutions can be found from charts similar to Fig. 7.1.7. Note that 40% glycol is needed to lower the freezing point to -10°F (-23°C). By definition, the freezing point is that at which the first ice crystal forms. Chilled-water piping has been protected from freeze damage with as little as 10% or even 5% glycol in 0°F (-18°C) weather. This is referred to as “burst protection.” Ice crystals may form at 25 to 29°F (-4 to -2°C) in the 10% or 5% water/glycol solution, but the solution merely forms a slush and does not freeze solid. The slush must be permitted to expand. If it is trapped between shutoff valves, check valves, automatic valves, etc., pipe rupture may occur. The slush must be permitted to melt before the chilled-water pumps are started.

Automotive antifreeze solutions should not be used. The corrosion inhibitors added to automotive antifreeze solutions are specifically made for the materials encountered in an automobile engine. Automotive antifreeze is not meant for long life, whereas industrial heat-transfer fluids may last 15 years with proper care.

A chemical analysis of the makeup water must be checked for compatibility with the proposed chiller, pump, piping, and coil materials, chemical treatment, antifreeze, corrosion inhibitors, etc., to preclude the formation of scale, sludge, and corrosion. The water should be checked regularly for depletion of any components.

7.1.10 STRATIFIED CHILLED-WATER STORAGE SYSTEM

The chilled-water storage system is a conventional chilled-water system with the addition of a thermally insulated storage tank. Figure 7.1.9 shows a typical chilled-

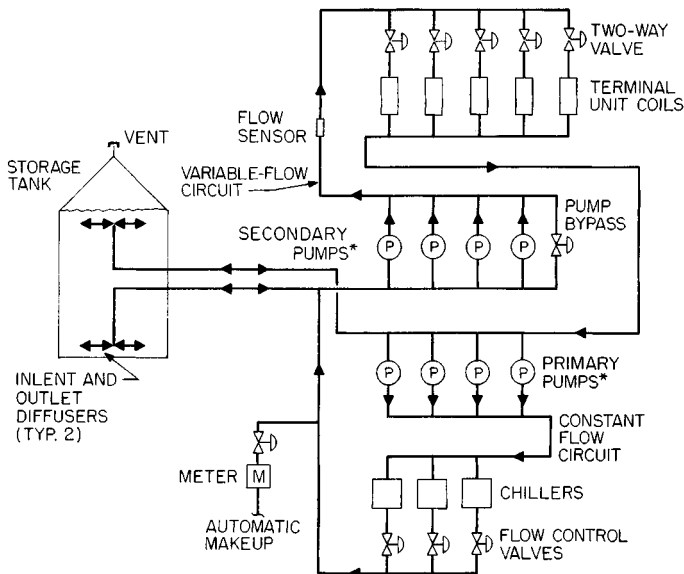


FIGURE 7.1.9 Chilled-water storage arrangement. Asterisk indicates stand-by pumps.

water storage arrangement. During the daily cooling cycle, the chillers operate to maintain cooling until the load exceeds the capacity of the system. At that point, the chillers and tank work in conjunction to handle the peak demand. As the load falls below the chiller's capacity, the chillers continue to operate to recharge the tank for the next day's demand.

The advantage of this arrangement is that a portion of the equipment required for a conventional system to handle peak loss can be replaced by a less expensive storage tank. In addition, the owner's electric power rates are reduced since the tank has shaved the monthly peak power demand.

The system is classified as a stratified storage system because warm water and cold water within the storage tank remain separated by stratification. During operation, as a portion of chilled water is removed from the tank bottom for cooling, the identical portion of warm return water is discharged back into the tank at the top. A thermal boundary forms with the warmer, less dense water stratifying at the top and the denser, colder water remaining below. During periods of reduced load, the tank is recharged by removing the warm stratified water from the tank top, chilling it, and returning it to the tank bottom. During a daily cycle, the thermal boundary moves up and down within the tank, but the total water quantity remains unchanged.

7.1.10.1 Load Profile

Figure 7.1.10 shows a typical building cooling load profile utilized for storage applications. Curve *ABCDE* represents the cooling load profile during the day. Point *C* represents the maximum instantaneous peak. line *FG* represents the installed chiller capacity. The area within *ABDE* represents the portion of cooling provided by the chillers, and area *BCD* represents the portion of cooling provided by the tank. The remaining areas, *FBA* and *DGE*, represent the chiller capacity available to recharge the tank. For storage applications, units of ton-hours (kWh) are used to determine the cooling load and storage requirements of the system.

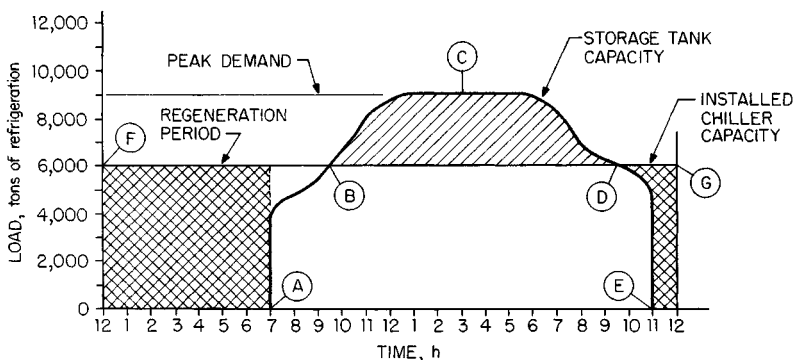


FIGURE 7.1.10 Cooling-load profile. See text explanation of letter symbols.

7.1.10.2 Where Used

Chilled-water storage systems generally become economical in systems in excess of 400 tons (1407 kW) of refrigeration. In all cases, the economics of the installation should be the deciding factor for choosing a storage system.

7.1.10.3 Design Considerations

The design engineer must analyze the operating parameters of the facility in order to accurately predict the load-cycle hours for a given day. The number of hours required for cooling and available for tank regeneration must be determined. Once established, the daily cooling load can be calculated. If available, a computer cooling-load program capable of providing an hour-by-hour analysis is recommended for predicting the cooling load profile. Example 7.1.1 demonstrates the method for creating a load profile diagram and determining the refrigeration and tank capacity for a storage installation.

EXAMPLE 7.1.1

FIND: (1) Total refrigeration load

(2) Tank capacity

GIVEN: Cooling period, 8 a.m. to 8 p.m. (12 h)

Regeneration period, 8 p.m. to 8 a.m. (12 h)

Note: Hourly loads provided are not from Fig. 7.1.10 load profile

Daily cooling load, hour-by-hour analysis

Time	Total load	
8 a.m.	30,084 MBtu	$(7.58 \times 10^6 \text{ kcal})$
9	36,972	(9.31×10^6)
10	45,144	(1.14×10^7)
11	59,268	(1.49×10^7)
12	72,168	(1.82×10^7)
1 p.m.	80,952	(2.04×10^7)
2	82,212	(2.07×10^7)
3	83,100	(2.09×10^7)
4	85,392	(2.15×10^7)
5	85,320	(2.15×10^7)
6	83,436	(2.10×10^7)
7	81,900	(2.06×10^7)

Total	825,948 MBtu	$(2.08 \times 10^8 \text{ kcal})$
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Total daily load (ton/h)

$$\begin{aligned}
 &= 825,948 \text{ MBtu} / [12 \text{ MBtu}/(\text{ton} \cdot \text{h})] \\
 &= 68,829 \text{ ton/h} \\
 &= (2.08 \times 10^8 \text{ kcal} / 3022 \text{ kcal}/(\text{ton} \cdot \text{h})) \\
 &= (68,829 \text{ ton/h})
 \end{aligned}$$

Solution (1) Total Chiller Capacity

$$\begin{aligned}
 &68,829 \text{ ton/h} / 24 \text{ hr} = 2868 \text{ tons} \\
 &(\text{Select two } 1500\text{-ton chillers})
 \end{aligned}$$

(2) Storage Tank Capacity

$$\begin{aligned} & 68,829 \text{ ton/h} - (3000 \text{ ton} \times 12 \text{ h}) \\ & = 32,829 \text{ ton/h} \end{aligned}$$

Based on 20°F temp. diff (11.1 K)

$$\frac{(7.48 \text{ gal/ft}^3 (32,829 \text{ ton/h}) [12,000 \text{ Btu/(h} \cdot \text{ton)}])}{(62.4 \text{ lb/ft}^3 (20^\circ\text{F}) [1.0 \text{ Btu/(lb} \cdot ^\circ\text{F)}])}$$

$$\left(\frac{(1000 \text{ L/m}^3) (32,829 \text{ ton/h}) [3022 \text{ kcal/(ton} \cdot \text{h)}]}{(1000 \text{ kg/m}^3) (11.1\text{K}) [0.998 \text{ kcal/(kg} \cdot \text{K)}]} \right)$$

$$\text{capacity} = 2,361,162 \text{ gal (8,955,680 L)}$$

Increasing tank volume 20% for mixed zone and internal piping:

$$\text{Total Capacity} = 2,833,395 \text{ gal (10,746,817 L)}$$

7.1.10.4 Design Temperatures

As previously described for the conventional chilled-water system, the design temperatures of the system must be established. Temperature differentials for storage applications typically range from 16 to 22°F (−10 to −6°C). A higher temperature differential is preferred since it will reduce the size of the tank and system flows. Before the design temperatures are finalized, the terminal unit coils must be checked to ensure that they can be sized for proper operation. High differential temperature can result in low water velocities within the coil tubes, which may lead to poor part-load performance. Tube water velocities between 2 and 5 ft/s (0.6 and 1.5 m/s) are preferred for efficient heat transfer.

7.1.10.5 Tank Sizing

After the design temperatures have been established, the storage tank capacity can be determined. The capacity previously calculated in ton-hours can be converted to gallons, as illustrated in Example 7.1.1. the tank sizing must also allow both for unused space due to piping and related apparatus and for the mixed thermal boundary between warm and cold fluids.

7.1.10.6 Installation Considerations

Tank. Tanks for storage applications are field-fabricated of steel or concrete. Steel is generally preferred over concrete for stratified storage because steel readily absorbs and rejects the changes in water temperature without disturbing the thermal boundary. For example when a concrete tank is recharged, the rising chilled water is warmed by the heat stored in the concrete.

The tank should have a roof, to keep out unwanted debris. The exterior of the tank must be insulated; spray foam or rigid board insulation is used for this type of installation. The tank should be equipped with provisions for access, filling and draining, venting, and overflow, with associated controls for temperature and level monitoring.

Diffuser. The size, number, and orientation of the diffusers within the tank depend on the design parameters of the system. The intent of the diffusers is to allow removal and replacement of the tank water without disrupting the stratified thermal boundary. The tank should be installed in parallel with the chillers. This arrangement will result in two diffusers one at the top and one at the bottom of the tank.

Radial-type diffusers have been used with success for cylindrically shaped tanks. The diffuser consists of two steel plates with the inlet pipe located at the center. As the water enters the diffuser, the flow is distributed in all directions toward the outside wall. The diffuser should be designed for low outlet velocity. This is determined by designing for a Froude number of 1 or below. Example 7.1.2 demonstrates the method for sizing a radial diffuser.

EXAMPLE 7.1.2

FIND: Radial diffuser height between plates

GIVEN: Maximum flow = 3500 gal/min (220 L/s)

Diffuser diameter = 10 ft (3.048 m)

Water temperatures = 44°F (6.6°C), 65°F (17.7°C)

Solution The Froude number is dimensionless and defined by the following equation:

$$F = \frac{Q}{\sqrt{g(\Delta p/p)h^3}}$$

where F = Froude number = 1

Q = outlet/inlet flow ratio, defined as ft^3/s (m^3/s) flow divided by outlet perimeter in ft (m)

g = gravitational effect, ft/s^2 (m/s^2)

p = water density, lb/ft^3 (kg/m^3)

Δp = difference in water density between inlet and outlet, lb/ft^3 (kg/m^3)

h = distance between plates, ft (m)

$$1 = \frac{3500 \text{ gal/min}}{7.48 (\text{gal} \cdot \text{min})/\text{ft}^3 \times (60 \text{ s/min}) \times 10 \text{ ft} \times 3.14} \sqrt{32.2[(62.40 - 62.31)/62.40]h^3}$$

or

$$1 = \frac{220 \text{ L/s}}{1000 \text{ L}/\text{m}^3 \times 3.048 \text{ m} \times 3.14} \sqrt{9.8 \text{ m}/\text{s}^2[(999.7 - 998.3)/999.7]h^3}$$

Solving for h gives 1.13 ft (0.34 m).

Temperature Monitoring. The tank should be equipped with temperature controls to monitor the temperature gradient within the tank. Thermocouples are typically used and installed vertically along the tank wall. The spacing should be adequate to identify the location of the thermal layer at all times. Generally, spacing of 1 to 2 ft (0.3 to 0.6 m) should be adequate. However, judgment is required depending on the tank's dimension.

7.1.11 REFERENCES

1. *Primary Secondary Pumping Application Manual*, International Telephone and Telegraph Corp., Bulletin TEH-775, 1968.
2. The American Society of Heating, Refrigeration, and Air-Conditioning Engineers, Inc., 1992 *ASHRAE Handbook, Systems and Equipment* ASHRAE, Atlanta, GA, chaps. 12 and 13.
3. The American Society of Heating, Refrigeration, and Air-Conditioning Engineers, Inc., 1991 *ASHRAE Handbook, Applications*, ASHRAE, Atlanta, GA, chap 39.

CHAPTER 7.2

ALL-AIR SYSTEMS

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7.2.1 SINGLE-ZONE CONSTANT-VOLUME SYSTEM

The single-zone constant-volume system is the basic all-air system used in a temperature-controlled area (Fig. 7.2.1). This system will maintain comfort conditions in an area where the heating or cooling load is fairly uniform throughout the space.

The method of controlling the temperature in the area can vary based on the functions performed in the space. The basic general-use area will require only modulating the cooling or heating medium at the air-handling equipment to maintain desired space conditions. Areas requiring a closer temperature-humidity control

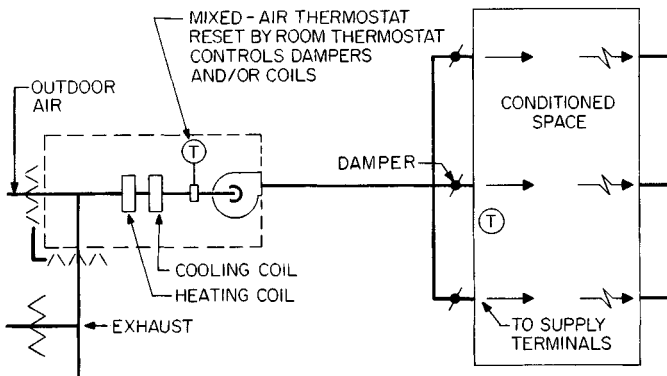


FIGURE 7.2.1 Single-zone system.

will need a unit that can cool and reheat at the same time, to maintain a close temperature range.

7.2.1.1 Central Equipment

A single-zone unit, shown in Fig. 7.2.1, consists of a supply fan, cooling coil, heating coil, filter section, and return-air/outdoor-air mixing plenum.

The combination of these components in the unit provides a system that can maintain a basic temperature-controlled environment with a change in either heating or cooling loads.

Variations and additions to these components can provide a system that can maintain a closely controlled temperature and humidity environment throughout the year. With the addition of a return-air fan to systems having long return air ducts, outdoor air can provide the required cooling medium during certain periods of the year. Adding a humidifier to the unit provides means to control and maintain a precise level of humidity to match the area function.

7.2.1.2 Ductwork System

The layout of the single supply duct to the conditioned space should be routed with a minimum of abrupt directional and size configuration changes.

The supply ductwork should be sized by using the static regain method to assist in a balanced air distribution in the duct system. The branch or zone mains should be provided with a balancing damper at the point of connection to the supply main. This will enable fine adjustments to be made to the air distribution system within the zone.

The supply air terminals in the space should be selected, sized, and located to provide even distribution throughout the space without creating drafts or excessive noise. Each terminal should have a volume damper to permit individual air balancing.

The return-air ductwork should be sized by using the equal-friction method from the space return registers back to the central equipment. The same ductwork configuration considerations and accessories should be used in laying out the return ductwork as listed for the supply duct system.

The location of the central equipment relative to the conditioned space should be considered when one is evaluating the need for acoustically lined ductwork or sound traps at the central equipment, to prevent transmission of noise through the duct system to the space.

7.2.1.3 Applications

The single-zone systems are generally used for small offices, classrooms, and stores. The single-package type of individual air-handling unit, complete with refrigeration and heating capabilities, can be roof-mounted or located in a mechanical space adjacent to or remote from the conditioned space.

7.2.2 SINGLE-ZONE CONSTANT-VOLUME SYSTEM WITH REHEAT

A single-zone constant-volume system with reheat has the same equipment and operating characteristics as the single-zone system, but has the advantage of being able to control temperatures in a number of zones with varying load conditions. See Fig. 7.2.2.

Areas made up of zones with varying loads can be supplied by a single supply air system of constant volume and temperature. The air quantity and air temperature are based on the maximum load and comfort conditions established for the area. The individual rooms or zones within the area can be temperature-controlled with the addition of a reheat coil to the branch supply duct.

7.2.2.1 Central Equipment

The same equipment as described for the single-zone constant-volume system is used, except for the addition of reheat coils in the branch ducts serving zones with changing heating and cooling loads.

The heating coil can be electric, hot water, or steam. A space thermostat modulates the reheat coil to maintain the desired space temperature.

7.2.2.2 Ductwork System

The ductwork for the constant-volume system with reheat requires the same considerations as listed for the single-zone constant-volume system.

The addition of a reheat coil will require ductwork enlargement transition before the coil and a reducing fitting after the coil, to ensure proper air flow over the entire

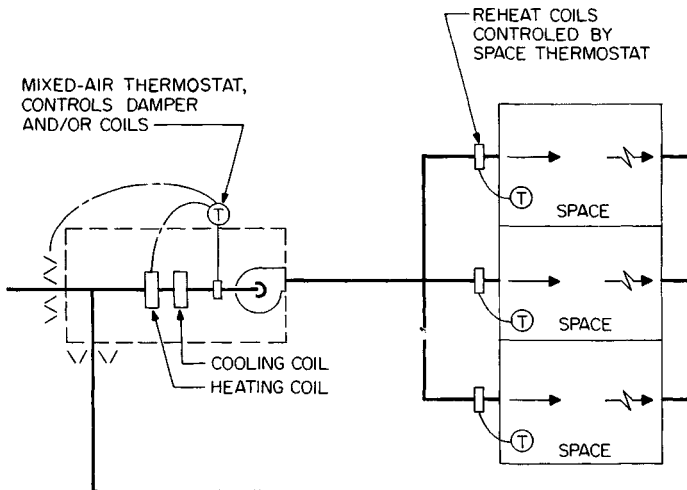


FIGURE 7.2.2 Reheat system.

face area of the coil. Access doors should be provided in the ductwork on the entering and leaving sides of the coil for cleaning and inspection.

7.2.2.3 Applications

The single-zone system with reheat coils is used for small commercial facilities which may be divided into a number of areas and/or offices with varying internal and perimeter loads. These systems, which use reheat to maintain comfort, should be provided with controls to automatically reset the system cold-air supply to the highest temperature level that will satisfy the zone requiring the coolest air.

The leaving air temperature of a reheat coil depends on several factors:

- The design space heating temperature
- Whether there is a supplementary heating system along the exterior perimeter of the building (such as fin pipe convectors, fan coil units, etc.) for the zone served by the reheat coil
- Whether there is a space equipment cooling load during the heating season

For instance, one of the following conditions can determine the leaving air temperature of a reheat coil:

- *Condition 1:* When the space or zone does not have an exterior exposure or has a supplementary perimeter heating system and there is no equipment cooling load during the heating season, the reheat coil leaving air temperature should nearly equal the space design temperature.
- *Condition 2:* When the space or zone has an exterior exposure without a supplementary perimeter heating system and there is no equipment cooling load during the heating season, the reheat coil leaving air temperature should equal the space design temperature plus the temperature difference calculated to offset the space or zone heating loss from the exterior exposure.
- *Condition 3:* The space or zone is the same as for condition 1 except there is an equipment cooling load requirement during the heating season. Then the reheat coil leaving air temperature should be equal to the space design temperature minus the temperature difference calculated to offset the space equipment cooling load.
- *Condition 4:* The space or zone is the same as for condition 2 except there is an equipment cooling load requirement during the heating season. Then the reheat coil leaving air temperature should be equal to the space design temperature plus the temperature difference calculated to offset the space or zone heat loss from the exterior exposure minus the temperature difference calculated to offset the space equipment cooling load.

7.2.3 MULTIZONE SYSTEM

This type of system (Fig. 7.2.3) is used when the area being served is made up of rooms or zones with varying loads. Each room or zone is supplied by means of a single duct from a common central air-handling unit.

The central air-handling unit consists of a hot-air plenum and cold-air plenum with individual modulating zone dampers mixing hot and cold air streams and

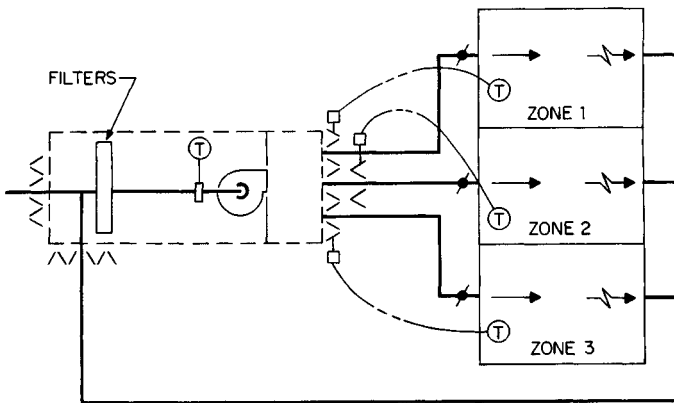
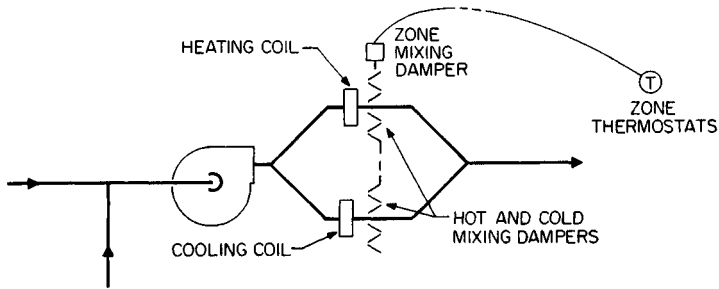


FIGURE 7.2.3 Multizone system.

supplying the mixture through a dedicated duct to the space. A thermostat located in the occupied space modulates the zone dampers at the unit to achieve the desired temperature conditions.

7.2.3.1 Central Equipment

The multizone unit shown in Fig. 7.2.3 may be a factory-assembled package unit consisting of a mixing plenum, filter section, supply fan, heating coil, cooling coil and damper assemblies on the discharge side of coils. A humidifier can be added to the unit to maintain a winter humidity level.

7.2.3.2 Ductwork System

The supply ductwork for the multizone system originates at the central unit dampered discharge outlet from the hot and cold deck. Each zone will be supplied by a single duct with a number of supply air terminals. The supply ductwork should be sized by using the static region method, to assist in a balanced air distribution

in the duct system. Branch duct takeoffs from the duct mains should be provided with balancing dampers to permit fine adjustments to the air distribution system within the individual zones.

The supply air terminals in the space should be selected, sized, and located to provide even distribution throughout the zones without creating drafts or excessive noise. The supply air terminal should be provided with a volume damper to balance air quantities at the individual outlets.

The return-air ductwork should be sized by using the equal-friction method from the space return registers back to the central equipment. The same ductwork configuration considerations and accessories shall be used in laying out the return duct as was used in designing the supply duct system.

The location of the central equipment relative to the conditioned space should be considered when one is evaluating the need for acoustically lined ductwork or sound traps at the central equipment, to prevent transmission of noise through the duct system to the space.

7.2.3.3 Application

The multizone type of system is considered for office buildings, schools, or buildings with a number of floors and interior zones with varying loads.

The multizone system and dual-duct system, to an extent, will give similar performances inasmuch as the dual-duct system is sometimes described merely as a multizone system with extended hot and cold decks. However, the following real differences do exist:

- Packaged multizone air handlers are available with up to 14 zones whereas dual-duct systems have virtually no limit as to zones.
- Building configuration may be better suited to the numerous small ducts from a multizone system than to the two large ducts off a dual-duct air handler.
- The small zone off a multizone which also has large zones will have erratic air flow when the large zone dampers are modulating. The pressure-independent mixing boxes of a dual-duct system preclude this.
- The damper leakage at “economy” multizone units can be excessive, especially when maintenance is poor.
- It is undoubtedly more costly and cumbersome to add a zone to an existing multizone system than to use a dual-duct system.
- Packaged multizone systems are suitable for small systems and as such may include direct expansion cooling and gas-fired heating equipment. The step capacity control included with this equipment can result in noticeable cycling of space temperatures. The larger cooling and heating equipment generally accompanying dual-duct systems includes modulating capacity controls, and this precludes the space temperature cycling.
- The air in the short hot and cold plenums of multizone units can experience the same temperature gradient as that of a heating coil which has a “hot end,” especially during low loads (and similarly for cooling coils). This temperature gradient can result in improper hot (or cold) air entering the zone duct. The long hot and cold ducts of the dual-duct system permit thorough mixing of air off the coils and eliminate the gradient.

7.2.4 INDUCTION UNIT SYSTEM

The induction unit system is used for the perimeter rooms in multistory buildings such as office buildings, hotels, hospital patient rooms, and apartments. See Fig. 7.2.4.

The system consists of a central air-handling unit which supplies primary air, heated or cooled to offset the building transmission loss or gain; a high-velocity duct system for conveying the primary air to the induction units; an induction unit with a coil for each room or office; and a secondary water system, which is supplied from central equipment. The secondary water system is heated or cooled depending on the time of year and the requirements of the space being served.

A constant volume of primary air is supplied from the central air-handling unit through a high-pressure duct system to induction units located in the rooms. The air is introduced to the room through the high-pressure nozzles located within the unit that cause the room air to be drawn over the unit coil. The induced air is heated or cooled depending on the secondary water temperature and is discharged into the room.

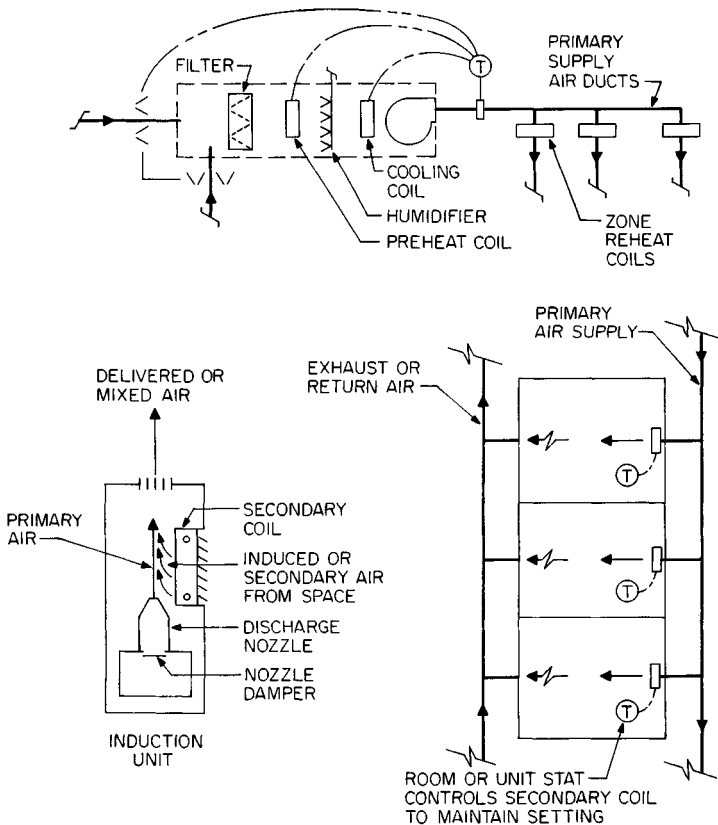


FIGURE 7.2.4 Induction unit system.

7.2.4.1 Central Equipment

The primary air supply unit for the induction system generally includes a filter, humidifier, cooling coil, heating coil, and fan. A preheat coil is also included when the unit handles large quantities of outdoor air which is less than 32°F (0°C). The heating coil may be in the form of zone reheat coils when the unit supplies induction units on more than one exposure (north, east, south, or west).

The supply fan is a high-static unit sized to provide the primary air requirements for each induction unit. The chilled water or refrigerant cooling coil dehumidifies and cools the primary air during the summer months. Primary air is supplied at a constant rate to the induction units and is generally 40 to 50°F (4 to 10°C) year-round. The final room temperature is maintained by the secondary coil.

7.2.4.2 Ductwork System

The air supply to the induction units originates from a central air-handling unit. The supply header ductwork should be routed around the perimeter of the building, with individual risers routed up through the floors supplying primary air to the induction units.

Limited available duct space frequently dictates that velocities in the risers be maintained at 4000 to 5000 ft/min (20 to 26 m/s). Rigid spiral ductwork is used, with elbows and takeoffs being of welded construction. Close attention must be paid to prevent noisy air leakage in the duct system.

A sound-absorbing section of ductwork should be provided at the discharge of the central air handler to absorb noise generated by the high-pressure fan.

The supply header and risers should be thermally insulated to prevent heat gain and sweating during summer operation and heat loss during winter operation.

The supply ductwork system should be sized by using the static regain method.

7.2.4.3 Application

The induction unit system is well suited to the multistory, multiroom buildings with perimeter rooms that require individual temperature selection.

The benefits in using the induction system in these types of buildings is in the reduced amount of space required for air distribution and equipment. The secondary coil in the induction unit is frequently connected to a two-pipe dual-temperature system which provides the coil with hot water during the winter and chilled water during the remaining seasons. The thermostat modulates water flow and therefore varies the temperature of the delivered or mixed air to compensate for the room heat loss or heat gain.*

7.2.5 VARIABLE-AIR-VOLUME SYSTEM

This system is used primarily when a cooling load exists throughout the year, such as the interior zone of office buildings. This air supply system uses varying amounts

*For more details see *1987 ASHRAE Handbook, HVAC Systems and Applications*, American Society of Heating, Refrigeration, and Air-Conditioning Engineers, Atlanta, GA, 1984.

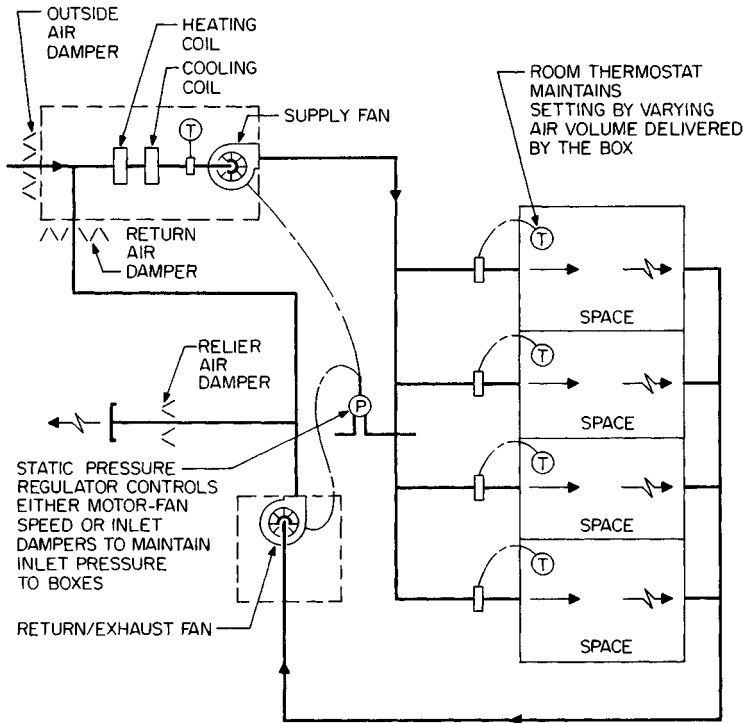


FIGURE 7.2.5 Variable-volume system.

of constant-temperature air induced to a space to offset cooling loads and to maintain comfort conditions. The system operates equally well at exterior zones during the cooling season, but during the heating season supplementary equipment such as reheat coils, finned radiation, etc., must be provided at all spaces with an exterior exposure. When a reheat coil is added to a variable-air-volume (VAV) box, the temperature controls should reduce air flow to the minimum acceptable level for room air motion and make-up air and then activate the reheat coil.

The system typically consists of a central air-handling unit with heating and cooling coil, single-duct supply system, VAV box, supply duct with air diffuser, return air duct, and return air fan.

Constant-temperature air is provided from the central air-handling unit through a single-supply air duct to the individual VAV box which regulates supply air to zone to offset cooling load requirements.

7.2.5.1 Central Equipment

The air supply unit consists of a supply fan with variable inlet vanes, variable speed control or discharge dampers, cooling coils using refrigerant or chilled water, filters, heating coil using steam or hot water for morning warmup, return air fan which is modulated through controls to match supply fan demands, and mixed air plenum to provide outdoor air requirements to the system.

The supply fan should be selected for the calculated load and system static pressure. During system operation, the supply air demand varies with space load requirements. To meet this demand, the supply and return fans' discharge air quantities must be modulated in unison with variable inlet vanes, variable speed control, or discharge dampers.

The cooling coil, being either a direct-expansion or chilled-water type, automatically controls the discharge air temperature for the unit. During the winter cycle, the mixed air damper and return air damper are modulated to maintain the discharge air temperature.

The following items may need special consideration when VAV systems are designed:

Minimum Outdoor Air. The outdoor air drawn into a building will tend to reduce as the supply fan volume reduces. This can become detrimental when the supply fan has a large turndown from its maximum flow and/or when the VAV system provides makeup air for constant-flow exhaust systems. The minimum outdoor air can be maintained by providing a short duct with a flow sensor downstream of the minimum outdoor air damper. The flow sensor modulates the outdoor and return air dampers to maintain the required minimum.

Building Static Pressure. It is important that return fan volume be properly reduced when the supply fan volume reduces. The return should reduce at a greater rate so as to leave a fixed flow rate for the constant-flow exhaust systems and building pressurization. Flow sensors at the supply and return fans can monitor and maintain a constant *difference* between supply and return air by modulating the return air and exhaust air flow.

Room Air Motion. Select air diffusion devices for proper performance at minimum as well as maximum flow to preclude "dumping" of air.

Building Heating. Calculations frequently show that the internal heat gain (lights, equipment, people) during occupied hours of basement, interior, and sometimes perimeter spaces is more than sufficient to keep these spaces warm. So it may appear that a "mechanical" heat source is not required. But these heat gains might not exist during unoccupied nights, weekends, and shutdown periods, and the spaces will cool down even when the only exposure is a well-insulated wall or roof. The central equipment of VAV systems is sometimes designed without heating coils and in itself cannot heat the building (it "heats" by providing less cooling). Unit heaters, radiation elements, convectors, or heating coils including controls coordinated with the VAV system at zero outdoor air are required for a timely morning warmup and heat when the space is unoccupied.

Calculations for winter usually show a need for heat at perimeter spaces. If the VAV boxes have "stops" for minimum air supply, there must be sufficient heat to warm this minimum air [usually 55 to 60°F (13 to 15°C)] in addition to that required for transmission losses.

7.2.5.2 Ductwork System

The supply air mains and branch ducts can be sized for either low or high velocity depending on the space available in the ceiling. A low-velocity design will result in lower operating costs.

Supply ducts should have pressure relief doors, or be constructed to withstand full fan pressure in the event of a static pressure-regulator failure concurrent with closed VAV boxes.

7.2.5.3 Application

The variable-air-volume system is considered for a building with a large interior zone requiring cooling all year. The varying amounts of cool air match the varying internal loads as people and lighting loads change throughout the day and night. When used in conjunction with perimeter heating systems, the VAV system can be used for the perimeter zone of a building also. These conditions describe the typical operations of office buildings, schools, and department stores which are the prime users of this type of system.

7.2.6 DUAL-DUCT SYSTEM

This type of system is used when the area served is made up of rooms or zones with varying loads with the entire area being supplied from a central air-handling unit. See Fig. 7.2.6. The central unit supplies both cold and hot air through separate duct mains to a mixing box at each zone. The zone box controlled by the space thermostat mixes the two air streams to control the temperature conditions within the zone.

7.2.6.1 Central Equipment

The dual-duct unit may be a factory-assembled packaged unit consisting of a mixing plenum, filter section, supply fan, heating coil, cooling coil, and discharge plenums. A humidifier can be added to maintain winter humidity level.

7.2.6.2 Ductwork System

The hot and cold supply ductwork headers originate at the unit discharges and run parallel throughout the building, connecting to the individual mixing boxes which supply the zone ductwork.

The cold duct header is sized to carry the peak air volumes of all zones. The hot duct header is sized to carry a certain percentage of the cold air, usually 70 percent.

The zone mixing box responds to a space thermostat and modulates and mixes quantities of cold and hot air and delivers a constant volume of air to the space to maintain desired temperature levels. The size of the mixing box is based on the peak air volume of the zone or room.

The mixed air from the mixing box is discharged through a single duct terminating with a number of supply air diffusers, chosen and sized to provide even distribution throughout the zone.

Supply ductwork may be sized by the static regain or equal-friction method with aerodynamically smooth fittings and velocities not exceeding 3000 ft/min (15 m/s).

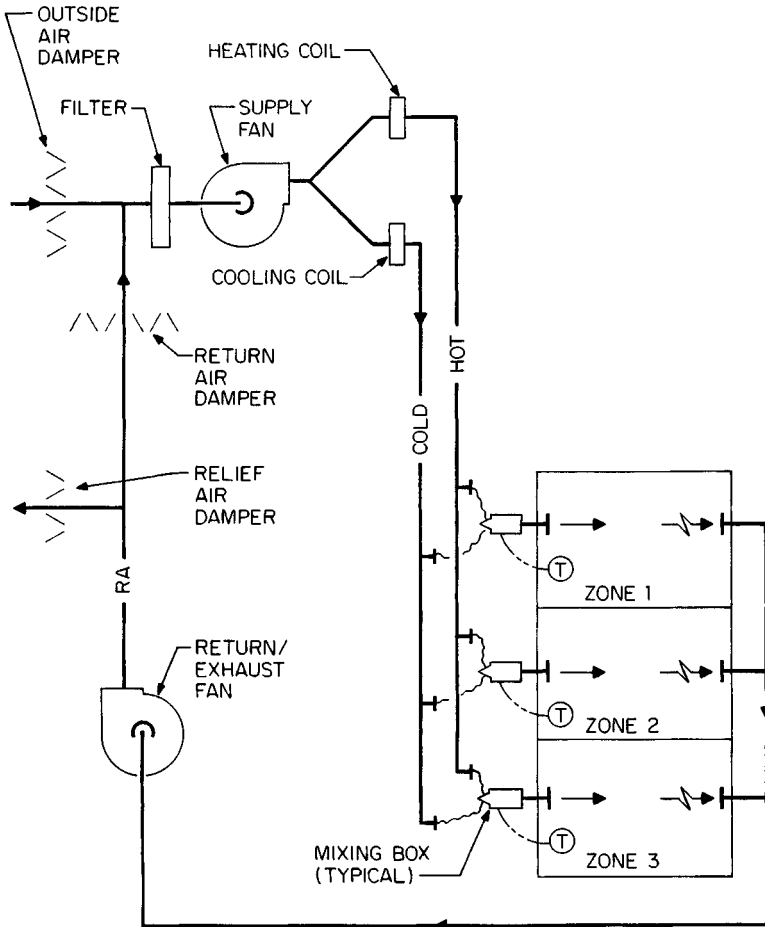


FIGURE 7.2.6 Constant-volume dual-duct system.

The return-air ductwork is often sized by the equal-friction method, but it does not exceed 1500 ft/min (7.6 m/s). The routing and configuration of the supply headers must satisfy the space limitations in ceiling and shaft areas. Access and space requirements for the mixing box should also be considered when the routing of the duct system is laid out.

The location of the central equipment relative to the conditioned space should be considered when deciding the need for acoustically lined ductwork or sound traps at the central equipment, to prevent transmission of noise through the duct system to the space.

7.2.6.3 Application

The dual-duct type system is considered for office buildings, schools, or buildings with a number of floors and zones with varying loads. Generally, however, this

system has been “replaced” by the VAV system because of higher operating and first costs and increased duct space requirements.

7.2.7 BIBLIOGRAPHY

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CHAPTER 7.3

DIRECT EXPANSION SYSTEMS

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7.3.1 SYSTEM DESCRIPTION

To air-condition the interior space of a building, recirculating and makeup air is commonly cooled and simultaneously dehumidified while passing across a cooling coil. This coil could be a direct expansion (DX) type.

DX refrigeration utilizes refrigerant (working fluid of the cycle) fluid temperature, pressure, and latent heat of vaporization to cool the air. To evaporate liquid refrigerant to a vapor, the latent heat of vaporization has to be applied to the liquid. The quantity of heat necessary to evaporate 1 lb (0.45 kg) of liquid refrigerant varies with the thermal characteristics of different refrigerants. The boiling point of an ideal refrigerant has to be below the supply air temperature and above 32°F (0°C), so as to not freeze the moisture condensed from the building supply air. Most refrigerants in use today have a relatively low boiling point and nonirritant, non-toxic, nonexplosive, nonflammable, and noncorrosive characteristics for use in commercially available piping materials.

Early air-conditioning and refrigeration systems used toxic and hazardous substances such as ether, chloroform, ammonia, carbon dioxide, sulfur dioxide, butane, and propane as refrigerants. The most common refrigerants in use today are numbers 11, 12, 22, and 502 (for low-temperature applications such as food freezing).^{*} There is concern that these refrigerants may be damaging the earth's ozone layer and alternative refrigerants are being offered for use in new equipment. Refrigerant 22 is still in use, but efforts are underway to find a suitable replacement. Production of refrigerant 11 and 12 is ending in 1995 and their replacement refrigerants are R134a and R123, respectively. See also Chapter 6.1.

A simple refrigeration system is illustrated in Fig. 7.3.1. Basic elements of the system are the expansion valve (pressure-reducing valve), evaporator (cooling coil or DX coil), compressor, condenser, and interconnecting piping. The compressor and expansion valve are points of the system at which the refrigerant pressure changes. The compressor maintains a difference in pressure between the suction

^{*}These refrigerants can be found on the market under the various trade names of Freon (registered trademark of E. I. du Pont de Nemours Co.), Genetron (registered trademark of Allied Chemical Corp.), and Isotron (registered trademark of the Pennsalt Chemicals Corp.).

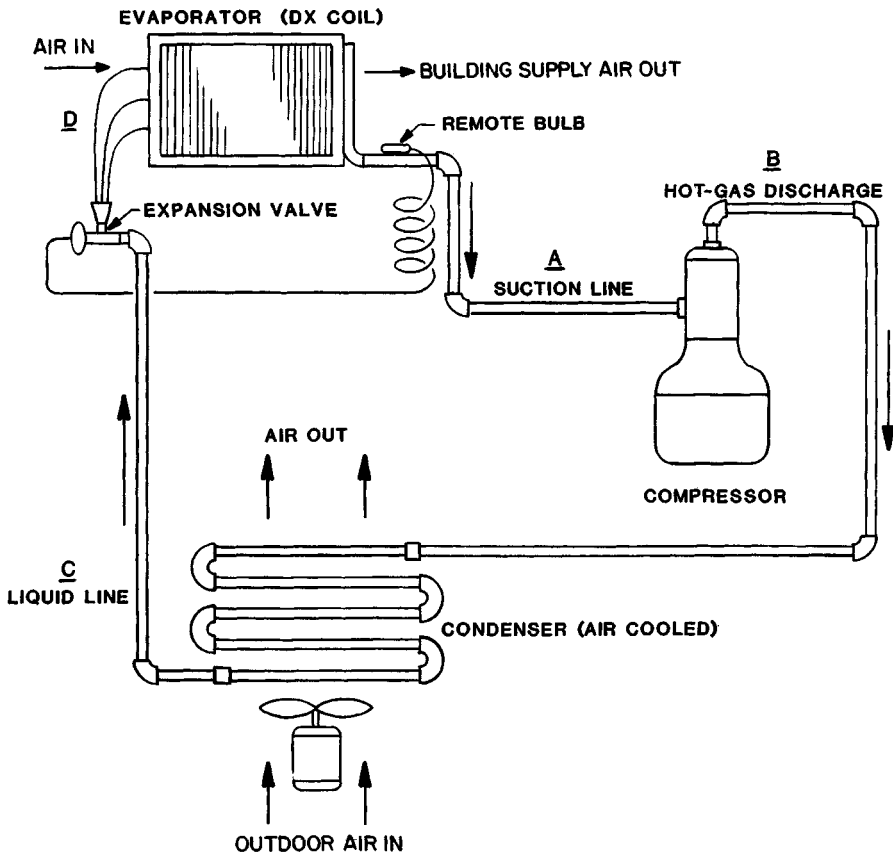


FIGURE 7.3.1 Mechanical refrigeration system.

and discharge sides of the system (between the DX coil and condenser), and the expansion valve separates high- and low-pressure sides of the system. The function of the expansion valve is to meter the refrigerant from the high-pressure side (where it acts as a pressure-reducing valve) to the low-pressure side (where it undergoes a phase change from a liquid to a vapor during the process of heat absorption).

The compressor draws vaporized refrigerant from the evaporator through a suction line A. In the compressor, the refrigerant pressure is raised from evaporation temperature and pressure to a much higher discharge pressure and temperature. In the discharge line B, refrigerant is still in the vapor state at high temperature, usually between 105 and 115°F (40 and 46°C). A relatively warm cooling medium (water or air) can be used to condense and subcool the hot vapor. In the condenser, heat of vaporization and of compression is transferred from the hot refrigerant gas to the cooling medium through the walls of the condenser heat-exchange surfaces while the gas becomes liquefied at or below the corresponding compressor discharge pressure C.

The expansion valve separates the high-pressure or condenser side of the system from the low-pressure or evaporator side. The purpose of the expansion valve is to control the amount of liquid entering the evaporator such that there is a sufficient amount to evaporate but not flood the evaporator *D*.

In the evaporator, liquid refrigerant is entirely vaporized by the heat of the building supply air. Heat equivalent to the latent heat of vaporization has been transferred from building air through the walls of the evaporator to the low-temperature refrigerant. Thus the building supply air is cooled and dehumidified. The boiling point (temperature of evaporation) at the evaporator pressure is usually between 34 and 45°F (1 and 7°C) for refrigerants 11, 12, and 22 or their replacements. It is even lower for refrigerant 502.

From the evaporator, vaporized refrigerant is drawn through suction piping to the compressor, and the cycle is repeated.

All refrigerants have different physical and thermal characteristics. Depending on the available condensing temperature, required evaporation temperature, and cooling capacity, different refrigerants are used for different applications.

Figure 7.3.2 illustrates the theoretical refrigeration cycle (without pressure losses in the system and without subcooling of the liquid or superheating of the vapor) shown on a Mollier or pressure-enthalpy diagram.

7.3.2 EQUIPMENT

The purpose of this section is to describe the components of a DX refrigeration system and their functions. In addition to the basic elements already mentioned (compressor, condenser, expansion valve, evaporator, and refrigerant piping), the typical refrigeration system incorporates other components or accessories for various purposes. Figure 7.3.3 illustrates a simple DX refrigeration system with an air-cooled condenser designed for operation in cold weather.

7.3.2.1 Compressor

This is a vapor-phase fluid pump which maintains a difference in refrigerant gas pressure between the DX coil (low-pressure or suction side) and the condenser (high-pressure or discharge side) of the system. Compressors can be categorized as to construction, i.e., hermetic, semihermetic, and open (direct- or belt-driven). They also can be categorized by the type of machine, i.e., reciprocating, centrifugal, and screw. More about compressors can be found in Chaps. 6.2 to 6.5.

7.3.2.2 Condenser

This is a heat-exchange device where heat of vaporization and compression is transferred from hot refrigerant gas to the cooling medium in order to change the refrigerant from a superheated vapor to a liquid state and sometimes to subcool the refrigerant. Condensers can be air-cooled, where outdoor air is used to condense and subcool the refrigerant, or water-cooled, where city water or cooling tower water is used as the cooling medium. Evaporative condensers use both water and

TEMPERATURE, °F (°C)
 SP. VOLUME, FT³/lb (m³/kg)
 ENTHALPY, BTU/lb (kJ/kg)
 ENTROPY, BTU/lb • R (kJ/kg • K)
 PRESSURE, lb/in² (kPa)
 QUALITY, %

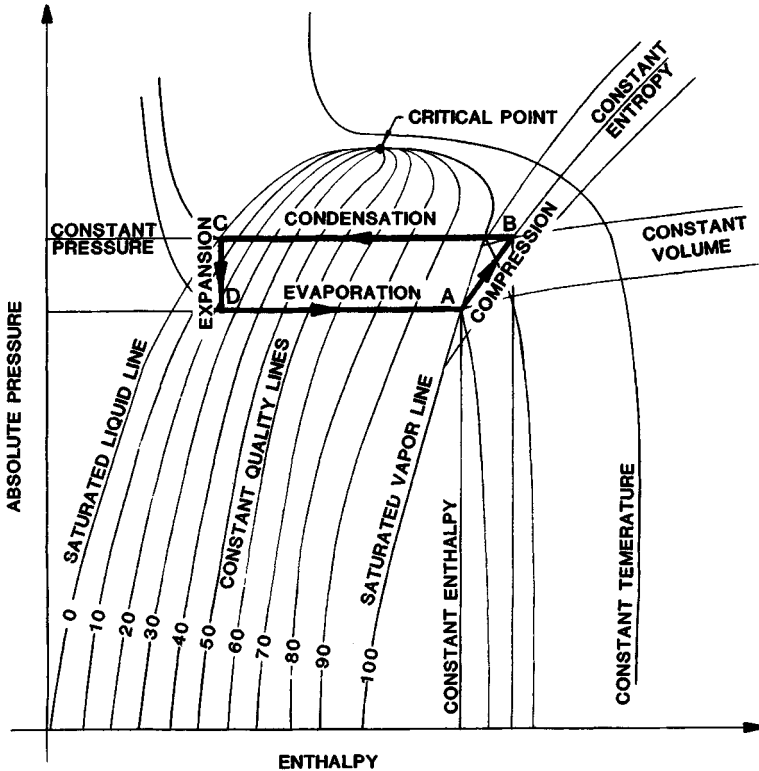


FIGURE 7.3.2 Mollier diagram.

air to condense and subcool refrigerant: recirculating water is sprayed over tubes containing hot refrigerant and is evaporated by moving outdoor air, thus removing heat from the refrigerant.

Air-cooled condensers can be single- or multifan types. Axial fans are most commonly used because axial fans economically handle large air volumes at low static pressure. Centrifugal fans, which are capable of generating higher static pressures, are used in certain applications.

Condensers can also be categorized as single- and multicircuit, according to whether they are connected to one or multiple compressors.

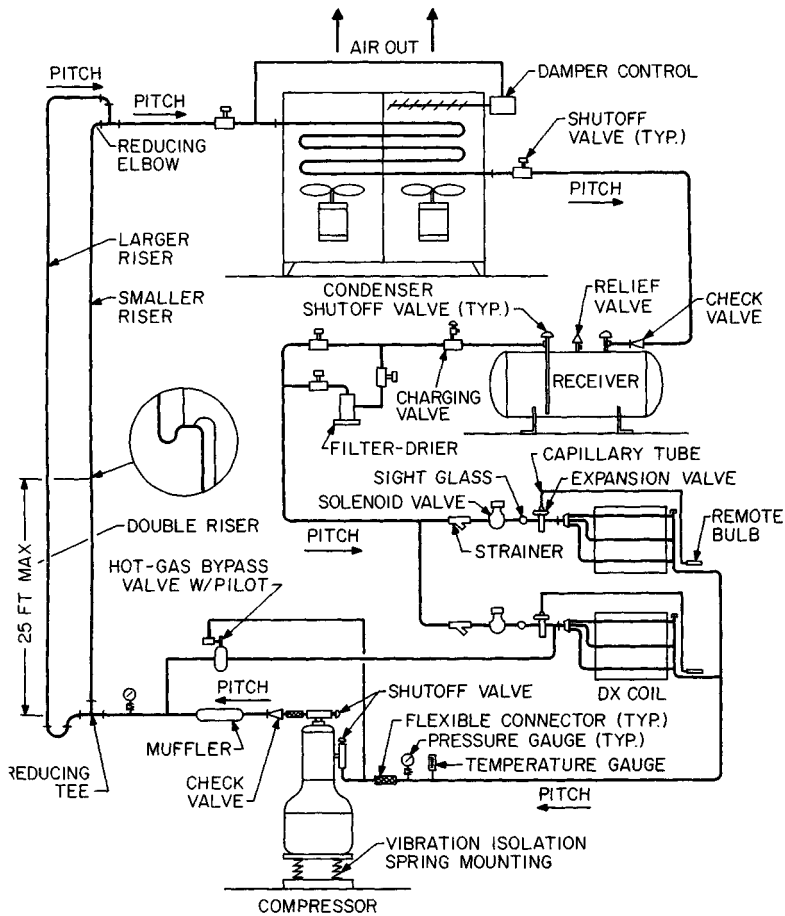


FIGURE 7.3.3 Reciprocating DX system.

7.3.2.3 Expansion Valve

This is a throttling or metering device with a diaphragm operator. The space above the diaphragm is connected to a remote sensor bulb with capillary tubing and filled with the same refrigerant as is used in the system. The valve controls flow of fluid refrigerant to maintain a set-point pressure in the evaporator.

The remote temperature-sensing bulb is normally strapped or soldered to the suction line (leaving evaporator) for maximum surface contact. An increase in heat load on the evaporator is sensed by the bulb, causing a corresponding increase in vapor pressure within the bulb, capillary tube, and space above the diaphragm. This pressure, transmitted by the diaphragm, moves the valve off its seat, to admit more liquid refrigerant into the evaporator, for evaporation by the increased heat load.

When the cooling requirements are satisfied, the process reverses.

7.3.2.4 Evaporator (DX Coil)

This is an extended-surface (finned tube) device where heat exchange occurs between building supply air and the liquid refrigerant in the coil tubes, causing the refrigerant to vaporize.

DX coils are either dry or flooded (with refrigerant liquid). The coils can have 20 or more parallel circuits and are of one- or multirow construction.

7.3.2.5 Refrigerant Piping

Typically, Type L copper tubing is used for handling chlorinated fluorocarbon refrigerants, discussed earlier.

7.3.2.6 Hot-Gas Bypass Control

This control is a way to maintain a reasonably stable evaporator suction pressure when a refrigeration system is operating at minimum load. Two hot-gas bypass methods are in use today on direct expansion systems: hot-gas bypass to the evaporator inlet and to the suction line.

Hot-gas bypass to the evaporator inlet introduces compressor discharge vapor to the DX coil after the expansion valve (see Fig. 7.3.3). This acts as an artificial heat load on the DX coil and raises the temperature at the coil outlet. The remote bulb of the expansion valve senses this temperature rise and opens the valve to increase the flow of refrigerant through the coil, resulting in a rise of the suction pressure and stabilization thereof.

The effectiveness of this method is a function of the distance between the (compressor) and the DX coil. This method should not be used when the distance is greater than 50 ft (15 m) for hermetic compressors (hot gas might start condensing, resulting in oil “holdup” and compressor lubrication problems).

Hot-gas bypass to the suction line introduces hot vapor from the compressor discharge to the inlet (suction) side of the compressor (see Fig. 7.3.4). This method requires an additional liquid line solenoid valve and expansion valve. When low suction pressure is sensed because of reduced heat load, hot gas is introduced to the compressor inlet through the hot-gas bypass valve. This causes the supplementary expansion valve to open and to introduce liquid refrigerant into the hot gas. The liquid refrigerant evaporates and increases compressor suction pressure to stabilize operation.

Disadvantages of this method include the additional cost of expansion and solenoid valves, oil trapping in the DX coil, and the possibility of liquid slugs entering the compressor.

7.2.3.7 Suction and Hot-Gas Double Riser

This is a pipe assembly which promotes oil movement to the compressor on the suction side and from the compressor on the hot-gas side. A double-riser arrangement is used in a vertical piping layout when the compressor is below the condenser and/or when the compressor is above the DX coil. Figure 7.3.3 illustrates the double riser on the hot-gas side. A similar setup would be provided on the suction side if the compressor were located above the DX coil.

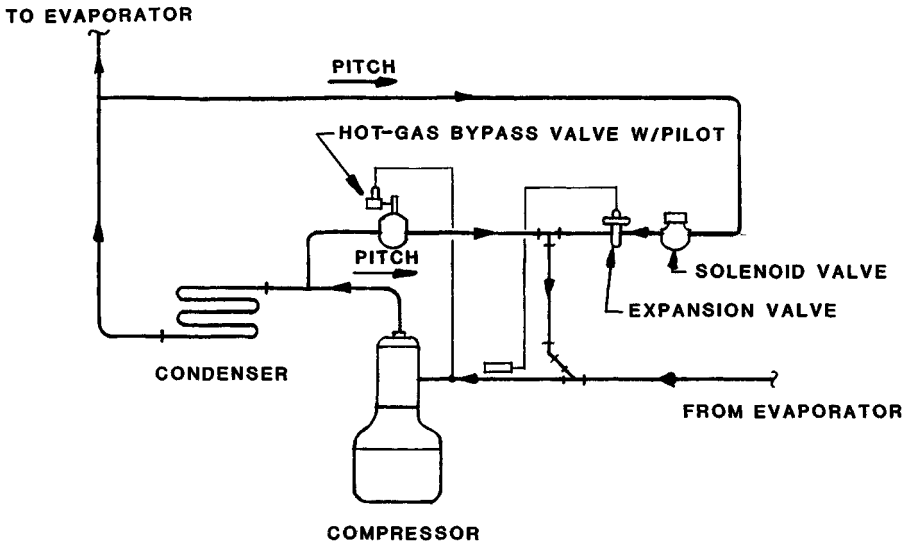


FIGURE 7.3.4 Hot-gas bypass to suction line.

Operation at minimum compressor capacity (with the compressor unloaded or one compressor running in multiple-compressor installations) reduces oil conveying velocities in the system which causes compressor oil to fill the trap, thus directing gas flow to the smaller riser. This riser is sized to produce a velocity of 1000 ft/min (5 m/s) which is sufficient to convey oil upward for return to the compressor. When full-load capacity is restored, pressure clears the trap and flow is established through both risers. The larger riser is sized for velocities between 1000 and 4000 ft/min (5 and 20 m/s) at full compressor load.

The maximum vertical rise of a double riser should not exceed 25 ft (8 m). If greater than 1 25-ft (8 m) rise is required, an intermediate trap should be incorporated every 25 ft (8 m) of rise (see Fig. 7.3.3).

7.3.2.8 Filter-Drier

This is usually installed in the liquid line to protect the expansion valve from dirt or moisture that may freeze in the expansion valve and to protect motor windings from moisture. The filter-drier core has an affinity for and retains water while simultaneously removing foreign particles from the liquid refrigerant (see Fig. 7.3.3).

7.3.2.9 Condenser Pressure Control

This control is necessary with lower outdoor air temperatures when the air-cooled condenser capacity increases and system load decreases, causing low condenser pressure. This is controlled by modulating air flow through the condenser with an outlet damper whose operator is driven by condensing pressure. In multifan con-

densers, cycling of the fans by outdoor air temperature thermostat provides step control of the air flow through the condenser. The last operating fan might have an outlet damper operated by condensing pressure.

In water-cooled condensers, water flow is usually controlled by a flow-modulating valve controlled by condensing pressure.

7.3.2.10 Hot-Gas Muffler

This muffler is usually installed on the discharge side of reciprocating compressors with long piping systems. This reduces gas pulsation and noise produced by reciprocating equipment.

7.3.2.11 Solenoid Valves

These valves are electrically operated two-position valves. They permit isolation of coil circuits to reduce the cooling produced and pumpdown of the low pressure side for eventual compressor shutoff when heat load is zero.

7.3.2.12 Sight Glasses

These should be installed in every system in front of the expansion valve. The operator can verify the flow of liquid and absence of gases or vapors upstream of cooling coils.

7.3.2.13 Shutoff Valves

These valves are usually the capped, packed, angle type mounted directly on the compressor or liquid receiver. The purpose of shutoff valves is to isolate portions of the refrigeration circuit to enable maintenance or repair.

7.3.2.14 Charging Valve

This is the point at which refrigerant is introduced (charged) into the system. Normally the charging valve is installed in the liquid line after the condenser or after the liquid receiver, if one is used.

7.3.2.15 Relief Valves and Fusible Plugs

These devices protect the refrigeration system from excessive pressure buildup. In the case of fusible plug activation, all refrigerant charge is released when the plug melts because of excessive temperature.

7.3.2.16 Check Valves

These are usually used in front of the liquid receiver and after the compressor, to prevent vapor migration from the receiver to the condenser or liquid migration from

the condenser to the discharge of the compressor, after the system shutdown. This is especially important in systems where the receiver is located in a hot space or the compressor is located in a space cooler than the condenser.

7.3.2.17 Strainers

These are installed in liquid lines to protect solenoid and expansion valves from dirt.

7.3.2.18 Liquid Receiver

When condensers (evaporative, air-cooled) which inherently have a small refrigerant storage volume are used and the system is sufficiently large, a liquid receiver is installed after the condenser to collect and hold the system liquid refrigerant until it is required by the peak load.

7.3.2.19 Pressure and Temperature Gauges

These are used to indicate suction and discharge compressor pressures and temperatures, condenser water temperature, and compressor lubricating oil pressure.

7.3.3 APPLICATIONS

Direct expansion air-conditioning systems are available as window-type units with capacities from less than $\frac{1}{2}$ ton (1.8 kW) to large packaged or built-up units of over 100 tons (352 kW) of refrigeration. They can be of self-contained construction (rooftop) where all elements, including the controls, are built in one cabinet, or they can be a split system where the condenser and compressor are located in one cabinet outside the building (condensing unit) and the DX coil and expansion valve are located inside the building in the air-handling unit. In the latter case, interconnecting refrigerant piping (liquid and suction lines) has to be field-installed and -insulated.

Capacities of split and self-contained air-conditioning systems (excluding window air conditioners) normally start at 2 tons (7 kW) and for self-contained systems range to 100 tons (352 kW), while split systems range to 120 tons (422 kW) of refrigeration.

Split and self-contained DX systems are normally used in situations where individual temperature and humidity control is required for numerous small spaces within a large building. Typical applications are apartment buildings, condominiums, small shopping malls (strip stores), office buildings with multiple tenants, medical buildings (doctors' and dentists' offices), and various departments within a manufacturing building. These systems are also used in nonair-conditioned plants where it is necessary to air-condition in-plant offices and spaces. In this case, packaged systems are frequently located on the roof of the in-plant space with heat rejection to plant space. These air-conditioning systems can be individually shut off when not needed, thereby saving energy. Also the systems can be individually metered to facilitate allocating the operating cost directly to the tenants or depart-

ments. In case of failure in one system, only the space being served will be affected whereas failure of a large, central built-up system would affect the entire building or buildings.

Typical use of DX system air conditioning is found in churches and restaurants where zoning of different spaces is important (different temperatures or times of use of different spaces, such as bar, kitchen, dining area, recreation hall, and church area).

The most common use of small DX systems is in residential spaces. Capacities of these systems start at 2 tons (7 kW) of refrigeration, which is enough for a small home. When two or more small DX systems are installed in larger homes, separate zones with independent temperature control and operating periods are established, e.g., sleeping areas which are cooled only at night, living quarters cooled only during the day, rooms with west exposure cooled only in late afternoon or evening, etc.

Special fields of use of DX cooling systems include computer rooms and vehicles. Self-contained, water-cooled condenser units are often used for computer rooms. They are normally designed for recirculation air only with bottom (under floor) discharge and sized to handle large sensible loads. However, computer rooms which have uniform heat release throughout the room can be conditioned by ceiling air distribution systems. For transportation vehicles (subway cars, public buses, and cars), modular systems with air-cooled condensers are used. All three major parts of the system (condenser, evaporator, and compressor) are in different locations in the vehicle and are connected with insulated piping.

Split systems are applicable as a retrofit or an option to standard air-handling units. A typical case is a residential furnace where space for a future DX coil is provided.

7.3.4 DESIGN CONSIDERATIONS

During initial design development, the designer must consider the type and function of facility to be air-conditioned, cooling loads involved, building layout, provisions for future expansions, and degree of required temperature and humidity control. If the designer decides that a DX system is suitable for the project, the next steps include evaluation of available condenser cooling media, type of system to be used, and location of the condenser and air handler if the system is comprised of multipackage units.

The simplest approach is to provide an air-cooled, single-package, rooftop-mounted air-conditioning unit. This system is completely self-contained including controls, so that the designer has only to connect ductwork to the unit and to bring in electric power and thermostat wiring. With restrictions on water use and the high cost of water in many areas of the country, air-cooled condensers have long been popular.

In general, air-cooled condensers have lower initial cost, they are lighter, maintenance is easier, and there is no liquid disposal problem. However, there are certain disadvantages and design considerations that the designer has to recognize before choosing a type of condenser. Air-cooled condensers require large amounts of relatively cool air, which could be a problem, especially with an indoor location of the condenser. Axial-flow fan condensers can be noisy. They require relatively clean air (condenser plugging problem). Startup difficulties at low outdoor air temperatures, capacity reduction on high outdoor temperatures, and operating problems at

part load are common problems. Air-cooled condensers require locations free of any obstructions on both inlet and outlet sides. Usually clearance of 1.5 times the condenser height is required around the condenser. If a possibility of air short-circuiting (recirculation of hot air) occurs, the designer should consider condenser fan discharge stacks. Since the north side of the building is cooler and is in shade for most of the day, the condenser should be located in this area, if possible.

When a system operates for a longer period on minimum load, the suction pressure drops, as does the corresponding temperature. This can result in frost or ice on the cooling coil, restricting air flow through it. Also reduced refrigerant flow through the system may cause compressor lubrication problems and motor cooling problems in hermetic compressors.

In general, capacity control in a reciprocating compressor DX system is a problem. Control is achieved in steps, either with multiple-compressor arrangements or by compressor valve control (unloading compressor cylinders). In any case, this is step (nor modulating) control, therefore, precise temperature control cannot be expected from DX systems. For more precise capacity control, multispeed and variable-speed motors are usually considered.

Temperature and humidity control can be achieved with parallel- and series-arrangement DX coils. A parallel coil arrangement is less expensive and provides better humidity control, but maintenance of constant leaving air temperature is difficult. Therefore, parallel coil arrangements are not recommended for reheat air distribution systems where a constant air temperature in front of reheat coils is important. Coils arranged in series are usually split to carry half the capacity each and are connected to separate compressors of the same capacity (two circuits). This division is done so that the first coil has one-third of the total number of rows and the second coil has the remaining two-thirds, because the first coil has greater air temperature differences and still will carry one-half of the total cooling load. The disadvantage in this arrangement is that one compressor (the one connected to the upstream coil) is always leading on load demand and is therefore wearing faster.

Air velocity through the cooling coil is limited to 550 ft/min (2.8 m/s) maximum because of condensate moisture carryover from coil fins.

Part-load system operation can increase lubricating oil migration problems. On long vertical piping runs, this is solved with double-riser piping arrangements, as discussed earlier.

If a split system is selected, the designer must consider the distance between the condensing unit and DX coil. This distance is limited to 50 ft (15 m) total length of piping for hermetic compressors of 20 tons (70 kW) of refrigeration capacity and under and to 150 ft (46 m) for semihermetic compressors with capacity of over 20 tons (70 kW) of refrigeration.

When modular systems are used, compressor vibration and noise factors must be recognized. These disadvantages can be mitigated by installing vibration isolators under the compressor and by providing muffler and flexible connectors at the compressor. Piping flexibility can be improved by using two or three 90° elbows in the piping near the compressor.

If the air-conditioning unit is not easily accessible, remote panel indication of air filter status, different pressures, and temperatures should be considered.

All equipment requiring maintenance should be provided with manual shutoff valves.

Some municipalities require licensed operators for compressor motors above certain sizes. This can be avoided by use of multiple compressors of smaller size.

7.3.5 REFERENCES

ASHRAE Handbook, HVAC Systems and Equipment, 1996, Ch. 5, 21, 34, 35, The American Society of Heating, Refrigeration and Air-Conditioning Engineers, Inc. Atlanta, GA.

P · A · R · T C

GENERAL CONSIDERATIONS

CHAPTER 8.1

AUTOMATIC TEMPERATURE, PRESSURE, FLOW CONTROL SYSTEMS

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8.1.1 CONTROL BASICS

8.1.1.1 Control Systems

Elements of Control Systems. Control loops consist of several elements and are used to match equipment capacity to load by changing system variables. Figure 8.1.1 is a block diagram of a control loop and shows the relation of the elements.

The controlled variable is the condition being controlled; for HVAC systems this is typically temperature, humidity, or pressure. A sensor is the device that measures a variable and transmits its value to the controller. The controller compares the value of the variable with the set point or desired value, and outputs a signal based on the difference between the variable and the set point.

The final control element responds to the controller signal and varies the *manipulated variable*. *Control elements* may be valves, dampers, electric relays, or electronic motor speed controllers, and manipulated variables may be air, water, steam, or electricity. The *process plant* is the equipment being controlled and whose output is the controlled variable. It may be a coil, fan, steam generator, or heat exchanger.

Types of Control Loops. There are two basic types of control loops, *open loop* and *closed loop*. With open-loop control, the system sensor measures a variable external to the system yet has some relation to the controlled variable. An example is sensing outdoor temperature to control heat flow into a building to maintain indoor temperature. Thus, a fixed relationship between outdoor temperature and required heat input is assumed and the control system programmed accordingly.

*The parts of this chapter covering boilers, refrigeration, central plants and building management systems were written by Donald H. Spethman for the first edition and were updated by Edward B. Gut for this edition.

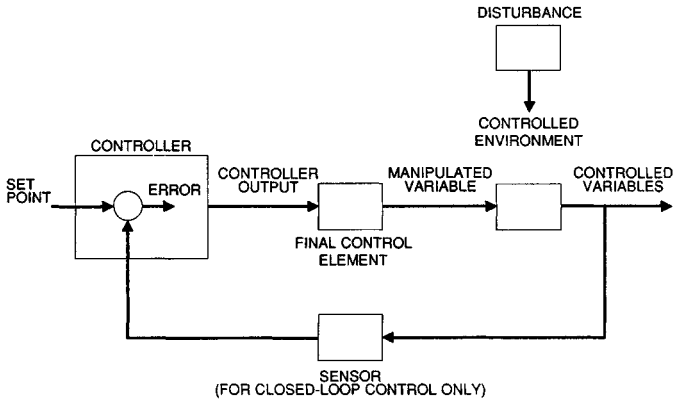


FIGURE 8.1.1 Basic elements of a control loop.

Closed-loop control pertains when the system sensor measures the controlled variable, resulting in variations in the manipulated variable to maintain the desired value of the controlled variable. Closed-loop control is also called “feedback control,” and results of a corrective action are fed back within the controlled system, therefore providing true control of the controlled variable.

8.1.1.2 Modes of Feedback Control

Feedback-controlled systems are categorized by the type of corrective action a controller is designed to output. For all types, the set point is the desired value of the controlled variable to which the controller is set. The control point is the actual value of the controlled variable as maintained by the controller’s action.

Two-Position Control. The final control element may be in one or the other position, i.e., maximum or minimum, except for the brief time when it changes positions. There are two values of the controlled variable which establish the position of the controlled element: set point and differential. Differential is the smallest range through which the controlled variable must pass to move the control element from one position to the other. Figure 8.1.2 shows a temperature controller or thermostat with a 70°F (21.1°C) set point. At 70°F (21.1°C) this electric thermostat would open its contacts and stop a burner. For the thermostat contacts to close, turning on the

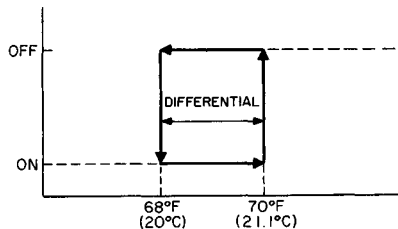


FIGURE 8.1.2 Two-position control.

burner, the temperature must drop below the set point by the amount of the differential, 2°F (1.1°C) in this example. Differential may be subtracted from or added to set point depending on controller design.

Two-position control is a low-cost device and provides acceptable control of slow-reacting systems that have minimum-lag between controller outputs and control-element response. Fast-reacting systems may overshoot excessively and be unstable. Examples of two-position control are domestic hot-water heaters, residential space-temperature controls, and HVAC system electric preheat elements.

Timed Two Position Control. The final control element may be in one of two positions, as for a two-position control, but a timer is incorporated in the controller so that it responds to the average value of the controlled variable rather than the peak fluctuations. Timed two-position control greatly reduces the variations or swings in the control variable by anticipating controlled-variable changes due to control-system action.

A typical example of timed two-position control is residential space-heating temperature control. The thermostat has an electric heating element that is energized during the on period. The heat from the element warms the temperature sensor more quickly than the rising space temperature, shortening the on time and reducing temperature overshoot. During the off period the sensor heater is also off, allowing the sensor to respond directly to space temperature. This results in a relatively constant cycle time with a variable on-off ratio dependent on space load.

Timed two-position control is low-cost and may be applied to slow-reacting systems that have some lag between controller output and control-element response. The timer will anticipate the response and minimize variations in the controlled variable.

Proportional Control. A *proportional controller* has a linear relationship between the value of the incoming sensor signal and the controller's output. The relationship is generally adjustable in the controller but once adjusted remains fixed during operation. There is therefore only one value of the final control element for each value of the controlled variable within the operating range of a proportional control system.

The variation in control variable required to move the final control element through its operating range is the throttling range of the control system and is expressed in the measuring units of the controlled variable. The variable in the sensor signal required to operate a proportional controller through its range is called the "proportional band" and is expressed as a percentage of sensor span.

The "set point" of a proportional controller is defined as the sensor input which results in the controller output at the midpoint of its range.

"Offset" is the difference between the set point and the controlled variable at any instant. Sometimes offset is also referred to as "deviation," "droop," or "drift." Offset results from the fixed linear relationship between control input, sensor signal, and output. Therefore, under full-load conditions, control input must be offset by one-half the proportional band for the controller to output a signal at one extreme of its range. Similarly, at minimum load the offset will be one-half the proportional band (see Figure 8.1.3).

Proportional control is used with slow stable systems, allowing narrow throttling ranges and therefore small offset. Fast-reacting systems require large throttling ranges to avoid instability and cycling of the controlled variable. This, of course, increases offset.

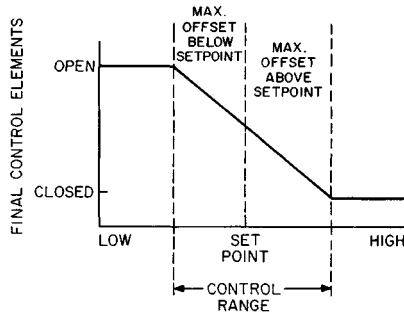


FIGURE 8.1.3 Proportional-controller offset.

Proportional-plus-Reset Control. A proportional-plus-reset controller has proportional action plus an automatic means of resetting the set point to eliminate offset. This controller action is also called “proportional-plus-integral,” or PI control.

A PI controller’s initial output signal has a fixed relationship to a changed sensor input signal, the same as a proportional controller, but then continues to change until the control variable equals set point. The rate at which this additional change occurs is called the “reset rate” or “repeats per minute” and is the number of times the original proportional change in controller output is repeated per minute. The reset rate may also be expressed as reset or integral time, which is the amount of time for the controller to change its output as much as the first proportional change.

PI control may be applied in fast-acting systems that require large proportional bands for stability but where the resultant offset between set point and control point is undesirable due to comfort and/or energy-conservation considerations. Typical applications are mixed-air, duct-static, chiller-discharge, and coil-discharge control.

Proportional-plus-Rate Control. Also called “proportional-plus-derivative,” or PD, control, this control mode adds to proportional control an automatic means of varying controller output based on changes in deviation or difference between the set point and the control variable.

When deviation increases, rate action adds to the controller’s output, causing the final control element to respond an additional amount to stabilize the controlled variable more quickly than proportional control alone can. Conversely, when deviation decreases, rate action subtracts from the controller’s output. When there is no change in deviation, rate action stops and the deviation is determined only by the proportional band of the controller.

Proportional-plus-Integral-plus-Derivative (PID) Control. This combination of control modes is useful for controlling fast-acting systems that tend to be unstable, such as duct static-pressure control. For these applications, the controller may be set with a large proportional band for system stability, a slow reset to eliminate deviation from set point yet retain stability, and derivative action to speed control response when the system is upset due to changes.

Adaptive Control. A PID controller must be properly tuned to the system it is controlling to achieve stable and accurate control. The proportional, integral, and derivative parameters of the controllers are dependent on the characteristics of the system being controlled and can be time-consuming to establish. Even then they

will be optimum for one operating condition and compromised for the remaining operating range.

Adaptive control is the ability of a controller to adapt to the system it is controlling by determining the ideal PID parameters and adjusting itself according. Two types of adaptive control have been developed, self-tuning and model reference.

A *self-tuning controller* begins with initial PID parameters. With input from the controller's output and the control-variable value, it establishes new parameters. After a few cycles of control-system operations, the controller determines the optimum parameters. The process continues as the system operates, so every time the system changes, the controller reestablishes the parameters so that they are optimum for every condition of system operation.

The *model-reference controller* compares its output with that of a fixed model and develops the PID parameters to achieve control-system operation for the model. While the model may not be exactly the same as the actual system, it is very close and allows the controller to develop the parameter values quickly.

Floating Control. A floating control outputs a corrective signal when the difference between the set point and sensor signal is greater than a set amount or differential. The output signal will increase or decrease a final control element depending on if the controlled variable is below or above set point. If the difference is less than the differential, the controller output is zero and the final control element remains in the position it was last driven to. Floating controls may be applied to systems that react quickly with little lag and have slow load changes.

Time-Proportioning Control. Time proportioning is a method of controlling loads like electric heating elements. The final control element is either on or off, but the ratio of on-to-off time is varied depending on system load, therefore varying the energy inputs.

The sum of on and off time, or the total time per cycle, is constant. Time-proportioning control is also called "average-position control" and is a relatively low-cost way to simulate proportional control.

8.1.1.3 Flow-Control Characteristics

Flow Control. Proper volume or flow control of one form or another is essential to the successful operation of most HVAC systems. Usually the flow of water, steam, and/or air is controlled to modulate system outputs or capacity as required by changing loads. As in other control loops, a sensor measures the control variable and a controller compares the sensor signal to a set point and outputs a corrective signal as required to a final control element. For water and steam flow the final control element is a valve, and for air flow a damper.

The flow-control characteristics of valves and dampers are designated in terms of the flow versus opening based on a constant pressure drop across the element. The three common characteristics are quick opening, linear, and equal percentage.

As shown in Fig. 8.1.4, quick opening provides for more percentage of full flow than when the valve or damper is opened. Linear characterization has the same percentage of full flow as when the valve is open, while equal percentage increases flow by an equal percentage over the previous value for each equal increment of opening. In other words, a 10 percent change in opening from 20 to 30 percent increases the flow by the same percentage of flow at 20 percent opening as an

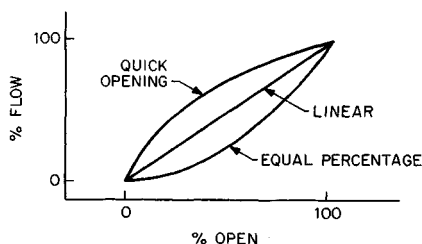


FIGURE 8.1.4 Flow-control characteristics.

increase in opening from 70 to 80 percent would increase flow from the 70 percent opening position. These different characteristics are required to match the control needs of water, steam, and air flow.

Pressure drop across a valve or damper in a system rarely stays constant. Therefore actual opening-flow characteristics vary from manufacturer's ratings, which are based on constant pressure drop. The amount of this variation depends on how much the pressure drop changes and is determined by overall system design. The pressure drop is minimum when the valve or damper is full open and increases as the valve or damper closes. When fully closed, the entire pressure drop is across the valve or damper.

For the valve or damper to provide approximately its design characteristic, the design or full-open pressure drop should be a fairly large percentage of the total system drop. As a high pressure drop consumes energy, consideration should be given to design or control a system to provide a more constant pressure drop, allowing the valve or damper to be sized for a lower pressure drop at full flow.

Control of Water Flow. One of the primary uses of water-flow control is to modulate the capacity of a heating or cooling coil. However, the capacity of a coil is not linear with water flow; instead, as the flow is reduced, more energy is transferred from the water, partly offsetting the reduction in flow. Figure 8.1.5 shows the relationship of capacity versus flow for a heating or cooling coil. This nonlinearity is primarily a consideration with hot-water coils due to the large temperature difference between the water and air flow through a coil. For hot-water coils, this nonlinear variation may be reduced by designing the coil for a higher water-temperature drop or by reducing water temperature as system load decreases.

Since hot-water coils have a significantly nonlinear relationship between heat transfer and water flow, equal-percentage valves are used for coil water-flow control, resulting in a more linear relationship between valve position and coil heat output.

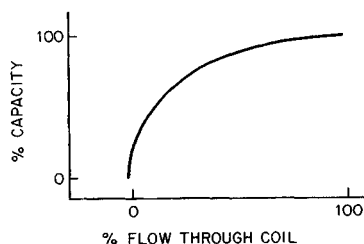


FIGURE 8.1.5 Heating- or cooling-coil capacity versus flow.

The capacity of a water valve is a function of valve design and the pressure drop across the valve and is independent of the supply pressure. Valve capacity is rated by a flow coefficient, or C_v , which is defined as the amount of water in gal/min (m^3/h) that will flow through an open valve at 1 lb/in² (1 atm, 101.325 kPa) pressure drop. For valves in systems, the pressure drop increases as the valve closes, offsetting part of the desired flow reduction. To minimize this, valves should be sized so that they constitute about 25 to 50 percent of the system resistance that a valve controls.

Valve pressure-drop changes can be minimized by providing a system bypass valve to maintain total system flow even when control valves close. Also system flow may be modulated by an automatic flow-control valve in series with the pump, or the pump may be operated at varying speeds based on system pressures near the far end of the piping circuit.

Control of Steam Flow. Control of steam flow is usually applied to modulate the heat output of a steam-to-water and steam-to-air heat exchanger.

For one-pipe steam systems, line-size two-position valves are used to ensure proper flow of steam and simultaneous drainage of condensate. Two-pipe steam systems may be controlled by two-position or modulating valves which must be sized properly for good control. Since output of a steam heat exchanger is linear with steam flow, valves with linear flow-opening characteristics should be used for modulating control.

The capacity of a steam valve is determined by valve design, the pressure drop across it and the inlet pressure. Valves for two-position applications are sized to provide the required full flow with minimum pressure drop and to be able to close against system pressure.

Modulating steam valves must be sized to *only* full-load flows, which may be less than full heat-exchanger flow, to avoid system instability due to excessive capacity. Since steam valve capacity depends on pressure drop and inlet pressure, it is important that valve inlet and outlet pressures are kept fairly constant to maintain a linear relationship between valve opening and heat-exchanger output.

Supply pressures can be controlled by automatic pressure-reducing valves in the supply lines or by a narrow differential controller. The effect of variations in return pressures can be minimized by sizing the valve so that the outlet pressure is near its minimum value or at a pressure resulting in critical velocity in the fully open valve, whichever is higher.

Critical velocity in a valve is the velocity at which an increase in pressure drop will not result in an increase in velocity or flow through the valve. This occurs when outlet pressure is about 58 percent of inlet pressure. For some applications with large-capacity modulating, two steam valves in parallel may be used for better full-range control. The valves should be sized so that one valve has about one-third full-load capacity and the other valve about two-thirds full-load capacity. The valves are operated in sequence so that the smaller valve controls during low loads and the larger valves operates when the smaller valve is fully open.

Control of Air Flow. Air flow in HVAC systems is controlled in an *on-off mode* or *modulating mode*. The on-off mode is generally used to allow outside air into a building when desired such as during occupied times and to prevent outside air from entering at other times. Modulating air flow is used to blend air from more than one source to achieve a desired temperature or to vary the volume of air delivered to match load requirements.

Dampers are used to control air flow and are produced in two basic designs, parallel-blade and opposing-blade configurations (see Fig. 8.1.6). The opening-flow

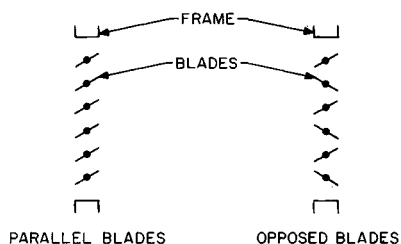


FIGURE 8.1.6 Damper-blade configurations.

characteristics of these configurations for constant pressure drop and for various ratios of system pressure drop without the damper to damper pressure drop at full open flow is shown in Figs. 8.1.7 and 8.1.8.

However, as with valves, dampers installed in systems have varying pressure drops as they modulate, being minimum when full open and maximum when closed. For two-position applications, dampers should be selected on the basis of full-flow pressure drop, leakage, and closed-pressure differential ability. Modulating characteristics are not important.

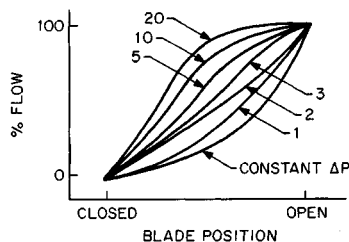


FIGURE 8.1.7 Characteristics of parallel-blade dampers. Curves other than the constant- ΔP curve represent ratios of system pressure drop to open-damper pressure drop at full flow.

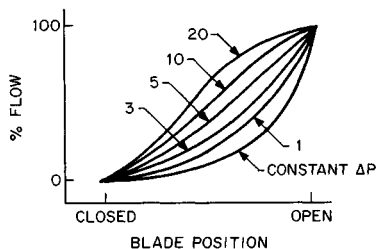


FIGURE 8.1.8 Characteristics of opposed-blade dampers. Curves other than the constant- ΔP curve represent ratios of system pressure drop to open-damper pressure drop at full flow.

Typical modulating applications are mixed air, face and bypass, and volume control. Mixed air, or control of outside, return, and exhaust air, requires the coordination of three dampers for modulating outside and return air to maintain a constant supply volume and for modulating exhaust-air volume as outside air varies. Face and bypass control is used to vary the amount of air through and around a coil to vary the temperature of the total air flow after the coil. The face damper controls air flow through the coil, and the bypass damper the air flow around the coil. The dampers are arranged so that when one opens the other closes, and the sum of the air flow through both dampers is constant. To achieve this relationship it is important that both dampers are selected for linear control.

Volume control of air flow may be used to maintain static pressure in a duct or space or to match space- or zone-conditioning needs. Variable air flow is achieved by changing duct system resistance to air flow or by diverting air flow through an alternative or bypass route. Dampers should be selected to provide equal changes in air flow for equal changes in control variables, which may be temperature, pressure, or flow volume in these specified systems, for stable control over the full operating range.

8.1.2 CONTROL EQUIPMENT TYPES

The elements of a control loop are divided into four categories: sensors, controllers, final control elements, and auxiliary equipment, and may be pneumatic, electric, or electronic.

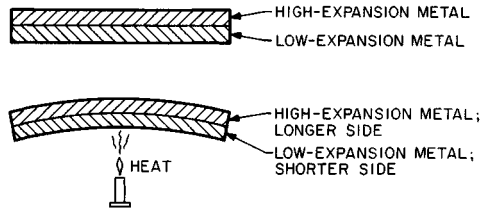
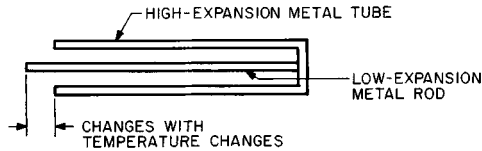
8.1.2.1 Sensors

The controlled variable of a system is measured by a *sensor*. A sensor output signal, whether pneumatic or electric, may change electrical resistance depending on the value of the sensed variable. The usual pneumatic sensor-signal range is 3 to 15 lb/in² (20.7 to 103.4 kPa), while electric sensors output 2 to 10 V dc or 4 to 20 mA. Resistance sensors have a nominal resistance of 500, 1000 and 2000 Ω . Temperature-sensing elements are usually bimetal, rod and tube, sealed bellows, and resistance.

Bimetal is the oldest and most common type of temperature-sensing element. Its operation is based on the principal that the change in size with the change in temperature is different for different metals. Combining two metals, one with a large expansion coefficient and one with a small coefficient, into a strip, the strip will deflect with temperature changes due to the different amounts of expansion (see Fig. 8.1.9). The amount of deflection is proportional to temperature and can therefore be used to measure or sense temperature and generate a proportional pneumatic or electric signal. Bimetal strips may be used as straight elements or may be U-shaped or spiral-wound depending on the space available and the temperature-deflection characteristics desired.

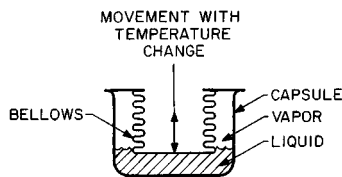
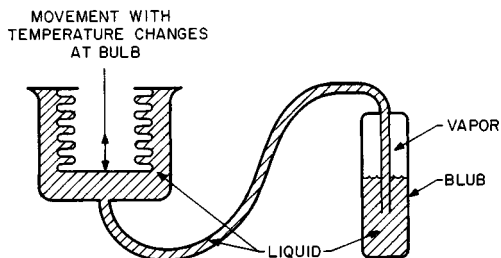
Rod-and-tube elements also use the different expansion rates of metals to generate movement with temperature changes. However, they are constructed with a low-expansion rod, high-expansion tube (see Fig. 8.1.10) and are usually used for insertion directly into the medium, such as water, steam, or air.

Two versions of the remote bulb element have a long, (10 to 20 ft or 3 to 7 m), capillary tube in place of the bulb. One is liquid filled and senses the average

**FIGURE 8.1.9** Bimetal strip.**FIGURE 8.1.10** Rod-and-tube element.

temperature over its length. The other is vapor filled and senses the coldest (12 in or 30 cm) along its length.

Sealed bellows (see Fig. 8.1.11) consist of a capsule and bellows evacuated of air and filled with a vapor or liquid. As a vapor or liquid changes pressure or volume with temperature changes, the bellows moves, providing an indication of sensed temperature. A variation of sealed bellows is the remote-bulb element (see Fig. 8.1.12). A bulb is attached to the bellows assembly by a capillary tube so that

**FIGURE 8.1.11** Sealed bellows.**FIGURE 8.1.12** Remote-bulb element.

temperature changes at the bulb result in pressure changes which are transmitted to the bellows, resulting in movement corresponding to temperature at the bulb.

Resistance elements consist of an element with a known temperature life vs. resistance characteristic. The element may be wire, wound on the bobbin, or a thermister, which is a semiconductor, or a stable metal line platinum, plated on a ceramic base, whose resistance depends on temperature.

Humidity-sensing elements are hydroscopic or electric. The hydroscopic elements are based on the fact that certain materials change size as they absorb or release moisture. Typical materials are hair, wood, leather or nylon, whose size changes due to moisture absorption or release based on the moisture content of surrounding air, will indicate humidity of the air. This size change is used to develop a pneumatic signal proportional to humidity or to turn an electric switch on and off.

Electric humidity-sensing elements are constructed to provide either a resistance change with ambient humidity changes or a capacitor change and are generally used with electronic controllers. They generally respond quicker than hydroscopic elements.

Dew-point sensors are constructed by winding two wires around a hollow tube impregnated with lithium chloride. The conductivity of the lithium chloride varies as it absorbs or releases moisture to the surrounding air. Electric power supplied to the two wires around the sleeve will flow through the lithium chloride at a rate depending on its conductivity, which varies with dew points. As the electricity flows through the wires, the temperature of the cavity of the tube is elevated and is a measure of dew point (see Fig. 8.1.13). The cavity temperature may be sensed with any temperature sensor that will fit inside the tube.

Pressure sensors may be high-range (psi or Pa) or low-range (in or cm of water). High-range sensor elements usually are Bourdon tubes, bellows, or diaphragms to provide movement based on pressure. Low-range pressure sensors generally use large slack diaphragms or flexible metal bellows to transduce low pressures into usable forces for indicating pressure. If one side of the element is open to the atmosphere, the element responds to sensed pressure above or below atmospheric. For differential pressure sensing, both sides of an element are connected to sense pressure variables. Outputs of pressure sensors may be pneumatic, electric analog, or electric on-off.

Pneumatic air-velocity sensors are of the differential-pressure or of the deflected-jet type. The differential-pressure types use a restriction in the air stream, such as an orifice plate, or sense static and total pressure to generate differential pressures that represent air velocity (see Fig. 8.1.14). The deflected-jet type has a small air jet flowing across the measured air stream from an emitter tube. The air is captured in a collector tube and generates a recovery pressure (see Fig. 8.1.15). When the velocity of the measured air stream is low, most of the air jet depinges on the collector tube and the recovery pressure is high. As the air-stream velocity in-

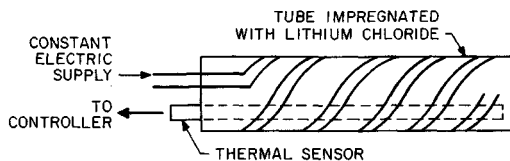


FIGURE 8.1.13 Dew-point sensor.

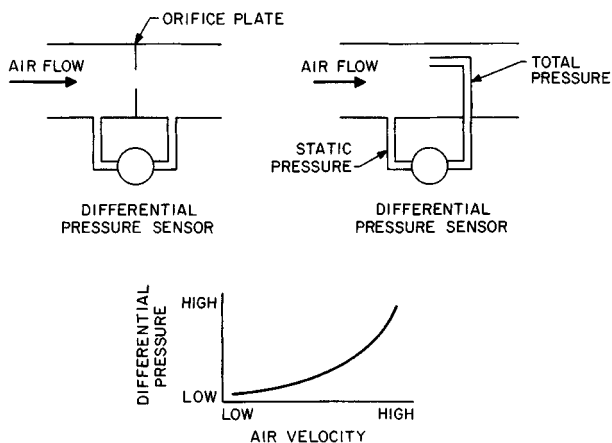


FIGURE 8.1.14 Differential-pressure air-velocity sensors.

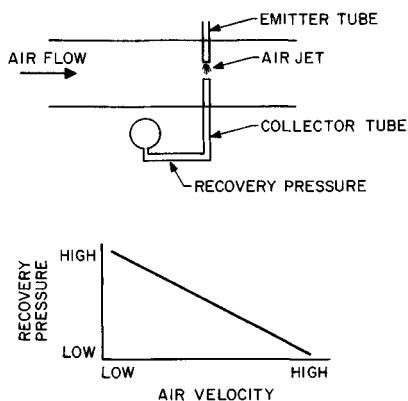


FIGURE 8.1.15 Deflected-jet air-velocity sensor.

creases, the air jet is deflected and recovery pressure diminishes. The recovery pressure is, therefore, a direct indication of air-stream velocity.

Electric air-velocity sensors use a heated wire or thermistor placed in the air stream. The amount of current required to maintain the wire or thermistor temperature varies with the cooling effect of differing air velocities and, therefore, is a measure of air velocity. A reference wire or thermistor shielded from the air stream compensates for varying air temperatures. The sensor may be solid state with all sensing elements on a chip.

Water-flow sensors may be differential-pressure types, such as orifice plates, pitot tubes, or flow nozzles, that have limited range or vortex-shedding, turbine, or magnetic types that have greater range but are more expensive.

Other sensing elements including smoke and high-temperature detectors, specific-gravity, current, CO, and CO₂ sensors are often used for complete control of HVAC systems.

8.1.2.2 Controllers

Controllers provide the set-point, and for some, the proportional-band, integral, and derivative parameters of a control loop. They compare the sensor signal with the set point and output a corrective signal as determined by the controlled settings. This signal may be *direct-acting*, increasing with sensor-signal increases, or *reverse-acting*, decreasing with sensor-signal increasing. Controllers may incorporate a sensing element for sensing and controlling in one device. Proportional controllers may also be designed to use remote sensors and are called sensor-controller systems.

Controllers may be pneumatic or electric powered. Pneumatic controllers receive a sensor signal and output a proportional signal typically 3 to 13 lb/in² (20.6 and 270 kPa). The controller may be a nonrelay or relay type. Nonrelay types use a restricted supply air, bleeding varying amounts to the atmosphere to generate a corrective output signal (see Fig. 8.1.16). Since the capacity of the output signal is restricted, amplification should be limited to small volume-control elements or where long response times are acceptable. Relay-type controllers incorporate a capacity amplifier for the corrective signal for greater output volume.

Electric controllers also may have integral or remote sensors. Outputs are two-position to cycle equipment, floating to open, hold, or close a final-control element or proportioning to position a final-control element. Proportioning electric controllers may be analog or digital.

Electric analog controllers are similar to pneumatic controllers. That is, their response to a sensor signal is fixed by their design and only by their parameters, such as set-point, direct- or reverse-action, proportional-band, and if included integral- and derivative-timing, are adjustable. Digital controllers are microprocessor-based, and their response to a sensor signal is programmable. This provides great flexibility for the application of a digital controller and allows control strategy changes after installation.

Digital controllers measure signals from sensors, perform control routines in software programs, and take corrective action in the form of output signals to actuators. Since the programs are in digital form, the controllers perform what is known as direct digital control (DDC). Microprocessor-based controllers can be used as stand-alone controllers or they can be incorporated in a building management system utilizing a minicomputer or a personal computer (PC) as a host to provide additional functions. A stand alone controller can take several forms. The simplest generally controls only one control loop while larger versions can control

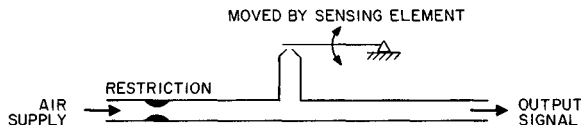


FIGURE 8.1.16 Nonrelay pneumatic controller.

from eight or ten to 30 or 40 loops. As the systems get larger, they generally incorporate more programming features and functions.

Pneumatic and electric controllers may also provide indication and/or recording of the value of the sensed variable for visual checks or for a history of system operation. Transducers may be used with controllers to convert sensor signals and controller outputs from pneumatic to electric, or vice versa, as required by the controller or final controlled elements.

8.1.2.3 Final-Control Elements

Final-control elements are valves, dampers, electric heaters, relays, and motors for fans, pumps, burners, refrigeration, and other HVAC equipment. All these elements may be operated on-off or two position, while valves and dampers and motors may also be used with floating-control and proportional-control modes. Final-control elements may be normally open, that is, open with no controller signal, or normally closed.

Pneumatic valve and damper operators have a flexible diaphragm or bellows attached to a valve stem or damper linkage (see Fig. 8.1.17). Movement is opposed by a compression spring, while a pneumatic controller signal is connected to the operator and generates a force depending on the pressure of the signal in the area of the diaphragm. When the signal pressure multiplied by the area exceeds the force of the spring, the operator moves, also moving the valve or damper until the spring force and controller's signal generated force are in balance. When the controller signal reduces, the spring causes the operator to retract. By selection of springs, various operator position-controller signal characteristics can be attained. Since operator position depends on the balance between the diaphragm and spring force, any external force from a valve or damper will offset the operator. For control systems requiring accurate synchronization of final-control elements, this may be a problem. For precise positioning, a positive positioner is used. It senses controller input signal and operator position and feeds or bleeds air to or from the operator to position it regardless of external load.

Electric motors are *unidirectional*, *spring-return*, or *reversible*. Unidirectional motors are for two-position operation: opening a valve or damper in half a revolution and closing it in the second half. Once initiated, the motor continues through half a revolution. When it receives a second signal, the controller continues through the next half-revolution cycle.

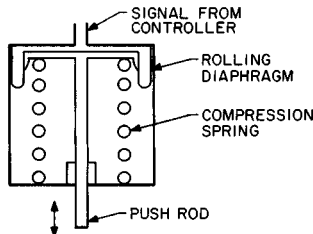


FIGURE 8.1.17 Pneumatic valve or damper operation.

Spring-return motors are also used for two-position operation. A control signal drives the motor to one end of its movement and holds it there. When the controller is satisfied and ends its output, the motor is driven back by an internal spring which was wound during its initial movement.

Reversible motors are used with floating- or proportional-control modes. The motor can be operated in either direction, depending on the controller signal; it stops when the signal stops. For proportional control, a potentiometer on the motor shaft is used to signal the motor position to the controller.

8.1.2.4 Auxiliary Equipment

Many control systems require auxiliary equipment for complete system operation. For pneumatic control systems, these include:

- Compressed air systems with compressors, dryers, and filters to provide clean dry air at the proper pressures to power the system
- Pneumatic-electric relays for switching electric loads with pneumatic signals and electric-pneumatic relays for switching pneumatic lines with electric signals
- Two-position relays for converting proportional pneumatic signals to two-position and proportional relays for reversing signals, selecting the higher or lower of two or more signals, averaging two signals, adding or subtracting a constant from a signal, and amplifying signal pressure or air-flow capacity
- Switching relays to divert signals automatically or manually
- Gradual switches to manually vary air pressure in a circuit

Electric systems utilize transformers to provide required voltage, relays to switch electric loads larger than a controller's capacity, potentiometers for manual positioning of proportional control devices or for remote set-point adjustments, manual on-off switches, and auxiliary switches on dampers and valves for control of sequence operation.

Other auxiliary devices are common to pneumatic and electric systems. These include step controllers for operating a number of electric switches by a proportional operator to control stages of electric heating or refrigeration. Power controllers may be solid-state, saturable-core, or variable autotransformers and are used to control electric resistance heaters with a proportional pneumatic or electric control signal. Clocks and timers are used to control apparatus or control-system sequences based on time of day or elapsed time.

8.1.2.5 Pneumatic, Electric, Electronic Comparisons

A pneumatic control system may be as shown in Fig. 8.1.18. Advantages of pneumatic controls are the inherent modulation sensors and controller signals and the low cost of modulating operators. The system is explosion proof, and the control elements require little maintenance and are easy to troubleshoot. Disadvantages are the need for an air-compressor system which may be too expensive for small systems. Compressed air must be piped to all controls, increasing installation costs, and transducers are required for interfacing to automation systems.

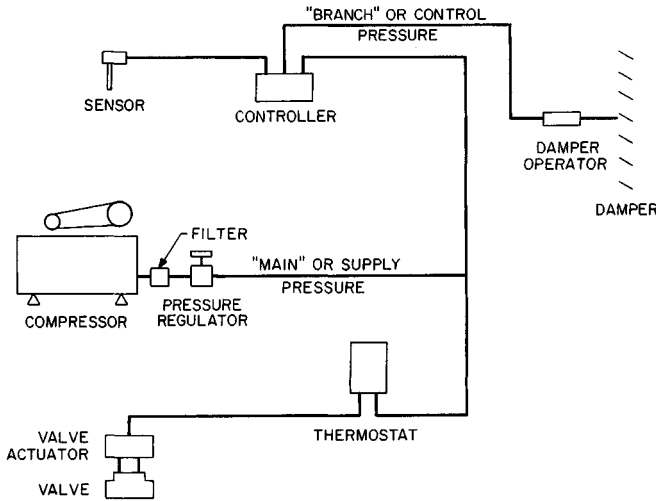


FIGURE 8.1.18 Pneumatic control system.

Electric control systems, as shown in Fig. 8.1.19, can be installed wherever electric power is available and are low-cost for small simple systems. However, modulating operators are expensive, and explosionproof housings are required in hazardous areas. Electronic modulating controls allow remote sensors and set-point adjustment, provide high accuracy, and readily interface with automation systems. They are higher in cost and require more skilled personnel for trouble-shooting.

Direct digital controllers (see Fig. 8.1.20), offer many advantages. They have very high accuracy, so control-loop accuracy is limited only by the sensor and final-control element. They are capable of complex control algorithms which may easily be changed by reprogramming. This allows flexible building operation during construction, startup, occupancy, full-occupancy, and expansion phases. An entire building can be controlled from one location, and building-wide energy-management strategies can be accomplished. Direct digital controllers are higher in cost but can control multiple loops and share sensors.

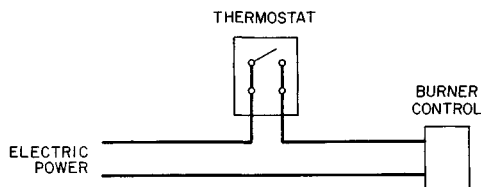


FIGURE 8.1.19 Electric two-position control system.

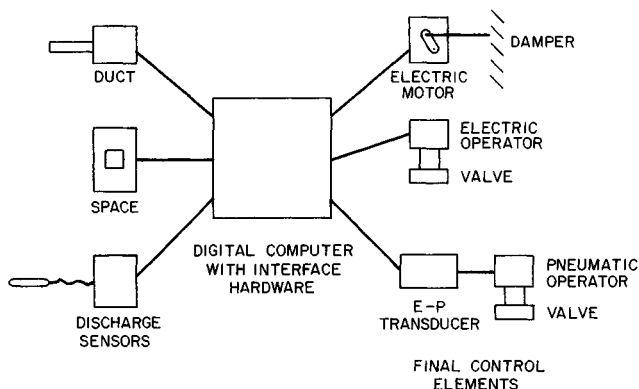


FIGURE 8.1.20 Direct digital control system.

8.1.3 CONTROL APPLICATIONS

These are defined by general functions in the total HVAC system.

8.1.3.1 Boiler Control

The control of steam or water boilers involves three types of functionality: flame safeguard, load control, and control of excess air. Steam boilers also include water-level control. The means of accomplishing each of these functions is influenced by the size of the boiler involved. In general, small boilers have a single control package which accomplishes flame safeguard and load control with no need for excess air control. Large boilers can have different control packages for all three functions. The application of flame-safeguard control is very dependent upon boiler and burner design and therefore is normally supplied by a complete package by the boiler-burner manufacturer. The type of fuel(s) selected, the size of design load, and the type of approval required are the primary decisions of the HVAC designer that establish the type of boiler controls that are appropriate. On larger-size installations, the method of load control, the use of multiple boilers, and the cost effectiveness of appropriate types of excess air control are additional considerations for the HVAC designer. This section explains means of accomplishing the three basic types of boiler-control functionality. It also explains an auxiliary function of monitoring smoke control.

Flame-Safeguard Control. The objective of flame-safeguard control is to ensure that safe conditions exist for initiating and sustaining combustion. On small- to medium-size [up to 400,000 Btu/h (422,000 kJ/h)] burners, the flame-safeguard-control function is provided by a package called a primary control. The primary control starts the burner in the proper sequence, proves that combustion air is available, purges the combustion chambers and proves the burner flame is established, and supervises the flame during burner operation. It causes safety shutdown on failure to ignite the pilot or main burner or on loss of flame. In addition, the primary control checks itself against unsafe failure. Typically, a check for flame-simulating

CHAPTER 8.2

NOISE CONTROL

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8.2.1 INTRODUCTION

Is noise control engineering a science or an art? It is a bit of both.

Acoustic theory helps explain the acoustic world we live in and enables us to establish general design parameters for engineered noise control solutions and products, but it does not always do so very accurately. For instance, it is impossible to calculate the noise reduction of barriers, walls, enclosures, rooms, and silencers or the propagation of sound waves over open surfaces with the degree of accuracy needed for reliability. There are just too many variables. Consequently, we cannot rely on theory for more than directional indicators.

Optimum noise control solutions must therefore be based on engineered products with performance characteristics obtained from repeatable laboratory tests and/or extensive field data—because if we overdesign, it costs too much money, and if we do not adequately provide for noise control, we may have an unacceptable job. For critical jobs, where there are significant uncertainty factors, model testing is essential. This may include power plants, aviation terminals and test facilities, industrial factories, and air-handling units in high-rise buildings. Furthermore, apart from economics considerations, the structural, mechanical, aerodynamic, and thermodynamic engineering aspects of the solution to a noise control problem are often more complex than its acoustic components, so in many instances a multidisciplinary approach is essential.

This chapter is concerned primarily with basic acoustic engineering principles and how they can be applied to solve noise problems inherent in HVAC systems. The chapter first discusses the theory of sound, with emphasis on the acoustic engineering aspects, and then examines the nature of noise in HVAC systems and the means available for controlling noise.

8.2.2 THE NATURE OF SOUND

Sound is essentially the sensation produced through the ear by fluctuations of pressure in the adjacent air, and “noise” can be defined as sound that annoys, usually

because the sound pressure level is too high. High noise levels not only interfere with direct voice communications and electronically transmitted speech; they are also considered a health hazard in both the working and living environments.

Sound waves are propagated in air as compressional waves. Although compressional waves are generally caused by vibrations of solid bodies, they can also be caused by pressure waves generated by the gas discharge of a jet engine or the subsonic velocities in an air-conditioning duct. When these waves strike solid bodies, they cause the bodies to vibrate, or oscillate.

To illustrate what happens, we can think of sound being generated by a piston oscillating back and forth in an air-filled tube (Fig. 8.2.1). This action of the compressor causes the air molecules adjacent to the piston to be alternately crowded together (or compressed) and then moved apart (or rarefied). The oscillation generated by the piston in this manner is referred to as "simple harmonic motion." And as shown in Fig. 8.2.2, a plot of the piston displacement can be presented as a sinusoidal function; that is, the sound wave generated in its purest form for a

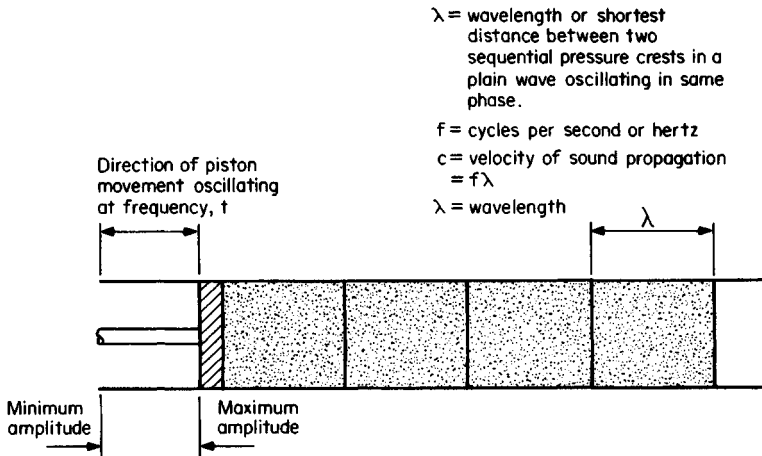


FIGURE 8.2.1 Sound wave being propagated through a compressible medium in a tube.

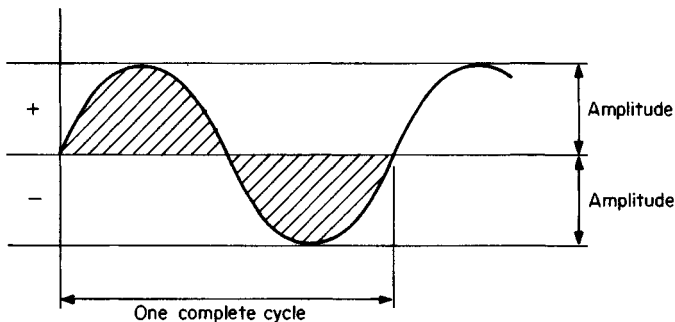


FIGURE 8.2.2 Sine wave of the simple harmonic motion characterizing a pure tone.

discrete sound is sinusoidal and has a frequency equal to the number of times per second that the piston moves back and forth.

8.2.2.1 Displacement Amplitude and Particle Velocity

Specifically, sound is transmitted through individual vibrating air particles. The vibration causes the particles to move, but they do not change their average positions if the transmitting medium itself is not in motion. The average maximum distance moved by individual particles is called the “displacement amplitude,” and the speed at which they move is called the “particle velocity.”

In air, the displacement amplitude may range from 4×10^{-9} in (10^{-7} mm) to a few millimeters per second. The smallest amplitude would be the lowest discernible sound, and the largest amplitude would be the loudest sound the human ear can perceive as a proper sound.

8.2.2.2 Frequency

The frequency of a sound wave is expressed in hertz (Hz). The range of human hearing extends from 20 to 20,000 Hz, but 12,000 to 13,000 Hz is the limit for many adults (and the exposure of teenagers to noisy rock music is likely to result in “old-age deafness” before they reach the age of 30). Figure 8.2.3 plots the threshold of hearing for young adults with normal hearing.

8.2.2.3 Wavelength

The wavelength of sound is the distance between analogous points of two successive waves. It is denoted by the Greek letter λ and can be calculated from the relationship

$$\lambda = \frac{c}{f} \quad (8.2.1)$$

where c is the speed of sound in ft/s (m/s), and f is the frequency in Hz.

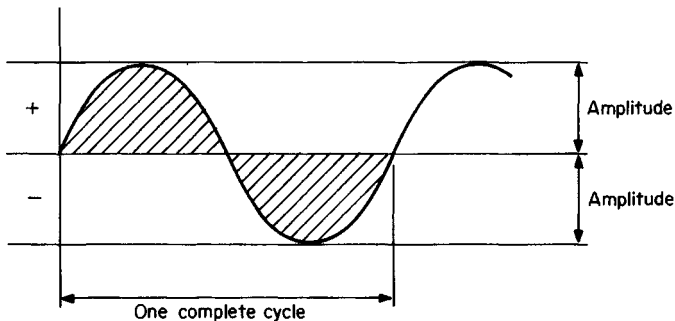


FIGURE 8.2.3 Threshold of hearing for young adults with normal hearing. [C. M. Harris (ed.), *Handbook of Noise Control*, 2d ed., McGraw-Hill, New York, 1979, part 1, p. 8–4.]

8.2.2.4 Sound Level

For convenience in measuring sound without having to take data at a large number of discrete frequencies, sound levels are often measured in one-third octave bands or in full octaves, as per Table 8.2.1. This table shows, for instance, that the one-third octave band with a center frequency of 63 Hz has a range from 56 to 71 Hz, while the corresponding full octave has a range from 45 to 89 Hz.

8.2.3 THE SPEED OF SOUND IN AIR

The speed of sound in air can be calculated from the expression

$$c = \begin{cases} 49.03\sqrt{460 + ^\circ\text{F}} & \text{in English units} \\ 20.05\sqrt{273 + ^\circ\text{C}} & \text{in metric units} \end{cases} \quad (8.2.2)$$

TABLE 8.2.1 Nominal One-Third Octave Band Center Frequencies and Ranges

Third octave band no.	Center frequency, Hz	Frequency range, Hz	Corresponding full octave band
14	25	22–28	Suboctave 22–45
15	—31.5—	28–36	
16	40	35–45	
17	50	45–56	1 45–89
18	—63—	56–71	
19	80	71–89	
20	100	89–112	2 89–178
21	—125—	112–141	
22	160	141–178	
23	200	178–224	3 178–355
24	—250—	224–282	
25	315	282–355	
26	400	355–447	4 354–709
27	—500—	447–563	
28	630	562–708	
29	800	708–892	5 707–1414
30	—1000—	891–1123	
31	1250	1122–1413	
32	1600	1412–1779	6 1411–2822
33	—2000—	1778–2240	
34	2500	2238–2819	
35	3150	2817–3549	7 2815–5630
36	—4000—	3547–4469	
37	5000	4465–5625	
38	6300	5621–7082	8 5617–11234
39	—8000—	7077–8916	
40	10000	8909–11225	

Note: Band numbers and center frequencies (nominal for ordinary use) are per ANSI/IEC standards. Frequency band limits are rounded to the nearest hertz.

where c is the speed of sound in ft/s (m/s). Note that for all practical purposes, the speed of sound in air is independent of pressure.

For example, if the temperature is 70°F (21°C), the speed of sound is

$$c = 49.03\sqrt{530} = 1129 \text{ ft/s (344 m/s)}$$

We can then use this value of c to compute the wavelength λ at various frequencies f at 70°F (21°C). At a frequency of 1000 Hz, for instance, we find from Eq. (8.2.1) that $\lambda = c/f = 1129/1000 = 1.129 \text{ ft (0.344 m)}$; likewise, at 20 Hz the wavelength would measure about 56.5 ft (17.2 m), and at 20,000 Hz it would measure about $\frac{2}{3}$ in (17.2 mm). For 70°F (21°C), Table 8.2.2 gives the wavelengths of sound in air at several frequencies.

As a practical matter, because the thickness of walls and the absorptive sections of silencers are small in relation to the wavelengths of low frequencies, such structures generally attenuate sound much better in the middle and high frequencies than in the low ones. Larger and more complex structures are required for reducing low-frequency noise.

8.2.4 THE SPEED OF SOUND IN SOLIDS

The speed of sound in longitudinal waves in a solid bar can be shown to be

$$c_s = \sqrt{\frac{E}{\rho}} \quad (8.2.3)$$

where c_s is the speed of sound in solids (m/s), E is the bar's modulus of elasticity (N/m²), and ρ is its density (kg/m³). This obviously means that sound travels faster through media of high modulus of elasticity and of low density. Accordingly, because rubber has a much higher elasticity and lower density than steel does (as one example), a rubber insert in a steel pipe will tend to slow down sound transmission along the pipe. Table 8.2.3 shows the speed of sound in various media.

One can speculate that since the elasticity and density in an absolute vacuum are zero, theoretically no sound waves should be able to travel through it. An absolute vacuum may thus be the ultimate noise barrier. However, no one is yet known to have been able to come up with a practical earthborn design.

8.2.5 THE DECIBEL

In using the term "decibel" it is important to understand the difference between sound power levels and sound pressure levels, since both are expressed in decibels.

TABLE 8.2.2 Wavelengths of Sound in Air at 70°F (21°C) $c = 1129 \text{ ft/s}$

f , Hz	63	125	250	500	1000	2000	4000	8000
λ , ft	17.92	9.03	4.52	2.26	1.129	0.56	0.28	0.14
λ , m	5.46	2.75	1.38	0.69	0.34	0.17	0.085	0.043

TABLE 8.2.3 Speed of Sound in Various Media (Shown in Descending Order of Magnitude)

Medium	Speed		Medium	Speed	
	ft/s	m/s		ft/s	m/s
Steel	16,500	5029	Concrete	10,600	3231
Aluminum	16,000	4877	Water	4,700	1433
Brick	13,700	4176	Lead	3,800	1158
Wood (hard)	13,000	3962	Cork	1,200–1,700	366–518
Glass	13,000	3962	Air	1,129	344
Copper	12,800	3901	Rubber	130–490	40–149
Brass	11,400	3475			

Sound pressure levels, which can readily be measured, quantify in decibels the intensity of given sound sources. Sound pressure levels vary substantially with distance from the source, and they also diminish as a result of intervening obstacles and barriers, air absorption, wind, and other factors.

Sound power levels, on the other hand, are constants independent of distance. It is very difficult to establish the sound power level of any given source because this level cannot be measured directly, but must be calculated by means of elaborate procedures; thus, as a practical matter, sound power levels are converted to sound pressure levels, which form the basis of practically all noise control criteria.

(As one example, Sec. 8.2.24 illustrates how the sound power level of a fan in an HVAC system is a critical element in the silencer selection procedure to meet specified sound pressure level criteria in an office or space.)

8.2.5.1 Sound Power Level

The lowest sound level that people of excellent hearing can discern has an acoustic power, or sound power, of about 10^{-12} W. On the other hand, the loudest sound generally encountered is that of a jet aircraft, with a sound power of about 10^5 W. Thus the ratio of loudest to softest sounds generally encountered is $10^{17}:1$.

A tenfold increase is called a “bel,” so the intensity of the jet aircraft’s noise can also be referred to as “17 bels.” This cuts the expression of immense ranges of intensity down to manageable size. However, since the bel is still a rather large unit, it is divided into 10 subunits called “decibels” (dB). Thus the jet noise is 170 dB, and to avoid confusion with any other reference intensity, we can say that it is 170 dB with reference to 10^{-12} W.

Sound power level L_w in decibels is therefore defined as

$$L_w = 10 \log \frac{W}{10^{-12}} \quad \text{dB re } 10^{-12} \text{ W} \quad (8.2.4)$$

where W is the sound power in watts. The sound power level in decibels can also be computed from

$$L_w = 10 \log W + 120 \quad (8.2.5)$$

Since 10^{-12} as a power ratio corresponds to -120 dB, we can see that by definition

1 W is equivalent to a 120-dB power level. Table 8.2.4 shows the sound power levels of typical sources.

Note that certain older literature may contain sound power level data referenced to 10^{-13} W, an absolute standard. Where this is the case, deduct 10 dB to convert to the current standard of 10^{-12} W.

The question now is, How does one measure sound power W ? This is where another way of looking at sound power helps. As shown in Fig. 8.2.4, consider a

TABLE 8.2.4 Sound Power Level L_w of Typical Sources

Source	Sound power W , W	L_w , dB re 10^{-12} W
Saturn rocket	100,000,000	200
Afterburning jet engine	100,000	170
Large centrifugal fan at 500,000 ft ³ /min (849,500 m ³ /h)	100	140
Seventy-five-piece orchestra/vaneaxial fan at 100,000 ft ³ /min (169,900 m ³ /h)	10	130
Large chipping hammer	1	120
Blaring radio	0.1	110
Centrifugal fan at 13,000 ft ³ /min (22,087 m ³ /h)	0.1	110
Automobile on highway	0.01	100
Food blenders—upper range	0.001	90
Dishwashers—upper range	0.0001	80
Voice—conversational level	0.00001	70
Quiet-Duct silencer, self-noise at + 1000 ft/min (5.1 m/s)	0.00000001	40
Voice—very soft whisper	0.000000001	30
Quietest audible sound for persons with excellent hearing	0.000000000001	0

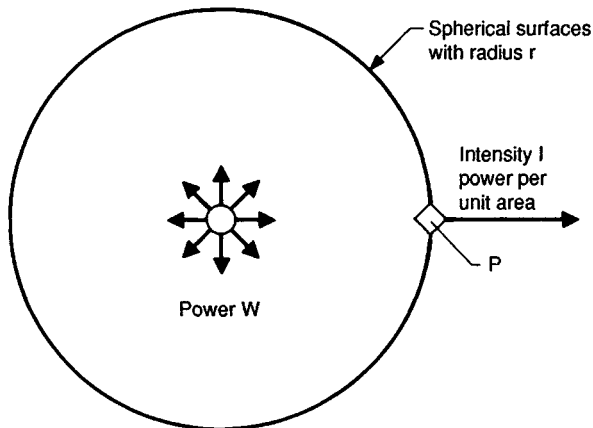


FIGURE 8.2.4 Ideal nondirectional sound source radiating W watts and producing a sound intensity I in watts per unit.

simple nondirectional source located at the center of a spherical surface (or at the center of a number of expanding spherical surfaces). Here the total sound power in watts is equal to sound intensity I (W/m^2) times the surface area S (m^2):

$$W = IS \quad (8.2.6)$$

where S for a spherical surface is $4\pi r^2$. Of course, as the sound waves move farther from their source, the surrounding spherical surface will become larger, and less power will pass through any unit element of the surface.

If the sound source is directional, the intensity will vary over the surface and the radiated power must be found by integration:

$$W = \int_S I S^{ds} \quad (8.2.7)$$

Since sound intensity I is rather difficult to measure, we measure sound pressure p instead. The relationship between sound pressure and intensity is

$$I = \frac{p_{\text{rms}}^2}{\rho c} \quad \text{W}/\text{m}^2 \quad (8.2.8)$$

where p = root-mean-square (rms) sound pressure, N/m^2
 ρ = density of air, kg/m^3
 c = speed of sound in air, m/s

The form of this equation will be familiar to many since it is analogous to the formula relating to electric power, voltage, and resistance:

$$P = \frac{E^2}{R}$$

where P = power, W
 E = voltage, V
 R = resistance, Ω

Sound intensity level L_i is defined as

$$L_i = 10 \log \frac{I}{I_{\text{ref}}} \quad \text{dB re } I_{\text{ref}} \quad (8.2.9)$$

where I_{ref} is $10^{-12} \text{ W}/\text{m}^2$.

8.2.5.2 Sound Pressure Level

Since sound-measuring instruments respond to sound pressure, the word “decibel” is generally associated with sound pressure level, but it is also a unit of sound power level. The square of sound pressure is proportional to, though not equal to, sound power.

Assuming a point source of sound radiating spherically in all directions, Eq. (8.2.6) tells us that $W = IS$. Accordingly, $10 \log W = 10 \log I + 10 \log S$, where S is the surface of the radiating sphere in ft^2 (m^2). Equation (8.2.5), however, tells us that $10 \log W = L_w - 120$. It can also be shown that $10 \log I = L_i - 120$. As a result, we get

$$L_w - 120 = L_i - 120 + 10 \log S$$

or

$$L_w = L_i + 10 \log S$$

Since $L_i = 10 \log (I/I_{\text{ref}})$ and $I = p^2/\rho c$, and since L_i can also be expressed as $10 \log (p^2/p_{\text{ref}}^2)$, which is also referred to as sound pressure level L_p , then

$$L_p = 10 \log \left(\frac{p}{p_{\text{ref}}} \right)^2 = 20 \log \frac{p}{p_{\text{ref}}} \quad (8.2.10)$$

Accordingly,

$$L_w = L_p + 10 \log S \quad (8.2.11)$$

8.2.6 DETERMINATION OF SOUND POWER LEVELS

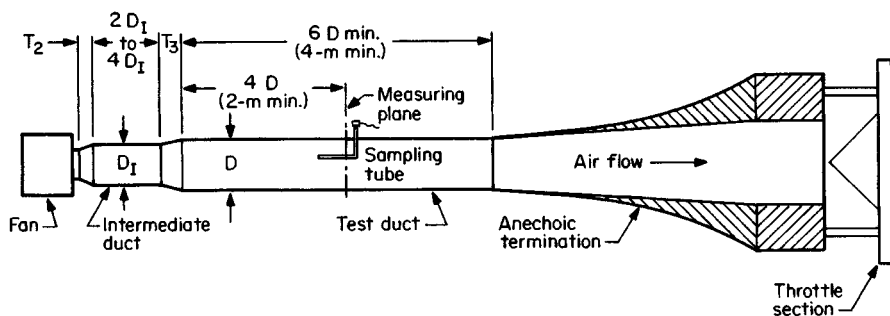
The concept of the imaginary radiating sphere emanating from the sound source will be referred to again in Sec. 8.2.8, Propagation of Sound Outdoors. Here, on the other hand, without considering imaginary spheres, we are concerned with measuring the sound power of a source that is confined within a structurally rigid space; for very large pieces of equipment and operating machinery in a plant, this approach may be the most practical way to estimate sound power.

The best method for determining the sound power level of a source is to measure it inside a good reverberant room with a truly diffuse sound field. With the sound power thus contained within the room, and with its intensity evenly distributed throughout the room, often only one sound pressure level measurement has to be taken. Then the sound power level can be calculated from $L_w = L_p + K$, where K is a constant dependent on the room volume, on the reverberation time at a given frequency or frequency band, and on the humidity.

Another method consists of containing the sound within the rigid walls of a pipe or duct equipped with an anechoic termination to minimize end reflections. Here all the sound energy must travel through the duct, and its sound field can be measured at a suitable measuring plane by averaging the sound pressure level across it. Equation (8.2.11) can then be used for calculating the power level of the noise-maker. Figure 8.2.5 illustrates such an arrangement for a ducted fan, which is also the basis of U.S. and British standards. (The 1986 U.S. standard was published jointly by ASHRAE, ANSI, and AMCA: ANSI/ASHRAE 68-1986 and ANSI/AMCA 330-86.) Although the anechoic duct method must overcome some practical difficulties, such as allowing for aerodynamically induced noise at the microphone, it clearly illustrates the relationship between measurements of sound pressure level and sound power level.

A test code of the U.S. Air Movement and Control Association (AMCA) requires the use of a reverberant or semireverberant room for determination of fan sound power levels. In such an arrangement the microphone would not be affected by aerodynamic flow. These two methods can yield comparable results, but relative fan sound power levels are likely to be most comparable if they have been determined under identical conditions.

In the British Standard 848, *Methods of Testing Fans*, 1966, part 2, the sound power level L_w in each frequency band would be calculated after averaging the



Test on fan outlet with open inlet

FIGURE 8.2.5 Anechoic duct method for fan sound power level determination. (*British Standard 848, Methods of Testing Fans, 1966, Part 2; ASHRAE/AMCA, Laboratory Method of Testing In-Duct Sound Power Measurement Procedure for Fans, 1986.*)

sound pressure level L_p across the duct area according to $L_w = L_p + 10 \log A$, where A is the cross-sectional area in ft^2 (m^2) at the plane of measurement. The U.S. standard (as in Fig. 8.2.5) uses $L_w = L_p + 20 \log D - 1.1$, where D is the diameter in ft (m) of the test duct.

8.2.7 CALCULATING CHANGES IN SOUND POWER AND SOUND PRESSURE LEVELS

8.2.7.1 Sound Power Level

Let L_{w1} be the sound power level corresponding to sound power W , and let L_{w2} be the sound power level twice as great, or $2W$. Then from Eq. (8.2.4)

$$L_{w1} = 10 \log \frac{W}{W_{\text{ref}}}$$

$$\text{and} \quad L_{w2} = 10 \log \frac{2W}{W_{\text{ref}}} = 10 \log \frac{W}{W_{\text{ref}}} + 10 \log 2 = L_{w1} + 3 \text{ dB}$$

Note: In eq. (8.2.4), $W_{\text{ref}} = 10^{-12} \text{ W} = 1 \text{ pW}$ (picowatt).

8.2.7.2 Sound Pressure Level

Assume L_{p1} to correspond to sound pressure p , and L_{p2} to sound pressure $2p$. Then from Eq. (8.2.10),

$$L_{p1} = 20 \log \frac{p}{p_{\text{ref}}}$$

and

$$L_{p2} = 20 \log \frac{2p}{p_{\text{ref}}} = 20 \log \frac{p}{p_{\text{ref}}} + 20 \log 2 = L_{p1} + 6 \text{ dB}$$

The addition of two equal sound pressures results in an increase of 6 dB, and the addition of two equal sound powers results in an increase of 3 dB. However, when two equal sound pressure levels are added, we are adding in effect two equal sound power levels, therefore:

$$\begin{aligned} L_{p1} + L_{p1} &= 10 \log \left(\frac{p}{p_{\text{ref}}} \right)^2 + 10 \log \left(\frac{p}{p_{\text{ref}}} \right)^2 \\ &= 10 \log \left(\frac{p}{p_{\text{ref}}} \right)^2 \times 2 \\ &= 10 \log \left(\frac{p}{p_{\text{ref}}} \right)^2 + 3 \text{ dB} \end{aligned}$$

Similarly, it can be said that when N identical sound sources are added,

$$L_p(\text{total}) = L_p(\text{single source}) + 10 \log N \quad (8.2.12)$$

where N is the number of sources; $10 \log N$ is plotted as a function of N in Fig. 8.2.6.

Table 8.2.5 shows how to add two unequal decibel levels, and Fig. 8.2.7 presents Table 8.2.5 graphically. Examples:

1. Two fans produce an L_p of 95 dB each in the fourth octave band at a given location. The combined L_p in that band would then be 98 dB.
2. If one of these fans is slowed down to produce an L_p of 90 dB, the combination L_p would then be 96 dB.

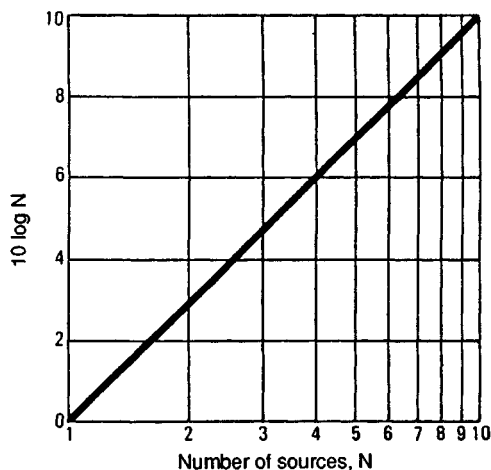
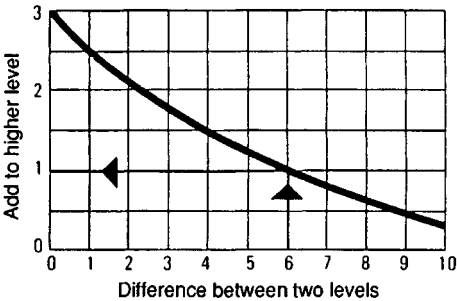


FIGURE 8.2.6 Predicting the combined noise level of identical sources.

TABLE 8.2.5 Addition of Sound Levels

Difference between the two levels, dB	Add to the higher level, dB
0	3
1	2.5
2	2
3	2
4	1.5
5	1
6	1
7	1
8	0.5
9	0.5
10 or more	0



Example: 80 dB + 74 dB = 81 dB

FIGURE 8.2.7 Decibel addition.

8.2.8 PROPAGATION OF SOUND OUTDOORS

Section 8.2.5.1 (and Fig. 8.2.4) introduced the concept of sound propagating through a series of spheres increasing in size as the distance r from its source increases. We now need to differentiate between hemispherical and spherical sound sources. If the source is considered hemispherical, the surface area $S = 2\pi r^2$; if the source is spherical, $S = 4\pi r^2$.

A fully spherical source would not be encountered frequently in a practical situation (examples would be an aircraft flying overhead, a rocket in flight, or noise emanating from the top of a tall building or a vertical stack or from a bird flying through air). If the source radiates hemispherically, as most sources do when close to the ground or to other reflective surfaces, then for a uniformly directional source (such as a siren), the relationship between sound pressure and sound power would be

$$\begin{aligned}
 L_p &= L_w - 10 \log 2\pi r^2 \\
 &= L_w - 20 \log r - 10 \log 2\pi \\
 &= \begin{cases} L_w - 20 \log r + 2.3 & \text{if } r \text{ is in feet} \\ L_w - 20 \log r - 8 & \text{if } r \text{ is in meters} \end{cases} \quad (8.2.13)
 \end{aligned}$$

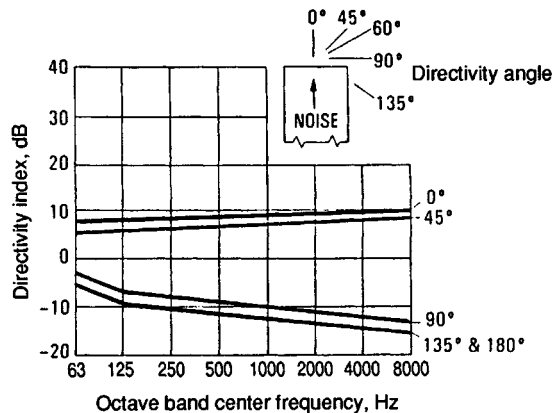
For a spherical source, the relationship would be

$$\begin{aligned}
 L_p &= L_w - 10 \log 4\pi r^2 \\
 &= L_w - 20 \log r - 20 \log 4\pi \\
 &= \begin{cases} L_w - 20 \log r + 0.7 & \text{if } r \text{ is in feet} \\ L_w - 20 \log r - 11 & \text{if } r \text{ is in meters} \end{cases} \quad (8.2.14)
 \end{aligned}$$

It will be noted that the sound pressure level for a hemispherical source is 3 dB higher than for a spherical source because the same sound intensity is considered to pass through an area half the size of a full sphere.

Not all sound sources radiate uniformly. If a sound source has a marked directional characteristic, this characteristic has to be taken into account; it is called the "directivity index" (DI). Figure 8.2.8 illustrates how noise emanating from an opening, stack, or pipe will vary with the directivity angle.

Other factors affecting the radiation of sound might be barriers and the attenuation of sound due to atmospheric conditions (such as molecular absorption in the air, wind, and rain) and ground conditions (including grass, trees, shrubbery, snow, paving, and water). Attenuation due to such factors is generally significant in the



The noise emanating from an opening, stack or pipe, will vary with directivity angle between the point of measurement and the conduit centerline. Data shown for stack or pipe diameter of approximately 10-ft (3.05-m) equivalent diameter.

FIGURE 8.2.8 Directivity index from openings, stacks, or pipes. (NEMA Standards Publication SM 33, *Directivity in Openings, Stacks or Pipes*, 1964.)

high frequencies and over long distances and makes reliable and reputable outdoor measurements very difficult to obtain.

For a directional noise source, we can therefore estimate sound pressure levels by modifying Eq. (8.2.13) as follows:

$$L_p = \begin{cases} L_w - 20 \log r + \text{DI} - A_a - A_b + 2.3 & \text{if } r \text{ is in feet} \\ L_w - 20 \log r + \text{DI} - A_a - A_b - 8 & \text{if } r \text{ is in meters} \end{cases} \quad (8.2.15)$$

where DI = directivity index

A_a = attenuation due to atmospheric conditions

A_b = attenuation due to barriers

r = distance from source, ft (m)

For instance, if we know or estimate the L_w of a fan (now provided by many manufacturers), we can also estimate the L_p at a distance r by taking into account directivity and the other factors indicated in Eq. (8.2.15).

8.2.9 THE INVERSE-SQUARE LAW

From Eq. (8.2.15) we can see that if the sound pressure level of a source is measured at two different distances from the source, the difference in sound pressure levels at those locations is

$$L_{p2} - L_{p1} = 10 \log \left(\frac{r_2}{r_1} \right)^2 = 20 \log \frac{r_2}{r_1} \quad (8.2.16)$$

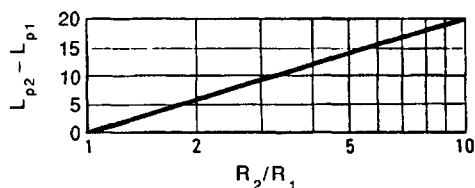
where L_{p1} = sound pressure level at location 1, dB

L_{p2} = sound pressure level at location 2, dB

r_1 = distance from source to location 1, ft (m)

r_2 = distance from source to location 2, ft (m)

The relationship between $(L_{p2} - L_{p1})$ and r_2/r_1 is shown in Fig. 8.2.9.



R_1 = distance from source to location 1

R_2 = distance from source to location 2

L_{p1} = sound pressure level, location 1

L_{p2} = sound pressure level, location 2

FIGURE 8.2.9 Inverse-square law.

Equation (8.2.16) shows that the sound pressure level varies inversely with the square of the distance from the source, with L_p decreasing by 6 dB for each doubling of distance from the source. This relationship is known as the “inverse-square law.”

At locations very close to a sound source, a measurement point will be in what is known as the “near field” or the source. In the near field, neither Eq. (8.2.15) nor Eq. (8.2.16) applies, and L_p will vary substantially with small changes in position. As the distance increases, however, L_p will decrease according to the inverse-square law; Eqs. (8.2.15) and (8.2.16) will apply, and a measurement point can be said to be in the “far field” of the source.

For all practical purposes, the inverse-square law functions only in a “free field,” which is defined as a space with no reflective boundaries or surfaces. Outdoors, such conditions are likely to exist only in an open field. In a reverberant field, such as might exist in the courtyard of a building or in a narrow street, the sound pressure level may decrease by a factor of less than 6 dB for each doubling of the distance. On the other hand, in a field of freshly fallen snow the decrease may be more than that predicted by the inverse-square law.

8.2.10 PARTIAL BARRIERS

Unobstructed sound propagates directly along a straight-line path from the source. If a barrier is interposed between that source and a receiver, some of the sound will be reflected back toward the source. These reflections can, of course, be attenuated by placing sound-absorptive surfaces on the barrier side facing the source.

Another portion of the sound emanating from the source is transmitted through the barrier (Fig. 8.2.10). To meet structural and wind loading criteria, however, most barrier designs significantly inhibit noise transmission to the extent that sound reaches the receiver primarily by diffracting over and around the barrier. As shown in Fig. 8.2.11, the presence of the barrier creates a “shadow zone” in which diffraction attenuates the noise reaching the receiver; the extent of this attenuation is the angle Θ between the straight and diffracted sound paths. Angle Θ (and thereby barrier attenuation) increases if the receiver or source is placed closer to the barrier or (assuming that the barrier is long enough to prevent sound from diffracting around the ends) if the barrier height is increased.

The theoretical relationship between barrier height, source and receiver position, and barrier attenuation from diffraction can be mathematically expressed as a function of Fresnel number N , as shown in Fig. 8.2.12.

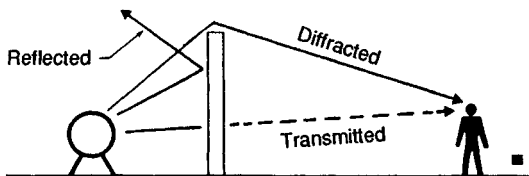


FIGURE 8.2.10 Barrier reflection, diffraction, and transmission.

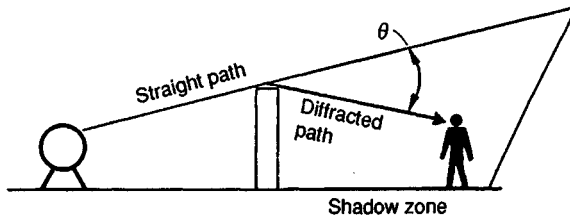
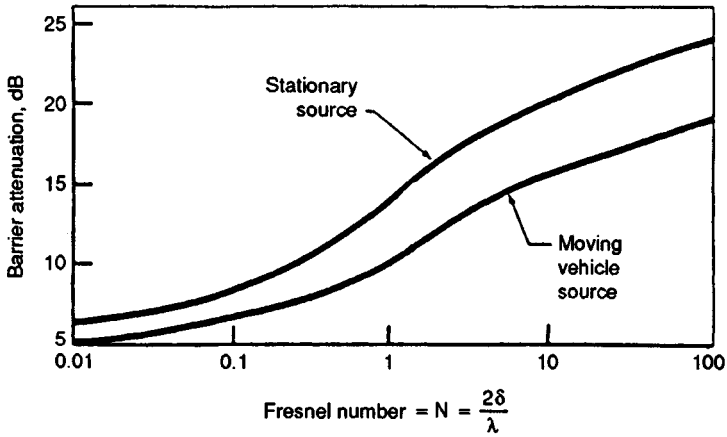
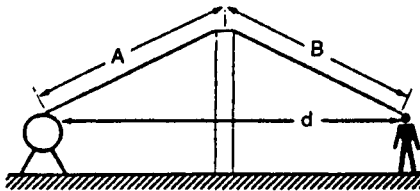


FIGURE 8.2.11 The shadow zone behind a barrier.



where λ = wavelength of sound, ft or m

$\delta = A + B - d$, ft or m



Note: N is a dimensionless number and can be used in English or metric units on a consistent basis.

FIGURE 8.2.12 Barrier attenuation as a function of Fresnel number. (Z. Maekawa, "Shielding Highway Noise," *Noise Control Engineering*, vol. 9, no. 1, July-Aug, 1977.)

8.2.11 PROPAGATION OF SOUND INDOORS

Assume that a sound source is on the floor of an enclosed space and that there are no partitions or barriers between the source and the receiver, and assume further that none of the sound leaves the space and reaches the receiver by a flanking path. Under these conditions, the sound in the space will reach the receiver by two paths: a direct sound path and a reverberant sound path.

8.2.11.1 Direct Sound Path

In the far field of the source, sound from a source on or near the center of a wall or floor in a room will propagate to the receiver according to the inverse-square law:

$$L_{pd} = \begin{cases} L_w - 20 \log r + 2.3 & \text{if } r \text{ is in feet} \\ L_w - 20 \log r - 8 & \text{if } r \text{ is in meters} \end{cases} \quad (8.2.17)$$

where L_{pd} is the sound pressure level from direct sound.

8.2.11.2 Reverberant Sound Path

Reverberant sound will reach the receiver after reflecting off surfaces in the space. If the sound in the space is diffuse (essentially equal at all locations), Eq. (8.2.18) applies:

$$L_{pr} = \begin{cases} L_w - 10 \log A + 16.3 & \text{if } A \text{ is in sabins} \\ L_w - 10 \log A + 6.0 & \text{if } A \text{ is in metric sabins} \end{cases} \quad (8.2.18)$$

where L_{pr} is the sound pressure level from reverberant sound, and A is the total absorption. The "total absorption" of a surface is the product of the surface area S and the absorption coefficient α of that surface:

$$A = S\alpha \quad (8.2.19)$$

where the units of A are sabins if S is in ft^2 , and metric sabins if S is in m^2 ; α is the sound absorption coefficient, the dimensionless ratio of sound energy absorbed by a given surface to that incident upon the surface (see Sec. 8.2.22 and Table 8.2.43).

Total room absorption can be calculated as follows:

$$A = \sum S\alpha = S_1\alpha_1 + S_2\alpha_2 + S_3\alpha_3 + \cdots + S_n\alpha_n \quad (8.2.20)$$

where

A = total absorption in room, sabins (metric sabins)

S = total surface area in room, ft^2 (m^2)

$\bar{\alpha}$ = average room absorption coefficient

$S_1, S_2, S_3, \dots, S_n$ = surface area of different segments of wall, ceiling, and floor surfaces in room

$\alpha_1, \alpha_2, \alpha_3, \dots, \alpha_n$ = corresponding sound absorption coefficients

Reverberant sound may be reduced by adding sound-absorptive materials to reflective room surfaces. The theoretical reduction in reverberant sound due to add-

ing sound-absorptive treatment to the surfaces of a room containing a diffuse sound field is equal to

$$\text{Reduction in reverberant sound} = 10 \log \frac{A_2}{A_1} \quad (8.2.21)$$

where A_1 is the total room absorption after adding sound-absorptive treatment, and A_2 is the total room absorption before adding treatment. This is illustrated in Fig. 8.2.13.

8.2.11.3 Effects of Direct and Reverberant Sound

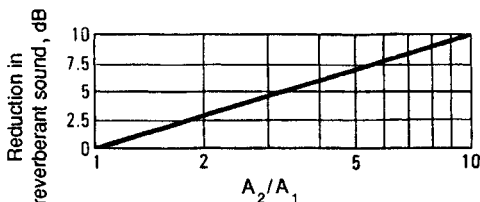
The effects of direct and reverberant sound are shown in Fig. 8.2.14. Direct sound predominates close to the source, but direct sound diminishes with distance. Thus, farther from the source, reverberant sound predominates; under ideal conditions this occurs when the L_p in the room levels off with increasing distance from the source.

The quantitative relationship between L_w and L_p from both direct and reflected sound paths is shown in Fig. 8.2.15 as a function of distance from the source and total room absorption. Add 3 dB to L_p if the source is on the wall or floor of the room, add 6 dB if the source is at the intersection of two walls (or a wall and ceiling), and add 9 dB if the source is in a corner (Ref. 1).

Note in Fig. 8.2.15 [and also Eq. (8.2.17)] that increasing the total room absorption has no effect on direct sound; accordingly, adding sound-absorptive materials to room surfaces will show maximum reduction in L_p in areas where reverberant sound predominates. Also, note that small increases in total room absorption will not produce significant decreases in sound pressure level; even in a location dominated by reverberant sound, doubling the room absorption will decrease the sound pressure level by only 3 dB.

8.2.12 SOUND TRANSMISSION LOSS

Figure 8.2.16 shows that when a sound path is broken by a partition, part of the sound is reflected, part is absorbed, and part is transmitted through the partition.



A_2 = Total room absorption after adding sound absorptive treatment

A_1 = Total room absorption before adding treatment

FIGURE 8.2.13 Effect of increasing room absorption.

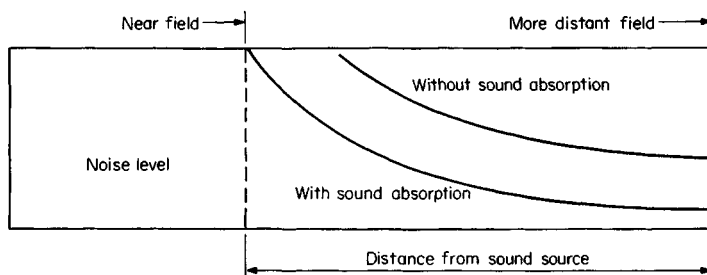
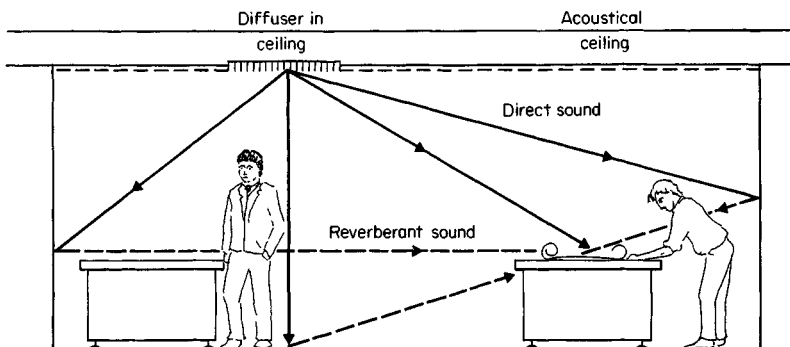


FIGURE 8.2.14 Effects of direct and reverberant sound on listeners in the source's near and far fields. Close to the source, direct sound predominates; at a distance, reverberant sound predominates.

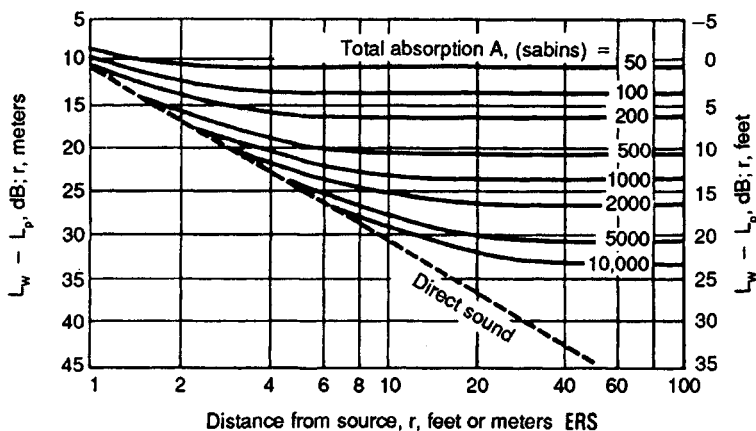


FIGURE 8.2.15 Effects of direct and reverberant sound in rooms. [C. M. Harris (ed.), *Handbook of Noise Control*, 2d ed., McGraw-Hill, New York, 1979, part 1, p. 8-4.]

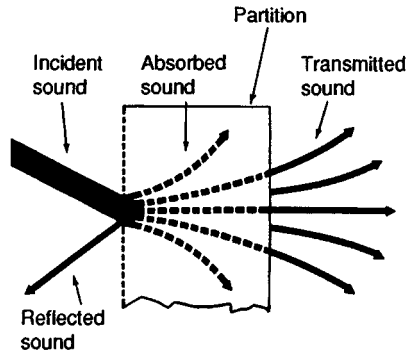


FIGURE 8.2.16 Effect of partitions on incident sound. (*Noise Control: A Guide for Workers and Employees, U.S. Department of Labor, Occupational Safety and Health Administration, 1980.*)

Ten times the logarithmic ratio of incident sound power to transmitted sound power is defined as “sound transmission loss” (TL). As shown in Fig. 8.2.17,

$$TL = 10 \log \frac{W_i}{W_t} = 10 \log \frac{1}{\tau} \quad (8.2.22)$$

where τ = sound transmission coefficient

W_i = incident sound power, W

W_t = transmitted sound power, W

8.2.12.1 The Mass Law

The mass law provides a theoretical relationship between the sound transmission loss of a single-wall (solid) partition, its weight, and the frequency of sound being transmitted through it. For normal incidence (NI), the relationship is

$$TL = \begin{cases} 10 \log w + 20 \log f - 33.5 & \text{if } w \text{ is in lb/ft}^2 \\ 10 \log w + 20 \log f - 47.5 & \text{if } w \text{ is in kg/m}^2 \end{cases} \quad (8.2.23)$$

where w is the weight (or mass density), and f is the frequency in hertz.

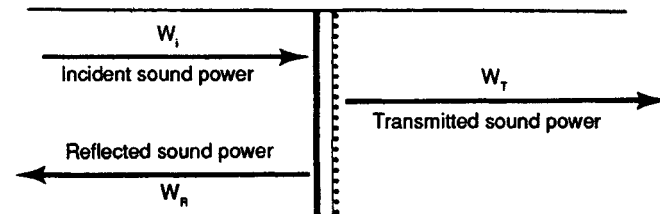


FIGURE 8.2.17 Incident sound power versus reflected and transmitted sound power.

Equation (8.2.23) tells us that for each doubling of the barrier's weight, the transmission loss increases by 6 dB. Equally, by doubling or halving the frequency, a 6-dB shift in TL occurs.

Equation (8.2.23) is commonly known as "the mass law," but more accurately it is an approximation. Actual data can deviate from mass law predictions by 10 dB or more, and the law generally does not apply to nonhomogeneous structures. As will be shown in Sec. 8.2.21.1, for example, multilayer walls or double walls separated by an air space generally provide greater TL than that predicted by the mass law.

8.2.12.1 The Effect of Openings on Partition TL

Windows, access ports, door seals, wall-to-ceiling joints, cutouts for wiring or plumbing, and other openings can significantly diminish the TL capabilities of a structure. As an example, if a 100-ft² (9.3-m²) partition has a TL rating of 40 dB at a given frequency, a 1 percent [or 1-ft² (0.093-m²)] opening in that partition will reduce the overall TL to 20 dB unless noise control measures are applied to the opening. The theoretical effect of openings in partitions or complete enclosures is shown in Fig. 8.2.18.

8.2.12.3 Single-Number TL Ratings: STC Ratings

For engineering rating purposes, the TL of partitions is frequently defined in terms of a single-number decibel rating known as "sound transmission class" (STC). STC ratings are determined by plotting contours of TL versus frequency in one-third octave bands from 125 to 4000 Hz and comparing the results with standard contours defined in ASTM E413 (Fig. 8.2.19). The TL data and STC ratings of typical structures are listed in Table 8.2.3a. The total deficiencies must not be greater than 32 dB, but any single band's deficiency cannot be greater than 8 dB.

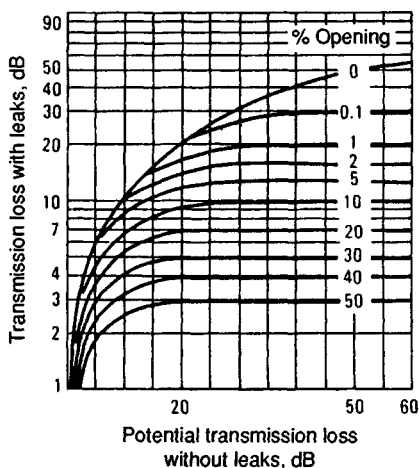


FIGURE 8.2.18 Effect of openings on partition TL.

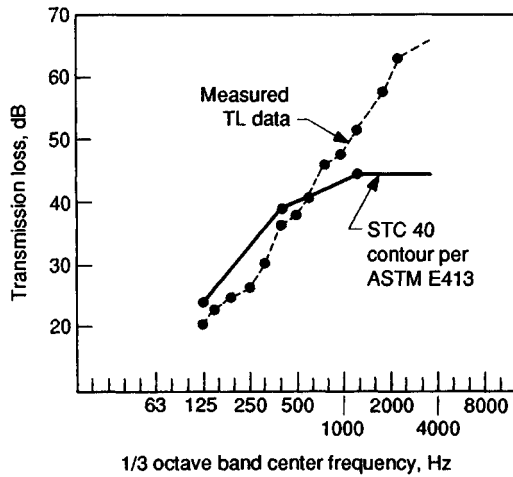


FIGURE 8.2.19 ASTM E413 contours for sound transmission class (STC) and noise isolation class (NIC). (*ASTM E413, Standard Classification for Determination of Sound Transmission Class, 1973.*)

8.2.13 NOISE REDUCTION AND INSERTION LOSS

As shown in Fig. 8.2.20, “noise reduction” (NR) is simply the difference in sound pressure level between any two points along the sound path from a noise source:

$$NR = L_{p1} - L_{p2} \quad (8.2.24)$$

“Insertion loss” (IL), on the other hand is the before-versus-after difference at the same measurement point, brought about by interposing a means of noise control between the source and the receiver (Fig. 8.2.21):

$$IL = L_{p0} - L_{p2} \quad (8.2.25)$$

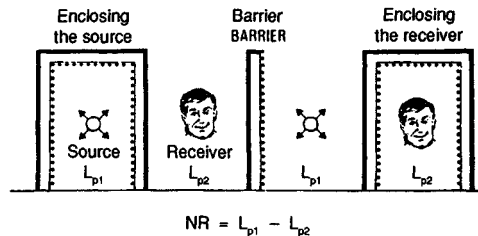


FIGURE 8.2.20 Illustration of noise reduction: $NR = L_{p1} - L_{p2}$. (Lawrence G. Copley, “Control of Noise by Partitions and Enclosures,” *Tutorial Papers on Noise Control for Inter-Noise, Institute of Noise Control Engineers, 1972.*)

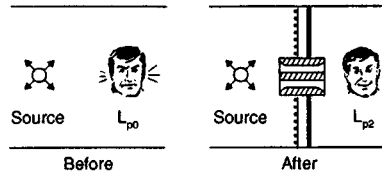


FIGURE 8.2.21 Illustration of insertion loss: $IL = L_{p0} - L_{p2}$. (Lawrence G. Copley, "Control of Noise by Partitions and Enclosures," *Tutorial Papers on Noise Control for Inter-Noise*, 1972.)

Like TL, NR and IL are typically rates as a function of full octave bands or one-third octave bands. The NR ratings of several types of soundproof room are listed in Table 8.2.42. A single-number NR rating system called "noise isolation class" (NIC) is often used for such rooms. Similar to the STC ratings described in Sec. 8.2.12.3, NIC ratings are established by plotting NR as a function of frequency and comparing the results against standard contours defined in ASTM E413.

8.2.14 THE EFFECTS OF SOUND ABSORPTION ON RECEIVING-ROOM NR CHARACTERISTICS

Figure 8.2.22 shows a receiver located within a room outside of which is a noise source. The relationship between the NR and TL characteristics of such a room can be shown to be represented by

$$NR = TL + 10 \log \frac{\bar{\alpha}_2 A_2}{S} \quad (8.2.26)$$

where $NR = L_{p1} - L_{p2}$

L_{p1} = sound pressure level in source room, dB

L_{p2} = sound pressure level in receiving room, dB

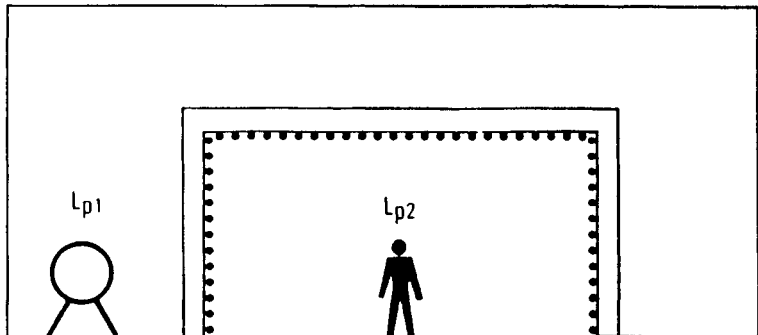


FIGURE 8.2.22 Noise source in outer room, and receiver in inner room.

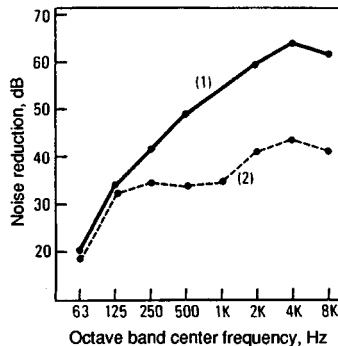
- TL = transmission loss of receiving-room walls, dB
 $\bar{\alpha}_2$ = average sound absorption coefficient in receiving room
 A_2 = total wall area in receiving room, ft² (m²)
 S = surface area separating the two rooms, ft² (m²)

If the source room is highly reverberant and if the receiving room is highly absorptive such that $\bar{\alpha}_2$ is close to unity, then $NR = TL$. In the event that the receiving room is highly reflective, however, $\bar{\alpha}_2$ will be very low; for instance, if $S = A_2$ and if $\bar{\alpha}_2 = 0.01$, then $NR = TL - 20$ dB.

Accordingly, a highly absorptive receiving room can be seen to have a potential of 20-dB more noise reduction than a reflective receiving room with the same TL . This effect is illustrated in Fig. 8.2.23, which shows the NR of a 6-ft 4-in by 6-ft 0-in by 6-ft 6-in (1930- by 1829- by 1981-mm) room, which could be a fan plenum, tested with and without 2 in (51 mm) of sound-absorptive materials on the otherwise highly reflective steel inside walls. The sound absorption coefficient of a 2-in (51-mm) liner is relatively low at low frequencies, so the liner has little effect on NR . At the higher frequencies, however, NR is approximately 20 dB higher with the absorptive liner in place.

8.2.15 FAN NOISE

Fans are the primary source of noise generation in HVAC systems. It is always best to use fan L_w data provided by the fan manufacturer. However, if these data are not available, Eq. (8.2.27) can be used to predict the estimated fan L_w (dB re $1p_w$); see note in section 7.1 (Ref. 2):



1. 2-in (51-mm) sound absorptive materials on inside wall surface
2. Reflective steel walls

FIGURE 8.2.23 Additional NR demonstrated by adding sound absorption to the inside surface of a reflective receiving room. (Martin Hirschorn, *Noise Control Reference Handbook*, Industrial Acoustics Company, 1982.)

$$L_w = K_w + 10 \log \frac{Q}{Q_1} + 20 \log \frac{P}{P_1} + C + \text{BFI} \quad (8.2.27)$$

where K_w = specific sound power level, dB re $1 p_w$ (from Table 8.2.6)

Q = flow rate, ft^3/min (L/s)

$Q_1 = 1$ when Q is in ft^3/min , 0.472 when Q is in L/s

P = fan pressure head, in WG (Pa) [in WG is "inch water gauge"]

$P_1 = 1$ when P is in inches WG, 249 when P is in Pa

C = correction factor for point of operation, dB

BFI = blade frequency increment to be added only to octave band containing blade pass frequency

The values of K_w and BFI are shown in Table 8.2.6, and Table 8.2.7 shows the octave band in which the BFI is likely to occur for different fan types. Values for C are given in Table 8.2.8.

Fans can generate high-intensity noise levels of a discrete tone at the blade pass frequency (BPF). The noise level's intensity will vary with the type of fan. The BPF can be established if the rpm and number of blades of the fan are known; the following equation can then be used:

$$\text{BPF} = \frac{\text{rpm} \times \text{number of blades}}{60} \text{ Hz} \quad (8.2.28)$$

For example, if the rpm is 1200 and the number of blades is 8, $\text{BPF} = 160 \text{ Hz}$.

These discrete tones at the BPF are usually the most predominant noises emanating from large fans, but such discrete frequencies may not show up in an octave band analysis. To find them, narrower frequency ranges may have to be measured, such as one-third octave bands or even one-tenth octave bands.

BFI's vary from 2 dB for centrifugal radial-blade fans to 8 dB for centrifugal pressure-blower fans (see Table 8.2.6). BFI's and second harmonic frequencies generally occur in the 63- to 500-Hz region (see Table 8.2.7).

Once a decision has been made as to the type of fan to be used, it is best to select one that operates close to the peak of its efficiency curve. Such a fan will typically generate the lowest noise level. The correction factor C for off-peak operation is shown in Table 8.2.8.

EXAMPLE 8.2.1 A 35.5-in-diameter vaneaxial fan with eight blades has a 20,000- ft^3/min flow rate, develops a 4-in WG head at a speed of 1765 r/min, and operates at 95 percent of peak efficiency. Determine the fan's L_w and BPF.

Solution Calculate the fan's total L_w from Eq. (8.2.27):

- For K_w , Table 8.2.6 gives a range of octave band center frequencies for a vaneaxial fan with a diameter (or wheel size) under 40 in.
- Flow rate $Q = 20,000 \text{ ft}^3/\text{min}$, and $Q_1 = 1$. Thus $10 \log (Q/Q_1) = 43$.
- Fan pressure head $P = 4$ in WG, and $P_1 = 1$. Thus $20 \log (P/P_1) = 12$.
- Correction factor C comes from Table 8.2.8; at 95 percent peak efficiency, $C = 0$.
- From Table 8.2.6, for vaneaxial fans, $\text{BFI} = 6$. Furthermore, Table 8.2.7 and its note show that this BFI occurs in the 250-Hz octave band.

These data are tabulated in Table 8.2.9, which shows the total L_w .

TABLE 8.2.6 Specific Sound Power Levels K_w (dB re $1p_w$) and Blade Frequency Increment (BFI) for Various Types of Fans

Fan type	Wheel size	Octave band center frequency, Hz							BFI
		63	125	250	500	1000	2000	4000	
Centrifugal									
Airfoil, backward-curved,	Over 36 in (900 mm)	32	32	31	29	28	23	15	3
backward-inclined	Under 36 in (900 mm)	36	38	36	34	33	28	20	
Forward-curved	All	47	43	39	33	28	25	23	2
Radial blade and pressure blower	Over 40 in (1000 mm)	45	39	42	39	37	32	30	8
	From 40 in (1000 mm) to 20 in (500 mm)	55	48	48	45	45	40	38	
	Under 20 in (500 mm)	63	57	58	50	44	39	38	
Vaneaxial	Over 40 in (1000 mm)	39	36	38	39	37	34	32	6
	Under 40 in (1000 mm)	37	39	43	43	43	41	28	
Tubeaxial	Over 40 in (1000 mm)	41	39	43	41	39	37	34	5
	Under 40 in (1000 mm)	40	41	47	46	44	43	37	
Propeller									
Cooling tower	All	48	51	58	56	55	52	46	5

Note: These values are the specific sound power levels radiated from either the inlet or the outlet of the fan. If the total sound power level being radiated is desired, add 3 dB to the above values.

Source: 1987 ASHRAE Handbook, Systems, ASHRAE, Atlanta, 1987, chap. 52, "Sound and Vibration Control."

TABLE 8.2.7 Octave Band in Which Blade Frequency Increment (BFI) Is Likely to Occur

Fan type	Octave band in which BFI occurs*
Centrifugal	
Airfoil, backward-curved, backward-inclined	250 Hz
Forward-curved	500 Hz
Radial blade and pressure blower	125 Hz
Vaneaxial	125 Hz
Tubeaxial	63 Hz
Propeller	
Cooling Tower	63 Hz

*Use for estimating purposes. For speeds of 1750 r/min (29 r/s) or more, move the BFI to the next higher octave band. Where the actual fan is known, use the manufacturer's data.

Source: 1987 ASHRAE Handbook, Systems, ASHRAE, Atlanta, 1987, chap. 52, "Sound and Vibration Control."

TABLE 8.2.8 Correction Factor C for Off-Peak Operation

Static efficiency, % of peak	Correction factor, dB
90 to 100	0
85 to 89	3
75 to 84	6
65 to 74	9
55 to 64	12
50 to 54	15

Source: 1984 ASHRAE Handbook, Systems, ASHRAE, Atlanta, GA, 1984, chap. 32, "Sound and Vibration Control."

To calculate the fan's BPF, use Eq. (8.2.28). Given 1765 r/min and eight blades, $BPF = (1765 \times 8)/60 = 235$ Hz, and Table 8.2.9 shows that the nearest octave band to 235 Hz in this example is 250 Hz.

TABLE 8.2.9 Calculation of Total Fan L_w in Example 49.1

Calculation	Octave band center frequency, Hz						
	63	125	250	500	1000	2000	4000
Specific fan K_w	37	39	43	43	43	41	28
$10 \log (Q/Q_1) + 20 \log (P/P_1)$	55	55	55	55	55	55	55
C	0	0	0	0	0	0	0
BFI	...	...	6				
+3 dB to get total L_w (see note below Table 49.6)	3	3	3	3	3	3	3
Total L_w	95	97	107	101	101	99	86

8.2.16 COOLING TOWER NOISE

In the typical mechanically induced-draft cooling tower (Fig. 8.2.24), noise is generated by fan noise and water impact; at most locations of interest, however, fan noise predominates. For evaluation and control of cooling tower noise, see Refs. 3 to 6. A typical cooling tower noise control installation, consisting of air-intake and -discharge silencers, is shown in Fig. 8.2.25.

Cooling tower fan noise, if not available from the manufacturer, can be estimated from Eq. (8.2.27) and Tables 8.2.6 to 8.2.8. It should be noted, however, that the intake noise must propagate upstream against the air flow, make a 90° turn, divide as it disperses through the side of the tower, and pass through the louvers. This tortuous path results in the cooling tower fan's intake noise being less than its discharge noise. Typical fan attenuation at the air intake can amount to as much as 3, 7, 11, and 9 dB in the first four octave bands, respectively; however, in the last four bands water noise predominates. Clearly, wherever possible, data based on actual measurements and provided by the cooling tower manufacturer should be used.

8.2.17 DUCT SILENCERS—TERMINOLOGY AND TYPES

Duct silencers reduce the air-flow noise inside air-handling systems that is caused by the following:

- The fan—the air's prime mover
- The passage of air through straight ducts

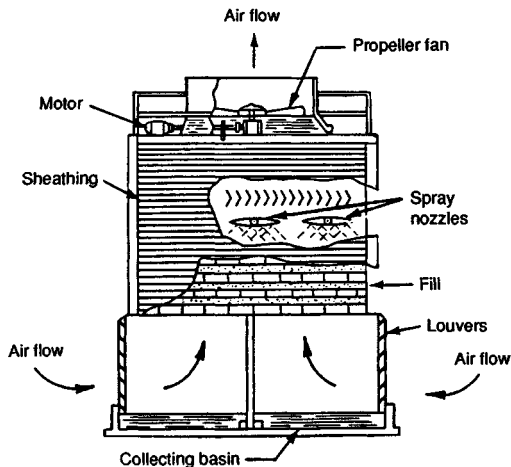


FIGURE 8.2.24 Mechanically induced-draft cooling tower.

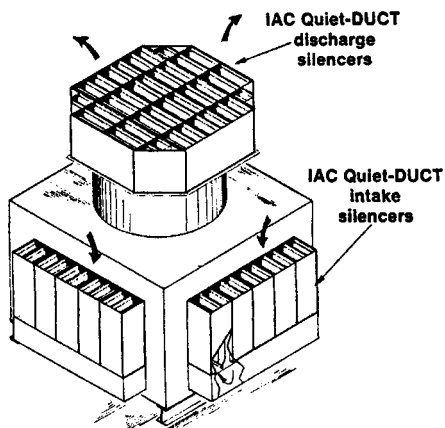


FIGURE 8.2.25 Silencers for cooling towers.
(*Application Manual for Duct Silencers, Bulletin 1.0301.4, Industrial Acoustics Company, 1989.*)

- The impact of air flowing through duct components, such as elbows, branches, mixing boxes, rods, and orifices

We can generalize that any form of air movement will generate noise. If V is the velocity of air flow in a straight duct, the sound power level may be a function of V^5 to V^7 , depending on the frequency and the duct component. This means that the noise generated by air flow inside a duct may increase or decrease by 15 to 21 dB every time the velocity is doubled or halved.

Six principal parameters are generally used to describe the aeroacoustic characteristics of silencers:

1. *Dynamic insertion loss (DIL)*: The DIL is the difference between two sound power levels or intensity levels when measured at the same point in space before and after a silencer has been inserted between the measuring point and the noise source.
2. *Self-noise (SN)*: The SN is the sound power level in decibels generated by a given volume of air flowing through a silencer of stated cross-sectional area.
3. *Air flow*: Accurate aerodynamic measurements are essential in describing any component of an air-handling system. DIL and SN data are always reported as a function of silencer face air-flow velocity.
4. *Static pressure drop*: This is generally related to silencer face velocity and volumetric air-flow capacity for a given silencer face area. For energy conservation considerations, it can also be related to the horsepower (kilowatts) required to overcome the pressure drop.
5. *Forward flow*: This applies to DIL and SN ratings with the air flow moving in the same direction as the noise propagation, such as in a fan discharge system.
6. *Reverse flow*: This applies to DIL and SN ratings with the air flow and noise propagation moving in opposite directions, such as in a fan inlet system.

There are many types of silencers, including the following:

Reactive Silencers. These have tuned cavities and/or membranes and are designed mainly to attenuate low-frequency noise in diesel, gasoline, and similar engines. Such silencers, however, are rarely used in HVAC systems.

Diffuser-Type Silencers. These are used primarily for jet engine test facilities and pneumatic cleaning nozzles in manufacturing operations. They often employ perforated “pepper pots” that slow down the flow velocities and/or prevent the generation of low-frequency noise.

Active Attenuators. Much work has been done during the last 10 years on “active” silencers. These attenuate noise by means of electronic cancellation techniques involving microphones, speakers, synchronizing sensors, and microprocessors.

Such silencers are effective at low frequencies under 300 Hz but are not suitable for broadband noise reduction without the addition of a dissipative silencer.

Moreover, this cost and maintenance requirements do not make such silencers a practical proposition. However, they might constitute an answer in unusual situations where there is no room for conventional silencers and where very low frequency noise must be controlled.

Packless Silencers. These can be used where the acoustic infill of conventional silencers could become a breeding ground for disease-carrying bacteria or where particulate matter from fiber erosion can contaminate streams of air or gas. This makes packless silencers particularly suitable for microchip manufacturers, food processing plants, hospitals, and pharmaceutical and other manufacturing plants requiring clean-room environments.

The absence of acoustic materials also reduces fire hazards where flammable materials could saturate the infill. Other applications therefore include engine test cell, kitchen exhausts, and facilities in general where fuels, grease, acids, and solvents might be carried in streams of air or gas.

Packless silencers could well become more important for general use if it becomes established that fiberglass causes lung illnesses.

Dissipative Silencers. These are widely used in HVAC duct systems. Figures 8.2.26 and 8.2.27 show the general configuration of rectangular splitter silencers. The splitter, consisting of a strong, perforated-steel envelope containing sound-absorptive materials, divides the air or gas flow into smaller sound-attenuating passages. Rectangular silencers are used in rectangular ducts and are sometimes set up in very large tiers, or banks, on the intakes and exhausts of fans.

Figure 8.2.28 shows a tubular, or cylindrical, silencer. At first sight it looks similar in cross section to the rectangular silencer, but it consists of an outer cylindrical shell and an inner concentric bullet. Cylindrical silencers are often used in circular duct systems in conjunction with vaneaxial fans.

Dissipative silencers are available in a variety of executions, lengths, and cross sections to meet almost any noise-reduction and pressure-drop requirement of an HVAC system. The use of dissipative silencers is further discussed and illustrated in Secs. 8.2.23 to 8.2.29 in terms of applications. For discussions of the principles of silencer performance and duct break-out noise, respectively, see Sec. 8.2.18 and 8.2.20.

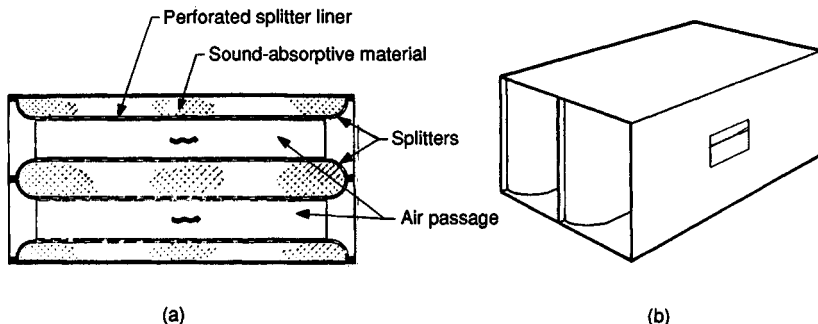
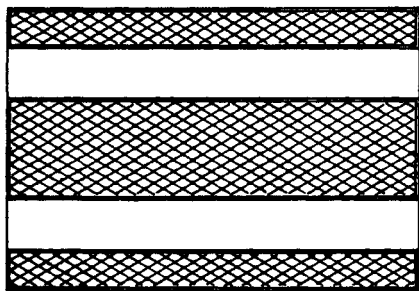


FIGURE 8.2.26 "Round-nosed" rectangular silencer. (a) Cross section; (b) external view. (*Application Manual for Duct Silencers, Bulletin 1.0301.4, Industrial Acoustics Company, 1989.*)



(Frequently used in Europe. Constitutes poor aerodynamic and self-noise design. See Sect. 49.23.3 and Fig. 49.59.)

FIGURE 8.2.27 "Flat-nosed" rectangular silencer. (*Martin Hirschorn, "The Aero-Acoustic Rating of Silencers for 'Forward' and 'Reverse' Flow of Air and Sound," Noise Control Engineering, vol. 2, no. 1, Winter 1974.*)

8.2.18 EFFECTS OF FORWARD AND REVERSE FLOW ON SILENCER SN AND DIL

The self-noise (SN) of a silencer varies by 7 to 26 dB for each doubling and halving of flow velocity, depending on the frequency, on the silencer's configuration, and on whether the noise and air flow are traveling in the same direction (i.e., forward or reverse flow).

As explained in Sec. 8.2.17, forward flow occurs if the air flow is traveling in the same direction as the sound propagation, as on the supply side of an HVAC system, and reverse flow occurs when air is traveling in a direction opposite to the direction of sound propagation, such as in a duct's return-air system. Both are illustrated in Fig. 8.2.29.

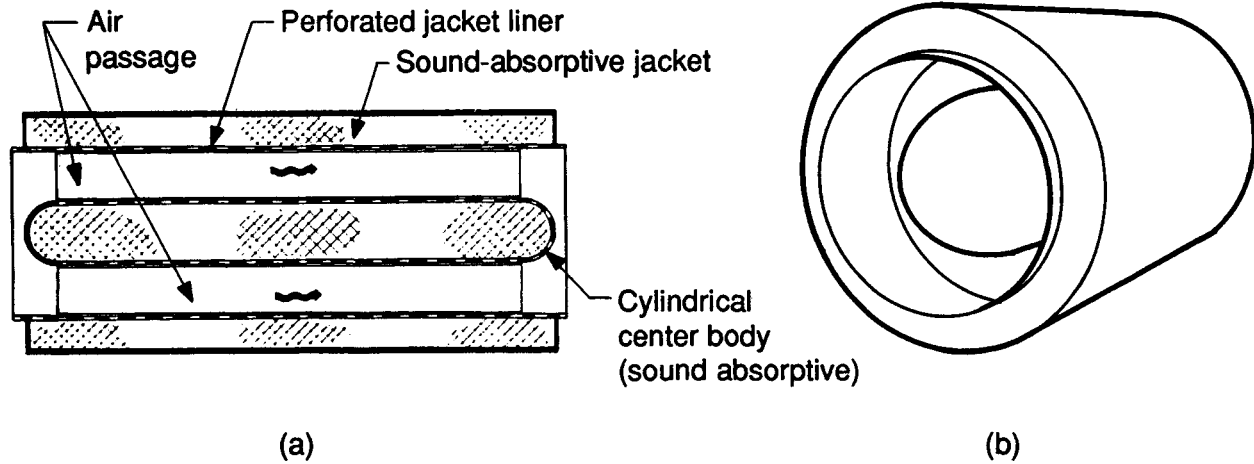
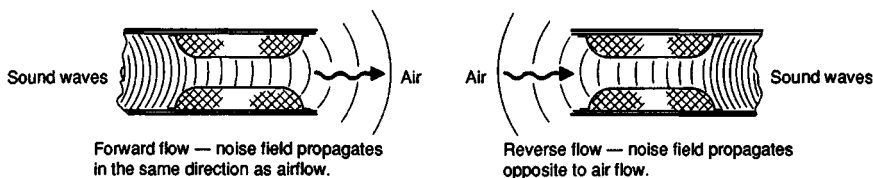


FIGURE 8.2.28 Cylindrical silencer. (a) Cross section; (b) external view. (*Application Manual for Duct Silencers, Bulletin 1.0301.3, Industrial Acoustics Company.*)



Note: If velocity of air through silencer is 70 ft/s (21.3 m/s), the speed of sound in the forward-flow direction would be $1100 + 70 = 1170$ ft/s ($335.3 + 21.3 = 356.6$ m/s). Similarly in the reverse-flow direction, the speed of sound through the silencer would be $1100 - 70 = 1030$ ft/s ($335.3 - 21.3 = 314$ m/s). Approximate velocity of sound at sea level = 1100 ft/s (335.3 m/s).

FIGURE 8.2.29 Schematic of reverse flow versus forward flow. (*Application Manual for Duct Silencers, Bulletin 1.0301.4, Industrial Acoustics Company, 1989.*)

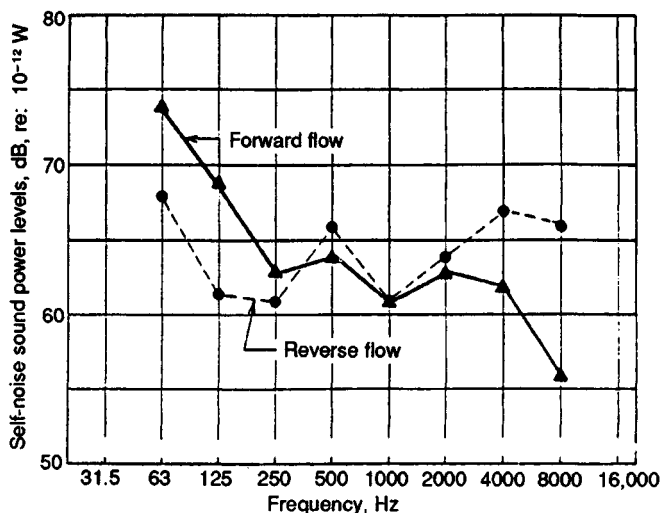
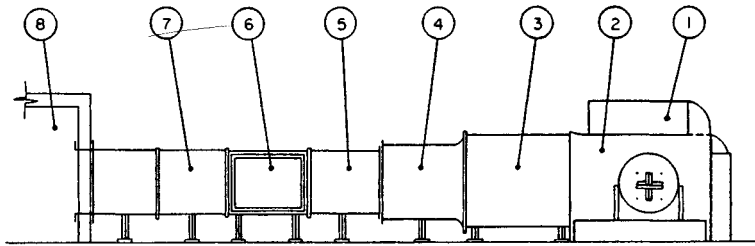


FIGURE 8.2.30 Characteristic self-noise spectra for rectangular silencers with 30 percent free area. (*M. Hirschorn, "Acoustic and Aerodynamic Characteristics of Duct Silencers for Airhandling Systems," ASHRAE Paper CH-81-6, 1981.*)

Figure 8.2.30 illustrates the effects of forward and reverse flow on silencer SN. Low-frequency SN is the greatest in the forward-flow mode, while high-frequency SN is the greatest in the reverse-flow mode.

Because of the forward- and reverse-flow phenomena, silencer performance is best rated with air flow in terms of dynamic insertion loss (DIL) determined in accordance with ASTM E477 (Ref. 7) in a reverberant room in the reverse and forward modes. The test arrangement is shown in Fig. 8.2.31 and 8.2.32. See Ref. 8.



1. Air flow measurements station
2. System fan
3. System silencer
4. Signal source chamber
5. Upstream pressure test station
6. Silencer under test
7. Downstream pressure test station
8. Reverberation room

FIGURE 8.2.31 Typical facility for rating duct silencers with or without air flow. (ASTM E477, *Standard Method of Testing Duct Liner Materials and Prefabricated Silencers for Acoustical and Airflow Performance*, American Society for Testing and Materials, 1973.)

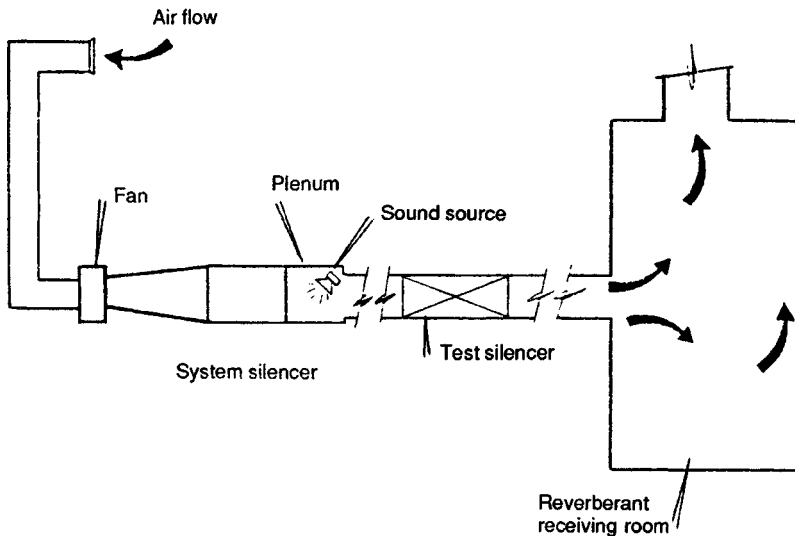


FIGURE 8.2.32 Schematic of the facility shown in Fig. 8.2.31; forward flow illustrated. (M. Hirschorn, "Acoustic and Aerodynamic Characteristics of Duct Silencers for Airhandling Systems," ASHRAE Paper CH-81-6, 1981.)

8.2.18.1 Brief Theory of the Effects of Air-Flow Direction on Silencer Performance

In examining the influence of air flow on the acoustic DIL, observers have found that air flow affects sound transmission in three major ways: (1) convection, (2) refraction, and (3) flow modification of the acoustic impedance of the duct walls. Since the third effect is rather insignificant for silencers using absorptive materials, it will not be discussed here.

Convection. The term “convection” signifies that the speed of sound in the forward direction is greater than in the reverse direction. As a result, the sound waves (previously referred to as the “noise field”) maintain longer contact with the absorptive boundary in the silencer in the reverse direction than in the forward-flow mode. This results in higher attenuation in the reverse direction than in the forward direction. Quantitatively, this difference between reverse-and forward-flow attenuation depends on the Mach number M in the duct, which is defined as

$$M = \frac{V}{c} \quad (8.2.29)$$

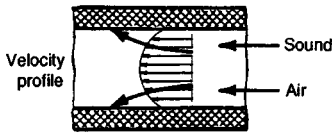
where V is the velocity of air, and c is the velocity of sound. At sea level, c is approximately 1100 ft/s (335.3 m/s), and V in an air-conditioning silencer might typically be on the order of 70-ft/s (21.3-m/s) throat velocity, or a Mach number of about 0.064.

This dependence on the Mach number is modified by whether the air-flow pattern in the flow sublayer close to the boundary is streamlined or turbulent. If the pattern is streamlined, the ratio between reverse- and forward-flow attenuation can be shown to be $(1 + M)/(1 - M)^1$; if the pattern is turbulent, the ratio is expected to be $(1 + M^2)/(1 - M^2)^2$. If the Mach number is about 0.064 and if the turbulent sublayer is streamlined, this would correspond to a theoretical ratio between reverse- and forward-flow attenuation of about 14 percent; however, much wider fluctuations have been measured under actual test conditions.

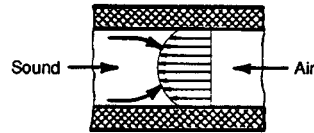
Where turbulent flow conditions control, the ratio between reverse- and forward-flow attenuation might then be on the order of 30 percent of more; consequently, it follows that shape and construction can have a major effect on silencer attenuation values and that it cannot be concluded that all silencers will necessarily behave alike. There is only one way to be sure that silencers will provide the performance specified, and that is on the basis of actual test data.

(It is interesting to note that if the velocity of air through a duct equals Mach 1, then theoretically no noise at all should be transmitted in the reverse-flow direction. In fact, experimental jet engine intake silencers have been constructed on this principle.)

Refraction. At higher frequencies, refraction begins to be significant, and it works in opposition to the effect of convection. That is, refraction tends to increase high-frequency attenuation in the forward-flow direction and decrease it in the reverse-flow direction. This situation is illustrated schematically in Fig. 8.2.33. As a sound wave travels in the forward-flow direction, there is a tendency for it to be refracted toward the boundary, which leads to smaller attenuation in the reverse-flow direction. This effect is significant only at higher frequencies when the wavelength is smaller than the cross-sectional dimensions of the duct.



Under forward-flow conditions, high-frequency sound is refracted into the duct-silencer walls.



Under reverse-flow conditions, sound is refracted away from the walls and toward the center of the duct silencer.

FIGURE 8.2.33 The refraction of sound under forward- and reverse-flow conditions. (*Application Manual for Duct Silencers, Bulletin 1.0301.3, Industrial Acoustics Company.*)

It will be noted from the data in Fig. 8.2.55 that in the reverse-flow mode, silencer attenuation falls off markedly from the sixth octave band upward and increases for the forward-flow mode (Refs. 9 to 11).

8.2.19 COMBINING ACTIVE AND DISSIPATIVE SILENCERS:

Active noise control presently is not a broadly used method for achieving HVAC noise control because of relatively higher costs compared with dissipative silencers. However, for selected applications, there may be significant benefits in combining the active technology with the broad band performance of dissipative sound absorptive silencers.

In active noise cancellation, sound is cancelled by destructive interference. The basis of all active attenuation systems is that the noise from a secondary source is generated with a mirror image wave form of the primary sound field to cancel unwanted sound downstream of the attenuator. The secondary noise source must also be controlled.

The secondary source must be of the same order of magnitude as the noise to be cancelled and must also be controlled. Fig. 8.3.34 shows how noise and anti-noise sources cancel each other out.

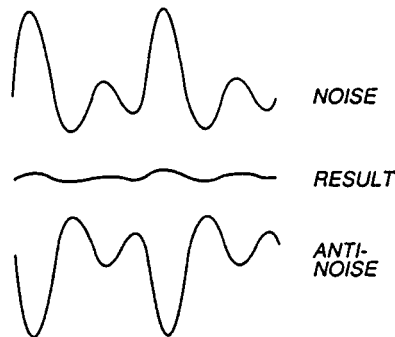


FIGURE 8.2.34 Noise Cancellation Theory. (Ref. 20)

A simple active noise control system, Fig. 8.2.35, utilizes two microphones; one for input and one for error corrections, a loudspeaker and a controller. Unlike dissipative silencers, active silencers will consume small amounts of electrical power and will require equipment maintenance from time to time. This may entail replacement of loudspeakers, which must often operate continuously in some times rugged environments.

Hybrid Active Dissipative Silencers: The most effective application of the active silencer principle in HVAC Systems is a “hybrid” combination of active and dissipative silencers. Table 8.2.10 shows performance of one combination. Active duct silencers for frequencies in excess of 500 Hz are generally not considered practical due to increasingly complex “cross modes” at the higher frequencies.

The active silencer performance shown in Table 8.2.10 provides attenuation up to 500 Hz. The acoustical characteristics of dissipative silencers for 3 m and 900 mm long silencers respectively, provide additional low frequency attenuation as well as greater amounts of mid and high frequency attenuation. Other selections of dissipative silencers can be combined with the active silencer where the dissipative silencer provides a larger amount of low frequency but most of the attenuation above 500 Hz. Depending on space considerations, these can also be designed with very minimal pressure drop.

For most active silencer systems performance can be limited by the presence of excessive turbulence in the airflow detected by the microphones. Manufacturers recommend using active silencers only where duct velocities are less than 1500 fpm and where the duct configuration is conducive to smooth, evenly distributed airflow. (Ref: 1995 ASHRAE Handbook, HVAC Applications).

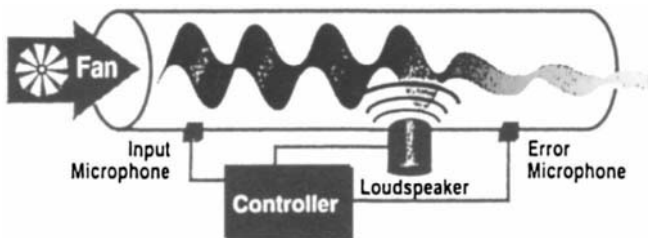


FIGURE 8.2.35 Active Duct Silencer. (Ref. 21)

TABLE 8.2.10 DIL of Dissipative Silencers and TCM (Tight Coupled Monopole) Active Attenuator.

	63	125	205	500	1K	2K	4K	8K
3 m (10 ft)	13	26	42	52	55	53	51	42
900 mm (3 ft)	2	7	11	15	21	26	18	11
TCM	10	11	16	13	1	0	0	0

8.2.20 SOUND TRANSMISSION THROUGH DUCT WALLS—DUCT BREAK-OUT AND BREAK-IN NOISE

The break-out phenomenon in particular illustrates the importance of reducing fan noise by means of silencers directly after the fan. Otherwise, duct runs that lack an adequate acoustic design may radiate unacceptably high noise levels into occupied spaces.

Air ducts are commonly manufactured from light-gauge sheet materials, which provide only partial containment of the sound field within the duct. Internal noise can be transmitted into the surrounding space (break-out), and in some cases external noise can pass into the duct (break-in), which then becomes a path for noise to travel into other occupied areas.

The phenomena of break-out and break-in sound transmission are illustrated in Fig. 8.2.35 and 8.2.36.

The magnitude of the sound transmission loss (TL) of a duct wall differs from that of a plenum wall panel due to the frequency-dependent nature of the sound propagation within the duct. If the cross-sectional dimensions of the duct are smaller than one-half of the wavelength, only plane waves can propagate within the duct. The vibration response of the duct walls and the pattern of radiation of sound from the duct are governed by the directional characteristics of the internal sound field. The forced response of the duct wall is proportional to the local sound pressure, which propagates in an axial direction at a speed that is equal to or greater than the speed of sound.

Practical TL curves have been developed that are divided into two regions (Ref. 2): one where plane-mode transmission within the duct predominates, and another where cross-modes prevail.

For break-out, the limiting frequency f_1 between these curves is given by

$$f_1 = \frac{24,134}{(ab)^{0.5}} \quad (8.2.30)$$

where a and b are the duct cross-sectional dimensions in inches; when working

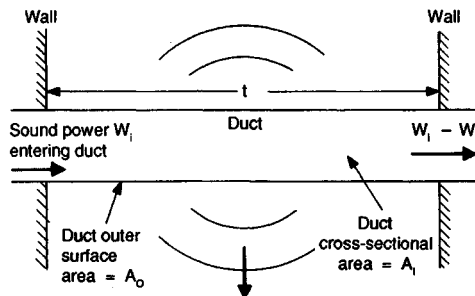


FIGURE 8.2.36 Break-out sound transmission through duct walls. (*ASHRAE Handbook 1987 Systems*, chap. 52, "Sound and Vibration Control," American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, 1987.)

with metric units, convert millimeters to inches before using Eq. (8.2.30) (mm divided by 25.4).

For break-in, the lowest acoustic cross-mode frequency is used as the limiting frequency:

$$f_1 = \frac{6764}{a} \quad (8.2.31)$$

where a is the larger duct dimension in inches; when working with metric units, convert millimeters to inches before using Eq. (8.2.31) (mm divided by 25.4).

Below the limiting frequencies, break-out TL is given by

$$TL_{out} = 10 \log \left(\frac{fq^2}{a+b} \right) + 17 \quad (8.2.32)$$

and break-in TL is given by the larger of

$$TL_{in} = TL_{out} - 4 - 10 \log \frac{a}{b} + 20 \log \frac{f}{f_1} \quad (8.2.33)$$

or

$$TL_{in} = 10 \log \left[12l \left(\frac{1}{a} + \frac{1}{b} \right) \right] \quad (8.2.34)$$

where f = frequency, Hz

q = mass per unit area of duct wall, lb_m/ft² (kg/m² × 0.2048)

l = duct length, ft (m)

Above the limiting frequencies, break-out TL is given by

$$TL_{out} = 20 \log qf - 31 \quad (8.2.35)$$

and break-in TL is given by

$$TL_{in} = TL_{out} - 3 \quad (8.2.36)$$

Air ducts are frequently installed above suspended ceilings or under access flooring. These confined spaces have the effect of modifying the radiating pattern of sound around the duct. Close proximity of the duct to a concrete slab will modify the response of the duct wall; the slab will also act as a reflecting plane to the overall sound radiation. Rigid partitions perpendicular to the axis of the duct may cause standing waves at frequencies where the wavelength is equal to the distance between the partitions. These standing waves can raise the local sound pressure levels in the occupied space by as much as 10 dB.

Circular ducts provide much higher TL than do rectangular ducts in the low frequencies, where most duct break-out problems occur. At higher frequencies, however, circular ducts can exhibit a resonance phenomenon at the duct's so-called "ring frequency," where the TL is sharply reduced.

Examples of break-out and break-in TL values are shown in Table 8.2.10 for an unlined rectangular duct made from 22-gauge [0.034-in (0.85-mm)] sheet steel and measuring 44 in (1118 mm) wide by 12 in (305 mm) deep; the break-out values for an equal-area circular-section duct 26 in (660 mm) in diameter made from spiral-wound 24-gauge [0.028-in (0.7-mm)] steel are shown for comparison. More

comprehensive listings of break-out and break-in TL are shown in Tables 8.2.11 to 8.2.17.

Lagging on the outside of ductwork is often used to increase the TL values. The increase in performance due to the lagging will depend on the type and rigidity of the lagging material. A hard outer layer of sheet metal or gypsum board may not be a very effective means of reducing low-frequency noise caused by resonance effects in rectangular ducts. Limp covering materials that effectively add mass to the duct wall may improve the TL values by reducing wall response without adding stiffness. In critical situations, it may be necessary to apply panels with air spaces to the duct surfaces for maximum noise reduction.

Undoubtedly, more correlation between field and empirical data is required on break-out and break-in noise. Some acoustic consultants and engineers consider

TABLE 8.2.11 Examples of Duct Break-out and Break-in TL versus Frequency

Duct and TL type	Octave band center frequency, Hz							
	63	125	250	500	1000	2000	4000	8000
Rectangular,* TL _{out}	19	22	25	28	31	37	43	45
Rectangular,* TL _{in}	14	14	22	25	28	34	40	42
Circular,† TL _{out}	45	50	26	26	25	22	36	43

*Duct size: 44 by 12 in (1118 by 305 mm), 22 ga [0.034 in (0.85 mm)].

†Duct size: 26-in (660-mm) diameter, 24 ga [0.028 in (0.7 mm)].

TABLE 8.2.12 TL_{out} versus Frequency for Various Rectangular Ducts

Duct size*		Octave band center frequency, Hz							
in (mm)	Gauge								
	in (mm)	63	125	250	500	1000	2000	4000	8000
12 × 12 (300 × 300)	24 ga								
	0.028 (0.7)	21	24	27	30	33	36	41	45
12 × 24 (300 × 600)	24 ga								
	0.028 (0.7)	19	22	25	28	31	35	41	45
12 × 48 (300 × 1200)	22 ga								
	0.034 (0.85)	19	22	25	28	31	37	43	45
24 × 24 (600 × 600)	22 ga								
	0.034 (0.85)	20	23	26	29	32	37	43	45
24 × 48 (600 × 1200)	20 ga								
	0.04 (1.0)	20	23	26	29	31	39	45	45
48 × 48 (1200 × 1200)	18 ga								
	0.052 (1.3)	21	24	27	30	35	41	45	45
48 × 96 (1200 × 2400)	18 ga								
	0.052 (1.3)	19	22	25	29	35	41	45	45

*All duct lengths are 20 ft (6 m).

TABLE 8.2.13 TL_{in} versus Frequency for Various Rectangular Ducts

Duct size*		Octave band center frequency, Hz							
in (mm)	Gauge								
	in (mm)	63	125	250	500	1000	2000	4000	8000
12 × 12 (300 × 300)	24 ga								
	0.028 (0.7)	16	16	16	25	30	33	38	42
12 × 24 (300 × 600)	24 ga								
	0.028 (0.7)	15	15	17	25	28	32	38	42
12 × 48 (300 × 1200)	22 ga								
	0.034 (0.85)	14	14	22	25	28	34	40	42
24 × 24 (600 × 600)	22 ga								
	0.034 (0.85)	13	13	21	26	29	34	40	42
24 × 48 (600 × 1200)	20 ga								
	0.04 (1.0)	12	15	23	26	28	36	42	42
48 × 48 (1200 × 1200)	18 ga								
	0.052 (1.3)	10	19	24	27	32	38	42	42
48 × 96 (1200 × 2400)	18 ga								
	0.052 (1.3)	11	19	22	26	32	38	42	42

*All duct lengths are 20 ft (6 m).

that the TL data presented here (from Ref. 2) may be overstated when translated to field installations; for instance, the break-out noise sound levels are likely to be higher than would be arrived at by using the TL figures in Tables 8.2.10 to 8.2.16. However, in the meantime, the above procedures (including Tables 8.2.10 to 8.2.16) can be used, bearing in mind that the introduction of safety factors might be in order.

8.2.21 NOISE CRITERIA

Noise is unwanted or objectionable sound, and numerous standards define its limits for specific types of noisemakers, specify how the sound is to be measured, and in certain instances specify when. These standards are published by local and national government agencies, national and international standards organizations, the military, professional societies, and others. A few of these standards (or criteria) are given below.

8.2.21.1 dBA Criteria

One way of rating sounds is by means of the A scale, a sound-level meter weighing network that approximates the response of the human ear to sound. Both the human ear and the A-weighting network are more sensitive to high-frequency than low-frequency sound. In decibels, A-scale levels are expressed as dBA. Typical noise source dBA levels are shown in Table 8.2.18.

CHAPTER 8.3

VIBRATION CONTROL

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8.3.1 INTRODUCTION

Chillers, pumps, blowers, cooling towers, ducts, piping, rigid electrical conduits, etc., that are rigidly bolted to a structure, transmit 100 percent of their vibratory energy. The introduction of properly selected vibration isolators will reduce this transmitted energy to the point where it is completely imperceptible or so minor as to no longer be annoying to the occupants, structurally destructive, or detrimental to critical manufacturing processes. "Vibration isolation efficiency" is defined as the percentage of vibration that is no longer transmitted to the structure because of the introduction of vibration isolators.

8.3.2 THEORY

The vibration problem can be approached on a theoretical basis by using Eq. (8.3.1), the theoretical vibration efficiency equation, which represents an isolated machine as shown in Fig. 8.3.1.

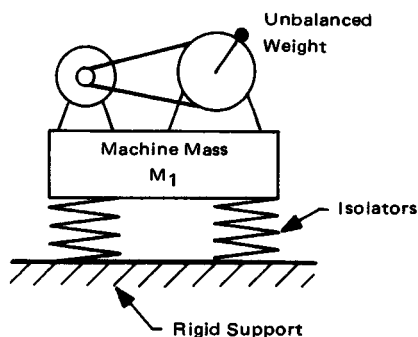


FIGURE 8.3.1 Isolated machine.

$$E = 100 \left[1 - \frac{1}{(f_d/f_n)^2 - 1} \right] \quad (8.3.1)$$

where E = percentage of vibration isolated (efficiency)
 f_d = disturbing frequency of the isolated machine
 f_n = natural frequency of the isolated machine

The disturbing frequency should be taken as the r/min of either the equipment or the driver, whichever is lower. All equipment has some unbalance at the primary speed, and this approach is conservative, since any higher-frequency vibration used in the formula would result in overly optimistic values for the primary disturbance.

This equation states that a given percentage of the vibration can be kept out of the structure by installing the machine which is creating the vibration at a known frequency (usually the machine r/min) on a system of vibration mountings. These should resonate at a frequency that is very much lower than the disturbing frequency described above. When this ratio of disturbing frequency to the natural frequency is 3:1, about 90 percent of the transmitted vibration is theoretically eliminated. Refer to Fig. 8.3.1 to help visualize the mechanical system represented by Eq. (8.3.1). A motor is shown driving a rotating component. Assume that there is one point of unbalance in the rotating machine and none in the motor. These two components would be kept in their relative positions by either a steel base or a steel base filled with concrete. The total weight of the system would be the weight of the motor plus the weight of the machine plus the weight of the base.

Assuming that the system has only one degree of freedom and that there are no external connections, when we push the system down it will bounce back up and continue to oscillate vertically at a specific frequency. This frequency f_n is known as the "natural" frequency because it continues to move at this rate with no further introduction of energy.

When we turn the machine on, the unbalanced force that is centrifugally generated will force the system to vibrate at the operating speed of the driven machine. This frequency is known as the "forcing" or "disturbing" frequency f_d . Examination of the efficiency equation (8.3.1) shows us that the larger the ratio between this forcing frequency and the natural frequency, the greater will be the degree of vibration isolation.

The only unknown in the equation is f_n . The equation used in determining this frequency is a simple relationship to the static deflection, provided the isolation material that is used has a straight-line deflection curve and virtually no damping. (Mass in no way enters the equation.) The frequency is dependent upon acceleration provided by the return spring force acting against the mass. When the deflection is directly proportionate to increased mass, as a function of the spring stiffness, mass and stiffness conveniently drop out and deflection is all that is needed to determine frequency. Static deflection refers to the actual difference in height between the unloaded spring and the spring in the deflected condition. For example, if the spring were 6 in (150 mm) tall to begin with and 4 in (100 mm) tall under load, the static deflection would be 2 in (50 mm).

If the material does not have a straight-line deflection curve, as in the case of some rubber materials in compression, or the material has appreciable dynamic stiffness (which means that in motion it acts like a stiffer material than the deflection would indicate), frequency tests can be run to determine the frequency at various loads and deflections and the curves plotted. This is normally the case for natural rubber and other elastomers and for materials such as fiberglass, felt, cork, and sisal. When using these materials, deflection is a very poor indicator of the natural

frequency, and test curves must be referred to for any degree of accuracy when inserting f_n in the efficiency equation. The same applies to more exotic devices like air springs where there is no deflection under load but only a change in air pressure. Frequencies can be calculated, but test data are far more reliable.

Equation (8.3.2) refers to helical steel springs which are free of damping and have uniform deflection rates. The natural frequency f_n is then expressed as

$$f_n = \sqrt{\frac{188(25.4)}{d_1}} \quad (8.3.2)$$

where f_n = natural frequency, cycles/min (Hz)

d_1 = static deflection in the spring supports, in (mm)

In looking at this as a mechanical system, it is simpler to think in terms of the pendulum. The frequency of a pendulum is dependent on the length of the pendulum from the rotation point to the center of the bob or mass and is in no way a function of the mass. The frequency drops as the pendulum becomes longer and increases as it is shortened. Pendulum length is analogous to spring deflection. The longer the pendulum, the lower the frequency. The more static deflection, the lower the natural frequency of the system.

Returning to the efficiency equation, let us try a solution referencing a machine with a disturbing frequency of 600 cycles/min. A natural frequency of about 200 cycles/min would be required to attain a ratio of 3:1 and to arrive at 87.5 percent efficiency. Since exact numbers are not critical, it is easier to use an efficiency chart (Fig. 8.3.2).

This is still a theoretical discussion and not to be used for practical purposes. This is only introductory to aid in understanding.

The operating speeds or disturbing frequencies are listed across the bottom of Fig. 8.3.2. The vertical scale is a continuous solution of the natural frequency equation, with the deflection on the left and corresponding natural frequencies on the right. The efficiency lines run diagonally.

To use this chart, start at the bottom with the lowest operating speed of the equipment. Move up vertically and intersect the desired efficiency line and then move over to the left to pick up the natural frequency of the mountings required for that efficiency and the corresponding spring deflection. We have outlined this procedure for 90 percent at 600 r/min in the dark lines. For all practical purposes, we would need mountings with about 1 in (25.4 mm) of static deflection in order to obtain the 90 percent efficiency.

It is important to emphasize that *commercial vibration control is not an exact science*. The weight of the equipment provided by manufacturers is approximate at best, and the location of the center of gravity is even more nebulous. Using commercially available mountings, *it is impossible to select mountings for exact deflection*. Vibration isolation commodities are not precision devices, and there cannot be exact cataloged springs for every load. For these reasons, deflections are generally specified as minimums. Deflections beyond those that are minimal will merely lower the natural frequency and improve performance.

Transmitted vibration is that portion of the vibration that is still sensed by the structure. At 90 percent efficiency, there is 10 percent transmission. Efficiency is basically a salesperson's term, probably developed because of the grading system that we all grew up with: 85 percent fair, 90 percent good, and 95 percent excellent. In our work, these numbers can be extremely deceptive. In comparing 95 to 90 percent, we assume an improvement of only 5 percent. However, the comparison

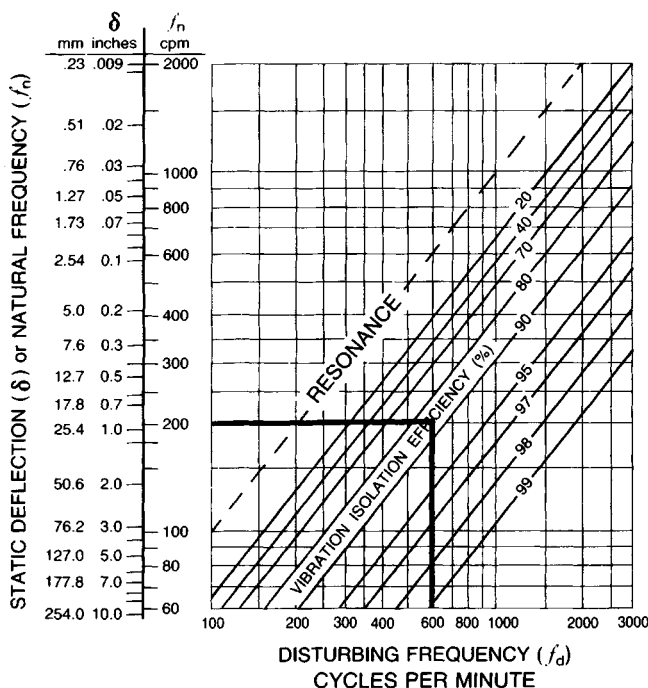


FIGURE 8.3.2 Theoretical isolation efficiency chart.

of 5 to 10 percent transmission shows that only half the remaining force is transmitted. Neither number can be considered alone as the source of the vibration, and what we have to eliminate is the deciding factor.

In broad terms a 125-hp (93-kW) pump will generate five times the vibratory energy of a 25-hp (19-kW) pump. Therefore, the isolation provided for the 125-hp (93-kW) pump must be five times more efficient to reduce the force transmission to a similar level. An 80 percent efficiency with 20 percent transmission for the 25-hp (19-kW) pump is equivalent to 96 percent efficiency with 4 percent transmission for the 125-hp (93-kW) unit.

8.3.3 APPLICATION

8.3.3.1 Basic Considerations

A completely arithmetical, rather than conventional, approach provides a better mechanical visualization of what is really going on. The results can be confirmed on the efficiency chart.

There are six basic considerations once the installation is well away from resonance ($f_d/f_n \leq 3:1$):

1. Efficiency is controlled by static deflection, and transmissions reduced in direct proportion to the increase in deflection.
2. If the frequency of the isolator approaches the operating speed of the equipment, resonance is approached.
3. When approaching resonance, the dynamic motion generated by the unbalanced force is amplified.
4. Dynamic motion is controlled by the unbalanced force and its relationship to the total mass. For all practical purposes, at higher frequency ratios the frequency ratio itself no longer influences motion.
5. Once the frequency ratio is 3:1 or greater, motion should only be reduced by the introduction of mass.
6. Assuming the unbalanced forces act through the center of gravity, motion is reduced in direct proportion to the increase in mass.

In reality, when a rotating machine vibrates on isolation mountings, the foundation has a small rotary motion. Since floors are much more sensitive in the vertical direction, the other modes are usually ignored and installations are always visualized moving vertically.

A spring constant is referred to as k and is defined as the number of pounds (kilograms) required to deflect the spring 1 in (25 mm). Thus a spring with a constant k of 1000 lb/in (18 kg/mm) would deflect 0.5 in (13 mm) at 500 lb (227 kg) and 1.0 in (25 mm) at 1000 lb/in (454 kg).

A system constant is normally defined as the number of pounds (kilograms) required to deflect all the supports of a system 1 in (25 mm) simultaneously. Using four springs with a constant k of 1000 lb/in (18 kg/mm) each, one in each corner of an installation, the system constant would be 4000 lb/in (72 kg/mm). Thus 2000 lb (907 kg) of equipment would deflect the system 0.5 in (13 mm) and 4000 lb (1814 kg), 1.0 in (25 mm).

Since the spring rate is uniform, this also means that if the upward vibratory force pulled the equipment 0.10 in (2.5 mm) up from the neutral position, it would be reducing the spring load by 0.10 of 4000 lb (1814 kg), or 400 lb (181 kg). As the vibratory motion pushed the installation 0.10 in (2.5 mm) below the neutral position, there would be an increase in the spring load of the same 400 lb (181 kg). Since the bottom of the spring is attached to the structure, this change in force of ± 400 lb (181 kg) is what the structure sees as a change in the static loading. This occurs at 600 r/min. A change in static loading at a particular frequency is another definition of vibration.

This approach to the problem is best illustrated by Fig. 8.3.3.

If the machine is running at 600 r/min, the vibratory force transmitted would be ± 400 lb (181 kg) at 600 cycles/min.

Assume that the ± 400 lb (181 kg) is unacceptable in the structure. The instinctive solution would be to use a larger mass. A traditional mass ratio is three times the equipment weight, bringing the system to 16,000 lb (7258 kg) (Fig. 8.3.4). Continuing with a 1-in (25-mm) deflection spring grouping, the system constant would have to be raised to 16,000 lb/in (286 kg/mm) by using stiffer individual springs or clusters of four times the original spring groupings.

In basic rule 6, it was stated that the vibratory motion would be reduced in direct proportion to the increase in mass. Therefore, the amplitude of 0.10 in (2.5 mm), as in Fig. 8.3.3, would now become 0.025 in (0.64 mm). Following the sketches across, it is found that the reduced motion is merely acting against a proportionately

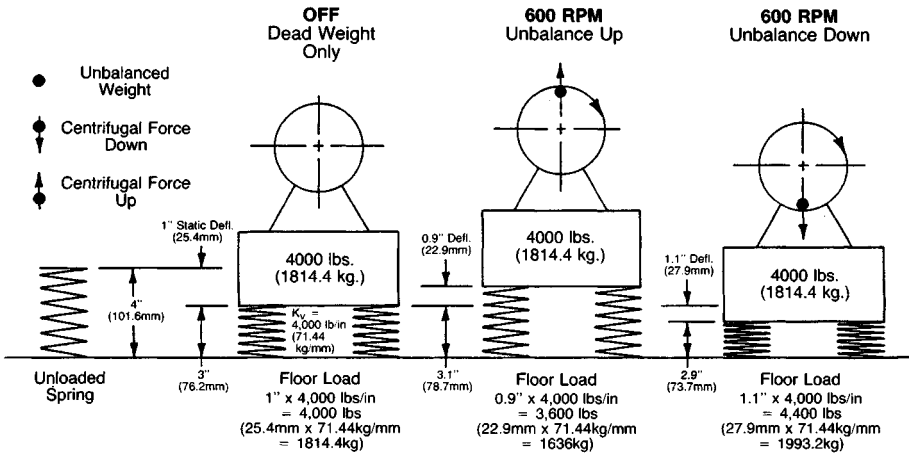


FIGURE 8.3.3 Vibratory transmission, 4000-lb (1800-kg) load. 1-in (25-mm) deflection.

stiffer spring constant so that there is no reduction in vibration transmission but only in amplitude.

If the machine is running at 600 r/min, the vibratory force transmitted remains $\pm 400 \text{ lb}$ (181 kg) at 600 cycles/min.

The question then comes up as to how to actually reduce the transmitted vibration. Assume we wish to reduce this transmission by 75 percent so that the end result is $\pm 100 \text{ lb}$ (45 kg). Basic rule 1 states the efficiency is controlled by static deflection and transmission reduced in direct proportion to the increase in deflection. Rule 4 also states that for all practical purposes, the frequency ratio no longer influences motion.

Therefore, increase the deflection to 4 in (100 mm), as in Fig. 8.3.5. Since this larger deflection will give us a lower natural frequency, there will be no noticeable

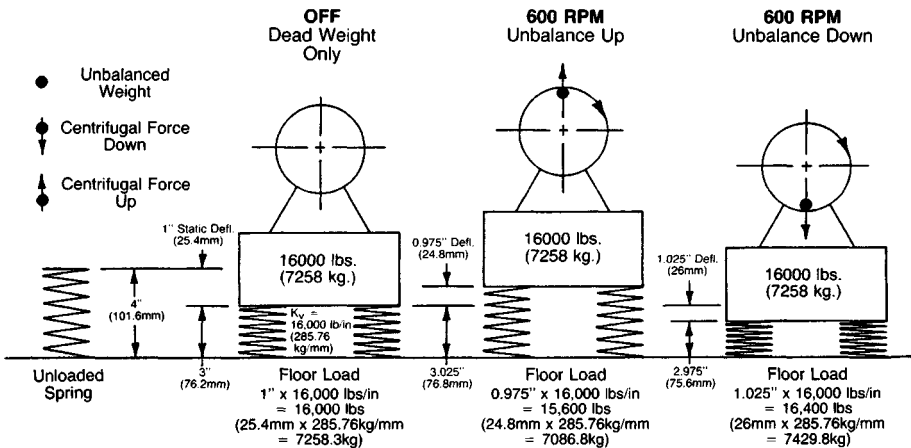


FIGURE 8.3.4 Vibratory transmission, 16,000-lb (7300-kg) load. 1-in (25-mm) deflection.

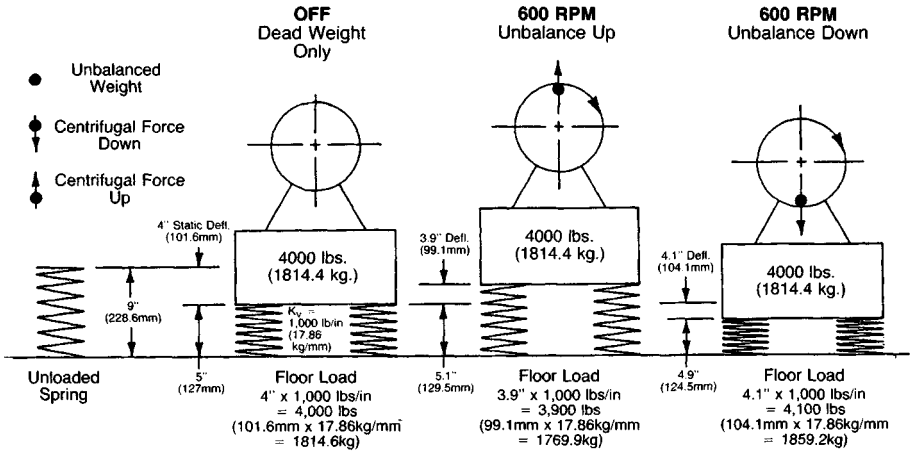


FIGURE 8.3.5 Vibratory transmission, 4000-lb (1800-kg) load. 4-in (100-mm) deflection.

difference in the amplitude. The example shows that the spring constant has dropped to 1000 lb/in (18 kg/mm). Since the amplitude remains at the original ± 0.10 in (2.54 mm), this amplitude multiplied by the new spring constant results in a force transmission of only ± 100 lb (45 kg) at 600 cycles/min.

The problem can now be approached on the basis of reducing both amplitude and transmission by reusing the total weight of 16,000 lb (7258 kg) and providing 4-in (100-mm) deflection, as shown in Fig. 8.3.6. A reduction is now made in both the amplitude to 0.025 in (0.64 mm) and in the transmitted force to ± 100 lb (45 kg).

The vibratory force transmitted would be ± 100 lb (45.5 kg) at 600 cycles/min.

This really agrees with the efficiency chart (Fig. 8.3.2), as a 600-r/min machine isolated by 1-in (25-mm) and 4-in (100-mm) deflection springs would show effi-

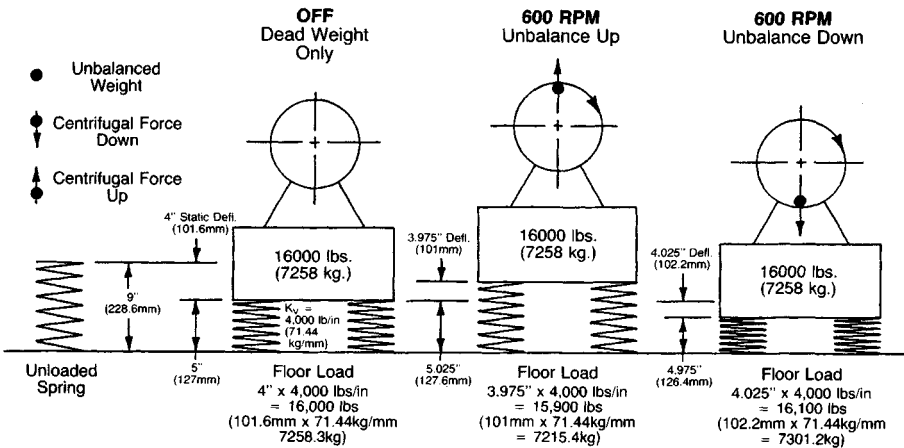


FIGURE 8.3.6 Vibratory transmission, 16,000-lb (7300-kg) load. 4-in (100-mm) deflection.

ciencies of 90 and 97.5 percent, respectively. Transmission reduction is 10:2.5, which is the same factor 4 shown by the arithmetic.

Both the efficiency equation and the efficiency chart are based on the completely false assumption that the floor stiffness or frequency in an upper story is very high as compared to the stiffness or frequency of the isolator. In reality, the floor has a deflection of its own and a natural frequency which can be low enough to mandate the use of isolators with very much higher deflections than indicated by the chart. Figure 8.3.7 shows the actual conditions in a structure. Rather than a simple system with the machine or machine foundation resting on springs on a relatively unyielding support, the springs are supported by a spring board with a finite mass of its own. Schematically, this is sketched in Fig. 8.3.8. The machine mass rests on springs on the floor mass, and the floor stiffness is shown by a second set of springs. Although floors are supported by beams connected to vertical columns, ground-level vertical stiffness really exists only at the columns and not in between.

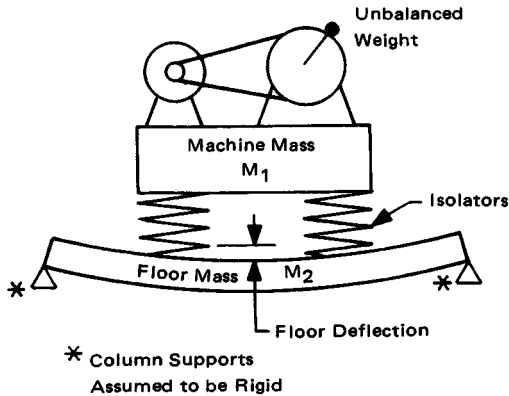


FIGURE 8.3.7 Actual structure conditions (floor deflection exaggerated).

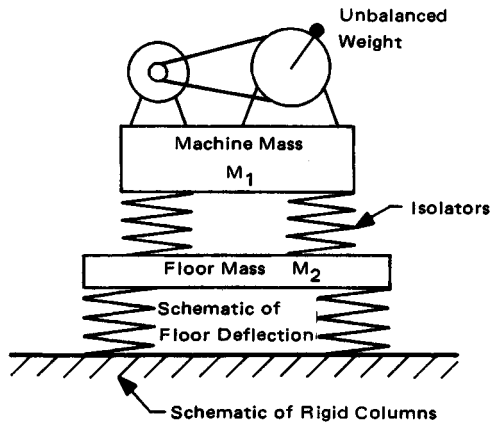


FIGURE 8.3.8 Schematic of floor deflection.

The worldwide structural limit on floor deflection is 1/360th of the span. In many commercial buildings design spans are at least 360 in (9144 mm), or 30 ft (9.14 m). This means that the structural engineer is allowed a floor deflection of 1 in (25 mm) at the center of the span when the floor is fully laden with both live and dead loads.

Let us make the assumption that in a particular area where a 125-hp (93 kW) pump is installed that the floor is loaded to half dead load plus live load. The floor deflection would then be 0.5 in (13 mm). If the pump is running at 1750 r/min, a quick reference to the efficiency chart shows that a mounting deflection of 0.1 in (2.54 mm) should provide 90 percent efficiency or 10 percent transmission. However, the 0.5-in (13 mm) floor deflection would be five times the deflection of the isolator. The actual efficiency would be influenced by the floor's 0.5-in (13-mm) deflection, the mass of the floor, and the floor's damping characteristics. The 90 percent theoretical efficiency could never be attained, and depending on the combination of conditions, the actual efficiency might slide down to 50 or 40 percent and not meet the requirement.

Therefore, rather than relying on the theoretical method, commercial selection of isolators has evolved into using isolators with deflections that equal or exceed the floor deflection to attain acceptable transmission levels. The efficiency chart should only be used as a tool to learn the subject and gain direction.

Studies show that the floor stiffness is greatly in excess of the isolator stiffness, because the mass of the floor is much greater than the mass of the machine that is to be isolated. While this may be of importance when isolating small equipment, it is certainly not significant with large pumps and chillers.

On a day-to-day basis, the cost of an involved engineering investigation of a commercial structure's stiffness and resonance, along with the possibility of error in these conclusions, dictates the continued use of the more conservative floor-deflection rather than stiffness approach. The cost of isolation is small as compared to the cost of an installed air-conditioning system. Possible savings in using lower-deflection materials are in no way proportionate to complete loss of occupancy or lower rental rates in a noisy structure or recourse to very expensive retrofits.

The recommended deflections shown in the selection guide (Table 8.3.6) are based on empirical data gathered through 40 years of installation experience, as well as discussions with mechanical and acoustical engineers, architects, and manufacturers. The deflections were influenced by operating speeds, size of equipment, the equipment as a vibratory source, and the sensitivity of the floor structure in terms of construction and floor span.

8.3.3.2 Isolation Materials

An "isolation material" can be defined broadly as any resilient material that will accept a load on a permanent basis and produce a resonant or natural frequency that is reasonably consistent and predictable. It is also important that any increase in this frequency is small as the material ages.

Vibration Pads. "Elastomeric" describes any rubberlike material. While natural rubber has the best performance characteristics, it is generally not used commercially because of aging when exposed to oxygen, ozone, or oil. A synthetic elastomer similar in properties to natural rubber but lower in cost is the oil derivative SBR (styrene-butadiene rubber). It is very commonly used where there are no specific aging requirements. The neoprenes are not quite as resilient as natural rubber or SBR, but because of their very excellent aging characteristics, better grades of

pad are either all neoprene or neoprene blended with SBR or natural rubber to reduce cost.

While the selection of the proper elastomer is an important choice in terms of performance and aging, the physical properties of the compound are specifically controlled by formulation. The polymers are not used by themselves but mixed with other materials such as carbon black and clay to provide reinforcement. Larger ratios of these fillers will reduce cost but with penalty to physical properties. As a general statement, a ratio of one-third fillers to the selected polymer produces the best physical and dynamic properties. When aging is the primary concern, the formulation should contain only neoprene and not a blend of the neoprene with natural rubber or SBR. Unless a specification states exactly what the physical properties and ingredients must be, the material that is furnished will only follow the manufacturer's conscience. Unfortunately, there are no industry standards as to when a pad can be called neoprene or natural rubber. Products are available that are made with as little as 5 percent neoprene but still referred to as commercial-grade neoprene. A good guide to quality are the AASHO standards shown in Table 8.3.1.

Other than foams, which do not have enough capacity or stability to be used as isolators, air-free rubber materials are incompressible. When a load is applied to a pad, it changes shape but does not lose volume. The ability to change shape is controlled by the shape factor and the material's hardness. Since a pad used in compression can only change its volume by bulging, unconfined edges are referred to as "escape area," whether internal in the form of holes or external. The term "shape factor" is the ratio of the loaded area to the escape area. The lower the shape factor, the more deflection at a particular load.

Thus a 4-in (100-mm) square pad that is 1 in (25 mm) thick, covered completely either by the equipment or by a steel plate, would have a loaded area of 16 in² (100 cm²). Since the perimeter is 16 in (400 mm) and the pad 1 in (25 mm) thick, the escape area is also 16 in² (100 cm²). The shape factor (load area divided by escape area) would be 16/16 (100/100), or 1. Assuming that the hardness remains the same, we could increase the load-carrying capacity of this pad by using two ½-in (12.5-mm) thick pads with a steel plate separating the two layers. Since the loaded area would remain 16 in² but the escape would not be 16 in (400 cm) multiplied by 0.5 in (12.5 mm), or 8 in² (50 cm²), the shape factor would be increased to 2 with a lower deflection for the same imposed load. These relationships are shown in shape-factor curves (Fig. 8.3.9). These curves are empirical based on test data, and small variations will be found from one publication to another.

Hardness is measured by a durometer (in units called "duros"), a clocklike gauge with a penetration probe on the bottom. Pads are normally used in 30 to 70 duro in 10-point increments. Since rubberlike materials cannot be exactly controlled, the normal acceptable variation is ± 5 of a nominal duro. To give you some idea of the feel of these durometers, common references are rubber band stock at about 30, red erasers at about 40, white erasers at 50 to 60, and the old-fashioned hard gray erasers at 70.

Automobile tires are 50 to 70 duro. The influence of hardness on load capacity is shown in Fig. 8.3.10. A 70-duro material will handle about four times the load that would be carried by the same shape in 30 duro.

Unfortunately, the harder the rubber material, the less it acts like a steel spring; the introduction of viscosity is similar to the introduction of a dashpot working in parallel with a spring. When the dashpot becomes large and the fluid stiff, vibratory forces are transmitted through the dashpot. It is for this reason that 70 duro is

TABLE 8.3.1 Physical Properties of Structural Bearings Made from Du Pont Neoprene

Physical Property	Test Method	Performance Requirements			
		HARDNESS GRADE			
		50	60	70	
Hardness, durometer A	ASTM D 2240	50±5	60±5	70±5	
Tensile strength	ASTM D 412	2500 (1725)	2500 (1725)	2500 (1725)	psi (kPa) minimum
Elongation at break	ASTM D 412	400	300	300	% maximum
Adhesion	ASTM D 429	40	40	40	lbs. per inch, minimum
Bond made during vulcanization					
Low-temperature performance	(Sample first prepared 96 hr. at - 20 ± 2°F (-29 ± 1°C) axial load 500 psi and strain of 20% “T” [effective thickness].) Shear resistance after 1 hr. at 25% shear strain not to exceed values shown	50 (35)	75 (52)	110 (76)	psi (kPa)
Resistance to heat	ASTM D 573				
Change in original properties after 70 hrs. at 212°F (100°C)					
Hardness		+15	+15	+15	points, maxi- mum
Elongation		-40	-40	-40	%, maximum
Tensile Strength		-15	-15	-15	%, maximum
Resistance to oil aging*	ASTM D 471*	+80	+80	+80	%, maximum*
Change in volume after 70 hrs. immersion in ASTM Oil No. 3 at 212°F (100°C)					
Resistance to ozone	ASTM D 1149				
Condition after exposure to 100 pphm ozone in air for 100 hrs. at 100±2°F (29±1°C) (sample under 20% strain)		No Cracks	No Cracks	No Cracks	
Resistance to permanent set	ASTM D 395	35	35	35	% maximum
Compression set after 22 hrs. at 212°F (100°C)					

*This oil aging requirement is not a part of the AASHTO Specification referenced. However, its inclusion is strongly recommended to assure use of a high-quality neoprene compound.

Source: American Association of State Highway Officials Standard Specification for Highway bridges, Table B.

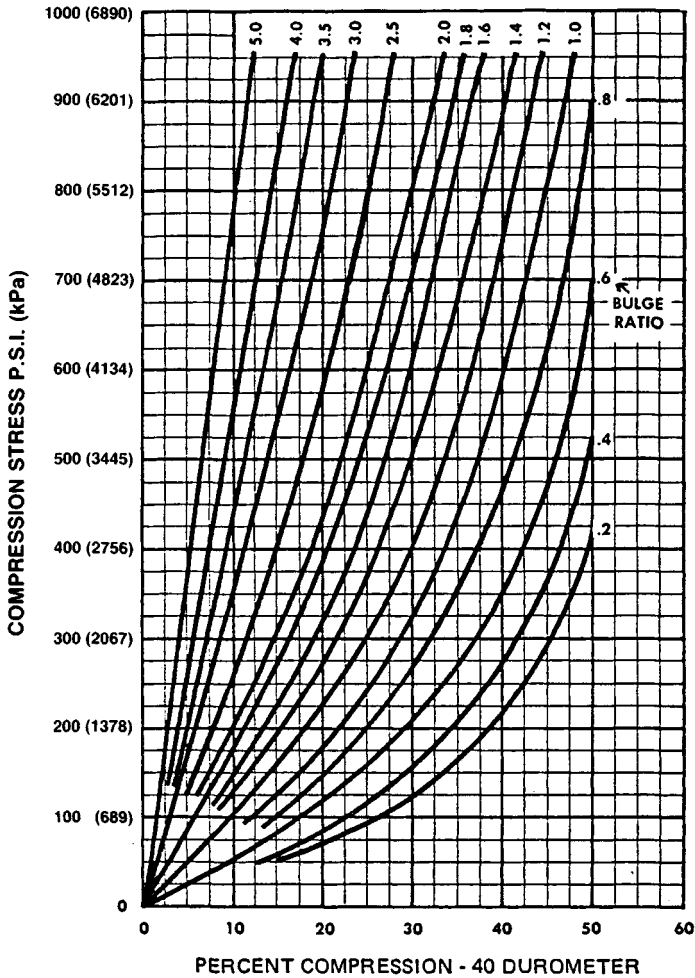


FIGURE 8.3.9 Shape-factor or bulge ratio.

considered the extreme hardness for vibration isolation. Since 30 duro becomes uneconomical for large loadings and hard to manufacture, most pad materials fall into the 40- to 60-duro range.

Figure 8.3.11 is a dynamic stiffness chart based on experimental work with neoprene compounds containing no other elastomer and minimum fillers and the actual frequency at various deflections and hardness. Increased use of fillers lowers cost and quality at the expense of performance. To see the influence of the dynamic stiffness, you need only compare these frequencies and deflections with the same deflections but lower frequencies shown in the efficiency chart, Fig. 8.3.2.

Pad deflection is limited by thickness. For the material to remain resilient and to control permanent set and creep, pad deflection should be limited to 15 percent of the thickness regardless of the rubber configuration or the rubber material.

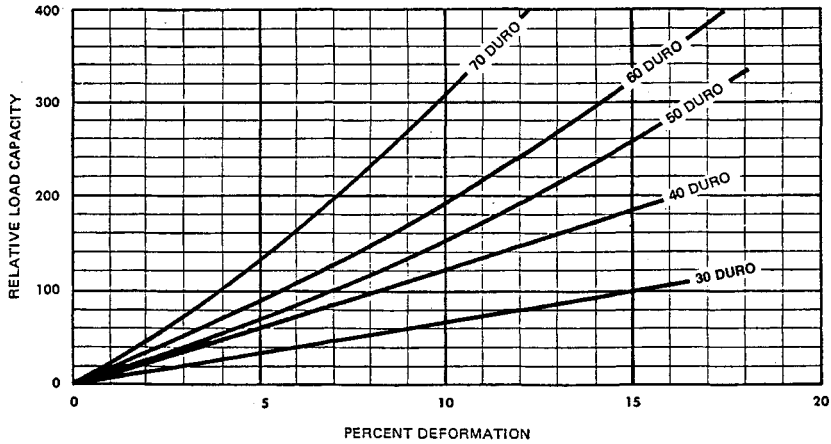


FIGURE 8.3.10 Influence of hardness on load capacity.

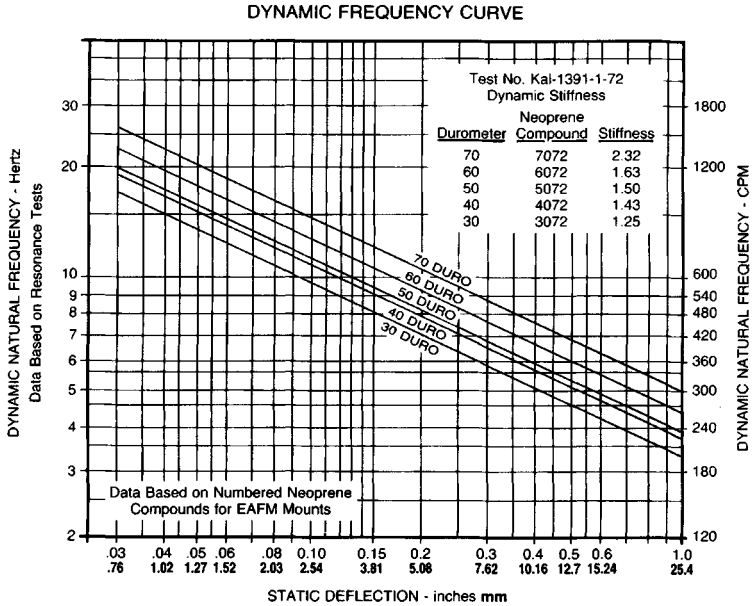


FIGURE 8.3.11 Dynamic stiffness chart.

Thus, the maximum deflection for a 1-in-thick (25-mm) pad is 0.15 in (3.8 mm) with corresponding reductions in deflection for thinner materials.

A vibration pad may be solid if for a given load it has the proper shape factor for the maximum 15 percent deflection in an acceptable durometer. Deflection can only be increased by increasing thickness. Pads may be molded thicker or made of multiple layers separated by steel shim plates.

In most cases, loadings per unit area are low, so rather than solid pads, additional escape area is needed to reduce the shape factor. Most vibration pads are molded with round or square holes and in cross-ribbed and waffle designs, as shown in Fig. 8.3.12.

Most commercial isolation pads are available in thicknesses up to $\frac{3}{8}$ in (9.5 mm). They should be used at a maximum deflection of 0.06 in (1.5 mm) per layer. Dynamic frequency is 16 Hz in the best materials. A new $\frac{3}{4}$ -in (19-mm) pad has been introduced recently with deflections of 0.11 in (2.8 mm) and a dynamic frequency of 12 Hz.

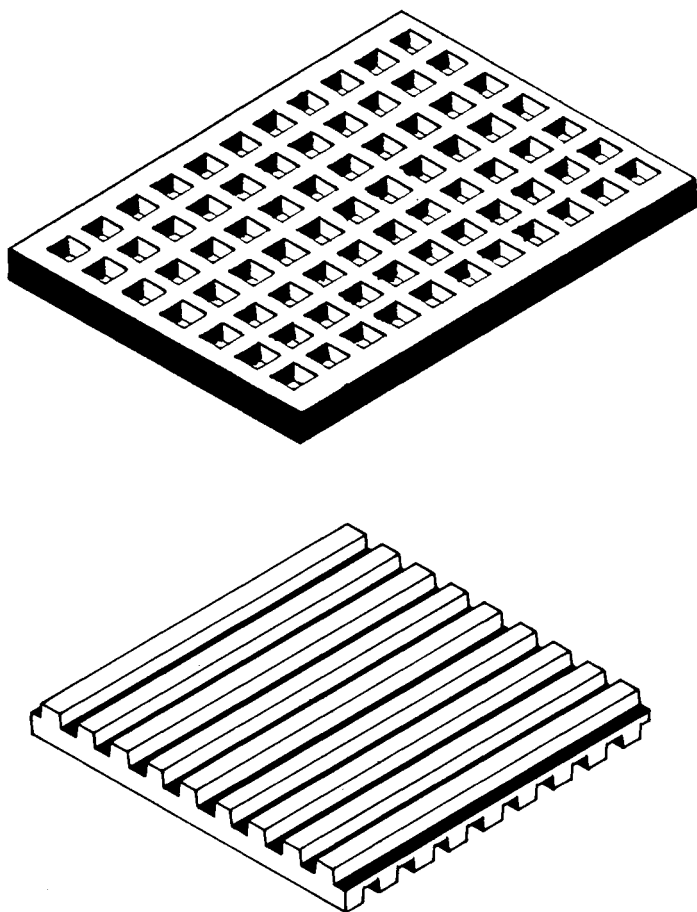


FIGURE 8.3.12 Typical vibration pads.

Other pad materials are cork, felt, sisal, and heavy-density precompressed fiberglass. Since the fiberglass is a fragile, spongelike material, it is normally covered with a neoprene or other coating to protect it against fraying and moisture. Fiberglass-pad frequency is not as sensitive to deflection and loading, so fiberglass pads are sometimes described as flat-frequency materials. However, this frequency is higher than neoprene or natural rubber at the same deflection. Table 8.3.2 shows published frequencies for 0.5-in (12.5-mm) and 1.0-in (25-mm) pads. Fiberglass pads should be avoided if there are large shear forces, such as those under a horizontal compressor.

Rubber mountings are sophisticated rubber pads. While the rubber may be loaded in shear, most commercial rubber mountings for air-conditioning applications are loaded in compression. Quality levels may be as described for pads, but the static deflections are higher only because the mountings are thicker. Rubber mountings have the advantage of provision for bolting to the equipment and to the floor when needed. Many of the newer designs have rubber under and over the base and top plates so that they can be slipped into place without bolting in stationary applications.

Springs. Spring mountings are generally required to provide the minimum deflection needed to compensate for structural flexibility. The heart of any steel spring mounting is the spring itself. It should be designed with a minimum diameter-to-deflected-height ratio of 0.8 so that the horizontal spring constant K_h is a minimum of 80 percent of the vertical K_v . Most designs end up with those 0.8 ratio, but this is not an exact rule. While this chapter is not meant to be a spring design handbook, Fig. 8.3.13 will give you the criteria for checking the horizontal stiffness as compared to the vertical.

An allowance of 50 percent additional capacity beyond rated load is also good practice. This means a spring rated for 2-in (50-mm) deflection would not have the coils touching (solid) before 3 in (75 mm). If it is rated for 1-in (25-mm) deflection, it should not go solid before 1½ in (37.5 mm). Overtravel allowance is needed as it is impossible to calculate exact weight distribution. Published equipment weight is inaccurate, and center-of-gravity locations are often unavailable. A 50 percent overtravel will allow for an acceptable 20 percent overload.

Isolation springs should be designed such that when the coils are touching, the elastic limit has not been exceeded so that springs will return to full height. If the spring is designed this way, it will be stressed two-thirds the elastic limit under normal load. Springs last indefinitely as vibration amplitudes are very small, and the spring movement per coil is the total amplitude divided by the number of active coils. With so little movement the stress cycle is close to zero, and these applications approach static loadings.

A simple spring can be considered a vibration isolator and often built into mechanical equipment. This low-cost method is satisfactory when thousands of springs are used in a repetitive application. When springs are used in the field, however, minimum additions to the design are normally a neoprene friction pad on the bottom to eliminate the need for bolting and to act in series with the steel coils to help eliminate high-frequency noise transmission. Since loadings are not easily or exactly determined, there must be a means of leveling, and this is usually done by means of an adjustment bolt. The bolt is often used to attach the mountings to the equipment as well.

Air Springs. The last remaining commodity of major importance is the air spring. Air springs are made of neoprene with nylon tire cord reinforcement and shaped like vertical bellows. Ethylene-propylene (EPDM) is also used for this purpose and

TABLE 8.3.2 Comparison of Natural Frequencies at Given Deflections: Heavy-Density Fiberglass, AASHO Neoprene, Steel Springs

Heavy-density fiberglass					AASHO neoprene					Steel springs		
Pad thickness		Deflection		Dyna- mic freq., Hz	Pad thickness		15% Deflection		Dyna- mic freq., Hz	Deflection		Freq., Hz
in	mm	in	mm		in	mm	in	mm		in	mm	
0.5	13	0.045	1.1 to	22	0.3	7.6	0.045	1.1	18.0	0.045	1.1	14.8
		to	2.3		0.5	12.7	0.075	1.9	14.5	0.075	1.9	11.4
		0.090			0.6	15.2	0.090	2.3	13.1	0.090	2.3	10.4
1.0	25	0.09 to	2.3 to	16	1.0	25.4	0.150	3.8	10.5	0.150	3.8	8.1
		0.18	4.6		1.2	30.5	0.180	4.6	9.5	0.180	4.6	7.4

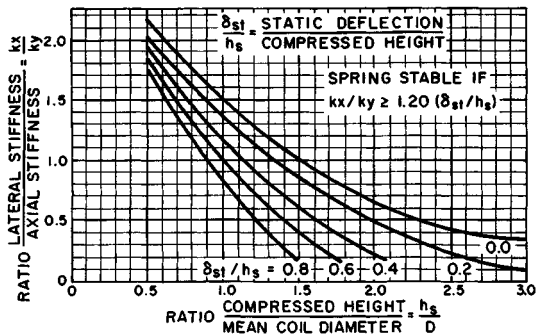


FIGURE 8.3.13 Horizontal versus vertical stiffness.

sometimes butyl as the inner liner. Butyl is the least permeable of the rubber materials and reduces air loss.

The rolling lobe is a design variation that provides for movement by the rubber walls literally rolling down a steel stanchion rather than flexing. Both designs are equally suitable, and it is merely a matter of selecting one over the other depending on what frequency is needed. In general, rolling lobes have lower natural frequencies as compared to single-convolution and double-convolution bellows of the same height.

All air springs leak. The leakage rate is very low, but it is generally impractical to set up supervisory replenishing procedures. All air spring systems should be installed with replenishing air lines connected to height-sensitive leveling valves. If an air spring or cluster of air springs loses air and the equipment settles, air will automatically be added. Where air springs might be installed on a hot roof, the leveling system would respond and bleed small quantities of air should the air expand.

Leveling valves also level equipment that goes out of level because of external forces when the equipment is running. For example, a top horizontal fan tends to rotate away from the point of discharge. It rears up on the discharge end and settles in the back. Leveling valves automatically compensate for this and return the installation to proper elevation.

Air springs have the advantages of low frequencies and low profiles. Since there is no steel continuity, there is no noise transmission. Air spring frequency varies very little with pressure, but since the capacity is directly proportional to increased pressure, air springs need not be selected as carefully as steel springs. Most devices will handle loadings at a minimum of 25 lb/in² (172 kPa) and as much as 100 lb/in² (689 Pa), which allows for a 4:1 ratio from minimum to maximum loading on a particular mounting.

Hangers. Hangers accommodate all the above devices in modified form so that they fit within steel frames which are usually open-sided. Very simple high-frequency neoprene hangers could be pad hangers, but more often rubber elements are designed for hangers by eliminating the base plates and the tapped holes on top. The elements are molded with a projecting bushing that passes through the hanger hole to prevent the rod from rubbing. Occasionally fiberglass pads are used in lieu of neoprene elements, but they generally have even higher frequencies. Steel springs are generally fitted into neoprene cups, and the cup itself has a no-rub

bushing arrangement. Rubber elements are often used in series with steel springs to combine the advantages of both materials.

It is very important to isolate and provide for flexibility in the piping so that the function of the floor mountings or equipment hangers connected to piped equipment is not interfered with or bypassed.

Connectors. Stainless steel connectors can be manufactured to specific lengths; they consist of a corrugated stainless steel body welded to the appropriate fittings. Even under moderate pressures, the closely spaced bellows become unstable and would spew out as the bellows expand. Therefore, virtually all metallic connectors have a stainless steel braid attached to the two ends to form a tube over the stainless steel corrugations. Stainless steel braid prevents elongation and adds to the radial strength.

Metallic connections are designed to allow flexing and reduce fatigue. The connector end away from the equipment should be rigidly secured so that the connector is forced to flex. This minimizes piping vibration after the anchor. Unfortunately, when any such connector is pressurized, there is a tremendous pull on the braid which makes the assembly extremely stiff. As a practical matter, it is very difficult to secure the afterend rigidly, so that in the average installation the flexible connector compensates for misalignment but does very little to reduce noise and vibration.

The next class of flexible connectors are Teflon® bellows which are an improvement on the metallic connectors because the Teflon® introduces a discontinuity in the metallic pipe wall. While Teflon® is an excellent material, temperature and pressure ratings are often too low for high-rise structures. Teflon® bellows are manufactured with built-in control rods to prevent excessive elongation. Control rods severely limit vibration and noise reduction as they bypass the flexible bellows.

Hand-built, single-arch rubber expansion joints are still being manufactured and are similar in function to Teflon®. The arch in the center is all that provides the flexibility, and because of the bellowslike shape these connectors also elongate unless control rods are used or the piping is anchored. Here again the control rods tend to bypass the action of the expansion joint. The stiff walls leading up to the arch have little function other than to provide room for the steel retaining rings and the bolt heads that go between the steel rings and the arch. Applications are generally reserved to industry as they can be built up to 144 in (3.7 m) in diameter and manufactured in exotic rubber materials, particularly for high temperatures and highly corrosive chemicals.

The most recent entrant is the spherical neoprene or ethylene-propylene (EPDM) expansion joint. Unlike the three commodities described above, spherical connectors are designed on the principle of the automobile tire. The reinforcement fabric is nylon tire cord. The tire cord forms a suspension bridge from one flange to the other. When these connectors are pressurized, the nylon stretches until the stretching force equals the pressure and then the connector remains stable at that length and diameter. In most cases they can be installed without control rods.

The volumetric expansion and contraction of the connector dissipates sound energy. They do an excellent job in reducing sound transmission at blade passage frequency (number of pump buckets times shaft r/min). Unfortunately, they are too stiff to handle the primary vibration at r/min , so it is still necessary to protect the structure by isolating the pipelines with isolation hangers. The connectors take care of misalignment and virtually eliminate the high-pitched whine that normally travels through a structure. Connectors are recommended in a double-sphere configuration, for equal ends or concentric reduction. Long-radius tapered elbows save space at pump connections.

Bases. When equipment is manufactured or supplied in multiple components not connected by a common base, a base must be used to connect the two elements before the vibration isolators can be applied under the entirety. A belt pulling on a flexibly mounted motor connected to a flexibly mounted blower would pull one or the other off the isolators. Torque causes similar problems when machinery is directly connected. Bases can be constructed of structural steel, or, where additional mass is required or advisable, of steel frames filled with concrete. Most air-conditioning equipment is so well-balanced that there is no base weight criteria and rigidity is the only concern. Descriptions of these bases are covered in the selection section.

8.3.4 SELECTION

Following is a complete guide specification written in engineering terms. The specification selection guide provides the proper prescription for the complete isolation of a unit in terms of the type of mounting or hanger, the recommended deflection, the need for a base (if there is one), and the recommendation for a specific flexible connector.

We suggest you include all these specification paragraphs in your standard engineering specifications. In addition, prepare a drawing of standard details that becomes part of the mechanical drawing set. This eliminates constant editing on each job since the materials are not actually used unless an extra column is added to your equipment schedule, Table 8.3.3. It is this callout that defines the isolation that is to be used under each piece of equipment. Table 8.3.3 refers to the appropriate specification paragraph by letter with the notation as to the proper static deflection. The recommendations come right from the selection guide (Table 8.3.6), so the table containing them can be prepared very quickly.

The selections were based on a 30-ft (9.1-m) floor span in the penthouse (9.1 m) and a 20-foot (6.1-m) span in other locations. Note that for pump no. 5 no isolation is called out as it is located in the basement under the garage where no one could be annoyed by the vibration. Fire pumps (no. 6) are seldom isolated.

In preparing a schedule similar to Table 8.3.3 there is an opportunity to consider every piece of equipment and very little possibility of skipping over some item in the rush of completing the project.

SPECIFICATION A

Double-deflection neoprene mountings (Fig. 8.3.14) shall have a minimum static deflection of 0.35 in (8.9 mm). All metal surfaces shall be neoprene-covered to avoid corrosion and have friction pads both top and bottom so they need not be bolted to the floor. Bolt holes shall be provided for these areas where bolting is required. Steel rails (Fig. 8.3.15) shall be used above the mountings to compensate for the overhang on small vent sets close-coupled pumps, etc.

SPECIFICATION B

Spring-type isolators (Fig. 8.3.16) shall be freestanding and laterally stable without any housing and complete with 1/4-in (6.4-mm) neoprene acoustical friction pads between the baseplate and the support. All mountings shall have leveling bolts that must be rigidly bolted to the equipment. Spring diameters shall be no less than 0.8 of the compressed height of the spring at rated load. Springs shall have a minimum additional travel to solid equal to 50 percent of the rated deflection. Submittals shall include spring diameters, deflections, compressed spring height, and solid spring height.

CHAPTER 8.4

ENERGY CONSERVATION PRACTICE

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8.4.1 INTRODUCTION

Energy conservation means many things to the design engineer. For instance:

- At one end of the scale it is the design of a system for new or retrofit projects that will have the *lowest* energy consumption over the operating life of the facility while meeting the owner's or user's needs. This is energy conservation in its pure sense, where costs are secondary to energy savings.
- At the other end of the scale it is the design of a system for new or retrofit* projects that will *minimize* energy consumption at *lowest* first cost of the project while meeting the owner's or user's needs. This is not pure energy conservation, since energy savings are secondary to costs. The prime consideration here is minimum *initial* cost; energy and maintenance cost are not included in the cost evaluation.
- Between these two extremes lies the area of design which offers the greatest challenge to the design engineer with respect to energy conservation. That is, to design the most efficient (minimized-energy-consumption) system for new or retrofit projects having the lowest life-cycle costs over the operating life of the facility and while meeting the owner's or user's needs.

The last concept of energy conservation, evaluated on life-cycle costs (LCC), will be discussed in this chapter.

*Retrofitting an existing building or facility for energy conservation means adding insulation, weather-stripping, storm windows, or replacement windows with insulated glass, or undertaking any other kind of remodeling that contributes to the prevention of unwanted heat loss or gain.

8.4.2 GENERAL

In new or retrofit energy conservation building design, innovation should be encouraged. However, any innovation will fail, no matter how beneficial from an energy conservation point of view, if it cannot be easily integrated into conventional construction practices and conform to established owner-user preferences, financing methods, building codes, and standards.

Though the design engineer uses the same procedures and information whether designing for energy conservation or not, there is significantly greater care and effort necessary in energy-saving design. Special attention must be given to the following factors:

- Overall values of the coefficient of heat transfer U for walls, floors, roofs, and glass
- Maximum percent fenestration (glass) area
- Building orientation with respect to fenestration per exposure
- Hours of operation of each space and area on weekdays, Saturdays, Sundays, and holidays
- Zoning of heating, ventilating, and air-conditioning (HVAC) systems
- System efficiencies at full load and at partial loads
- Ability to control, reset, start, stop, and reduce loads
- Heat recovery and heat storage
- Use of nondepletable energy sources
- Lighting illumination and fixture efficiencies
- Electrical motor efficiencies

Whether it is a new or retrofit project, reduction in one or more of the following general categories is required to reduce the energy consumed:

- Hours of system operation
- Air-conditioning loads
- Heating loads
- Ventilation and/or exhaust loads
- Domestic hot-water loads
- Lighting loads
- Off-peak loads

In addition, demand limiting and improvements in system efficiency and heat recovery are required.

Demand limiting and shifting electric loads to off-peak periods generally do not reduce the total energy required for the facility. They do reduce peak electric load, and therefore the utility or cogeneration plant energy requirement.

Of all the above energy-reduction items, it is the hours of operation that will usually have the most significant impact on energy conservation. Put another way, the energy consumption of an inefficient mechanical, plumbing, electrical, or process system that is turned off when not needed will generally be less than that of the most efficient system that is unnecessarily left on.

8.4.3 DESIGN PARAMETERS

Of all energy conservation factors, the major one determining the annual energy consumption of a facility is how that facility is used. This is more important than the type or capacity of the HVAC systems, boilers, chillers, and processes and the amount of glass or insulation or lighting.

It is therefore essential, if not mandatory, for the design engineer to have a definitive work schedule for each activity to be performed in the facility before energy conservation options can be considered. This schedule is part of the project design program, a topic discussed in various books (see list given in Preface), and should include the following items for each space and area:

- A detailed description of the work being performed.
- The type of process equipment and heating and cooling.
- The number of working staff or personnel by shifts for weekdays, Saturdays, Sundays, and holidays.
- The percent of equipment operating in a given hour and the average percent of full capacity for all the equipment by shifts for weekdays, Saturdays, Sundays, and holidays. If this information is not available, then the percent of maximum capacity of each operating piece of equipment for each hour of each shift for weekdays, Saturdays, Sundays, and holidays will be required.

The project annual energy budget must be determined. This establishes the maximum annual energy in Btu/ft² (MJ/m²) expected to be consumed by the project.

The energy budget depends on the type of facility (such as office, hospital, institution, or warehouse). The owner or user usually establishes the energy budget. If it is not available, the engineer should establish a budget for submission to the owner or user for approval before starting the design.

It is the designer's responsibility to select and design a totally integrated system whose annual energy consumption will not exceed the project's energy budget. If the project is a new facility, the design engineer can initiate the energy conservation design. However, if the project is a retrofit, an energy audit of the existing facilities must be performed before the design engineer can start the energy conservation design.

8.4.3.1 Energy Audit

The purpose of the typical energy audit is threefold:

- To learn how much energy is being used annually and for what purpose.
- To identify areas of potential energy saving (heat or cooling reclamation) and areas of energy waste.
- To obtain data required to prepare plans and specifications to reduce, reclaim, or eliminate the waste identified in the audit.

It is general practice to set priorities for the recommendations of the energy audit, starting with the most cost-effective and progressing down to the least cost-effective options. Before proposing or making any modifications to a particular system, the designer should carefully study all possible effects on the total facility.

For instance, a reduction in energy usage for one or more subsystems may result in an increase in the total facility energy consumption.

A typical energy-audit scope of work can be prepared for residential, commercial, institutional, or industrial facilities by selecting applicable items from the following procedures:

Utility Consumption. Obtain annual and daily records of the quantities and cost of each type of energy:

- Oil (by grade)
- Gas (natural and propane)
- Coal (by type and grade)
- Electricity

If this information is available by function, system, and process (such as office, cafeteria, or manufacturing), it should be recorded as such.

Identify all equipment observed to be idling for extended periods of time. Determine which could be turned off when not needed.

Insulation. Identify areas of damaged or missing insulation on piping systems, ductwork, and equipment.

Is the insulation type and thickness in walls and roof and on piping, ductwork, and equipment in compliance with current energy conservation standards? If not, will it be cost-effective to replace with insulation of the appropriate thickness and type or to add new insulation over the existing insulation?

If there is indication that the building, piping, ductwork, and equipment insulation may be inadequate, an infrared energy survey of the facilities should be performed to identify the hot spots (areas of greatest energy loss).

Fenestration. Is the percent of glass area high (25 percent or more of the total wall area)? Is there large glass exposure to the west and north? Is the glazing single-pane?

If the facility is fully air-conditioned, especially with large western glass exposures, the cost-effectiveness of replacing single-pane glazing with tinted Thermo-pane®, retrofitting shading devices in the summer, and reducing the glass area should be evaluated.

Infiltration. Is caulking around windows and exterior door frames in good condition? All defective or questionable caulking should be removed and replaced.

Is there weatherstripping around windows and exterior doors? Is it in good condition? If it is defective, it should be removed and replaced. If missing, it should be installed.

Broken windows should be replaced.

Do all building personnel entrances that are used daily have vestibules with double doors? If not, is it cost-effective to provide them? Especially in areas that have long winters, it is good practice to provide vestibules on all frequently used doors.

Do loading docks have shrouds or air-curtain fans? If not, is it cost-effective to retrofit the doors with them?

Ventilation. Is outside air set at minimum volume?

Is it cost-effective from an energy standpoint to recirculate all but the minimum ft^3/min (m^3/s) of outside air (that required to replenish oxygen and dilute unfilterable gases, e.g., carbon monoxide and carbon dioxide) through a filtering system using high-efficiency particle filters (to remove the particulate matter) in series with gas sorbers (to remove the pollutant gases).

Typical sorbers contain gas absorption materials and oxidizers such as activated charcoal and aluminum impregnated with potassium permanganate, depending on the particular gases present or anticipated in the air stream.

Exhaust and Makeup. Identify all systems that exhaust moderate to large volumes of air and fumes to the atmosphere. Can these quantities be reduced? Will it be cost-effective to recover the thermal energy being exhausted?

Identify areas and systems where the actual makeup air is excessive or deficient when compared to the required makeup air requirements. Determine the most cost-effective way to correct the makeup air volumes to the design specifications for all excessive and deficient areas or systems.

Air Systems. Is the time interval between morning startup of air-handling units and the start of the workday as short as possible but long enough to develop an acceptable temperature for arriving employees?

Is the time interval between shutdown of the refrigeration or heating system (depending on the mode of operation) and the end of the workday as long as possible but short enough to maintain an acceptable temperature at the close of the workday? For a discussion on determining the optimum startup and shutdown time periods, see Sec. 8.4.3.7, "Automatic Temperature Controls."

Night Setback. Do the heating coil controls of the air-handling and heating and ventilating units have night setback controls that close outside air dampers and reset the thermostat downward when the facility is unoccupied?

Is the night setback temperature in the unoccupied area at least 10°F (5.5°C) lower than the nominal (occupied) space temperature? Maximum setback should maintain at least 40°F (4.5°C), however.

If any air-handling and heating and ventilating units do not have night setback controls, the cost-effectiveness of adding them should be evaluated.

If the present night setbacks are not set to maintain temperatures in the 40 to 55°F (4.5 to 12.8°C) range, the reason should be determined. If there is no valid reason, they should be adjusted to do so.

Cooldown Cycle (Cooling Mode). Do the air-handling units have a cooldown control cycle? Does that cycle close the outside air damper (assuming the building is unoccupied), de-energize the heating cycle, reset the cooling thermostat (to the occupied settings), and energize the cooling cycle?

If there are air-handling units that are normally operated 12 h or less a day without a cooldown cycle, the cost-effectiveness of adding a cooldown cycle should be evaluated.

Warmup Cycle (Heating Mode). Do the air-handling and heating and ventilating units have a warmup control cycle? Does the cycle close the outside air damper (assuming the building is unoccupied), reset the heating thermostat (to occupied setting), and de-energize the cooling and ventilating cycles?

If there are air-handling or heating and ventilating units that do not have a warmup cycle, the cost-effectiveness of adding them should be evaluated.

Low-Leakage Dampers. Do air-handling, heating and ventilating and makeup air units have low-leakage outside air dampers? For energy conservation, a low-leakage damper is one having a maximum leakage rate less than 1 percent of the full flow in ft^3/min (m^3/s).

If there are air-handling, heating and ventilating, and makeup air units that do not have low-leakage outside air dampers and whose outside air dampers are normally closed when the building or facility they serve is unoccupied, the cost-effectiveness of retrofitting them, with respect to the energy saved, should be evaluated.

Coils (Heating and Cooling). Are coil surfaces clean? Are there any blockages or restrictions to uniform air flow across the coil face area?

Is the water side of the coil clean?

Are there any plugged tubes or indication that a coil has been frozen and repaired? It is not uncommon for repairs to frozen coils to seriously reduce heat transferability (efficiency). If this is the case, the cost-effectiveness of replacing all repaired coils should be evaluated.

All coils with dirty air side and fouled water side heat-transfer surfaces should be cleaned.

All coils found with blockages or restrictions to uniform air flow should be evaluated to determine if it will be cost-effective to correct this situation at this time.

Preheat Coils. Does the air-handling system have preheat coils? Can any of them be shut off?

If reheat is required for few zones, can variable-air-volume boxes that bypass air to the return be retrofitted to replace the reheat coils?

Is there reclaimable waste heat that could be used as an energy source for the zones that must have reheat?

Is it possible to reduce the heating-medium temperature and still maintain leaving air temperatures?

All reheat coils that are not needed should be valved off. Those reheat coils where valve turnoff is questionable—and where there is no possibility of freezing—should be shut off and their zones monitored to determine whether they can be permanently valved off.

For those instances where there are a few zones requiring reheat, the cost-effectiveness of replacing the reheat coils with variable-air-volume boxes that bypass air to the return should be evaluated.

For those instances where there are a significant number of reheat points and there is a source of waste heat that can be recovered, the cost-effectiveness of retrofitting a waste-heat recovery system for the reheater should be evaluated.

Ductwork. For comments on duct and equipment insulation, see Insulation, above.

Is there indication that the ductwork is not tight? For low-velocity systems, the leakage rate should not be greater than $7\frac{1}{2}$ percent of the supply fan ft^3/min (m^3/s). For high-velocity (or medium-velocity) systems, the leakage rate should not be greater than 5 percent of the supply fan cfm (m^3/s).

Are there indications of restrictions or poorly installed ductwork? Can the supply and return fans' static pressure (total pressure for axial flow fans) be significantly reduced by modifying the ductwork?

If any of these conditions is found in the existing duct systems, the cost-effectiveness of modifying the duct systems to correct it should be studied.

Types of Systems. If the air-conditioning system is constant-volume with terminal reheat or dual ducts, the cost-effectiveness of retrofitting it to a variable-air-volume (VAV) system should be evaluated. If the air-conditioning system is multizone, it may be cost-effective to retrofit it to a VAV system.

When it is not cost-effective to retrofit a dual-duct system to a VAV system, the hot-deck automatic control should remain closed during the cooling mode. Under these conditions the hot-deck temperature will be adequate for the commercial reheat requirements, even though it will be equal only to the mixed-air temperature plus temperature increase caused by the heat added by the supply fan (which is minor).

Where reheating cannot be eliminated, are the leaving air temperatures of the coils as low as possible, yet high enough to maintain space conditions?

Can the speeds of the air-handling system supply and return fans speed be reduced by replacing the drive pulleys and belts and rebalancing the system? If the answer is "yes" or "maybe," then a study should be made to determine if the changes will be cost-effective.

Do the air-conditioning systems that serve areas that must maintain design temperatures and relative humidity 365 days a year (computer facilities, constant-temperature rooms, calibration laboratories, etc.) have some means to utilize the cooler ambient temperatures during the spring, fall, and winter months to reduce the annual energy costs?

If not, will it be cost-effective to retrofit the existing systems to have a water-cooled condenser with dry coolers as described under Condenser Water/Precooling Recovery in Sec. 8.4.3.6?

Liquid Refrigeration Chillers. Is the chilled-water supply temperature set at the highest temperature possible but low enough to maintain space temperatures under maximum load conditions? If not, it should be reset.

Are the automotive controls capable of resetting the chilled-water supply temperature higher as the cooling load decreases?

Is the refrigerant compressor operating at the highest suction pressure and the lowest head pressure possible, yet able to maintain the required chilled-water supply temperature under maximum load conditions? If not, this should be corrected.

Are the automatic controls for the cooling tower capable of resetting (lowering) the condenser water supply temperature as the ambient wet-bulb temperature drops?

Are the evaporator and condenser tube surfaces clean, maximizing heat-transfer efficiencies? If not, they should be cleaned.

If the present automatic control cannot reset the chilled-water supply temperature higher as the cooling load decreases, or reset (lower) the condenser water supply temperature from the cooling tower as the wet-bulb temperature drops, determine the cost-effectiveness of modifying the controls to provide these capabilities.

For facilities that have a year-round cooling requirement that cannot be met by using 100 percent outside air (economizer cycle) and have a chilled-water system with water-cooled condenser (with cooling tower or spray pound), evaluate the cost-effectiveness of the following:

- Reclaiming and reusing the heat of the condenser by providing a double bundle condenser
- Installing a plate exchanger piped into the condenser and chilled-water system. Depending on the ambient wet-bulb temperature, the plastic exchanger will provide the chilled-water supply temperature and still maintain separate chilled and condenser water piping systems.

Refrigerant Compressors for DX Air-Handling Units. Is the suction pressure set at the highest temperature possible, yet able to maintain space temperatures under maximum load conditions? If not, it should be.

Is the refrigerant compressor operating at the lowest head pressure possible, yet able to maintain the required suction pressure? If not, it should be.

If the condenser is an air-cooled, is the automatic control for the condenser fans capable of maintaining the lowest head pressure recommended by the compressor manufacturer while maintaining the required (compressor) suction pressure?

If the condenser is water-cooled, can the automatic control for the cooling tower reset (lower) the condenser water supply temperature as the ambient wet-bulb temperature drops, corresponding to the lowest head pressure recommended by the compressor manufacturer, and still maintain the required (compressor) suction pressure?

If the condenser is water-cooled, can the automatic control for the cooling tower reset (lower) the condenser water supply temperature as the ambient wet-bulb temperature drops, corresponding to the lowest head pressure recommended by the compressor manufacturer, and still maintain the required (compressor) suction pressure?

If the condenser is an evaporative condenser, can the automatic controls for the spray pump and condenser fan lower the condensing temperature (as the ambient wet-bulb and dry-bulb temperatures decrease) to the lowest head pressure recommended by the compressor manufacturer and yet maintain the required (compressor) suction pressure?

If the automatic controls do not maintain the lowest recommended condensing temperature, the cost-effectiveness of modifying the condenser controls should be determined.

Cooling Towers. Will it be cost-effective to reduce the blowdown (makeup water) requirements by changing or modifying water treatment?

Can the makeup water required because of drift loss be reduced by modifying or adding drift eliminators on the existing towers? Will the change be cost-effective?

Do the towers have two-speed fan motors? If not, is the energy saved enough to justify the cost of modifying the fan motors and tower controls?

Can the tower fan volume be reduced and still supply condenser water at the required temperature under design load conditions? If not, will it be cost-effective to provide this feature?

Is the automatic temperature control for the cooling tower capable of resetting (lowering) the condenser water supply temperature as the ambient wet-bulb temperature drops? If not, will it be cost-effective to retrofit the control?

Boilers. Are tubes and breeching clean?

Is the flue gas continually analyzed and the air/fuel ratio adjusted for maximum combustion efficiency, corresponding to swings in heating load?

Is heat recovered from the flue gas to preheat the combustion air or for some other preheat service?

Is the stack gas temperature as low as possible, i.e., approximately 50°F (10°C) above the lowest combustion gas dew point?

Is the breeching installed properly? Is breeching the correct size for the maximum firing rate? Is breeching pitched up toward the stack or chimney connection without restrictions?

Are the stack diameter and height adequate for the maximum firing rate of the connected boilers?

Are the burner flame shape and capacity correct for the dimensions of the combustion chamber? Is the burner type the most efficient for the boiler? Is the burner the correct size (neither undersized nor grossly oversized) for the boiler and load?

Will it be cost-effective from an energy standpoint to modify any or all of the above items?

Waste Heat and Heat Recovery. Identify areas and systems where heat can be reclaimed or recovered.

Is there a requirement for chilled water or process cooling water during the heating season? If there is, will it be cost-effective to preheat the ventilation or makeup air (outside air) and precool the chilled water or process cooling water as it is returned to the chiller by retrofitting a water-to-water and water-to-air heat-recovery system? See Fig. 8.4.2 and the related discussion in Sec. 8.4.3.6.

If the electric transformers are located indoors, will it be cost-effective to reclaim the heat generated by them?

If there are large computer rooms that operate 24 h a day or throughout the night, do they have the ability to utilize the lower-temperature ambient air to reduce the refrigeration energy demand? If not, the cost-effectiveness of retrofitting them to provide this capability should be evaluated.

In areas where ceiling height is greater than 12 ft (3.5 m), is there temperature stratification near the ceiling with a temperature difference greater than 10°F (5.5°C) during the heating season? If so, the cost-effectiveness of reclaiming this wasted heat should be evaluated. Two types of heat recovery systems are discussed under Heat Recovery by Recirculating Warm Stratified Air in Sec. 8.4.3.6.

Hydronic Systems. Identify leaks in condenser water, chilled water, hot water, process water, etc.

Are three-way valves used to automatically control the heating and cooling coil capacities? If two-way valves are used to automatically control the heating and cooling coil capacities, are variable-speed pumps used? If three-way or two-way automatic coil control valves with constant-speed pumps are used, will it be cost-effective to retrofit the system to one using variable-speed pumps with two-way automatic control valves?

Is the water treatment optimum to provide maximum heat-transfer efficiency within the boilers, coils, and heat exchangers and minimize corrosion and fouling of the water distribution system? Refer to Chap. 8.5, "Water Conditioning," for a discussion on water treatment. If the water treatment is not optimum, the cost-effectiveness of providing one that is should be evaluated.

Is the hot-water supply temperature to the fin pipe radiators automatically reset on the basis of ambient temperature?

For comments on piping and equipment insulation see Insulation, above in this section.

All leaks in the valves, equipment, and piping system should be repaired.

If the facility has a three-pipe (independent hot-water and chilled-water supply pipes and a common return pipe) distribution system, is it feasible to retrofit a two-pipe or four-pipe distribution system? If so, which is more cost-effective?

Steam Systems. Identify leaks in steam and condensation piping systems. This is especially critical for vacuum steam heating systems. Identify malfunctioning and leaking steam traps. All leaks in the piping system, valves, equipment, and malfunctioning steam traps should be repaired.

Is any condensate wasted that is suitable to be returned to the boiler?—i.e., uncontaminated? Would it be cost-effective to return it?

If high-pressure steam [at least 125 lb/in² (8.5 bar)] is available, will it be cost-effective to use steam-driven turbine pumps and fans, since turbines can operate as a pressure-reducing valve to supply the low-pressure [under 15 lb/in² (1 bar)] needs?

Is the boiler feedwater treatment optimum to provide maximum heat-transfer efficiency within the boilers, coils, and heat exchangers and minimize corrosion and fouling of the steam and condensation piping distribution systems? If the water treatment is not optimum, the cost-effectiveness of optimizing it should be evaluated.

For comments on piping an equipment insulation, see Insulation, above in this section.

Self-contained automatic radiator control valves should be retrofitted on all steam radiators and fin pipe convectors that do not already have them.

Process Equipment. Is there cost justification for:

- Replacing old equipment with new equipment requiring less energy?
- Replacing an obsolete inefficient process and equipment with a modern process using less energy?

For batch-type processes, is it cost-effective to shut off equipment between batches?

Is the equipment startup period (the time it takes for the process to reach operating conditions) as short as possible? If the startup period is long, can the equipment be modified to shorten it? Will the modification be cost-effective?

Automatic Space Controls. Were the controls calibrated recently? If they have not been calibrated within the past 5 years, they should be recalibrated.

Are the space air-conditioning thermostats set for 78°F (25.5°C) dry-bulb temperature for comfort cooling and at the highest temperature at which the process and/or equipment can operate?

Are the space-heating thermostats set for 68°F (20°C) dry-bulb temperature for comfort heating and at the lowest temperature at which the process and/or equipment can operate?

Do thermostats reset at night or when the space is unoccupied? Can thermostats be reset by unauthorized personnel?

Are the air-handling units that have the economy cycle (provision to use 100 percent outside air for cooling) provided with enthalpy control?

Are the radiators controlled via hand valves?

Will it be cost-effective from an energy standpoint to modify any or all of the above items?

Does the facility have an energy management system? If it does, is it functionally satisfactory? If it does not, will it be cost-effective to install one?

Are the perimeter radiation hot-water supply temperature set points as low as possible for ambient air temperatures but high enough to maintain space conditions? Is the hot-water supply temperature to the heating coils as low as possible and yet able to maintain space and/or leaving air conditions? Are the controls set to prevent, or at least minimize, the effect of the perimeter system bucking the interior system in the cooling or heating mode? If the controls are not so set, they should be adjusted or modified so they will not waste energy.

Solar. Is the site's geographical location favorable for the application of solar collectors? If it is, will it be cost-effective to heat the domestic hot water or to preheat the process water?

Domestic Hot Water. Are flow restrictors installed at lavatory, bathtub, and shower fixtures?

What temperature is the hot-water supply set at? If the system supplies predominantly toilets and showers and the hot-water supply temperature is above 110°F (4.3°C), determine if it is cost-effective to install booster heaters locally at the equipment or fixtures that require higher temperatures and reduce the supply hot water to the 105 to 110°F (40.5 to 43°C) range.

Determine if the domestic hot-water heater is oversized? If so, is it cost-effective to reduce its capacity to match the connected load?

Does the domestic hot-water system have recirculating pumps? Do they run continuously? If so, evaluate the cost-effectiveness of shutting off the pumps after normal working hours and, if needed, installing supplementary domestic hot-water heaters for the toilets that are used during those hours.

Identify and fix all leaking fixtures, valves, and fittings.

Identify areas of damaged insulation, or those lacking insulation. Evaluate the cost-effectiveness of replacing and providing insulation where appropriate.

Is the geographical location favorable for the application of solar collectors? If it is, will it be cost-effective to install solar systems to preheat or heat the domestic hot water?

Compressed-Air Systems. Identify all leaks in compressed-air piping, valves, and fittings.

Determine if compressed-air supply pressure can be lowered. If so, the pressure control should be reset.

For central systems, determine if the compressed-air supply pressure was set for equipment in one or two areas where the required volume of compressed air is a small percentage of the plant's (volume) capacity. If so, determine if it is cost-effective to lower the supply pressure of the central system and install local air compressors in areas having equipment requiring higher pressures.

Is rejected heat from intercoolers, aftercoolers, and ventilation air reclaimed? If not, is it cost-effective to reclaim and use it?

Is the intake air to the compressor intake filter unrestricted at the pressure and quantity specified by the compressor manufacturer? If not, evaluate the cost-effectiveness of modifying the intake system to comply with the manufacturer's requirements.

Is the intake air to the compressor clean and at the lowest temperature possible? If not, will the increase in efficiency (reduction in energy consumption) produced by modifying the intake system justify the cost?

Lighting and Power. Identify areas with excessive illumination levels and areas where illumination levels can be reduced if task lighting is provided. Identify areas where lights are left on when not needed. Are there enough switches to permit leaving the lights on only in areas where persons are working (after the normal working day, etc.) and shutting off all other fixtures except for security requirements? Are the light fixtures wired to permit reducing the general illumination level by switching off alternate fixtures and reducing the number of active tubes in fluorescent fixtures? Are all fluorescent fixtures of the energy-efficient type with energy-saving ballasts?

Is exterior lighting (building, parking lot, advertising, etc.) controlled by timers or photocells?

Is the present lighting fixture maintenance program adequate to maintain maximum illuminating output? If not, determine if it will be cost-effective to increase or revise the maintenance program.

Are high-efficiency electric motors used? Are the electric motors oversized? Oversized motors operate at a lower power factor.

Determine the overall power factor for the installation. If the power factor is low (according to the electric utility standard), will it be cost-effective to provide power factor correction equipment?

From the electric utility billing criteria and the facility's hourly electric load profile, determine if it will be cost-effective to install demand-limiting equipment.

Determine the cost-effectiveness of the following:

- Reducing the illumination levels by adding task lighting where necessary.
- Retrofitting additional switches to permit shutting off lights in areas and rooms not used.
- Retrofitting the fixture circuits to permit switching off alternate fixtures and tubes.
- Relamping the facility with the most energy-efficient fixtures and bulbs for the type of work being performed in each area.
- Replacing the electric motors with those having the highest efficiency and power factor available.

8.4.3.2 Design

General. Though the energy required for a process normally does not vary with the seasons of the year, the energy consumed by HVAC systems does. On an annual basis, most of the energy use for building HVAC systems occurs when ambient temperatures are moderate and the systems are operating at part load. Only a small fraction of the annual hours of operation of HVAC equipment occurs when ambient summer and winter temperatures are at or near their respective design values. The designer should (from an energy consumption standpoint) be more concerned about minimizing energy consumption at various part-load conditions throughout the year than at the design heating and cooling loads.

The designer must consider carefully energy consumption of equipment that operates most of the time at or close to full load. Typically lights, fans, and pumps, before the energy crisis in the 1970s, were operated constantly and at full load. In many air-conditioned offices and institutional buildings, under such conditions, the HVAC fans and pumps on an annual basis use more energy than the central air-conditioning chillers.

Energy can be saved if the designer carefully considers the following:

1. Operating HVAC systems at part load, especially fans and pumps.
2. Selecting variable-capacity fans and pumps, capable of varying their capacities to meet their respective part-load requirements (this is usually required of systems employing variable-speed drives and/or multiple units).
3. Using high-efficiency motors.
4. Designing HVAC systems that can isolate areas having relatively constant occupancy during the normal working day from those having only part-time occupancy (such as conference rooms, auditoriums, etc.), spaces that are used 24 h a day (such as computer rooms, constant-temperature rooms, and calibration labs), and areas that are used after the normal working day and on Saturdays, Sundays, and holidays. The systems should be designed and zoned so that only the areas occupied or requiring constant exhaust, temperature, or humidity will be operating. All other cooling and exhaust systems will be off, and heating system temperatures should be reset as low as possible.
5. The lighting system should be designed to provide the minimum acceptable level of general illumination and task lighting for the working area. High-efficiency lamps and low-energy ballasts should be installed, and available daylight should be used whenever possible. Lighting circuitry should be designed to permit turning off lights in unoccupied areas and reducing lighting level for off-hours housecleaning.
6. Domestic hot-water temperature should be set as low as possible. Local generation of domestic hot water to eliminate long runs of recirculating piping should be evaluated. Water conservation fixtures should be used.
7. The design pressure in plant compressed air systems should be as low as possible.

Components. To assist the designer in selecting the proper components for an integrated energy-efficient design that will minimize energy usage and meet the project's energy budget, the following guidance is offered. This list of components must not be considered all-inclusive. Innovations and additions should be encouraged.

Utilities. When determining the most appropriate fuel or fuels to be used, the following should be considered:

- Present and long-term availability and costs of oil, coal, gas, and electricity available at the project site.
- The various grades of oil and coal available.
- For coal, the costs of unloading, storing, handling coal; controlling air pollution (particulate matter); and ash handling and disposal must be considered.
- In locations where natural gas is available, the cost-effectiveness of using dual fuels—especially oil and gas—should not be overlooked.

Alternative Energy Sources. To reduce dependence on oil and electricity (generated by burning oil), alternative energy sources such as coal; methane gas from wells, landfill, and sewage treatment plants; wood; hydropower; sun; wind; and tidal motion (to name the most common) should be considered. Although instal-

lations using one or more of these alternative energy sources have been successful, specific environmental, meteorologic, and site-related conditions must be favorable. When site conditions are favorable, alternative energy sources should be compared with oil, gas, coal, and electricity to determine those most cost-effective.

Transmission Values. When the maximum transmission values U are not determined by the user or owner, they should be selected by the engineer to minimize energy consumption. The author has used the values in Table 8.4.1 as the basis for his designs and as the base U values for calculating the cost-effectiveness of using lower U values in combination with additional insulation and triple-pane glazing.

The T values in Table 8.4.1 are selected on the basis of heating degree-days. However, the author suggests that the design U values, for projects where the air-conditioning load is predominant, should be based on the lower of two values, one based on the actual heating degree-days and the other based on one of the following conditions:

- When the summer air-conditioning design ambient temperature is above 95°F (35°C) 2½ percent of the time and the cooling season is at least 4 months long, the U values corresponding to 3001 to 4000 (1671 to 2220) heating degree-days should be used. (Values in parentheses are Celsius degree-days; others are Fahrenheit degree-days.)
- When the summer air-conditioning design ambient temperature is between 90 and 95°F (32 and 35°C) 2½ percent of the time and the cooling season is at least 3 months, the U values corresponding to 2001 to 3000 (1111 to 1670) heating degree-days should be used.
- When the summer air-conditioning design ambient temperature is below 90°F (32°C) 2½ percent of the time and the cooling season is at least 4 months, the U values corresponding to less than 1000 (560) heating degree-days should be used.
- For all other conditions, the U values should be selected on the basis of the actual heating degree-days.

If there is any question on the selection of a particular value, the decision should be based on a life-cycle cost analysis.

Fenestration. Traditionally the architect is the one who determines the glass area of a building. However, in order to design a facility that will meet the established energy budget, it is the engineer who must determine the maximum percentage of glass area that can be permitted in conjunction with the wall construction that will not exceed the overall design U_o value (see preceding discussion under Transmission Values).

The overall U_o value is determined by the following equation:

$$U_o = \frac{U_w A_w + U_g A_g + U_d A_d + \dots}{A_o} \quad (8.4.1)$$

where U_o = average or combined transmission of the gross exterior wall, floor, or roof-ceiling assembly area, Btu/(h · ft² · °F) [W/(m² · K)]

A_o = gross exterior wall, floor, or roof-ceiling assembly area, ft² (m²)

U_w = thermal transmission of the components of the opaque wall, floor, or roof-ceiling assembly area, Btu/(h · ft² · °F) [W/(m² · K)]

A_w = opaque wall, floor, or roof-ceiling assembly area, ft² (m²)

TABLE 8.4.1 Maximum Heat Transmission Values*

Heating degree-days, °F days (°C days)	Gross wall†		Walls Ceiling/roof		Floor	
	U_o (note 1)	U_o (note 2)	U_w (note 3)	U_r (note 4)	U_f (note 5)	U_f (note 6)
Less than 1000 (less than 560)	0.31 (1.760)	0.38 (2.15)	0.15 (0.853)	0.05 (0.284)	0.10 (0.568)	0.29 (1.647)
1000–2000 (561–1110)	0.23 (1.306)	0.38 (2.15)	0.15 (0.853)	0.05 (0.284)	0.08 (0.454)	0.24 (1.363)
2001–3000 (1111–1670)	0.18 (1.022)	0.36 (2.048)	0.10 (0.568)	0.04 (0.227)	0.07 (0.397)	0.21 (1.192)
3001–4000 (1671–2220)	0.16 (0.909)	0.36 (2.048)	0.10 (0.568)	0.03 (0.170)	0.07 (0.397)	0.18 (1.022)
4001–6000 (2221–3330)	0.13 (0.738)	0.31 (1.760)	0.08 (0.454)	0.03 (0.170)	0.05 (0.284)	0.14 (0.794)
6001–8000 (3331–4440)	0.12 (0.683)	0.28 (1.590)	0.07 (0.397)	0.03 (0.170)	0.05 (0.284)	0.12 (0.683)
Over 8001 (over 4441)	0.10 (0.568)	0.28 (1.590)	0.07 (0.397)	0.03 (0.170)	0.05 (0.284)	0.10 (0.568)

*Heat transmission values are expressed in English units, Btu/(ft² · h · °F), and, in parentheses, in SI units, W/(m² · K).

†Gross wall values include all doors and windows, window frames, metal ties through walls, structural steel members that protrude through all insulation to the exterior or adjacent to the exterior and continuous concrete or masonry walls or floors that extend from inside heated spaces through the building envelope to the exterior, e.g., fire walls that extend above the roof and concrete floor slabs that extend beyond the exterior wall to form a balcony or terrace.

Note 1: These gross wall U_o values are used for all new construction and major alteration of facilities other than hospitals and medical and dental clinics.

Note 2: These gross wall U_o values are to be used for hospitals and medical and dental clinics. The maximum U_o value will put a limitation on the allowable percentage of glass area to gross wall area in a building. Insulating glass will allow higher percentage of glass area than single glass.

Note 3: Wall U_w value is the thermal transmittance of all elements of the opaque wall area. U_w values are to be used for upgrading existing facilities where the alteration of walls and resizing of window glazing to meet gross wall values is not cost-effective.

Note 4: Ceiling/roof U_r values are for ceiling and roof areas where adequate space exists for insulation to be applied above the ceiling or below the roof structure. Built-up roof assemblies and ceiling assemblies in which the finished interior surface is essentially the underside of the roof deck shall have a maximum U_r value of 0.05 (0.284) for any heating degree-day area.

On existing buildings, use the maximum U_r value practical to accommodate the existing roof conditions where the life-cycle cost analysis indicates a higher life-cycle cost to implement U_r values required by Table 8.4.1. Examples of costs encountered on existing buildings related to implementing U_r values required by Table 8.4.1 are as follows: (a) cost of providing structural support to accommodate additional dead loads of new insulation and roofing system, and additional live loads from greater accumulations of snow (snow will melt more slowly because of increased insulation); (b) cost of raising roof curbs; (c) cost of raising cap flashings; (d) cost of raising roof drains.

Note 5: Floor U_f values are for floors of heated space over unheated areas such as garages, crawl spaces, and basements without a positive heat-supply to maintain a minimum temperature of 50°F (10°C).

Note 5: Floor U_f values are for slab-on-grade insulation around the perimeter of the floor.

Source: Department of Defense Construction Criteria, document DOD 4270.1-M, Office of the Deputy Assistant Secretary of Defense (Installations), Washington, DC, 15 Dec. 1983, chap. 8, table 8-1, p. 8-8.

- U_g = thermal transmission of the glazing (window or skylight area), Btu/(h · ft² · °F) [W/(m² · K)]
 A_g = glazing area (finished opening), ft² (m²)
 U_d = thermal transmission of the door or similar opening, Btu/(h · ft² · °F) [W/(m² · K)]
 A_d = door area (finished opening), ft² (m²)

From Eq. (8.4.1) it can be seen that the percentage of glass can be maximized without increasing the design U_o by selecting the lowest economical U_w (by changing the wall construction or adding insulation) and using triple Thermopane® glazing.

Although maximizing the percentage of glazing can have aesthetic, daylighting, and passive solar heating benefits, it generally increases wall construction costs.

Insulation. In residential facilities, most of the energy is used for environmental control. In such facilities the thermal (insulation) quality of the buildings and the severity of the weather become a predominant influence on energy consumption. Other major factors are how the systems perform with respect to space temperatures and hours of operation. In facilities such as these, the insulation thickness has a direct effect on reducing the amount of energy consumed. The more insulation, the less energy required to maintain space conditions.

In nonresidential facilities, energy usage is more complex. It is influenced by the function of the particular building, type and sophistication of control systems, type of fan and pump operation (constant speed or variable speed), hours of operation, ventilating rate, and thermal (insulation) quality of the building. Buildings such as these are relatively insensitive to energy savings resulting from insulation thickness alone. The primary reason for this is that, during the cooling season, most of the air-conditioning energy is used to offset heat gains from people, lights, and equipment, which are the same for facilities in Fairbanks, Alaska, or Miami, Florida. Another reason is that energy loss through exterior areas (building skin and roof) is a small percentage of the heating and cooling load; this is especially true in high-rise office buildings and institutions.

Selecting the optimum insulation thickness and type is important, since it can improve system efficiencies and reduce the amount of energy needed to maintain the same environmental condition or process load—or increase the energy available to maintain environmental conditions or process load.

The optimum insulation thickness is the thickness which will result in the lowest total of the cost of energy lost and the cost of insulation and installation. The method and procedure to calculate the optimum insulation thickness can be found in standard design handbook sources such as Ref. 1.

If the analytical method is not used to determine the optimum insulation thickness, the author recommends the following thickness guidelines. At the very least, they can be used as a basis for comparison of insulation thicknesses and types.

1. Duct insulation—outside air, supply, and return ductwork; plenums and casing of HVAC units
 - a. Indoors
 - (1) Blanket-type flexible fibrous-glass insulation, minimum density 1 lb/ft³ (16 kg/m³), minimum thickness 2 in (50.8 mm)
 - (2) Rigid-type fibrous-glass insulation, minimum density 3 lb/ft³ (48 kg/m³), minimum thickness 2 in (50.8 mm)
 - b. Outdoors—polyurethane or polyisocyanate board, minimum density 1.7 lb/ft³ (27.2 kg/m³), minimum thickness 3 in (76.2 mm)

2. Equipment

a. Pumps, chilled, dual-temperature, and hot water

- (1) Cellular glass insulation, minimum thickness 2 in (50.8 mm)
- (2) Fibrous-glass insulation, minimum density 6 lb/ft³ (96.1 kg/m³), minimum thickness 2 in (50.8 mm)
- (3) Polyurethane or polyisocyanate, minimum density 1.7 lb/ft³ (27.2 kg/m³), minimum thickness 2 in (50.8 mm)

b. Expansion tanks, condensate receivers, hot-water storage tanks, and converters

- (1) Cellular glass, minimum thickness 4 in (101.6 mm)
- (2) Fibrous-glass insulation, minimum density 6 lb/ft³ (96.1 kg/m³), minimum thickness 4 in (101.6 mm)
- (3) Calcium silicate, minimum thickness 4 in (101.6 mm)
- (4) Polyurethane or polyisocyanate, minimum thickness 2 in (50.8 mm)

c. Chillers

- (1) Polyurethane or polyisocyanate, minimum thickness 2 in (50.8 mm)
- (2) Plastic foam, minimum thickness 2 in (50.8 mm)

d. Piping systems—chilled water, dual temperature, hot-water heating, and domestic hot water

- (1) Fibrous glass, minimum density 3 lb/ft³ (48 kg/m³)
- (2) Pipes less than 3 in (76.2 mm) in diameter, minimum thickness 1 in (25.4 mm)
- (3) Pipes 3 in (76.2 mm) and 4 in (101.6 mm) in diameter, minimum thickness 1½ in (38.1 mm)
- (4) Pipes 5 in (127 mm) and larger, minimum thickness 2 in (50.8 mm)
- (5) Pipes less than 3 in (76.2 mm) in diameter, minimum thickness ¾ in (19 mm)
- (6) Pipes 3 in (76.2 mm) and 4 in (101.6 mm) in diameter, minimum thickness 1 in (25.4 mm)
- (7) Pipes 5 in (127 mm) and larger, minimum thickness 1½ in (38.1 mm)

e. Steam, condensate, and boiler-feed piping

- (1) Fibrous glass, minimum density 3 lb/ft³ (48 kg/m³); minimum thickness of pipe insulation is listed in Table 8.4.2
- (2) Calcium silicate, minimum thickness ½ inch (12.7 mm) greater than those in Table 8.4.2

TABLE 8.4.2 Minimum Thickness of Fibrous Glass Pipe Insulation
(Not exposed to weather)

Maximum temperature, °F (°C)	Nominal pipe sizes, in (mm)				
	Up to 1.25 (31.75)	1.5–2.5 (38.1–68.5)	3–4 (76.2–101.6)	5–6 (127–152.4)	8 (203.2) and larger
Up to 299 (148.3)	1 (25.4)	1.5 (38.1)	2 (50.8)	2.5 (63.5)	3 (76.2)
300–499 (148.9–259.4)	1.5 (38.1)	2.5 (63.5)	3 (76.2)	3.5 (88.9)	4 (101.6)
Condensate and boiler feedwater	1 (25.4)	1 (25.4)	1.5 (38.1)	2 (50.8)	2 (50.8)

For outdoor insulation it is a good rule of thumb to increase thickness by 1 in (25.4 mm) over that of indoor insulation. Similarly, when chilled-water and cooling-water piping as well as air-conditioning ducts must be routed through hot areas such as boiler rooms and laundries, additional insulation thickness should be considered.

When selecting insulation thicknesses for energy conservation purposes, the engineer must not overlook the fact that it may not be cost-effective to insulate piping and ductwork for liquids or gas in the temperature range of 55 to 120°F (12.8 to 48.9°C). As the temperature difference between the liquid or gas stream and the surrounding space decreases, so does the possibility of saving energy. A point is reached where this temperature difference is so small that heat loss or gain without insulation will not increase the annual energy requirements.

Infiltration. Infiltration is air flowing into a building or space through cracks around windows, doors, and skylights and through minute passageways and cracks within wall, floor, and roof structures. Infiltration always results in an additional heating load and an additional sensible and latent cooling load when portions of a building are under negative pressure because of stack effect in high-rise buildings or insufficient tempered makeup air. The heating and/or cooling infiltration load can be calculated from the formulas in standard handbook sources such as Refs. 2 and 3.

With present technology, it is not economically feasible to design a commercial, institutional, or industrial facility for zero infiltration. However, the engineer should select exterior wall components that will minimize the infiltration load. This will be cost-effective at the point when energy saved (by reducing the infiltration load) over the life of the facility is greater than the total cost of reducing the infiltration load.

The following ways of controlling the infiltration load are suggested to the designer:

1. Reduce the pressure differential across exterior doors and windows.
 - a. For exterior personnel entrances, provide vestibules with exterior and interior doors or revolving doors. The vestibules should have cabinet heaters and ducted tempered supply (pressurization) air. With revolving doors (four-section), tempered supply air can be ducted through the top of the two completely closed-in sections. The supply air volume should be automatically controlled to maintain the inside pressure equal to or slightly greater than the outside pressure.
 - b. Generally, space restraints preclude vestibules at loading-dock doors. However, air-curtain-type door heaters, mounted over the door and discharging downward, with shrouds (flexible closure pieces) to seal the space between the door opening and the truck or trailer body, have proven to be effective.
 - c. Infiltration at window areas can be controlled to acceptable levels by pressurizing the space (when the space does not have to be maintained under a negative pressure) by returning slightly less air ft³/min (m³/s) than one supplies to the space and selecting tightly closing, well-made window assemblies and hardware with good-quality seals around the perimeter and especially at all points where the sashes slide against the frame or past another sash.
2. Provide good-quality weather stripping seals around the perimeter of all doors.
3. Provide good-quality heavy building paper between the sheathing and exterior siding on all wood-constructed exterior walls.
4. Seal all exterior brick walls.

Ventilation. From an energy standpoint, when HVAC systems are operating in either the cooling or heating mode, ventilation air should be kept at the minimum quantity required to replenish the oxygen and dilute pollutants and contaminants in the indoor air to an acceptable level.

The American Society of Heating, Refrigerating, and Air Conditioning Engineers (ASHRAE) has recently revised its recommended outside ventilation air quantities upward in order to achieve acceptable indoor air quality by dilution with outdoor air only. (See Ref. 4 for details.) For a typical office, ASHRAE is recommending a four-fold increase from $5 \text{ ft}^3/(\text{min} \cdot \text{person})$ [$2.51 \text{ L}/(\text{s} \cdot \text{person})$] to $20 \text{ ft}^3/(\text{min} \cdot \text{person})$ [$10 \text{ L}/(\text{s} \cdot \text{person})$].

Before designing an HVAC system with these higher outdoor air ventilating quantities (dilution only), the engineer should evaluate the cost-effectiveness of (1) adding only the minimum quantity of outside air required to replenish oxygen and dilute unfilterable gases and (2) removing or reducing contaminants and pollutants in the return air by filtration. This procedure will create, at least, the same indoor-air quality as if higher outdoor-air-quantities were used.

The major design parameters for a typical dilution-removal filtration system are:

- The outdoor air quality will be set at the minimum required to replenish the oxygen (O_2) and dilute unfilterable gases, namely carbon monoxide (CO) and carbon dioxide (CO_2).
- The mixed air stream (outdoor and return air) first passes through a roughing filter, 2 in (50 mm) thick, in series with a 90 percent (minimum) ASHRAE 52.1-1992 efficiency and a filter at least 6 in (152 mm) deep.
- The air stream then passes through gas sorbers capable of removing a broad range of gases and vapors commonly found in a particular indoor and outdoor environment.
- The sorbers usually contain gas adsorbers and oxidizers, such as activated charcoal and alumina impregnated with potassium permanganate, depending on the gases present at the site or anticipated in the air stream.
- Odoroxidant media should be suitable for removing odorous, irritating, acidic gases from air by reacting chemically with the sorbed gases to prevent later desorption.
- The sorbers should be selected with sufficient capacity to remain active (effective) for a minimum service life of 4370 h (24 h per day for 6 months).
- The velocity through the sorber collection bed should provide a minimum residence time of approximately 0.06 s.
- High circulation rates (6 to 10 changes of the volume of air in each space per hour) are required to obtain effective mixing of the air within each space to capture and remove sufficient quantities of indoor contamination to provide the required indoor air quality.
- The filtered (supply) air should be discharged from diffusers that direct the air in a plug (flow predominately in one direction) or horizontal laminar flow pattern so the contaminants will be swept along with the flow across an occupied space to return-air intakes on the opposite side of the space.
- Since the static pressure drop across the combined high-efficiency particle filter and gas sorber is normally about 2 in water (497 Pa) and high circulation rates are required, it is not uncommon for this type of filtering system to have its own

fan system and operate either in conjunction with the building HVAC system or independent of it.

- When this type of filtering system is integrated with air-handling equipment containing cooling coils, the sorber section must be located downstream of the cooling coils and coil condensate drain pan to ensure that microbiological contaminants living on wet surfaces are removed before the air is distributed to the occupied spaces.

Exhaust and Makeup. For energy conservation, the engineer should determine the minimum exhaust air quantity for each system consistent with applicable codes and good engineering practices. To achieve this goal, the designer should evaluate each exhaust system with respect to the following items:

- When codes and good engineering practice permit shutting off the exhaust system when a facility is not occupied or a process is not operating, the designer should design a dedicated exhaust system that can be independently taken out of service. Though starting and stopping the exhaust system can be done manually, more energy will be saved if it is automatically done.
- Where the applicable codes do not mandate the exhaust air quantity for a particular type of space activity, it should be equal to the ventilation air quantity recommended for the activity in Ref. 4. Though not as current, Ref. 3 is also used.
- The industrial exhaust hoods should be as close to the source (oxygen or exhaust air) as possible to minimize the exhaust air volume.
- Push-pull exhaust systems should be considered for large tanks and vats.
- When possible, all tanks and vats should be provided with covers to reduce emission of vapors and odors.
- Generally, recirculating systems with adequate filtration should be used instead of exhausting air to the outside, whenever the particular industrial process or equipment and good engineering permit it.
- Low-volume, high-velocity exhaust systems should be used whenever possible to control dust from portable hand tools and machining operations.
- The industrial exhaust system should conform to the recommended practices set forth in the latest edition of Ref. 5.
- Will it be cost-effective to reduce the industrial exhaust air quantities by selecting less toxic or less hazardous materials or modernizing the process or equipment?

Once the exhaust air quantities have been established, the makeup air should be equal to the total exhaust air quantity unless there are specific areas that must be maintained at a negative or positive pressure. When there are equipment, processes, or areas that must be maintained at a negative or positive pressure, they should be enclosed in the smallest envelope possible and their makeup air should be supplied from a separate zone or unit.

The designer should evaluate the cost-effectiveness of recovering the heating or cooling energy in the exhaust air to heat or cool the makeup air. Thermal wheels, parallel-plate heat exchangers, coil runaround cycles, and heat-pipe recovery systems are discussed in Sec. 8.4.3.6.

Low-Leakage Dampers. High-performance, low-leakage dampers should be used for outside air, relief air, and return air and for mixing hot and cold air streams. The energy that can be saved by using high-performance, low-leakage dampers

instead of standard dampers is apparent when one compares the leakage rates at the same difference pressure drop across fully closed dampers.

Typical leakage rates are

Low-leakage dampers	Less than 1 percent of full flow
Standard dampers	5 percent minimum to 25 percent of full flow (depending on the quality of the manufacturing)

The static pressure drop across a fully open high-performance, low-leakage damper or standard damper is so small compared to the total system static pressure that, in general, there is no noticeable effect on the system energy usage with either type (in the fully open position).

The engineer may want to evaluate the cost-effectiveness of low-leakage dampers for other damper duties.

Coils (Heating and Cooling). From an energy conservation point of view, the engineer can reduce the energy used by the fans by selecting coils having minimum resistance to air flow. The heat-transfer surfaces on both air and fluid sides must be kept clean at all times with adequate water treatment on the fluid side and periodic cleaning on the air side. The following parameters can be used to select coils with low resistance to air flow:

For cooling coils:

Minimum velocity	400 ft/min (2 m/s)
Maximum velocity	500 ft/min (2.5 m/s)
Fan spacing	6 to 10 fins/in (0.24 to 0.39 fins/mm)

For heating coils:

Velocity range	500 to 800 ft/min (2.5 to 4 m/s)
Maximum fin spacing	14 fins/in (0.55 fins/mm)
Heat-recovery coils	Maximum velocity and fin spacing should be determined to maximize the energy recovered and minimize the cost of recovering it

The author acknowledges the manufacturers' position that for the same cooling load, coils with fin spacings of 6 to 10 fins/in (0.24 to 0.39 fins/mm) will probably require one and possibly two additional rows compared to a coil having 14 fins/in (0.55 fins/mm) and there will be no apparent difference in static pressure drop across the coil. That position is valid only when the coil is clean, however. With 14 fins/in (0.55 fins/mm) and wet fin surfaces, the particulate matter that passes through the filters will adhere to the wet fins and, in a relatively short time, will reduce the already narrow spacing between these fins. It has been the author's experience that in a short time, the static pressure drop across wet coils with 14 fins/in (0.55 fins/mm) becomes greater than for coils with 6 to 10 fins/in (0.24 to 0.39 fins/mm) even with the additional row or two. Considering the length of time between coil cleaning, the greater fin spacing can save energy.

Heating coils, on the other hand, always have dry fin surfaces and in general have no more than two rows. Under these conditions, the static pressure drop with 14 fins/in (0.55 fins/mm) will not result in a noticeable increase in fan energy.

Ductwork. Energy savings result when fans operate at the lower system pressures allowed by larger ductwork. The system design must provide operation cost savings (over the service life of the equipment) that more than offset the increased construction costs of the ductwork.

For a discussion of commercial and industrial system duct sizing procedures, the reader is referred to Chap. 3.2 of this book. For industrial exhaust system ducting, design in accordance with the procedures set forth in Ref. 5.

All duct seams of commercial and institutional duct systems should be taped and the maximum system leakage should not exceed the following:

- For a low-velocity system, $7\frac{1}{2}$ percent of the fan ft^2/min (m^3/s).
- For a medium- or high-velocity system, 5 percent of the fan ft^3/min (m^3/s).

All duct seams of industrial exhaust systems should be welded, brazed, or soldered, depending on the system temperature and duct material.

All hot and cold ducts should be insulated. See previous discussion of duct insulation in this section.

8.4.3.3 Types of Systems

General. When selecting air-handling units (especially for HVAC application) the engineer must always remember that probably less than 5 percent of the actual hours of heating or cooling system operation will be at the respective design load. The remainder of the time, the system will be operating at part load.

Though the actual part-load capacity and corresponding percent of operation time should be determined for each system, Table 8.4.3 can be used to estimate the order of magnitude of a typical heating or cooling HVAC system. From Table 8.4.3, it is apparent that the energy used (especially by fan and pump motors) in the 25 to 75 percent full-load range is extremely important. It is in this load range that one should concentrate on maximizing the system efficiency and minimizing the horsepower, and not at the design load. As a general rule, more energy can be saved by reducing the fan ft^3/min (m^3/s) and pump gal/min (m^3/s) than by reducing the supply air or water temperature to meet part-load conditions.

In order to conserve energy, areas and processes that are used after normal business hours should have their own HVAC and exhaust systems. Typically, these are the areas that must maintain design temperatures and relative humidity conditions 365 days a year (computer facilities, constant-temperature rooms, calibration laboratories, etc.). Also, auditoriums, cafeterias, conference rooms, and meeting rooms that are frequently used after normal business hours should be included in

TABLE 8.4.3 Heating or Cooling Operating Time at Various Loads for Typical HVAC Systems

Percent of full load	Percent of operating time
75–100	10
50–75	50
25–50	30
0–25	10

this category. The facilities that serve or support these areas, such as lobbies, corridors, toilets, lounges, and lunch rooms, should also be designed to operate independently of the main building system if one is to minimize energy costs and usage.

Though the engineer has six basic types of HVAC air-distribution systems from which to select that most appropriate for a design, those that can vary the air and liquid volume in accordance with variations in load generally have the lowest energy consumption.

Basic Systems. The six basic systems and their variations are:

1. *Single duct:* This is usually a low-velocity distribution system. The unit consists of filters, cooling and heating coils, supply fan, and sometimes a return fan.

The fans are generally centrifugal type, constant or variable volume. If the fans are variable volume, centrifugal or axial flow, they can be controlled by inlet vanes or variable-speed motors. Axial-flow fans, depending on size, can also be controlled by varying the pitch of the blades.

This system is suitable for single-zone application. When more than one zone is required, terminal reheats have been used to provide zone control. However, even when waste or reclaimed heat is then used for the reheat energy, it still may not have the lowest life-cycle costs.

2. *Dual duct:* This is usually a high-velocity supply and low-velocity return duct distribution system. The unit consists of filters, cooling and heating coils, and supply and return fans. The supply distribution mains consist of hot and cold ducts with mixing boxes at each zone. The ductwork from the mixing boxes to the diffusers is low-velocity.

The system is extremely flexible with respect to future modifications and has good temperature controls.

The size of the cold duct main should be based on the maximum building peak cooling load. The cold branch mains on a floor should be sized on the maximum simultaneous internal- and external-exposure peak loads of areas they serve. The hot duct is usually sized between 50 and 75 percent of the air capacity of the cold duct.

For energy conservation, the fans are generally airfoil variable-volume, centrifugal- or axial-flow types. Variable-frequency speed control is used on both types. Axial-flow fans are also available with adjustable-pitch blades.

The hot deck coil control valve should be closed during the cooling mode to conserve energy.

Even with these energy conservation measures, this system's energy consumption is relatively high.

3. *Multizone:* This is a low-velocity duct distribution system. The unit consists of filters, cooling and heating coils, hot and cold automatic modulating coil discharge air dampers, supply fan, and sometimes a return fan.

Depending on the size of the unit, six to ten zones with controls are common. The zone controls available with this type of unit are satisfactory for comfort air conditioning (such as in an office environment) but usually not for critical areas (such as laboratories).

The fans are centrifugal, constant-speed type.

This system varies each zone supply temperature by modulating its respective hot and cold deck dampers, as required, to satisfy the particular zone space temperature set point. It is not adaptable to varying the supply air volume. In some comfort air-conditioning installations, energy can be saved during the cooling cycle

by automatically closing the heating coil control valve during this mode of operation.

This system is generally relatively expensive to install and modify.

Even with the energy conservation measures noted above, the energy usage of this system will be higher than that of a variable air volume system.

A recent variation to the standard multizone uses individual zone heating and cooling coils instead of a common hot and cold deck with individual zone mixing dampers. The elimination of simultaneous heating and cooling and air-stream mixing losses can result in significant energy saving. Energy consumption of this unit can be as much as 40 percent less than a multizone unit with common hot and cold deck with individual zone mixing dampers. Only package rooftop units in the 15- to 37-ton refrigeration (52.8- to 130.1-kW) range are currently manufactured in this type.

These units are available with gas-fired heat, electric heat, or hot-water/glycol heat and direct-expansion cooling coils with multiple reciprocating compressors and air-cooled condensers. When high indoor relative humidity (in humid weather and during part load) is a concern, a direct expansion cooling coil in the outside air stream can be provided with this type of unit.

4. Variable air volume: This is usually a high-velocity supply, low-velocity return duct distribution system. The unit consists of filters, cooling and heating coils and supply and return fans. Return fans have been omitted on smaller systems.

Fans are variable-volume, centrifugal- or axial-flow type. Depending on fan size, the air volume can be varied by variable-frequency control or variable inlet vanes on smaller systems, or by variable blade pitch only on larger axial-flow fans.

The supply distribution main consists of a single duct with VAV boxes at the beginning of each zone duct. The ductwork leaving the VAV boxes to the diffusers is low-velocity.

The system is extremely flexible with respect to future modifications and has good temperature controls.

Care must be exercised in selecting the type of diffusers and controls. See discussion on VAV systems in Sec. 8.4.3.7.

The size of the main supply duct should be based on the maximum building peak cooling load. The branch mains on a floor should be sized on the maximum simultaneous interior and exterior exposure peak loads of the areas they serve.

For the commercial office, this system generally has the lowest energy usage and construction costs. However, there have been problems when VAV systems were used to air-condition laboratories and good-quality automatic temperature controls were not employed.

5. Fan coil unit: Each unit usually consists of a filter, combination heating and cooling coil, centrifugal fan, and supply and return grilles. Though not common, units are available with separate heating and cooling coils.

Although ceiling-mounted units are available, fan coil units are generally located at the floor against the exterior walls, preferably under the windows.

Since these units generally have no provision for ventilation air (that is, they recirculate 100 percent of the supply air), they are used in conjunction with single-duct, dual-duct, multizone, or variable-air-volume systems. The fan coil units are sized to handle the exterior (solar, transmission, and infiltration) cooling and heating load and the interior cooling load for the first 10 to 15 ft (3 to 4.6 m) from the exterior wall. The interior system will provide the ventilation air for the exterior zones. This combined system significantly reduces the size of the distribution ductwork and the associated construction cost, since the ducted system serves only the

interior loads and ventilation air requirements. The system combined with a VAV interior system is used most often in modern offices and is among the lowest energy users.

Units are available that have provision for ventilation air. They are generally self-contained, packaged heat pumps with their own air-cooled direct-expansion compressor, cooling coil, and supplementary electric heat. They are predominantly used in schools, motels, and hotels. If there are extended periods during the heating and cooling seasons when the spaces served are not occupied, energy usage is reasonable. However, in areas where the ambient heating design temperature is 12°F (−11°C) or lower and there are 5000 (2780) degree-days or more, energy usage is generally high, since under these conditions the heating is mostly electric.

The self-contained heat-pump units are thermostatically controlled. The other unit capacities can be regulated by varying the water flowing through the coil with an automatic temperature-controlled water-regulating valve or by varying the fan speed. Though varying the fan speed requires constant flow through the coil, and the choice of pump size is therefore restricted and the possibility of saving pump energy by reducing the flow is eliminated, it is economical and is the method most often provided for these units.

6. Induction unit: This is a constant-volume, low- or high-velocity system. It consists of a centrally located unit that filters, cools, and dehumidifies the primary air and induction units located generally at the floor along the walls. Each induction unit consists of a primary air plenum (which is sound-attenuated), primary air nozzle, mixing chamber, heating coil, and return and discharge grilles.

The primary air is ducted to each induction unit. At each induction unit the primary air flow enters the primary air plenum and leaves through the primary air nozzle at high velocity, inducing return air from the space to flow into the mixing chamber and mix with the primary air. The mixed air leaves the unit and enters the conditioned space.

The primary air provides the ventilation air and cooling requirements of the conditioned spaces. The heating coil in the return air stream provides the heating requirements.

Though this system was popular before the energy crisis and provides good temperature control, it is seldom selected any more for new facilities because of its high energy use.

8.4.3.4 Chillers

Centrifugal. To minimize energy use, the following guidelines should be considered:

- For commercial and institutional applications, the number and size of the refrigeration units should be determined so that the number of units on line (operating) will have the lowest kilowatts per ton (kW/W) ratio—in the range of 75 to 25 percent of design load—since approximately 80 percent of the hours of operation will be in this load range. If units have a significantly lower kilowatts per ton ratio in the 75 to 50 percent of design load range, they should be selected since approximately 50 percent of the hours of operation will occur in this load range. See the general discussion of this in Sec. 8.4.3.3 a preceding portion of this section “Types of Systems” for typical part-load operation.
- For industrial or other applications where the cooling load does not vary appreciably with the ambient weather conditions, the number and size of the refrigeration units should be determined so that the number of units on line (operating) will have the lowest kilowatts per ton (kW/W) ratio—in the range of 75 to 25 percent of design load—since approximately 80 percent of the hours of operation will be in this load range. If units have a significantly lower kilowatts per ton ratio in the 75 to 50 percent of design load range, they should be selected since approximately 50 percent of the hours of operation will occur in this load range. See the general discussion of this in Sec. 8.4.3.3 a preceding portion of this section “Types of Systems” for typical part-load operation.

eration units should be chosen to produce the lowest kilowatts per ton (kW/W) over the duration of the cooling load.

- Select chilled-water supply temperatures at the highest possible temperature that will maintain space design temperature and humidity under maximum load conditions.
- Select refrigerant compressors to operate at the highest suction pressure and the lowest head pressure possible and still maintain the required supply chilled-water temperature under maximum load conditions.
- Select refrigerant compressors that can maximize the energy reduction possible with lower condenser water-supply temperatures under part-load conditions.
- Provide automatic controls that can reset the supply chilled-water temperature to the highest level under part-load operation and still maintain space design temperature and humidity conditions.

For a discussion on heat recovery with double bundle condensers see Sec. 8.4.3.6, "Waste Heat and Heat Recovery."

Heat-transfer surfaces must be kept clean at all times with adequate water treatment and periodic cleaning.

Absorption. When waste heat [preferably steam around 12 lb/in² (0.8 bar)] is available and chilled water is required, absorption refrigeration units should seriously be considered to save energy and improve the overall plant efficiency.

However, when steam or hot water must be generated expressly for absorption units, the engineer must evaluate the following before selecting the type of refrigeration units:

- The water rate for a single-stage absorption unit for 12-lb/in² (0.8-bar) steam of about 18 to 20 lb/h of steam per ton of refrigeration (2.3 to 2.6 kg/kW), or its equivalent hot-water value, is not energy-efficient. Furthermore, the heat rejection to the cooling tower is about 200 percent greater than that of an electric-driven compressor unit for the same refrigeration capacity.
- The water rate for a two-stage absorption unit with 125- to 150-lb/in² (8.6- to 10.3-bar) steam entering the first stage is about 12 to 14 lb/h of steam per ton of refrigeration (1.5 to 1.8 kg/kW), which indicates a significant reduction in steam energy, or its equivalent high-temperature water at 355°F (179°C), for the same refrigeration capacity. However, the lithium bromide refrigerant solution used in absorption units is extremely corrosive at the elevated temperatures at which the first stage operates. Although manufacturers of two-stage units profess that corrosion will not be a problem if their water treatment requirements are strictly adhered to, it is the author's experience and position that corrosion and/or the potential corrosion-related problems are a major concern and repair expense for users of two-stage units.
- Guidelines for selecting the number and size of absorption units are similar to those noted under the heading Centrifugal, above.

Direct-Expansion Evaporators—Screw Compressors and Reciprocating Compressors

- Generally screw compressors are more economical above 100 tons (350 kW) of refrigeration, whereas reciprocating compressors are more economical below that capacity.

- The same criteria described under the heading Centrifugal should be used to select units with water-cooled condensers.
- If the condenser is *air-cooled*, the same criteria described under the heading Centrifugal should be used, except for the automatic controls for the condenser fans. These should be able to lower the condensing temperature (head pressure) as the ambient dry-bulb temperature drops to the lowest recommended by the compressor manufacturer, yet maintain the required (compressor) suction pressure.
- If the condenser is *evaporative*, the same criteria described under Centrifugal should be used except for the automatic controls for the evaporative condenser fans. These should automatically control the spray pump and condenser fan, so as to lower the condensing temperature (head pressure) as the ambient dry- and/or wet-bulb temperature decreases to the lowest recommended by the compressor manufacturer, yet maintain the required (compressor) suction pressure.

Cooling Tower. For energy conservation, towers should be selected in conjunction with the refrigeration unit to produce the lowest kilowatt per ton of refrigeration (kW/W) ratio. To achieve this goal, the following guidelines should be considered:

- Induced-draft towers should be selected over forced-draft towers since they require significantly less fan horsepower (kW) for the same cooling requirement.
- Hyperbolic natural-draft cooling towers are without question the most energy-efficient. However, their minimum effective size is approximately 250,000 gal/min (15.8 m³/8), which is far greater than the central refrigeration plant requirements we are concerned with in this book.
- Though it is possible to design a natural-draft tower (without mechanical fans) in the capacity range we would need, it would be inefficient and would need a large amount of space. However, if space is available, natural-draft cooling towers, as well as spray pounds, should be considered.
- If the project is located near a river, lake, or other large body of water, it should be considered as a source of condenser water before a mechanical-draft cooling tower is selected.
- Groundwater has been used for precooling and condenser water. However, requirements for recharging wells and restrictions on groundwater contamination generally make this source of condenser water uneconomical.
- The three major cooling tower parameters are:

Ambient wet-bulb temperature: This temperature should be selected with care, since the wet-bulb temperature of the air entering the tower is the basis for the thermal design of any evaporative-type cooling tower.

Range: This is the difference in temperature between hot water entering the tower [condenser water return (CWR)] and the cold water leaving the tower [condenser water supply (CWS)]. Of these two temperatures, the tower size is primarily affected by the CWS temperature.

Approach: This is the difference between the cold-water temperature leaving the tower and the entering air wet-bulb temperature. The approach is important for two primary reasons: first, it sets the CWS temperature; the lower this temperature is, the lower the refrigeration unit kilowatts per ton of refrigeration (kW/W) ratio will be. Second, it fixes the size and efficiency of the cooling tower. Although increasing the tower efficiency will measurably decrease the approach, there are limits. In practice it is the tower size that is significantly increased to achieve the

lower approach requirements. The closest approach that can be achieved is 5°F (2.8°C).

- It is generally more cost-effective to increase the tower size to obtain lower CWS than to increase the refrigeration unit kilowatt per ton (kW/W) ratio.
- Towers should be selected to minimize the drift and evaporation losses.
- Automatic temperature controls capable of resetting (lowering) the condenser water supply temperature as the ambient wet-bulb temperature drops should be provided.
- Tower fan motors should be two-speed to improve part-load efficiency.
- The heat-transfer surfaces must be kept clean at all times with adequate water treatment and periodic cleaning.

Several heat-recovery systems using cooling towers are discussed in Sec. 8.4.3.6, "Waste Heat and Heat Recovery."

8.4.3.5 Boilers

To minimize energy usage the following guidelines should be considered:

- For comfort heating, the number and size of the boilers should be determined so that the number of units on line (operating) will be close to their maximum efficiency point at part loads ranging from 75 to 25 percent of design load, since approximately 80 percent of the hours of operation will be in this load range. If significantly higher efficiencies can be obtained by selecting boilers operating in the 75 to 50 percent of design load range, then the boiler size and number should be determined at this load range, since they will be operating in this range approximately 50 percent of the time. See the general discussion in Sec. 8.4.3.3 for typical part-load application. For process boilers, the number and size of the boilers should be determined to maximize the plant efficiency.
- Boilers should be selected for lead-lag control with low fire rats to minimize the on and off cycling of the lead boiler.
- Boiler insulation type and thickness should be selected to minimize the cooldown and radiation heat loss. The burner flame shape and heat output must be selected to match the dimensions of the combustion chamber.
- The burner controls for multiple boilers should be capable of the following:
 - Automatically cycling the boiler on and off (lead-lag control) and modulating its firing rate in accordance with load swings.
 - Continually monitoring the flue gas for excess O₂ and CO content and excess temperature.
 - Automatically adjusting the firing rate according to the operating parameters (flue gas O₂, CO, and temperature) and actual plant load for the highest obtainable combustion efficiency.
- The combustion air volume should be set at the lowest safe maintainable value that the boiler controls can operate at.
- The stack gas temperature should be as low as possible, approximately 50°F (10°C) above the lowest combustion-gas dew point.

CHAPTER 8.5

WATER CONDITIONING

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8.5.1 INTRODUCTION

It is the object of this chapter to discuss state-of-the-art technology of water treatment for commercial and industrial heat-transfer equipment, with specific emphasis on heating, ventilating, and air-conditioning (HVAC) systems. Since water treatment for industrial processes requires a specific design for each process, it is beyond the scope of this chapter to cover all aspects of industrial water treatment. For fuller coverage of industrial water treatment, see the bibliography (Sec. 8.5.10) at the end of the chapter.

8.5.2 WHY WATER TREATMENT?

Water treatment for corrosion and deposit control is a specialized technology. Essentially, it can be understood when one first recognizes why treatment is necessary to prevent serious failures and malfunction of equipment which uses water as a heat-transfer medium. This is seen more easily when one observes the problems water can cause, the mechanism by which water causes these problems, which leads to solutions, and the actual solutions or cures available.

Water is a universal solvent. Whenever it comes into contact with a foreign substance, there is some dissolution of that substance. Some substances dissolve at faster rates than others, but in all cases a definite interaction occurs between water and whatever it contacts. It is because of this interaction that problems occur in equipment such as boilers or cooling-water systems in which water is used as a heat-transfer medium. In systems open to the atmosphere, corrosion problems are made worse by additional impurities picked up by the water from the atmosphere.

Most people have seen the most obvious examples of corrosion of metals in contact with water and its devastating effect. Corrosion alone is the cause of failure and costly replacement of equipment and is itself a good reason why water treatment is necessary.

8.5.2.1 Cost of Corrosion

The direct losses due to corrosion of metals for replacement and protection are reported to be \$10 to \$15 billion annually; over \$5 billion is spent for corrosion-resistant metallic and plastic equipment, almost \$3 billion for protective coatings, and over \$340 million for corrosion inhibitors (1978 dollars).¹ All this is just to minimize the losses due to corrosion. Typical examples of these losses resulting from failures of piping, boiler equipment, and heat-exchanger materials because of corrosion and deposits are depicted in this chapter. Only with correct application of corrosion inhibitors and water treatment will HVAC equipment, such as heating boilers and air-conditioning chillers and condensers, provide maximum economical service life. However, even more costly than failures and replacement costs, and less obvious, is the more insidious loss in energy and operating efficiency due to corrosion and deposits.

In heat-transfer equipment, corrosion and deposits will interfere with the normal efficient transfer of heat energy from one side to the other. The degree of interference with this transfer of heat in a heat exchanger is called the *fouling factor*. In the condenser of an air-conditioning machine, a high fouling factor causes an increase in condensing temperature of the refrigerant gas and thus an increase in energy requirements to compress the refrigerant at that higher temperature. The manufacturer's recommended design fouling factor for air-conditioning chillers and condensers is 0.0005. This means that the equipment cannot tolerate deposits with a fouling factor greater than 0.0005 without the efficiency of the machine being seriously reduced.

Figure 8.5.1 graphically illustrates the effect of scale on the condensing temperature of a typical water-cooled condenser. From this graph, we see that the condensing temperature increases in proportion to the fouling factor. An increase in condensing temperature requires a proportionate increase in energy or compressor horsepower to compress the refrigerant gas. Thus the fouling factor affects the compressor horsepower and energy consumption, as shown in Fig. 8.5.2. Condenser

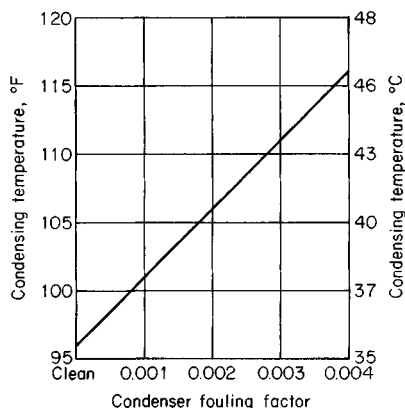


FIGURE 8.5.1 Effect of scale on condensing temperature. (From *Carrier System Design Manual*, part 5, "Water Conditioning," Carrier Corporation, Syracuse, NY, 1972, p. 5-2. Used with permission.)

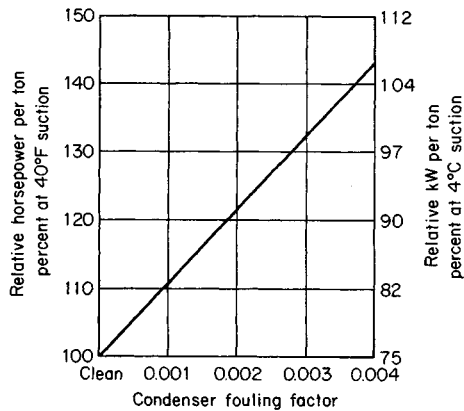


FIGURE 8.5.2 Effect of scale on compressor horsepower. (From *Carrier system Design Manual*, part 5, "Water conditioning," Carrier Corporation, Syracuse, NY, 1972, p. 5-2, Used with permission.)

tubes are quickly fouled by a hard water supply which deposits calcium carbonate on the heat-transfer surface. The explanation of the mechanism of this type of fouling is given in a later section.

Table 8.5.1 lists the fouling factors of various thicknesses of a calcium carbonate type of scale deposit most frequently found on condenser watertube surfaces where no water treatment or incorrect treatment is applied.

The additional energy consumption required to compensate for a calcium carbonate type of scale on condenser tube surfaces of a refrigeration machine is illustrated in Fig. 8.5.3. The graph shows that a scale thickness of only 0.025 in (0.635 mm) [fouling factor of 0.002] will result in a 22 percent increase in energy consumption, which is indeed wasteful.

8.5.2.2 Cost of Scale and Deposits

The actual cost of scale is even more surprising. For example, a 500-ton air-conditioning plant operating with a scale deposit of 0.025 in (0.635 mm) of a

TABLE 8.5.1 Fouling Factor of Calcium Carbonate Type of Scale

Approximate thickness of calcium carbonate type of scale, in (mm)	Fouling factor
0.000	Clean
0.006 (0.1524)	0.0005
0.012 (0.3048)	0.0010
0.024 (0.6096)	0.0020
0.036 (0.9144)	0.0030

Source: *Carrier System Design Manual*, part 5, "Water Conditioning," Carrier Corp., Syracuse, NY, 1972, p. 5-3. Used with permission.

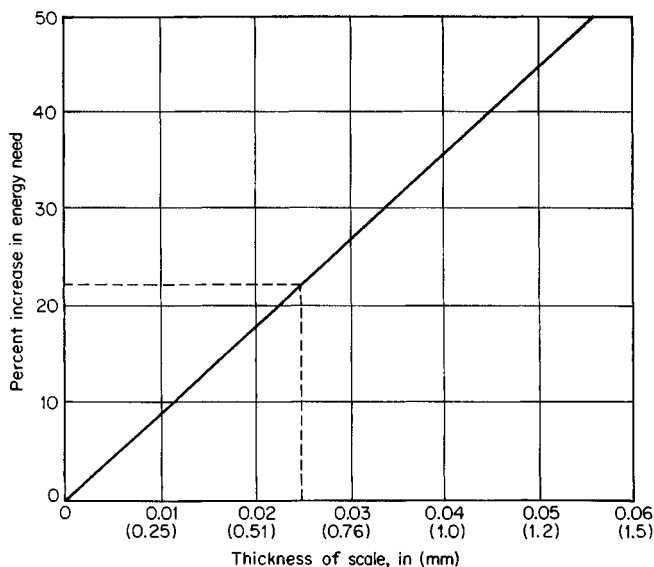


FIGURE 8.5.3 Effect of condenser tube scale on energy consumption, $K = 1.0 \text{ Btu}/(\text{h} \cdot \text{ft}^3 \cdot ^\circ\text{F})$. *Example:* Scale that is 0.025 in (0.6 mm) thick requires 22 percent increase in energy.

calcium carbonate type will increase energy requirements by 22 percent if the same refrigeration load is maintained and cost \$2870 in additional energy consumption required for only 1 month (720 h) of operation. This is based on an efficient electric-drive air-conditioning machine's requiring $0.75 \text{ kW}/(\text{h} \cdot \text{ton})$ of refrigeration for compressor operation. The average cost for this energy in early 1995 was 5.0 cents/kWh.

With proper care and attention to water treatment, wasteful use of energy can be avoided. Likewise, in a boiler operation for heating or other purposes, an insulating scale deposit on the heat-transfer surfaces can substantially increase energy requirements.

Boiler scale or deposits can consist of various substances including iron, silica, calcium, magnesium, carbonates, sulfate, and phosphates. Each of these, when deposited on a boiler tube, contributes in some degree to the insulation of the tube. That is, the deposits reduce the rate of heat transfer from the hot gases or fire through the boiler metal to the boiling water.

When this occurs, the temperature of the boiler tube metal increases. The scale coating offers a resistance to the rate of heat transfer from the furnace gases to the boiler water. This heat resistance results in a rapid rise in metal temperature to the point at which the metal bulges and eventual failure results. This is the most serious effect of boiler deposits, since failure of such tubes causes boiler explosions. Figure 8.5.4 shows a boiler tube blister caused by a scale deposit.

Table 8.5.2 shows the average loss of energy as a result of boiler scale. A normal scale of only $1/16$ -in (1.588-mm) thickness can cause an energy loss of 4 percent. For example, a loss of 4 percent in energy as a result of a scale deposit can mean that 864 gal (3270.6 L) more of No. 6 fuel oil than is normally used would be

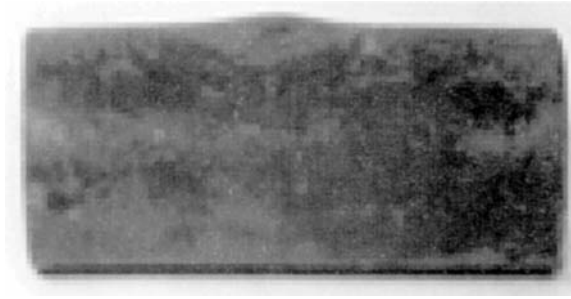


FIGURE 8.5.4 Boiler tube blister. (Courtesy of Metropolitan Refining Co., Inc.)

TABLE 8.5.2 Boiler Scaled Thickness vs. Energy Loss

Normal scale, calcium carbonate type, in (mm)	Dense scale (iron silica type)	Energy loss, %
$\frac{1}{32}$ (0.794)	$\frac{1}{64}$ (0.397)	2
$\frac{1}{16}$ (1.588)	$\frac{1}{32}$ (0.794)	4
$\frac{3}{32}$ (2.381)	$\frac{3}{64}$ (1.191)	6
$\frac{1}{8}$ (3.175)	$\frac{1}{16}$ (1.588)	8
$\frac{3}{16}$ (4.763)	$\frac{3}{32}$ (2.381)	12
$\frac{1}{4}$ (6.350)	$\frac{1}{8}$ (3.175)	16

required for the operation of a steam boiler at 100 boiler hp (bhp) (1564.9 kg for 1 month (720 h).

8.5.3 WATER CHEMISTRY

Water and its impurities are responsible for the corrosion of metals and formation of deposits on heat-transfer surfaces, which in turn reduce efficiency and waste energy. Having seen the effects of corrosion and deposits, let us see how this can be prevented. The path to their prevention can best be approached through understanding their basic causes, why and how they occur.

Water, the common ingredient present in heat-transfer equipment such as boilers, cooling towers, and heat exchangers, contains many impurities. These impurities render the water supply more or less corrosive and/or scale-forming.

8.5.3.1 Hydrologic Cycle

The hydrologic cycle (Fig. 8.5.5) consists of three stages: evaporation, condensation, and precipitation. This cycle begins when surface waters on the earth are heated by the rays of the sun, vaporized, and raised into the troposphere, a thin layer of air and moisture approximately 7 mi (11 Km) thick which surrounds the

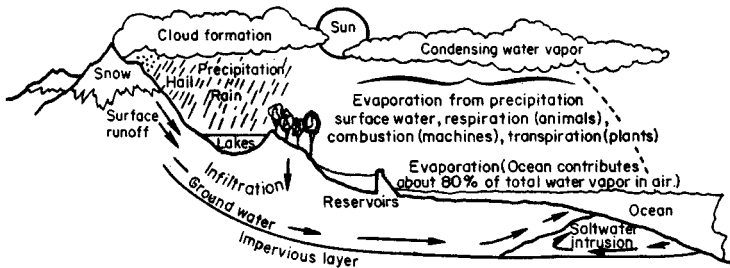


FIGURE 8.5.5 Hydrologic cycle.

earth. Clouds of condensed moisture form in the troposphere, and when carried over land by the wind, they contact cold-air currents. This causes precipitation or rain or snow. In this manner, water returns to the earth's surface, only to repeat the cycle.

Throughout the hydrologic cycle, the water absorbs impurities. While falling through the atmosphere, water dissolves the gases, oxygen, nitrogen, carbon dioxide, nitrogen oxides, sulfur oxides, and many other oxides present in the atmosphere in trace amounts.

The quantity of these gases in the atmosphere depends on the location. For example, in large urban areas rainwater contains high concentrations of carbon dioxide, sulfur oxides, and nitrogen oxides. In rural areas, water contains lesser amounts of these gases. A study made by Gene E. Likens of Cornell University noted that in the recent past the acidity of our rainfall has steadily increased.² This is caused by the increased amounts of sulfur and nitrogen oxide gases that pollute the atmosphere.

8.5.3.2 Water Impurities

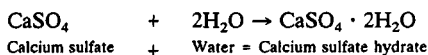
In contact with the earth surface, rainwater will tend to dissolve and absorb many of the minerals of the earth. The more acidic the rainfall, the greater the reaction with the earth's minerals. This reaction includes hydrolysis and hydration. As water passes over and through gypsum, calcite, dolomite, and quartz rock, it will dissolve calcium, silica, and magnesium minerals from these rocks (Table 8.5.3). In similar

TABLE 8.5.3 Reactions of Water with Minerals

Hydrolysis is the chemical reaction between water and minerals in which the mineral dissolves in the water:



Hydration is the absorption of water by minerals, changing the nature of the mineral:



manner, other minerals present in the earth's crust can be dissolved and taken up by the water. Table 8.5.4 shows some of the minerals present in the earth's surface which by reaction with water become impurities in water. Water accumulates on the earth's surface in lakes, rivers, streams, and ponds and can be collected in reservoirs. These surface water supplies usually contain fewer minerals but are more likely to contain dissolved gases.

Underground water supplies are a result of surface waters' percolating through the soil and rock. The water supplies usually contain large quantities of minerals and not much dissolved gases, although there are numerous exceptions to this general rule. Table 8.5.5 lists the various sources of water. Figures 8.5.6 through 8.5.10 show typical analyses of surface waters and underground well waters.

A brief observation of the analyses of these different water supplies shows that the natural impurities and mineral content do indeed vary with location. In fact, many well water supplies in a very proximate location exhibit vast differences in mineral content. Let us examine each of the basic impurities of water to see how they contribute to corrosion and deposits.

8.5.3.3 Dissolved Gases

Oxygen. One of the gases in the atmosphere is oxygen which makes up approximately 20 percent of air. Oxygen in water is essential for aquatic life; however, it is the basic factor in the corrosion process and is, in fact, one of the essential

TABLE 8.5.4 Mineral Groups

Silicates	Quartz, augite, mica, chert, feldspar, hornblend
Carbonates	Calcite, dolomite, limestone
Halides	Halite, fluorite
Oxides	Hematite, ice, magnetite, bauxite
Sulfates	Anhydrite, gypsum
Sulfides	Galena, pyrite
Natural elements	Copper, sulfur, gold, silver
Phosphates	Apatite

TABLE 8.5.5 Sources of Water

Surface water	Lakes and reservoirs of fresh water
Groundwater	Water below the land surface caused by surface run-off drainage and seepage
Water table	Water found in rock saturated with water just above the impervious layer of the earth
Wells	Water-bearing strata of the earth—water seeps and drains through the soil surface, dissolving and absorbing minerals of which the earth is composed (thus the higher mineral content of well water)



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CERTIFICATE OF ANALYSIS

WATER ANALYSIS

Divisions:
Metropolitan Refining
Consolidated Water Conditioning
Cosmopolitan Chemical
Petro Con Chemical

CLIENT: _____ DATE: _____
ADDRESS: _____ REPRESENTATIVE: _____ SAMPLE DATE: _____
NEW YORK, NY (CROTON RESERVOIR) ANALYSIS NO.: 339568 SOURCE: CITY _____

pH		6.9	PHOSPHATE	PO ₄	mg/L	
P ALKALINITY	CaCO ₃	mg/L	MOLYBDATE	Na ₂ MoO ₄	mg/L	
FREE CARBON DIOXIDE	CO ₂	mg/L	NITRITE	NaNO ₂	mg/L	
BICARBONATES	CaCO ₃	mg/L	ZINC	Zn	mg/L	
CARBONATES	CaCO ₃	mg/L	SPECIFIC CONDUCTANCE	ms/cm/cm		
HYDROXIDES	CaCO ₃	mg/L	TOTAL DISSOLVED SOLIDS	mg/L	33.5	
M (Total) ALKALINITY	CaCO ₃	mg/L	SUSPENDED MATTER			
TOTAL HARDNESS	CaCO ₃	mg/L	BIOLOGICAL GROWTHS	TOTAL BACTERIA COLONIES/ML		
SULFATE	SO ₄	mg/L	SPECIFIC GRAVITY @ 15.5°/15.5°C			
SILICA	SiO ₂	mg/L	FREEZING POINT			
IRON	Fe	mg/L	TRACE	% BY WEIGHT		
CHLORIDE	NaCl	mg/L	13			
ORGANIC INHIBITOR	PHOSPHONATE	mg/L				

RESULTS:

TREATMENT	TREATMENT CONTROL	FOUND	RECOMMENDED

REMARKS:

NOTES:

ANALYTICAL RESULTS EXPRESSED IN MILLIGRAMS PER LITRE (mg/L) ARE EQUIVALENT TO PARTS PER MILLION (ppm).
DIVIDE BY 17.1 TO OBTAIN GRAINS PER GALLON (gpg).
CYCLES OF CONCENTRATION = CHLORIDES IN SAMPLE/CHLORIDES IN MAKEUP

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FIGURE 8.5.6 New York City (Croton Reservoir) water analysis. (Courtesy of The Metro Group, Inc.)

elements in the corrosion process of metals. Therefore, dissolved oxygen in water is important to us in the study of corrosion and deposits.

Carbon Dioxide. Carbon dioxide is present in both surface and underground water supplies. These water supplies absorb small quantities of carbon dioxide from the atmosphere. Larger amounts of carbon dioxide are absorbed from the decay of organic matter in the water and its environs. Carbon dioxide contributes signifi-



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WATER ANALYSIS

Divisions:
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Consolidated Water Conditioning
Cosmopolitan Chemical
Petro Con Chemical

CLIENT: _____ DATE: _____
ADDRESS: _____ REPRESENTATIVE: _____ SAMPLE DATE: _____
SYRACUSE, N.Y. (OTISCO LAKE) ANALYSIS NO.: 57627 SOURCE: CITY _____

pH			7.4	PHOSPHATE	PO ₄	mg/L	
P ALKALINITY	CaCO ₃	mg/L	0.0	MOLYBDATE	Na ₂ MoO ₄	mg/L	
FREE CARBON DIOXIDE	CO ₂	mg/L		NITRITE	NaNO ₂	mg/L	
BICARBONATES	CaCO ₃	mg/L	85.	ZINC	Zn	mg/L	
CARBONATES	CaCO ₃	mg/L		SPECIFIC CONDUCTANCE	msiemens/cm		245.
HYDROXIDES	CaCO ₃	mg/L		TOTAL DISSOLVED SOLIDS		mg/L	148.
M (Total) ALKALINITY	CaCO ₃	mg/L	85.	SUSPENDED MATTER			TRACE
TOTAL HARDNESS	CaCO ₃	mg/L	132.	BIOLOGICAL GROWTHS TOTAL BACTERIA COLONIES/ML			TRACE
SULFATE	SO ₄	mg/L		SPECIFIC GRAVITY @ 15.5°/15.5°C			
SILICA	SiO ₂	mg/L	1.0	FREEZING POINT			
IRON	Fe	mg/L	0.0	% BY WEIGHT			
CHLORIDE	NaCl	mg/L	21.				
ORGANIC INHIBITOR	PHOSPHONATE	mg/L					

RESULTS:

TREATMENT	TREATMENT CONTROL	FOUND	RECOMMENDED

REMARKS:

NOTES:

ANALYTICAL RESULTS EXPRESSED IN MILLIGRAMS PER LITRE (mg/L) ARE EQUIVALENT TO PARTS PER MILLION (ppm).
DIVIDE BY 17.1 TO OBTAIN GRAINS PER GALLON (gpg).
CYCLES OF CONCENTRATION = CHLORIDES IN SAMPLE/CHLORIDES IN MAKEUP

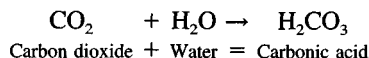
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FIGURE 8.5.7 Water analysis of Syracuse, NY (Otisco Lake). (Courtesy of The Metro Group, Inc.)

cantly to corrosion by making water acidic. This increases its capability to dissolve metals. Carbon dioxide forms the mild carbonic acid when dissolved in water, as follows:





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Divisions:
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Cosmopolitan Chemical
Petro Con Chemical

CLIENT: _____ DATE: _____
ADDRESS: _____ REPRESENTATIVE: _____ SAMPLE DATE: _____
WASHINGTON, D.C. (POTOMAC RIVER) ANALYSIS NO.: 20197 SOURCE: CITY _____

pH			7.3	PHOSPHATE	PO ₄	mg/L	
P ALKALINITY	CaCO ₃	mg/L		MOLYBDATE	Na ₂ MoO ₄	mg/L	
FREE CARBON DIOXIDE	CO ₂	mg/L		NITRITE	NaNO ₂	mg/L	
BICARBONATES	CaCO ₃	mg/L	90.	ZINC	Zn	mg/L	
CARBONATES	CaCO ₃	mg/L		SPECIFIC CONDUCTANCE	ms/cm @ 25°C		
HYDROXIDES	CaCO ₃	mg/L		TOTAL DISSOLVED SOLIDS		mg/L	195.
M (Total) ALKALINITY	CaCO ₃	mg/L	90.	SUSPENDED MATTER			
TOTAL HARDNESS	CaCO ₃	mg/L	140.	BIOLOGICAL GROWTHS TOTAL BACTERIA COLONIES/ML			
SULFATE	SO ₄	mg/L		SPECIFIC GRAVITY @ 15.5°/15.5°C			
SILICA	SiO ₂	mg/L	7.0	FREEZING POINT			
IRON	Fe	mg/L	0.0	% BY WEIGHT			
CHLORIDE	NaCl	mg/L	41.				
ORGANIC INHIBITOR	PHOSPHONATE	mg/L					

RESULTS:

TREATMENT	TREATMENT CONTROL	FOUND	RECOMMENDED

REMARKS:

NOTES:

ANALYTICAL RESULTS EXPRESSED IN MILLIGRAMS PER LITRE (mg/L) ARE EQUIVALENT TO PARTS PER MILLION (ppm).
DIVIDE BY 17.1 TO OBTAIN GRAINS PER GALLON (gpg).
CYCLES OF CONCENTRATION = CHLORIDES IN SAMPLE/CHLORIDES IN MAKEUP

SW:

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FIGURE 8.5.8 Potomac River (Washington, DC) water analysis. (Courtesy of The Metro Group, Inc.)

Sulfur Oxides. Sulfur oxide gases are present in the atmosphere as a result of sulfur oxides absorbed from the atmosphere, in which they are present as pollutants from the combustion of fuels containing sulfur, such as coal and fuel oil. In large urban areas, the quantity of sulfur oxides that are absorbed by surface water supplies and aerated waters used in cooling towers can be significant. Also when dissolved in water, sulfur oxides form acids which create a corrosive atmosphere.



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CERTIFICATE OF ANALYSIS

WATER ANALYSIS

Divisions:
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Cosmopolitan Chemical
Petro Con Chemical

CLIENT: _____ DATE: _____
ADDRESS: _____ REPRESENTATIVE: _____ SAMPLE DATE: _____
JAMAICA, N.Y. (WELLS) ANALYSIS NO.: 38140 SOURCE: CITY WATER

pH			7.0	PHOSPHATE	PO ₄	mg/L	
P ALKALINITY	CaCO ₃	mg/L	0.0	MOLYBDATE	Na ₂ MoO ₄	mg/L	
FREE CARBON DIOXIDE	CO ₂	mg/L		NITRITE	NaNO ₂	mg/L	
BICARBONATES	CaCO ₃	mg/L	30.	ZINC	Zn	mg/L	
CARBONATES	CaCO ₃	mg/L		SPECIFIC CONDUCTANCE	megohms/cm		154.
HYDROXIDES	CaCO ₃	mg/L		TOTAL DISSOLVED SOLIDS	mg/L		106.
M (Total) ALKALINITY	CaCO ₃	mg/L	30.	SUSPENDED MATTER			
TOTAL HARDNESS	CaCO ₃	mg/L	60.	BIOLOGICAL GROWTHS	TOTAL BACTERIA COLONIES/ML		
SULFATE	SO ₄	mg/L		SPECIFIC GRAVITY @ 15.5°/15.5°C			
SILICA	SiO ₂	mg/L	14.3	FREEZING POINT			
IRON	Fe	mg/L	0.07	% BY WEIGHT			
CHLORIDE	NaCl	mg/L	29.				
ORGANIC INHIBITOR	PHOSPHONATE	mg/L					

RESULTS:

TREATMENT	TREATMENT CONTROL	FOUND	RECOMMENDED

REMARKS:

NOTES:

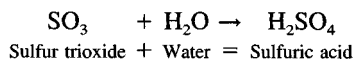
ANALYTICAL RESULTS EXPRESSED IN MILLIGRAMS PER LITRE (mg/L) ARE EQUIVALENT TO PARTS PER MILLION (ppm).
DIVIDE BY 17.1 TO OBTAIN GRAINS PER GALLON (gpg).
CYCLES OF CONCENTRATION = CHLORIDES IN SAMPLE/CHLORIDES IN MAKEUP

SW:

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FIGURE 8.5.9 Water analysis of Jamaica, NY (wells). (Courtesy of The Metro Group, Inc.)



Nitrogen Oxides. Nitrogen oxides are also present in the atmosphere both naturally and from pollutants created by the combustion process. These, too, form acids when absorbed by water and contribute to the corrosion process.



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CERTIFICATE OF ANALYSIS WATER ANALYSIS

Divisions:
Metropolitan Refining
Consolidated Water Conditioning
Cosmopolitan Chemical
Petro Con Chemical

CLIENT: _____ DATE: _____
ADDRESS: _____ REPRESENTATIVE: _____ SAMPLE DATE: _____
YELLOW SPRINGS, OHIO (WELLS) ANALYSIS NO.: 47588 SOURCE: CITY WATER

pH			7.8	PHOSPHATE	PO ₄	mg/L	
P ALKALINITY	CaCO ₃	mg/L	0.0	MOLYBDATE	Na ₂ MoO ₄	mg/L	
FREE CARBON DIOXIDE	CO ₂	mg/L		NITRITE	NaNO ₂	mg/L	
BICARBONATES	CaCO ₃	mg/L	300.	ZINC	Zn	mg/L	
CARBONATES	CaCO ₃	mg/L		SPECIFIC CONDUCTANCE	msiemens/cm		840.
HYDROXIDES	CaCO ₃	mg/L		TOTAL DISSOLVED SOLIDS		mg/L	514.
M (Total) ALKALINITY	CaCO ₃	mg/L	300.	SUSPENDED MATTER			ABS.
TOTAL HARDNESS	CaCO ₃	mg/L	454.	BIOLOGICAL GROWTHS	TOTAL BACTERIA COLONIES/ML		ABS.
SULFATE	SO ₄	mg/L		SPECIFIC GRAVITY @ 15.5°/15.5°C			
SILICA	SiO ₂	mg/L	9.5	FREEZING POINT			
IRON	Fe	mg/L	0.0	% BY WEIGHT			
CHLORIDE	NaCl	mg/L	58.				
ORGANIC INHIBITOR	PHOSPHONATE	mg/L					

RESULTS:

TREATMENT	TREATMENT CONTROL	FOUND	RECOMMENDED

REMARKS:

NOTES:

ANALYTICAL RESULTS EXPRESSED IN MILLIGRAMS PER LITRE (mg/L) ARE EQUIVALENT TO PARTS PER MILLION (ppm).
DIVIDE BY 17.1 TO OBTAIN GRAINS PER GALLON (gpgl).

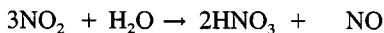
CYCLES OF CONCENTRATION = CHLORIDES IN SAMPLE/CHLORIDES IN MAKEUP

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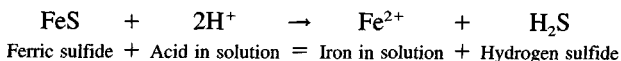
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FIGURE 8.5.10 Water analysis of Yellow Springs, OH (wells). (Courtesy of The Metropolitan Refining Co., Inc.)



Nitrogen + Water = Nitric acid + Nitric oxide

Hydrogen Sulfide. The odor typical of rotten eggs which is found in some water is due to the presence of hydrogen sulfide. This gas comes from decaying organic matter and from sulfur deposits. Hydrogen sulfide forms when acidic water reacts with sulfide minerals such as pyrite, an iron sulfide commonly called “fool’s gold”:



Hydrogen sulfide reacts with water to form hydrosulfuric acid, a slightly acidic solution. Its presence in water is also due to the decomposition of organic matter and protein which contain sulfur. Hydrogen sulfide is also a constituent of sewer gas, marsh gas, and coal gas. It can be present in water and also comes from these sources. Because of its acidic reaction in water, hydrogen sulfide is very corrosive and must be removed or neutralized.

8.5.3.4 Dissolved Minerals

Alkalinity. Alkalinity is the quantity of dissolved alkaline earth minerals expressed as calcium carbonate. It is the measured carbonate and bicarbonate minerals calculated as calcium carbonate since that is the primary alkaline earth mineral contributing to alkalinity. Alkalinity is also measured and calculated as the hydroxide when that is present. All natural waters contain some quantity of alkalinity. It contributes to scale formation because its presence encourages deposition of calcium carbonate, or lime scale.

pH Value. The *quality* of alkalinity, or the measure of the relative strength of acidity or alkalinity of a water, is the pH value, a value calculated from the hydrogen-ion concentration in water. The pH scale ranges from 0 to 14. A pH of 7.0 is neutral. It indicates a balance between the acidity and alkalinity. As the pH decreases to zero, the alkalinity decreases and the acidity increases. As the pH increases to 14, the alkalinity increases and the acidity decreases.

The pH scale (Fig. 8.5.11) is used to express the strength or intensity of the acidity or alkalinity of a water solution. This scale is logarithmic so that a pH change of 1 unit represents a tenfold increase or decrease in the strength of acidity or alkalinity. Hence water with a pH value of 4.0 is 100 times more acid in strength than water with a pH value of 6.0. Water is corrosive if the pH value is on the acidic side. It will tend to be scale-forming if the pH value is alkaline.

Hardness. Hardness is the total calcium, magnesium, iron, and trace amounts of other metallic elements in water which contribute to the hard feel of water. Hardness is also calculated as calcium carbonate, because it is the primary component contributing to hardness. Hardness causes lime deposits or scale in equipment.

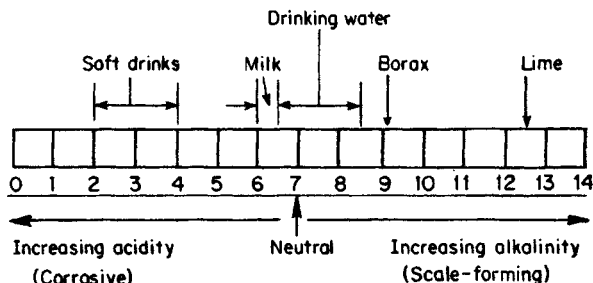


FIGURE 8.5.11 The pH scale.

Silica. Silica is dissolved sand or silica-bearing rock such as quartz through which the water flows. Silica is the cause of very hard and tenacious scales that can form in heat-transfer equipment. It is present dissolved in water as silicate or suspended in very fine, invisible form as colloidal silica.

Iron, Manganese, and Alumina. Iron, manganese, and alumina are dissolved or suspended metallic elements present in water supplies in varying quantities. They are objectionable because they contribute to a flat metallic taste and form deposits. These soluble metals, when they react with oxygen in water exposed to the atmosphere, form oxides which precipitate and cause cloudiness, or “red water.” This red color, particularly from iron, causes staining of plumbing fixtures, sinks, and porcelain china and is a cause of common laundry discoloration.

Chlorides. Chlorides are the sum total of the dissolved chloride salts of sodium, potassium, calcium, and magnesium present in water. Sodium chloride, which is common salt, and calcium chloride are the most common of the chloride minerals found in water. Chlorides do not ordinarily contribute to scale since they are very soluble. Chlorides are corrosive, however, and cause excessive corrosion when present in large volume, as in seawater.

Sulfates. Sulfates are the dissolved sulfate salts of sodium, potassium, calcium, and magnesium in the water. They are present due to dissolution of sulfate-bearing rock such as gypsum. Calcium and magnesium sulfate scale is very hard and difficult to remove and greatly interferes with heat transfer.

Total Dissolved Solids. The total dissolved solids (TDS) reported in water analyses are the sum of dissolved minerals including the carbonates, chlorides, sulfates, and all others that are present. The dissolved solids contribute to both scale formation and corrosion in heat-transfer equipment.

Suspended Matter. Suspended matter is finely divided organic and inorganic substances found in water. It is caused by clay silt and microscopic organisms which are dispersed throughout the water, giving it a cloudy appearance. The measure of suspended matter is turbidity. Turbidity is determined by the intensity of light scattered by the suspended matter in the water.

8.5.4 CORROSION

Corrosion is the process whereby a metal through reaction with its environment undergoes a change from the pure metal to its corresponding oxide or other stable combination. Usually, through corrosion, the metal reverts to its naturally occurring state, the ore. For example, iron is gradually dissolved by water and oxidized by oxygen in the water, forming the oxidation product iron oxide, commonly called rust.

This process occurs very rapidly in heat-transfer equipment because of the presence of heat, corrosive gases and dissolved minerals in the water, which stimulate the corrosion process.

The most common forms of corrosion found in heat-transfer equipment are

- General corrosion

- Oxygen pitting
- Galvanic corrosion
- Concentration cell corrosion
- Stress corrosion
- Erosion-corrosion
- Condensate grooving
- Microbiologically influenced corrosion (MIC)

8.5.4.1 General Corrosion

General corrosion is found in various forms in heat-transfer equipment. In a condenser water or cooling tower circuit, it can be seen as an overall deterioration of the metal surface with an accumulation of rust and corrosion products in the piping and water boxes. On copper condenser tubes, it is observed most frequently as a surface gouging or a uniform thinning of the tube metal.

In boilers, general corrosion is observed in the total overall disintegration of the tube metal surface in contact with the boiler water. (See Figs. 8.5.12 and 8.5.13.)

General corrosion occurs when the process takes place over the entire surface of the metal, resulting in a uniform loss of metal rather than a localized type of attack. It is often, but not always, accompanied by an accumulation of corrosion products over the surface of the metal (Fig. 8.5.14).

Iron and other metals are corroded by the metal going into solution in the water. It is necessary, therefore, to limit corrosion of these metals by reducing the activity of both hydroxyl ions and hydrogen ions, i.e., by maintaining a neutral environment.

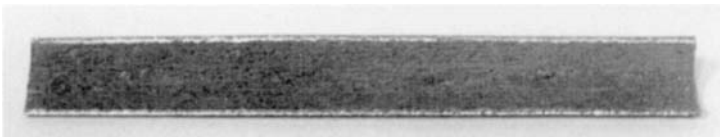


FIGURE 8.5.12 General corrosion on condenser tube. (Courtesy of The Metro Group, Inc.)

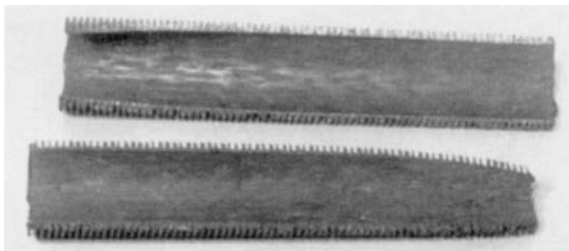


FIGURE 8.5.13 Pitting corrosion on condenser tubes. (Courtesy of The Metro Group, Inc.)

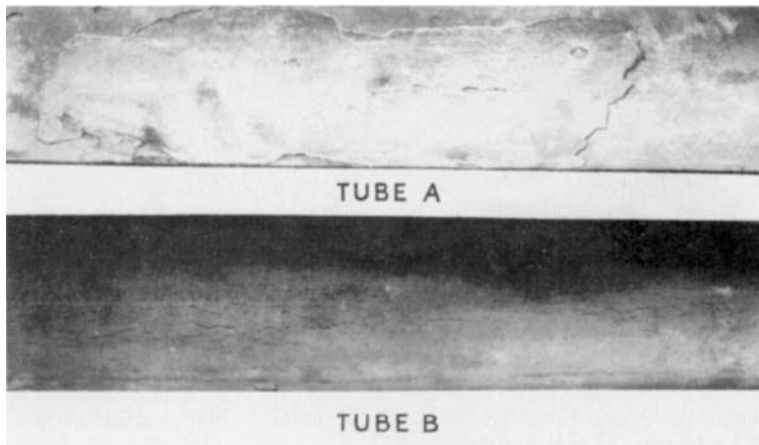


FIGURE 8.5.14 Boiler tube corrosion. (Courtesy of Babcock & Wilcox Co.)

Another important factor in the corrosion process is dissolved oxygen. The evolution of hydrogen gas in these reactions tends to slow the rate of the corrosion reaction and indeed, in many instances, to stop it altogether by forming an inhibiting film on the surface of the metal which physically protects the metal from the water.

Accumulation of rust and corrosion products is further promoted by the presence of dissolved oxygen. Oxygen reacts with the dissolved metal, eventually forming the oxide which is insoluble and in the case of iron builds up a voluminous deposit of rust. Since the role of dissolved oxygen in the corrosion process is important, removal of dissolved oxygen is an effective procedure in preventing corrosion.

8.5.4.2 Oxygen Pitting

The second type of corrosion frequently encountered in heat-transfer equipment is pitting. Pitting is characterized by deep penetration of the metal at a small area on the surface with no apparent attack over the entire surface as in general corrosion.

The corrosion takes place at a particular location on the surface, and corrosion products frequently accumulate over the pit. These appear as a blister, tubercle, or carbuncle, as in Fig. 8.5.15.

Oxygen pitting is caused by dissolved oxygen. It differs from localized pitting due to other causes, such as deposits of foreign matter, which is discussed in Sec. 8.5.4.4. Following are examples of pitting caused by dissolved oxygen (Figs. 8.5.16 and 8.5.17).

Oxygen pitting occurs in steam boiler systems where the feedwater contains dissolved oxygen. The pitting is found on boiler tubes adjacent to the feedwater entrance, throughout the boiler, or in the boiler feedwater line itself.

One of the most unexpected forms of oxygen pitting is commonly found in boiler feedwater lines following a deaerator. It is mistakenly believed that mechanically deaerated boiler feedwater will completely prevent oxygen pitting. However, quite to the contrary, water with a low concentration of dissolved oxygen frequently is more corrosive than that with a higher dissolved oxygen content. This is dem-

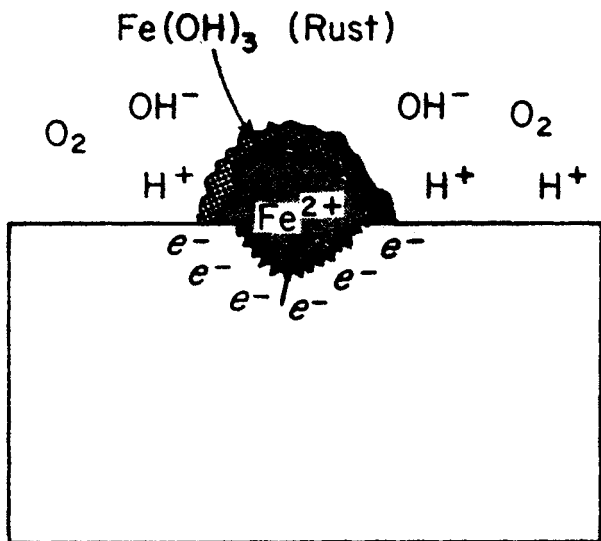
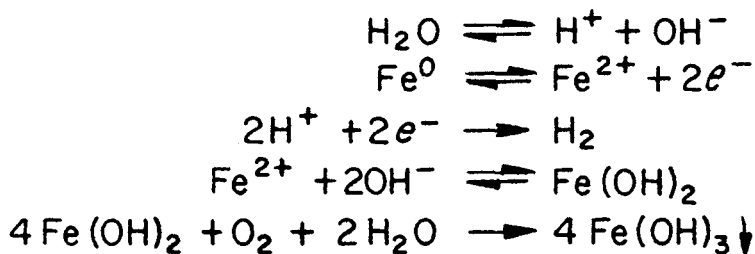


FIGURE 8.5.15 Reactions forming blisters over pit.

onstrated by the occurrence of oxygen pitting in boiler feedwater lines carrying deaerated water.

Mechanical deaerators are not perfect, and none can produce a feedwater with zero oxygen. The lowest guaranteed dissolved oxygen content that deaerators produce is $0.0005 \text{ cm}^3/\text{L}$. This trace quantity of dissolved oxygen is sufficient to cause severe pitting in feedwater lines or in boiler tubes adjacent to the feedwater entrance. This form of pitting is characterized by deep holes scattered over the surface of the pipe interior with little or no accumulation of corrosion products or rust, since there is insufficient oxygen in the environment to form the ferric oxide rust. (See Fig. 8.5.18.)

8.5.4.3 Galvanic Corrosion

Corrosion can occur when different metals come in contact with one another in water. When this happens, an electric current is generated similar to that of a storage



FIGURE 8.5.16 Pitting on boiler tube.
(Courtesy of The Metro Group, Inc.)



FIGURE 8.5.17 Blisters over pits on boiler tubes. (Courtesy of Babcock & Wilcox Co.)



FIGURE 8.5.18 Pitting in boiler feedwater line. (Courtesy of the Metro Group, Inc.)

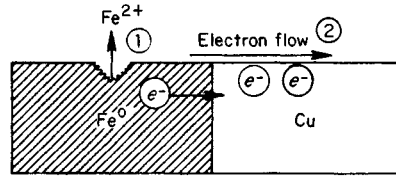


FIGURE 8.5.19 Galvanic corrosion caused by dissimilar-metal couple. (1) Iron going into solution loses two electrons: $\text{Fe}^0 \rightarrow \text{Fe}^{2+} + 2e^-$; (2) electrons flow to copper, the less reactive metal.

battery. The more active metal will tend to dissolve in the water, thereby generating an electric current (an electron flow) from the less active metal. This current is developed by a coupling of iron and copper, as in Fig. 8.5.19.

This tendency of a metal to give up electrons and go into solution is called the “electrode potential.” This potential varies greatly among metals since the tendency of different metals to dissolve and react with the environment varies.

In galvanic corrosion, commonly called “dissimilar-metal corrosion,” there are four essential elements:

1. A more reactive metal called the “anode”
2. A less reactive metal called the “cathode”
3. A water solution environment called the “electrolyte”
4. Contact between the two metals to facilitate electron flow

The rate of galvanic corrosion is strongly influenced by the electrode potential difference between the dissimilar metals. The galvanic series is a list of metals in order of their activity, the most active being at the top of the list and the least active at the bottom. *The farther apart two metals are on this list, the greater will be the reactivity between them and, therefore, the faster the anodic end will corrode.* The galvanic series is shown in Fig. 8.5.20.

If one or more of these four essential elements are eliminated, the corrosion reactions will be disrupted and the rate of corrosion slowed or halted altogether.

One method of preventing this type of corrosion is to eliminate contact of dissimilar metals in HVAC equipment by using insulating couplings or joints, such as a dielectric coupling which interferes with the electron flow from one metal to the other. Other forms of protection involve the removal of dissolved oxygen and use of protective coatings and inhibitors which provide a barrier between the corroding metal and its environment.

<i>Corroded end (anodic, or least noble)</i>
Magnesium alloys (1)
Zinc (1)
Beryllium
Aluminum alloys (1)
Cadmium
Mild steel, wrought iron
Cast iron, flake or ductile
Low-alloy high-strength steel
Nickel-resist, types 1 & 2
Naval bronze (CA464), yellow bronze (CA268), aluminum bronze (CA687), Red bronze (CA230), Admiralty bronze (CA443) manganese bronze
Tin
Copper (CA102, 110), silicon bronze (CA655)
Lead-tin solder
Tin bronze (G & M)
Stainless steel, 12–14% chromium (AISI Types 410, 416)
Nickel silver (CA 732, 735, 745, 752, 764, 770, 794)
90/10 Copper-nickel (CA 706)
80/20 Copper-nickel (CA 710)
Stainless steel, 16–18% chromium (AISI Type 430)
Lead
70/30 Copper-nickel (CA 715)
Nickel-aluminum bronze
Inconel* alloy 600
Silver braze alloys
Nickel 200
Silver
Stainless steel, 18 chromium, 8 nickel (AISI Types 302, 304, 321, 347)
Monel* Alloys 400, K-500
Stainless steel, 18 chromium, 12 nickel-molybdenum (AISI Types 316, 317)
Carpenter 20† stainless steel, Incoloy* Alloy 825
Titanium, Hastelloy‡ alloys C & C 276, Inconel* alloy 625
Graphite, graphitized cast iron
<i>Protected end (cathodic, or most noble)</i>

* International Nickel Trademark.

† Union Carbide Corp. Trademark.

‡ The Carpenter Steel Co. Trademark.

FIGURE 8.5.20 Galvanic Series.

8.5.4.4 Concentration Cell Corrosion

Concentration cell corrosion is a form of pitting corrosion that is a localized type of corrosion rather than a uniform attack. It is frequently called “deposit corrosion” or “crevice corrosion” since it occurs under deposits or at crevices of a metal joint.

Deposits of foreign matter, dirt, organic matter, corrosion products, scale, or any substance on a metal surface can initiate a corrosion reaction as a result of differences in the environment over the metal surface. Such differences may either be differences of solution ion concentration or dissolved oxygen concentration.

With concentration cell corrosion, the corrosion reaction proceeds as in galvanic corrosion since this differential also forms an electrode potential difference. This can best be prevented by maintaining clean surfaces.

8.5.4.5 Stress Corrosion

Stress corrosion is a combination of exposure of a metal to a corrosive environment and application of stress on the metal. It is frequently seen on condenser tubes and boiler tubes in the area where the tubes are rolled into the tube sheets. In steam boilers, stress corrosion has been referred to as “necking and grooving.” It is seen as a circumferential groove around the outside of a firetube where it enters the tube sheet. Figure 8.5.21 shows this type of corrosion.

The corrosion failure is a result of a corrosive environment and stresses and strains at the point of failure. Usually it occurs at the hottest end of the tube at the beginning of the first pass against the firewall. It concentrates at the tube end because of strains from two sources. First, when tubes are rolled in, stresses are placed on the metal, expanding the metal to fit the tube sheet. Second, when a boiler is fired, the heat causes rapid expansion of the tube, and consequently strains are greatest at the tube ends, which are fixed in the tube sheets. This actually causes a flexing and bowing of the tube, and sometimes the expansion is so severe that the tubes loosen in the sheets. During this bending of the tube, the natural protective iron oxide film forming at the tube ends tends to tear or flake off, exposing fresh steel to further attack. Eventually, the tube fails due to both corrosion and stress.

Stress corrosion can also occur on condenser tubes and heat-exchanger tubes from heat expansion that causes stresses in the metal at tube supports or tube sheets. This problem is reduced by more gradual firing practices in boilers, which allow more gradual temperature changes, and by using proper inhibitors to correct the corrosive environment.

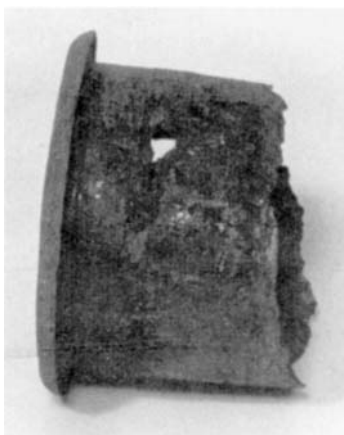


FIGURE 8.5.21 Necking and grooving on boiler firetube. (Courtesy of The Metro Group, Inc.)

8.5.4.6 Erosion-Corrosion

“Erosion-corrosion” is the gradual wearing away of a metal surface by both corrosion and abrasion. It is also commonly called “impingement corrosion.”

Water moving rapidly through piping can contain entrained air bubbles and suspended matter, sand, or other hard particulates. This is not uncommon in cooling tower waters where such particles are washed from the atmosphere. These abrasive particles remove natural protective oxide films present on the surface of the metal and cause general corrosion of the exposed metal. The higher the velocity of the impinging stream, the greater the rate of erosion-corrosion.

8.5.4.7 Condensate Grooving

Condensate grooving is a particular phenomenon of steam condensate line corrosion in HVAC equipment. It is found in steam condensate piping on all types of equipment, heat exchangers, steam-turbine condensers, unit heaters, steam absorption condensers, radiators, or any type of unit utilizing steam as a heat-transfer medium.

Condensate grooving is a direct chemical attack by the steam condensate on the metal over which it flows and is identified by the typical grooves found at the bottom of the pipe carrying the condensate. This is shown in Fig. 8.5.22.

The primary cause of condensate grooving is carbon dioxide. The dissolved carbon dioxide forms a mild carbonic acid. The methods available to prevent this type of corrosion include removal of bicarbonate and carbonate alkalinity from the boiler makeup water (dealkalinization) and use of carbonic acid neutralizers and filming inhibitors.

8.5.4.8 Microbiologically Influenced Corrosion (MIC)

Since the early 1980s the phenomenon of Microbiologically Influenced Corrosion (MIC) has become as a very serious problem in building HVAC recirculating water systems. MIC is the term given to corrosion involving the reaction of microbiological species with metals. It is corrosion caused or influenced by microbiological organisms or organic growths on metals.

There are many forms and mechanisms of MIC involving many types of microbiological organisms. The basic cause of MIC found in recirculating water systems are as follows:

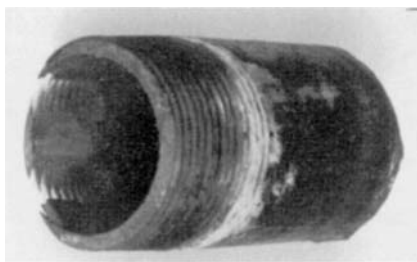


FIGURE 8.5.22 Steam condensate line corrosion. (Courtesy of The Metro Group, Inc.)

- Iron Related Bacteria (IRB)
- Sulfate Reducing Bacteria (SRB)
- Acid Producing Bacteria (APB)
- Biological Deposits

Iron Related Bacteria. A major group of organisms that are a direct cause of corrosion of iron and steel in recirculating water systems is the *iron related bacteria* (IRB). This class of organisms is responsible for causing corrosion of iron and steel by direct metabolism of iron. Some of these organisms actually consume iron by using it in their metabolic process and then deposit it in the form of hydrated ferric hydroxide along with the mucous secretions.

Sulfate Reducing Bacteria. The best known group of organisms involved in MIC are the Sulfate Reducing Bacteria (SRB). This group of organisms basically falls into three kinds, the *Desulfovibrio*, *Desulfotomaculum*, and *Desulfomonas* genera of organisms all of which metabolize sulfur in one form or another. All are anaerobic, which live without oxygen. The most widely known organism is the *Desulfovibrio*.

Acid Producing Bacteria. Another group of bacteria which cause MIC is the Acid Producing Bacteria (APB). There are many types of APB most of which are the slime forming bacteria such as *Pseudomonas*, *Aerobacter*, and *Bacillus* types which exude various organic acids in their metabolic process. Organic acids such as formic acid, acetic acid and oxalic acid have been identified in deposits of slime containing APB. These organic acids cause low pH conditions at local sites resulting in corrosion at these sites.

One APB that is commonly responsible for MIC is the *Thiobacillus*. These organisms oxidize sulfur compounds forming sulfuric acid which is extremely corrosive.

Biological Deposits. MIC can also be caused by other forms of organic growths such as algae, yeast, molds, and fungus along with bacterial slimes. Even in the absence of specific corrosive organisms such as the IRB, SRB or APB biological deposits provide the environment for corrosion through establishment of concentration cells resulting in under deposit corrosion. Biological deposits in general act as traps and food for other organisms resulting in rapid growth. This complex matrix sets up a corrosion potential between adjacent areas of a metal surface that may have a different type of deposit.

To control MIC it is important to understand the processes that cause it and therefore understand how to prevent it. It is clear that an essential control program will include control of all types of biological growths in recirculating water systems.

8.5.5 SCALE AND SLUDGE DEPOSITS

The most common and costly water-caused problem encountered in HVAC equipment is scale formation. The high cost of scale formation stems from the significant interference with heat transfer caused by water mineral scale deposits.

8.5.5.1 Mineral Scale and Pipe Scale

At this point, we should differentiate between mineral scale and pipe scale. *Mineral scale* is formed by deposits of the more insoluble minerals present in water, the heat-transfer medium (Fig. 8.5.23). *Pipe scale* (Fig. 8.5.24) is the natural iron oxide coating or corrosion products that form on the interior of piping which flake off and appear as a scale.



FIGURE 8.5.23 Pipe scale and iron corrosion products. (Courtesy of The Metro Group, Inc.)



FIGURE 8.5.24 Mineral scale deposits of water minerals. (Courtesy of The Metro Group, Inc.)

Mineral scale in steam boilers, heat exchangers, and condensers consists primarily of calcium carbonate, the least soluble of the minerals in water. Other scale components, in decreasing order of occurrence, are calcium sulfate, magnesium carbonate, iron, silica, and manganese. Present also in some scales are the hydroxides of calcium, magnesium, and iron as well as the phosphates of these minerals, where phosphates and alkalinity are used as a corrosion or scale inhibitor. Sludge is a softer form of scale and results when hard-water minerals reacting with phosphate and alkaline treatments forming a soft, pastelike substance rather than a hard, dense material. In most cases, scales contain a complex mixture of mineral salts because scale forms gradually and deposits the different minerals in a variety of forms.

The major cause of mineral scale is the inverse solubility of calcium and magnesium salts. Most salts or soluble substances, such as table salt or sugar, are more soluble in hot water than in cold.

Calcium and magnesium salts, however, dissolve more readily and in greater quantity in cold water than in hot, hence inverse solubility. This unique property is responsible for the entire problem of mineral scale on heat-transfer surfaces in HVAC equipment. From this property alone, we can readily understand why mineral scale forms on hot-water generator tubes, condenser tubes, boiler tubes, etc. It is simply the fact that the hottest surface in contact with the water is the tube surface of this type of equipment.

In condenser water systems using recirculating cooling tower water or once-through cooling water, the water temperature is much lower than that in steam boiler or hot-water systems. At these lower temperatures most of the scale-forming minerals will remain in solution, but the tendency will be to deposit calcium carbonate on the heat-transfer surfaces where there is a slight rise in temperature.

The primary factors which affect this tendency are:

- Alkalinity
- Hardness
- pH
- Total dissolved solids

The higher the alkalinity of a water, the higher the bicarbonate and/or carbonate content. As these minerals approach saturation, they tend to come out of solution.

Likewise, a higher concentration of hardness will increase the tendency of calcium and magnesium salts to come out of solution. The pH value reflects the ratio of carbonate to bicarbonate alkalinity. The higher the pH value, the greater the carbonate content of the water. Since calcium carbonate and magnesium carbonate are less soluble than the bicarbonate, they will tend to precipitate as the pH value and carbonate content increase.

Also affecting this tendency are the total dissolved solids and temperature. The higher the solids content, the greater the tendency to precipitate the least soluble of these solids. The higher the temperature, the greater the tendency to precipitate the calcium and magnesium salts because of their property of inverse solubility.

8.5.5.2 Langelier Index

The Langelier index is a calcium carbonate saturation index that is very useful in determining the scaling or corrosive tendencies of a water. It is based on the assumption that a water with a scaling tendency will tend to deposit a corrosion-

inhibiting film of calcium carbonate and hence will be less corrosive, whereas a water with a nonscaling tendency will tend to dissolve protective films and be more corrosive. This is not entirely accurate since other factors are involved in corrosion, as we have seen in Sec. 8.5.4 on corrosion, but it is an extremely valuable index in determining a tendency of a water.

In the 1950s, Eskell Nordell arranged five basic variables into an easy-to-use chart to quickly determine the pH of saturation of calcium carbonate and the Langelier index.³ This index is based on the pH of saturation of calcium carbonate.

The pH of saturation of calcium carbonate is the theoretical pH value of a particular water if that water is saturated with calcium carbonate. As the actual pH of a recirculating water approaches or even exceeds the pH of saturation of calcium carbonate, the tendency is to form a scale of calcium carbonate. If the actual pH is well below the pH of saturation of calcium carbonate, the tendency is to dissolve minerals and therefore to be corrosive. The Langelier index, therefore, is determined by comparing the actual pH of a recirculating water with the pH of saturation of calcium carbonate.

To determine the Langelier index, the actual pH of the water must be measured, and the pH of saturation of calcium carbonate, called the pHs, is calculated from a measure of the total alkalinity, hardness, total dissolved solids, and temperature.

A useful shortcut calculation of pHs can be made for cold well or municipal water supplies that are used for once-through cooling or service water. The reason why this rapid calculation is valid is that these supplies are usually consistent in temperature [49 to 57°F (10 to 14°C)] and total dissolved solids (50 to 300 mg/L). If a water supply has these characteristics, the following formula can be used (see Fig. 8.5.25).

$$\text{pHs @ } 50^{\circ}\text{F (10}^{\circ}\text{C)} = 11.7 - (C + D)$$

Likewise for hot-water supplies at 140°F (60°C), a short-form calculation of the pH of saturation of calcium carbonate can be done with the following formula:

$$\text{pHs @ } 140^{\circ}\text{F (60}^{\circ}\text{C)} = 10.8 - (C + D)$$

Once the pH of saturation of calcium carbonate has been calculated, the Langelier saturation index (SI) can be determined from the formula

$$\text{SI} = \text{pH} - \text{pHs}$$

where pH = actual measured pH of the water and pHs = pH of saturation of calcium carbonate as calculated from Fig. 8.5.25. Figure 8.5.26 can also be used to determine the pH of saturation.

A positive index indicates scaling tendencies; a negative one, corrosion tendencies. A very handy guide in predicting the tendencies of a water by using the Langelier saturation index is shown in Table 8.5.6.

8.5.5.3 Ryznar Index

Another useful tool for determining the tendencies of a water is the Ryznar index. This index is also based on the pH of saturation of calcium carbonate and was intended to serve as a more accurate index of the extent of scaling or corrosion in addition to the tendency. This index is calculated as follows:

$$\text{Ryznar index} = 2(\text{pHs}) - \text{pH}$$

A		C		D	
Total solids (mg/L)	A	Calcium hardness (mg/L of CaCO ₃)	C	M Alkalinity (mg/L of CaCO ₃)	D
50-300	0.1				
400-1000	0.2				
B		10- 11	0.6	10- 11	1.0
		12- 13	0.7	12- 13	1.1
		14- 17	0.8	14- 17	1.2
Temperature °F °C	B	18- 22	0.9	18- 22	1.3
		23- 27	1.0	23- 27	1.4
		28- 34	1.1	28- 35	1.5
32- 34 (0- 1.1)	2.6	35- 43	1.2	36- 44	1.6
36- 42 (2.2- 5.5)	2.5	44- 55	1.3	45- 55	1.7
44- 48 (6.7- 8.9)	2.4	56- 69	1.4	56- 69	1.8
50- 56 (10.0-13.3)	2.3	70- 87	1.5	70- 88	1.9
58- 62 (14.4-16.7)	2.2	88- 110	1.6	89- 110	2.0
64- 70 (17.8-21.1)	2.1	111- 138	1.7	111- 139	2.1
72- 80 (22.2-26.7)	2.0	139- 174	1.8	140- 176	2.2
82- 88 (27.8-31.1)	1.9	175- 220	1.9	177- 220	2.3
90- 98 (27.8-31.1)	1.8	230- 270	2.0	230- 270	2.4
100-110 (37.8-43.3)	1.7	280- 340	2.1	280- 350	2.5
112-122 (44.4-50.0)	1.6	350- 430	2.2	360- 440	2.6
124-132 (51.1-55.6)	1.5	440- 550	2.3	450- 550	2.7
134-142 (56.7-63.3)	1.4	560- 690	2.4	560- 690	2.8
148-160 (64.4-71.1)	1.3	700- 870	2.5	700- 880	2.9
162-178 (72.2-81.1)	1.2	800-1000	2.6	890-1000	3.0

$$\text{pH}_s = (9.3 + A + B) - (C + D)$$

$$\text{SI} = \text{pH} - \text{pH}_s$$

If index is 0, water is in chemical balance.

If index is positive, scale-forming tendencies are indicated.

If index is negative, corrosive tendencies are indicated.

FIGURE 8.5.25 Data for calculations of the pH of saturation of calcium carbonate. (From Eskell Nordell, *Water Treatment for Industrial and Other Uses*, 2d ed., © 1961 by Litton Educational Publishing Inc., reprinted with permission of Van Nostrand Reinhold Co.)

where pH_s = pH of saturation of calcium carbonate, as calculated from Fig. 8.5.25, and pH = actual measured pH of the water. Table 8.5.7 can be used to determine the tendency and extent of corrosion or scaling with the Ryznar index.

Let us see how these indices can help us in analyzing a particular water supply. Figure 8.5.8 depicts an analysis report on the Washington, DC, water supply. The Langlier saturation index at 50°F (10°C) is determined by using this analysis and the data shown on Fig. 8.5.25 as follows:

$$\begin{aligned}\text{pH}_s &= 9.3 + A + B - (C + D) \\ &= 9.3 + 0.1 + 2.3 - (1.8 + 2.0) \\ &= 8.2\end{aligned}$$

and

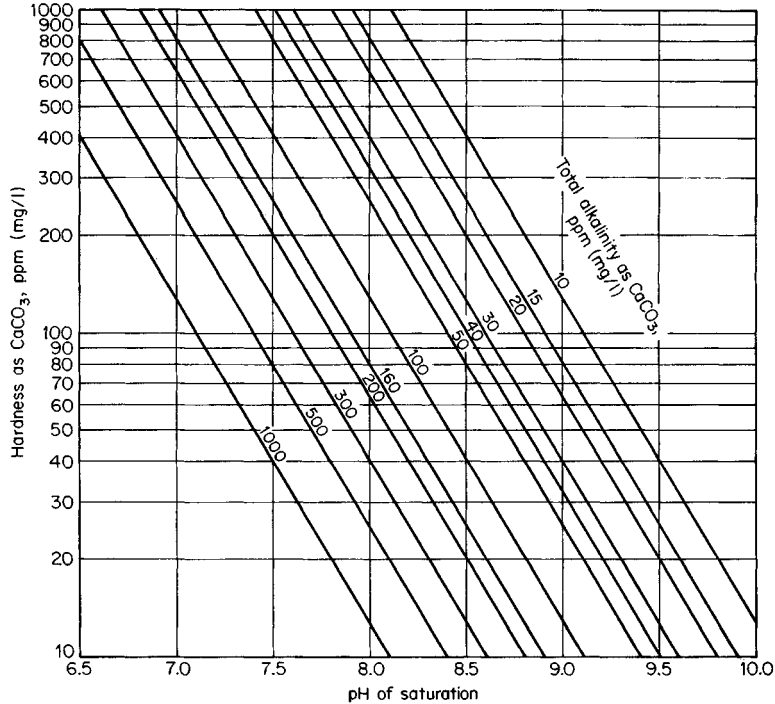


FIGURE 8.5.26 The pH of saturation for waters 49 to 57°F (10 to 14Ψ) and total dissolved solids of 50 to 300 mg/L.

TABLE 8.5.6 Prediction of Water Tendencies by the Langelier Index

Langelier saturation index	Tendency of water
2.0	Scale-forming and for practical purposes noncorrosive
0.5	Slightly corrosive and scale-forming
0.0	Balanced, but pitting corrosion possible
-0.5	Slightly corrosive and non-scale-forming
-2.0	Serious corrosion

Source: *Carrier System Design Manual*, part 5, “Water Conditioning,” Carrier Corp., Syracuse, NY, 1972, p. 5-12.

$SI = pH - pHs = 7.7 - 8.2 = -0.5$

From Table 8.5.6, according to the Langelier saturation index this water supply is somewhat more than “slightly corrosive and non-scale-forming.”

To learn more about this water, the Ryznar index (RI) can be calculated in the same manner:

$RI = 2(pHs) - pH = 16.4 - 7.7 = 8.7$

TABLE 8.5.7 Prediction of Water Tendencies by the Ryznar Index

Ryznar stability index	Tendency of water
4.0–5.0	Heavy scale
5.0–6.0	Light scale
6.0–7.0	Little scale or corrosion
7.0–7.5	Significant corrosion
7.5–9.0	Heavy corrosion
9.0+	Intolerable corrosion

Source: *Carrier System Design Manual*, part 5, "Water Conditioning," Carrier Corp., Syracuse, NY, 1972, p. 5–14.

According to Table 8.5.6, this water supply tendency indicates "heavy corrosion." The Ryznar index, being more quantitative, indicates that the degree of corrosion would be greater than we would anticipate from the tendency shown by the qualitative Langelier saturation index.

In an examination of a water supply, both the Langelier and the Ryznar indices are used to determine the scale-forming or corrosion tendencies.

In open cooling tower condenser water systems and steam boilers, however, there is a constant accumulation of minerals as a result of evaporation of pure water, such as distilled water, and makeup water containing the various mineral impurities. Therefore, in these systems the pH, concentration of hardness, total dissolved solids, and alkalinity are constantly changing, making a study of the Langelier and Ryznar indices relatively complex and subject to gross inaccuracies.

8.5.5.4 Boiler Scale

Scale in boilers is a direct result of precipitation of the calcium, magnesium, iron, and silica minerals present in the boiler feedwater. Scale can be prevented by removing a portion of the scale-forming ingredients prior to the boiler with external water-softening equipment or within the boiler itself with internal boiler water treatment.

One of the most troublesome deposits frequently encountered in steam boilers is iron and combinations of iron with calcium and phosphate used in boiler water treatment. These sticky, adherent sludge deposits are caused by excessive amounts of iron entering the boiler with the feedwater. The iron is in the form of iron oxide or iron carbonate corrosion products. It is a result of corrosion products from the sections prior to the boiler, such as steam and condensate lines, condensate receivers, deaerators, and boiler feedwater lines. A program for preventing scale deposits must include treatment to prevent this troublesome type of sludge deposit.

8.5.5.5 Condensate Scale

In recirculating cooling tower condenser water systems for air conditioning and refrigeration chillers, scale deposits are a direct result of precipitation of the carbonate, calcium sulfite, or silica minerals due to such an overconcentration of these minerals that their solubility or pH of saturation is exceeded and the minerals come

out of solution. Scale in this equipment can include foreign substances such as corrosion products, organic matter, and mud or dirt. These are usually called “foulants” rather than “scale.” Treatment to prevent mineral scale should, therefore, include sufficient dilution of the recirculating water to prevent the concentration of minerals from approaching the saturation point, pH control to prevent the pH from reaching the pH of saturation of calcium carbonate, and chemical treatments to inhibit and control scale crystal formation.

8.5.6 FOULANTS

In addition to water mineral scale, other deposits of mud, dirt, debris, foreign matter, and organic growth are a recurrent problem in recirculating water systems. Deposits of foreign matter plug narrow passages, interfere with heat transfer and foul heat-transfer surfaces, causing inefficient performance of the equipment and high energy consumption.

8.5.6.1 Mud, Dirt, and Clay

Open recirculating cooling tower systems are most subject to deposits of mud, dirt, and debris. A cooling tower is a natural air washer with water spraying over slats and tower fill washing the air blown through either naturally or assisted by fans.

Depending on the location, all sorts of airborne dust and debris end up in cooling tower recirculating water systems. These vary from fine dust particles to pollen, weeds, plant life, leaves, tree branches, grass, soil, and stones.

The fine particles of dust and dirt tend to collect and compact in the condenser water system, especially in areas of low circulation. At heat-transfer surfaces, the dust and dirt can deposit and compact into a sticky mud and seriously interfere with operating efficiency.

Muddy foulants are a common occurrence and form with the combination of airborne particles, corrosion products, scale, and organic matter. *Very rarely can one identify a foulant as a single compound because it is usually a complex combination of all these things.*

In closed recirculating water systems, foulants are not nearly as varied and complex as in open systems, but they are just as serious when they occur. Deposits in closed systems are usually caused by dirt or clay entering with the makeup water or residual construction debris. A break in an underground water line can result in dirt, sand, and organic matter being drawn into a system and is a common source of fouling.

Makeup water containing unusual turbidity or suspended matter is usually treated at the source by coagulation, clarification, and filtration so as to maintain its potability. Suspended matter and turbidity, therefore, are not common in makeup water in HVAC systems since the makeup water usually comes from a municipal or local source, over which there is a water authority responsible for delivery of clear, potable water.

Where a private well water, pond, or other nonpublic source of water is available for use as makeup water to recirculating water systems and boilers, it should be carefully examined for turbidity and suspended matter. The suspended matter measured as turbidity should be no more than the maximum of 1 turbidity unit for

drinking water recommended by the Environmental Protection Agency. When the supply is excessively turbid, some form of clarification such as coagulation, settling, filtration, and/or fine strainers should be used to remove the suspended matter and reduce the turbidity to below 1 unit.

The more common problem with suspended matter and turbidity results from makeup water that is temporarily or occasionally dirty. This may occur when the local water authority is cleaning sections of a distribution main or installing new mains or when water mains are cut into during some nearby construction project. This kind of work creates a disturbance of the water mains, causing settled and lightly adherent pipeline deposits to break off and be flushed into the water supply. These deposits consist mostly of iron oxide corrosion products and dirt, clay, or silt.

8.5.6.2 Black Mud and Mill Scale

One of the most common and difficult foulants found in closed systems is a black mud made up of compacted, fine, black magnetic iron oxide particles. This black mud not only deposits at heat-transfer surfaces, but also clogs or blocks narrow passages in unit heaters, fan-coil units, and cooling, reheat, and heating coils in air-handling units. This black mud is a result of wet very fine particles of black magnetic iron oxide being compacted into a dense adherent mud.

The interior of black iron piping, commonly used for recirculating water, has a natural black iron oxide protective coating ordinarily held intact by oil-based inhibitors used to coat the pipe to prevent corrosion during storage and layup. This natural iron oxide protective coating is called *mill scale*, a very general term which can be applied to any form of pipe scale or filings washed off the interior of the pipe. This mill scale film becomes disturbed and disrupted during construction due to the constant rough handling, cutting, threading, and necessary battering of the pipe. After construction, the recirculating water system is filled and flushed with water, which removes most of the loosened mill scale along with any other construction debris. However, very fine particles of magnetic iron oxide will continue to be washed off the metal surface during operation, and in many instances this washing persists for several years before it subsides. Mill scale plugging can be a serious problem. It is best alleviated in a new system by thorough cleaning and flushing with a strong, low-foaming detergent-dispersant cleaner. This, however, does not always solve the problem. Even after a good cleanout, gradual removal of mill scale during ensuing operation can continue.

8.5.6.3 Boiler Foulants

In steam boilers, foulants other than mineral scale usually consist of foreign contaminants present in the feedwater. These include oil, clay, contaminants from a process, iron corrosion products from the steam system, and construction debris in new boiler systems. Mud or sludge in a boiler is usually a result of scale-forming minerals combined with iron oxide corrosion products and treatment chemicals. Such foulants are controlled by using proper dispersants which prevent adherence on heat-transfer surfaces.

In heating boilers, the most frequent foulants other than sludge are oil and clay. Oil can enter a boiler system through leakage at oil lubricators, fuel oil preheaters, or steam heating coils in fuel oil storage tanks. When oil enters a boiler, it causes

priming and foaming by emulsifying with the alkaline boiler water. Priming is the bouncing of the water level that eventually cuts the boiler off at low water due to the very wide fluctuation of this level. Oil can also carbonize at hot boiler tubes, causing not only serious corrosion from concentration corrosion cells but also tube ruptures as a result of overheating due to insulating carbon deposits. Whenever oil enters a boiler system, it must be removed immediately to prevent these problems. This is easily done by boiling out with an alkaline detergent cleaner for boilers.

Clay is a less frequent foulant in boilers, but it, too, can form insulating deposits on tube surfaces. Clay enters a boiler with the boiler makeup water that is either turbid or contaminated with excessive alum, used as a coagulant in the clarification process. Clay can be dispersed with the use of dispersants in the internal treatment of the boiler, but makeup water should be clear and free of any turbidity before it is used as boiler feedwater. Where turbidity and clay are a constant problem, filtration of the boiler feedwater is in order.

8.5.6.4 Construction Debris

All new systems become fouled and contaminated with various forms of foreign matter during construction. It is not uncommon to find these in the interior of HVAC piping and heat exchangers: welding rods, beads, paper bags, plastic wrappings, soft drink can rings, pieces of tape, insulation wrappings, glass, and any other construction debris imaginable.

It is necessary not only to clean out construction debris from the interior of HVAC systems prior to initial operation, but also to clean the metal surfaces of oil and mill scale naturally present on the pipe interior. This oil and mill scale, as has been shown, can seriously foul and plug closed systems and cause boiler tube failures, if the oil is carbonized during firing. Every new recirculating water system and boiler must be cleaned thoroughly with a detergent-dispersant type of cleaner or, as in steam boilers, with an alkaline boilout compound. This initial cleanout will remove most of the foulants and prevent serious operational difficulties.

8.5.6.5 Organic Growths

Organic growths in HVAC equipment are usually found in open recirculating water systems such as cooling towers, air washers, and spray coil units. Occasionally closed systems become fouled with organic slimes due to foreign contamination. Open systems are constantly exposed to the atmosphere and environs which contain not only dust and dirt but also innumerable quantities of microscopic organisms and bacteria. Cooling tower waters, because they are exposed to sunlight, operate at ideal temperatures, contain mud as a medium and food in the form of inorganic and organic substances, and are a most favorable environment for the abundant growth of biological organisms. Likewise, air washers and spray coil units, as they wash dust and dirt from the atmosphere, collect microscopic organisms which tend to grow in the recirculating water due to the favorable environment. The organisms that grow in such systems consist primarily of algae, fungi, and bacterial slimes.

8.5.6.6 Algae

Algae are the most primitive form of plant life and together with fungus form the family of thallus plants. Algae are widely distributed throughout the world and

consist of many different forms. The forms found in open recirculating water systems are the blue-green algae, green algae, and brown algae. The blue-green algae, the simplest form of green plants, consist of a single cell and hence are called unicellular. Green algae are the largest group of algae and are either unicellular or multicellular. Brown algae are also large, plantlike organisms that are multicellular.

Large masses of algae can cause serious problems by blocking the air in cooling towers, plugging water distribution piping and screens, and accelerating corrosion by concentration cell corrosion and pitting. Algae must be removed physically before a system can be cleaned since the mass will provide a continuous source of material for reproduction and biocides will be consumed only at the surface of the mass, leaving the interior alive for further growth.

8.5.6.7 Fungi

Fungi are also a thallus plant similar to the unicellular and multicellular algae. They require air, water, and carbohydrates for growth. The source of carbohydrates can be any form of carbon. Fungi and algae can grow together; the algae living within the fungus mass are furnished with a moist, protected environment, while the fungus obtains carbohydrates from the algae.

8.5.6.8 Bacteria

Bacteria are microscopic unicellular living organisms that exhibit both plant and animal characteristics. They exist in rod-shaped, spiral and spherical forms. There are many thousands of strains of bacteria, and all recirculating waters contain some bacteria. The troublesome ones, however, are bacterial slimes, iron bacteria, sulfate-reducing bacteria, and pathogenic bacteria.

Pathogenic bacteria are disease-bearing bacteria. Cooling tower waters, having ideal conditions for the growth of bacteria and other organisms, can promote the growth of pathogenic bacteria. In isolated instances, pathogenic bacteria have been found growing in cooling tower waters. *Therefore, it is as important to keep these systems free of bacterial contamination, to inhibit growth of pathogenic bacteria, as it is to prevent growth of slime-forming and corrosion-promoting bacteria.*

8.5.7 PRETREATMENT EQUIPMENT

Prior to internal treatment of HVAC equipment, it is frequently necessary to use mechanical equipment to remove from the feedwater supply damaging impurities such as dissolved oxygen, excess hardness, or suspended solids.

The choice of proper equipment and its need can be determined by studying the quality and quantity of makeup water used in a boiler, condenser water system, and an open or a closed recirculating water system.

8.5.7.1 Water Softeners

Hardness in the makeup water is the cause of scale formation. In equipment using large volumes of a hard water, a substantial amount of scale can form on heat-

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