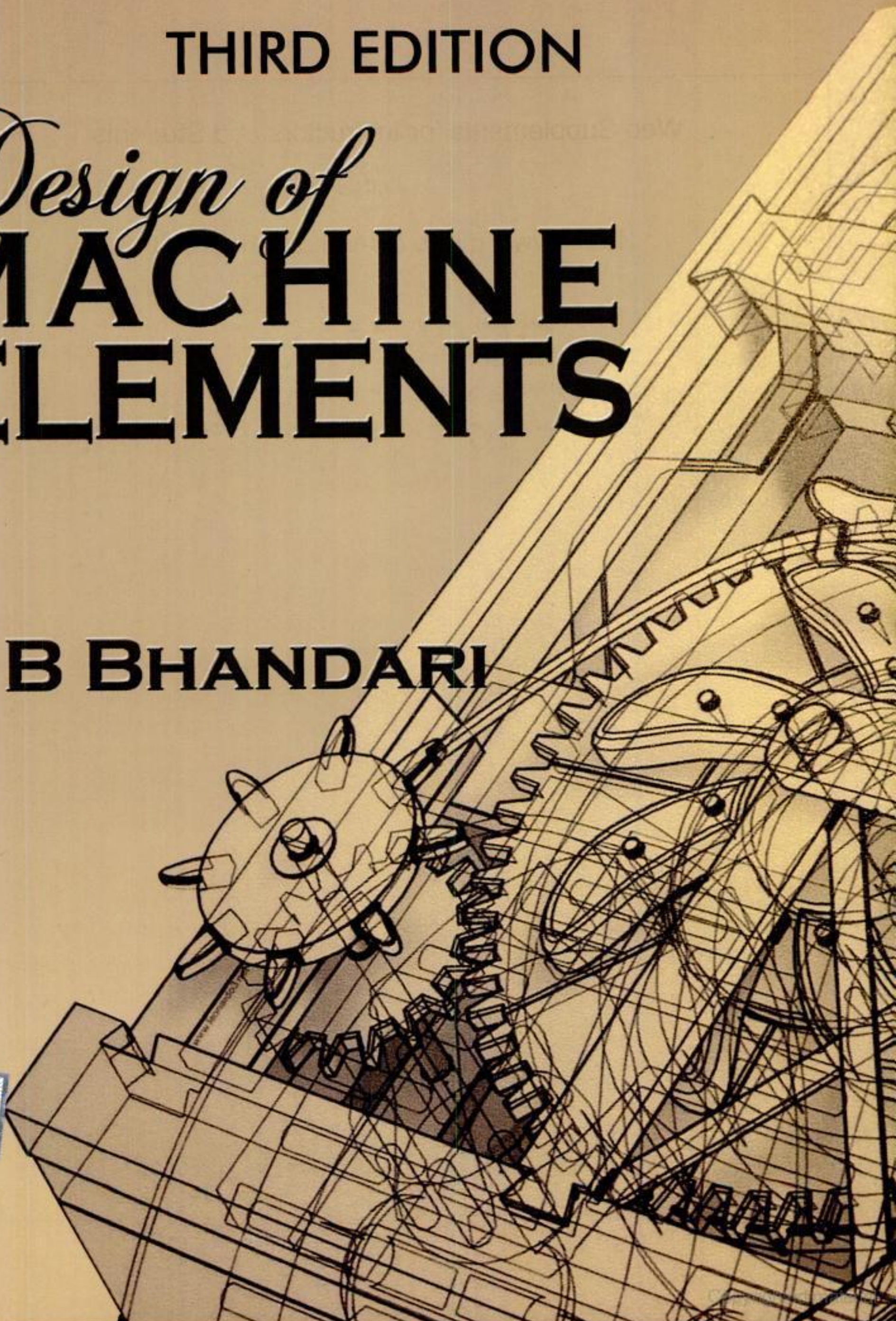
THIRD EDITION

Design of MACHINE ELEMENTS



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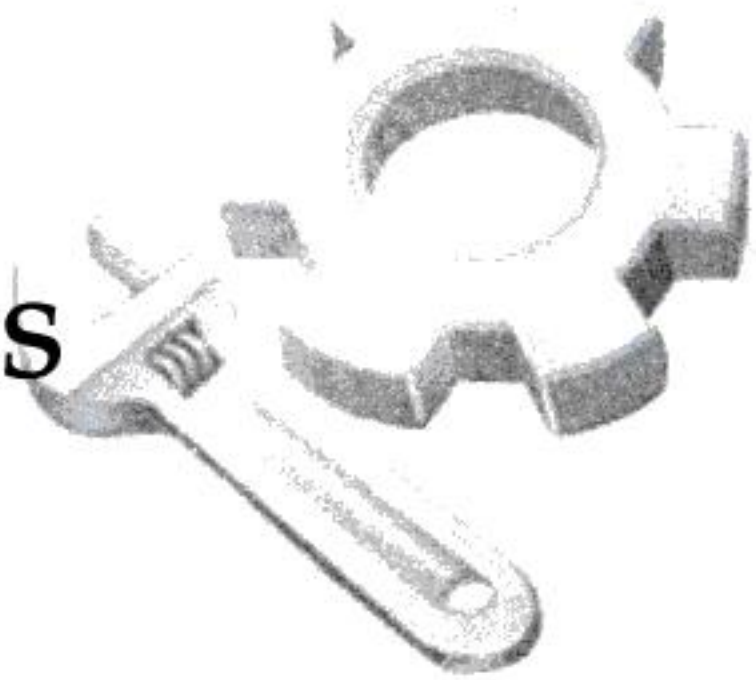

Preface xvii
Visual Walkthrough xxi

1. Introduction 1

1.1 Machine Design *1*
1.2 Basic Procedure of Machine Design *2*
1.3 Basic Requirements of Machine Elements *3*
1.4 Design of Machine Elements *4*
1.5 Traditional Design Methods *8*
1.6 Design Synthesis *8*
1.7 Use of Standards in Design *9*
1.8 Selection of Preferred Sizes *11*
1.9 Aesthetic Considerations in Design *14*
1.10 Ergonomic Considerations in Design *15*
1.11 Concurrent Engineering *17*
Short Answer Questions *19*
Problems for Practice *19*

2. Engineering Materials 20

2.1 Stress–Strain Diagrams *20*
2.2 Mechanical Properties of Engineering Materials *23*
2.3 Cast Iron *26*
2.4 BIS System of Designation of Steels *29*
2.5 Plain-carbon Steels *30*
2.6 Free-cutting Steels *32*
2.7 Alloy Steels *32*
2.8 Overseas Standards *34*
2.9 Heat Treatment of Steels *36*
2.10 Case Hardening of Steels *37*
2.11 Cast Steel *38*

This One
6FF3-8P5-RS1Y

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2.12 Aluminium Alloys 39
2.13 Copper Alloys 41
2.14 Die-casting Alloys 43
2.15 Ceramics 44
2.16 Plastics 45
2.17 Fibre-reinforced Plastics 48
2.18 Natural and Synthetic Rubbers 49
2.19 Creep 50
2.20 Selection of Material 51
2.21 Weighted Point Method 51
Short Answer Questions 53

3. Manufacturing Considerations in Design **55**

3.1 Selection of Manufacturing Method 55
3.2 Design Considerations of Castings 57
3.3 Design Considerations of Forgings 59
3.4 Design Considerations of Machined Parts 61
3.5 Hot and Cold Working of Metals 62
3.6 Design Considerations of Welded Assemblies 62
3.7 Design for Manufacture and Assembly (DFMA) 64
3.8 Tolerances 65
3.9 Types of Fits 66
3.10 BIS System of Fits and Tolerances 67
3.11 Selection of Fits 69
3.12 Tolerances and Manufacturing Methods 69
3.13 Selective Assembly 70
3.14 Tolerances For Bolt Spacing 72
3.15 Surface Roughness 73
Short Answer Questions 73
Problems for Practice 74

4. Design Against Static Load **76**

4.1 Modes of Failure 76
4.2 Factor of Safety 77
4.3 Stress–strain Relationship 79
4.4 Shear Stress and Shear Strain 80
4.5 Stresses Due To Bending Moment 81
4.6 Stresses Due To Torsional Moment 82
4.7 Eccentric Axial Loading 83
4.8 Design of Simple Machine Parts 84
4.9 Cotter Joint 85
4.10 Design Procedure for Cotter Joint 90
4.11 Knuckle Joint 94
4.12 Design Procedure for Knuckle Joint 99
4.13 Principal Stresses 104
4.14 Theories of Elastic Failure 106

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4.15 Maximum Principal Stress Theory *107*
4.16 Maximum Shear Stress Theory *108*
4.17 Distortion-Energy Theory *110*
4.18 Selection and Use of Failure Theories *112*
4.19 Levers *117*
4.20 Design of Levers *118*
4.21 Fracture Mechanics *128*
4.22 Curved Beams *130*
4.23 Thermal Stresses *135*
4.24 Residual Stresses *136*
Short Answer Questions *137*
Problems for Practice *138*

5. Design Against Fluctuating Load **141**

5.1 Stress Concentration *141*
5.2 Stress Concentration Factors *142*
5.3 Reduction of Stress Concentration *145*
5.4 Fluctuating Stresses *149*
5.5 Fatigue Failure *151*
5.6 Endurance Limit *152*
5.7 Low-cycle and High-cycle Fatigue *153*
5.8 Notch Sensitivity *154*
5.9 Endurance Limit—Approximate Estimation *155*
5.10 Reversed Stresses—Design for Finite and Infinite Life *159*
5.11 Cumulative Damage in Fatigue *166*
5.12 Soderberg and Goodman Lines *167*
5.13 Modified Goodman Diagrams *168*
5.14 Gerber Equation *174*
5.15 Fatigue Design under Combined Stresses *177*
5.16 Impact Stresses *180*
Short Answer Questions *182*
Problems for Practice *182*

6. Power Screws **184**

6.1 Power Screws *184*
6.2 Forms of Threads *185*
6.3 Multiple Threaded Screws *187*
6.4 Terminology of Power Screw *187*
6.5 Torque Requirement—Lifting Load *189*
6.6 Torque Requirement—Lowering Load *189*
6.7 Self-locking Screw *190*
6.8 Efficiency of Square Threaded Screw *190*
6.9 Efficiency of Self-locking Screw *192*
6.10 Trapezoidal and Acme Threads *192*
6.11 Collar Friction Torque *193*
6.12 Overall Efficiency *194*

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6.13 Coefficient of Friction *194*
6.14 Design of Screw and Nut *194*
6.15 Design of Screw Jack *206*
6.16 Differential and Compound Screws *214*
6.17 Recirculating Ball Screw *215*
Short-Answer Questions *216*
Problems for Practice *217*

7. Threaded Joints **219**

7.1 Threaded Joints *219*
7.2 Basic Types of Screw Fastening *220*
7.3 Cap Screws *222*
7.4 Setscrews *223*
7.5 Bolt of Uniform Strength *224*
7.6 Locking Devices *225*
7.7 Terminology of Screw Threads *227*
7.8 ISO Metric Screw Threads *228*
7.9 Materials and Manufacture *230*
7.10 Bolted Joint—Simple Analysis *231*
7.11 Eccentrically Loaded Bolted Joints in Shear *233*
7.12 Eccentric Load Perpendicular to Axis of Bolt *235*
7.13 Eccentric Load on Circular Base *242*
7.14 Torque Requirement for Bolt Tightening *248*
7.15 Dimensions of Fasteners *249*
7.16 Design of Turnbuckle *251*
7.17 Elastic Analysis of Bolted Joints *254*
7.18 Bolted Joint under Fluctuating Load *257*
Short-Answer Questions *269*
Problems for Practice *269*

8. Welded and Riveted Joints **272**

8.1 Welded Joints *272*
8.2 Welding Processes *273*
8.3 Stress Relieving of Welded Joints *274*
8.4 Butt Joints *274*
8.5 Fillet Joints *275*
8.6 Strength of Butt Welds *276*
8.7 Strength of Parallel Fillet Welds *277*
8.8 Strength of Transverse Fillet Welds *278*
8.9 Maximum Shear Stress in Parallel Fillet Weld *281*
8.10 Maximum Shear Stress in Transverse Fillet Weld *282*
8.11 Axially Loaded Unsymmetrical Welded Joints *284*
8.12 Eccentric Load in the Plane of Welds *285*
8.13 Welded Joint Subjected to Bending Moment *290*
8.14 Welded Joint Subjected to Torsional Moment *294*
8.15 Strength of Welded Joints *295*

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8.16 Welded Joints Subjected to Fluctuating Forces *296*
8.17 Welding Symbols *297*
8.18 Weld Inspection *298*
8.19 Riveted Joints *298*
8.20 Types of Rivet Heads *301*
8.21 Types of Riveted Joints *303*
8.22 Rivet Materials *305*
8.23 Types of Failure *306*
8.24 Strength Equations *306*
8.25 Efficiency of Joint *307*
8.26 Caulking and Fullering *307*
8.27 Longitudinal Butt Joint for Boiler Shell *311*
8.28 Circumferential Lap Joint for Boiler Shells *318*
8.29 Eccentrically Loaded Riveted Joint *321*
Short-Answer Questions *325*
Problems for Practice *325*

9. Shafts, Keys and Couplings **330**

9.1 Transmission Shafts *330*
9.2 Shaft Design on Strength Basis *331*
9.3 Shaft Design on Torsional Rigidity Basis *333*
9.4 ASME Code for Shaft Design *334*
9.5 Design of Hollow Shaft on Strength Basis *342*
9.6 Design of Hollow Shaft on Torsional Rigidity Basis *344*
9.7 Flexible Shafts *346*
9.8 Keys *346*
9.9 Saddle Keys *347*
9.10 Sunk Keys *348*
9.11 Feather Key *349*
9.12 Woodruff Key *350*
9.13 Design of Square and Flat Keys *350*
9.14 Design of Kennedy Key *352*
9.15 Splines *354*
9.16 Couplings *356*
9.17 Muff Coupling *357*
9.18 Design Procedure for Muff Coupling *357*
9.19 Clamp Coupling *359*
9.20 Design Procedure for Clamp Coupling *360*
9.21 Rigid Flange Couplings *362*
9.22 Design Procedure for Rigid Flange Coupling *364*
9.23 Bushed-pin Flexible Coupling *368*
9.24 Design Procedure for Flexible Coupling *371*
9.25 Design for Lateral Rigidity *376*
9.26 Castigliano's Theorem *380*

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9.27 Area Moment Method *382*
9.28 Graphical Integration Method *383*
9.29 Critical Speed of Shafts *385*
Short-Answer Questions *388*
Problems for Practice *389*

10. Springs **393**

10.1 Springs *393*
10.2 Types of Springs *393*
10.3 Terminology of Helical Springs *395*
10.4 Styles of End *396*
10.5 Stress and Deflection Equations *397*
10.6 Series and Parallel Connections *399*
10.7 Spring Materials *401*
10.8 Design of Helical Springs *403*
10.9 Spring Design—Trial-and-Error Method *405*
10.10 Design against Fluctuating Load *405*
10.11 Concentric Springs *425*
10.12 Optimum Design of Helical Spring *430*
10.13 Surge in Spring *432*
10.14 Helical Torsion Springs *433*
10.15 Spiral Springs *435*
10.16 Multi-Leaf Spring *437*
10.17 Nipping of Leaf Springs *439*
10.18 Belleville Spring *441*
10.19 Shot Peening *443*
Short-Answer Questions *443*
Problems for Practice *444*

11. Friction Clutches **448**

11.1 Clutches *448*
11.2 Torque Transmitting Capacity *450*
11.3 Multi-disk Clutches *456*
11.4 Friction Materials *459*
11.5 Cone Clutches *461*
11.6 Centrifugal Clutches *465*
11.7 Energy Equation *467*
11.8 Thermal Considerations *469*
Short-Answer Questions *470*
Problems for Practice *471*

12. Brakes **472**

12.1 Brakes *472*
12.2 Energy Equations *472*
12.3 Block Brake with Short Shoe *475*
12.4 Block Brake with Long Shoe *480*

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12.5 Pivoted Block Brake with Long Shoe *482*
12.6 Internal Expanding Brake *485*
12.7 Band Brakes *490*
12.8 Disk Brakes *493*
12.9 Thermal Considerations *496*
Short-Answer Questions *496*
Problems for Practice *497*

13. Belt Drives **499**

13.1 Belt Drives *499*
13.2 Belt Constructions *501*
13.3 Geometrical Relationships *503*
13.4 Analysis of Belt Tensions *504*
13.5 Condition for Maximum Power *507*
13.6 Condition for Maximum Power (Alternative Approach) *507*
13.7 Characteristics of Belt Drives *509*
13.8 Selection of Flat-belts from Manufacturer's Catalogue *514*
13.9 Pulleys for Flat Belts *517*
13.10 Arms of Cast-iron Pulley *520*
13.11 V-belts *522*
13.12 Selection of V-belts *534*
13.13 V-grooved Pulley *535*
13.14 Belt-Tensioning Methods *540*
13.15 Ribbed V-belts *540*
Short-Answer Questions *542*
Problems for Practice *542*

14. Chain Drives **544**

14.1 Chain Drives *544*
14.2 Roller Chains *546*
14.3 Geometric Relationships *548*
14.4 Polygonal Effect *549*
14.5 Power Rating of Roller Chains *549*
14.6 Sprocket Wheels *551*
14.7 Design of Chain Drive *553*
14.8 Chain Lubrication *555*
14.9 Silent Chain *562*
Short-Answer Questions *562*
Problems for Practice *563*

15. Rolling Contact Bearings **564**

15.1 Bearings *564*
15.2 Types of Rolling-contact Bearings *565*
15.3 Principle of Self-aligning Bearing *568*
15.4 Selection of Bearing-type *569*
15.5 Static Load Carrying Capacity *569*

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15.6 Stribeck's Equation *569*
15.7 Dynamic Load Carrying Capacity *571*
15.8 Equivalent Bearing Load *571*
15.9 Load-Life Relationship *572*
15.10 Selection of Bearing Life *572*
15.11 Load Factor *573*
15.12 Selection of Bearing from Manufacturer's Catalogue *573*
15.13 Selection of Taper Roller Bearings *580*
15.14 Design for Cyclic Loads and Speeds *588*
15.15 Bearing with Probability of Survival other than 90 Per Cent *592*
15.16 Needle Bearings *595*
15.17 Bearing Failure—Causes and Remedies *596*
15.18 Lubrication of Rolling Contact Bearings *596*
15.19 Mounting of Bearing *597*
Short-Answer Questions *598*
Problems for Practice *599*

16. Sliding Contact Bearings **601**

16.1 Basic Modes of Lubrication *601*
16.2 Viscosity *604*
16.3 Measurement of Viscosity *605*
16.4 Viscosity Index *605*
16.5 Petroff's Equation *606*
16.6 McKee's Investigation *607*
16.7 Viscous Flow through Rectangular Slot *608*
16.8 Hydrostatic Step Bearing *609*
16.9 Energy Losses in Hydrostatic Bearing *611*
16.10 Reynold's Equation *619*
16.11 Raimondi and Boyd Method *622*
16.12 Temperature Rise *624*
16.13 Bearing Design—Selection of Parameters *625*
16.14 Bearing Constructions *634*
16.15 Bearing Materials *635*
16.16 Sintered Metal Bearings *637*
16.17 Lubricating Oils *637*
16.18 Additives for Mineral Oils *639*
16.19 Selection of Lubricants *640*
16.20 Greases *641*
16.21 Bearing Failure—Causes and Remedies *641*
16.22 Comparison of Rolling and Sliding Contact Bearings *642*
Short-Answer Questions *643*
Problems for Practice *644*

17. Spur Gears **646**

17.1 Mechanical Drives *646*
17.2 Gear Drives *647*

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17.3 Classification of Gears *647*
17.4 Selection of Type of Gears *648*
17.5 Law of Gearing *649*
17.6 Terminology of Spur Gears *650*
17.7 Standard Systems of Gear Tooth *653*
17.8 Gear Trains *656*
17.9 Interference and Undercutting *657*
17.10 Backlash *658*
17.11 Force Analysis *658*
17.12 Gear Tooth Failures *665*
17.13 Selection of Material *666*
17.14 Gear Blank Design *667*
17.15 Number of Teeth *670*
17.16 Face Width *671*
17.17 Beam Strength of Gear Tooth *672*
17.18 Permissible Bending Stress *673*
17.19 Effective Load on Gear Tooth *674*
17.20 Estimation of Module Based on Beam Strength *677*
17.21 Wear Strength of Gear Tooth *678*
17.22 Estimation of Module Based on Wear Strength *680*
17.23 Internal Gears *688*
17.24 Gear Lubrication *690*
Short-Answer Questions *690*
Problems for Practice *690*

18. Helical Gears **694**

18.1 Helical Gears *694*
18.2 Terminology of Helical Gears *694*
18.3 Virtual Number of Teeth *695*
18.4 Tooth Proportions *696*
18.5 Force Analysis *697*
18.6 Beam Strength of Helical Gears *702*
18.7 Effective Load on Gear Tooth *702*
18.8 Wear Strength of Helical Gears *703*
18.9 Herringbone Gears *706*
18.10 Crossed Helical Gears *708*
Short-Answer Questions *709*
Problems for Practice *710*

19. Bevel Gears **711**

19.1 Bevel Gears *711*
19.2 Terminology of Bevel Gears *713*
19.3 Force Analysis *715*
19.4 Beam Strength of Bevel Gears *720*
19.5 Wear Strength of Bevel Gears *722*
19.6 Effective Load on Gear Tooth *722*

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19.7 Spiral Bevel Gears *727*
Short-Answer Questions *728*
Problems for Practice *728*

20. Worm Gears **730**

20.1 Worm Gears *730*
20.2 Terminology of Worm Gears *731*
20.3 Proportions of Worm Gears *733*
20.4 Force Analysis *735*
20.5 Friction in Worm Gears *737*
20.6 Selection of Materials *741*
20.7 Strength Rating of Worm Gears *742*
20.8 Wear Rating of Worm Gears *744*
20.9 Thermal Considerations *745*
Short-Answer Questions *747*
Problems for Practice *747*

21. Flywheel **749**

21.1 Flywheel *749*
21.2 Flywheel and Governor *750*
21.3 Flywheel Materials *750*
21.4 Torque Analysis *751*
21.5 Coefficient of Fluctuation of Energy *752*
21.6 Solid Disk Flywheel *753*
21.7 Rimmed Flywheel *755*
21.8 Stresses in Rimmed Flywheel *756*
Short-Answer Questions *767*
Problems for Practice *767*

22. Cylinders and Pressure Vessels **768**

22.1 Thin Cylinders *768*
22.2 Thin Spherical Vessels *769*
22.3 Thick Cylinders—Principal Stresses *770*
22.4 Lame's Equation *771*
22.5 Clavarino's and Birnie's Equations *772*
22.6 Cylinders with External Pressure *774*
22.7 Autofrettage *775*
22.8 Compound Cylinder *775*
22.9 Gaskets *779*
22.10 Gasketed Joint *780*
22.11 Unfired Pressure Vessels *783*
22.12 Thickness of Cylindrical and Spherical Shells *785*
22.13 End Closures *785*
22.14 Openings in Pressure Vessel *791*
Short-Answer Questions *794*
Problems for Practice *794*

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23. Miscellaneous Machine Elements **796**

23.1 Oil Seals *796*
23.2 Wire Ropes *797*
23.3 Stresses in Wire Ropes *800*
23.4 Rope Sheaves and Drums *804*
23.5 Buckling of Columns *806*
Short-Answer Questions 812
Problems for Practice 812

24. Statistical Considerations in Design **814**

24.1 Frequency Distribution *814*
24.2 Characteristics of Frequency Curves *816*
24.3 Measures of Central Tendency and Dispersion *817*
24.4 Probability *819*
24.5 Probability Distribution *819*
24.6 Normal Curve *821*
24.7 Population Combinations *823*
24.8 Design and Natural Tolerances *825*
24.9 Reliability *829*
24.10 Probabilistic Approach to Design *830*
Short-Answer Questions 840
Problems for Practice 841

25. Design of IC Engine Components **843**

25.1 Internal Combustion Engine *843*
25.2 Cylinder and Cylinder Liner *844*
25.3 Bore and Length of Cylinder *845*
25.4 Thickness of Cylinder Wall *845*
25.5 Stresses in Cylinder Wall *846*
25.6 Cylinder Head *847*
25.7 Design of Studs for Cylinder Head *847*
25.8 Piston *853*
25.9 Piston Materials *854*
25.10 Thickness of Piston Head *854*
25.11 Piston Ribs and Cup *855*
25.12 Piston Rings *856*
25.13 Piston Barrel *857*
25.14 Piston Skirt *858*
25.15 Piston Pin *858*
25.16 Connecting Rod *867*
25.17 Buckling of Connecting Rod *868*
25.18 Cross-section for Connecting Rod *869*
25.19 Big and Small End Bearings *871*

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25.20 Big End Cap and Bolts *873*
25.21 Whipping Stress *875*
25.22 Crankshaft *880*
25.23 Design of Centre Crankshaft *881*
25.24 Centre Crankshaft at Top-Dead Centre Position *881*
25.25 Centre Crankshaft at Angle of Maximum Torque *883*
25.26 Side Crankshaft at Top-Dead Centre Position *892*
25.27 Side Crankshaft at Angle of Maximum Torque *895*
25.28 Valve-Gear Mechanism *903*
25.29 Design of Valves *904*
25.30 Design of Rocker Arm *906*
25.31 Design of Valve Spring *910*
25.32 Design of Push Rod *911*
Short-Answer Questions *922*
Problems for Practice *923*

References *927*
Index *930*

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Preface

It was really a pleasure to receive an overwhelming response to the textbook *Design of Machine Elements* since it was published first in 1994. In fact, whenever I visit an engineering college in any part of the country, students and staff members of the Mechanical Engineering Department know me as the 'Machine Design author' and the book has become my identity.

Machine design occupies a prominent position in the curriculum of Mechanical Engineering. It consists of applications of scientific principles, technical information and innovative ideas for the development of a new or improved machine. The task of a machine designer has never been easy, since he has to consider a number of factors, which are not always compatible with the present-day technology. In the context of today's technical and social climate, the designer's task has become increasingly difficult. Today's designer is required to account for many factors and considerations that are almost impossible for one individual to be thoroughly conversant with. At the same time, he cannot afford to play a role of something like that of a music director. He must have a special competence of his own and a reasonable knowledge of other 'instruments.'

New to this Edition

After the publication of the second edition in 2007, it was observed that there was a need to incorporate a broader coverage of topics in the textbook to suit the content of 'Machine Design' syllabi of various universities in our country. One complete chapter on **'Design of Engine Components'** (Chapter 25) and half a chapter on **'Design of Riveted Joints'** (Chapter 8) are added to fulfill this requirement. *Design of Engine Components* includes cylinders, pistons, connecting rods, crankshafts and valve-gear mechanism. *Design of Riveted Joints* includes strength equations, eccentrically loaded joints and riveted joints in boiler shells.

Another important feature of the current edition is changing the style of solutions to numerical examples. A 'step-by-step' approach is incorporated in all solved examples of the book. This will further simplify and clarify the understanding of the examples.

Target Audience

This book is intended to serve as a textbook for all the courses in Machine Design. It covers the syllabi of all universities, technical boards and professional examining bodies such as Institute of Engineers in the country. It is also useful for the preparation of competitive examinations like UPSC and GATE.

This textbook is particularly written for the students of the Indian subcontinent, who find it difficult to adopt the textbooks written by foreign authors.

Salient Features

The main features of the book are the following:

(i) SI system of units used throughout the book

(ii) Indian standards used throughout the book for materials, tolerances, screw threads, springs, gears, wire ropes and pressure vessels

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(iii) The basic procedure for selection of a machine component from the manufacturer's catalogue discussed with a particular reference to Indian products
(iv) Step by step approach of problem solving

Organization

The book is divided into 25 chapters. **Chapter 1** is an introductory chapter on machine design and discusses the various procedures, requirements, design methods and ergonomic considerations for design. **Chapter 2** is on engineering materials and describes the different kinds of irons, steels and alloys used in engineering design. **Chapter 3** explains in detail the manufacturing considerations in design. **Chapters 4 and 5** discuss the various procedures for design against static load and fluctuating load correspondingly.

Chapter 6 describes power screws in detail while **chapters 7 and 8** specify the features and varieties of threaded joints, and welded and riveted joints in that order. Similarly, **chapters 9 to 22** are each devoted to a particular design element, that is, shafts, keys and couplings; springs; friction clutches; brakes; belt drives; chain drives; rolling contact bearings; sliding contact bearings; spur gears; helical gears; bevel gears; worm gears; flywheel; cylinders and pressure vessels respectively.

Chapter 23 describes miscellaneous machine elements like oil seals, wire ropes, rope sheaves and drums. **Chapter 24** details the various statistical considerations in design. Finally, **Chapter 25** explains the design of IC engine components.

Web Resources

The readers should note that there is a website of this textbook which can be accessed at
http://www.mhhe.com/bhandari/dme3e that contains the following.

For Instructors:
(i) Solution Manual
(ii) Power Point Lecture Slides

For Students:
(i) Interactive 643 Objective Type Questions
(ii) 803 Short Answer Questions
(iii) Glossary
(iv) Bibliography

The above additional information will be useful for students in preparing for competitive examinations.

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Visual Walkthrough

Introduction

Each chapter begins with an Introduction of the Machine Element designed in the chapter and its functions. This helps the reader in gaining an overview of the machine element.

Friction Clutches

Chapter 11

11.1 CLUTCHES

The clutch is a mechanical device, which is used to connect or disconnect the source of power from the remaining parts of the power transmission system at the will of the operator. An automotive clutch can permit the engine to run without driving the car. This is desirable when the engine is to be started or stopped, or when the gears are to be shifted.

Very often, three terms are used together, namely, couplings, clutches and brakes. There is a basic difference between the coupling and the clutch. A coupling, such as a flange coupling, is a permanent connection. The driving and driven shafts are permanently attached by means of coupling and it is not possible to disconnect the shafts, unless the coupling is dismantled. On the other hand, the clutch can connect or disconnect the driving and driven shafts, as and when required by

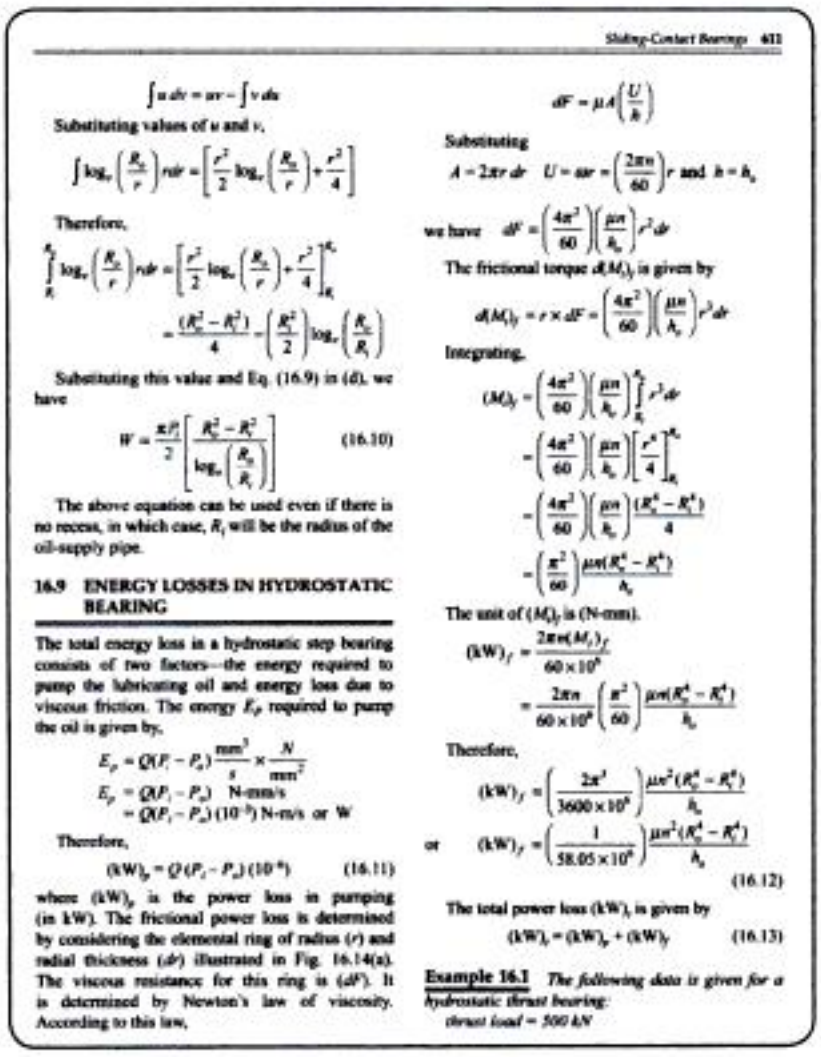
(i) Initial Condition One member such as the brake drum is rotating and the braking member such as the brake shoe is at rest.

(ii) Final Condition Both members are at rest and have no relative motion.

Clutches are classified into the following four groups:

(1) Positive contact Clutches They include square jaw clutches; spiral jaw clutches and toothed clutches. In these clutches, power transmission is achieved by means of interlocking of jaws or teeth. Their main advantage is positive engagement and once coupled, they can transmit large torque with no slip.

(2) Friction Clutches They include single and multi-plate clutches, cone clutches and centrifugal clutches. In these clutches, power transmission is achieved by means of friction between connecting



Sliding-Contact Bearings 411

$$\int u\,dv = uv - \int v\,du$$

Substituting values of u and v,

$$\int \log_e\left(\frac{R_o}{r}\right) r\,dr = \left[\frac{r^2}{2}\log_e\left(\frac{R_o}{r}\right) + \frac{r^2}{4}\right]$$

Therefore,

$$\int_{R_i}^{R_o} \log_e\left(\frac{R_o}{r}\right) r\,dr = \left[\frac{r^2}{2}\log_e\left(\frac{R_o}{r}\right) + \frac{r^2}{4}\right]_{R_i}^{R_o}$$

$$= \frac{(R_o^2 - R_i^2)}{4} - \left(\frac{R_i^2}{2}\right)\log_e\left(\frac{R_o}{R_i}\right)$$

Substituting this value and Eq. (16.9) in (d), we have

$$W = \frac{\pi p_i}{2}\left[\frac{R_o^2 - R_i^2}{\log_e\left(\frac{R_o}{R_i}\right)}\right] \qquad (16.10)$$

The above equation can be used even if there is no recess, in which case, R_i will be the radius of the oil-supply pipe.

16.9 ENERGY LOSSES IN HYDROSTATIC BEARING

The total energy loss in a hydrostatic step bearing consists of two factors—the energy required to pump the lubricating oil and energy loss due to viscous friction. The energy E_p required to pump the oil is given by,

$$E_p = Q(P_i - P_a)\frac{\text{mm}^3}{s} \times \frac{N}{\text{mm}^2}$$

$$E_p = Q(P_i - P_a) \text{ N-mm/s}$$

$$= Q(P_i - P_a)(10^{-3}) \text{ N-m/s or W}$$

Therefore,

$$(\text{kW})_p = Q(P_i - P_a)(10^{-6}) \qquad (16.11)$$

where $(\text{kW})_p$ is the power loss in pumping (in kW). The frictional power loss is determined by considering the elemental ring of radius (r) and radial thickness (dr) illustrated in Fig. 16.14(a). The viscous resistance for this ring is (dF). It is determined by Newton's law of viscosity. According to this law,

$$dF = \mu A\left(\frac{U}{h}\right)$$

Substituting

$$A = 2\pi r\,dr \quad U = \omega r = \left(\frac{2\pi n}{60}\right) r \text{ and } h = h_o$$

we have $dF = \left(\frac{4\pi^2}{60}\right)\left(\frac{\mu n}{h_o}\right) r^2\,dr$

The frictional torque $d(M_t)_f$ is given by

$$d(M_t)_f = r \times dF = \left(\frac{4\pi^2}{60}\right)\left(\frac{\mu n}{h_o}\right) r^3\,dr$$

Integrating,

$$(M_t)_f = \left(\frac{4\pi^2}{60}\right)\left(\frac{\mu n}{h_o}\right)\int_{R_i}^{R_o} r^3\,dr$$

$$= \left(\frac{4\pi^2}{60}\right)\left(\frac{\mu n}{h_o}\right)\left[\frac{r^4}{4}\right]_{R_i}^{R_o}$$

$$= \left(\frac{4\pi^2}{60}\right)\left(\frac{\mu n}{h_o}\right)\frac{(R_o^4 - R_i^4)}{4}$$

$$= \left(\frac{\pi^2}{60}\right)\frac{\mu n(R_o^4 - R_i^4)}{h_o}$$

The unit of $(M_t)_f$ is (N-mm).

$$(\text{kW})_f = \frac{2\pi n(M_t)_f}{60 \times 10^6}$$

$$= \frac{2\pi n}{60 \times 10^6}\left(\frac{\pi^2}{60}\right)\frac{\mu n(R_o^4 - R_i^4)}{h_o}$$

Therefore,

$$(\text{kW})_f = \left(\frac{2\pi^3}{3600 \times 10^6}\right)\frac{\mu n^2(R_o^4 - R_i^4)}{h_o}$$

or $$(\text{kW})_f = \left(\frac{1}{58.05 \times 10^6}\right)\frac{\mu n^2(R_o^4 - R_i^4)}{h_o} \qquad (16.12)$$

The total power loss $(\text{kW})_t$ is given by

$$(\text{kW})_t = (\text{kW})_p + (\text{kW})_f \qquad (16.13)$$

Example 16.1 *The following data is given for a hydrostatic thrust bearing:*

thrust load = 500 kN

Theoretical Considerations

Basic equations for design are derived from first principle, with a step-by-step approach.

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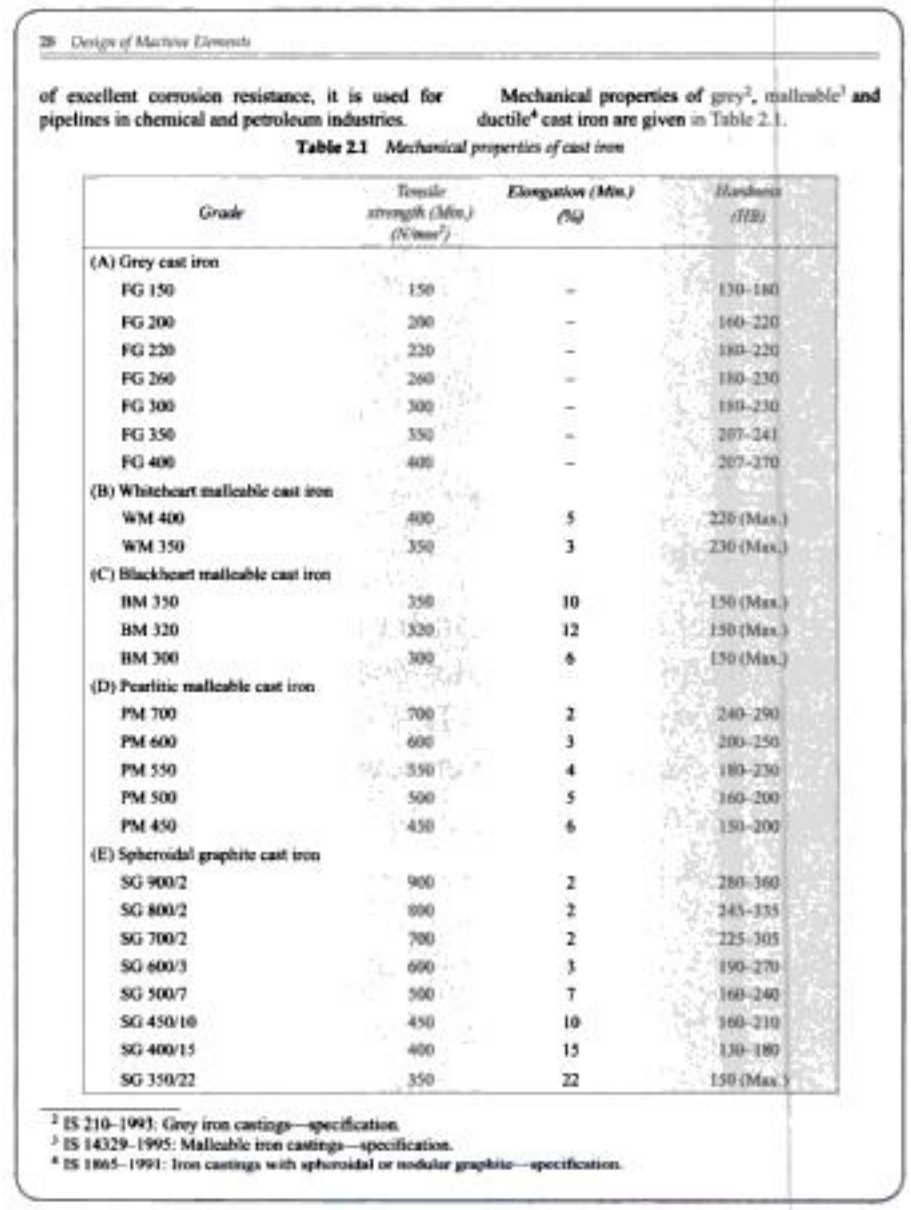
Properties of Materials

Exhaustive tables are provided from Indian Standards for Mechanical Properties of Engineering Materials.



28 Design of Machine Elements

of excellent corrosion resistance, it is used for pipelines in chemical and petroleum industries.

Mechanical properties of grey[2], malleable[3] and ductile[4] cast iron are given in Table 2.1.

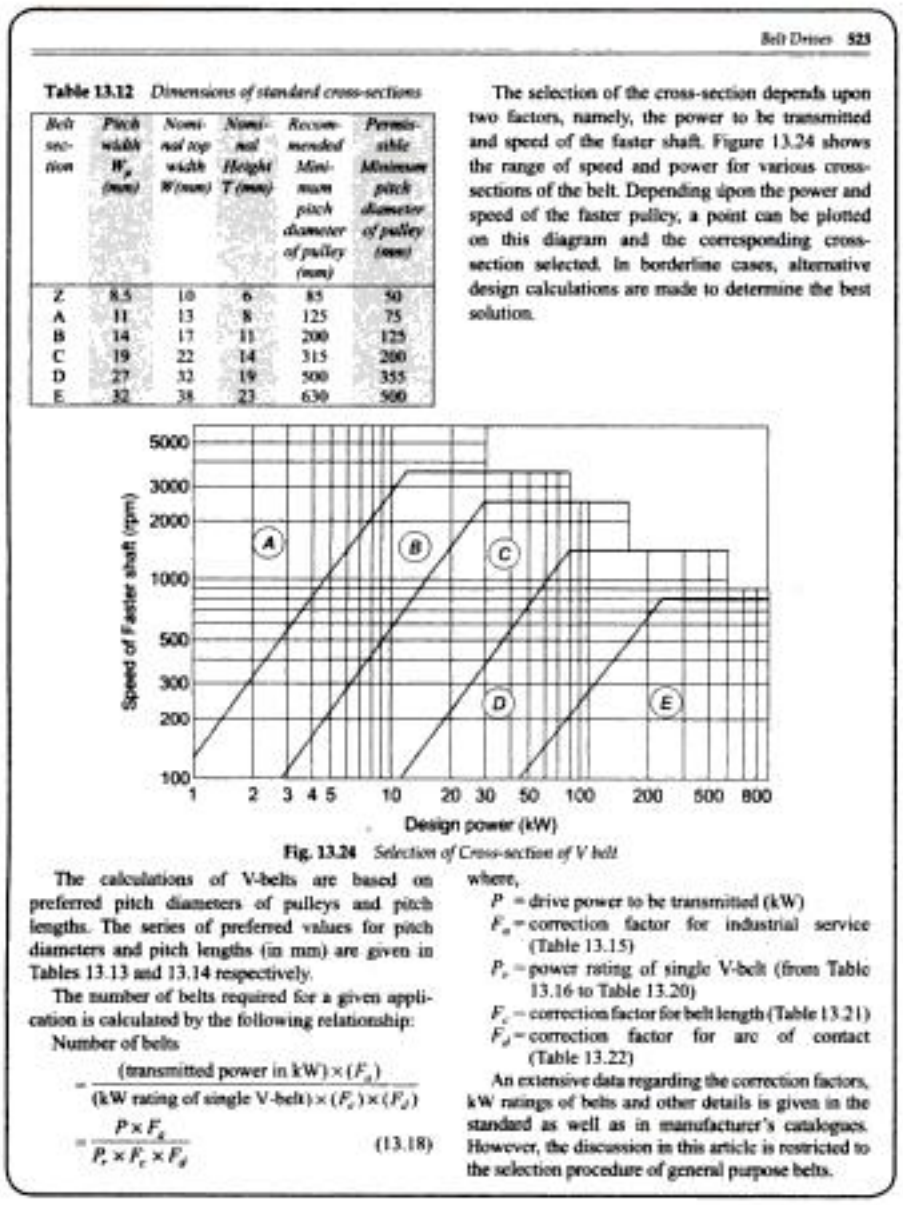
Table 2.1 *Mechanical properties of cast iron*

Grade	Tensile strength (Min.) (N/mm²)	Elongation (Min.) (%)	Hardness (HB)
(A) Grey cast iron			
FG 150	150	–	130–180
FG 200	200	–	160–220
FG 220	220	–	180–220
FG 260	260	–	180–230
FG 300	300	–	180–230
FG 350	350	–	207–241
FG 400	400	–	207–270
(B) Whiteheart malleable cast iron			
WM 400	400	5	220 (Max.)
WM 350	350	3	230 (Max.)
(C) Blackheart malleable cast iron			
BM 350	350	10	150 (Max.)
BM 320	320	12	150 (Max.)
BM 300	300	6	150 (Max.)
(D) Pearlitic malleable cast iron			
PM 700	700	2	240–290
PM 600	600	3	200–250
PM 550	550	4	180–230
PM 500	500	5	160–200
PM 450	450	6	150–200
(E) Spheroidal graphite cast iron			
SG 900/2	900	2	280–360
SG 800/2	800	2	245–335
SG 700/2	700	2	225–305
SG 600/3	600	3	190–270
SG 500/7	500	7	160–240
SG 450/10	450	10	160–210
SG 400/15	400	15	130–180
SG 350/22	350	22	150 (Max.)

[2] IS 210–1993: Grey iron castings—specification.
[3] IS 14329–1995: Malleable iron castings—specification.
[4] IS 1865–1991: Iron castings with spheroidal or nodular graphite—specification.

Indian Standards

Indian Standards are used for Machine Elements like screw threads, belts, springs, gears, wire ropes and pressure vessels.



Belt Drives 523

Table 13.12 *Dimensions of standard cross-sections*

Belt section	Pitch width W_p (mm)	Nominal top width W (mm)	Nominal Height T (mm)	Recommended Minimum pitch diameter of pulley (mm)	Permissible Minimum pitch diameter of pulley (mm)
Z	8.5	10	6	85	50
A	11	13	8	125	75
B	14	17	11	200	125
C	19	22	14	315	200
D	27	32	19	500	355
E	32	38	23	630	500

The selection of the cross-section depends upon two factors, namely, the power to be transmitted and speed of the faster shaft. Figure 13.24 shows the range of speed and power for various cross-sections of the belt. Depending upon the power and speed of the faster pulley, a point can be plotted on this diagram and the corresponding cross-section selected. In borderline cases, alternative design calculations are made to determine the best solution.

Fig. 13.24 *Selection of Cross-section of V belt*

The calculations of V-belts are based on preferred pitch diameters of pulleys and pitch lengths. The series of preferred values for pitch diameters and pitch lengths (in mm) are given in Tables 13.13 and 13.14 respectively.

The number of belts required for a given application is calculated by the following relationship:

$$\text{Number of belts} = \frac{(\text{transmitted power in kW}) \times (F_s)}{(\text{kW rating of single V-belt}) \times (F_c) \times (F_d)} = \frac{P \times F_s}{P_r \times F_c \times F_d} \quad (13.18)$$

where,

P = drive power to be transmitted (kW)
F_s = correction factor for industrial service (Table 13.15)
P_r = power rating of single V-belt (from Table 13.16 to Table 13.20)
F_c = correction factor for belt length (Table 13.21)
F_d = correction factor for arc of contact (Table 13.22)

An extensive data regarding the correction factors, kW ratings of belts and other details is given in the standard as well as in manufacturer's catalogues. However, the discussion in this article is restricted to the selection procedure of general purpose belts.

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operation (24 h per day) such as pumps, compressors and conveyors

The values given in the above tables are only for general guidance. For a particular application, the designer should consider the past experience, the difficulties faced by the customer in replacing the bearing and the economics of breakdown costs.

15.11 LOAD FACTOR

The forces acting on the bearing are calculated by considering the equilibrium of forces in vertical and horizontal planes. These elementary equations do not take into consideration the effect of dynamic load. The forces determined by these equations are multiplied by a *load factor* to determine the dynamic load carrying capacity of the bearing. Load factors are used in applications involving gear, chain and belt drives. In gear drives, there is an additional dynamic load due to inaccuracies of the tooth profile and the elastic deformation of teeth. In chain and

15.12 SELECTION OF BEARING FROM MANUFACTURER'S CATALOGUE

The basic procedure for the selection of a bearing from the manufacturer's catalogue consists of the following steps:

(i) Calculate the radial and axial forces acting on the bearing and determine the diameter of the shaft where the bearing is to be fitted.

(ii) Select the type of bearing for the given application.

(iii) Determine the values of X and Y, the radial and thrust factors, from the catalogue. The values of X and Y factors for single-row deep groove ball bearings are given in Table 15.4. The values depend upon two ratios, $\left(\frac{F_a}{F_r}\right)$ and $\left(\frac{F_a}{C_0}\right)$, where C_0 is the static load capacity. The selection of the bearing is, therefore, done by trial and error. The static and dynamic load capacities of

Selection Procedure

When a machine component is to be selected from manufacturer's catalogue, the selection processes are discussed with a particular reference to Indian products.

Free-Body Diagram of Forces

Whenever required, free-body diagrams are constructed to help the reader understand the forces acting on individual components.

478 *Design of Machine Elements*

The rate of heat generated during the braking period is equal to the rate of work done by the frictional force.

Rate of heat generated = frictional force × average velocity

$= \mu M(1.047)$
$= 0.35\,(3571.43)\,(1.047)$
$= 1308.75$ N-m/s or W (iii)

Step IV Dimensions of the block

From Eq. (12.7),

$N = plw$

$3571.43 = (1)(2w)(w)$

$\therefore\ w = 42.26$ mm or 45 mm

$l = 2w = 90$ mm (iv)

Step V Self-locking property

Referring to Fig. 12.5 (a),

$a = 200$ mm $\quad c = 50$ mm $\quad \mu = 0.35$

$\left(\frac{a}{c}\right) = 4 \qquad \therefore \left(\frac{a}{c}\right) > \mu$

The brake is not self-locking.

Example 12.5 *A double block brake is shown in Fig. 12.6. The brake drum rotates in a clockwise direction and the actuating force is 500 N. The coefficient of friction between the blocks and the drum is 0.35. Calculate the torque absorbing capacity of the brake.*

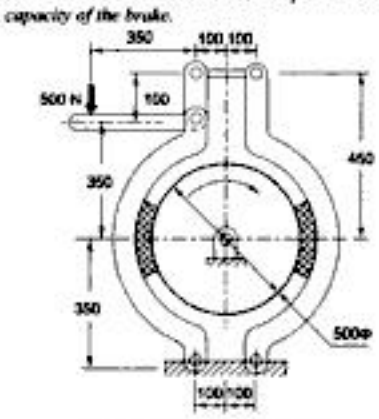

Fig. 12.6 *Double Block Brake*

Solution

Given $P = 500$ N $\quad \mu = 0.35 \quad R = 250$ mm

Step I Force analysis

The free-body diagram of forces acting on various parts is shown in Fig. 12.7. Considering the forces

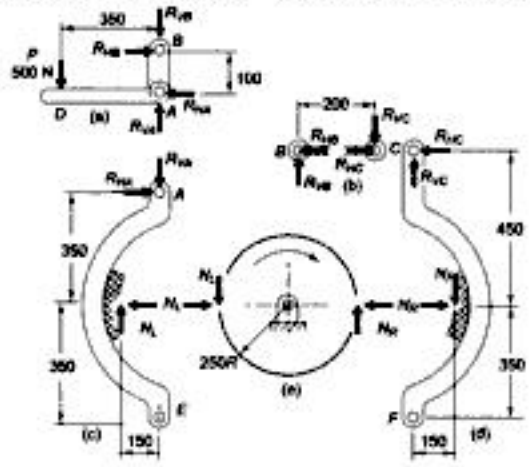

Fig. 12.7 *Free-body Diagram of Forces*

172 *Design of Machine Elements*

$\sigma_{min} = 0$

$\sigma_a = \sigma_m = \frac{1}{2}\sigma_{max} = \frac{1}{2}\left(\frac{750}{t}\right) = \left(\frac{375}{t}\right)$ N/mm²

$\tan\theta = \frac{\sigma_a}{\sigma_m} = 1 \qquad$ or $\qquad \theta = 45°$

The modified Goodman diagram for this example is shown in Fig. 5.45.

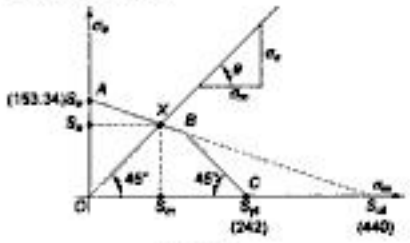

Fig. 5.45

Step III Permissible stress amplitude

Refer to Fig. 5.45. The coordinates of the point X are determined by solving the following two equations simultaneously.

(i) Equation of line AB

no stress concentration and the expected reliability is 50%. Calculate the number of stress cycles likely to cause fatigue failure.

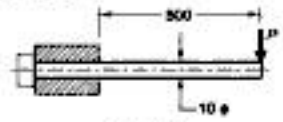

Fig. 5.46

Solution

Given For cantilever spring, $d = 10$ mm
$l = 500$ mm $\quad P = 75$ to 150 N $\quad S_{ut} = 860$ N/mm²
$S_{yt} = 690$ N/mm² $\quad R = 50\%$

Step I Endurance limit stress for cantilever beam

$S_e' = 0.5S_{ut} = 0.5(860) = 430$ N/mm²

From Fig. 5.24 (machined surface and $S_{ut} = 860$ N/mm²),

$K_a = 0.72$

For 10 mm diameter wire, $K_b = 0.85$

For 50% reliability,

$K_c = 1.0$

$S_e = K_a K_b K_c S_e'$
$= 0.72(0.85)(1.0)(430) = 263.16$ N/mm²

Fatigue Diagrams

Fatigue diagrams are constructed for design of machine components subjected to fluctuating loads.

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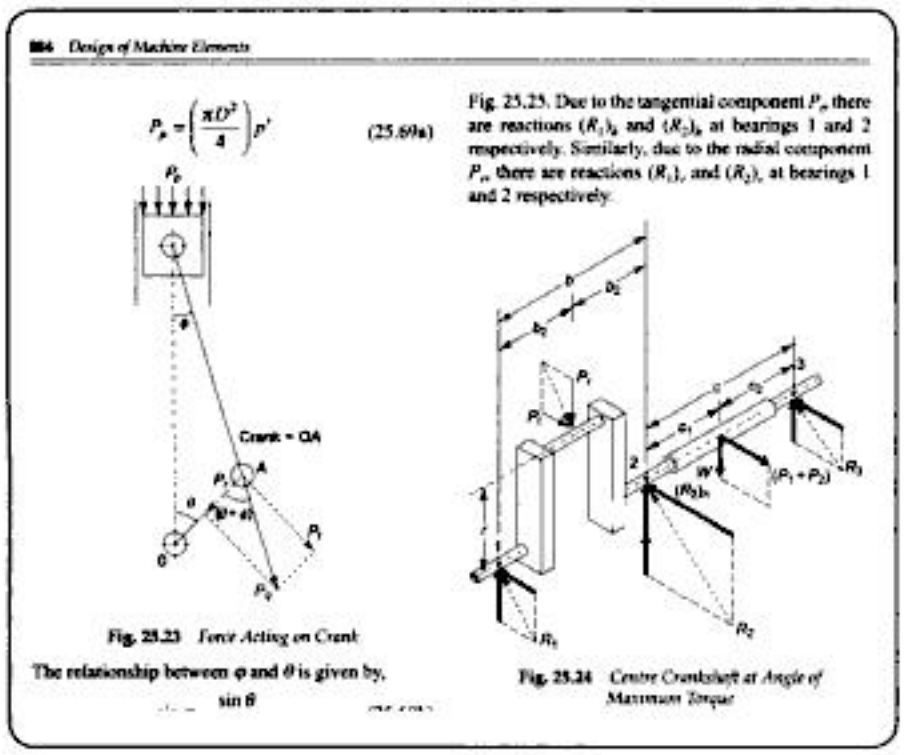

$$P_p = \left(\frac{\pi D^2}{4}\right) p' \qquad (25.69a)$$

Fig. 25.23 *Force Acting on Crank*

The relationship between φ and θ is given by,

$$\sin\theta$$

Fig. 25.25. Due to the tangential component P_t there are reactions $(R_1)_t$ and $(R_2)_t$ at bearings 1 and 2 respectively. Similarly, due to the radial component P_r there are reactions $(R_1)_r$ and $(R_2)_r$ at bearings 1 and 2 respectively.

Fig. 25.24 *Centre Crankshaft at Angle of Maximum Torque*

Isometric Views

When it is difficult to understand the forces in three dimensions, isometric views are given for clear understanding.

Statistical Considerations in Design

Chapter **24**

24.1 FREQUENCY DISTRIBUTION

Statistics deals with drawing conclusions from a given or observed data. Statistical techniques are used for collection, processing, analysis and interpretation of numerical data. On the basis of statistical analysis, valid conclusions are drawn and reasonable decisions are taken. Statistics enables the engineers to understand the phenomena of variations and how to effectively predict and control them. Statistics has made valuable contributions in the areas of product design and manufacture and effective use of material and labour.

The basic data consists of observations, such as the diameters of shafts manufactured in one shift. In this case, the group of all shafts is called '*population*'. When the group is large, it is not possible to take observations of the entire population. In such cases, only a small portion of the population is examined and this portion is called a '*sample*'. *Population is defined as a collection of all elements we are studying and about which we are trying to draw conclusions. On the other hand, a sample is defined as a collection of some, but not all, of the elements of the population.* A sample is a part of the population but the converse is not true. It is easier to study a sample than studying the whole population. It costs less and takes less time. A representative sample has the characteristics of the population in the same proportions, as they are included in the entire population. Therefore, many times a sample is analysed instead of the entire population.

Let us consider an example of 100 shafts of hydrodynamic bearing, with recommended tolerance of 40e7. The shafts are manufactured on lathe and finished on grinding machine. Their diameters are measured by micrometer and the readings are tabulated in Table 24.1. The readings in Table 24.1 are called '*raw data*'. *A data is defined as the collection of numbers belonging to observations of one or more variables.* In this case, the diameter of shaft is a variable and one hundred numerical readings taken by micrometer are a data. Raw data is a data before it is arranged or analysed by any statistical method. Raw data does not show any pattern or trend of population and does not lead to any conclusions. In this chapter, the objective of statistical techniques is to rearrange and present the data in a useful way so that the decisions can be taken.

Let us rearrange the data given in Table 24.1 on the basis of diameter of shaft against the number of shafts. As shown in Table 24.2, a particular diameter such as 39.926 mm is written in the first column and each shaft with this diameter is shown by a mark X against it. Finally, the number of marks are counted and written in the last column. For example, the total number of shafts with a diameter of 39.940 mm is 8. The data in Table 24.2 is further

Statistical Considerations in Design

A separate chapter on Statistical Considerations in Design is included and examples are solved on the basis of reliability.

Example 10.5 *A helical tension spring is used in the spring balance to measure the weights. One end of the spring is attached to the rigid support while the other end, which is free, carries the weights to be measured. The maximum weight attached to the spring balance is 1500 N and the length of the scale should be approximately 100 mm. The spring index can be taken as 6. The spring is made of oil-hardened and tempered steel wire with ultimate tensile strength of 1360 N/mm² and modulus of rigidity of 81 370 N/mm². The permissible shear stress in the spring wire should be taken as 50% of the ultimate tensile strength. Design the spring and calculate*

(i) wire diameter,
(ii) mean coil diameter,
(iii) number of active coils,
(iv) required spring rate, and
(v) actual spring rate.

Solution

Given $P = 1500$ N $\quad C = 6 \quad S_{ut} = 1360$ N/mm² $\quad G = 81\,370$ N/mm² $\quad \tau = 0.5\,S_{ut}$

Step I Wire Diameter

The working principle of the spring balance is illustrated in Fig. 10.18. As the load acting on the spring varies from 0 to 1500 N, the pointer attached to the free end of the spring moves over a scale between highest and lowest positions. The length of the scale between these two positions of the pointer is 100 mm. In other words, the spring deflection is 100 mm when the force is 1500 N.

The permissible shear stress for spring wire is given by,

$$\tau = 0.5\,S_{ut} = 0.5\,(1360) = 680 \text{ N/mm}^2$$

$$K = \frac{4C-1}{4C-4} + \frac{0.615}{C} = \frac{4\times6-1}{4\times6-4} + \frac{0.615}{6} = 1.2525$$

From Eq. (10.13),

$$\tau = K\left(\frac{8PC}{\pi d^3}\right) \quad \text{or} \quad 680 = 1.2525\left\{\frac{8(1500)(6)}{\pi d^3}\right\}$$

$$\therefore \quad d = 6.5 \text{ or } 7 \text{ mm} \qquad \text{(i)}$$

Step II Mean coil diameter

$$D = Cd = 6\,(7) = 42 \text{ mm} \qquad \text{(ii)}$$

Step III Number of active coils

From Eq. (10.8),

$$\delta = \frac{8PD^3N}{Gd^4} \quad \text{or} \quad 100 = \frac{8(1500)(42)^3 N}{(81\,370)(7)^4}$$

$$\therefore \quad N = 21.97 \text{ or } 22 \text{ coils} \qquad \text{(iii)}$$

Step IV Required spring rate

$$k = \frac{P}{\delta} = \frac{1500}{100} = 15 \text{ N/mm} \qquad \text{(iv)}$$

Step V Actual spring rate

$$k = \frac{Gd^4}{8D^3N} = \frac{(81\,370)(7)^4}{8(42)^3(22)} = 14.98 \text{ N/mm} \qquad \text{(v)}$$

Numerical Examples

Numerical Examples solved by step by step approach are provided in sufficient number in each chapter to help the reader understand the design procedures.

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Short Answer Questions

25.1 What are the functions of engine cylinder?
25.2 What are the cooling systems for engine cylinders? Where do you use them?
25.3 What are the advantages of cylinder liner?
25.4 What are dry and wet cylinder liners?
25.5 What are the desirable properties of cylinder materials?
25.6 Name the materials used for engine cylinder.
25.7 What do you understand by 'bore' of cylinder?
25.8 What are the functions of piston?
25.9 What are the design requirements of piston?
25.10 Name the materials used for engine piston.
25.11 What are the advantages and disadvantages of aluminium piston over cast iron piston?
25.12 Why is piston made lightweight?
25.13 Name two criteria for calculating the thickness of piston head.
25.14 Why is piston clearance necessary? What is its usual value?
25.15 What are the functions of piston ribs?
25.16 What is the function of the cup on piston head?
25.17 What are the functions of compression piston rings?
25.18 What are the functions of oil scraper rings?
25.19 Name the materials used for piston rings.
25.20 Why are more number of thin piston rings preferred over small number of thick rings?
25.30 What is the force on bolts of of big end connecting rod?
25.31 What is the difference between centre and overhung crankshafts?
25.32 Where do you use overhung crankshafts?
25.33 Where do you use centre crankshafts?
25.34 What is the main advantage of overhung crankshafts?
25.35 Name the materials for crankshafts.
25.36 What is the manufacturing method for crankshaft?
25.37 When do you use push rod?
25.38 Why is the design of exhaust valve more critical than that of an inlet valve?
25.39 Why is the area of inlet valve port more than that of an exhaust valve?
25.40 Why do inlet and exhaust valves have conical heads and seats?
25.41 What is the function of rocker arm?
25.42 Why is rocker arm made of I section?
25.43 Name the materials for rocker arm.
25.44 What is tappet? What is the stress in tappet?
25.45 What is the purpose of valve spring?
25.46 What is the criterion for design of push rod?

Problems for Practice

25.1 The cylinder of a four-stroke diesel engine has the following specifications:
Brake power = 3 kW
Speed = 800 rpm

Short-Answer Questions

At the end of each chapter, Short-Answer Questions are provided for the students for preparation of oral and theory examinations.

Problems for Practice

At the end of each chapter, a set of examples with answers is given as exercise to students. It is also helpful to teachers in setting classwork and homework assignments.

Problems for Practice

11.1 A single plate clutch consists of one pair of contacting surfaces. The inner and outer diameters of the friction disk are 125 and 250 mm respectively. The coefficient of friction is 0.25 and the total axial force is 15 kN. Calculate the power transmitting capacity of the clutch at 500 rpm using :
(i) uniform wear theory; and
(ii) uniform pressure theory.
[(i) 18.41 kW (ii) 19.09 kW]

11.2 An automotive single plate clutch consists of two pairs of contacting surfaces. The outer diameter of the friction disk is 270 mm. The coefficient of friction is 0.3 and the maximum intensity of pressure is 0.3 N/mm^2. The clutch is transmitting a torque of 531 N-m. Assuming uniform wear theory, calculate:
(i = 1.1258)

11.5 A leather faced cone clutch transmits power at 500 rpm. The semi-cone angle α is 12.5°. The mean diameter of the clutch is 300 mm, while the face width of the contacting surface of the friction lining is 100 mm. The coefficient of friction is 0.2 and the maximum intensity of pressure is limited to 0.07 N/mm^2. Calculate the force to engage the clutch and the power transmitting capacity.
[1324.68 N and 9.61 kW]

11.6 A centrifugal clutch, transmitting 18.5 kW at 720 rpm, consists of four shoes. The clutch is to be engaged at 75% of the running speed. The inner radius of the drum is 165 mm, while the radius of the centre of gravity of each shoe, during engaged position, is 140 mm. The coefficient of friction is 0.25. Calculate the mass of each shoe.
[4.27 kg]

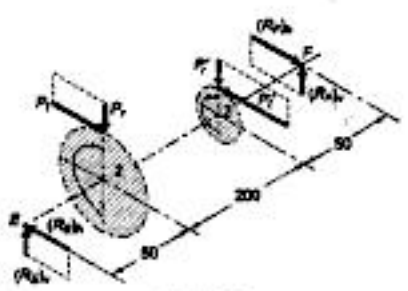

Fig. 17.30

Step II Reactions at bearings E and F
The forces acting in vertical and horizontal planes are shown in Fig. 17.31.

Considering vertical forces and taking moments about the bearing E,

$$P_v \times 50 + P'_v \times 250 = (R_F)_v \times 300$$
$$482.73 \times 50 + 1206.83 \times 250 = (R_F)_v \times 300$$
$$\therefore \quad (R_F)_v = 1086.15 \text{ N} \quad \text{(i)}$$
$$P_v + P'_v = (R_F)_v + (R_E)_v$$
$$482.73 + 1206.83 = 1086.15 + (R_E)_v$$
$$\therefore \quad (R_E)_v = 603.41 \text{ N} \quad \text{(ii)}$$

(b) Horizontal plane

Fig. 17.31

17.12 GEAR TOOTH FAILURES

There are two basic modes of gear tooth failure—breakage of the tooth due to static and dynamic loads and the surface destruction[4, 5]. The complete breakage of the tooth can be avoided by adjusting the parameters in the gear design, such as the module and the face width, so that the beam strength of the gear tooth is more than the sum of static and dynamic loads. The surface destruction or tooth wear is classified according to the basis of their primary causes. The principal types of gear tooth wear are as follows:

(i) Abrasive Wear Foreign particles in the lubricant, such as dirt, rust, weld spatter or metallic debris can scratch or brinell the tooth surface. Remedies against this type of wear are provision of oil filters, increasing surface hardness and use of high viscosity oils. A thick lubricating film developed by these oils allows fine particles to pass without scratching.

(ii) Corrosive Wear The corrosion of the tooth surface is caused by corrosive elements, such as

[4] RH Pearson–'Gear overdesign and how to avoid it'–Machine Design—May 9, 1968, vol. 40, no. 11, p. 153.
[5] Eugene E Shipley – 'Gear failures—how to recognize them, what causes them and how to avoid them?'—Machine Design—Dec 7, 1967, vol. 39, no. 28, p. 152.

References

The list of textbooks, journals and company catalogues is provided at the end of respective pages for quick reference.

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Introduction

Chapter 1

If the point of contact between the product and people becomes a point of friction, then the industrial designer has failed. On the other hand, if people are made safer, more efficient, more comfortable—or just plain happier—by contact with the product, then the designer has succeeded.

Henry Dreyfuss[1]

1.1 MACHINE DESIGN

Machine design is defined as the use of scientific principles, technical information and imagination in the description of a machine or a mechanical system to perform specific functions with maximum economy and efficiency. This definition of machine design contains the following important features:

(i) A designer uses principles of basic and engineering sciences such as physics, mathematics, statics and dynamics, thermodynamics and heat transfer, vibrations and fluid mechanics. Some of the examples of these principles are
   (a) Newton's laws of motion,
   (b) D' Alembert's principle,
   (c) Boyle's and Charles' laws of gases,
   (d) Carnot cycle, and
   (e) Bernoulli's principle.

(ii) The designer has technical information of the basic elements of a machine. These elements include fastening devices, chain, belt and gear drives, bearings, oil seals and gaskets, springs, shafts, keys, couplings, and so on. A machine is a combination of these basic elements. The designer knows the relative advantages and disadvantages of these basic elements and their suitability in different applications.

(iii) The designer uses his skill and imagination to produce a configuration, which is a combination of these basic elements. However, this combination is unique and different in different situations. The intellectual part of constructing a proper configuration is creative in nature.

(iv) The final outcome of the design process consists of the description of the machine. The description is in the form of drawings of assembly and individual components.

(v) A design is created to satisfy a recognised need of customer. The need may be to perform a specific function with maximum economy and efficiency.

[1] Henry Dreyfuss–*The Profile of Industrial Designer*—Machine Design, July 22, 1967.

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Machine design is the creation of plans for a machine to perform the desired functions. The machine may be entirely new in concept, performing a new type of work, or it may more economically perform the work that can be done by an existing machine. It may be an improvement or enlargement of an existing machine for better economy and capability.

1.2 BASIC PROCEDURE OF MACHINE DESIGN

The basic procedure of machine design consists of a step-by-step approach from given specifications about the functional requirements of a product to the complete description in the form of drawings of the final product. A logical sequence of steps, usually common to all design projects, is illustrated in Fig. 1.1. These steps are interrelated and interdependent, each reflecting and affecting all other steps. The following steps are involved in the process of machine design.

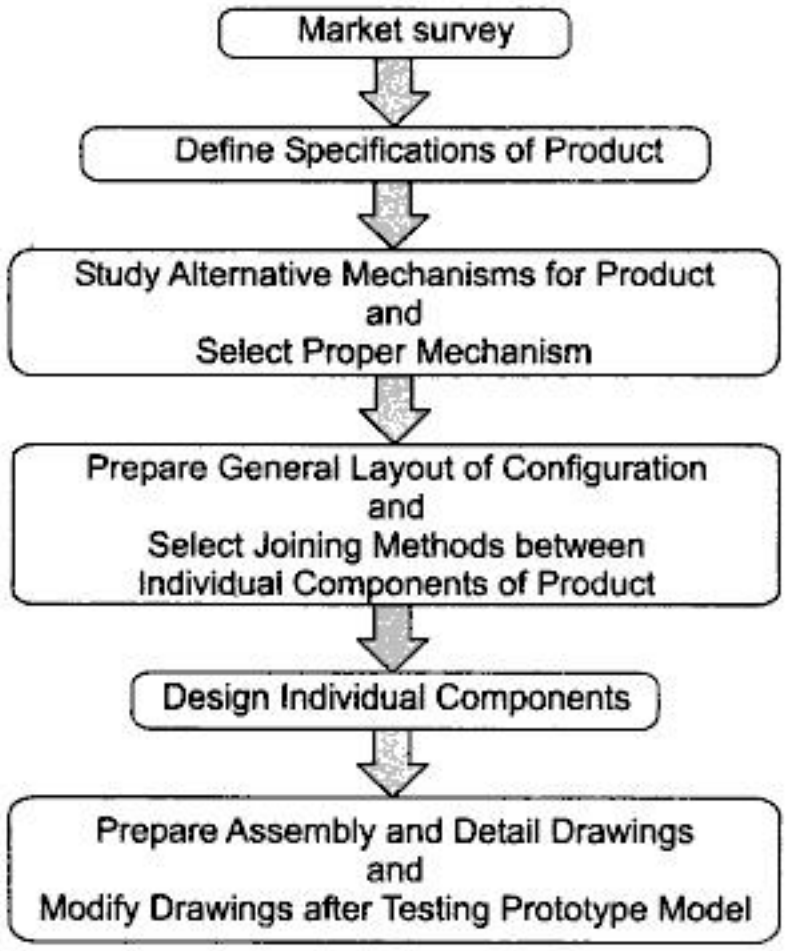

Fig. 1.1 *The Design Process*

Step 1: Product Specifications

The first step consists of preparing a complete list of the requirements of the product. The requirements include the output capacity of the machine, and its service life, cost and reliability. In some cases, the overall dimensions and weight of the product are specified. For example, while designing a scooter, the list of specifications will be as follows:

(i) Fuel consumption = 40 km/l
(ii) Maximum speed = 85 km/hr
(iii) Carrying capacity = two persons with 10 kg luggage
(iv) Overall dimensions
Width = 700 mm
Length = 1750 mm
Height = 1000 mm
(v) Weight = 95 kg
(vi) Cost = Rs 40000 to Rs 45000

In consumer products, external appearance, noiseless performance and simplicity in operation of controls are important requirements. Depending upon the type of product, various requirements are given weightages and a priority list of specifications is prepared.

Step 2: Selection of Mechanism

After careful study of the requirements, the designer prepares rough sketches of different possible mechanisms for the product. For example, while designing a blanking or piercing press, the following mechanisms are possible:

(i) a mechanism involving the crank and connecting rod, converting the rotary motion of the electric motor into the reciprocating motion of the punch;
(ii) a mechanism involving nut and screw, which is a simple and cheap configuration but having poor efficiency; and
(iii) a mechanism consisting of a hydraulic cylinder, piston and valves which is a costly configuration but highly efficient.

The alternative mechanisms are compared with each other and also with the mechanism of the products that are available in the market. An approximate estimation of the cost of each alternative configuration is made and compared with the cost of existing products. This will reveal the competitiveness of the product. While selecting the final configuration, the designer should

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consider whether the raw materials and standard parts required for making the product are available in the market. He should also consider whether the manufacturing processes required to fabricate the non-standard components are available in the factory. Depending upon the cost-competitiveness, availability of raw materials and manufacturing facility, the best possible mechanism is selected for the product.

Step 3: Layout of Configuration

The next step in a design procedure is to prepare a block diagram showing the general layout of the selected configuration. For example, the layout of an Electrically-operated Overhead Travelling (EOT) crane will consist of the following components:

(i) electric motor for power supply;
(ii) flexible coupling to connect the motor shaft to the clutch shaft;
(iii) clutch to connect or disconnect the electric motor at the will of the operator;
(iv) gear box to reduce the speed from 1440 rpm to about 15 rpm;
(v) rope drum to convert the rotary motion of the shaft to the linear motion of the wire rope;
(vi) wire rope and pulley with the crane hook to attach the load; and
(vii) brake to stop the motion.

In this step, the designer specifies the joining methods, such as riveting, bolting or welding to connect the individual components. Rough sketches of shapes of the individual parts are prepared.

Step 4: Design of Individual Components

The design of individual components or machine elements is an important step in a design process. It consists of the following stages:

(i) Determine the forces acting on the component.
(ii) Select proper material for the component depending upon the functional requirements such as strength, rigidity, hardness and wear resistance.
(iii) Determine the likely mode of failure for the component and depending upon it, select the criterion of failure, such as yield strength, ultimate tensile strength, endurance limit or permissible deflection.
(iv) Determine the geometric dimensions of the component using a suitable factor of safety and modify the dimensions from assembly and manufacturing considerations.

This stage involves detailed stress and deflection analysis. The subjects 'Machine Design' or 'Elements of Machine Design' cover mainly the design of machine elements or individual components of the machine. Section 1.4 on Design of Machine Elements, elaborates the details of this important step in design procedure.

Step 5: Preparation of Drawings

The last stage in a design process is to prepare drawings of the assembly and the individual components. On these drawings, the material of the component, its dimensions, tolerances, surface finish grades and machining symbols are specified. The designer prepares two separate lists of components—standard components to be purchased directly from the market and special components to be machined in the factory. In many cases, a prototype model is prepared for the product and thoroughly tested before finalising the assembly drawings.

It is seen that the process of machine design involves systematic approach from known specifications to unknown solutions. Quite often, problems arise on the shop floor during the production stage and design may require modifications. In such circumstances, the designer has to consult the manufacturing engineer and find out the suitable modification.

1.3 BASIC REQUIREMENTS OF MACHINE ELEMENTS

A machine consists of machine elements. Each part of a machine, which has motion with respect to some other part, is called a machine element. It is important to note that each machine element may consist of several parts, which are manufactured separately. For example, a rolling contact bearing is a machine element and it consists of an inner race, outer race,

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cage and rolling elements like balls. Machine elements can be classified into two groups—*general-purpose* and *special-purpose* machine elements. General-purpose machine elements include shafts, couplings, clutches, bearings, springs, gears and machine frames Special-purpose machine elements include pistons, valves or spindles. Special-purpose machine elements are used only in certain types of applications. On the contrary, general-purpose machine elements are used in a large number of machines.

The broad objective of designing a machine element is to ensure that it preserves its operating capacity during the stipulated service life with minimum manufacturing and operating costs. In order to achieve this objective, the machine element should satisfy the following basic requirements:

(i) Strength: A machine part should not fail under the effect of the forces that act on it. It should have sufficient strength to avoid failure either due to fracture or due to general yielding.

(ii) Rigidity: A machine component should be rigid, that is, it should not deflect or bend too much due to forces or moments that act on it. A transmission shaft in many times designed on the basis of lateral and torsional rigidities. In these cases, maximum permissible deflection and permissible angle of twist are the criteria for design.

(iii) Wear Resistance: Wear is the main reason for putting the machine part out of order. It reduces useful life of the component. Wear also leads to the loss of accuracy of machine tools. There are different types of wear such as abrasive wear, corrosive wear and pitting. Surface hardening can increase the wear resistance of the machine components, such as gears and cams.

(iv) Minimum Dimensions and Weight: A machine part should be sufficiently strong, rigid and wear-resistant and at the same time, with minimum possible dimensions and weight. This will result in minimum material cost.

(v) Manufacturability: Manufacturability is the ease of fabrication and assembly. The shape and material of the machine part should be selected in such a way that it can be produced with minimum labour cost.

(vi) Safety: The shape and dimensions of the machine parts should ensure safety to the operator of the machine. The designer should assume the worst possible conditions and apply 'fail-safe' or 'redundancy' principles in such cases.

(vii) Conformance to Standards: A machine part should conform to the national or international standard covering its profile, dimensions, grade and material.

(viii) Reliability: Reliability is the probability that a machine part will perform its intended functions under desired operating conditions over a specified period of time. A machine part should be reliable, that is, it should perform its function satisfactorily over its lifetime.

(ix) Maintainability: A machine part should be maintainable. Maintainability is the ease with which a machine part can be serviced or repaired.

(x) Minimum: Life-cycle Cost: Life-cycle cost of the machine part is the total cost to be paid by the purchaser for purchasing the part and operating and maintaining it over its life span.

It will be observed that the above mentioned requirements serve as the basis for design projects in many cases.

1.4 DESIGN OF MACHINE ELEMENTS

Design of machine elements is the most important step in the complete procedure of machine design. In order to ensure the basic requirements of machine elements, calculations are carried out to find out the dimensions of the machine elements. These calculations form an integral part of the design of machine elements. The basic procedure of the design of machine elements is illustrated in Fig. 1.2. It consists of the following steps:

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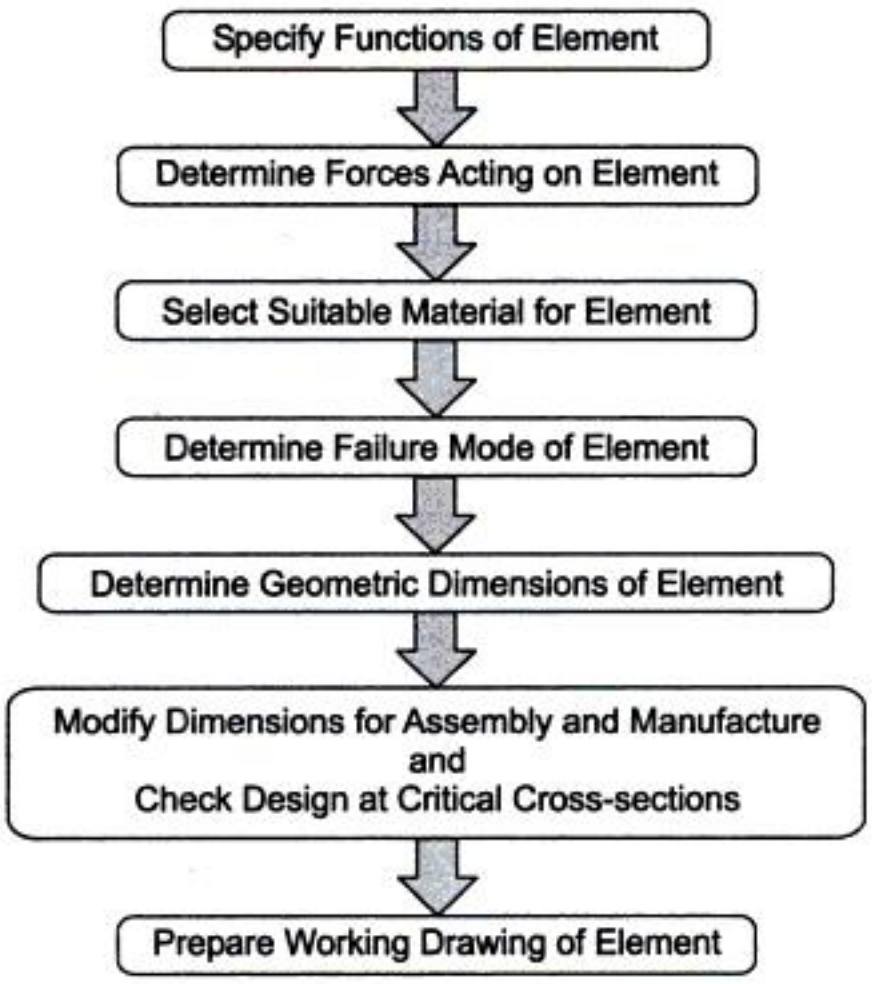

Fig. 1.2 *Basic Procedure of Design of Machine Element*

Step 1: Specification of Function

The design of machine elements begins with the specification of the functions of the element. The functions of some machine elements are as follows:

(i) ***Bearing*** To support the rotating shaft and confine its motion
(ii) ***Key*** To transmit the torque between the shaft and the adjoining machine part like gear, pulley or sprocket
(iii) ***Spring in Clock*** To store and release the energy
(iv) ***Spring in Spring Balance*** To measure the force
(v) ***Screw Fastening*** To hold two or more machine parts together
(vi) ***Power Screw*** To produce uniform and slow motion and to transmit the force

Step 2: Determination of Forces

In many cases, a free-body diagram of forces is constructed to determine the forces acting on different parts of the machine. The external and internal forces that act on a machine element are as follows:

(i) The external force due to energy, power or torque transmitted by the machine part, often called 'useful' load
(ii) Static force due to deadweight of the machine part
(iii) Force due to frictional resistance
(iv) Inertia force due to change in linear or angular velocity
(v) Centrifugal force due to change in direction of velocity
(vi) Force due to thermal gradient or variation in temperature
(vii) Force set up during manufacturing the part resulting in residual stresses
(viii) Force due to particular shape of the part such as stress concentration due to abrupt change in cross-section

For every machine element, all forces in this list may not be applicable. They vary depending on the application. There is one more important consideration. The force acting on the machine part is either assumed to be concentrated at some point in the machine part or distributed over a particular area. Experience is essential to make such assumptions in the analysis of forces.

Step 3: Selection of Material

Four basic factors, which are considered in selecting the material, are availability, cost, mechanical properties and manufacturing considerations.

For example, flywheel, housing of gearbox or engine block have complex shapes. These components are made of cast iron because the casting process produces complicated shapes without involving machining operations. Transmission shafts are made of plain carbon steels, because they are available in the form of rods, besides their higher strength. The automobile body and hood are made of low carbon steels because their cold formability is essential to press the parts. Free cutting steels have excellent machinability due to addition of sulphur. They are ideally suitable for bolts and studs because of the ease with which the thread profiles can be machined. The crankshaft and connecting rod are subjected to fluctuating forces and nickel–chromium steel is used for these components due to its higher fatigue strength.

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Step 4: Failure Criterion

Before finding out the dimensions of the component, it is necessary to know the type of failure that the component may fail when put into service. The machine component is said to have 'failed' when it is unable to perform its functions satisfactorily. The three basic types of failure are as follows:

(i) failure by elastic deflection;
(ii) failure by general yielding; and
(iii) failure by fracture.

In applications like transmission shaft, which is used to support gears, the maximum force acting on the shaft is limited by the permissible deflection. When this deflection exceeds a particular value (usually, 0.001 to 0.003 times of span length between two bearings), the meshing between teeth of gears is affected and the shaft cannot perform its function properly. In this case, the shaft is said to have 'failed' due to elastic deflection. Components made of ductile materials like steel lose their engineering usefulness due to large amount of plastic deformation. This type of failure is called failure by yielding. Components made of brittle materials like cast iron fail because of sudden fracture without any plastic deformation. There are two basic modes of gear-tooth failure—breakage of tooth due to static and dynamic load and surface pitting. The surface of the gear tooth is covered with small 'pits' resulting in rapid wear. Pitting is a surface fatigue failure. The components of ball bearings such as rolling elements, inner and outer races fail due to fatigue cracks after certain number of revolutions. Sliding contact bearings fail due to corrosion and abrasive wear by foreign particles.

Step 5: Determination of Dimensions

The shape of the machine element depends on two factors, viz., the operating conditions and the shape of the adjoining machine element. For example, involute profile is used for gear teeth because it satisfies the fundamental law of gearing. A V-belt has a trapezoidal cross-section because it results in wedge action and increases the force of friction between the surfaces of the belt and the pulley. On the other hand, the pulley of a V-belt should have a shape which will match with the adjoining belt. The profile of the teeth of sprocket wheel should match the roller, bushing, inner and outer link plates of the roller chain. Depending on the operating conditions and shape of the adjoining element, the shape of the machine element is decided and a rough sketch is prepared.

The geometric dimensions of the component are determined on the basis of failure criterion. In simple cases, the dimensions are determined on the basis of allowable stress or deflection. For example, a tension rod, illustrated in Fig. 1.3, is subjected to a force of 5 kN. The rod is made of plain carbon

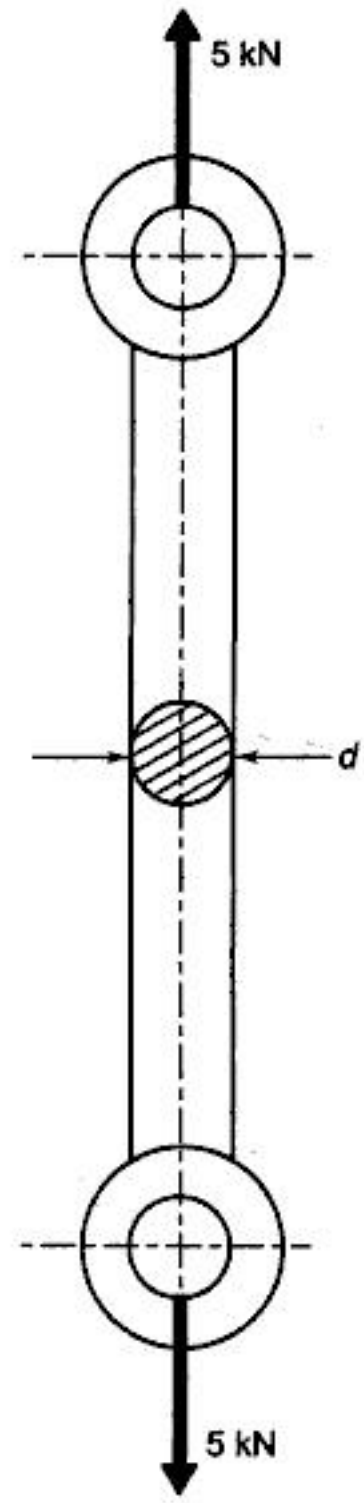

Fig. 1.3 *Tension Rod*

steel and the permissible tensile stress is 80 N/mm^2. The diameter of the rod is determined on the basis of allowable stress using the following expression:

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$$\text{stress} = \frac{\text{force}}{\text{area}} \quad \text{or} \quad 80 = \frac{(5 \times 10^3)}{\left(\frac{\pi d^2}{4}\right)}$$

Therefore,

$$d = 8.92 \text{ or } 10 \text{ mm}$$

As a second example, consider a transmission shaft, shown in Fig. 1.4, which is used to support a gear. The shaft is made of steel and the modulus

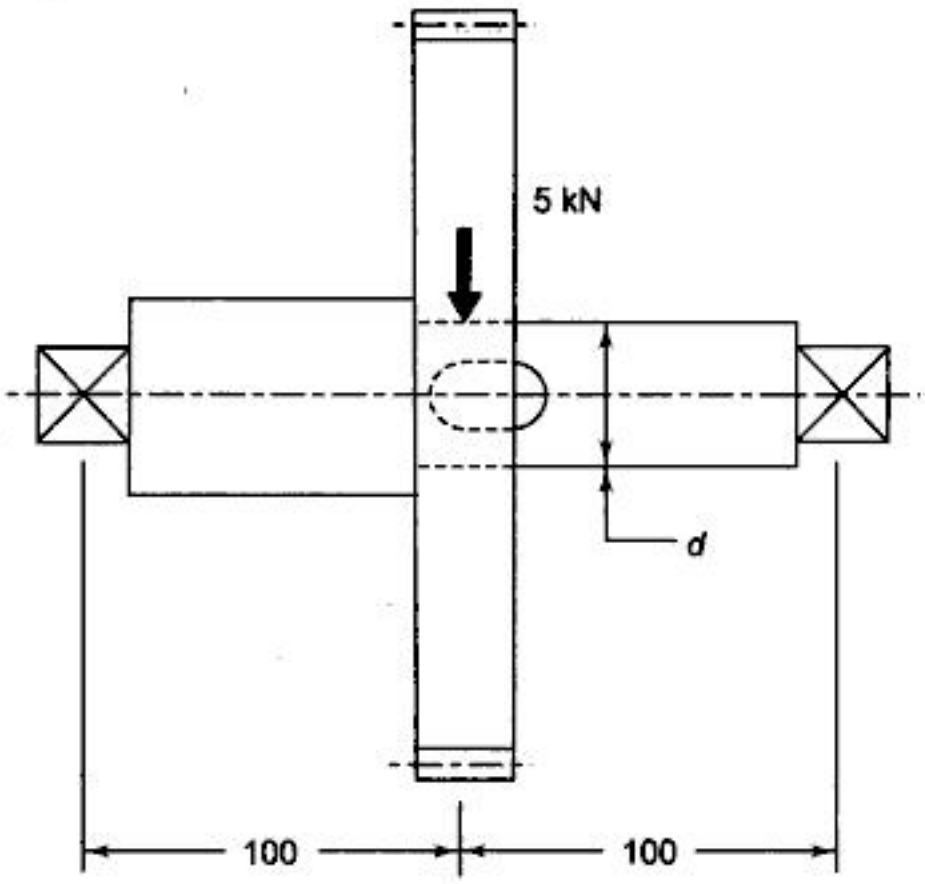

Fig. 1.4 *Transmission Shaft*

of elasticity is 207 000 N/mm^2. For proper meshing between gear teeth, the permissible deflection at the gear is limited to 0.05 mm. The deflection of the shaft at the centre is given by,

$$\delta = \frac{Pl^3}{48EI} \text{ or } 0.05 = \frac{(5 \times 10^3)(200)^3}{48(207\,000)\left(\frac{\pi d^4}{64}\right)}$$

Therefore,

$$d = 35.79 \text{ or } 40 \text{ mm}$$

The following observations are made from the above two examples:

(i) Failure mode for the tension rod is general yielding while elastic deflection is the failure criterion for the transmission shaft.

(ii) The permissible tensile stress for tension rod is obtained by dividing the yield strength by the factor of safety. Therefore, yield strength is the criterion of design. In case of a transmission shaft, lateral deflection or rigidity is the criterion of design. Therefore, modulus of elasticity is an important property for finding out the dimensions of the shaft.

Determination of geometric dimensions is an important step while designing machine elements. Various criteria such as yield strength, ultimate tensile strength, torsional or lateral deflection and permissible bearing pressure are used to find out these dimensions.

Step 6: Design Modifications

The geometric dimensions of the machine element are modified from assembly and manufacturing considerations. For example, the transmission shaft illustrated in Fig. 1.4 is provided with steps and shoulders for proper mounting of gear and bearings. Revised calculations are carried out for operating capacity, margin of safety at critical cross-sections and resultant stresses taking into consideration the effect of stress concentration. When these values differ from desired values, the dimensions of the component are modified. The process is continued till the desired values of operating capacity, factor of safety and stresses at critical cross-sections are obtained.

Step 7: Working Drawing

The last step in the design of machine elements is to prepare a working drawing of the machine element showing dimensions, tolerances, surface finish grades, geometric tolerances and special production requirements like heat treatment. The working drawing must be clear, concise and complete. It must have enough views and cross-sections to show all details. The main view of the machine element should show it in a position, it is required to occupy in service. Every dimension must be given. There should not be scope for guesswork and a necessity for scaling the drawing. All dimensions that are important for proper assembly and interchangeability must be provided with tolerances.

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1.5 TRADITIONAL DESIGN METHODS

There are two traditional methods of design—*design by craft evolution* and *design by drawing*. Bullock cart, rowing boat, plow and musical instruments are some of the products, which are produced by the craft-evolution process. The salient features of this age-old technique are as follows:

(i) The craftsmen do not prepare dimensioned drawings of their products. They cannot offer adequate justification for the designs they make.

(ii) These products are developed by trial and error over many centuries. Any modification in the product is costly, because the craftsman has to experiment with the product itself. Moreover, only one change at a time can be attempted and complete reorganization of the product is difficult.

(iii) The essential information of the product such as materials, dimensions of parts, manufacturing methods and assembly techniques is transmitted from place to place and time to time by two ways. First, the product, which basically remains unchanged, is the main source of information. The exact memory of the sequence of operations required to make the product is second source of information. There is no symbolic medium to record the design information of the product.

With all these weaknesses, the craft-evolution process has successfully developed some of the complex structures. The craft-evolution method has become obsolete due to two reasons. This method cannot adapt to sudden changes in requirement. Secondly, the product cannot be manufactured on a mass scale.

The essential features of design by drawing method are as follows:

(i) The dimensions of the product are specified in advance of its manufacture.

(ii) The complete manufacturing of the product can be subdivided into separate pieces, which can be made by different people. This division of work is not possible with craft-evolution.

(iii) When the product is to be developed by trial and error, the process is carried out on a drawing board instead of shop floor. The drawings of the product are modified and developed prior to manufacture.

In this method, much of the intellectual activity is taken away from the shop floor and assigned to design engineers.

1.6 DESIGN SYNTHESIS

Design synthesis is defined as the process of creating or selecting configurations, materials, shapes and dimensions for a product. It is a decision making process with the main objective of optimisation. There is a basic difference between design analysis and design synthesis. In design analysis, the designer assumes a particular mechanism, a particular material and mode of failure for the component. With the help of this information, he determines the dimensions of the product. However, design synthesis does not permit such assumptions. Here, the designer selects the optimum configuration from a number of alternative solutions. He decides the material for the component from a number of alternative materials. He determines the optimum shape and dimensions of the component on the basis of mathematical analysis.

In design synthesis, the designer has to fix the objective. The objective can be minimum cost, minimum weight or volume, maximum reliability or maximum life. The second step is mathematical formulation of these objectives and requirements. The final step is mathematical analysis for optimisation and interpretation of the results. In order to illustrate the process of design synthesis, let us consider a problem of designing cylindrical cans. The requirements are as follows:

(i) The cylindrical can is completely enclosed and the cost of its material should be minimum.

(ii) The cans are to be stored on a shelf and the dimensions of the shelf are such that the radius of the can should not exceed R_{max}.

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The following notations are used in the analysis:

r = radius of can
h = height of can
A = surface area of can
V = volume of can

Therefore,

$$A = 2\pi r^2 + 2\pi rh \quad \text{(a)}$$

$$V = \pi r^2 h \quad \text{(b)}$$

Substituting Eq. (b) in Eq. (a),

$$A = 2\pi r^2 + \frac{2V}{r} \quad \text{(c)}$$

For minimum cost of material of the can,

$$\frac{dA}{dr} = 0 \quad \text{or} \quad 4\pi r - \frac{2V}{r^2} = 0$$

or $$r = \left(\frac{V}{2\pi}\right)^{1/3}.$$

Let us call this radius as r_1 giving the condition of minimum material. Therefore,

$$r_1 = \left(\frac{V}{2\pi}\right)^{1/3} \quad \text{(d)}$$

In order to satisfy the second requirement,

$$0 < r < R_{max.} \quad \text{(e)}$$

In Eqs (d) and (e), r_1 and $R_{max.}$ are two independent variables and there will be two separate cases as shown in Fig. 1.5.

Case (a)

$$r_1 > R_{max.}$$

The optimum radius will be,

$$r = R_{max.} \quad \text{(i)}$$

Case (b)

$$r_1 < R_{max.}$$

The optimum radius will be

$$r = r_1 \quad \text{(ii)}$$

It is seen from the above example, that design synthesis begins with the statement of requirements, which are then converted into mathematical expressions and finally, equations are solved for optimisation.

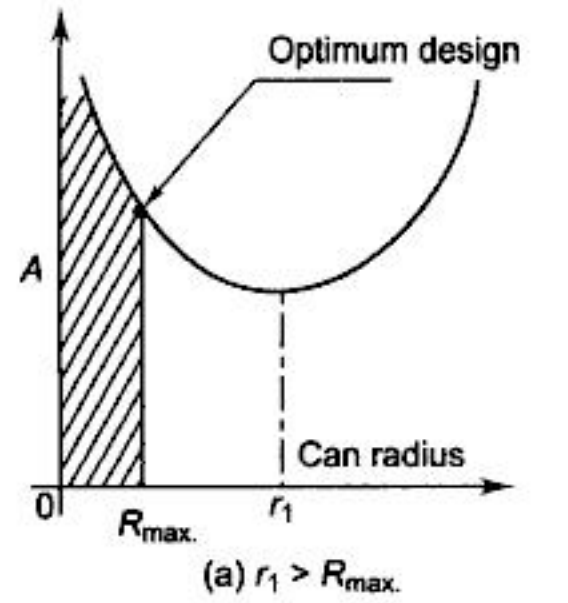

Fig. 1.5 *Optimum Solution to Can Radius*

1.7 USE OF STANDARDS IN DESIGN

Standardization is defined as obligatory norms, to which various characteristics of a product should conform. The characteristics include materials, dimensions and shape of the component, method of testing and method of marking, packing and storing of the product. The following standards are used in mechanical engineering design:

(i) Standards for Materials, their Chemical Compositions, Mechanical Properties and Heat Treatment For example, Indian standard IS 210 specifies seven grades of grey cast iron designated as FG 150, FG 200, FG 220, FG 260, FG 300, FG 350 and FG 400. The number indicates ultimate tensile strength in N/mm^2. IS 1570 (Part 4) specifies chemical composition of various grades of alloy steel. For example, alloy steel designated by 55Cr3 has 0.5–0.6% carbon, 0.10–0.35% silicon, 0.6–0.8% manganese and 0.6–0.8% chromium.

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(ii) Standards for Shapes and Dimensions of Commonly used Machine Elements The machine elements include bolts, screws and nuts, rivets, belts and chains, ball and roller bearings, wire ropes, keys and splines, etc. For example, IS 2494 (Part 1) specifies dimensions and shape of the cross-section of endless V-belts for power transmission. The dimensions of the trapezoidal cross-section of the belt, viz. width, height and included angle are specified in this standard. The dimensions of rotary shaft oil seal units are given in IS 5129 (Part 1). These dimensions include inner and outer diameters and width of oil seal units.

(iii) Standards for Fits, Tolerances and Surface Finish of Component For example, selection of the type of fit for different applications is illustrated in IS 2709 on 'Guide for selection of fits'. The tolerances or upper and lower limits for various sizes of holes and shafts are specified in IS 919 on 'Recommendations for limits and fits for engineering'. IS 10719 explains method for indicating surface texture on technical drawings. The method of showing geometrical tolerances is explained in IS 8000 on 'Geometrical tolerancing on technical drawings'.

(iv) Standards for Testing of Products These standards, sometimes called 'codes', give procedures to test the products such as pressure vessel, boiler, crane and wire rope, where safety of the operator is an important consideration. For example, IS 807 is a code of practice for design, manufacture, erection and testing of cranes and hoists. The method of testing of pressure vessels is explained in IS 2825 on 'Code for unfired pressure vessels'.

(v) Standards for Engineering Drawing of Components For example, there is a special publication SP46 prepared by Bureau of Indian Standards on 'Engineering Drawing Practice for Schools and Colleges' which covers all standards related to engineering drawing.

There are two words—standard and code—which are often used in standards. *A standard is defined as a set of specifications for parts, materials or processes*. The objective of a standard is to reduce the variety and limit the number of items to a reasonable level. On the other hand, *a code is defined as a set of specifications for the analysis, design, manufacture, testing and erection of the product*. The purpose of a code is to achieve a specified level of safety.

There are three types of standards used in design office. They are as follows:

(i) Company standards They are used in a particular company or a group of sister concerns.

(ii) National standards These are the IS (Bureau of Indian Standards), DIN (German), AISI or SAE (USA) or BS (UK) standards.

(iii) International standards These are prepared by the International Standards Organization (ISO).

Standardization offers the following advantages:

(a) The reduction in types and dimensions of identical components to a rational number makes it possible to manufacture the standard component on a mass scale in a centralised process. For example, a specialised factory like Associated Bearing Company (SKF) manufactures ball and roller bearings, which are required by all engineering industries. Manufacture of a standard component on mass production basis reduces the cost.

(b) Since the standard component is manufactured by a specialised factory, it relieves the machine-building plant of the laborious work of manufacturing that part. Availability of standard components like bearings, seals, knobs, wheels, roller chains, belts, hydraulic cylinders and valves has considerably reduced the manufacturing facilities required by the individual organisation.

(c) Standard parts are easy to replace when worn out due to interchangeability. This facilitates servicing and maintenance of machines. Availability of standard spare parts is always assured. The work of servicing and maintenance can be carried out even at an ordinary service station. These factors reduce the maintenance cost of machines.

(d) The application of standard machine elements and especially the standard units

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(e.g. couplings, cocks, pumps, pressure reducing valves and electric motors) reduce the time and effort needed to design a new machine. It is no longer necessary to design, manufacture and test these elements and units, and all that the designer has to do is to select them from the manufacturer's catalogues. On the other hand, enormous amount of work would be required to design a machine if all the screws, bolts, nuts, bearings, etc., had to be designed anew each time. Standardization results in substantial saving in the designer's effort.

(e) The standards of specifications and testing procedures of machine elements improve their quality and reliability. Standard components like SKF bearings, Dunlop belts or Diamond chains have a long-standing reputation for their reliability in engineering industries. Use of standard components improves the quality and reliability of the machine to be designed.

In design, the aim is to use as many standard components as possible for a given machine. The selection of standard parts in no way restricts the creative initiative of the designer and does not prevent him from finding better and more rational solutions.

1.8 SELECTION OF PREFERRED SIZES

In engineering design, many a times, the designer has to specify the size of the product. The '*size*' of the product is a general term, which includes different parameters like power transmitting capacity, load carrying capacity, speed, dimensions of the component such as height, length and width, and volume or weight of the product. These parameters are expressed numerically, e.g., 5 kW, 10 kN or 1000 rpm. Often, the product is manufactured in different sizes or models; for instance, a company may be manufacturing seven different models of electric motors ranging from 0.5 to 50 kW to cater to the need of different customers. Preferred numbers are used to specify the 'sizes' of the product in these cases.

[2] IS 1076–1985: Preferred Numbers (in three parts).

French balloonist and engineer *Charles Renard* first introduced preferred numbers in the 19th century. The system is based on the use of geometric progression to develop a set of numbers. There are five basic series[2], denoted as R5, R10, R20, R40 and R80 series, which increase in steps of 58%, 26%, 12%, 6%, and 3%, respectively. Each series has its own *series factor*. The series factors are given in Table 1.1.

Table 1.1 *Series factors*

R5 Series	$\sqrt[5]{10} = 1.58$
R10 Series	$\sqrt[10]{10} = 1.26$
R20 Series	$\sqrt[20]{10} = 1.12$
R40 Series	$\sqrt[40]{10} = 1.06$
R80 Series	$\sqrt[80]{10} = 1.03$

The series is established by taking the first number and multiplying it by a series factor to get the second number. The second number is again multiplied by a series factor to get the third number. This procedure is continued until the complete series is built up. The resultant numbers are rounded and shown in Table 1.2. As an example, consider a manufacturer of lifting tackles who wants to introduce nine different models of capacities ranging from about 15 to 100 kN. Referring to the R10 series, the capacities of different models of the lifting tackle will be 16, 20, 25, 31.5, 40, 50, 63, 80 and 100 kN.

Table 1.2 *Preferred numbers*

R5	*R10*	*R20*	*R40*
1.00	1.00	1.00	1.00
			1.06
		1.12	1.12
			1.18
	1.25	1.25	1.25
			1.32
		1.40	1.40
			1.50

(*Contd*)

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Table 1.2 *Contd*

R5	*R10*	*R20*	*R40*
1.60	1.60	1.60	1.60
			1.70
		1.80	1.80
			1.90
	2.00	2.00	2.00
			2.12
		2.24	2.24
			2.36
2.50	2.50	2.50	2.50
			2.65
		2.80	2.80
			3.00
	3.15	3.15	3.15
			3.35
		3.55	3.55
			3.75
4.00	4.00	4.00	4.00
			4.25
		4.50	4.50
			4.75
	5.00	5.00	5.00
			5.30
		5.60	5.60
			6.00
6.30	6.30	6.30	6.30
			6.70
		7.10	7.10
			7.50
	8.00	8.00	8.00
			8.50
		9.00	9.00
			9.50
10.00	10.00	10.00	10.00

It is observed from Table 1.2 that small sizes differ from each other by small amounts, while large sizes by large amounts. In the initial stages, the product is manufactured in a limited quantity and use is made of the R5 series. As the scale of production is increased, a change over is made from R5 to R10 series, introducing new sizes of intermediate values of R10 series. Preferred numbers minimise unnecessary variation in sizes. They assist the designer in avoiding selection of sizes in an arbitrary manner. The complete range is covered by minimum number of sizes, which is advantageous to the producer and consumer.

There are two terms, namely, '*basic series*' and '*derived series*', which are frequently used in relation to preferred numbers. R5, R10, R20, R40 and R80 are called *basic series*. Any series that is formed on the basis of these five basic series is called *derived series*. In other words, derived series are derived from basic series. There are two methods of forming derived series, namely, reducing the numbers of a particular basic series or increasing the numbers.

In the first method, a derived series is obtained by taking every second, third, fourth or *p*th term of a given basic series. Such a derived series is designated by the symbol of the basic series followed by the number 2, 3, 4 or *p* and separated by '/' sign. If the series is limited, the designation also includes the limits inside the bracket. If the series is unlimited, at least one of the numbers of that series is mentioned inside the bracket. Let us consider the meaning of these designations.

(i) Series R 10/3 (1, … ,1000) indicates a derived series comprising of every third term of the R10 series and having the lower limit as 1 and higher limit as 1000.

(ii) Series R 20/4 (…, 8, …) indicates a derived series comprising of every fourth term of the R20 series, unlimited in both sides and having the number 8 inside the series.

(iii) Series R 20/3 (200, …) indicates a derived series comprising of every third term of the R20 series and having the lower limit as 200 and without any higher limit.

(iv) Series R 20/3 (…200) indicates a derived series comprising of every third term of the R20 series and having the higher limit as 200 and without any lower limit.

In the second method, the derived series is obtained by increasing the numbers of a particular basic series. Let us consider an example of a derived series of numbers ranging from 1 to 1000 based on the R5 series. From Table 1.2, the

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numbers belonging to the R5 series from 1 to 10 are as follows:

1, 1.6, 2.5, 4, 6.3, 10

The next numbers are obtained by multiplying the above numbers by 10. They are as follows:

16, 25, 40, 63, 100

The same procedure is repeated and the next numbers are obtained by multiplying the above numbers by 10.

160, 250, 400, 630, 1000

Therefore, the complete derived series on the basis of R5 series is as follows:

1, 1.6, 2.5, 4, 6.3, 10, 16, 25, 40, 63, 100, 160, 250, 400, 630, 1000

The advantage of derived series is that one can obtain geometric series for any range of numbers, that is, with any value of the first and the last numbers. Also, one can have any intermediate numbers between these two limits.

Example 1.1 *Find out the numbers of the R5 basic series from 1 to 10.*

Solution

Step I *Calculation of series factor*

The series factor for the R5 series is given by

$$\sqrt[5]{10} = 1.5849$$

Step II *Calculation of numbers*

The series R5 is established by taking the first number and multiplying it by a series factor to get the second number. The second number is again multiplied by a series factor to get the third number. This procedure is continued until the complete series is built up. The numbers thus obtained are rounded.

First number = 1

Second number = 1 (1.5849) = 1.5849 = (1.6)

Third number = $(1.5849)(1.5849) = (1.5849)^2$
$= 2.51 = (2.5)$

Fourth number = $(1.5849)^2(1.5849) = (1.5849)^3$
$= 3.98 = (4)$

Fifth number = $(1.5849)^3(1.5849) = (1.5849)^4$
$= (6.3)$

Sixth number = $(1.5849)^4(1.5849) = (1.5849)^5$
$= (10)$

In above calculations, the rounded numbers are shown in brackets.

Example 1.2 *Find out the numbers of R20/4(100, ..., 1000) derived series.*

Solution

Step I *Calculation of series factor*

The series factor for the R20 series is given by

$$\sqrt[20]{10} = 1.122$$

Step II *Calculation of ratio factor*

Since every fourth term of the R20 series is selected, the ratio factor (ϕ) is given by,

$$\phi = (1.122)^4 = 1.5848$$

Step III *Calculation of numbers*

First number = 100

Second number = 100(1.5848) = 158.48 = (160)

Third number = $100(1.5848)(1.5848) = 100(1.5848)^2$
$= 251.16 = (250)$

Fourth number = $100(1.5848)^2(1.5848)$
$= 100(1.5848)^3 = 398.04 = (400)$

Fifth number = $100(1.5848)^3(1.5848)$
$= 100(1.5848)^4 = 630.81 = (630)$

Sixth number = $100(1.5848)^4(1.5848)$
$= 100(1.5848)^5 = 999.71 = (1000)$

In the above calculations, the rounded numbers are shown in brackets. The complete series is given by

100, 160, 250, 400, 630 and 1000

Example 1.3 *A manufacturer is interested in starting a business with five different models of tractors ranging from 7.5 to 75 kW capacities. Specify power capacities of the models. There is an expansion plan to further increase the number of models from five to nine to fulfill the requirement of farmers. Specify the power capacities of the additional models.*

Solution

Part I *Starting Plan*

Step I *Calculation of ratio factor*

Let us denote the ratio factor as (ϕ). The derived series is based on geometric progression. The power rating of five models will as follows,

$7.5(\phi)^0$, $7.5(\phi)^1$, $7.5(\phi)^2$, $7.5(\phi)^3$ and $7.5(\phi)^4$

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The maximum power rating is 75 kW. Therefore,

$$7.5(\phi)^4 = 75 \quad \text{or} \quad \phi = \left(\frac{75}{7.5}\right)^{1/4}$$
$$= (10)^{1/4} = \sqrt[4]{10} = 1.7783$$

Step II *Power rating of models*

Rating of first model = (7.5) kW

Rating of second model = 7.5(1.7783) = 13.34
= (13) kW

Rating of third model = 7.5(1.7783)^2 = 23.72
= (24) kW

Rating of fourth model = 7.5(1.7783)^3 = 42.18
= (42) kW

Rating of fifth model = 7.5(1.7783)^4 = 75.0
= (75) kW

Part II *Expansion Plan*

Step III *Calculation of ratio factor*

When the number of models is increased to nine, the power rating of nine models will be as follows:

$7.5(\phi)^0$, $7.5(\phi)^1$, $7.5(\phi)^2$, $7.5(\phi)^3$, $7.5(\phi)^4$, ..., $7.5(\phi)^8$

The maximum power rating is 75 kW. Therefore,

$$7.5(\phi)^8 = 75 \quad \text{or} \quad \phi = \left(\frac{75}{7.5}\right)^{1/8}$$
$$= (10)^{1/8} = 1.3335$$

Step IV *Power rating of models*

The power rating of the nine models will be as follows:

First model = 7.5 $(1.3335)^0$ = (7.5) kW

Second model = 7.5 $(1.3335)^1$ = 10.00 = (10) kW

Third model = 7.5 $(1.3335)^2$ = 13.34 = (13) kW

Fourth model = 7.5 $(1.3335)^3$ = 17.78 = (18) kW

Fifth model = 7.5 $(1.3335)^4$ = 23.72 = (24) kW

Sixth model = 7.5 $(1.3335)^5$ = 31.62 = (32) kW

Seventh model = 7.5 $(1.3335)^6$ = 42.17 = (42) kW

Eighth model = 7.5 $(1.3335)^7$ = 56.24 = (56) kW

Ninth model = 7.5 $(1.3335)^8$ = 74.99 = (75) kW

Part III *Power capacities of additional models*

It is observed that there are four additional models having power ratings as 10, 18, 32 and 56 kW.

Example 1.4 *It is required to standardize eleven shafts from 100 to 1000 mm diameter. Specify their diameters.*

Solution

Step I *Calculation of ratio factor*

The diameters of shafts will be as follows:

$100(\phi)^0$, $100(\phi)^1$, $100(\phi)^2$, $100(\phi)^3$, ..., $100(\phi)^{10}$

The maximum diameter is 1000 mm. Therefore,

$$100(\phi)^{10} = 1000 \quad \text{or} \quad \phi = \left(\frac{1000}{100}\right)^{1/10}$$
$$= (10)^{1/10} = \sqrt[10]{10}$$

Therefore the diameters belong to the R10 series.

Step II *Calculation of shaft diameters*

Since the minimum diameter is 100 mm, the values of the R10 series given in Table 1.2 are multiplied by 100. The diameter series is written as follows:

100, 125, 160, 200, 250, 315, 400, 500, 630, 800 and 1000 mm

1.9 AESTHETIC CONSIDERATIONS IN DESIGN

Each product has a definite purpose. It has to perform specific functions to the satisfaction of customer. The contact between the product and the people arises due to the sheer necessity of this functional requirement. The functional requirement of an automobile car is to carry four passengers at a speed of 60 km/hr. There are people in cities who want to go to their office at a distance of 15 km in 15 minutes. So they purchase a car. The specific function of a domestic refrigerator is to preserve vegetables and fruits for a week. There is a housewife in the city who cannot go to the market daily and purchase fresh vegetables. Therefore, she purchases the refrigerator. It is seen that such functional requirements bring products and people together.

However, when there are a number of products in the market having the same qualities of efficiency, durability and cost, the customer is attracted towards the most appealing product.

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External appearance is an important feature, which not only gives grace and lustre to the product but also dominates sale in the market. This is particularly true for consumer durables like automobiles, household appliances and audiovisual equipment.

The growing realisation of the need of aesthetic considerations in product design has given rise to a separate discipline known as '*industrial design*'. The job of an industrial designer is to create new forms and shapes, which are aesthetically pleasing. The industrial designer has, therefore, become the fashion maker in hardware.

Like in fashion, the outward appearance of a product undergoes many changes over the years. There are five basic forms—step, stream, taper, shear and sculpture. The *step form* is similar to the shape of a 'skyscraper' or multistorey building. This involves shapes with a vertical accent rather than a horizontal. The *stream* or *streamline form* is seen in automobiles and aeroplane structures. The *taper form* consists of tapered blocks interlocked with tapered plinths or cylinders. The *shear form* has a square outlook, which is suitable for free-standing engineering products. The *sculpture form* consists of ellipsoids, paraboloids and hyperboloids. The sculpture and stream forms are suitable for mobile products like vehicles, while step and shear forms are suitable for stationary products.

There is a relationship between functional requirement and appearance of the product. In many cases, functional requirements result in shapes which are aesthetically pleasing. The evolution of the streamlined shape of the Boeing is the result of studies in aerodynamics for effortless speed. The robust outlook and sound proportions of a high-capacity hydraulic press are the results of requirements like rigidity and strength. The objective of chromium plating of the parts of household appliances is corrosion resistance rather than pleasing appearance.

Selection of proper colour is an important consideration in product aesthetics. The choice of colour should be compatible with the conventional ideas of the operator. Many colours are associated with different moods and conditions. Morgan has suggested the meaning of colours that are given in Table 1.3.

Table 1.3 *Meaning of colour*

Colour	*Meaning*
Red	Danger-Hazard-Hot
Orange	Possible danger
Yellow	Caution
Green	Safety
Blue	Caution-Cold
Grey	Dull

The external appearance of the product does not depend upon only the two factors of form and colour. It is a cumulative effect of a number of factors such as rigidity and resilience, tolerances and surface finish, motion of individual components, materials, manufacturing methods and noise. The industrial designer should select a form which is in harmony with the functional requirements of the product. The economics and availability of surface-treating processes like anodizing, plating, blackening and painting should be taken into account before finalising the external appearance of the product.

1.10 ERGONOMIC CONSIDERATIONS IN DESIGN

Ergonomics is defined as the relationship between man and machine and the application of anatomical, physiological and psychological principles to solve the problems arising from man–machine relationship. The word 'ergonomics' is coined from two Greek words—'*ergon*', which means 'work' and '*nomos*', which means 'natural laws'. Ergonomics means the natural laws of work. From design considerations, the topics of ergonomic studies are as follows:

(i) Anatomical factors in the design of a driver's seat
(ii) Layout of instrument dials and display panels for accurate perception by the operators
(iii) Design of hand levers and hand wheels
(iv) Energy expenditure in hand and foot operations

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(v) Lighting, noise and climatic conditions in machine environment

Ergonomists have carried out experiments to determine the best dimensions of a driver's seat, the most convenient hand or foot pressure or dimensions of levers and hand wheels.

The machine is considered as an entity in itself in machine design. However, ergonomists consider a man–machine joint system, forming a closed loop as shown in Fig. 1.6. From display instruments, the operator gets the information about the operations of the machine. If he feels that a correction is necessary, he will operate the levers or controls. This, in turn, will alter the performance of the machine, which will be indicated on display panels. The contact between man and machine in this closed-loop system arises at two places—display instruments, which give information to the operator, and controls with which the operator adjusts the machine.

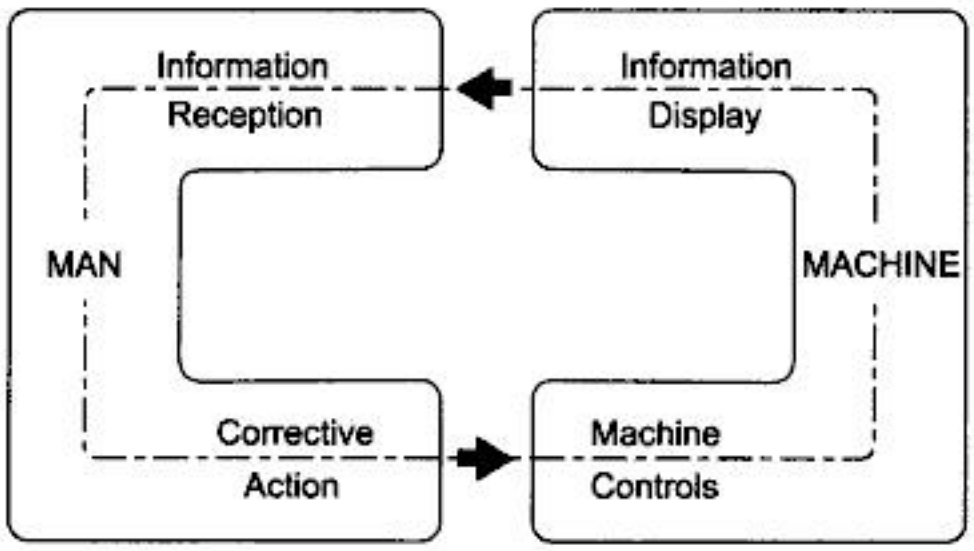

Fig. 1.6 *Man–Machine Closed-Loop System*

The visual display instruments are classified into three groups:

(i) Displays giving quantitative measurements, such as speedometer, voltmeter or energy meter

(ii) Displays giving the state of affairs, such as the red lamp indicator

(iii) Displays indicating predetermined settings, e.g., a lever which can be set at 1440 rpm, 720 rpm or 'off' position for a two-speed electric motor.

Moving scale or dial-type instruments are used for quantitative measurements, while lever-type indicators are used for setting purposes. The basic objective behind the design of displays is to minimise fatigue to the operator, who has to observe them continuously. The ergonomic considerations in the design of displays are as follows:

(i) The scale on the dial indicator should be divided in suitable numerical progression like 0–10–20–30 and not 0–5–30–55.

(ii) The number of subdivisions between numbered divisions should be minimum.

(iii) The size of letters or numbers on the indicator should be as follows:

$$\text{Height of letter or number} \geq \frac{\text{Reading distance}}{200}$$

(iv) Vertical figures should be used for stationary dials, while radially oriented figures are suitable for rotating dials.

(v) The pointer should have a knife-edge with a mirror in the dial to minimise parallax error.

The controls used to operate the machines consist of levers, cranks, hand wheels, knobs, switches, push buttons and pedals. Most of them are hand operated. When a large force is required to operate the controls, levers and hand wheels are used. When the operating forces are light, push buttons or knobs are preferred. The ergonomic considerations in the design of controls are as follows:

(i) The controls should be easily accessible and logically positioned. The control operation should involve minimum motions and avoid awkward movements.

(ii) The shape of the control component, which comes in contact with hands, should be in conformity with the anatomy of human hands.

(iii) Proper colour produces beneficial psychological effects. The controls should be painted in red colour in the grey background of machine tools to call for attention.

The aim of ergonomics is to reduce the operational difficulties present in a man–machine joint system, and thereby reduce the resulting physical and mental stresses.

The shape and dimensions of certain machine elements like levers, cranks and hand wheels are decided on the basis of ergonomic studies. The resisting force, i.e., the force exerted by the operator without undue fatigue is also obtained by

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ergonomic considerations. Ergonomic textbooks[3,4] give exhaustive details of the dimensions and resisting forces of different control elements. In this article, we will restrict the consideration to levers, cranks and hand wheels, which are frequently required in machine design. The nomenclature for the dimensions of lever, crank and hand wheel are shown in Fig. 1.7. Their dimensions and magnitude of resisting force are given in Tables 1.4 to 1.6.

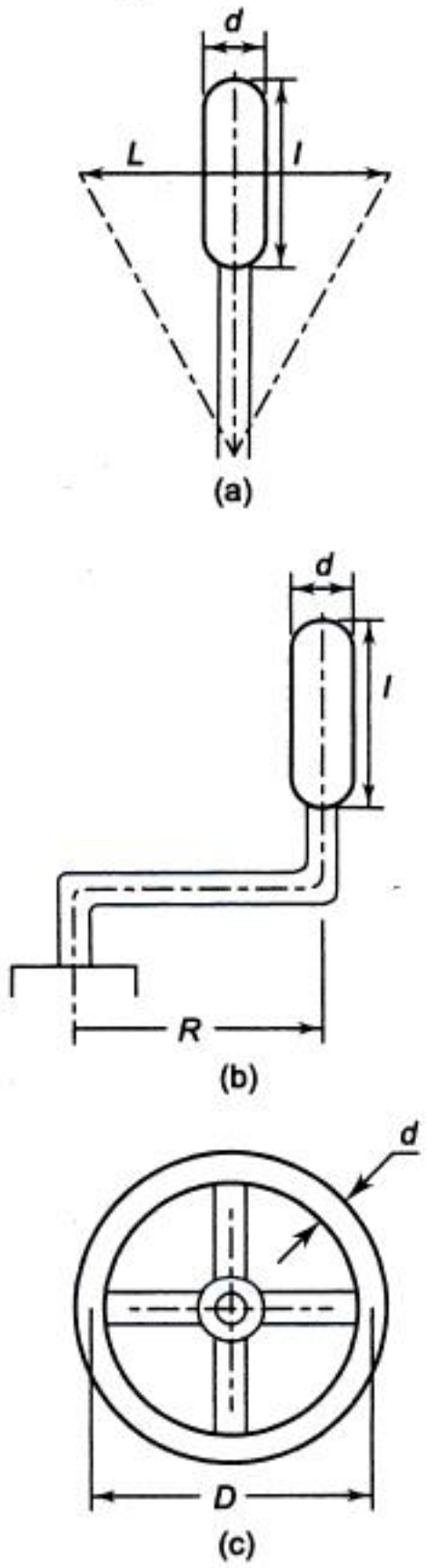

Fig. 1.7 *Control Elements: (a) Lever (b) Crank (c) Hand Wheel*

Table 1.4 *Dimensions and resisting force for lever*

	d	*l*	*L*	*P*
Minimum	40	75	–	–
Maximum	70	–	950	90

d = handle diameter (mm)
l = grasp length (mm)
L = displacement of lever (mm)
P = resisting force (N)

Table 1.5 *Dimensions and resisting force for crank with heavy load (more than 25 N)*

	d	*l*	*R*	*P*
Minimum	25	75	125	–
Maximum	75	–	500	40

d = handle diameter (mm)
l = grasp length (mm)
R = crank radius (mm)
P = resisting force (N)

Table 1.6 *Dimensions and resisting force for hand wheel*

	D	*d*	*P*
Minimum	175	20	–
Maximum	500	50	240

D = mean diameter of hand wheel (mm)
d = rim diameter (mm)
P = resisting force with two hands (N)

1.11 CONCURRENT ENGINEERING

Conventional design process is sequential, where the main activities are executed in a sequence as shown in Fig. 1.8. It begins with market survey, with the objective of finding out the requirements of the customer. This information is then handed over to the design department in the form of a '*product brief*'. The design department prepares the design and makes a few prototype samples for testing. The assembly and detail drawings are then prepared and passed on to the production department for their approval. Usually, the

[3] B.M. Pulat—*Fundamentals of Industrial Ergonomics*—Prentice Hall Inc.—1992.
[4] Hywel Murell—*Ergonomics—Man in his Working Environment*'—Chapman and Hall—1986.

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production department suggests changes in design from manufacturing considerations and the drawings are sent back to the design department. This process of sending the drawings by the design department to the production department and from the production department back to the design department continues till the design is finalised and in between valuable time is lost. Engineers who design new products and manufacturing personnel who have to figure out how to make the product are often at odds because of their different backgrounds and points of view. Many times, designers are creative artists who overlook the capabilities of the plant's machinery. Manufacturing engineers on the other hand, are realistic. They prefer standard materials, simple manufacturing methods, standard components and as few of them as possible. Due to this difference in background, many a times the designer comes up with a new product and throws it 'over the wall' to the manufacturing personnel, who says "We can't make this". Then there is a lot of finger pointing. In the *sequential design process*, the production department suggests a number of design modifications, after the design is finalised. In past, this has resulted in time consuming re-designs and missed time schedules in a number of projects. Personality conflicts and departmental barriers often create more problems. Due to these reasons, sequential design is often called 'over the wall' design.

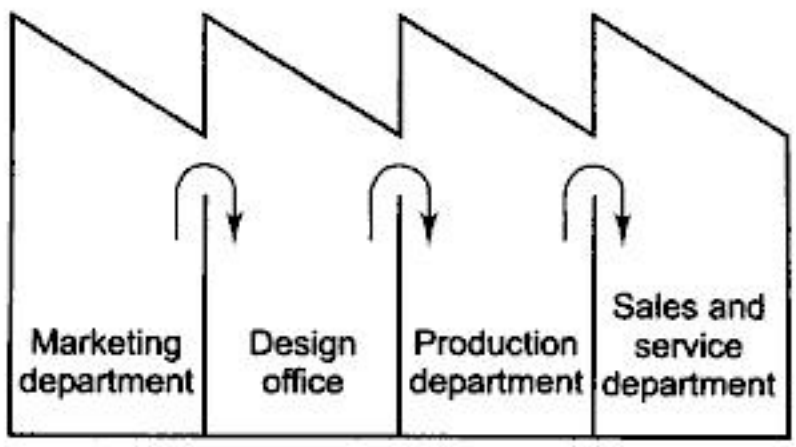

Fig. 1.8 *Sequential Design Process*

In recent years, there is a fundamental shift in the way the designs are prepared. The sequential design process is being replaced by *simultaneous* or *concurrent engineering*, where various activities are carried out in parallel, instead of in series. The trend is to bring the design and manufacturing activities together as a single engineering discipline.

Concurrent engineering is defined as the design process that brings both design and manufacturing engineers together during the early phases of design process. In this process, a team of specialists examines the design from different angles as shown in Fig. 1.9. The specialists include a manufacturing engineer, tool engineer, field personnel, reliability engineer and safety engineer. They consider various aspects of the product such as feasibility, manufacturability, assembly, testability, performance, reliability, maintainability, safety and cost. All these aspects are simultaneously considered early in the design stage. For example, manufacturing and assembly is simultaneously considered with stress analysis. This results in smaller number of modifications in the design at a later stage and reduces the 'time interval' from the conceptual stage to the marketing stage.

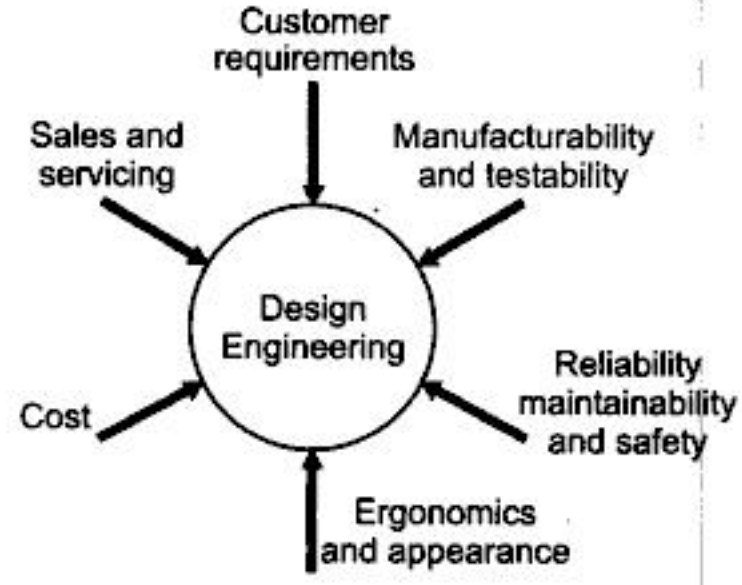

Fig. 1.9 *Simultaneous Design Process*

An example of a company making measuring instruments is interesting[5]. Keithley Instruments, Solon, USA applied the concept of concurrent engineering in development of a digital multi-meter. The number of parts in the new instrument were reduced from 131 to 76, the number of assembly screws were reduced from 30 to 8 and assembly time was reduced by 35% requiring only one screwdriver instead of multiple assembly tools.

[5] 'Design for Manufacture: key to success'—Machine Design—Vol. 64, No. 10—May, 1992.

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Short-Answer Questions

1.1 Define machine design.
1.2 What is the final outcome of a machine design process?
1.3 Name the various requirements of a product giving suitable example.
1.4 What are the basic requirements of a machine element?
1.5 What are the steps involved in design of a machine element?
1.6 Define design synthesis.
1.7 Distinguish between design synthesis and design analysis.
1.8 What is standardization?
1.9 What are the three basic types of standards used in a design office?
1.10 What do you understand by size of a product? Give examples.
1.11 What are preferred numbers?
1.12 How many basic series are used? How will you denote them?
1.13 What is a derived series?
1.14 How will you form a derived series?
1.15 What is industrial design?
1.16 Define ergonomics.
1.17 Explain man–machine joint system.
1.18 What is concurrent engineering?
1.19 Distinguish between sequential design and concurrent engineering.

Problems for Practice

1.1 Find out the numbers of R10 basic series from 1 to 10.
1.2 Find out the numbers of R20/3 (200,...) derived series.
[200, 280(282.5), 400(399.03), 560(563.63), 800(796.13), 1120(1124.53), ...] ($\phi = 1.4125$)
1.3 It is required to standardise load-carrying capacities of dumpers in a manufacturing unit. The maximum and minimum capacities of such dumpers are 40 and 630 kN, respectively. The company is interested in developing seven models in this range. Specify their load carrying capacities.
[40, 63(63.33), 100(100.26), 160(158.73), 250(251.31), 400(397.87), 630(629.90)] ($\phi = 1.5832$)
1.4 It is required to standardise 11 speeds from 72 to 720 rpm for a machine tool. Specify the speeds.
[72, 90.64, 114.11, 143.65, 180.84, 227.66, 286.60, 360.80, 454.22, 571.81, 719.85 rpm]

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Engineering Materials

Chapter 2

2.1 STRESS–STRAIN DIAGRAMS

A very useful information concerning the behaviour of material and its usefulness for engineering applications can be obtained by making a tension test and plotting a curve showing the variation of stress with respect to strain. A *tension test* is one of the simplest and basic tests and determines values of number of parameters concerned with mechanical properties of materials like strength, ductility and toughness. The following information can be obtained from a tension test:

(i) Proportional limit
(ii) Elastic limit
(iii) Modulus of elasticity
(iv) Yield strength
(v) Ultimate tension strength
(vi) Modulus of resilience
(vii) Modulus of toughness
(viii) Percentage elongation
(ix) Percentage reduction in area

The specimen used in a tension test is illustrated in Fig. 2.1. The shape and dimensions of this specimen are standardised. They should conform to IS 1608 : 1972[1]. The cross-section of the specimen can be circular, square or rectangular. The standard gauge length l_0 is given by,

$$l_0 = 5.65\sqrt{A_0},$$

where A_0 is the cross-sectional area of the specimen.

For circular cross-section,

$$l_0 \approx 5d_0$$

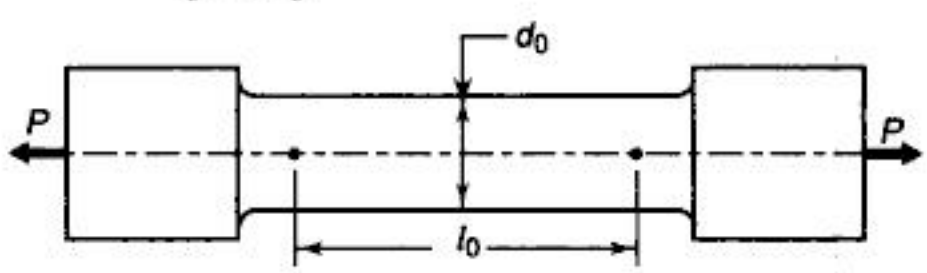

Fig. 2.1 *Specimen of Tension-test*

In a tension test, the specimen is subjected to axial tension force, which is gradually increased and the corresponding deformation is measured. Initially, the gauge length is marked on the specimen and initial dimensions d_0 and l_0 are measured before starting the test. The specimen is then mounted on the machine and gripped in the jaws. It is then subjected to an axial tension force, which is increased by suitable increments. After each increment, the amount by which the gauge length l_0 increases, i.e., deformation of gauge length, is measured by an extensometer. The procedure of measuring the tension force and corresponding deformation is continued till fracture

[1] IS 1608–1972: Method for tensile testing of steel products.

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occurs and the specimen is broken into two pieces. The tensile force divided by the original cross-sectional area of the specimen gives stress, while the deformation divided by gauge length gives the strain in the specimen.

Therefore, the results of a tension test are expressed by means of stress–strain relationship and plotted in the form of a graph. A typical stress–strain diagram for ductile materials like mild steel is shown in Fig. 2.2. The following properties of a material can be obtained from this diagram:

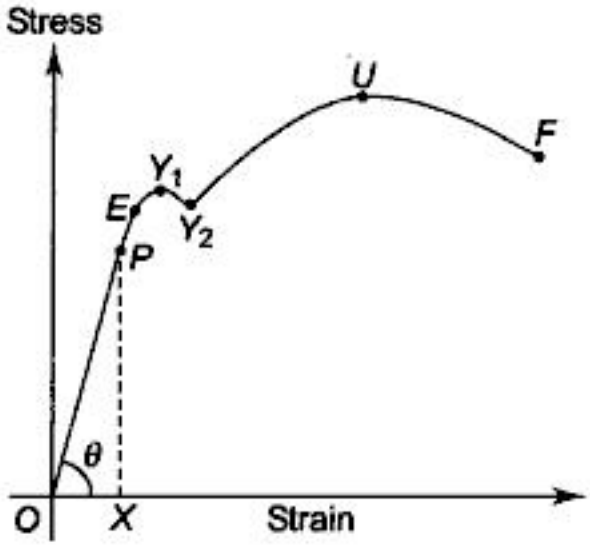

Fig. 2.2 *Stress–Strain Diagram of Ductile Materials*

(i) Proportional Limit It is observed from the diagram that stress–strain relationship is linear from the point O to P. OP is a straight line and after the point P, the curve begins to deviate from the straight line. *Hooke's law* states that stress is directly proportional to strain. Therefore, it is applicable only up to the point P. *The term proportional limit is defined as the stress at which the stress–strain curve begins to deviate from the straight line.* Point P indicates the proportional limit.

(ii) Modulus of Elasticity *The modulus of elasticity or Young's modulus (E) is the ratio of stress to strain up to the point P.* It is given by the slope of the line OP. Therefore,

$$E = \tan\theta = \frac{PX}{OX} = \frac{\text{stress}}{\text{strain}}$$

(iii) Elastic Limit Even if the specimen is stressed beyond the point P and up to the point E, it will regain its initial size and shape when the load is removed. This indicates that the material is in elastic stage up to the point E. Therefore, E is called the elastic limit. *The elastic limit of the material is defined as the maximum stress without any permanent deformation.*

The proportional limit and elastic limit are very close to each other, and it is difficult to distinguish between points P and E on the stress–strain diagram. In practice, many times, these two limits are taken to be equal.

(iv) Yield Strength When the specimen is stressed beyond the point E, plastic deformation occurs and the material starts yielding. During this stage, it is not possible to recover the initial size and shape of the specimen on the removal of the load. It is seen from the diagram that beyond the point E, the strain increases at a faster rate up to the point Y_1. In other words, there is an appreciable increase in strain without much increase in stress. In case of mild steel, it is observed that there is a small reduction in load and the curve drops down to the point Y_2 immediately after yielding starts. The points Y_1 and Y_2 are called the upper and lower yield points, respectively. For many materials, the points Y_1 and Y_2 are very close to each other and in such cases, the two points are considered as same and denoted by Y. The stress corresponding to the yield point Y is called the yield strength. *The yield strength is defined as the maximum stress at which a marked increase in elongation occurs without increase in the load.*

Many varieties of steel, especially heat-treated steels and cold-drawn steels, do not have a well-defined yield point on the stress–strain diagram. As shown in Fig. 2.3, the material yields gradually after passing through the elastic limit E. If the loading is stopped at the point Y, at a stress level slightly higher than the elastic limit E, and the specimen is unloaded and readings taken, the curve would follow the dotted line and a permanent set or plastic deformation will exist. The strain corresponding to this permanent deformation is indicated by OA. *For such materials, which do not exhibit a well-defined yield point, the yield strength is defined as the stress corresponding to*

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a permanent set of 0.2% of gauge length. In such cases, the yield strength is determined by the *offset method*. A distance *OA* equal to 0.002 mm/mm strain (corresponding to 0.2% of gauge length) is marked on the *X*-axis. A line is constructed from the point *A* parallel to the straight line portion *OP* of the stress–strain curve. The point of intersection of this line and the stress–strain curve is called *Y* or the yield point and the corresponding stress is called 0.2% yield strength.

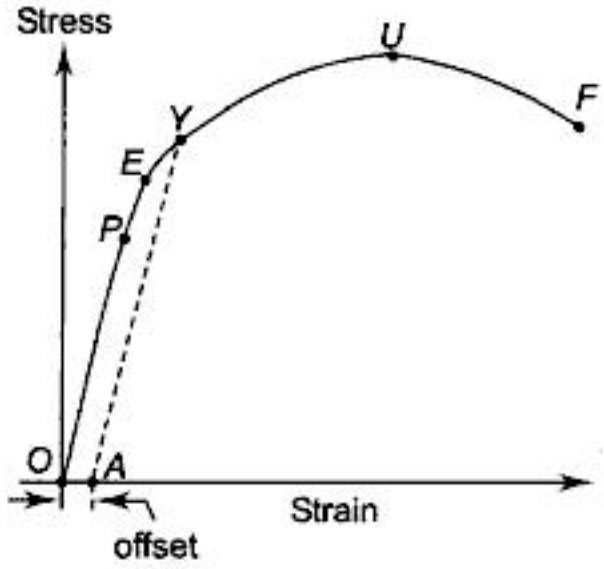

Fig. 2.3 *Yield Stress by Offset Method*

The terms *proof load* or *proof strength* are frequently used in the design of fasteners. Proof strength is similar to yield strength. It is determined by the offset method; however the offset in this case is 0.001 mm/mm corresponding to a permanent set of 0.1% of gauge length. *0.1% Proof strength, denoted by symbol Rp0.1, is defined as the stress which will produce a permanent extension of 0.1% in the gauge length of the test specimen*. The proof load is the force corresponding to proof stress.

(v) Ultimate Tensile Strength We will refer back to the stress–strain diagram of ductile materials illustrated in Fig. 2.2. After the yield point Y_2, plastic deformation of the specimen increases. The material becomes stronger due to strain hardening, and higher and higher load is required to deform the material. Finally, the load and corresponding stress reach a maximum value, as given by the point *U*. *The stress corresponding to the point U is called the ultimate strength. The ultimate tensile strength is the maximum stress that can be reached in the tension test.*

For ductile materials, the diameter of the specimen begins to decrease rapidly beyond the maximum load point *U*. There is a localised reduction in the cross-sectional area, called *necking*. As the test progresses, the cross-sectional area at the neck decreases rapidly and fracture takes place at the narrowest cross-section of the neck. This fracture is shown by the point *F* on the diagram. The stress at the time of fracture is called *breaking strength*. It is observed from the stress–strain diagram that there is a downward trend after the maximum stress has been reached. The breaking strength is slightly lower than the ultimate tensile strength.

The stress–strain diagram for brittle materials like cast iron is shown in Fig. 2.4. It is observed that such materials do not exhibit the yield point. The deviation of the stress–strain curve from straight line begins very early and fracture occurs suddenly at the point *U* with very small plastic deformation and without necking. Therefore, ultimate tensile strength is considered as failure criterion in brittle materials.

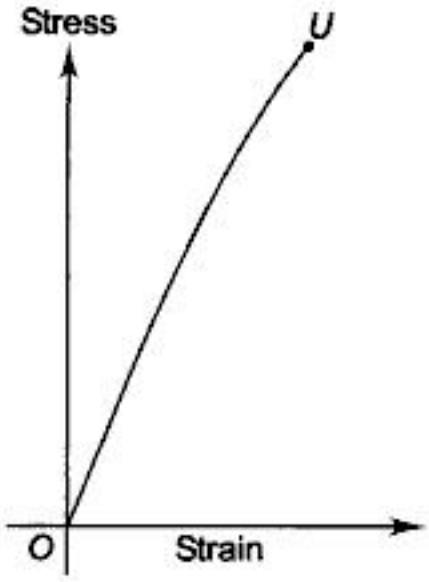

Fig. 2.4 *Stress-Strain Diagram of Brittle Materials*

(vi) Percentage Elongation After the fracture, the two halves of the broken test specimen are fitted together as shown in Fig. 2.5(b) and the extended gauge length *l* is measured. *The percentage elongation is defined as the ratio of the increase in the length of the gauge section of the specimen to original gauge length, expressed in per cent*. Therefore,

$$\text{percentage elongation} = \left(\frac{l - l_0}{l_0}\right) \times 100 \quad .$$

Ductility is measured by percentage elongation.

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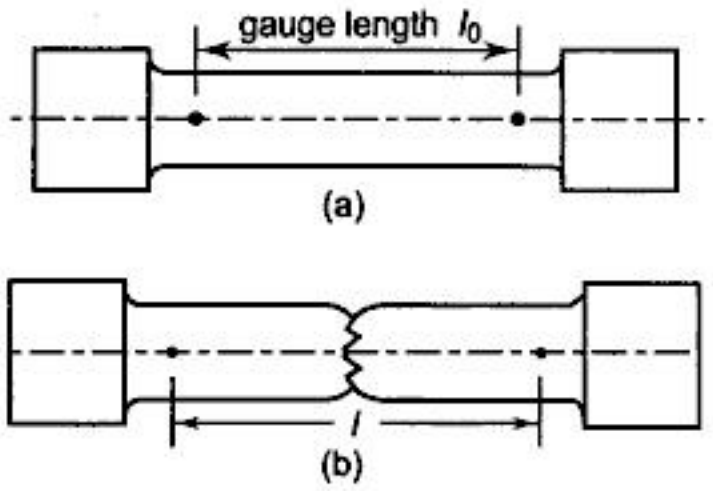

Fig. 2.5 *Determination of Percentage Elongation: (a) Original Test Piece (b) Broken Test Piece*

(vii) Percentage Reduction in Area *Percentage reduction in area is defined as the ratio of decrease in cross-sectional area of the specimen after fracture to the original cross-sectional area, expressed in per cent.* Therefore,

$$\text{percentage reduction in area} = \left(\frac{A_0 - A}{A_0}\right) \times 100$$

where,

A_0 = original cross-sectional area of the test specimen

A = final cross-sectional area after fracture

Percentage reduction in area, like percentage elongation, is a measure of the ductility of the material. If porosity or inclusions are present in the material or if damage due to overheating of the material has occurred, the percentage elongation as well as percentage reduction in area are drastically decreased. Therefore, percentage elongation or percentage reduction in area is considered as an *index of quality* for the material.

2.2 MECHANICAL PROPERTIES OF ENGINEERING MATERIALS

Materials are characterised by their properties. They may be hard, ductile or heavy. Conversely, they may be soft, brittle or light. The mechanical properties of materials are the properties that describe the behaviour of the material under the action of external forces. They usually relate to elastic and plastic behaviour of the material. Mechanical properties are of significant importance in the selection of material for structural machine components. In this article, we will consider the following mechanical properties:

(1) strength
(2) elasticity
(3) plasticity
(4) stiffness
(5) resilience
(6) toughness
(7) malleability
(8) ductility
(9) brittleness
(10) hardness

Strength is defined as the ability of the material to resist, without rupture, external forces causing various types of stresses. Strength is measured by different quantities. Depending upon the type of stresses induced by external loads, strength is expressed as tensile strength, compressive strength or shear strength. Tensile strength is the ability of the material to resist external load causing tensile stress, without fracture. Compressive strength is the ability to resist external load that causes compressive stress, without failure. The terms yield strength and ultimate tensile strength are explained in the previous article.

Elasticity is defined as the ability of the material to regain its original shape and size after the deformation, when the external forces are removed. All engineering metals are elastic but the degree of elasticity varies. Steel is perfectly elastic within a certain elastic limit. The amount of elastic deformation which a metal can undergo is very small. During the elastic deformation, the atoms of the metal are displaced from their original positions but not to the extent that they take up new positions. Therefore, when the external force is removed, the atoms of the metal return to their original positions and the metal takes back its original shape.

Plasticity is defined as the ability of the material to retain the deformation produced under the load on a permanent basis. In this case, the external forces deform the metal to such an extent that it cannot fully recover its original dimensions. During plastic deformation, atoms of the metal are permanently displaced from their original positions and take up new positions. The ability

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of some metals to be extensively deformed in the plastic range without fracture is one of the useful engineering properties of materials. For example, the extensive plastic deformability of low carbon steels enables automobile parts such as the body, hood and doors to be stamped out without fracture. The difference between elasticity and plasticity is as follows:

(i) Elasticity is the ability of a metal to regain its original shape after temporary deformation under an external force. Plasticity is the ability to retain the deformation permanently even after the load is removed.

(ii) The amount of elastic deformation is very small while plastic deformation is relatively more.

(iii) During elastic deformation, atoms of metal are temporarily displaced from their original positions but return back when the load is removed. During plastic deformation, atoms of metal are permanently displaced from their original positions and take up new positions.

(iv) For majority of materials, the stress–strain relationship is linear in the elastic range and non-linear in the plastic range.

(v) Elasticity is an important consideration in machine-tool components while plasticity is desirable for components made by press working operations.

Stiffness or rigidity is defined as the ability of the material to resist deformation under the action of an external load. All materials deform when stressed, to a more or less extent. For a given stress within elastic limit, the material that deforms least is the stiffest. *Modulus of elasticity* is the measure of stiffness. The values of the modulus of elasticity for aluminium alloy and carbon steel are 71 000 and 207 000 N/mm^2 respectively. Therefore, carbon steel is stiffer than aluminium alloy. Stiffness is an important consideration in the design of transmission shafting.

Resilience is defined as the ability of the material to absorb energy when deformed elastically and to release this energy when unloaded. A resilient material absorbs energy within elastic range without any permanent deformation. This property is essential for spring materials. Resilience is measured by a quantity, called *modulus of resilience*, which is the strain energy per unit volume that is required to stress the specimen in a tension test to the elastic limit point. It is represented by the area under the stress–strain curve from the origin to the elastic limit point.

Toughness is defined as the ability of the material to absorb energy before fracture takes place. In other words, toughness is the energy for failure by fracture. This property is essential for machine components which are required to withstand impact loads. Tough materials have the ability to bend, twist or stretch before failure takes place. All structural steels are tough materials. Toughness is measured by a quantity called *modulus of toughness*. Modulus of toughness is the total area under stress–strain curve in a tension test, which also represents the work done to fracture the specimen. In practice, toughness is measured by the *Izod and Charpy impact testing machines*. Toughness decreases as the temperature increases. The difference between resilience and toughness is as follows:

(i) Resilience is the ability of the material to absorb energy within elastic range. Toughness is the ability to absorb energy within elastic and plastic range.

(ii) Modulus of resilience is the area below the stress–strain curve in a tension test up to the yield point. Modulus of toughness is the total area below the stress–strain curve.

(iii) Resilience is essential in spring applications while toughness is required for components subjected to bending, twisting, stretching or to impact loads. Spring steels are resilient while structural steels are tough.

Figures 2.6(a) and (b) show the difference between modulii of resilience and toughness.

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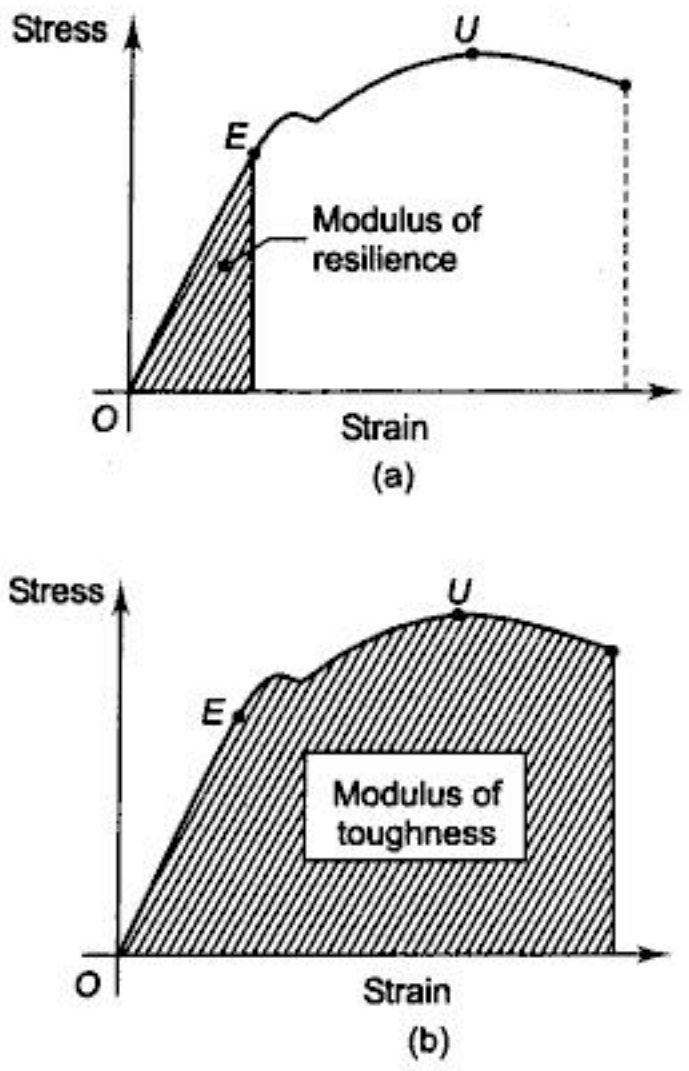

Fig. 2.6 *Modulii of Resilience and Toughness*

Malleability is defined as the ability of a material to deform to a greater extent before the sign of crack, when it is subjected to compressive force. The term 'malleability' comes from a word meaning 'hammer', and in a narrow sense, it means the ability to be hammered out into thin sections. Malleable metals can be rolled, forged or extruded because these processes involve shaping under compressive force. Low carbon steels, copper and aluminium are malleable metals. In general, malleability increases with temperature. Therefore, processes like forging or rolling are hot working processes where hot ingots or slabs are given a shape.

Ductility is defined as the ability of a material to deform to a greater extent before the sign of crack, when it is subjected to tensile force. In other words, ductility is the permanent strain that accompanies fracture in a tension test. Ductile materials are those materials which deform plastically to a greater extent prior to fracture in a tension test. Mild steel, copper and aluminium are ductile materials. Ductile metals can be formed, drawn or bent because these processes involve shaping under tension. Ductility is a desirable property in machine components which are subjected to unanticipated overloads or impact loads. Ductility is measured in units of percentage elongation or percentage reduction in area in a tension test. The ductility of metal decreases as the temperature increases because metals become weak at increasing temperature. All ductile materials are also malleable; however, the converse is not always true. Some metals are soft but weak in tension and, therefore, tend to tear apart under tension. Both malleability as well as ductility are reduced by the presence of impurities in the metal. The difference between malleability and ductility is as follows:

(i) Malleability is the ability of a material to deform under compressive force. Ductility is the ability to deform under tensile force.

(ii) Malleability increases with temperature, while ductility decreases with increasing temperature.

(iii) All ductile materials are also malleable, but the converse is not true.

(iv) Malleability is an important property when the component is forged, rolled or extruded. Ductility is desirable when the component is formed or drawn. It is also desirable when the machine component is subjected to shock loads.

Brittleness is the property of a material which shows negligible plastic deformation before fracture takes place. Brittleness is the opposite to ductility. A brittle material is that which undergoes little plastic deformation prior to fracture in a tension test. Cast iron is an example of brittle material. In ductile materials, failure takes place by yielding. Brittle components fail by sudden fracture. *A tensile strain of 5% at fracture in a tension test is considered as the dividing line between ductile and brittle materials*. The difference between ductility and brittleness is as follows:

(i) Ductile materials deform to a greater extent before fracture in a tension test. Brittle materials show negligible plastic deformation prior to fracture.

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(ii) Steels, copper and aluminium are ductile materials, while cast iron is brittle.

(iii) The energy absorbed by a ductile specimen before fracture in a tension test is more, while brittle fracture is accompanied by negligible energy absorption.

(iv) In ductile materials, failure takes place by yielding which is gradual. Brittle materials fail by sudden fracture.

Hardness is defined as the resistance of the material to penetration or permanent deformation. It usually indicates resistance to abrasion, scratching, cutting or shaping. Hardness is an important property in the selection of material for parts which rub on one another such as pinion and gear, cam and follower, rail and wheel and parts of ball bearing. Wear resistance of these parts is improved by increasing surface hardness by case hardening. There are four primary methods of measuring hardness—Brinell hardness test, Rockwell hardness test, Vicker hardness test and Shore scleroscope. In the first three methods, an indenter is pressed onto the surface under a specific force. The shape of the indenter is either a ball, pyramid or cone. The indenters are made of diamond, carbide or hardened steel, which are much more harder than the surface being tested. Depending upon the cross-sectional area and depth of indentation, hardness is expressed in the form of an empirical number like *Brinell hardness number*. In a Shore scleroscope, the height of rebound from the surface being tested indicates the hardness. Hardness test is simpler than tension test. It is non-destructive because a small indentation may not be detrimental to the performance of the product.

Hardness of the material depends upon the resistance to plastic deformation. Therefore, as the hardness increases, the strength also increases. For certain metals like steels, empirical relationships between strength and hardness are established. For steels,

$$S_{ut} = 3.45 \text{ (BHN)}$$

where S_{ut} is ultimate tensile strength in N/mm^2.

2.3 CAST IRON

Cast iron is a generic term, which refers to a family of materials that differ widely in their mechanical properties. *By definition, cast iron is an alloy of iron and carbon, containing more than 2% of carbon.* In addition to carbon, cast iron contains other elements like silicon, manganese, sulphur and phosphorus. There is a basic difference between steels and cast iron. Steels usually contain less than 1% carbon while cast iron normally contains 2 to 4% carbon. Typical composition of ordinary cast iron is as follows:

carbon = 3.0 – 4.0%
silicon = 1.0 – 3.0%
manganese = 0.5 – 1.0%
sulphur = up to 0.1%
phosphorus = up to 0.1%
iron = remainder

The mechanical properties of cast iron components are inferior to the parts, which are machined from rolled steels. However, even with this drawback, cast iron offers the only choice under certain conditions. From design considerations, cast iron offers the following advantages:

(i) It is available in large quantities and is produced on a mass scale. The tooling required for the casting process is relatively simple and inexpensive. This reduces the cost of cast iron products.

(ii) Cast iron components can be given any complex shape without involving costly machining operations.

(iii) Cast iron has a higher compressive strength. The compressive strength of cast iron is three to five times that of steel. This is an advantage in certain applications.

(iv) Cast iron has an excellent ability to damp vibrations, which makes it an ideal choice for machine tool guides and frames.

(v) Cast iron has more resistance to wear even under the conditions of boundary lubrication.

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(vi) The mechanical properties of cast iron parts do not change between room temperature and 350°C.

(vii) Cast iron parts have low notch sensitivity.

Cast iron has certain drawbacks. It has a poor tensile strength compared to steel. Cast iron parts are section-sensitive. Even with the same chemical composition, the tensile strength of a cast iron part decreases as the thickness of the section increases. This is due to the low cooling rate of thick sections. For thin sections, the cooling rate is high, resulting in increased hardness and strength. Cast iron does not offer any plastic deformation before failure and exhibits no yield point. The failure of cast iron parts is sudden and total. Cast iron parts are, therefore, not suitable for applications where permanent deformation is preferred over fracture. Cast iron is brittle and has poor impact resistance. The machinability of cast iron parts is poor compared to the parts made of steel.

Cast irons are classified on the basis of distribution of carbon content in their micro-structure. There are three popular types of cast iron—grey, malleable and ductile. *Grey cast iron* is formed when the carbon content in the alloy exceeds the amount that can be dissolved. Therefore, some part of carbon precipitates and remains present as 'graphite flakes' distributed in a matrix of ferrite or pearlite or their combination. When a component of grey cast iron is broken, the fractured surface has a grey appearance due to the graphite flakes. Grey cast iron is specified by the symbol FG followed by the tensile strength in N/mm^2 for a 30-mm section. For example, FG 200, in general, means a grey cast iron with an ultimate tensile strength of 200 N/mm^2. Grey cast iron is used for automotive components such as cylinder block, brake drum, clutch plate, cylinder and cylinder head, gears and housing of gear box, flywheel and machine frame, bed and guide.

White cast iron is formed when most of the carbon content in the alloy forms iron carbide and there are no graphite flakes. *Malleable cast iron* is first cast as white cast iron and then converted into malleable cast iron by heat treatment. In malleable cast iron, the carbon is present in the form of irregularly shaped nodules of graphite called 'temper' carbon. There are three basic types of malleable cast iron—blackheart, pearlitic and whiteheart—which are designated by symbols BM, PM and WM, respectively and followed by minimum tensile strength in N/mm^2. For example,

(i) BM 350 is blackheart malleable cast iron with a minimum tensile strength of 350 N/mm^2;

(ii) PM 600 is pearlitic malleable cast iron with a minimum tensile strength of 600 N/mm^2; and

(iii) WM 400 is whiteheart malleable cast iron with a minimum tensile strength of 400 N/mm^2.

Blackheart malleable cast iron has excellent castability and machinability. It is used for brake shoes, pedal, levers, wheel hub, axle housing and door hinges. Whiteheart malleable cast iron is particularly suitable for the manufacture of thin castings which require ductility. It is used for pipe fittings, switch gear equipment, fittings for bicycles and motorcycle frames. Pearlitic malleable iron castings can be hardened by heat treatment. It is used for general engineering components with specific dimensional tolerances.

Ductile cast iron is also called *nodular cast iron* or *spheroidal graphite cast iron*. In ductile cast iron, carbon is present in the form of spherical nodules called 'spherulites' or 'globules' in a relatively ductile matrix. When a component of ductile cast iron is broken, the fractured surface has a bright appearance like steel. Ductile cast iron is designated by the symbol SG (spheroidal graphite) followed by the minimum tensile strength in N/mm^2 and minimum elongation in per cent. For example, SG 800/2 is spheroidal graphite cast iron with a minimum tensile strength of 800 N/mm^2 and a minimum elongation of 2%. Ductile cast iron is used for crankshaft, heavy duty gears and automobile door hinges. Ductile cast iron combines the processing advantages of grey cast iron with the engineering properties of steel. Spheroidal graphite cast iron is dimensionally stable at high temperatures and, therefore, used for furnace doors, furnace components and steam plants. Because

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of excellent corrosion resistance, it is used for pipelines in chemical and petroleum industries.

Mechanical properties of grey[2], malleable[3] and ductile[4] cast iron are given in Table 2.1.

Table 2.1 *Mechanical properties of cast iron*

Grade	*Tensile strength (Min.) (N/mm^2)*	*Elongation (Min.) (%)*	*Hardness (HB)*
(A) Grey cast iron			
FG 150	150	–	130–180
FG 200	200	–	160–220
FG 220	220	–	180–220
FG 260	260	–	180–230
FG 300	300	–	180–230
FG 350	350	–	207–241
FG 400	400	–	207–270
(B) Whiteheart malleable cast iron			
WM 400	400	5	220 (Max.)
WM 350	350	3	230 (Max.)
(C) Blackheart malleable cast iron			
BM 350	350	10	150 (Max.)
BM 320	320	12	150 (Max.)
BM 300	300	6	150 (Max.)
(D) Pearlitic malleable cast iron			
PM 700	700	2	240–290
PM 600	600	3	200–250
PM 550	550	4	180–230
PM 500	500	5	160–200
PM 450	450	6	150–200
(E) Spheroidal graphite cast iron			
SG 900/2	900	2	280–360
SG 800/2	800	2	245–335
SG 700/2	700	2	225–305
SG 600/3	600	3	190–270
SG 500/7	500	7	160–240
SG 450/10	450	10	160–210
SG 400/15	400	15	130–180
SG 350/22	350	22	150 (Max.)

[2] IS 210–1993: Grey iron castings—specification.
[3] IS 14329–1995: Malleable iron castings—specification.
[4] IS 1865–1991: Iron castings with spheroidal or nodular graphite—specification.

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2.4 BIS SYSTEM OF DESIGNATION OF STEELS

A large number of varieties of steel are used for machine components. Steels are designated by a group of letters or numbers indicating any one of the following three properties: [5, 6]

(i) tensile strength;
(ii) carbon content; and
(iii) composition of alloying elements.

Steels, which are standardised on the basis of their tensile strength without detailed chemical composition, are specified by two ways—a symbol Fe followed by the minimum tensile strength in N/mm^2 or a symbol FeE followed by the yield strength in N/mm^2. For example, Fe 360 indicates a steel with a minimum tensile strength of 360 N/mm^2. Similarly, FeE 250 indicates a steel with a minimum yield strength of 250 N/mm^2.

The designation of plain carbon steel consists of the following three quantities:

(i) a figure indicating 100 times the average percentage of carbon;
(ii) a letter C; and
(iii) a figure indicating 10 times the average percentage of manganese.

As an example, 55C4 indicates a plain carbon steel with 0.55% carbon and 0.4% manganese. A steel with 0.35–0.45% carbon and 0.7–0.9% manganese is designated as 40C8.

The designation of unalloyed free cutting steels consists of the following quantities:

(i) a figure indicating 100 times the average percentage of carbon;
(ii) a letter C;
(iii) a figure indicating 10 times the average percentage of manganese;
(iv) a symbol 'S', 'Se', 'Te' or 'Pb' depending upon the element that is present and which makes the steel free cutting; and
(v) a figure indicating 100 times the average percentage of the above element that makes the steel free cutting.

As an example, 25C12S14 indicates a free cutting steel with 0.25% carbon, 1.2% manganese and 0.14% sulphur. Similarly, a free cutting steel with an average of 0.20% carbon, 1.2% manganese and 0.15% lead is designated as 20C12Pb15.

The term 'alloy' steel is used for low and medium alloy steels containing total alloying elements not exceeding 10%. The designation of alloy steels consists of the following quantities:

(i) a figure indicating 100 times the average percentage of carbon; and
(ii) chemical symbols for alloying elements each followed by the figure for its average percentage content multiplied by a factor. The multiplying factor depends upon the alloying element. The values of this factor are as follows:

Elements	Multiplying factor
Cr, Co, Ni, Mn, Si and W	4
Al, Be, V, Pb, Cu, Nb, Ti, Ta, Zr and Mo	10
P, S, N	100

In alloy steels, the symbol 'Mn' for manganese is included only if the content of manganese is equal to or greater than 1%. The chemical symbols and their figures are arranged in descending order of their percentage content.

As an example, 25Cr4Mo2 is an alloy steel having average 0.25% of carbon, 1% chromium and 0.2% molybdenum. Similarly, 40Ni8Cr8V2 is an alloy steel containing average 0.4% of carbon, 2% nickel, 2% chromium and 0.2% vanadium. Consider an alloy steel with the following composition:

carbon = 0.12–0.18%
silicon = 0.15–0.35%
manganese = 0.40–0.60%
chromium = 0.50–0.80%

[5] IS 1762–1974: Code for designation of steels.
[6] IS 1570–1978: Schedule for wrought steels.

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The average percentage of carbon is 0.15%, which is denoted by the number (0.15 × 100) or 15. The percentage content of silicon and manganese is negligible and, as such, they are deleted from the designation. The significant element is chromium and its average percentage is 0.65. The multiplying factor for chromium is 4 and (0.65 × 4) is 2.6, which is rounded to 3. Therefore, the complete designation of steel is 15Cr3. As a second example, consider a steel with the following chemical composition:

carbon = 0.12–0.20%
silicon = 0.15–0.35%
manganese = 0.60–1.00%
nickel = 0.60–1.00%
chromium = 0.40–0.80%

The average percentage of carbon is 0.16% and multiplying this value by 100, the first figure in the designation of steel is 16. The average percentage of silicon and manganese is very small and, as such, the symbols Si and Mn are deleted. Average percentages of nickel and chromium are 0.8 and 0.6, respectively, and the multiplying factor for both elements is 4. Therefore,

nickel: 0.8 × 4 = 3.2 rounded to 3 or Ni3
chromium: 0.6 × 4 = 2.4 rounded to 2 or Cr2.

The complete designation of steel is 16Ni3Cr2.

The term '*high alloy steels*' is used for alloy steels containing more than 10% of alloying elements. The designation of high alloy steels consists of the following quantities:

(i) a letter 'X';
(ii) a figure indicating 100 times the average percentage of carbon;
(iii) chemical symbol for alloying elements each followed by the figure for its average percentage content rounded off to the nearest integer; and
(iv) chemical symbol to indicate a specially added element to attain desired properties, if any.

As an example, X15Cr25Ni12 is a high alloy steel with 0.15% carbon, 25% chromium and 12% nickel. As a second example, consider a steel with the following chemical composition:

carbon = 0.15–0.25%
silicon = 0.10–0.50%
manganese = 0.30–0.50%
nickel = 1.5–2.5%
chromium = 16–20%

The average content of carbon is 0.20%, which is denoted by a number (0.20 × 100) or 20. The major alloying elements are chromium (average 18%) and nickel (average 2%). Hence, the designation of steel is X20Cr18Ni2.

2.5 PLAIN CARBON STEELS

Depending upon the percentage of carbon, plain carbon steels are classified into the following three groups:

(i) Low Carbon Steel Low carbon steel contains less than 0.3% carbon. It is popular as '*mild steel*'. Low carbon steels are soft and very ductile. They can be easily machined and easily welded. However, due to low carbon content, they are unresponsive to heat treatment.

(ii) Medium Carbon Steel Medium carbon steel has a carbon content in the range of 0.3% to 0.5%. It is popular as *machinery steel*. Medium carbon steel is easily hardened by heat treatment. Medium carbon steels are stronger and tougher as compared with low carbon steels. They can be machined well and they respond readily to heat treatment.

(iii) High Carbon Steel High carbon steel contains more than 0.5% carbon. They are called *hard steels* or *tool steels*. High carbon steels respond readily to heat treatments. When heat treated, high carbon steels have very high strength combined with hardness. They do not have much ductility as compared with low and medium carbon steels. High carbon steels are difficult to weld. Excessive hardness is often accompanied by excessive brittleness.

Plain carbon steels are available in the form of bar, tube, plate, sheet and wire. The mechanical properties of plain carbon steels[7] are given in Table 2.2.

[7] IS 1570 (Part 2/Sec1)–1979: Schedules for wrought steels—carbon steels (unalloyed steels).

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Table 2.2 *Mechanical properties of plain carbon steels*

Grade	*Tensile strength (Min.) (N/mm²)*	*Yield strength (Min.) (N/mm²)*	*Hardness (HB)*	*Elongation %*
7C4	320	–	–	27
10C4	340	–	–	26
30C8	500	400	179	21
40C8	580	380	217	18
45C8	630	380	229	15
50C4	660	460	241	13
55C8	720	460	265	13
60C4	750	–	255	11
65C6	750	–	255	10

(Note: Minimum yield strength = 55% of minimum tensile strength)

Many times, a designer is faced with the aspect of choosing a correct carbon content for a particular application. This is an important decision because by merely changing the carbon content, one can get totally different properties of steels.

The guidelines for deciding carbon content in plain carbon steels are as follows:

(i) In applications like automobile bodies and hoods, the ability of the material to deform to a grater extent or 'ductility' is the most important consideration. Such a material should have high ductility. Ductility is measured in terms of percentage elongation. It is observed from Table 2.2, that lower the percentage of carbon, higher is the percentage of elongation or ductility. Therefore, a plain carbon steel, like 7C4, which has a lower percentage of carbon and a higher percentage of elongation, is selected for these parts.

(ii) In applications like gears, machine tool spindles and transmission shafts, strength, toughness and response to heat treatment are important considerations. In these components, the surface is heavily stressed, while the stresses in the core are of comparatively small magnitude. These components require a soft core and a hard surface. This is achieved by case hardening of gears, shafts and spindles. Medium and high carbon steels, such as 40C8, 45C8, 50C4, 55C8, and 60C4 which are stronger, tougher and respond readily to heat treatment are, therefore, selected for these components. They can also be machined well to the required accuracy.

(iii) Spring wires are subjected to severe stress and strength is the most important consideration in selection of their material. High carbon steel, such as 65C6, having maximum tensile strength, is selected for helical and leaf springs.

(iv) Low and medium carbon steels can be satisfactorily welded. However, low carbon steels are the most easily welded. Higher the percentage of carbon in steel, more difficult it is to weld. Therefore, welded assemblies are made of low and medium carbon steels.

(v) Low and medium carbon steels can be satisfactorily forged. However, low carbon steels that are very soft and ductile, are the most easily forged. Higher the percentage of carbon in steel, the more difficult it is to forge the part. Therefore, forged components such as levers, rocker arm, yoke or tie rod are made of low carbon steel 30C8. However, there are some forged components like connecting rod and crankshaft, which also require heat treatment after forging. They are

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made of medium carbon steel 40C8, which responds readily to heat treatment.

(vi) All steels have essentially the same modulus of elasticity. Thus, if rigidity is the requirement of the component, all steels perform equally well. In this case, the least costly steel should be selected.

Some of the important applications of plain carbon steels are as follows:

7C4 Components made by severe drawing operation such as automobile bodies and hoods

10C4 Case hardened components such as cams and cam shaft, worm, gudgeon pin, sprocket and spindle

30C8 Cold formed and case hardened parts such as socket, tie rod, yoke, lever and rocker arm

40C8 Transmission shaft, crank shaft, spindle, connecting rod, stud and bolts

45C8 Transmission shaft, machine tool spindle, bolts and gears of large dimensions

50C4 Transmission shaft, worm, gears and cylinder

55C8 Components with moderate wear resistance such as gears, cam, sprocket, cylinder and key

60C4 Machine tool spindle, hardened bolt and pinion

65C6 Coil and leaf springs

2.6 FREE CUTTING STEELS

Steels of this group include carbon steel and carbon–manganese steel with a small percentage of sulphur. Due to addition of sulphur, the machinability of these steels is improved. *Machinability is defined as the ease with which a component can be machined.* It involves three factors—the ease of chip formation, the ability to achieve a good surface finish and ability to achieve an economical tool life. Machinability is an important consideration for parts made by automatic machine tools. Typical applications of free cutting steels are studs, bolts and nuts. Mechanical properties of free cutting steels[8] are given in Table 2.3.

Table 2.3 *Mechanical properties of free cutting steels (cold drawn bars) (20–40 mm diameter)*

Grade	*Tensile strength (Min.) (N/mm²)*	*Elongation (Min.) (%)*
10C8S10	460	10
14C14S14	520	11
25C12S14	560	10
40C10S18	600	10
40C15S12	640	8

2.7 ALLOY STEELS

Alloy steel is defined as carbon steel to which one or more alloying elements are added to obtain certain beneficial effects. The commonly added elements include silicon, manganese, nickel, chromium, molybdenum and tungsten. The term '*alloy steels*' usually refers to '*low*' alloy steels containing from about 1 to 4 per cent of alloying elements. On the other hand, stainless and heat-resisting steels are called '*high*' alloy steels. Plain carbon steels are successfully used for components subjected to low or medium stresses. These steels are cheaper than alloy steels. However, plain carbon steels have the following limitations:

[8] IS 1570 (Part 3)–1979: Schedules for wrought steel—carbon and carbon-manganese free cutting steels.

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(i) The tensile strength of plain carbon steels cannot be increased beyond 700 N/mm^2 without substantial loss in ductility and impact resistance.
(ii) Components with large section thickness cannot be produced with martensitic structure. In other words, plain carbon steels are not deep hardenable.
(iii) Plain carbon steels have low corrosion resistance.
(iv) Medium carbon steels must be quenched rapidly to obtain a fully martensitic structure. Rapid quenching results in distortion and cracking in heat-treated components.
(v) Plain carbon steels have poor impact resistance at low temperatures.

To overcome these deficiencies of plain carbon steels, alloy steels have been developed. Alloy steels cost more than plain carbon steels. However, in many applications, alloy steel is the only choice to meet the requirements. Alloy steels have the following advantages:

(i) Alloy steels have higher strength, hardness and toughness.
(ii) High values of hardness and strength can be achieved for components with large section thickness.
(iii) Alloy steels possess higher hardenability, which has great significance in heat treatment of components.
(iv) Alloy steels retain their strength and hardness at elevated temperatures.
(v) Alloy steels have higher resistance to corrosion and oxidation compared with plain carbon steels.

Alloying elements can affect constitution, characteristics and behaviour of these steels. The effects of major alloying elements are as follows:

(i) Silicon Silicon is present in almost all steels. It increases strength and hardness without lowering the ductility. Silicon is purposely added in spring steel to increase its toughness.

(ii) Manganese Most steels contain some manganese remaining from the deoxidisation and desulphurisation processes. However, when it exceeds 1 per cent, it is regarded as an alloying element. Manganese is one of the least expensive alloying elements. It increases hardness and strength. It also increases the depth of hardening. Manganese is an important alloying element in free cutting steels.

(iii) Nickel Nickel increases strength, hardness and toughness without sacrificing ductility. It increases hardenability of steel and impact resistance at low temperature. The main effect of nickel is to increase toughness by limiting grain growth during the heat treatment process.

(iv) Chromium Chromium increases hardness and wear resistance. Chromium steel components can be readily hardened in heavy sections. They retain strength and hardness at elevated temperatures. Chromium steels containing more than 4 per cent chromium have excellent corrosion resistance.

(v) Molybdenum Molybdenum increases hardness and wear resistance. It resists softening of steel during tempering and heating.

(vi) Tungsten Tungsten and molybdenum have similar effects. It is an expensive alloying element and about 2 to 3 per cent tungsten is required to replace 1 per cent of molybdenum. It is an important alloying element in tool steels.

Many times, a designer is faced with the aspect of choosing the correct alloying element of steels for a particular application. This is an important decision because by merely changing an alloying element, one can get totally different combinations of mechanical properties for steels.

The guidelines for selecting alloy steels are as follows:

(i) Spring wires are subjected to severe stresses, and strength is the most important consideration in selection of their material. Silicon increases strength. Therefore, silicon steel, such as 55Si7, is selected for helical and leaf springs.
(ii) In case of highly stressed screws, bolts and axles, high strength and toughness are important considerations. Nickel increases strength and toughness without loss of

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ductility. Therefore, nickel steel such as 40Ni14 is used for these severely stressed components.

(iii) In applications like gears, surface hardness, wear resistance and response to heat treatment are important considerations. In these components, the surface is heavily stressed, while the stresses in the core are of comparatively small magnitude. These components require a soft core and a hard surface. Chromium increases hardness and wear resistance. Also, chromium steels are readily hardened in heavy sections. Therefore, chromium steels, such as 40Cr4 is selected for all types of gears.

(iv) In a number of components like gears, cams, camshafts, and transmission shafts, combined properties such as hardness and toughness, strength and ductility are required. This is achieved by using nickel and chromium as alloying elements and selecting proper heat treatment. Nickel–chromium steels, like 16Ni3Cr2 or 30Ni16Cr5, which combines hardness and toughness, are selected for these parts.

Some of the important applications of alloy steels are as follows:

55Si7 Leaf and coil springs

37C15 Axle, shaft and crankshaft

35Mn6Mo3 Bolt, stud, axle, lever and general engineering components

16Mn5Cr4 Gears and shaft

40Cr4 Gears, axle and steering arm

50Cr4 Coil, laminated and volute springs

40Cr4Mo2 Shaft, axle, high tensile bolt, stud and popeller shaft

40Cr13Mo10V2 Components subjected to high tensile stresses

40Ni14 Severely stressed screw, nut and bolt

16Ni3Cr2 Gears, transmission components, cam and camshaft

30Ni16Cr5 Heavy duty gears

35Ni5Cr2 Gear shaft, crankshaft, chain parts, camshaft and planetary gears

40Ni6Cr4Mo2 General machine parts, nuts and bolts, gears, axles, shafts and connecting rod

40Ni10Cr3Mo6 High strength machine components, bolts and studs, axles and shafts, gears and crankshafts

Mechanical properties of alloy steels[9] are given in Table 2.4 on next page.

2.8 OVERSEAS STANDARDS

Cast iron and steel are the essential ingredients in any product. A large variety of steel and cast iron is developed for a number of applications. In our country, collaborations with foreign industries have resulted in use of different overseas standards and designations. Some important designations for ferrous materials are as follows,[10, 11, 12]

(i) The American Society for Testing Materials (ASTM) has classified grey cast iron by means of a number. This class number gives minimum tensile strength in kpsi. For example, ASTM Class No. 20 has minimum ultimate tensile strength of 20000 psi. Similarly, a cast iron with minimum ultimate tensile strength of 50000 psi is designated as ASTM Class No. 50. Commonly used ASTM classes of cast iron are 20, 25, 30, 35, 40, 50 and 60.

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[9] IS 1570 (Part 4)–1988: Schedules for wrought steels---Alloy steels with specified chemical composition and mechanical properties.

[10] SAE J402: 1984–SAE Numbering system for wrought or rolled steel (SAE standard).

[11] 'SAE Handbook'—Vol.1—'Materials'—Society of Automotive Engineers Inc., 1987.

[12] 'Metals Handbook'—Vol.1—'Properties and selection: Iron, steels and high performance alloys'—American Society of Metals Inc., 1990.

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Table 2.4 *Mechanical properties of alloy steels*

Grade	*Tensile strength (N/mm^2)*	*0.2 percent Proof stress (Min.) (N/mm^2)*	*Elongation (Min.) (%)*	*Hardness (HB)*	*Condition*
37C15	590–740	390	18	170–217	Oil–hardened and tempered
35Mn6Mo3	690–840	490	14	201–248	Oil-hardened and tempered
16Mn5Cr4	790 (Min.)	–	10	–	Case-hardened steel (core properties)
40Cr4	690–840	490	14	201–248	Oil-hardened and tempered
40Cr4Mo2	700–850	490	13	201–248	Oil-hardened and tempered
40Cr13Mo10V2	1340 (Min.)	1050	8	363 (Min.)	Oil-hardened and tempered
40Ni14	790–940	550	16	229–277	Oil-hardened and tempered
16Ni3Cr2	690 (Min.)	–	15	–	Case-hardened steel (core properties)
30Ni16Cr5	1540 (Min.)	1240	8	444 (Min.)	Oil-hardened and tempered
35Ni5Cr2	690–840	490	14	201–248	Oil-hardened and tempered
40Ni6Cr4Mo2	790–940	550	16	229–277	Oil-hardened and tempered
40Ni10Cr3Mo6	990–1140	750	12	285–341	Oil-hardened and tempered

(ii) In Germany, Deutches Institut Fuer Normung (DIN) has specified grey cast iron by minimum ultimate tensile strength in kgf/mm^2. For example, GG-12 indicates grey cast iron with minimum ultimate tensile strength of 12 kgf/mm^2. Similarly, grey cast iron with minimum ultimate tensile strength of 26 kgf/mm^2 is designated as GG-26. The common varieties of grey cast iron according to DIN standard are GG-12, GG-14, GG-18, GG-22,GG-26 and GG-30.

(iii) A numbering system for carbon and alloy steels is prescribed by the Society of Automotive Engineers (SAE) of USA and American Iron and Steel Institute (AISI). It is based on chemical composition of the steel. The number is composed of four or five digits. The first two digits indicate the type or alloy classification. The last two or three digits give the carbon content. Since carbon is the most important element in steel, affecting the strength and hardness, it is given proper weightage in this numbering system. The basic numbers for various types of steel are given in Table 2.5. For example, plain carbon steel has 1 and 0 as its first two digits. Thus, a steel designated as 1045 indicates plain carbon steel with 0.45% carbon. Similarly, a nickel–chromium steel with 1.25% Ni, 0.60% Cr and 0.40% carbon is specified as SAE 3140.

The AISI number for steel is the same as the SAE number. In addition, there is a capital letter A, B, C, D or E that is prefixed to the number. These capital letters indicate the manufacturing process of steel. The meaning of these letters is as follows:

A—Basic open-hearth alloy steel
B—Acid Bessemer carbon steel
C—Basic open-hearth carbon steel
D—Acid open-hearth carbon steel
E—Electric furnace alloy steel

The British system designates steel is a series of numbers known as 'En' series. The En number of a steel has no correlation either with the chemical composition such as carbon content and types of alloying element or mechanical properties such as ultimate tensile strength. For example, the number 3 in En3 steel has no relationship with the carbon content, alloying element or strength of steel. Table 2.6 shows the overseas equivalent designations of some popular varieties of steel.

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Table 2.5 *Basic numbering system of SAE and AISI steels*

Material	*SAE or AISI Number*
Carbon steels	1xxx
plain carbon	10xx
free-cutting, screw stock	11xx
Chromium steels	5xxx
low chromium	51xx
medium chromium	52xxx
corrosion and heat resisting	51xxx
Chromium–nickel–molybdenum steels	86xx
Chromium–nickel–molybdenum steels	87xx
Chromium–vanadium steels	6xxx
1.00% Cr	61xx
Manganese steels	13xx
Molybdenum steels	4xxx
carbon–molybdenum	40xx
chromium–molybdenum	41xx
chromium–nickel–molybdenum	43xx
nickel–molybdenum; 1.75% Ni	46xx
nickel-molybdenum; 3.50% Ni	48xx
Nickel–chromium steels	3xxx
1.25% Ni, 0.60% Cr	31xx
1.75% Ni, 1.00% Cr	32xx
3.50% Ni, 1.50% Cr	33xx
Silicon–manganese steels	9xxx
2.00% Si	92xx
Nickel steels	2xxx
3.5% Ni	23xx
5.0% Ni	25xx

2.9 HEAT TREATMENT OF STEELS

The heat treatment process consists of controlled heating and cooling of components made of either plain carbon steel or alloy steel, for the purpose of changing their structure in order to obtain certain desirable properties like hardness, strength or ductility. The major heat treatment processes are as follows:

(i) **Annealing** consists of heating the component to a temperature slightly above the critical temperature, followed by slow cooling. It reduces hardness and increases ductility.

(ii) **Normalising** is similar to annealing, except that the component is slowly cooled in air. It is used to remove the effects of the previous heat treatment processes.

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Table 2.6 *Overseas equivalent designations of steel*

BIS designation	*En Number*	*SAE*	*AISI*	*DIN*
Plain-carbon steels				
7C4	2A	1010	C 1010	17210
10C4	32A	1012	C 1012	17155
30C8	5	1030	C 1030	–
45C8	43B	1045	C 1045	17200
50C4	43A	1049, 1050	C 1049, C 1050	–
55C8	43J, 9K	1055	C 1055	–
60C4	43D	1060	C 1060	17200
65C6	42B	1064	C 1064	17222
Free cutting steels				
10C8S10	–	1109	C 1109	–
14C14S14	7A, 202	1117, 1118	C 1117, C 1118	–
25C12S14	7	1126	C 1126	–
40C10S18	8M	1140	C 1140	–
40C15S12	15AM	1137	C 1137	–
Alloy steels				
40Cr4	18	5135	5135	–
40Ni14	22	2340	2340	–
35Ni5Cr2	111	3140	3140	1662
30Ni16Cr5	30A	–	-	–
40Ni6Cr4Mo2	110	4340	4340	17200
27C15	14B	1036	C 1036	17200
37C15	15, 15A	1041, 1036	C 1041, C 1036	17200
50Cr4V2	47	6150	6150	17221

(iii) **Quenching** consists of heating the component to the critical temperature and cooling it rapidly in water or air. It increases hardness and wear resistance. However, during the process, the component becomes brittle and ductility is reduced.

(iv) **Tempering** consists of reheating the quenched component to a temperature below the transformation range, followed by cooling at a desired rate. It restores the ductility and reduces the brittleness due to quenching.

The recommended hardening and tempering treatments and temperature ranges can be obtained from the standards. The selection of a proper heat treatment process depends upon the desirable properties of the component.

2.10 CASE HARDENING OF STEELS

In number of situations, the stress distribution across the cross-section of a component is not uniform. In some cases, the surface is heavily stressed, while the stress in the core is of comparatively small magnitude. The examples of this type of stress distribution are gear, cam and rolling contact bearing. The surface failure of these components can be avoided by case hardening.

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Case hardening can be achieved by the following two ways:

(i) by altering the structure at the surface by local hardening, e.g., flame or induction hardening.

(ii) by altering the structure as well as the composition at the surface, e.g., case carburising, nitriding, cyaniding and carbo-nitriding.

Flame hardening consists of heating the surface above the transformation range by means of a flame, followed by quenching. The distortion of the component is low because the bulk of the work piece is not heated. Flame hardening can be done in stages, such as hardening of tooth by tooth of a gear blank. The minimum case depth obtained by this process is 1 mm, although case depths up to 6 mm are quite common. Flame hardening is recommended under the following situations:

(i) where the component is large;

(ii) where a small area of the work piece is to be hardened; and

(iii) where dimensional accuracy is desirable.

Carbon steels containing more than 0.4% carbon are generally employed for flame hardening.

The **induction-hardening process** consists of heating the surface by induction in the field of an alternating current. The amount of heat generated depends upon the resistivity of the material. Induction hardening produces case depths as small as 0.1 mm. There is not much difference between flame and induction hardening, except for the mode of heating and minimum case depth.

Case carburising consists of introducing carbon at the surface layer. Such a component has a high-carbon surface layer and a low-carbon core with a gradual transformation from one zone to the other. Different methods are used to introduce carbon, but all involve heating from 880 to 980°C. The carburising medium can be solid, liquid or gas. Case carburising is recommended for case depths up to 2 mm.

Carbo-nitriding consists of introducing carbon and nitrogen simultaneously at the surface layer. The component is heated from 650 to 920°C in the atmosphere of anhydrous ammonia and then quenched in a suitable medium. Nitrogen is concentrated at the surface and backed up by a carburised case. Medium carbon steels are carbo-nitrided with case depths up to 0.6 mm. The process gives a higher wear resistance compared to the case-carburising process. *Cyaniding* is similar to carbo nitriding except that the medium is liquid.

Nitriding consists of exposing the component to the action of nascent nitrogen in a gaseous or liquid medium from 490 to 590°C. This process does not involve any subsequent quenching. The gaseous medium consists of dry ammonia. The liquid medium can be of cyanides and cyanates. The nitrided case consists of two zones—a brittle white zone next to the surface, consisting of nitrides, followed by a tougher diffusion zone, where nitrides are precipitated in the matrix. Case depths up to 0.1 mm are obtained by this process. Nitrided components are used for applications requiring high resistance to abrasion, high endurance limit and freedom from distortion. The disadvantages of this process are as follows:

(i) The components cannot be used for concentrated loads and shocks;

(ii) The case depth is limited to 0.5 mm; and

(iii) Considerable time is required for the process due to long cycle time.

The applications of the nitrided component are indexing worms, high-speed spindles and crankshafts.

2.11 CAST STEEL

Pouring molten steel of desired chemical composition into a mould and allowing the steel to solidify produces cast steel components. Steel castings can be made from any type of carbon and low alloy steels. Cast steel components and wrought steel components of equivalent chemical composition respond similarly to heat treatment, have the same weldability and have similar physical and mechanical properties. However, cast steel components do not exhibit the effects of directionality on mechanical properties that are typical of wrought steels. This non-directional characteristic of cast steel components is advantageous where application involves multi-directional forces. Compared with cast

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iron components, the cast steel parts are lighter for the same strength. However, cast steels are costly and their superior mechanical properties are justified only in certain applications. Cast steel in its liquid form has poor fluidity compared with cast iron. Therefore, the wall thickness or section thickness cannot be made less than 6.5 mm. During solidification, cast steel shrinks to a greater extent. This results in residual stresses which cannot be completely relieved by the normalising process. Poor fluidity and excessive contraction should be taken into account while designing a component made of cast steel. It is always advisable to divide complicated one piece castings into simple parts and join them at the stage of assembly.

There are two varieties of steel castings: carbon steel castings and high tensile steel castings.[13, 14] *Carbon steel castings* are used for heavy duty machinery, highly stressed parts and gears. There are five grades of carbon steel castings. They are classified on the basis of minimum yield stress and tensile strength values respectively, expressed in N/mm^2. For example, a carbon steel casting of grade 200–400 has a yield stress of 200 N/mm^2 and an ultimate tensile strength of 400 N/mm^2. A carbon steel casting of yield stress and ultimate tensile strength of 280 and 520 N/mm^2 respectively, will be designated as a carbon steel casting of grade 280–520. The mechanical properties of carbon steel castings are given in Table 2.7.

Table 2.7 *Mechanical properties of carbon steel castings*

Grade	*Yield stress (Min.) (N/mm^2)*	*Tensile strength (Min.) (N/mm^2)*	*Elongation (Min.) (%)*
200–400	200	400	25
230–450	230	450	22
260–520	260	520	18
280–520	280	520	18
340–570	340	570	15

High tensile steel castings have higher strength, good toughness and high resistance to wear. These castings are used in transportation equipment, and agricultural machinery. There are five grades of high tensile steel castings which are classified according to the ultimate tensile strength, for example CS 640 is a steel casting with minimum ultimate tensile strength of 640 N/mm^2. A high tensile steel casting with minimum ultimate tensile strength of 1030 N/mm^2 is designated as CS 1030. The mechanical properties of high tensile steel castings are given in Table 2.8.

Table 2.8 *Mechanical properties of high tensile steel castings*

Grade	*Tensile strength (Min.) (N/mm^2)*	*Yield stress[1] (Min.) (N/mm^2)*	*Elongation (Min.) (%)*	*Hardness (HB)*
CS 640	640	390	15	190
CS 700	700	580	14	207
CS 840	840	700	12	248
CS 1030	1030	850	8	305
CS 1230	1230	1000	5	355

Note: 1. Yield stress is 0.5 per cent proof stress.

2.12 ALUMINIUM ALLOYS

Aluminium alloys are recent in origin compared with copper or steel. However, due to a unique combination of certain mechanical properties, they have become the most widely used nonferrous metal. Aluminium alloys offer the following design advantages:

[13] IS 1030—1998: Carbon steel castings for general engineering purposes—specification.
[14] IS 2644–1994: High strength steel castings for general engineering and structural purposes—specification.

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(i) Low Specific Gravity The relative density of aluminium alloys is 2.7 compared with 7.9 of steel, i.e., roughly one third of steel. This results in light weight construction and reduces inertia forces in applications like connecting rod and piston, which are subjected to reciprocating motion.

(ii) Corrosion Resistance Aluminium has high affinity for oxygen and it might be expected that aluminium components will oxidise or rust very easily. However, in practice it has an excellent resistance to corrosion. This is due to the thin but very dense film of oxide (alumina skin) which forms on the surface of metal and protects it from further atmospheric attack. It is due to alumina skin that there is comparatively dull appearance on the surface of a polished aluminium component.

(iii) Ease of Fabrication Aluminium alloys have a face-centered cubic crystal structure with many slip planes. This makes the material ductile and easily shaped. They can be cast, rolled, forged or extruded. Aluminium alloys can be formed, hot or cold, with considerable ease. They can be machined easily if suitable practice and proper tools are used. They can be joined by fusion welding, resistance welding, soldering and brazing. Due to excellent malleability, it is possible to produce very thin aluminium foil suitable for food packaging.

(iv) High Thermal Conductivity As a material for constructional components, aluminium has poor strength compared with steel. In '*soft*' condition, the tensile strength of pure aluminium is only 90 N/mm^2, while even in work-hardened state, it is no more than 135 N/mm^2. Hence, for engineering applications, aluminium is alloyed in order to obtain high strength to weight ratio. Some of the high strength aluminium alloys with suitable heat treatment have tensile strength in excess of 600 N/mm^2. There are two varieties of aluminium alloys–wrought and cast—which are used for machine components. Wrought aluminium alloys are available in the form of plates, sheets, strips, wires, rods and tubes. There are three methods of casting aluminium alloys, viz., sand casting, gravity die casting and pressure die casting.

Aluminium alloys are designated by a particular numbering system. The numbers given to alloying elements are given in Table 2.9.

Table 2.9

Aluminium	—	1	Magnesium	—	5
Copper	—	2	Magnesium silicide	—	6
Manganese	—	3	Zinc	—	7
Silicon	—	4	Other elements	—	8

Cast aluminium alloys are specified by a '*four-digit*' system while wrought alloys by a '*five-digit*' system[15, 16, 17, 18]. The meaning of these digits is as follows:

First digit It identifies the major alloying element.

Second digit It identifies the average percentage of the major alloying element, halved and rounded off.

Third, fourth and fifth digits They identify the minor alloying elements in order of their decreasing percentage.

As an example, consider an aluminium alloy casting with 9.8% Cu, 1.0% Fe and 0.25% Mg.

First digit	identification of copper	:	2
Second digit	(9.8/2 = 4.9 or 5)	:	5
Third digit	identification of iron	:	8

[15] IS 617–1994: Cast aluminium and its alloys—ingots and casting for general engineering purposes—specification.
[16] IS 733–1983: Specifications for wrought aluminium and aluminium alloy bars, rods and sections for general engineering purposes.
[17] IS 5052–1993: Aluminium and its alloys—Temper designations.
[18] SP: 1–1967: Comparison of Indian and overseas standards on aluminium alloy castings—Bureau of Indian Standards.

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Fourth digit identification of magnesium : 5
Complete designation = 2585

Some of the important applications of cast aluminium alloy are as follows:

Alloy 4450 Engine cylinder blocks, castings for valve body and large fan blades

Alloy 4600 Intricate and thin-walled castings, motor housings, water cooled manifolds and pump casings

Alloy 2280 Connecting rods and flywheel housings

Alloy 2285 (Y-alloy) Pistons and cylinder heads

Alloy 2250 Castings for hydraulic equipment

Alloy 4652 Pistons for internal combustion engines.

The alloy 4600 is used for pressure die casting parts. It has excellent fluidity, which facilitates the production of complex castings of large surface area and thin walls. The mechanical properties of aluminium alloy castings are given in Table 2.10.

The important applications of wrought aluminium alloy are as follows:

Alloy 24345 Heavy duty forging and structures

Alloy 24534 Stressed components of aircraft

Alloy 54300 Welded structures and tank

Alloy 64430 Roof truss and deep-drawn container

Alloy 74530 Welded pressure vessels.

Table 2.10 *Mechanical properties of cast aluminium alloys*

Alloy	*Condition*	*Tensile Strength (Min.) (N/mm^2)*		*Elongation (Min.) (%)*	
		Sand-cast	*Chill-cast*	*Sand-cast*	*Chill-cast*
4450	M	135	160	2	3
	T5	160	190	1	2
	T7	160	225	2.5	5
	T6	225	275	–	2
4600	M	165	190	5	7
2280	T4	215	265	7	13
	T6	275	310	4	9
2285	T6	215	280	–	–
2550	M	–	170	–	–
4652	T5	–	210	–	–
	T6	140	200	–	–
	T7	175	280	–	–

(M = as cast; T5 = precipitation treated; T4 = solution treated; T7 = solution treated and stabilised; T6 = solution and precipitation treated)

The mechanical properties of wrought aluminium alloy are given in Table 2.11 on next page.

2.13 COPPER ALLOYS

Copper possesses excellent thermal and electrical conductivity. It can be easily cast, machined and brazed. It has good corrosion resistance. However, even with these advantages, pure copper is not used in any structural application due to its poor strength. The tensile strength of copper is about 220 N/mm^2. Pure copper is mainly used for electrical and thermal applications. For structural applications, copper alloys are used instead of pure copper. Some of the popular copper alloys are brass, bronze, gunmetal and monel metal. Their properties and applications are briefly discussed in this article.

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Table 2.11 *Mechanical properties of wrought aluminium and aluminium alloys*

Alloy	*Condition*	*Diameter(mm)*		*0.2 per cent proof stress (N/mm^2)*		*Tensile strength (N/mm^2)*		*Elongation (Min.) (%)*
		over	*upto*	*Min.*	*Max.*	*Min.*	*Max.*	
24345	M	–	–	90	–	150	–	12
	O	–	–	–	175	–	240	12
	W	–	10	225	–	375	–	10
		10	75	235	–	385	–	10
		75	150	235	–	385	–	8
		150	200	225	–	375	–	8
	WP	–	10	375	–	430	–	6
		10	25	400	–	460	–	6
		25	75	420	–	480	–	6
		75	150	405	–	460	–	6
		150	200	380	–	430	–	6
24534	M	–	–	90	–	150	–	12
	O	–	–	–	175	–	240	12
	W	–	10	220	–	375	–	10
		10	75	235	–	385	–	10
		75	150	235	–	385	–	8
		150	200	225	–	375	–	8
54300	M	–	150	130	–	265	–	11
	O	–	150	125	–	–	350	13
64430	M	–	–	80	–	110	–	12
	O	–	–	–	–	–	150	16
	W	–	150	120	–	185	–	14
		150	200	100	–	170	–	12
	WP	–	5	225	–	295	–	7
		5	75	270	–	310	–	7
		75	150	270	–	295	–	7
		150	200	240	–	280	–	6
74530	W[1]	–	6	220	–	255	–	9
		6	75	230	–	275	–	9
		75	150	220	–	265	–	9
	WP	–	6	245	–	285	–	7
		6	75	260	–	310	–	7
		75	150	245	–	290	–	7

Note: [1] Naturally aged for 30 days (M = as manufactured; O = annealed; W = solution treated and naturally aged; WP = solution and precipitation treated).

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(i) Brass The most commonly used copper alloy is brass. It is an alloy of copper and zinc. Sometimes, it may contain small amounts of tin, lead, aluminium and manganese. Brass has the following advantages:

(i) The tensile strength of brass is higher than that of copper.
(ii) Brass is cheaper than copper.
(iii) Brass has excellent corrosion resistance.
(iv) Brass has better machinability.
(v) Brass has good thermal conductivity.

The strength and ductility of brass depend upon the zinc content. As the amount of zinc increases, the strength of brass increases and ductility decreases. The best combination of strength and ductility is obtained when the amount of zinc is 30 per cent. Brass can be used either in rolled condition or as cast. Some of the commonly used varieties of brass are yellow brass, naval brass, cartridge brass and muntz metal. Typical applications of brass in the field of mechanical engineering are tubes for condensers and heat exchangers, automotive radiator cores, rivets, valve stems and bellow springs.

(ii) Bronze Bronze is an alloy of copper and elements other than zinc. In some cases, bronze may contain a small amount of zinc. There are three important varieties of bronze—aluminium bronze, phosphor bronze and tin bronze. *Aluminium bronze* contains 5 to 10 per cent aluminium. It has excellent corrosion resistance, high strength and toughness, low coefficient of friction and good damping properties. It is used for hydraulic valves, bearings, cams and worm gears. The colour of aluminium bronze is similar to that of 22 carat gold and it is frequently called '*imitation*' gold. Aluminium bronze is difficult to cast because its casting temperature is around 1000°C. At this temperature, aluminium oxidises and creates difficulties in casting.

Phosphor bronze contains about 0.2 per cent phosphorus. The effect of phosphorus is to increase the tensile strength and corrosion resistance and reduce the coefficient of friction. In cast form, phosphor bronze is widely used for worm wheels and bearings. In wrought form, it is used for springs, bellows, pumps, valves and chemical equipment. *Tin bronze* contains up to 18 per cent tin and sometimes small amounts of phosphorus, zinc or lead. It has excellent machinability, wear resistance and low coefficient of friction. It is used for pump castings, valve fittings and bearings. High prices of both copper and tin, put limitations on the use of tin bronze.

In general, all varieties of bronze have the following advantages:

(i) excellent corrosion resistance;
(ii) low coefficient of friction; and
(iii) higher tensile strength than copper or brass.

The main limitation of bronze is its high cost.

(iii) Gunmetal Gunmetal is an alloy of copper which contains 10% tin and 2% zinc. The presence of zinc improves fluidity of gunmetal during casting process. Zinc is considerably cheaper than tin. Therefore, the total cost of gunmetal is less than that of bronze. In cast form, gunmetal is used for bearings. It has excellent corrosion resistance, high strength and low coefficient of friction.

(iv) Monel Metal Monel metal is a copper–nickel alloy. It contains 65% nickel and 32% copper. It has excellent corrosion resistance to acids, alkalis, brine water, sea water and other chemicals. It is mainly used for handling sulphuric and hydrochloric acids. It is also used for pumps and valves for handling the chemicals in process equipment.

2.14 DIE CASTING ALLOYS

The die casting process consists of forcing the molten metal into a closed metal die. This process is used for metals with a low melting point. The advantages of the die casting process are as follows:

(i) Small parts can be made economically in large quantities.
(ii) Surface finish obtained by this method is excellent and requires no further finishing.
(iii) Very thin sections or complex shapes can be obtained easily.

The drawbacks include the cost of dies and restriction on the size of the component, since only small parts can be die cast.

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Die casting alloys are made from zinc, aluminium and magnesium. Brass can be die cast but its casting temperature is high. Zinc die castings are more popular due to their high strength, long die life and moderate casting temperature. Aluminium and magnesium die castings are light in weight but their casting temperature is higher than that of zinc die castings.

2.15 CERAMICS

Ceramics can be defined as a compound of metallic and non-metallic elements with predominantly 'ionic' interatomic bonding. The word '*ceramic*' is derived from the Greek word keramos which means '*potter's clay*'. Traditional ceramics include refractories, glass, abrasives, enamels and insulating materials. However, many substances, which are now classed as ceramics in fact, contain no clay. Modern ceramics include metal oxides, carbides, borides, nitrides and silicates. Some of their examples are Magnesia (M_gO), Alumina (Al_2O_3), Zirconia (Z_rO_2), Beryllia (B_eO), Silicon carbide (S_iC) and Tungsten carbide (T_iC).

The advantages of modern engineering ceramics are as follows:

(i) Most of the ceramics possess high hardness. This increased hardness is due to the operation of strong covalent bonds between atoms in their crystal structure. Materials like silicon carbide and boron nitride are the examples of ceramics with high hardness. It is this property which makes them useful as abrasive powder and cutting tools.

(ii) Ceramics have high melting points. Materials such as magnesia and alumina have melting points of 2800°C and 2040°C respectively. This property makes them excellent refractory materials for the lining of the furnaces.

(iii) Ceramics are good thermal insulators. In most of the ceramics, there are no free conducting electrons and heat is conducted only by transfer of vibration energy from one atom to another. This is unlike free electrons in metals. Hence, ceramics possess excellent insulating property. This is another reason to use them as refractory material.

(iv) Ceramics have extremely high electrical resistivity. Hence, they are used for electrical insulators. Porcelain is a popular insulating material. Alumina is used for spark-plug insulation.

(v) The densities of ceramics are low compared with those of engineering metals. This results in lightweight components.

(vi) Ceramics are chemically resistant to most of the acids, alkalis and organic substances. They are also unaffected by oxygen. This increases the durability of ceramic components.

Ceramics have certain drawbacks. Their main disadvantages are as follows:

(i) Ceramics are brittle in nature. They are highly susceptible to stress concentration. Presence of even a micro-crack may lead to failure because it acts as a stress raiser. In ceramics, there is no plastic deformation like metals and no redistribution of stresses. This results in brittle fracture like cast iron components.

(ii) In ceramics, ductility is almost zero. This is mainly due to the presence of small voids in the structure of ceramic parts.

(iii) Ceramics have poor tensile strength.

(iv) There is a wide variation in strength values of ceramics. Even in identical specimens, the properties vary due to variation of internal pores. Hence, in design of ceramic components, a statistical approach is essential for calculating the values of strength.

(v) Ceramics are difficult to shape and machine.

In this article, we will consider only recent applications of advanced ceramics in the automobile industry. A number of parts of automotive engines are nowadays made of ceramics. They include cylinder liners, pistons, valves and engine blocks. The principal advantages of ceramic engine components over conventional metal parts are as follows:

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(i) ability to withstand higher operating temperature;
(ii) excellent wear and corrosion resistance;
(iii) lower frictional loss;
(iv) ability to operate without cooling system; and
(v) light weight construction with low inertia force.

Research is being conducted on gas turbine engines that employ ceramic rotors, stators, regenerators and combustion housings. Other applications include turbine blades for aircraft engines and surface coatings for the engine parts.

2.16 PLASTICS

Plastics are synthetic materials processed by heat and pressure. They are perhaps the most widely used group of polymers. There are two terms, which are used in understanding the construction of plastics, viz., monomer and polymer. *A monomer is a group of atoms that constitutes one unit of a polymer chain.* When monomers are subjected to heat and pressure, they join together to form a chain called *polymer. A polymer is a non-metallic organic compound of high molecular weight consisting of a very long chain of monomers.* The process of combining monomers into polymers is called *polymerization.* Figure 2.7 shows the construction of typical monomers and their corresponding polymers. In this figure, atoms of carbon, hydrogen and other elements are represented by their chemical symbols and their bonds by radial lines.

When a short polymer chain is lengthened by adding more and more monomer units, the material becomes more dense and passes from gaseous state to liquid state, from liquid state to semi-solid state and finally becomes a tough solid material. Let us consider the example of addition of (CH_2) unit to a polymer chain as shown in Fig. 2.8.

(i) Initial composition is (CH_4) which is methane gas.
(ii) Addition of one unit of (CH_2) to a methane molecule results in heavier ethane gas with (C_2H_6) composition.
(iii) Further addition of (CH_2) units to the ethane chain results in pentane, which is in liquid form with (C_5H_{12}) composition.

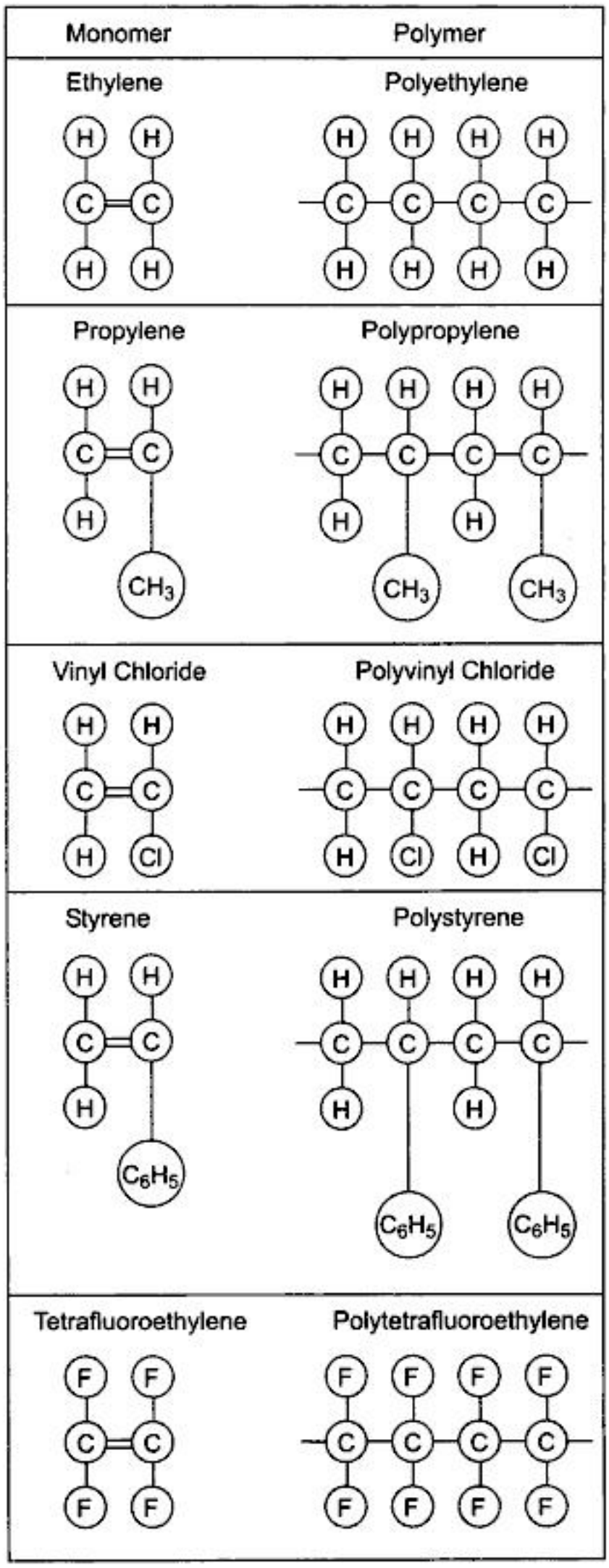

Fig. 2.7 *Construction of Monomer and Polymer*

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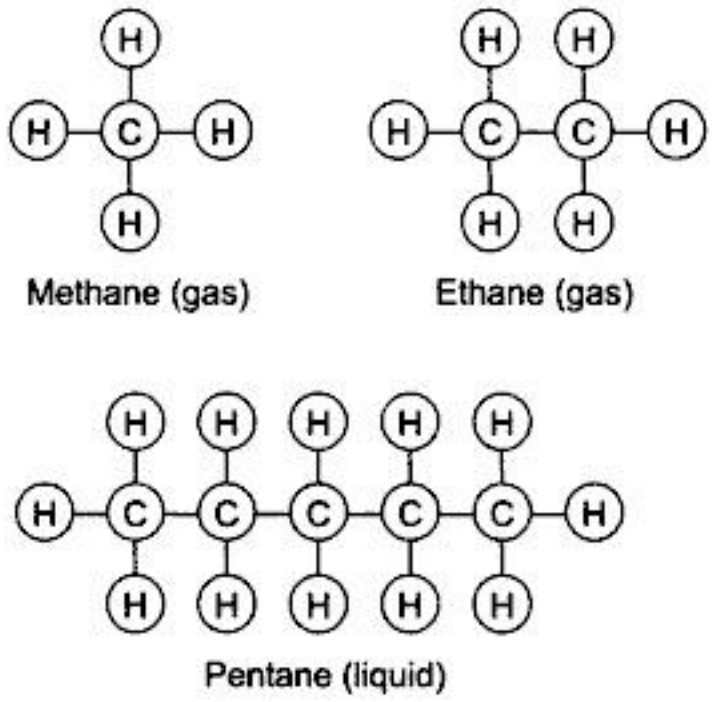

Fig. 2.8 *Monomer Chains*

(iv) If the process of adding (CH_2) units to the pentane chain is continued, paraffin wax is obtained. It is in a semi-solid stage with ($C_{18}H_{38}$) composition.

(v) If the process is further continued, a solid plastic called low-density polyethylene is obtained at approximately (C_{100} H_{102}) composition.

(vi) In the next stage, high-density polyethylene is obtained. It contains about half-million (CH_2) units in a single chain. It is a very tough solid plastic.

Adding a terminal link called *terminator*, which satisfies the bonds at each end of the chain, stops the linking of monomer units.

Knowing the construction of a monomer and polymer, we can define the term *plastic* at this stage. *A plastic can be defined as a solid material consisting of an organic polymer of a long molecular chain and high molecular weight.* It may also contain some additives like fillers, plasticisers, flame retardants and pigments. A *filler* is an inert foreign substance added to a polymer to improve certain properties such as tensile and compressive strengths, abrasion resistance, toughness and dimensional and thermal stability. Filler materials include finely powdered sawdust, silica floor and sand, clay, limestone and talc. A *plasticiser* is low-molecular weight polymer additive that improves flexibility, ductility and toughness and reduces brittleness and stiffness. They include polyvinyl chloride and acetate copolymers. A *flame retardant* is an additive which increases flammability resistance. Most polymers are flammable in their pure form. A flame retardant interferes with the combustion process and prevents burning. A *pigment* or *colourant* imparts a specific colour to the plastic material.

Plastics are divided into two basic groups depending on their behaviour at elevated temperatures, viz., thermoplastics and thermosetting plastics. *A thermoplastic is a polymeric material which softens when heated and hardens upon cooling. A thermosetting plastic is a polymeric material, which once having cured or hardened by a chemical reaction does not soften or melt upon subsequent heating.* In short, a thermoplastic softens with heat while a thermosetting plastic does not. A thermoplastic material can be moulded and remoulded repeatedly. This difference in properties of thermoplastic and thermosetting plastic materials is due to molecular structures of their polymer chains. A thermoplastic material has a linear polymer chain while a thermosetting plastic material consists of a cross-linked polymer chain as

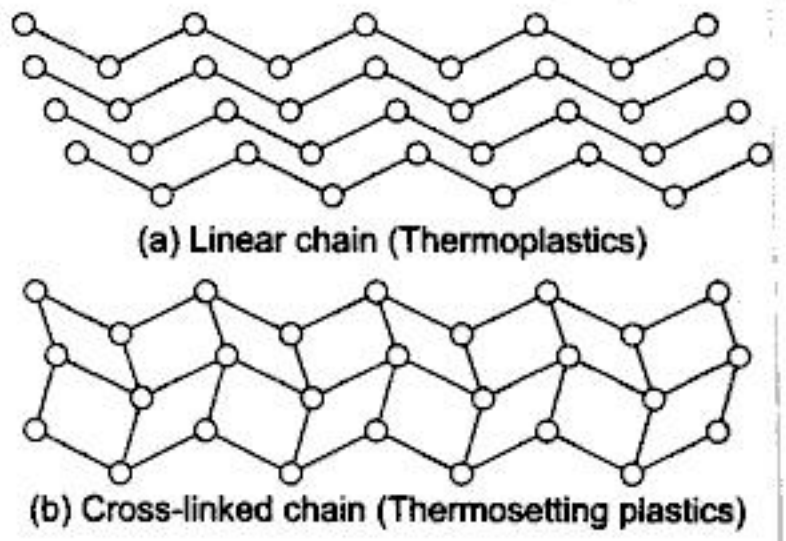

Fig. 2.9 *Linear and Cross-linked Polymer Chains*

shown in Fig. 2.9. The difference between the two categories of plastic is as follows:

(i) A thermoplastic material has a linear polymer chain. A thermosetting plastic material has cross-linked polymer chain.

(ii) A thermoplastic material can be softened, hardened or resoftened repeatedly by the application of heat. Alternate heating and

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cooling can reshape them. On the other hand, thermosetting plastic materials, once set and hardened, cannot be remelted or reshaped.

(iii) Thermoplastic materials can be recycled. Therefore, thermoplastic components are environmentally friendly. It is not possible to recycle a thermosetting plastic material. Disposal of components made of thermosetting plastic material, after their useful life, creates a problem.

(iv) Molecules in a linear chain can slide over each other. Therefore, thermoplastic materials are flexible. On the other hand, cross-linked thermosetting materials are more rigid. Their rigidity increases with the number of cross-links.

(v) Common examples of thermoplastic materials are polyethylene, polypropylene, polyvinylchloride (PVC), polystyrene, polytetrafluoethylene (PTFE) and nylon. Common examples of thermosetting plastic materials are phenolics, aminos, polyesters, epoxies and phenal-formaldehyde.

As a material for machine component, plastics offer the following advantages:

(i) Plastic materials have low specific gravity resulting in lightweight construction. The specific gravity of the heaviest plastic is 2.3 compared with 7.8 of cast iron.

(ii) Plastics have high corrosion resistance in any atmospheric condition. This is the most important advantage of plastic materials over metals. Many varieties of plastic materials are acid-resistant and can endure chemicals for a long period of time. PVC has excellent resistance to acids and alkalis.

(iii) Some plastic materials have low coefficient of friction and self-lubricating property. The coefficient of friction of polytetrafluoroethylene, commonly called *Teflon*, is as low as 0.04. Such materials are ideally suitable for bearings.

(iv) Fabrication of plastic components is easy. Raw material is available in the form of powders, granules or compressed pills. The raw material is converted into plastic parts by compression moulding, injection moulding, transfer moulding or extrusion process. Compression moulding is commonly used for components made of thermosetting plastic materials. Injection moulding is widely used for parts made of thermoplastics. Complicated parts performing several functions can be moulded from plastic material in a single operation.

Plastic materials have the following disadvantages:

(i) Plastic materials have poor tensile strength compared with other construction materials. The tensile strength of plastic materials varies from 10 N/mm^2 to 80 N/mm^2.

(ii) Mechanical properties of engineering metals do not vary much within the range of ambient temperatures encountered in practice. For many polymers, particularly thermoplastic materials, the mechanical properties vary considerably with temperature in the ambient region. For example, a thermoplastic material may have a tensile strength of 70 N/mm^2 at 0°C, falling to 40 N/mm^2 at 25°C and further to 10 N/mm^2 at 80°C.

(iii) A number of polymeric materials display viscoelastic mechanical behaviour. These materials behave like a glass at low temperatures, a rubbery solid at intermediate temperatures and a viscous liquid as the temperature is further increased. Therefore, such materials are elastic at low temperature and obey Hooke's law and at high temperatures, a liquid-like behaviour prevails. At intermediate temperatures, the rubberlike solid state exhibits combined mechanical characteristics of these two extremes. This condition is called *viscoelasticity*.

(iv) Many plastic materials are susceptible to time-dependent deformation when the stress level is maintained constant. Such deformation is called *creep*. This type of deformation is significant even at room temperature and under moderate stresses, which are even below the yield strength

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of the material. Due to creep, a machine component of plastic material under load may acquire a permanent set even at room temperature.

Although a large number of plastics are developed, we will consider a few materials in this article. These materials are mainly used for machine components. The names in brackets indicate popular trade names of the material.

(i) Polyamide (Nylon, Capron Nylon, Zytel, Fosta) Polyamide is a thermoplastic material. It has excellent toughness and wear resistance. The coefficient of friction is low. It is used for gears, bearings, conveyor rollers and automotive cooling fans.

(ii) Low-density Polyethylene (Polythene) It is a thermoplastic material. It is flexible and tough, light in weight, easy to process and a low-cost material. It is used for gaskets, washers and pipes.

(iii) Acetal (Delrin) It is a thermoplastic and a strong engineering material with exceptional dimensional stability. It has low coefficient of friction and high wear resistance. It is used for self-lubricating bearings, cams and running gears.

(iv) Polyurethane (Duthane, Texin) It is a thermoplastic and a tough, abrasion-resistant and impact-resistant material. It has good dimensional properties and self-lubricating characteristics and is used for bearings, gears, gaskets and seals.

(v) Polytetrafluoroethylene (Teflon) It is a thermoplastic material. It has low coefficient of friction and self-lubricating characteristics. It can withstand a wide range of temperatures from –260 to + 250°C. It is ideally suitable for self-lubricating bearing.

(vi) Phenolic It is a thermosetting plastic material. It has low cost with a good balance of mechanical and thermal properties. It is used in clutch and brake linings as filler material. Glass reinforced phenolic is used for pulleys and sheaves.

The mechanical properties of plastic materials are given in Table 2.12.

Table 2.12 *Mechanical properties of plastics*

Material	*Specific gravity*	*Tensile strength (N/mm^2)*	*Compressive strength (N/mm^2)*
Polyamide	1.04–1.14	70	50–90
Low-density Polythene	0.92–0.94	7–20	–
Acetal	1.41–1.42	55–70	–
Polyurethane	1.21–1.26	35–60	25–80
Teflon	2.14–2.20	10–25	10–12
Phenolic	1.30–1.90	30–70	–

2.17 FIBRE REINFORCED PLASTICS

Fibre reinforced plastic (FRP) is a composite material in which the low strength of the polymeric material is increased by means of high strength fibres. There are two main constituents of fibre reinforced plastic, viz., matrix and fibres. The function of the *matrix* is to provide a rigid base for holding the fibres in correct position. The function of the *fibres* is to transmit the load acting on the component. The bond between the surface of the fibres and surrounding matrix is usually chemical. The matrix protects the fibres from surface damage and from the action of environment. The fibres used in composite should be long enough so that the bonding force between the surface of the fibre and the surrounding matrix is greater than the tensile strength of the fibre.

Two types of fibres are widely used, viz., glass and carbon fibres. The advantages of glass reinforced plastics (GRP) are as follows:

(i) Glass can be easily drawn into fibres from the molten state.
(ii) Glass is cheaper and readily available.
(iii) Glass fibre is relatively strong.
(iv) Glass is chemically inert with respect to plastic matrix materials.

The disadvantages of glass reinforced plastic are as follows:

(i) Glass reinforced plastic has poor rigidity and stiffness.

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(ii) Its application is limited up to a temperature of 300°C.

Glass reinforced plastic is used for automotive bodies, pipes, valve bodies, pump casings and storage containers. It is more popular for vehicle bodies due to low specific gravity resulting in lightweight construction.

The advantages of carbon reinforced plastics (CRP) are as follows:

(i) Carbon fibre has maximum strength compared with all other fibre materials.
(ii) Carbon fibre retains its strength at elevated temperature.
(iii) Moisture, acids and solvents at ambient temperature do not affect carbon fibre.
(iv) Carbon reinforced plastic is relatively cheap.

There is one limitation for carbon reinforced plastic. Manufacturing techniques required to produce carbon fibre are relatively complicated. Carbon reinforced plastic is used for pressure vessels, aircraft components and casings of rocket motors.

Matrix material used in GRP and CRP should remain stable at the temperatures encountered in the application. It should not be affected by moisture and surrounding atmosphere. Both thermoplastic and thermosetting plastics are used as matrix materials. Nylon is a commonly used thermoplastic, while phenolic resins are popular thermosetting plastics for the matrices of the reinforced composites.

As an engineering material for structural components, fibre reinforced plastic offers the following advantages:

(i) It has low specific gravity resulting in lightweight construction.
(ii) It has high specific strength and modulus of elasticity.
(iii) It has good resistance to fatigue failure, particularly parallel to the direction of the fibres.
(iv) It has good resistance to corrosion.

The term *specific strength* means the ratio of the tensile strength to the specific gravity of the material.

The disadvantages of fibre reinforced plastic are as follows:

(i) A composite material containing fibres in a single direction is extremely anisotropic. The tensile strength of such a material in a direction perpendicular to that of fibres may be 5% or even less than that measured in the direction of fibres. Many times, the transverse strength is less than that of the matrix material because of the presence of discontinuities and insufficient binding between the fibres and the surrounding matrix material.
(ii) The design of components made of fibre reinforced plastics is complex. It is necessary to know the direction of principal stresses in such components. The fibres are aligned along the direction of principal stresses.
(iii) The manufacturing and testing of fibre reinforced components is highly specialised.

Fibre reinforced composite is a comparatively new material. It is being increasingly used for machine and structural parts such as motor shafts, gears and pulleys. Such materials are 'custom-made' materials, which combine the desirable characteristics of two or more materials in a given required manner.

2.18 NATURAL AND SYNTHETIC RUBBERS

Natural rubber is obtained from rubber latex, which is a milky liquid obtained from certain tropical trees. It is a low cost elastomer. Different varieties of rubber are obtained by adding carbon, silica and silicates. *Vulcanised rubber* is obtained by adding sulphur, which is followed by heating. Addition of carbon makes the rubber hard. Natural rubber, in hard and semi-hard conditions, is used for belts, bushes, flexible tubes and vibration mounts. It is also used for production of coatings, protective films and adhesives. Rubber coatings provide protection in a chemical environment.

Synthetic rubber has properties similar to those of natural rubber. It can be thermoplastic or a thermosetting plastic. It is, however, costlier than

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natural rubber. A few applications of synthetic rubber are as follows:

(i) Chloroprene (Neoprene) Conveyors and V belts, brake diaphragms and gaskets

(ii) Nitrile Butadiene (NBR) Bushes for flexible coupling and rubber rollers

(iii) Polysulfide (Thikol) Gaskets, washers and diaphragms

(iv) Chlorosulfonyl Polyethylene (Hypalon) Tank lining, high temperature conveyor belts, seals and gaskets

(iv) Silicone Seals, gaskets and O-rings

2.19 CREEP

When a component is under a constant load, it may undergo progressive plastic deformation over a period of time. This time-dependent strain is called creep. *Creep is defined as slow and progressive deformation of the material with time under a constant stress*. Creep deformation is a function of stress level and temperature. Therefore, creep deformation is higher at higher temperature and creep becomes important for components operating at elevated temperatures. Creep of bolts and pipes is a serious problem in thermal power stations. The material of steam or gas turbine blades should have a low creep rate, so that blades can remain in service for a long period of time before having to be replaced due to their reaching the maximum allowable strain. These blades operate with very close clearances and permissible deformation is an important consideration in their design. Design of components working at elevated temperature is based on two criteria. Deformation due to creep must remain within permissible limit and rupture must not occur during the service life. Based on these two criteria, there are two terms—creep strength and creep rupture strength. *Creep strength of the material is defined as the maximum stress that the material can withstand for a specified length of time without excessive deformation. Creep rupture strength of the material is the maximum stress that the material can withstand for a specified length of time without rupture.*

An idealised creep curve is shown in Fig. 2.10. When the load is applied at the beginning of the creep test, the instantaneous elastic deformation *OA* occurs. This elastic deformation is followed

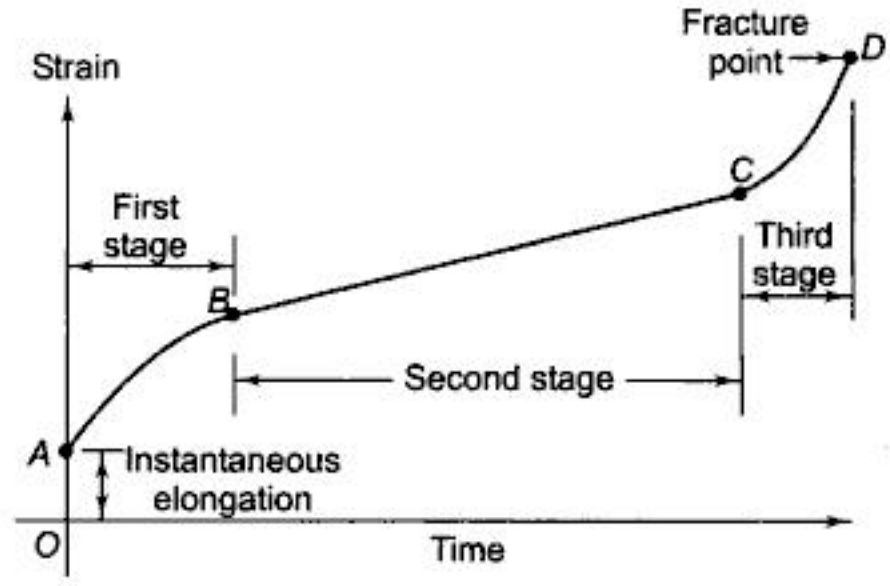

Fig. 2.10 *Creep Curve*

by the creep curve *ABCD*. Creep occurs in three stages. The first stage called *primary creep* is shown by *AB* on the curve. During this stage, the creep rate, i.e., the slope of the creep curve from *A* to *B* progressively decreases with time. The metal strain hardens to support the external load. The creep rate decreases because further strain hardening becomes more and more difficult. The second stage called *secondary creep* is shown by *BC* on the curve. During this stage, the creep rate is constant. This stage occupies a major portion of the life of the component. The designer is mainly concerned with this stage. During secondary creep, recovery processes involving highly mobile dislocations counteract the strain hardening so that the metal continues to elongate at a constant rate. The third stage called *tertiary creep* is shown by *CD* on the creep curve. During this stage, the creep rate is accelerated due to necking and also due to formation of voids along the grain boundaries. Therefore, creep rate rapidly increases and finally results in fracture at the point *D*. Creep properties are determined by experiments and these experiments involve very long periods stretching into months.

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2.20 SELECTION OF MATERIAL

Selection of a proper material for the machine component is one of the most important steps in the process of machine design. The best material is one which will serve the desired purpose at minimum cost. It is not always easy to select such a material and the process may involve the trial and error method. The factors which should be considered while selecting the material for a machine component are as follows:

(i) Availability The material should be readily available in the market, in large enough quantities to meet the requirement. Cast iron and aluminium alloys are always available in abundance while shortage of lead and copper alloys is a common experience.

(ii) Cost For every application, there is a limiting cost beyond which the designer cannot go. When this limit is exceeded, the designer has to consider other alternative materials. In cost analysis, there are two factors, namely, cost of material and the cost of processing the material into finished goods. It is likely that the cost of material might be low, but the processing may involve costly manufacturing operations.

(iii) Mechanical Properties Mechanical properties are the most important technical factor governing the selection of material. They include strength under static and fluctuating loads, elasticity, plasticity, stiffness, resilience, toughness, ductility, malleability and hardness. Depending upon the service conditions and the functional requirement, different mechanical properties are considered and a suitable material is selected. For example, the material for the connecting rod of an internal combustion engine should be capable to withstand fluctuating stresses induced due to combustion of fuel. In this case, endurance limit becomes the criterion of selection. The piston rings should have a hard surface to resist wear due to rubbing action with the cylinder surface, and surface hardness is the selection criterion. In case of bearing materials, a low coefficient of friction is desirable while clutch or brake lining requires a high coefficient of friction. The material for automobile bodies and hoods should have the ability to be extensively deformed in plastic range without fracture, and plasticity is the criterion of material selection.

(iv) Manufacturing Considerations In some applications, machinability of material is an important consideration in selection. Sometimes, an expensive material is more economical than a low priced one, which is difficult to machine. Free cutting steels have excellent machinability, which is an important factor in their selection for high strength bolts, axles and shafts. Where the product is of complex shape, castability or ability of the molten metal to flow into intricate passages is the criterion of material selection. In fabricated assemblies of plates and rods, weldability becomes the governing factor. The manufacturing processes, such as casting, forging, extrusion, welding and machining govern the selection of material.

Past experience is a good guide for the selection of material. However, a designer should not overlook the possibilities of new materials.

2.21 WEIGHTED POINT METHOD

In recent years, systematic methods have been developed for selection of materials. One such method is the weighted point method. It consists of the following four steps:

(i) The first step consists of the study of the given application and preparing a list of the desirable properties of the material for the application.

(ii) The desirable properties are then assigned values. The approximate range of these properties, such as yield strength, endurance strength, hardness, etc., is specified.

(iii) The desirable properties are divided into two groups—Go-no-go *parameters* and *discriminating parameters*. The Go-no-go parameters are the constraints. As an example, if a material is not available, or if it cannot be fabricated into a given shape, it is totally rejected. This is a screening step

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and only those materials, which meet the essential requirement, are allowed further consideration.

(iv) The discriminating parameters are the properties of the material which can be given quantitative values. The weightage depends upon the importance of that particular property in the given application. As an example, in case of a connecting rod, the endurance strength may be given a weighting factor of 5, compared with the cost having a weighting factor of 1. In general, the weighting factor varies from 1 to 5, with 1 for the poorest and 5 for the best.

Then each property of the candidate material is assigned a rating, ranging from 1 to 5, depending upon how closely it meets the requirements. These ratings are multiplied by the weighting factors for each property. These numbers are finally added and materials are arranged in descending order of their total points.

The main drawback of this method is the skill and judgement required for assigning the weightage. The results may not be numerically correct; however, one can get a priority list of materials for a given application.

Example 2.1 *It is required to select a material by the weighted point method. There are four candidate materials, viz., low alloy steel, plain carbon steel, stainless steel and chromium steel, which have passed through screening test. For a particular application, the designer has given a 5-point weightage for ultimate tensile strength, 3-point weightage for hardenability and 2-point weightage for cost-economy. Table 2.13 gives the data for the candidate materials.*

Table 2.13

Sr. No.	Material Property	Materials: Low alloy steel	Plain carbon steel	Stainless steel	Chromium steel
1	Ultimate tensile strength (N/mm^2)	850	850	1200	950
2	Hardenability Index	60	80	30	100
3	Cost (Rs / unit)	40	50	100	80

Select the most suitable material for the given application.

Solution

Part I *Calculation of weightage points for low alloy steel*

Step I *Points for ultimate tensile strength*

The sum of ultimate tensile strength of four materials is given by

$$850 + 850 + 1200 + 950 = 3850$$

Therefore, for low alloy steel, the per cent strength is given by

$$\frac{850}{3850} = 0.22$$

Since weightage for strength is 5, the points for low alloy steel are given by

$$0.22 \times 5 = 1.1 \quad \text{(a)}$$

Step II *Points for hardenability index*

The sum of hardenability index of four materials is given by

$$60 + 80 + 30 + 100 = 270$$

Therefore, for low alloy steel, the per cent hardenability index is given by

$$\frac{60}{270} = 0.222$$

Since weightage for hardenability is 3, the points for low alloy steel are given by

$$0.22 \times 3 = 0.666 \quad \text{(b)}$$

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Step III *Points for cost*

The points for cost are inversely proportional because a material with lower cost or points is a better material.

The sum of cost factor is given by

$$\frac{1}{40}+\frac{1}{50}+\frac{1}{100}+\frac{1}{80}=0.025+0.02$$
$$+0.01+0.0125=0.0675$$

Therefore, for low alloy steel, the per cent for cost factor is given by

$$\frac{0.0250}{0.0675}=0.37$$

Since weightage for cost factor is 2, the points for low alloy steel are given by

$$0.37\times 2=0.74 \qquad (c)$$

Step IV *Total points*

From (a), (b) and (c), the total points for low alloy steel are given by

$$1.1+0.666+0.74=2.506$$

Part II *Tabulation of weightage points*

Similarly, total points for other materials are calculated and given in Table 2.14.

Table 2.14

	Material Property	*Low alloy steel*	*Plain carbon steel*	*Stainless steel*	*Chromium steel*
(a)	Tensile strength				
	Per cent	0.22	0.22	0.312	0.247
	Points	1.10	1.10	1.56	1.235
(b)	Hardenability				
	Per cent	0.222	0.296	0.111	0.37
	Points	0.666	0.888	0.333	1.11
(c)	Cost				
	Per cent	0.37	0.296	0.148	0.185
	Points	0.74	0.592	0.296	0.37
	Total Points	2.506	2.58	2.189	2.715

Part III *Selection of material*

The list of material according to descending order of points will be

(i) Chromium steel (2.715 points)
(ii) Plain carbon steel (2.58 points)
(iii) Low alloy steel (2.506 points)
(iv) Stainless steel (2.189 points)

Therefore, for this particular application, chromium steel is selected as the best material for the component.

Short-Answer Questions

2.1 What is cast iron?
2.2 What is the percentage of carbon in cast iron and steel?
2.3 What are the advantages of cast iron form design considerations?
2.4 What are the disadvantages of cast iron form design considerations?
2.5 What is grey cast iron?
2.6 How will you designate grey cast iron?
2.7 Name the components made of grey cast iron.
2.8 What are white cast irons?
2.9 How will you designate white cast iron?
2.10 Name the components made of white cast iron.
2.11 What is spheroidal graphite cast iron?
2.12 How will you designate plain carbon steels?
2.13 What is high alloy steel?
2.14 How will you designate high alloy steels?
2.15 What is X20Cr18Ni2 designation of steel?
2.16 What is low carbon steel?
2.17 What is medium carbon steel?
2.18 What is high carbon steel?

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2.19 What is mild steel?
2.20 What is the percentage of carbon in mild steel?
2.21 Define alloy steel.
2.22 Name the various alloying elements in 'alloy' steels.
2.23 What are the advantages of alloy steel?
2.24 What are the important components made of alloy steels?
2.25 Compare cast iron and cast steel components.
2.26 Compare steel and cast steel components.
2.27 Name the components made of carbon steel castings?
2.28 Name the components made of high tensile steel castings?
2.29 What are the advantages of aluminum alloy for mechanical components?
2.30 Name the components made of aluminum alloy castings?
2.31 Name the components made of wrought aluminum alloy.
2.32 What are the advantages of copper alloys from design considerations?
2.33 What are the disadvantages of copper alloys from design considerations?
2.34 What are ceramics?
2.35 What are the advantages and drawbacks of ceramics?
2.36 What are the applications of ceramics in engineering industries?
2.37 What is plastic?
2.38 What is a monomer? Give its examples.
2.39 What is a polymer? Give its examples.
2.40 What are the types of plastics?
2.41 What is a thermoplastic? Give its examples.
2.42 What is a thermosetting plastic? Give its examples.
2.43 What is Teflon? Where do you use it?
2.44 What is fibre reinforced plastic (FRP)?
2.45 What are the advantages of fibre reinforced plastics?
2.46 What are the disadvantages of fibre reinforced plastics?
2.47 What is creep?
2.48 Explain the situations where creep is a serious problem.
2.49 What are the factors to be considered for selection of material for a machine component?
2.50 Explain the principle of weighted point method for selection of material for a machine component.

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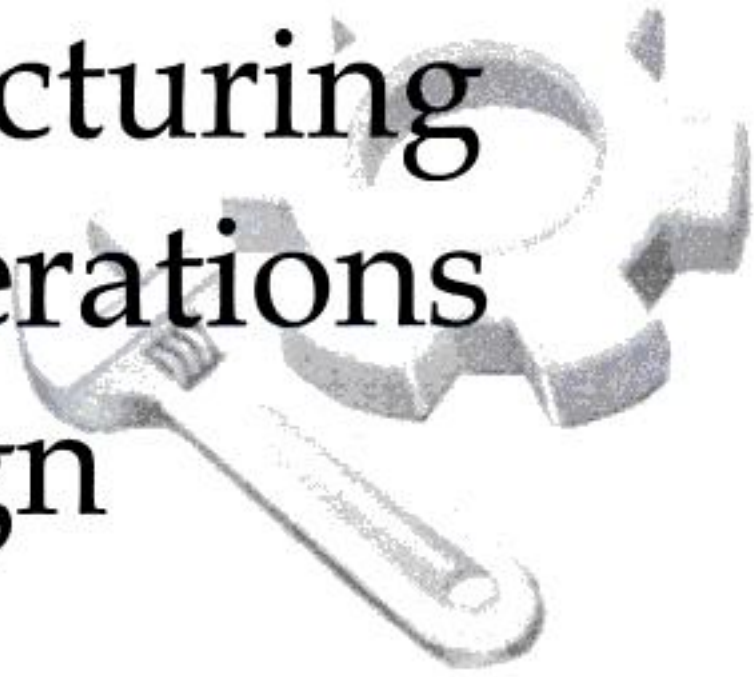
Manufacturing Considerations in Design



Chapter 3

3.1 SELECTION OF MANUFACTURING METHOD

Manufacturing of the product is an important link in the chain of events that begins with the concept of a probable product and ends with a competitive product in the market place. Product design, selection of materials and processing the materials into finished components are closely related to one another. Manufacturing can be considered as processing the available material into useful components of the product, e.g., converting a mild steel sheet into car body, converting a billet of cast iron into a machine tool bed or converting a steel bar into a transmission shaft. The manufacturing processes can be broadly classified into the following three categories:

(i) Casting Processes In these processes, molten metals such as cast iron, copper, aluminium or non-metals like plastic are poured into the mould and solidified into the desired shape, e.g., housing of gear box, flywheel with rim and spokes, machine tool beds and guides.

(ii) Deformation Processes In these processes, a metal, either hot or cold, is plastically deformed into the desired shape. Forging, rolling, extrusion, press working are the examples of deformation processes. The products include connecting rods, crankshafts, I-section beams, car bodies and springs.

(iii) Material Removal or Cutting Processes In these processes, the material is removed by means of sharp cutting tools. Turning, milling, drilling, shaping, planing, grinding, shaving and lapping are the examples of material removal processes. The products include transmission shafts, keys, bolts and nuts.

In addition, there are joining processes like bolting, welding and riveting. They are essential for the assembly of the product.

Many times, a number of manufacturing methods are available to make the component. In such cases, the optimum manufacturing method is selected by considering the following factors:

(i) Material of the component
(ii) Cost of manufacture
(iii) Geometric shape of the component
(iv) Surface finish and tolerances required
(v) Volume of production

One of the easiest methods to convert the raw material into the finished component is *casting*. There are several casting processes such as sand casting, shell-mould casting, permanent mould casting, die casting, centrifugal casting or investment casting. *Sand casting* is the most popular casting process. The advantages of sand casting process as a manufacturing method are as follows:

(i) The tooling required for casting process is relatively simple and inexpensive. This

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reduces the cost. Sand casting is one of the cheapest methods of manufacturing.

(ii) Almost any metal such as cast iron, aluminium, brass or bronze can be cast by this method.

(iii) Any component, even with a complex shape, can be cast. There is no limit on the size of the component. Even large components can be cast.

The disadvantages of the sand casting process are as follows:

(i) It is not possible to achieve close tolerances for cast components. Therefore, cast components require additional machining and finishing, which increases cost.

(ii) Cast components have a rough surface finish.

(iii) Long and thin sections or projections are not possible for cast components.

One of the important deformation processes is *forging*. In forging, the metal in the plastic stage, rather than in the molten stage, is forced to flow into the desired shape. There are a number of forging processes such as hand forging, drop forging, press forging or upset forging. The *drop-forging method* accounts for more than 80% of the forged components. The advantages of forging as a manufacturing method are as follows:

(i) The fibrelines of a forged component can be arranged in a predetermined way to suit the direction of external forces that will act on the component when in service. Therefore, forged components have inherent strength and toughness. They are ideally suitable for applications like connecting rods and crankshafts.

(ii) In forging, there is relatively good utilisation of material compared with machining.

(iii) Forged components can be provided with thin sections, without reducing the strength. This results in lightweight construction.

(iv) The tolerances of forged parts can be held between close limits, which reduce the volume of material removal during the final finishing stages.

(v) The forging process has a rapid production rate and good reproducibility.

The disadvantages of the forging process are as follows:

(i) Forging is a costly manufacturing method. The equipment and tooling required for forging is costly.

(ii) Forging becomes economical only when the parts are manufactured on a large scale.

Material removal or cutting processes are the most versatile and most common manufacturing methods. Almost every component is subjected to some kind of machining operation in its final finishing stage. Metal removal processes are broadly divided into three categories—metal cutting processes, grinding processes and unconventional machining processes. Depending upon the shape of machined surfaces, the metal removal processes are selected in the following way:

(i) For machining flat surfaces, shaping, planing and milling processes are usually used. A flat surface can also be machined on a lathe by the facing operation. Broaching and surface grinding are finishing methods for flat surfaces.

(ii) For machining external cylindrical surface, turning on lathe is a popular method. Such surfaces are finished by the cylindrical grinding method.

(iii) For machining internal cylindrical surfaces, drilling and boring are popular processes. Reaming and cylindrical grinding are finishing processes.

The advantages of metal cutting processes as a manufacturing method are as follows:

(i) Almost any metal can be machined.

(ii) It is possible to achieve close tolerances for machined components.

(iii) Machined components have a good surface finish.

The disadvantages of machining processes are as follows:

(i) Machining processes are costly and the rate of production is low compared with casting or forging.

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(ii) It is not possible to machine thin sections or projections.

(iii) There is wastage of material during material removal process.

In drilling operation, the cost of the hole increases linearly with the depth of the hole. However, when the depth is more than three times the diameter, the cost increases more rapidly.

3.2 DESIGN CONSIDERATIONS OF CASTINGS

Complex parts, which are otherwise difficult to machine, are made by the casting process using sand mould. Almost any metal can be melted and cast. Most of the sand cast parts are made of cast iron, aluminium alloys and brass. The size of the sand casting can be as small as 10 g and as large as 200×10^3 kg. Sand castings have irregular and grainy surfaces and machining is required if the part is moving with respect to some other part or structure. Cast components are stable, rigid and strong compared with machined or forged parts. Typical examples of cast components are machine tool beds and structures, cylinder blocks of internal combustion engines, pumps and gear box housings.

Poor shaping of a cast iron component can adversely affect its strength more than the composition of the material. Before designing castings, the designer should consult the foundry man and the patternmaker, whose cooperation is essential for a successful design. The general principles for the design of casting[1] are as follows:

Always Keep the Stressed Areas of the Part in Compression Cast iron has more compressive strength than its tensile strength. The balanced sections with equal areas in tension and compression are not suitable for cast iron components. The castings should be placed in such a way that they are subjected to compressive rather than tensile stresses as illustrated in Fig. 3.1. When tensile stresses are unavoidable, a clamping device such as a tie rod or a bearing cap as illustrated in Fig. 3.2 should be considered. The clamping device relieves the cast iron components from tensile stresses.

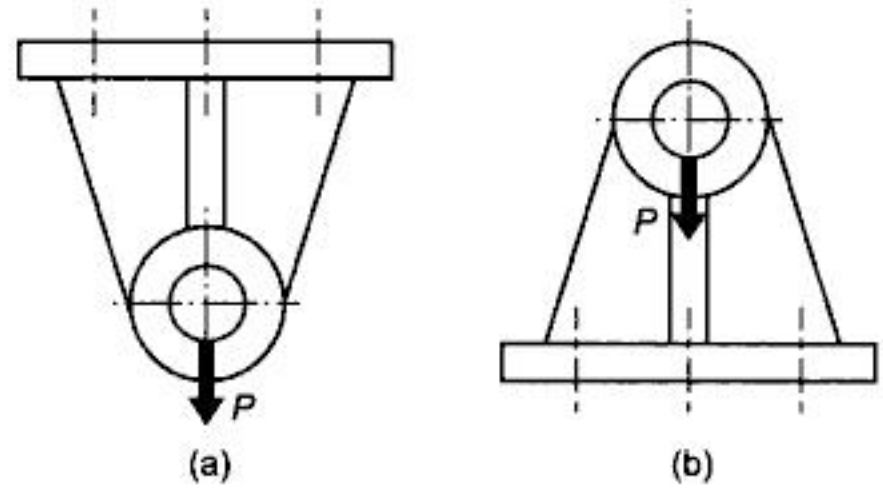

Fig. 3.1 *(a) Incorrect (Part in Tension) (b) Correct (Part in Compression)*

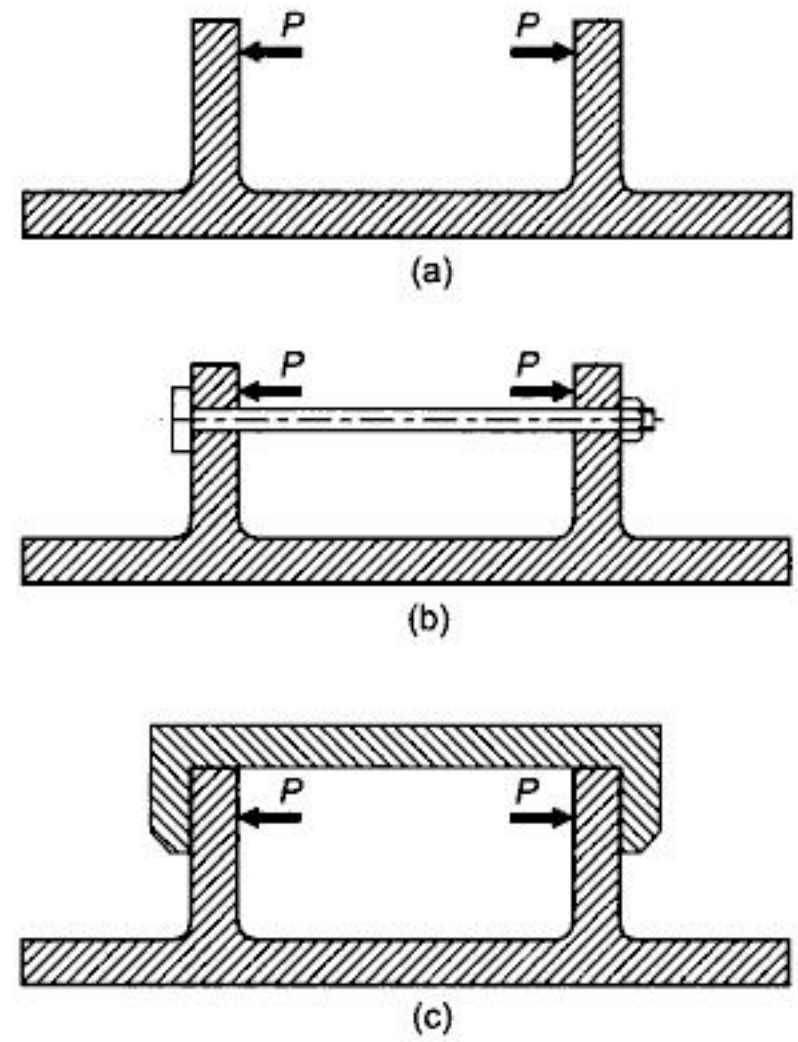

Fig. 3.2 *(a) Original Component (b) Use of Tie-rod (c) Use of Bearing-cap*

Round All External Corners It has two advantages—it increases the endurance limit of the component and reduces the formation of brittle chilled edges. When the metal in the corner cools faster than the metal adjacent to the corner, brittle chilled edges are formed due to iron carbide.

[1] John B. Caine, 'Five ways to improve grey iron-castings', Machine Design, vol. 36, no. 25, October 22, 1964.

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Appropriate fillet radius, as illustrated in Fig. 3.3 reduces the stress concentration. The values of the corner radii for different section thickness are given in Table 3.1.

Table 3.1

Wall thickness (mm)	*Inside corner radius (mm) (minimum)*
0–30	10
30–50	15
50–80	20
80–120	30

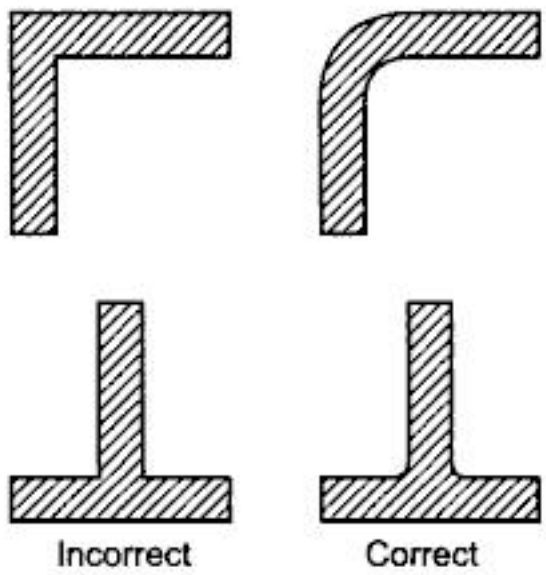

Fig. 3.3 *Provision of Fillet Radius*

Wherever Possible, the Section Thickness throughout should be Held as Uniform as Compatible with Overall Design Considerations Abrupt changes in the cross-section result in high stress concentration. If the thickness is to be varied at all, the change should be gradual as illustrated in Fig. 3.4.

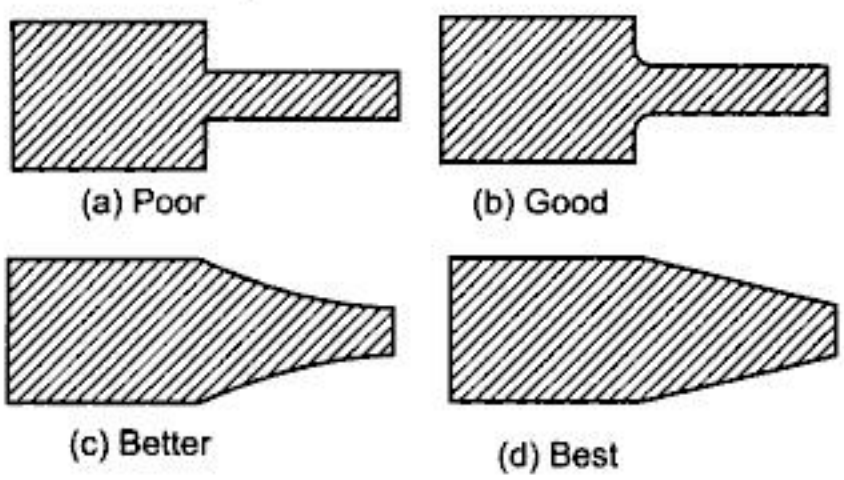

Fig. 3.4 *Change in Section-thickness*

Avoid Concentration of Metal at the Junctions At the junction as shown in Fig. 3.5, there is a concentration of metal. Even after the metal on the surface solidifies, the central portion still remains in the molten stage, with the result that a shrinkage cavity or blowhole may appear at the centre. There are two ways to avoid the concentration of metal. One is to provide a cored opening in webs and ribs, as illustrated in Fig. 3.6. Alternatively, one can stagger the ribs and webs, as shown in Fig. 3.7.

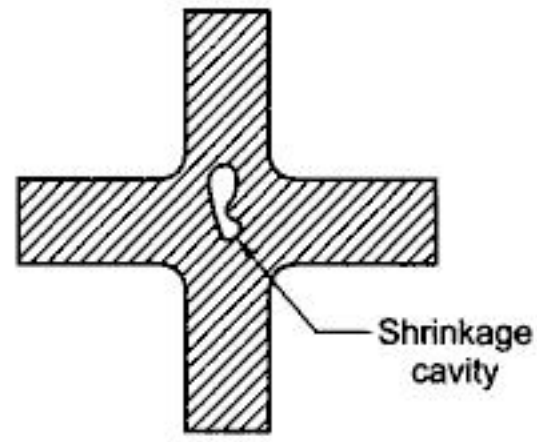

Fig. 3.5

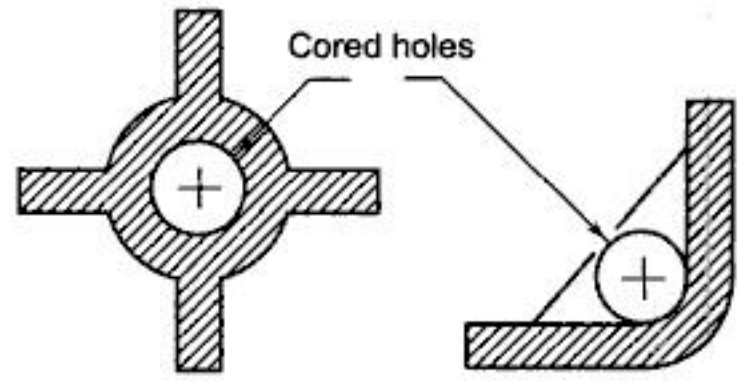

Fig. 3.6 *Cored Holes*

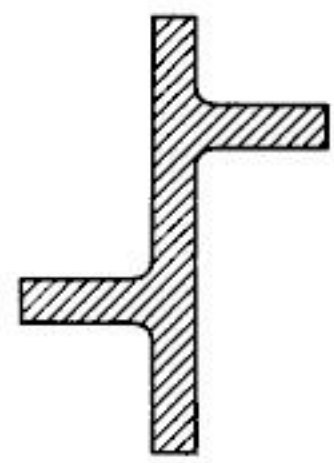

Fig. 3.7 *Staggered Ribs*

Avoid Very Thin Sections In general, if the thickness of a cast iron component is calculated from strength considerations, it is often too small. In such cases, the thickness should be increased to certain practical proportions. The minimum section thickness depends upon the process of casting, such as sand casting, permanent mould casting or die

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casting. The minimum thickness for a grey cast iron component is about 7 mm for parts up to 500 mm long, which gradually increases to 20 mm for large and heavy castings.

Shot Blast the Parts wherever Possible The shot blasting process improves the endurance limit of the component, particularly in case of thin sections.

Some ways to improve the strength of castings are illustrated in Figs 3.8 to 3.11. In Fig. 3.8, the inserted stud will not restore the strength of the original thickness. The wall adjacent to the drilled hole should have a thickness equivalent to the thickness of the main body. Figure 3.9 shows cored holes in webs or ribs. Oval-shaped holes are preferred with larger dimensions along the direction of forces. Patterns without a draft make a mould difficult and costly. A minimum draft of 3° should be provided, as illustrated in Fig. 3.10. Outside bosses should be omitted to facilitate a straight pattern draft as shown in Fig. 3.11.

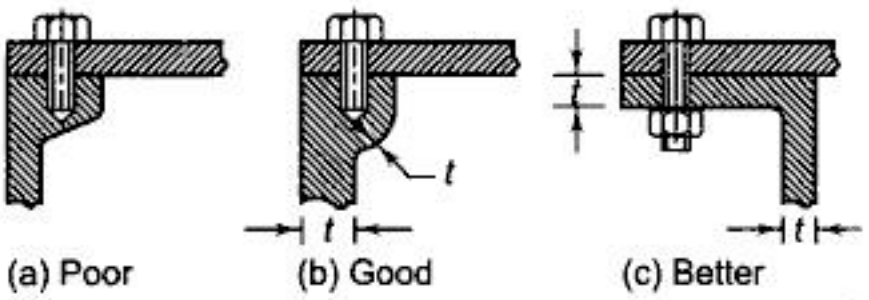

Fig. 3.8 *Uniform Wall-thickness*

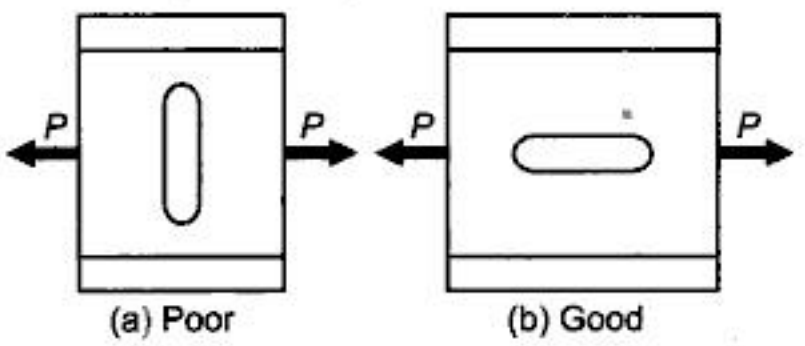

Fig. 3.9 *Cored Holes in Ribs*

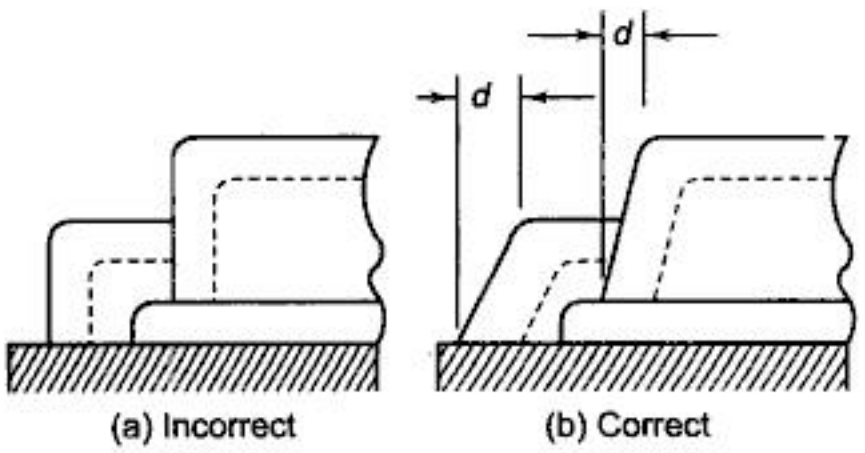

Fig. 3.10 *Provision of Draft*

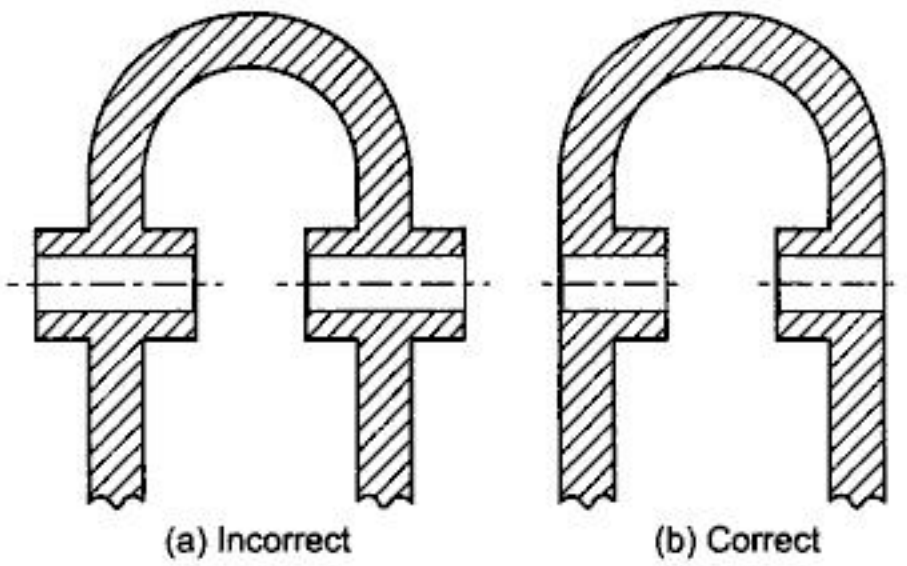

Fig. 3.11

3.3 DESIGN CONSIDERATIONS OF FORGINGS

Forged components are widely used in automotive and aircraft industries. They are usually made of steels and non-ferrous metals. They can be as small as a gudgeon pin and as large as a crankshaft. Forged components are used under the following circumstances:

(i) Moving components requiring light weight to reduce inertia forces, e.g., connecting rod of IC engines.

(ii) Components subjected to excessive stresses, e.g., aircraft structures.

(iii) Small components that must be supported by other structures or parts, e.g., hand tools and handles.

(iv) Components requiring pressure tightness where the part must be free from internal cracks, e.g., valve bodies.

(v) Components whose failure would cause injury and expensive damage are forged for safety.

In order to obtain maximum benefit from forged components, the following principles should be adopted:

(i) While designing a forging, advantage should be taken of the direction of fibre lines. The grain structure of a crankshaft manufactured by the three principal methods, viz.,

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casting, machining and forging, is shown in Fig. 3.12. There are no fibre lines in the cast component and the grains are scattered. In case of a component prepared by machining methods, such as turning or milling, the original fibre lines of rolled stock are broken. It is only in case of forged parts that the fibre lines are arranged in a favourable way to withstand stresses due to external load. While designing a forging, the profile is selected in such a way that fibre lines are parallel to tensile forces and perpendicular to shear forces. Machining that cuts deep into the forging should be avoided, otherwise the fibre lines are broken and the part becomes weak.

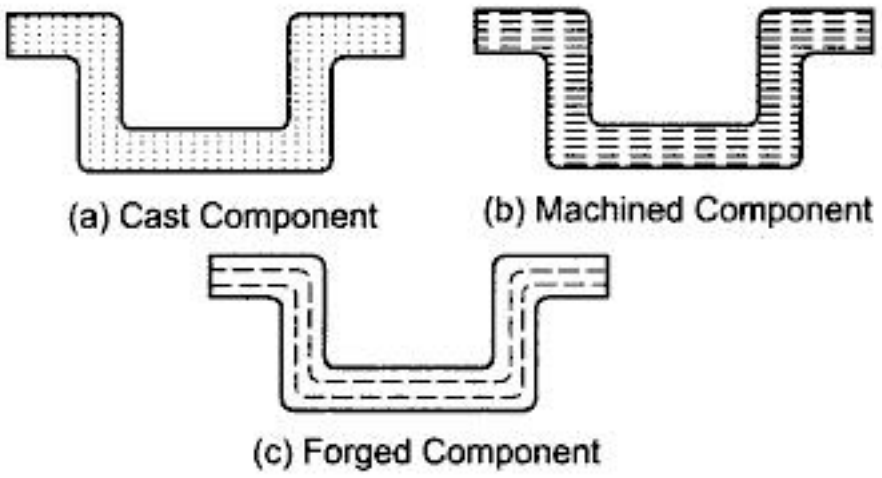

Fig. 3.12 *Grain Structure*

(ii) The forged component should be provided with an adequate draft as illustrated in Fig. 3.13. The draft angle is provided for an easy removal of the part from the die impressions. The angles α and β are drafts for outside and inside surfaces. As the material cools, it shrinks, and a gap is formed between the outer surface of the forging and the inner surfaces of the die cavity, with the result that the draft angle for the outer surface is small. On the other hand, when the material cools, its inner surfaces tend to shrink and grip the projecting surface of the die, with the result that the draft angle for the inner surface is large. For steels, the recommended values of α and β are 7° and 10° respectively.

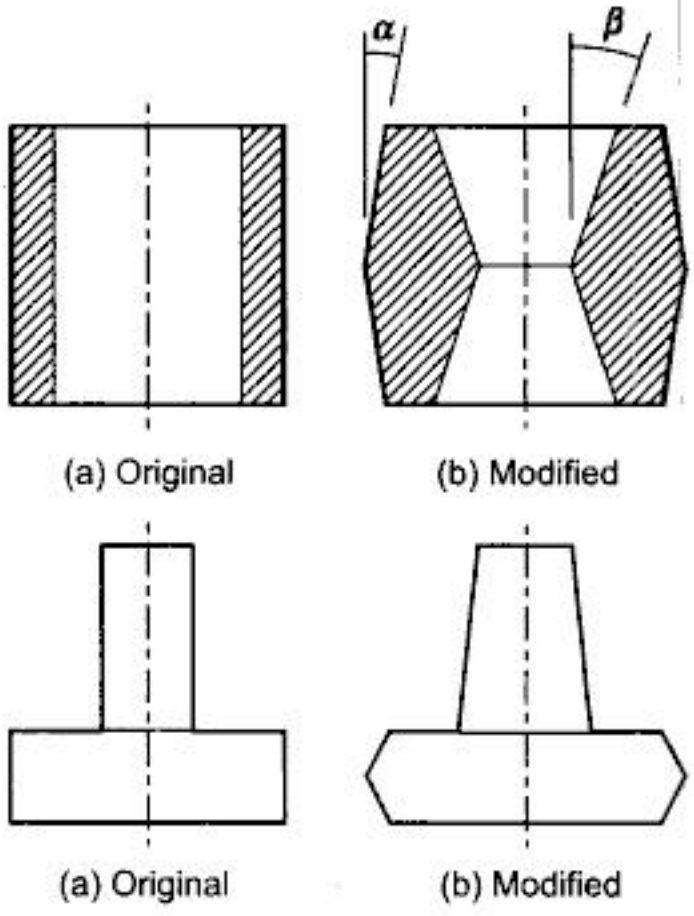

Fig. 3.13 *Draft for Forgings*

(iii) There are two important terms related to forgings, as illustrated in Fig. 3.14. The *parting line* is a plane in which the two halves of the forging dies meet and in which flash is formed. A *forging plane* is a plane, which is perpendicular to the die motion. In most of the cases, the parting line and forging plane coincide, as illustrated in Fig. 3.15. There are two basic principles for the location of the parting line—the parting line should be in one plane as far as possible and it should divide the forging into two equal parts. When the parting line is broken, as shown in Fig. 3.16, it results in unbalanced forging forces, which tend to displace the two die halves. Such forces are balanced either by a counter lock or by forging the two components simultaneously in a mirror-image position. A parting line that divides the forging into two halves ensures the minimum depth to which the steel must flow to fill the die impressions.

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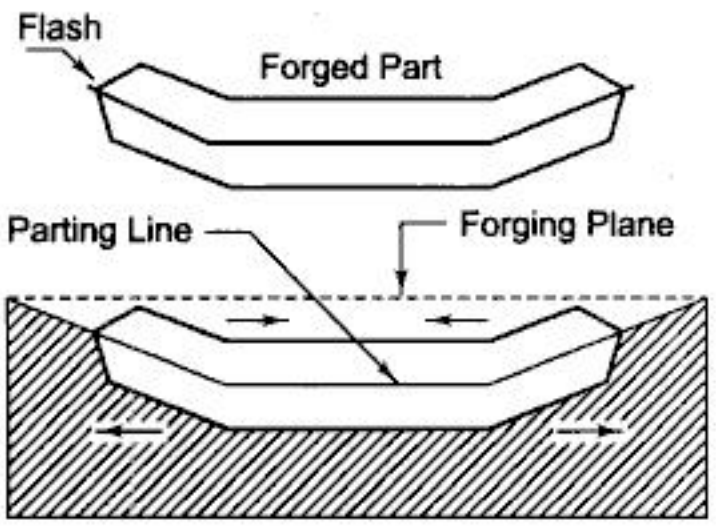

Fig. 3.14

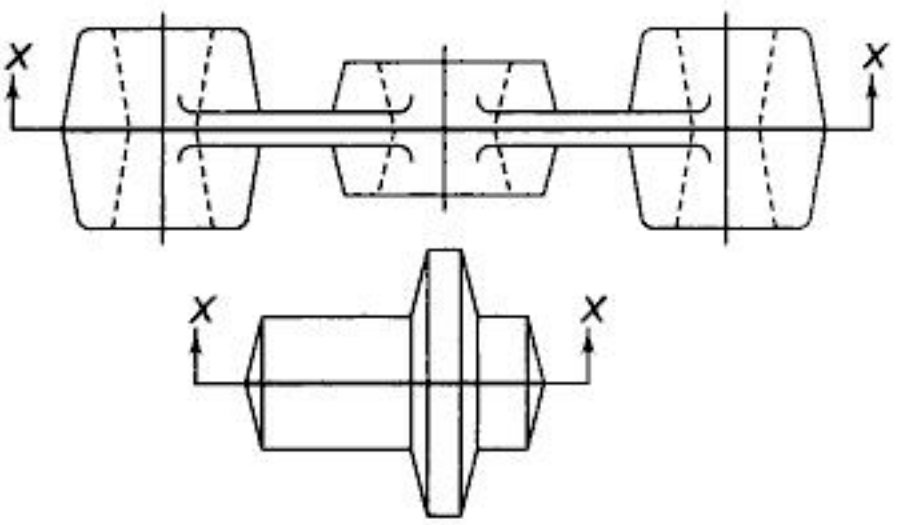

Fig. 3.15 *Location of Parting Line and Forging Plane (XX)*

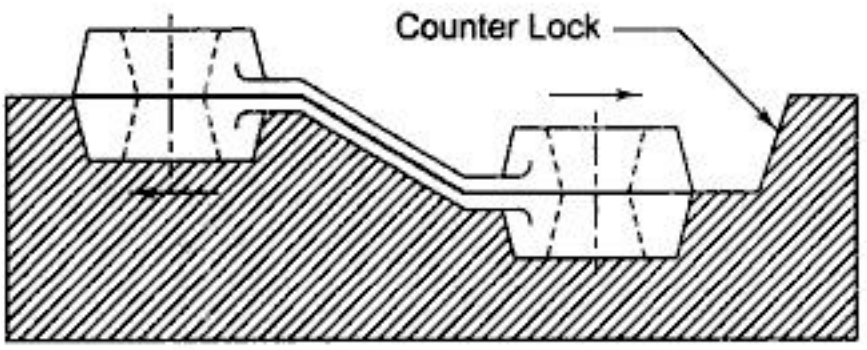

Fig. 3.16 *Unbalanced Forces*

(iv) The forging should be provided with adequate fillet and corner radii. A small radius results in folds on the inner surface and cracks on the outer surface. A large radius is undesirable, particularly if the forged component is to be machined, during which the fibre lines are broken. Sharp corners result in increasing difficulties in filling the material, excessive forging forces, and poor die life. The magnitude of fillet radius depends upon the material, the size of forging and the depth of the die cavity. For moderate size steel forgings, the minimum corner radii are 1.5, 2.5 and 3.5 mm for depths up to 10, 25 and 50 mm respectively.

(v) Thin sections and ribs should be avoided in forged components. A thin section cools at a faster rate in the die cavity and requires excessive force for plastic deformation. It reduces the die life, and the removal of the component from the die cavities becomes difficult. For steel forgings, the recommended value of the minimum section thickness is 3 mm.

A properly designed forging is not only sound with regard to strength but it also helps reduce the forging forces, improves die life and simplifies die design. If the design is poor, the best of steel and forging methods will not give a satisfactory component.

3.4 DESIGN CONSIDERATIONS OF MACHINED PARTS

Machined components are widely used in all industrial products. They are usually made from ferrous and non-ferrous metals. They are as small as a miniature gear in a wristwatch and as large as a huge turbine housing. Machined components are used under the following circumstances:

(i) Components requiring precision and high dimensional accuracy
(ii) Components requiring flatness, roundness, parallelism or circularity for their proper functioning
(iii) Components of interchangeable assembly
(iv) Components, which are in relative motion with each other or with some fixed part

The general principles for the design of machined parts are as follows:

(i) Avoid Machining Machining operations increase cost of the component. Components made by casting or forming methods are usually cheaper. Therefore, as far as possible, the designer should avoid machined surfaces.

(ii) Specify Liberal Tolerances The secondary machining operations like grinding or reaming are costly. Therefore, depending upon the functional

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requirement of the component, the designer should specify the most liberal dimensional and geometric tolerances. Closer the tolerance, higher is the cost.

(iii) Avoid Sharp Corners Sharp corners result in stress concentration. Therefore, the designer should avoid shapes that require sharp corners.

(iv) Use Stock Dimensions Raw material like bars are available in standard sizes. Using stock dimensions eliminates machining operations. For example, a hexagonal bar can be used for a bolt, and only the threaded portion can be machined. This will eliminate machining of hexagonal surfaces.

(v) Design Rigid Parts Any machining operation such as turning or shaping induces cutting forces on the components. The component should be rigid enough to withstand these forces. In this respect, components with thin walls or webs should be avoided.

(vi) Avoid Shoulders and Undercuts Shoulders and undercuts usually involve separate operations and separate tools, which increase the cost of machining.

(vii) Avoid Hard Materials Hard materials are difficult to machine. They should be avoided unless such properties are essential for the functional requirement of the product.

3.5 HOT AND COLD WORKING OF METALS

The temperature at which new stress-free grains are formed in the metal is called the *recrystallization temperature*. There are two types of metal deformation methods, namely, hot working and cold working. *Metal deformation processes that are carried out above the recrystallization temperature are called hot working processes*. Hot rolling, hot forging, hot spinning, hot extrusion, and hot drawing are hot working processes. *Metal deformation processes that are carried out below the recrystallization temperature are called cold working processes*. Cold rolling, cold forging, cold spinning, cold extrusion, and cold drawing are *cold working processes*. Hot working processes have the following advantages:

(i) Hot working reduces strain hardening.
(ii) Hot rolled components have higher toughness and ductility. They have better resistance to shocks and vibrations.
(iii) Hot working increases the strength of metal by refining the grain structure and aligning the grain of the metal with the final counters of the part. This is particularly true of forged parts.
(iv) Hot working reduces residual stresses in the component.

Hot working processes have the following disadvantages:

(i) Hot working results in rapid oxidation of the surface due to high temperature.
(ii) Hot rolled components have poor surface finish than cold rolled parts.
(iii) Hot working requires expensive tools.

Cold working processes have the following advantages:

(i) Cold rolled components have higher hardness and strength.
(ii) Cold worked components have better surface finish than hot rolled parts.
(iii) The dimensions of cold rolled parts are very accurate.
(iv) The tooling required for cold working is comparatively inexpensive.

Cold working processes have the following disadvantages:

(i) Cold working reduces toughness and ductility. Such components have poor resistance to shocks and vibrations.
(ii) Cold working induces residual stresses in the component. Proper heat treatment is required to relieve these stresses.

3.6 DESIGN CONSIDERATIONS OF WELDED ASSEMBLIES

Welding is the most important method of joining the parts into a complex assembly. Design of welded joints is explained in a separate chapter on welded joints. In this article, general principles in design of welded assemblies are discussed. The guidelines are as follows:

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(i) Select the Material with High Weldability In general, a low carbon steel is more easily welded than a high carbon steel. Higher carbon content tends to harden the welded joint, as a result of which the weld is susceptible to cracks. For ease in welding, maximum carbon content is usually limited to 0.22 per cent[2].

(ii) Use Minimum Number of Welds Distortion is a serious problem in welded assemblies. It creates difficulties in maintaining correct shape, dimensions and tolerances of finished assemblies. A metallic plate or component does not distort, when it is heated or cooled as a total unit uniformly and it has freedom to expand or contract in all directions. In welding, however, only the adjoining area of the joint is heated up, which has no freedom to expand or contract. Uneven expansion and contraction in this adjoining area and parent metal results in distortion. When distortion is prevented by clamping fixtures, residual stresses are built up in the parts and annealing is required to relieve these stresses. Since distortion always occurs in welding, the design should involve a minimum number of welds and avoid over welding. It will not only reduce the distortion but also the cost.

(iii) Do not Shape the Parts Based on Casting or Forging In designing a welded assembly to replace a casting, it is incorrect to duplicate its appearance or shape by providing protrusions, brackets and housing. The designer should appreciate that welded assemblies are different from castings, having an appearance of their own. A correctly designed welded assembly is much lighter than the corresponding casting. It should reflect its lightweight characteristic, flexibility and economy of the material.

(iv) Use Standard Components The designer should specify standard sizes for plates, bars and rolled sections. Non-standard sections involve flame cutting of plates and additional welding. Standard tubular sections should be used for torsional resistance. As far as possible, the designer should select plates of equal thickness for a butt joint.

(v) Avoid Straps, Laps and Stiffeners The stiffness of a plate can be increased by making bends, indentations in the form of ribs or corrugations by press working. If at all a stiffener is required to provide rigidity to the plate, it should be designed properly with minimum weight. Use of a separate stiffener involves additional welding increasing distortion and cost.

(vi) Select Proper Location for the Weld There are two aspects of selecting the correct location for a welded joint. The welded joint should be located in an area where stresses and deflection are not critical. Also, it should be located at such a place that the welder and welding machine has unobstructed access to that location. It should be possible to carry out pre-weld machining, post-weld heat treatment and finally weld inspection at the location of the weld.

(vii) Prescribe Correct Sequence of Welding The designer should consider the sequence in which the parts should be welded together for minimum distortion. This is particularly important for a complex job involving a number of welds. An incorrect sequence of welding causes distortion and sometimes cracks in the weld metal due to stress concentration at some point in fabrication. A correct welding sequence distributes and balances the forces and stresses induced by weld contraction. For example, to prevent angular distortion in a double V butt joint, a sequence is recommended to lay welding runs alternatively on opposite sides of the joint.

Examples of 'incorrect' and 'correct' ways of welded design are illustrated from Fig. 3.17 to Fig. 3.19. In Fig. 3.17(a), it is necessary to prepare bevel edges for the components prior to welding operation. This preparatory work can be totally eliminated by making a slight change in the arrangement of components, which is shown in Fig. 3.17(b). Many times, fabrication is carried out by cutting steel plates followed by welding. The aim

[2] IS 1871–1965: Commentary on Indian Standard wrought steels for general engineering purposes.

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of the designer is to minimise scrap in such process. This is illustrated in Fig. 3.18. The circular top plate and annular bottom plate are cut from two separate plates resulting in excess scrap as shown in Fig. 3.18(a). Making a slight change in design, the top plate and annual bottom plate can be cut from one plate reducing scrap and material cost, which is shown in Fig. 3.18(b). Accumulation of welded joints results in shrinkage stresses. A method to reduce this accumulation is illustrated in Fig. 3.19.

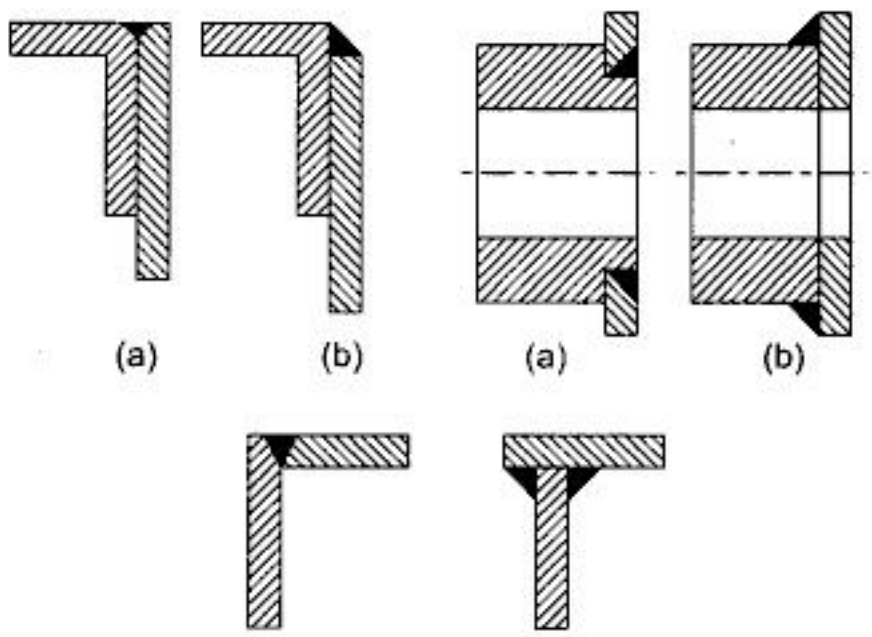

Fig. 3.17 *Saving of Preparatory Bevelling: (a) Incorrect (b) Correct*

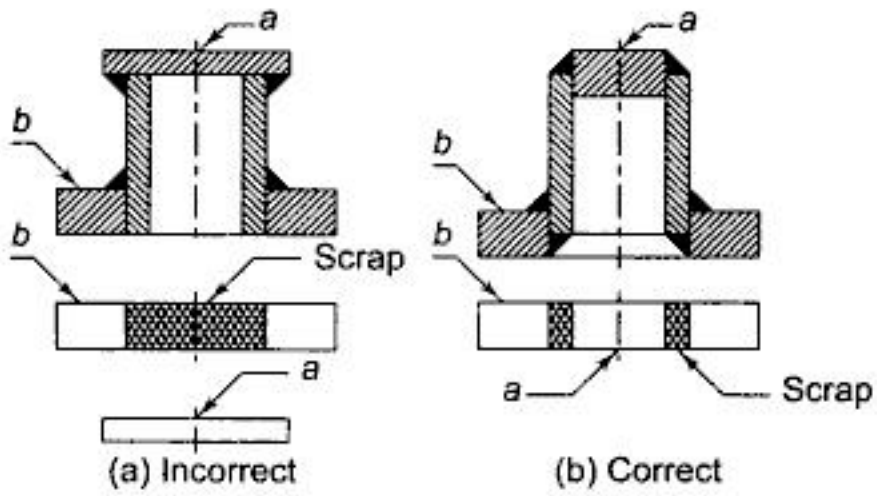

Fig. 3.18 *Reduction of Scrap*

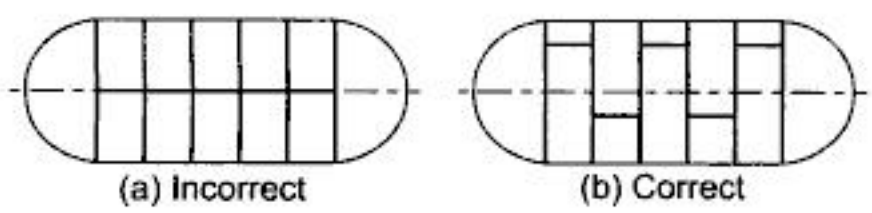

Fig. 3.19 *Avoiding Weld Accumulation*

3.7 DESIGN FOR MANUFACTURE AND ASSEMBLY (DFMA)

The design effort makes up only about 5% of the total cost of a product. However, it usually determines more than 70% of the manufacturing cost of the product. Therefore, at best only 30% of the product's cost can be changed once the design is finalised and drawings are prepared. Using statistical process control or improving manufacturing methods during production stage after the design has been finalised has marginal or little effect on the product's cost. The best strategy to lower product cost is to recognise the importance of manufacturing early in the design stage. Design for manufacture and assembly are simple guidelines formulated by Bart Huthwaite, Director of Institute for Competitive Design, Rochester, USA[3]. These guidelines, although simple, are used to simplify design, decrease assembly cost, improve product reliability and reduce operation time required to bring a new product into the market. Little technology is required to implement these guidelines. The guidelines are as follows:

(i) Reduce the Parts Count Design engineers should try for a product design that uses a minimum number of parts. Fewer parts result in lower costs. It also makes the assembly simpler and less prone to defects.

(ii) Use Modular Designs Modular design reduces the number of parts being assembled at any one time and also simplifies final assembly. Field service becomes simple, fast and cheap because dismantling is faster and requires fewer tools.

(iii) Optimize Part Handling Parts should be designed so that they do not become tangled, struck together or require special handling prior to assembly. Flexible parts such as those with wires or cables should be avoided because they complicate automated assembly. Whenever possible, parts should retain the same orientation from the point of manufacture to assembly.

[3] Bart Huthwaite—*Checklist for DFM*—Machine Design—February 8, 1990—vol. 8, no. 3.

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(iv) Assemble in the Open Assembly operation should be carried out in clear view. This is important for manual assembly. It also decreases the chances of manufacturing defects slipping past the inspector.

(v) Do not Fight Gravity Design products so that they can be assembled from the bottom to top along the vertical axis. This allows simple robots and insertion tools because gravity is used to assist the assembly process. On the other hand, expensive clamping fixtures are required for assembly along horizontal axes.

(vi) Design for Part Identity For both manual and automated assembly, symmetric parts are easier to handle and orient. As assembly rate increases, symmetry becomes more important. Features should be added to enhance symmetry. Asymmetric parts should be designed so that their other surfaces make them easily identifiable. Asymmetry can be added or exaggerated to force correct alignment and orientation and make mistakes impossible.

(vii) Eliminate Fasteners Fasteners are a major obstacle to efficient assembly and should be avoided wherever possible. They are difficult to handle and can cause jamming, if defective. *In manual assembly, the cost of driving a screw can be six to ten times the cost of the screw itself.* If the use of fasteners cannot be avoided, limit the number of different types of fasteners used.

(viii) Design Parts for Simple Assembly Assembling parts can be a major challenge in automated operations. Misalignment is a serious problem when parts from different vendors are put together. Part compliance is the ability of one part to move so that it is seated properly with another. The product should be designed for part compliance. Features such as added chamfers on both parts and adequate guiding surfaces make assembly faster and more reliable.

(ix) Reduce, Simplify and Optimise Manufacturing Process The number of processes needed to assemble a product should be kept to a minimum. Processes that are difficult to control, such as welding or brazing, should be avoided.

3.8 TOLERANCES

Due to the inaccuracy of manufacturing methods, it is not possible to machine a component to a given dimension. The components are so manufactured that their dimensions lie between two limits—maximum and minimum. The basic dimension is called the *normal* or *basic size*, while the difference between the two limits is called *permissible tolerance*. *Tolerance is defined as permissible variation in the dimensions of the component.* The two limits are sometimes called the *upper* and *lower deviations*. These definitions are illustrated in Fig. 3.20, with reference to shaft and hole in clearance fit.

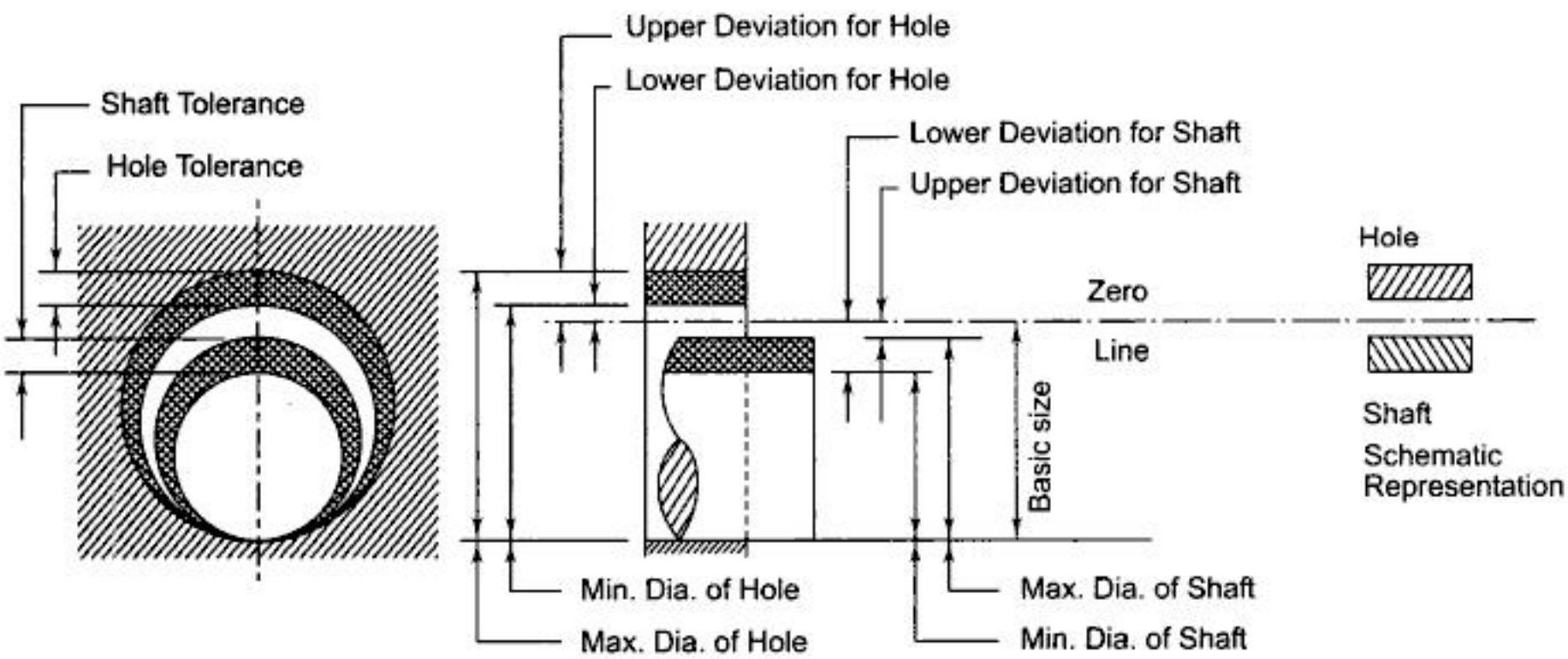

Fig. 3.20 *Tolerance Nomenclature*

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There are two systems of specification for tolerances, namely, unilateral and bilateral. In the *unilateral system*, one tolerance is zero, while the other takes care of all permissible variation in basic size. For example,

$$100^{+0.04}_{+0.00} \quad \text{or} \quad 100^{+0.00}_{-0.04}$$

In case of *bilateral tolerances*, the variations are given in both directions from normal size. The upper limit in this case is the basic size plus non-zero positive tolerance, and the lower limit is the basic size plus non-zero negative tolerance. For example,

$$25^{\pm 0.4} \quad \text{or} \quad 25^{+0.4}_{-0.2}$$

In bilateral tolerances, the two tolerances are often equal, but this is not a necessary condition. Unilateral tolerances are used for shafts and holes.

3.9 TYPES OF FITS

When two parts are to be assembled, the relationship resulting from the difference between their sizes before assembly is called a fit. Depending upon the limits of the shaft and the hole, fits are broadly classified into three groups—clearance fit, transition fit and interference fit. *Clearance fit is a fit which always provides a positive clearance between the hole and the shaft over the entire range of tolerances.* In this case, the tolerance zone of the hole is entirely above that of the shaft. *Interference fit is a fit which always provides a positive interference over the whole range of tolerances.* In this case, the tolerance zone of the hole is completely below that of the shaft. *Transition fit is a fit which may provide either a clearance or interference, depending upon the actual values of the individual tolerances of the mating components.* In this case, the tolerance zones of the hole and the shaft overlap. These definitions are illustrated in Fig. 3.21.

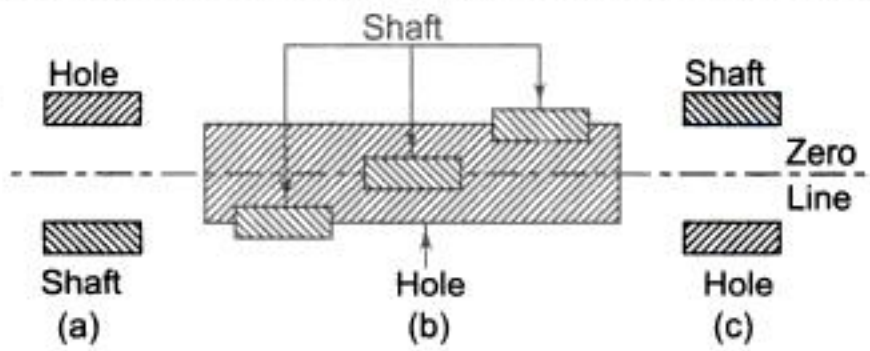

Fig. 3.21 *Types of Fits: (a) Clearance Fit (b) Transition Fit (c) Interference Fit*

There are two basic systems for giving tolerances to the shaft and the hole, namely, the hole-basis system and the shaft-basis, system. In the *hole-basis system*, the different clearances and interferences are obtained by associating various shafts with a single hole, whose lower deviation is zero. The system is illustrated in Fig. 3.22. In this case, the size of the hole is the basic size, and the clearance or interference is applied to the shaft dimension. The system is denoted by the symbol 'H'. This system has an advantage over the shaft-basis system, because holes are machined by standard drills or reamers having fixed dimensions, while the shafts can be turned or ground to any given dimension. Due to this reason, the hole-basis system is widely used.

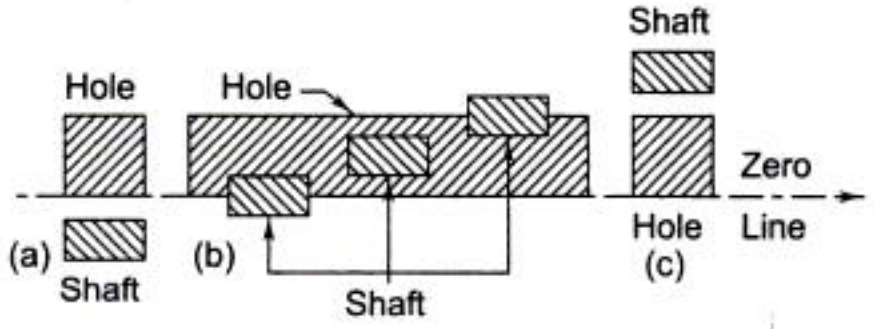

Fig. 3.22 *Hole Basis System: (a) Clearance Fit (b) Transition Fit (c) Interference Fit*

In the *shaft-basis system*, the different clearances or interferences are obtained by associating various holes with a single shaft, whose upper deviation is zero. This principle is illustrated in Fig. 3.23. In this system, the size of the shaft is the basic size, while the clearance or interference is applied to the dimensions of the hole. The system is denoted by the symbol 'h'. The shaft-basis system is popular in industries using semi-finished or finished shafting, such as bright bars, as raw material.

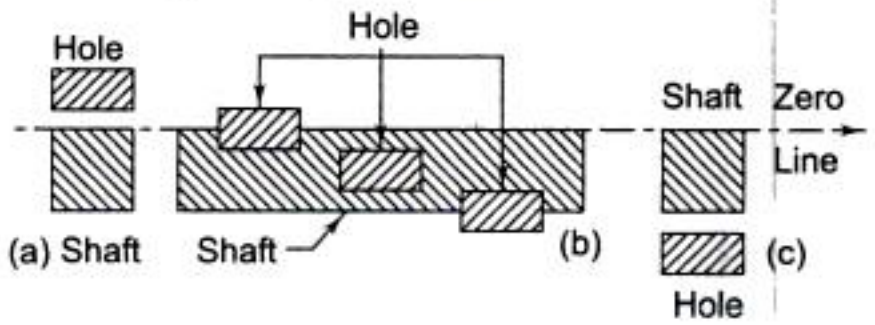

Fig. 3.23 *Shaft Basis System: (a) Clearance Fit (b) Transition Fit (c) Interference Fit*

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3.10 BIS SYSTEM OF FITS AND TOLERANCES

According to a system recommended by the Bureau of Indian Standards,[4, 5, 6] tolerance is specified by an alphabet, capital or small, followed by a number, e.g., H7 or g6. The description of tolerance consists of two parts—*fundamental deviation* and *magnitude of tolerance*, as shown in Fig. 3.24. The fundamental deviation gives the location of the tolerance zone with respect to the zero line. It is indicated by an alphabet—capital letters for holes and small letters

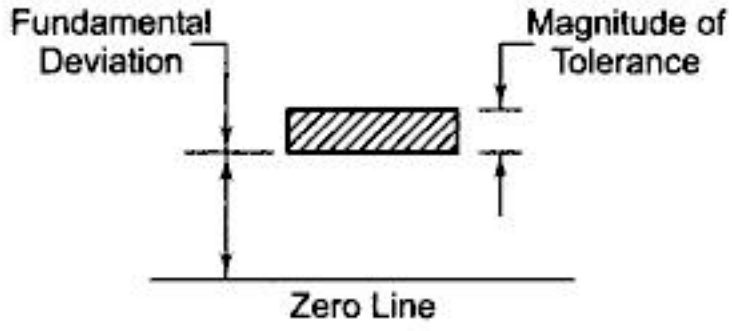

Fig. 3.24 *Description of Tolerance*

for shafts. The magnitude of tolerance is designated by a number, called the *grade*. *The grade of tolerance is defined as a group of tolerances, which are considered to have the same level of accuracy for all basic sizes*. There are eighteen grades of tolerances with designations IT1, IT2, ..., IT17, and IT18. The letters of symbol IT stand for 'International Tolerance' grade. The tolerance for a shaft of 50 mm diameter as the basic size, with the fundamental deviation denoted by the letter 'g' and the tolerance of grade 7 is written as 50g7.

The fit is indicated by the basic size common to both components followed by symbols for tolerance of each component. For example,

$$50\ \text{H8/g7} \quad \text{or} \quad 50\ \text{H8-g7} \quad \text{or} \quad 50\,\frac{\text{H8}}{\text{g7}}$$

The formulae for calculating the fundamental deviation and magnitude of tolerance of various grades are given in the standards. However, the designer is mainly concerned with readymade tables for determining tolerances. Table 3.2 and Tables 3.3a and 3.3b give tolerances for holes and shafts up to 100-mm size. For other sizes and grades, one can refer to relevant standards.

Table 3.2 *Tolerances for holes of sizes up to 100 mm (H5 to H11)*

Diameter steps in mm		*H*							
		5	*6*	*7*	*8*	*9*	*10*	*11*	*5–11*
over	*to*				*es*				*ei*
0	3	+4	+6	+10	+14	+25	+40	+60	0
3	6	+5	+8	+12	+18	+30	+48	+75	0
6	10	+6	+9	+15	+22	+36	+58	+90	0
10	18	+8	+11	+18	+27	+43	+70	+110	0
18	30	+9	+13	+21	+33	+52	+84	+130	0
30	50	+11	+16	+25	+39	+62	+100	+160	0
50	80	+13	+19	+30	+46	+74	+120	+190	0
80	100	+15	+22	+35	+54	+87	+140	+220	0

[4] IS 919—1993: ISO System of limits and fits (in two parts).
[5] IS 2709—1964: Guide for selection of fits.
[6] IS 2101—1990: Recommendations for limits and fits for sizes above 500 mm and up to 3150 mm.

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Table 3.3a *Tolerances for shafts of sizes up to 100 mm (from d to h)*

Diameter steps in mm		d					e					f				g			h						
		8–11	8	9	10	11	6–9	6	7	8	9	6–8	6	7	8	6–7	6	7	5–10	5	6	7	8	9	10
over	to	es	ei				es	ei				es	ei			es	ei		es	ei					
0	3	–20	–34	–45	–60	–80	–14	–20	–24	–28	–39	–6	–12	–16	–20	–2	–8	–12	0	–4	–6	–10	–14	–25	–40
3	6	–30	–48	–60	–78	–105	–20	–28	–32	–38	–50	–10	–18	–22	–28	–4	–12	–16	0	–5	–8	–12	–18	–30	–48
6	10	–40	–62	–76	–98	–130	–25	–34	–40	–47	–61	–13	–22	–28	–35	–5	–14	–20	0	–6	–9	–15	–22	–36	–58
10	18	–50	–77	–93	–120	–160	–32	–43	–50	–59	–75	–16	–27	–34	–43	–6	–17	–24	0	–8	–11	–18	–27	–43	–70
18	30	–65	–98	–117	–149	–195	–40	–53	–61	–73	–92	–20	–33	–41	–53	–7	–20	–28	0	–9	–13	–21	–33	–52	–84
30	50	–80	–119	–142	–180	–240	–50	–66	–75	–89	–112	–25	–41	–50	–64	–9	–25	–34	0	–11	–16	–25	–39	–62	–100
50	80	–100	–146	–174	–220	–290	–60	–79	–90	–106	–134	–30	–49	–60	–76	–10	–29	–40	0	–13	–19	–30	–46	–74	–120
80	100	–120	–174	–207	–260	–340	–72	–94	–107	–126	–159	–36	–58	–71	–90	–12	–34	–47	0	–15	–22	–35	–54	–87	–140

Table 3.3b *Tolerances for shafts of sizes up to 100 mm (from j to s)*

Diameter steps in mm		j						k			m			n			p			r			s			
		5		6		7		5	6	5–6	6	7	6–7	6	7	6–7	6	7	6–7	5	6	5–6	5	6	7	5–7
over	to	es	ei	es	ei	es	ei	es		ei	es		ei	es		ei	es		ei	es		ei	es			ei
0	3	+2	–2	+4	–2	+6	–4	+4	+6	0	+8	–	+2	+10	+14	+4	+12	+16	+6	+14	+16	+10	+18	+20	+24	+14
3	6	+3	–2	+6	–2	+8	–4	+6	+9	+1	+12	+16	+4	+16	+20	+8	+20	+24	+12	+20	+23	+15	+24	+27	+31	+19
6	10	+4	–2	+7	–2	+10	–5	+7	+10	+1	+15	+21	+6	+19	+25	+10	+24	+30	+15	+25	+28	+19	+29	+32	+38	+23
10	18	+5	–3	+8	–3	+12	–6	+9	+12	+1	+18	+25	+7	+23	+30	+12	+29	+36	+18	+31	+34	+23	+36	+39	+46	+28
18	30	+5	–4	+9	–4	+13	–8	+11	+15	+2	+21	+29	+8	+28	+36	+15	+35	+43	+22	+37	+41	+28	+44	+48	+56	+35
30	50	+6	–5	+11	–5	+15	–10	+13	+18	+2	+25	+34	+9	+33	+42	+17	+42	+51	+26	+45	+50	+34	+54	+59	+68	+43
50	80	+6	–7	+12	–7	+18	–12	+15	+21	+2	+30	+41	+11	+39	+50	+20	+51	+62	+32	+55	+61	+42	+69	+75	+86	+56
80	100	+6	–9	+13	–9	+20	–15	+18	+25	+3	+35	+48	+13	+45	+58	+23	+59	+72	+37	+66	+73	+54	+86	+93	+106	+71

Note: 1. In Tables 3.1 and 3.2a and b, the tolerances are given in microns (1 micron = 0.001 mm).
2. es = upper deviation and ei = lower deviation.

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3.11 SELECTION OF FITS

The guidelines for the selection of clearance fits are as follows:

(i) The fits H7-d8, H8-d9 and H11-d11 are loose running fits, and are used for plumber-block bearings and loose pulleys.

(ii) The fits H6-e7, H7-e8 and H8-e8 are loose clearance fits, and are used for properly lubricated bearings, requiring appreciable clearances. The finer grades are used for heavy-duty, high-speed bearings and large electric motors.

(iii) The fits H6-f6, H7-f7 and H8-f8 are normal running fits, widely used for grease or oil lubricated bearings having low temperature rise. They are also used for shafts of gearboxes, small electric motors and pumps.

(iv) The fits H6-g5, H7-g6 and H8-g7 are expensive from manufacturing considerations. They are used in precision equipment, pistons, slide valves and bearings of accurate link mechanisms.

The typical types of transition fits are H6-j5, H7-j6 and H8-j7. They are used in applications where slight interference is permissible. Some of their applications are spigot and recess of the rigid coupling and the composite gear blank, where a steel rim is fitted on an ordinary steel hub.

The general guidelines for the selection of interference fits are as follows:

(i) The fit H7-p6 or H7-p7 results in interference, which is not excessive but sufficient to give non-ferrous parts a light press fit. Such parts can be dismantled easily as and when required, e.g., fitting a brass bush in the gear.

(ii) The fit H6-r5 or H7-r6 is a medium drive fit on ferrous parts, which can be easily dismantled.

(iii) The fits H6-s5, H7-s6 and H8-s7 are used for permanent and semi-permanent assemblies of steel and cast iron parts. The amount of interference in these fits is sufficiently large to provide a considerable gripping force. They are used in valve seats and shaft collars.

The selection of interference fit depends upon a number of factors, such as materials, diameters, surface finish and machining methods. It is necessary to calculate the maximum and minimum interference in each case. The torque transmitting capacity is calculated for minimum interference, while the force required to assemble the parts is decided by the maximum value of interference.

3.12 TOLERANCES AND MANUFACTURING METHODS

The specification of machining method for a given component depends upon the grade of tolerances specified by the designer. The approximate relationship between the grade of tolerance and the manufacturing method with desirable accuracy under normal working conditions, is as follows:

Grade 16 Sand casting–flame cutting
Grade 15 Stamping
Grade 14 Die casting–moulding
Grade 11 Drilling–rough turning–boring
Grade 10 Milling–slotting–planning–rolling–extrusion
Grade 9 Horizontal and vertical boring–turning on automatic lathes
Grade 8 Turning, boring and reaming on centre, capstan and turret lathes
Grade 7 High precision turning–broaching–honing
Grade 6 Grinding–fine honing
Grade 5 Lapping–fine grinding–diamond boring
Grade 4 Lapping

In this chapter, the hole-basis system is discussed. The manufacturing processes for obtaining different grades of tolerances for holes are as follows:

H5 Precision boring–fine internal grinding–honing
H6 Precision boring–honing–hand reaming
H7 Grinding–broaching–precision reaming
H8 Boring–machine reaming

Finer grades of tolerances result in costly manufacturing methods. They are to be specified only on grounds of functional requirement. Coarse

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grades of tolerances reduce the cost of production. The designer should always take into consideration the available manufacturing facilities and their cost competitiveness before specifying the grade of tolerances.

Example 3.1 *The main bearing of an engine is shown in Fig. 3.25. Calculate*

(i) the maximum and minimum diameters of the bush and crank pin; and

(ii) the maximum and minimum clearances between the crank pin and bush.

Suggest suitable machining methods for both.

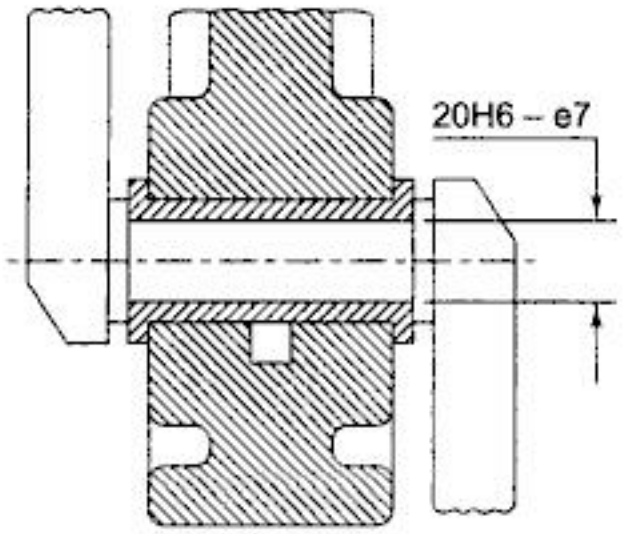

Fig. 3.25 *Engine Bearing*

Solution

Step I *Maximum and minimum diameters of the bush and crank pin*

The tolerances for the bush and crank pin from Tables 3.2 and 3.3a are as follows:

Bush:

$$20^{+0.013}_{+0.000} \quad \text{or} \quad \frac{20.013}{20.000}\ \text{mm}$$

Crankpin:

$$20^{-0.040}_{-0.061} \quad \text{or} \quad \frac{19.960}{19.939}\ \text{mm} \qquad \text{(i)}$$

Step II *Maximum and minimum clearances*

Maximum clearance = 20.013 – 19.939 = 0.074 mm

Minimum clearance = 20.000 – 19.960 = 0.040 mm

(ii)

Step III *Machining methods*

The bush (H6) is usually machined by precision boring, honing or hand reaming methods. The crank pin (e7) is machined by high precision turning or broaching.

Example 3.2 *The valve seat fitted inside the housing of a pump is shown in Fig. 3.26. Calculate*

(i) the maximum and minimum diameters of the housing and valve seat; and

(ii) the magnitude of the maximum and minimum interferences between the seat and housing.

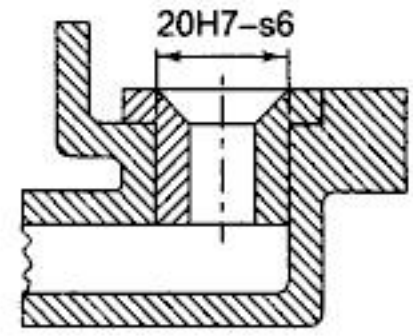

Fig. 3.26 *Valve Seat*

Solution

Step I *Maximum and minimum diameters of the housing and valve seat*

From Tables 3.2 and 3.3b, the tolerances for valve seat and housing are as follows:

Housing:

$$20^{+0.021}_{+0.000} \quad \text{or} \quad \frac{20.021}{20.000}\ \text{mm}$$

Valve seat (outer diameter):

$$20^{+0.048}_{+0.035} \quad \text{or} \quad \frac{20.048}{20.035}\ \text{mm} \qquad \text{(i)}$$

Step II *Maximum and minimum interferences*

Minimum interference = 20.035 – 20.021 = 0.014 mm

Maximum interference = 20.048 – 20.000 = 0.048 mm

(ii)

3.13 SELECTIVE ASSEMBLY

The selective assembly is a process of sorting the manufactured components into different groups according to their sizes, and then assembling the components of one group to the corresponding components of a matching group. In this method, larger shafts are assembled with larger holes and

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smaller shafts with smaller holes. This results in closer clearance or interference without involving costly machining methods.

As an example, a hydrodynamic journal bearing with a recommended class of fit 50 H8-d8 is considered. From Tables 3.2 and 3.3a, the limiting dimensions are as follows:

Bearing:

$$50^{+0.039}_{+0.000} \quad \text{or} \quad \frac{50.039}{50.000}\ \text{mm}$$

Journal:

$$50^{-0.080}_{-0.119} \quad \text{or} \quad \frac{49.920}{49.881}\ \text{mm}$$

Maximum clearance = 50.039 – 49.881 = 0.158 mm
Minimum clearance = 50.000 – 49.920 = 0.080 mm

The range of clearance is from 0.080 to 0.158 mm. Suppose it is desired to hold the clearance in a narrow range from 0.100 to 0.140 mm. Then the bearings and the journals are divided into two groups with the following limiting dimensions:

	Bearings	*Journals*
Group A	$\frac{50.039}{50.020}$	$\frac{49.920}{49.900}$
Group B	$\frac{50.020}{50.000}$	$\frac{49.900}{49.881}$

Group A
Maximum clearance = 50.039 – 49.900 = 0.139 mm
Minimum clearance = 50.020 – 49.920 = 0.100 mm

Group B
Maximum clearance = 50.020 – 49.881 = 0.139 mm
Minimum clearance = 50.000 – 49.900 = 0.100 mm

It is seen that both groups have clearance in the range 0.100 to 0.139 mm.

In the above example, only two groups are considered. When the parts are sorted into more number of groups, the range of clearance is further reduced. Selective assembly results in finer clearance or interference, even with large manufacturing tolerances. The main drawbacks of this method are as follows:

(i) Hundred per cent inspection is required for components and there is additional operation of sorting the components.
(ii) Interchangeability is affected and servicing or replacement of worn-out components becomes difficult.
(iii) The method can be used only when a large number of components are manufactured, otherwise some groups may not contain sufficient number of components.

Selective assembly is particularly useful in case of interference fits where a tight control over the range of interference is essential to avoid loosening of mating components.

Example 3.3 *The recommended class of fit for the hub shrunk on a shaft is 50H7-s6. However, it is necessary to limit the interference from 0.030 to 0.050 mm between the hub and the shaft. Specify the groups for selective assembly.*

Solution

Step I Maximum and minimum diameters of hub and shaft

From Tables 3.2 and 3.3b,

Hub:

$$50^{+0.025}_{+0.000} \quad \text{or} \quad \frac{50.025}{50.000}\ \text{mm}$$

Shaft:

$$50^{+0.059}_{+0.043} \quad \text{or} \quad \frac{50.059}{50.043}\ \text{mm}$$

Step II Maximum and minimum interferences
Maximum interference = 50.059 – 50.000 = 0.059 mm
Minimum interference = 50.043 – 50.025 = 0.018 mm

Step III Selection of groups
Suppose the components are sorted into three groups (A, B and C) with the following dimensions:

Group	*A*	*B*	*C*
Hub	$\frac{50.008}{50.000}$	$\frac{50.016}{50.008}$	$\frac{50.025}{50.016}$
Shaft	$\frac{50.048}{50.043}$	$\frac{50.055}{50.048}$	$\frac{50.059}{50.055}$

Step IV Maximum and minimum interferences for A, B and C

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The interferences I_A, I_B and I_C are as follows:

I_A max = 50.048 – 50.000 = 0.048 mm

I_B max = 50.055 – 50.008 = 0.047 mm

I_C max = 50.059 – 50.016 = 0.043 mm

The maximum interference is within the limit of 0.050 mm.

I_A min = 50.043 – 50.008 = 0.035 mm

I_B min = 50.048 – 50.016 = 0.032 mm

I_C min = 50.055 – 50.025 = 0.030 mm

The minimum interference is above the limit of 0.03 mm.

3.14 TOLERANCES FOR BOLT SPACING

When two or more components are assembled by means of bolts, it is often required to specify tolerances for the centre to centre distance between bolts. As shown in Fig. 3.27, two plates A and B are assembled by bolts with a centre distance of $(a \pm x)$. Consider the worst geometric situation, when the following conditions exist:

(i) The holes are of the smallest size, D_{min},

(ii) The bolts are of the largest size, d_{max},

(iii) The plate A has a minimum spacing $(a - x)$, and

(iv) The plate B has maximum spacing $(a + x)$.

Considering the plate A, the centre to centre distance between bolts is given by

$$(a-x)+2\left(\frac{D_{\text{min.}}}{2}\right)-2\left(\frac{d_{\text{max.}}}{2}\right)$$

or $(a-x)+(D_{\text{min.}}-d_{\text{max.}})$ (a)

Similarly, considering the plate B, the centre to centre distance is given by

$$(a+x)-2\left(\frac{D_{\text{min.}}}{2}\right)+2\left(\frac{d_{\text{max.}}}{2}\right)$$

or $(a+x)-(D_{\text{min.}}-d_{\text{max.}})$ (b)

Equating the expressions (a) and (b),

$$x=(D_{\text{min.}}-d_{\text{max.}}) \quad (3.1)$$

The tolerance on centre distance is given by

$(a \pm x)$

In case there are a number of holes, one hole is considered as the master or reference hole and the location of other holes should be provided with the required tolerance.

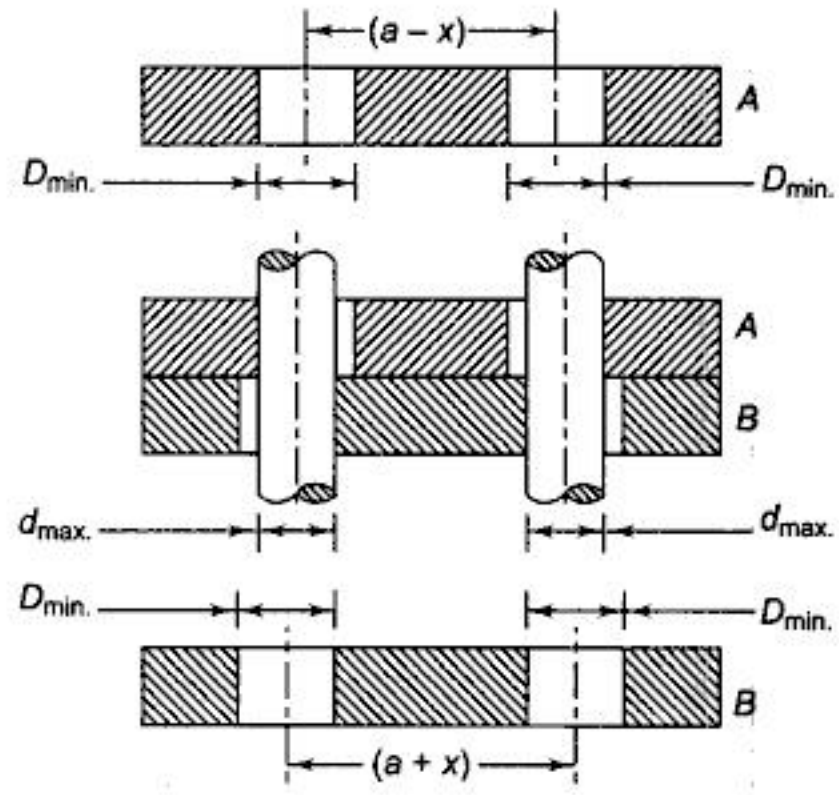

Fig. 3.27 *Assembly of Two Plates*

Example 3.4 *Two bolts, with a centre distance of 50 mm, are used to assemble two plates. The recommended class of fit between bolts and holes is 12H11-d11. Specify tolerance for the centre to centre distance between bolts.*

Solution

Step I *Maximum and minimum diameters of hole and bolt*

Hole: 12H11 (Table 3.2)

$$12^{+0.110}_{+0.000} \quad \text{or} \quad \frac{12.110}{12.000}\ \text{mm}$$

Bolt: 12d11 (Table 3.3a)

$$12^{-0.050}_{-160} \quad \text{or} \quad \frac{11.950}{11.840}\ \text{mm}$$

Step II *Calculation of x*

From Eq. (3.1),

$x = (D_{\text{min.}} - d_{\text{max.}}) = (12.000 - 11.950) = 0.05$ mm

Step III *Tolerance for the centre distance*

The tolerance for the centre distance is given as

50 ± 0.05 mm

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3.15 SURFACE ROUGHNESS

The surface roughness or surface finish plays an important role in the performance of certain machine elements. Some of the examples are as follows:

(i) Friction and wear increases with surface roughness, adversely affecting the performance of bearings.

(ii) Rough surfaces have a reduced contact area in interference fits, which reduces the holding capacity of the joints.

(iii) The endurance strength of the component is greatly reduced due to poor surface finish.

(iv) The corrosion resistance is adversely affected by poor surface finish.

It is necessary for the designer to specify an optimum surface finish from the considerations of functional requirement and the cost of manufacture. A magnified profile of the surface is shown in Fig. 3.28. There are two methods to specify surface roughness – **cla** value and **rms** value.

The **cla** value is defined as

$$\text{cla} = \frac{1}{L}\int_0^L y\,dx \qquad (3.2)$$

and the **rms** (root mean square) value as

$$\text{rms} = \left[\frac{1}{L}\int_0^L y^2\,dx\right]^{1/2} \qquad (3.3)$$

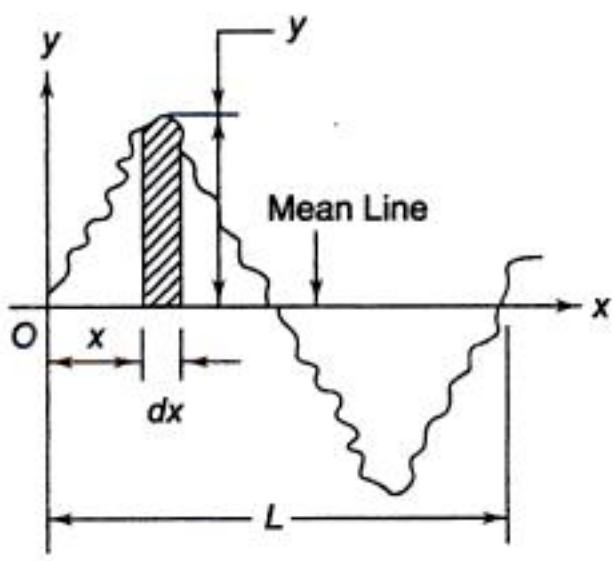

Fig. 3.28 *Surface Roughness*

A widely used symbol for surface finish is shown in Fig. 3.29. The sides of the symbol are inclined at 60° to the surface and the number indicates the surface roughness (**rms**) in microns. The surface roughness produced by various machining methods is given in Table 3.4.

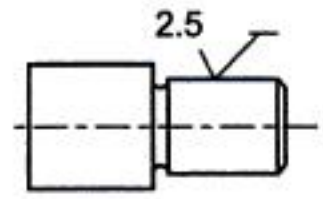

Fig. 3.29 *Symbol for Surface Roughness*

Table 3.4 *Surface roughness* (**rms**) *in microns*

Machining method	*Roughness (microns)*
turning–shaping–milling	12.5–1.0
boring	6.5–0.5
drilling	6.25–2.5
reaming	2.5–0.5
surface grinding	6.25–0.5
cylindrical grinding	2.5–0.25
honing–lapping	0.5–0.05
polishing–buffing	0.5–0.05

Typical values of surface roughness for machine components are as follows:

1.5 micron — Gear shafting and bores
0.75 micron — Bronze bearings
0.40 micron — Splined shafts, O-ring grooves, gear teeth and ball bearings
0.30 micron — Cylinder bores and pistons
0.20 micron — Crankshaft, connecting rod, cams and hydraulic cylinders

Short-Answer Questions

3.1 What is a casting process? Give examples of components made by casting.

3.2 What is deformation process? Give examples of components made by deformation process.

3.3 What is cutting process? Give examples of components made by cutting process.

3.4 What are the advantages of the sand casting process?

3.5 What are the disadvantages of the sand casting process?

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3.6 What are the advantages of the forging process?
3.7 What are the disadvantages of the forging process?
3.8 What are the advantages of the cutting process as a manufacturing method?
3.9 What are the disadvantages of the cutting process as a manufacturing method?
3.10 What are the principles of designing cast iron components?
3.11 Compare grain structure of a crankshaft manufactured by casting, forging and machining processes.
3.12 How will you select direction of fibre lines in forged components?
3.13 Where do you use machined components? Give practical examples.
3.14 What are the principles for the design of machined components?
3.15 What are the advantages of the hot working process?
3.16 What are the disadvantages of the hot working process?
3.17 What are the advantages of the cold working process?
3.18 What are the disadvantages of the cold working process?
3.19 What are the principles for design of welded assemblies?
3.20 What is DFM? What is DFMA?
3.21 What are the principles of Design for Manufacture and Assemblies (DFMA)?
3.22 What is tolerance?
3.23 What are unilateral and bilateral tolerances?
3.24 What is fit?
3.25 What is a clearance fit? Give examples.
3.26 What is a transition fit? Give examples.
3.27 What is an interference fit? Give examples.
3.28 What is the shaft-basis system for giving tolerances?
3.29 What is the hole-basis system for giving tolerances?
3.30 What are the advantages of the hole-basis system over the shaft-basis system?
3.31 What is fundamental deviation?
3.32 How will you designate fundamental deviation?
3.33 How will you designate magnitude of tolerance?
3.34 What are the guidelines for selection of clearance fits? Give examples.
3.35 What are the guidelines for selection of transition fits? Give examples.
3.36 What are the guidelines for selection of interference fits? Give examples.
3.37 What is selective assembly?
3.38 Distinguish between interchangeable and selective assemblies.
3.39 What are the advantages of selective assembly?
3.40 What are the disadvantages of selective assembly?
3.41 Explain the symbol for surface roughness.

Problems for Practice

3.1 The bush of the small end of a connecting rod is shown in Fig. 3.30. Calculate
(i) the maximum and minimum diameters of the bush and connecting rod; and
(ii) the maximum and minimum interference between them.

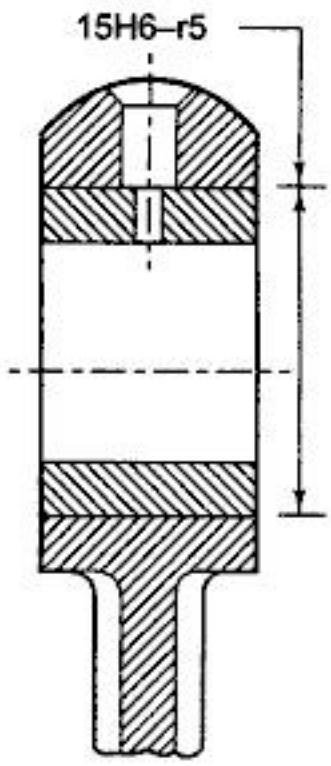

Fig. 3.30

[*(i)* $\frac{15.031}{15.023}$ *mm and* $\frac{15.011}{15.000}$ *mm*
(ii) 0.031 and 0.012 mm]

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3.2 The exhaust valve of an IC engine is shown in Fig. 3.31. There is a clearance fit between the valve stem and its guide and an interference fit between the valve seat and its housing. Determine

(i) diameters of the valve stem
(ii) inner diameters of guide for valve stem
(iii) the clearances between the stem and guide
(iv) diameters of the valve seat
(v) inner diameters of housing of the valve seat
(vi) the interferences between the valve seat and its housing

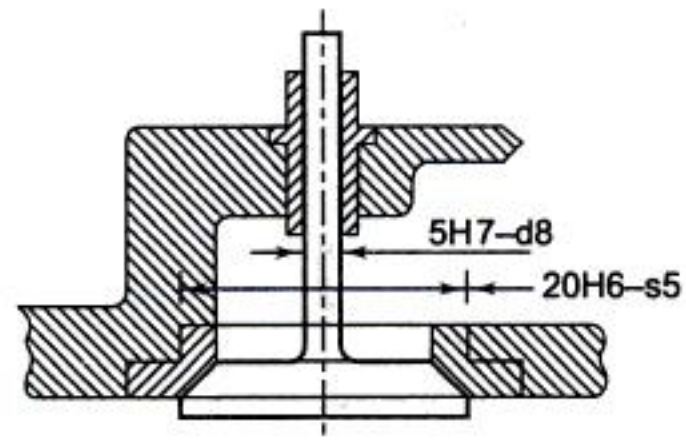

Fig. 3.31

3.3 The recommended class of fit for a hydrodynamic bearing is 40H6-e7. The maximum and minimum clearances are limited to 0.08 and 0.06 mm respectively. Specify groups for selective assembly.

The five groups are as follows:

Group	*A*	*B*	*C*	*D*	*E*
Hole (mm)	$\frac{40.016}{40.013}$	$\frac{40.013}{40.010}$	$\frac{40.010}{40.007}$	$\frac{40.007}{40.004}$	$\frac{40.004}{40.000}$
Shaft (mm)	$\frac{39.950}{39.945}$	$\frac{39.945}{39.940}$	$\frac{39.940}{39.935}$	$\frac{39.935}{39.930}$	$\frac{39.930}{39.925}$

[*(i)* $\frac{4.970}{4.952}$ *mm*

(ii) $\frac{5.012}{5.000}$ *mm*

(iii) 0.06 and 0.03 mm

(iv) $\frac{20.044}{20.035}$ *mm*

(v) $\frac{20.013}{20.000}$ *mm*

(vi) 0.044 and 0.022 mm]

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Design against Static Load

Chapter 4

4.1 MODES OF FAILURE

A static load is defined as a force, which is gradually applied to a mechanical component and which does not change its magnitude or direction with respect to time. It was discussed in Chapter 2 that engineering materials are classified into two groups—ductile and brittle materials. A ductile material is one which has a relatively large tensile strain before fracture takes place. On the other hand, a brittle material has a relatively small tensile strain before fracture. A tensile strain of 5% is considered to be the dividing line between brittle and ductile materials. Structural steels and aluminium are ductile materials, while cast iron is an example of a brittle material.

A mechanical component may fail, that is, may be unable to perform its function satisfactorily, as a result of any one of the following three modes of failure[1]:

(i) failure by elastic deflection;
(ii) failure by general yielding; and
(iii) failure by fracture.

In applications like transmission shaft supporting gears, the maximum force acting on the shaft, without affecting its performance, is limited by the permissible elastic deflection. Lateral or torsional rigidity is considered as the criterion of design in such cases. Sometimes, the elastic deflection results in unstable conditions, such as buckling of columns or vibrations. The design of the mechanical component, in all these cases, is based on the permissible lateral or torsional deflection. The stresses induced in the component are not significant and the properties of the material, such as yield strength or ultimate tensile strength, are not of primary importance. The modules of elasticity and rigidity are the important properties and the dimensions of the component are determined by the load-deflection equations.

A mechanical component made of ductile material loses its engineering usefulness due to a large amount of plastic deformation after the yield point stress is reached. Considerable portion of the component is subjected to plastic deformation, called *general yielding*. There is a basic difference between general yielding and localized yielding. The *localized yielding* in the region of stress concentration is restricted to a very small portion of the component and is not considered significant. The yield strength of a material is an important property when a component is designed against failure due to general yielding.

Components made of brittle material cease to function satisfactorily because of the sudden fracture without any plastic deformation. The

[1] FB Seely and JO Smith, *Advanced Mechanics of Materials*, John Wiley and Sons. Inc.

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failure in this case is sudden and total. In such cases, ultimate tensile strength of the material is an important property to determine the dimensions of these components.

The failure by elastic deflection is separately considered in the chapter on transmission shafting. The discussion in this chapter is restricted to the design of components on the strength basis.

4.2 FACTOR OF SAFETY

While designing a component, it is necessary to provide sufficient reserve strength in case of an accident. This is achieved by taking a suitable factor of safety (fs).

The factor of safety is defined as

$$(fs) = \frac{\text{failure stress}}{\text{allowable stress}}$$

or

$$(fs) = \frac{\text{failure load}}{\text{working load}}$$

The allowable stress is the stress value, which is used in design to determine the dimensions of the component. It is considered as a stress, which the designer expects will not be exceeded under normal operating conditions. For ductile materials, the allowable stress σ is obtained by the following relationship:

$$\sigma = \frac{S_{yt}}{(fs)} \tag{4.1}$$

For brittle materials, the relationship is,

$$\sigma = \frac{S_{ut}}{(fs)} \tag{4.2}$$

where S_{yt} and S_{ut} are the yield strength and the ultimate tensile strength of the material respectively.

There are a number of factors which are difficult to evaluate accurately in design analysis. Some of the factors are as follows:

(i) Uncertainty in the magnitude of external force acting on the component
(ii) Variations in the properties of materials like yield strength or ultimate strength
(iii) Variations in the dimensions of the component due to imperfect workmanship

In addition to these factors, the number of assumptions made in design analysis, in order to simplify the calculations, may not be exactly valid in working conditions. The factor of safety ensures against these uncertainties and unknown conditions.

The magnitude of factor of safety depends upon the following factors:

(i) Effect of Failure Sometimes, the failure of a machine element involves only a little inconvenience or loss of time, e.g., failure of the ball bearing in a gearbox. On the other hand, in some cases, there is substantial financial loss or danger to the human life, e.g., failure of the valve in a pressure vessel. The factor of safety is high in applications where failure of a machine part may result in serious accidents.

(ii) Type of Load The factor of safety is low when the external force acting on the machine element is static, i.e., a load which does not vary in magnitude or direction with respect to time. On the other hand, a higher factor of safety is selected when the machine element is subjected to impact load. This is due to the fact that impact load is suddenly applied to the machine component, usually at high velocities.

(iii) Degree of Accuracy in Force Analysis When the forces acting on the machine component are precisely determined, a low factor of safety can be selected. On the contrary, a higher factor of safety is necessary when the machine component is subjected to a force whose magnitude or direction is uncertain and unpredictable.

(iv) Material of Component When the component is made of a homogeneous ductile material like steel, yield strength is the criterion of failure. The factor of safety is usually small in such cases. On the other hand, a cast iron component has non-homogeneous structure and a higher factor of safety based on ultimate tensile strength is chosen.

(v) Reliability of Component In certain applications like continuous process equipment, power stations or defense equipment, high

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reliability of components is expected. The factor of safety increases with increasing reliability.

(vi) Cost of Component As the factor of safety increases, dimensions of the component, material requirement and cost increase. The factor of safety is low for cheap machine parts.

(vii) Testing of Machine Element A low factor of safety can be chosen when the machine component can be tested under actual conditions of service and operation. A higher factor of safety is necessary, when it is not possible to test the machine part or where there is deviation between test conditions and actual service conditions.

(viii) Service Conditions When the machine element is likely to operate in corrosive atmosphere or high temperature environment, a higher factor of safety is necessary.

(ix) Quality of Manufacture When the quality of manufacture is high, variations in dimensions of the machine component are less and a low factor of safety can be selected. Conversely, a higher factor of safety is required to compensate for poor manufacturing quality.

The selection of magnitude of the factor of safety is one of the difficult tasks faced by the designer. The guidelines for selection of quantitative values of the factor of safety are as follows:

(i) For cast iron components, *ultimate tensile strength* is considered to be the failure criterion. Failure occurs when the maximum stress in the component due to external force exceeds the ultimate tensile strength even once. Cast iron components have a non-homogeneous structure. Many times, there are residual stresses in the component. To account for these factors, a large factor of safety, usually 3 to 5, based on ultimate tensile strength, is used in the design of cast iron components.

(ii) For components made of ductile materials like steel and which are subjected to external static forces, *yield strength* is considered to be the criterion of failure. When such components are overloaded and the stress due to external force exceeds the yield strength of the material, there is a small amount of plastic deformation, which usually does not put the component out of service. Ductile components have a homogeneous structure and the residual stresses can be relieved by proper heat treatment. The stress analysis is more precise in case of static forces. Due to these reasons, the factor of safety is usually small in such cases. The recommended factor of safety is 1.5 to 2, based on the yield strength of the material.

(iii) For components made of ductile materials and those subjected to external fluctuating forces, *endurance limit* is considered to be the criterion of failure. Such components fail on account of fatigue. Fatigue failure depends upon the amplitude of fluctuating stresses and the number of stress cycles. The nature of fatigue failure is discussed in Section 5.5. The number of factors affect endurance limit, such as stress concentration, notch sensitivity, surface finish and even the size of the component. Therefore, the endurance limit of the component is reduced to account for these factors. The recommended factor of safety based on this endurance limit of component is usually 1.3 to 1.5.

(iv) The design of certain components such as cams and followers, gears, rolling contact bearings or rail and wheel is based on the calculation of contact stresses by the Hertz' theory. Failure of such components is usually in the form of small pits on the surface of the component. *Pitting* is surface fatigue failure, which occurs when contact stress exceeds the surface endurance limit. The damage due to pitting is local and does not put the component out of operation. The surface endurance limit can be improved by increasing the surface hardness. The recommended factor of safety for such components is 1.8 to 2.5 based on surface endurance limit.

(v) Certain components, such as piston rods, power screws or studs, are designed on the

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basis of *buckling consideration*. Buckling is elastic instability, which results in a sudden large lateral deflection. The critical buckling load depends upon yield strength, modulus of elasticity, end conditions and radius of gyration of the column. The recommended factor of safety is 3 to 6 based on the critical buckling load of such components.

The above-mentioned values of the factor of safety are based on past experience. They are applicable under normal circumstances. However, a higher factor of safety is chosen under the following conditions:

(i) The magnitude and nature of external forces acting on the machine component cannot be precisely estimated.
(ii) It is likely that the material of the machine component has a non-homogeneous structure.
(iii) The machine component is subjected to impact force in service.
(iv) There is possibility of residual stresses in the machine component.
(v) The machine component is working in a corrosive atmosphere.
(vi) The machine part is subjected to high temperatures during operation.
(vii) The failure of the machine part may hazard the lives of people (hoist, lifting machinery and boilers) and substantial loss to property.
(viii) It is not possible to test the machine component under actual conditions of service and there is variation in actual conditions and standard test conditions.
(ix) Higher reliability is demanded in applications like components of aircrafts.
(x) There is possibility of abnormal variation in external load on some occasions.
(xi) The quality of manufacture of the machine part is poor.
(xii) The exact mode of failure of the component is unpredictable.
(xiii) There is stress concentration in a machine component.

A higher factor of safety increases the reliability of the component. However, it increases the dimensions, the volume of material and consequently the cost of the machine component. In recent years, attempts have been made to obtain precise values of factor of the safety based on statistical considerations and reliability analysis.

4.3 STRESS–STRAIN RELATIONSHIP

When a mechanical component is subjected to an external static force, a resisting force is set up within the component. *The internal resisting force per unit area of the component is called stress*. The stresses are called *tensile* when the fibres of the component tend to elongate due to the external force. On the other hand, when the fibres tend to shorten due to the external force, the stresses are called *compressive stresses*. A tension rod subjected to an external force P is shown in Fig. 4.1. The tensile stress is given by,

$$\sigma_t = \frac{P}{A} \tag{4.3}$$

where

σ_t = tensile stress (N/mm^2)
P = external force (N)
A = cross-sectional area (mm^2)

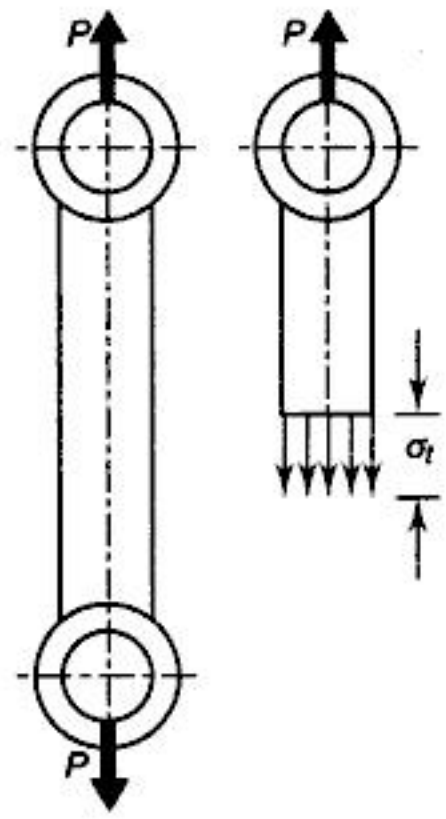

Fig. 4.1 *Tensile Stress*

Many times, the unit for stress or strength is taken as MPa.

1 MPa = 1 mega Pascal = 10^6 Pascal

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$$= (10^6)\left(\frac{\text{N}}{\text{m}^2}\right) = (10^6)\frac{\text{N}}{[10^3\,\text{mm}]^2} = 1\text{N/mm}^2$$

Therefore, two units (N/mm^2) and (MPa) are same. In this book, the unit (N/mm^2) is used. However, it can be replaced by (MPa) without any conversion factor.

The strain is deformation per unit length. It given by

$$\varepsilon = \frac{\delta}{l} \tag{4.4}$$

where,

ε = strain (mm/mm)

δ = elongation of the tension rod (mm)

l = original length of the rod (mm)

According to Hooke's law, the stress is directly proportional to the strain within elastic limit. Therefore,

$$\sigma_t \alpha \varepsilon$$

or
$$\sigma_t = E\,\varepsilon \tag{4.5}$$

where E is the constant of proportionality known as *Young's modulus* or *modulus of elasticity* (in N/mm^2 or MPa).

For carbon steels, $E = 207\,000$ N/mm^2

For grey cast iron, $E = 100\,000$ N/mm^2

Substituting Eqs (4.3) and (4.4) in Eq. (4.5),

$$\delta = \frac{Pl}{AE} \tag{4.6}$$

A component subjected to a compressive force is shown in Fig. 4.2. The compressive stress σ_c is given by,

$$\sigma_c = \frac{P}{A} \tag{4.7}$$

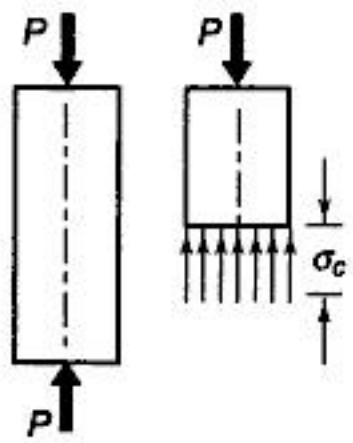

Fig. 4.2 *Compressive Stress*

The following assumptions are made in the analysis of stress and strain:

(i) The material is homogeneous.

(ii) The load is gradually applied.

(iii) The line of action of force P passes through the geometric axis of the cross-section.

(iv) The cross-section is uniform.

(v) There is no stress concentration.

4.4 SHEAR STRESS AND SHEAR STRAIN

When the external force acting on a component tends to slide the adjacent planes with respect to each other, the resulting stresses on these planes are called direct shear stresses. Two plates held together by means of a rivet are shown in Fig. 4.3 (a). The average shear stress in the rivet is given by

$$\tau = \frac{P}{A} \tag{4.8}$$

where,

τ = shear stress (N/mm^2 or MPa)

A = cross-sectional area of the rivet (mm^2)

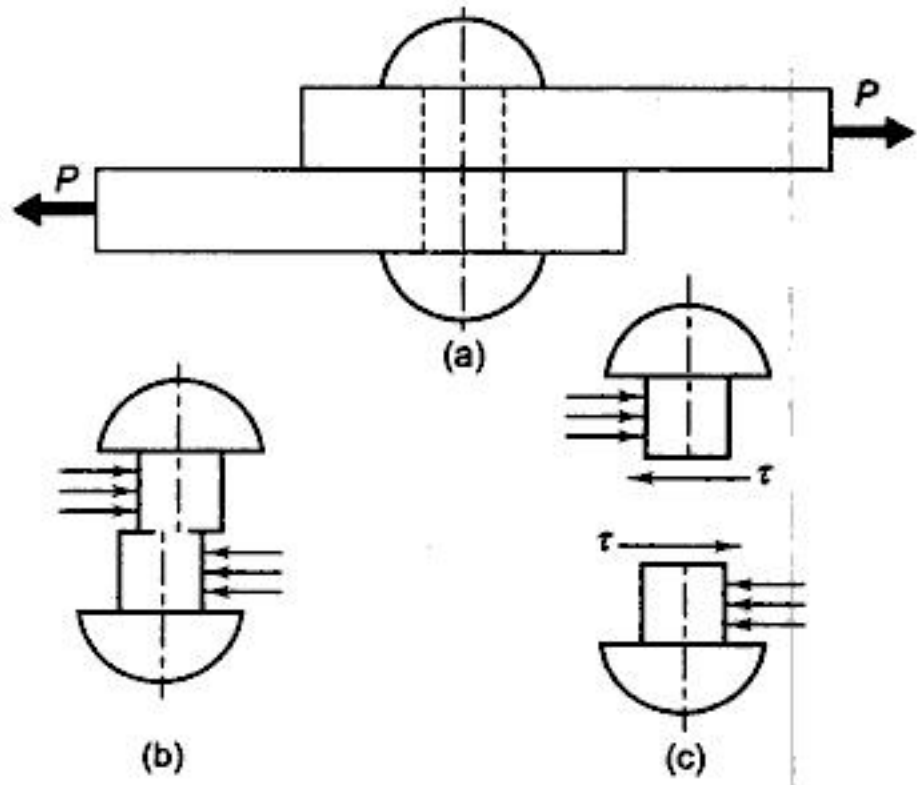

Fig. 4.3 *(a) Riveted Joint (b) Shear Deformation (c) Shear Stress*

A plane rectangular element, cut from the component and subjected to shear force, is shown in Fig. 4.4(a). Shear stresses cause a distortion in the original right angles. *The shear strain (γ) is defined as the change in the right angle of a shear element.* Within the elastic limit, the stress–strain relationship is given by

$$\tau = G\gamma \tag{4.9}$$

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where,

γ = shear strain (radians)

G is the constant of proportionality known as *shear modulus* or *modulus of rigidity* (in N/mm^2 or MPa).

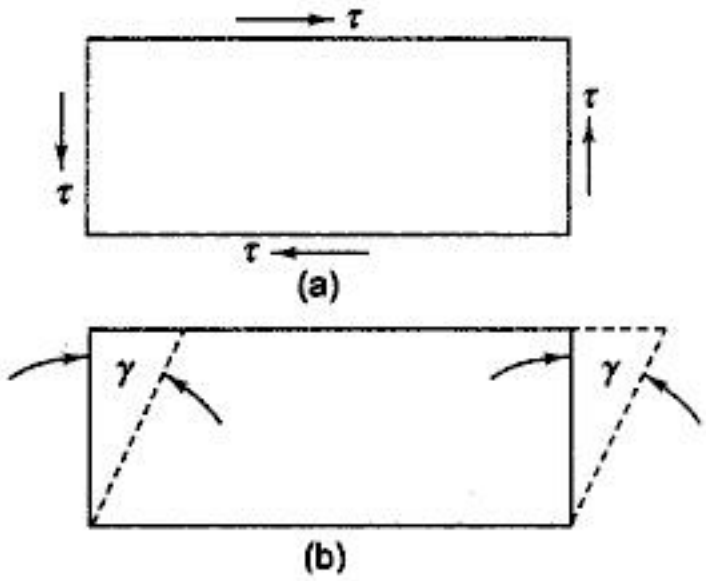

Fig. 4.4 *(a) Element Loaded in Pure Shear (b) Shear Strain*

For carbon steels, $G = 80\,000$ N/mm^2

For grey cast iron, $G = 40\,000$ N/mm^2

The relationship between the modulus of elasticity, the modulus of rigidity and the Poisson's ratio is given by,

$$E = 2G(1 + \mu) \qquad (4.10)$$

where μ is Poisson's ratio. *Poisson's ratio* is the ratio of strain in the lateral direction to that in the axial direction.

For carbon steels, $\mu = 0.29$

For grey cast iron, $\mu = 0.21$

The permissible shear stress is given by,

$$\tau = \frac{S_{sy}}{(fs)} \qquad (4.11)$$

where,

S_{sy} = yield strength in shear (N/mm^2 or MPa)

It will be proved at a later stage that the yield strength in shear is 50% of the yield strength in tension, according to the principal shear stress theory of failure.

4.5 STRESSES DUE TO BENDING MOMENT

A straight beam subjected to a bending moment M_b is shown in Fig. 4.5(a). The beam is subjected to a combination of tensile stress on one side of the neutral axis and compressive stress on the other. Such a stress distribution can be visualized by bending a thick leather belt. Cracks will appear on the outer surface, while folds will appear on the inside. Therefore, the outside fibres are in tension, while the inside fibres are in compression. The bending stress at any fibre is given by,

$$\sigma_b = \frac{M_b\, y}{I} \qquad (4.12)$$

where,

σ_b = bending stress at a distance of y from the neutral axis (N/mm^2 or MPa)

M_b = applied bending moment (N-mm)

I = moment of inertia of the cross-section about the neutral axis (mm^4)

The bending stress is maximum in a fibre, which is farthest from the neutral axis. The distribution of stresses is linear and the stress is proportional to the distance from the neutral axis.

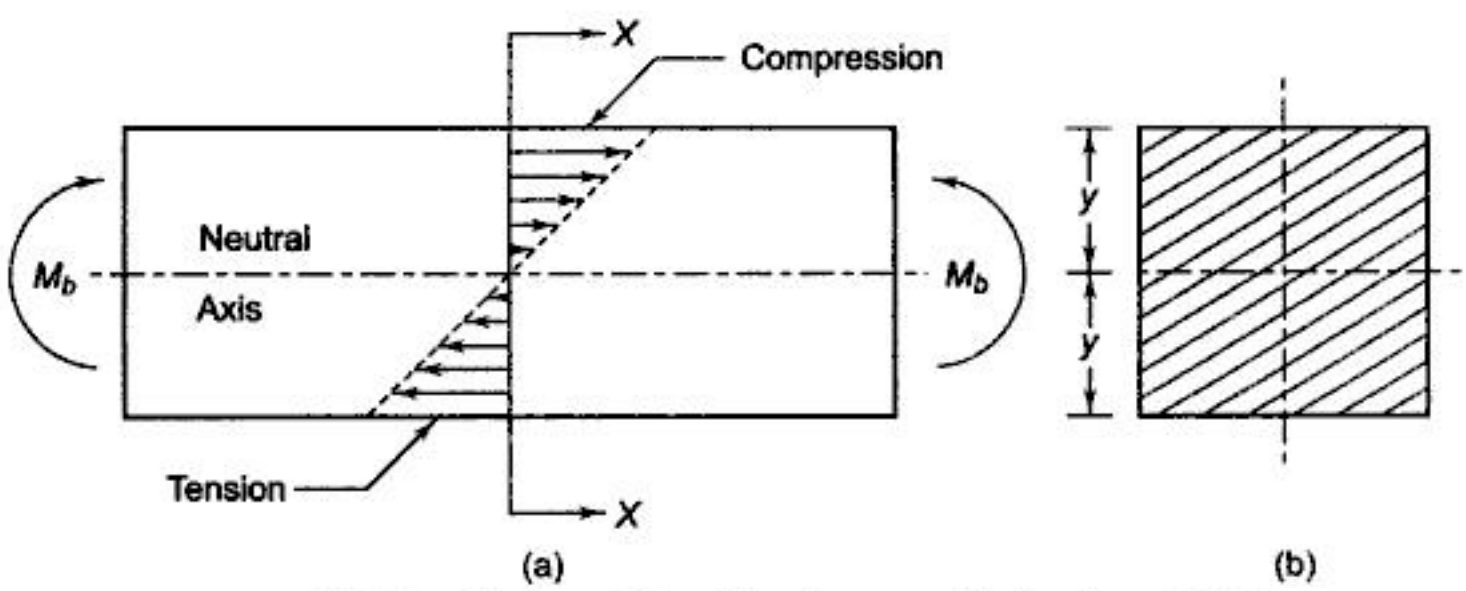

Fig. 4.5 *(a) Distribution of Bending Stresses (b) Section at XX*

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Equation (4.12) is based on the following assumptions:

(i) The beam is straight with uniform cross-section.
(ii) The forces acting on the beam lie in a plane perpendicular to the axis of the beam.
(iii) The material is homogeneous, isotropic and obeys Hooke's law.
(iv) Plane cross-sections remain plane after bending.

The moment of inertia in Eq. (4.12) is the area moment of inertia.

For a rectangular cross-section,

$$I = \frac{bd^3}{12} \tag{4.13}$$

where,

b = distance parallel to the neutral axis (mm)

d = distance perpendicular to the neutral axis (mm)

For a circular cross-section,

$$I = \frac{\pi d^4}{64} \tag{4.14}$$

where d is the diameter of the cross-section.

When the cross-section is irregular, as shown in Fig. 4.6, the moment of inertia about the centroidal axis X_g is given by,

$$I_{xg} = \int y^2 \, dA \tag{4.15}$$

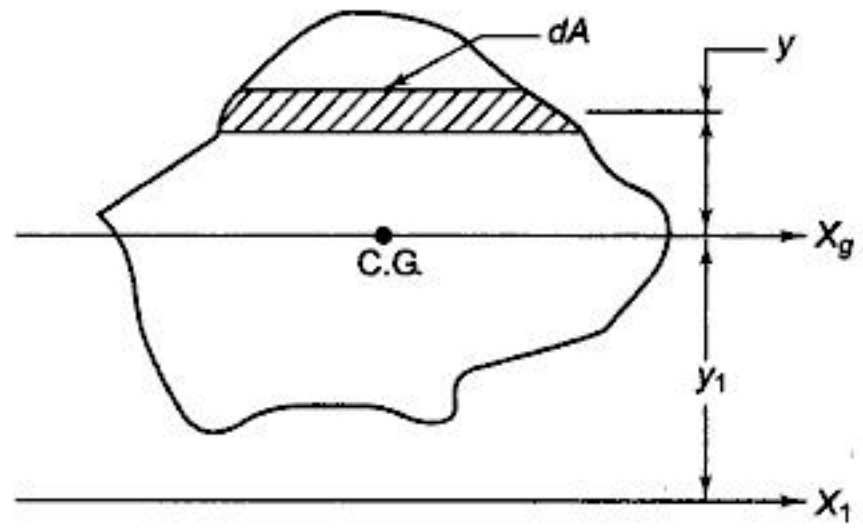

Fig. 4.6 *Parallel-axis Theorem*

The parallel-axis theorem for this area is given by the expression,

$$I_{x1} = I_{xg} + Ay_1^2 \tag{4.16}$$

where,

I_{x1} = moment of inertia of the area about X_1 axis, which is parallel to the axis X_g, and located at a distance y_1 from X_g

I_{xg} = moment of inertia of the area about its own centroidal axis

A = area of the cross-section.

In design of machine elements like transmission shaft, axle or lever, it is required to find out the maximum bending moment by constructing the bending moment diagram. There is a particular sign convention for bending moment diagram, which is illustrated in Fig. 4.7. For positive bending, the bending moment diagram is constructed on the positive side of the Y-axis. For negative bending, the diagram is on the negative side of the Y-axis. There is a simple way to remember positive bending. Imagine the crescent shaped moon—it is positive bending.

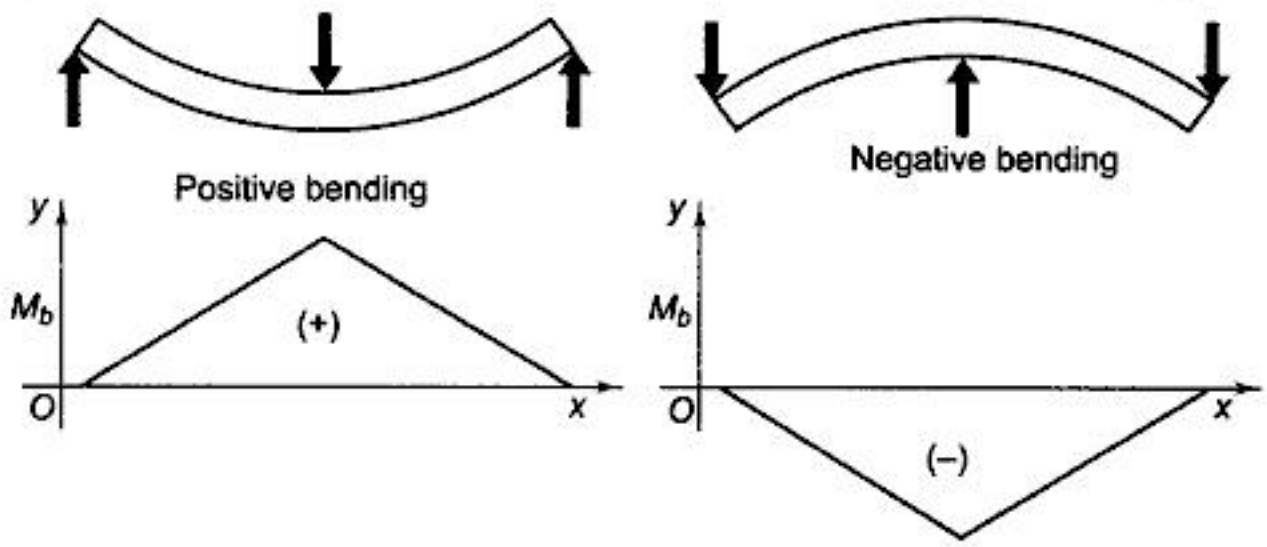

Fig. 4.7 *Sign Convention for Bending Moment*

4.6 STRESSES DUE TO TORSIONAL MOMENT

A transmission shaft, subjected to an external torque, is shown in Fig. 4.8 (a). *The internal stresses, which are induced to resist the action*

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of twist, are called torsional shear stresses. The torsional shear stress is given by

$$\tau = \frac{M_t\, r}{J} \tag{4.17}$$

where,

τ = torsional shear stress at the fibre (N/mm^2 or MPa)

M_t = applied torque (N-mm)

r = radial distance of the fibre from the axis of rotation (mm)

J = polar moment of inertia of the cross-section about the axis of rotation (mm^4)

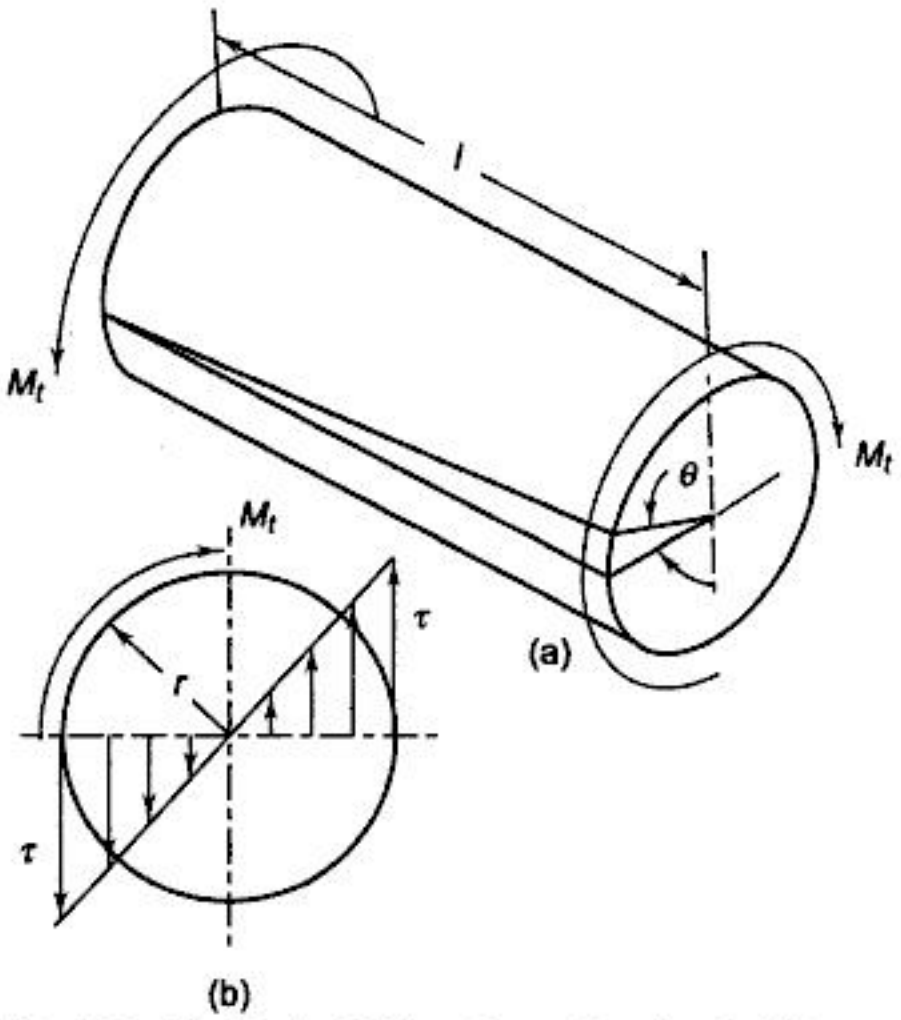

Fig. 4.8 *(a) Shaft Subjected to Torsional Moment (b) Distribution of Torsional Shear Stresses*

The distribution of torsional shear stresses is shown in Fig. 4.8 (b). The stress is maximum at the outer fibre and zero at the axis of rotation. The angle of twist is given by

$$\theta = \frac{M_t l}{JG} \tag{4.18}$$

where,

θ = angle of twist (radians)

l = length of the shaft (mm)

Equations (4.17) and (4.18) are based on the following assumptions:

(i) The shaft is straight with a circular cross-section.

(ii) A plane transverse section remains plane after twisting.

(iii) The material is homogeneous, isotropic and obeys Hooke's law.

The polar moment of inertia of a solid circular shaft of diameter d is given by

$$J = \frac{\pi d^4}{32} \tag{4.19}$$

For a hollow circular cross-section,

$$J = \frac{\pi\,(d_o^4 - d_i^4)}{32} \tag{4.20}$$

Substituting Eqs (4.19) and (4.20) in Eq. (4.18) and converting θ from radians to degrees,

$$\theta = \frac{584\ M_t l}{G d^4} \quad \text{(deg)} \tag{4.21}$$

$$\theta = \frac{584\ M_t l}{G\,(d_o^4 - d_i^4)} \quad \text{(deg)} \tag{4.22}$$

In many problems of machine design, it is required to calculate torque from the power transmitted and the speed of rotation. This relationship is given by,

$$kW = \frac{2\pi n M_t}{60 \times 10^6} \tag{4.23}$$

where,

kW = transmitted power (kW)

M_t = torque (N-mm)

n = speed of rotation (rpm)

4.7 ECCENTRIC AXIAL LOADING

In Section 4.3, it was assumed that the line of action of force passes through the centroid of the cross-section. There are certain mechanical components subjected to an external force, tensile or compressive, which does not pass through the centroid of the cross-section. A typical example of such an eccentric loading is shown in Fig. 4.9(a). According to the principle of statics, the eccentric force P can be replaced by a parallel force P passing through the centroidal axis along with a couple $(P \times e)$ as shown in Figs 4.9(b) and (c) respectively. In Fig. 4.9(b), the force P causes a uniformly distributed tensile stress of magnitude (P/A). In Fig. 4.9(c), the couple causes bending stress of magnitude (Pey/I). The resultant stresses at the cross-section are obtained by the principle of superimposition of stresses. They are given by,

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$$\sigma = \frac{P}{A} \pm \frac{Pey}{I} \tag{4.24}$$

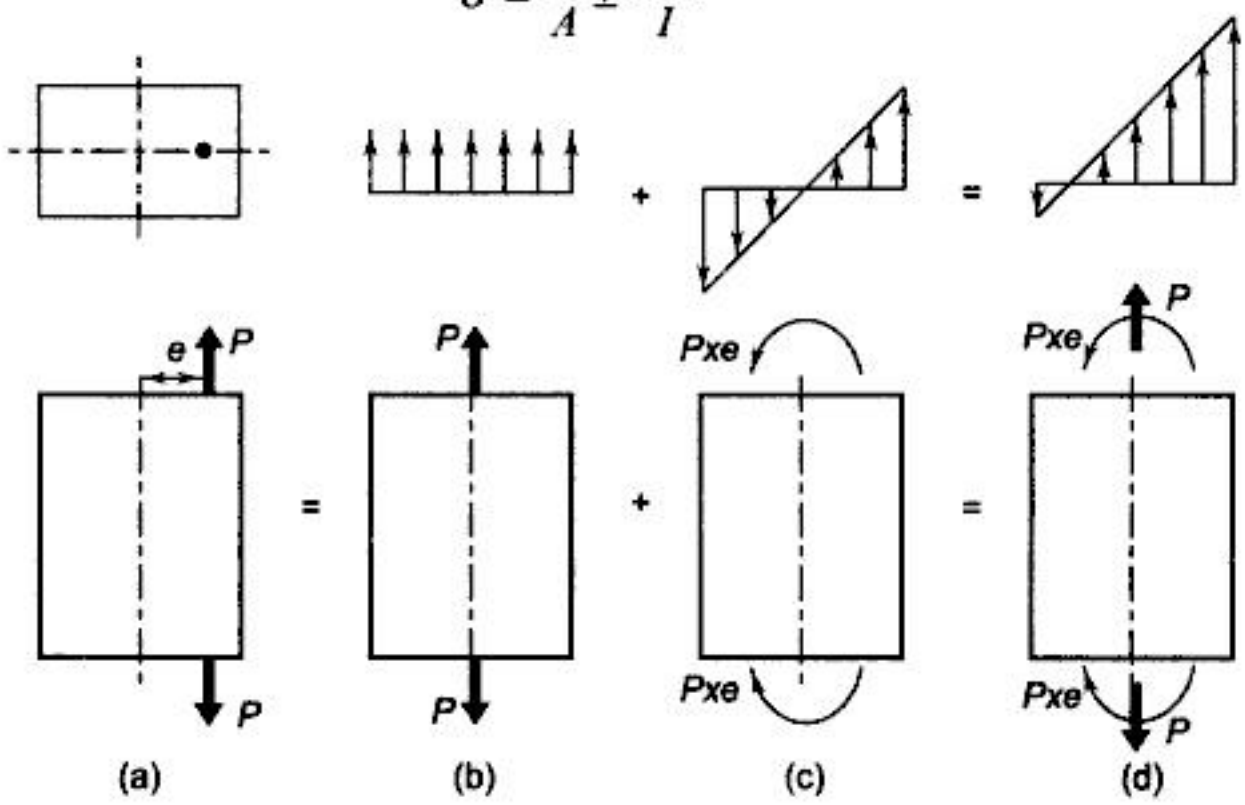

Fig. 4.9 *Eccentric Axial Load*

4.8 DESIGN OF SIMPLE MACHINE PARTS

In this chapter, design of simple machine parts is illustrated with the help of examples. The following points should be remembered in solving such problems:

(i) The dimensions of simple machine parts are determined on the basis of pure tensile stress, pure compressive stress, direct shear stress, bending stress or torsional shear stress. The analysis is simple but approximate, because a number of factors such as principal stresses, stress concentration, and reversal of stresses is neglected. Therefore, a higher factor of safety of up to 5 is taken to account for these factors.

(ii) It is incorrect to assume allowable stress as data for design. The allowable stress is to be obtained from published values of ultimate tensile strength and yield strength for a given material by Eqs (4.1) and (4.2) respectively.

(iii) It will be proved at a later stage that according to maximum shear stress theory, the yield strength in shear is 50% of the yield strength in tension. Therefore,

$$S_{sy} = 0.5\, S_{yt}$$

and the permissible shear stress (τ) is given by

$$\tau = \frac{S_{sy}}{(fs)}$$

The above value of allowable shear stress is used in determination of dimensions of the component.

The design analysis in this chapter is approximate and incomplete. The purpose of such problems is to illustrate the application of design equations to find out the geometric dimensions of the component. In practice, much more stress analysis is involved in the design of machine parts.

Example 4.1 *Two plates, subjected to a tensile force of 50 kN, are fixed together by means of three rivets as shown in Fig. 4.10 (a). The plates and rivets are made of plain carbon steel 10C4 with a tensile yield strength of 250 N/mm^2. The yield strength in shear is 50% of the tensile yield strength, and the factor of safety is 2.5. Neglecting stress concentration, determine*

(i) the diameter of the rivets; and

(ii) the thickness of the plates.

Solution

Given $P = 50 \times 10^3$ N $\quad S_{yt} = 250$ N/mm^2

$(fs) = 2.5$

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Step I *Permissible shear stress for rivets*

$$\tau = \frac{S_{sy}}{(fs)} = \frac{0.5\ S_{yt}}{(fs)} = \frac{0.5\ (250)}{(2.5)} = 50\ \text{N/mm}^2$$

Step II *Diameter of rivets*

Since there are three rivets,

$$3\left[\frac{\pi}{4}d^2\right]\tau = P \quad \text{or} \quad 3\left[\frac{\pi}{4}d^2\right]50 = 50\times10^3$$

$$\therefore\ d = 20.60 \text{ or } 22 \text{ mm} \qquad \text{(i)}$$

Step III *Permissible tensile stress for plates*

$$\sigma_t = \frac{S_{yt}}{(fs)} = \frac{250}{2.5} = 100\ \text{N/mm}^2$$

Step IV *Thickness of plates*

As shown in Fig. 4.10(b),

$$\sigma_t\,(200 - 3d)\,t = P$$

or $\quad 100(200 - 3 \times 22)\,t = 50 \times 10^3$

$$\therefore\ t = 3.73 \text{ or } 4 \text{ mm} \qquad \text{(ii)}$$

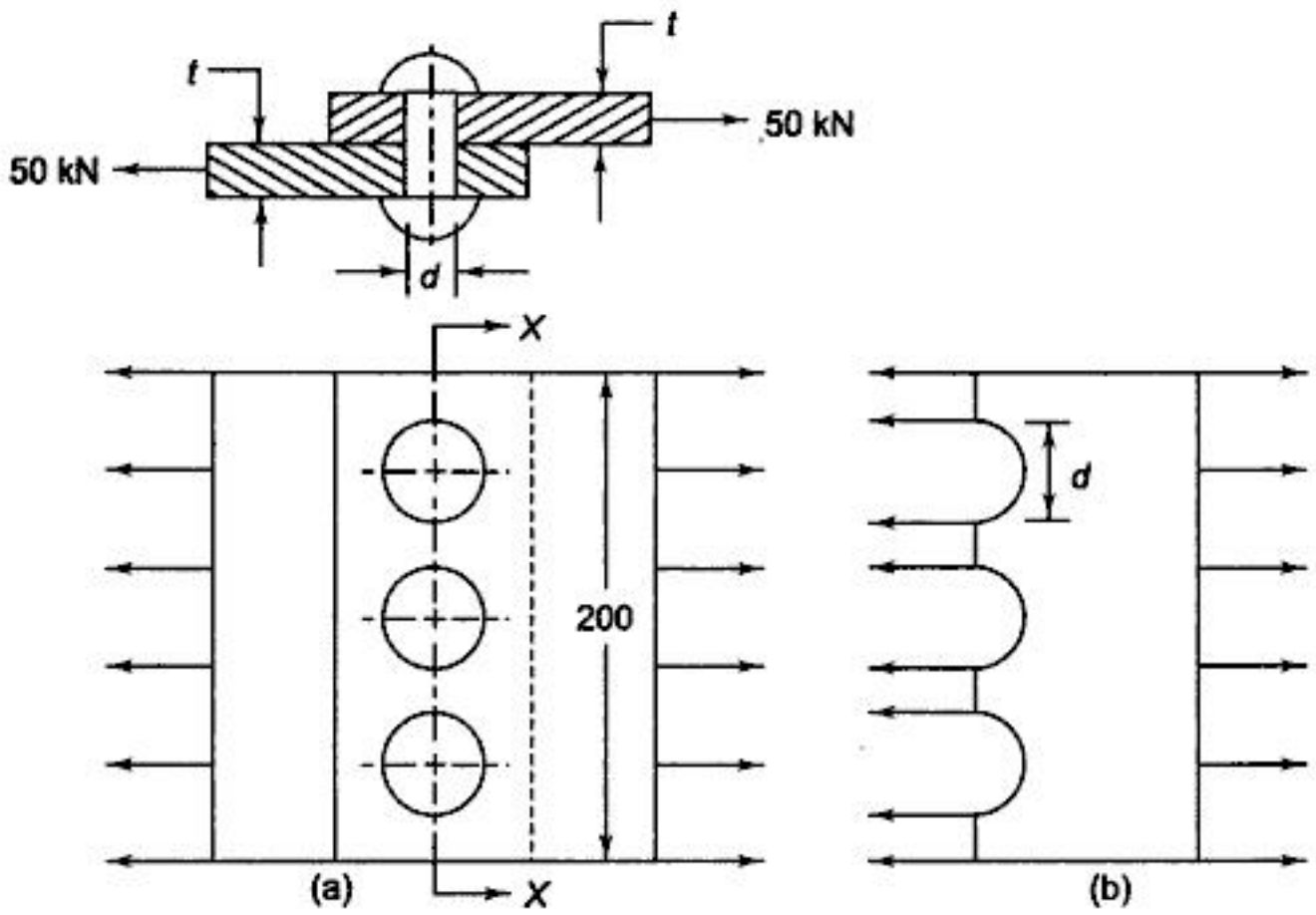

Fig. 4.10 *(a) Riveted Joint (b) Tensile Stress in Plate*

4.9 COTTER JOINT

A cotter joint is used to connect two co-axial rods, which are subjected to either axial tensile force or axial compressive force. It is also used to connect a rod on one side with some machine part like a crosshead or base plate on the other side. It is not used for connecting shafts that rotate and transmit torque. Typical applications of cotter joint are as follows:

(i) Joint between the piston rod and the crosshead of a steam engine

(ii) Joint between the slide spindle and the fork of the valve mechanism

(iii) Joint between the piston rod and the tail or pump rod

(iv) Foundation bolt

The principle of wedge action is used in a cotter joint. A cotter is a wedge-shaped piece made of a steel plate. The joint is tightened and adjusted by means of a wedge action of the cotter. The construction of a cotter joint, used to connect two rods *A* and *B* is shown in Fig. 4.11. Rod-*A* is provided with a socket end, while rod-*B* is provided with a spigot end. The socket end of rod-*A* fits over the spigot end of rod-*B*. The socket as well as the spigot is provided with a narrow rectangular slot. A cotter is tightly fitted in this slot passing through the socket and the spigot. The cotter has uniform thickness and the width dimension *b* is given a slight taper. The taper is usually 1 in 24. The taper is provided for the following two reasons:

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(i) When the cotter is inserted in the slot through the socket and the spigot and pressed by means of hammer, it becomes tight due to wedge action. This ensures tightness of the joint in operating condition and prevents loosening of the parts.

(ii) Due to its taper shape, it is easy to remove the cotter and dismantle the joint.

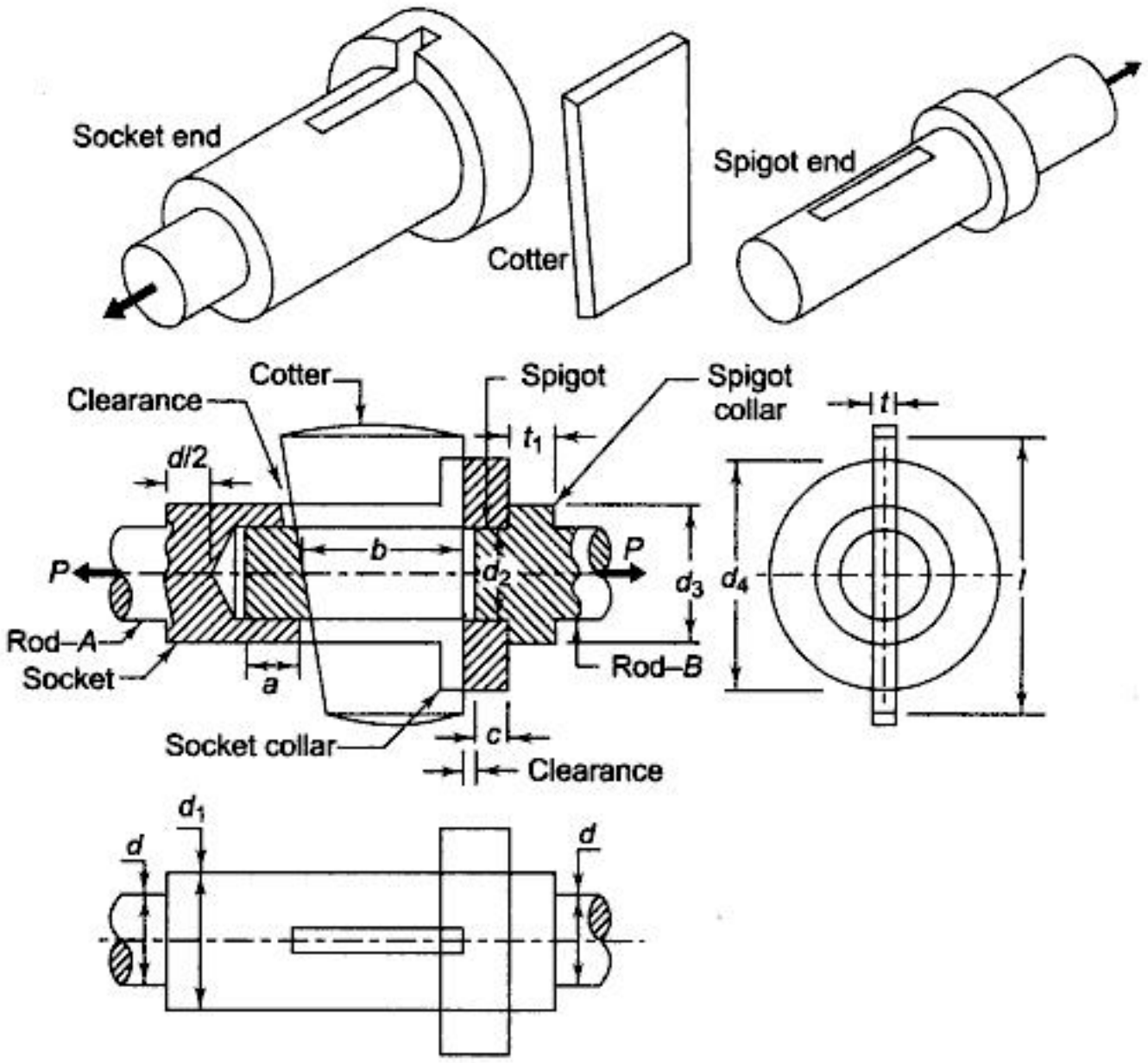

Fig. 4.11 *Cotter Joint*

The taper of the cotter as well as slots is on one side. Machining a taper on two sides of a machine part is more difficult than making a taper on one side. Also, there is no specific advantage of a taper on two sides. A clearance of 1.5 to 3 mm is provided between the slots and the cotter. When the cotter is driven in the slots, the two rods are drawn together until the spigot collar rests on the socket collar. The amount by which the two rods are drawn together is called the *draw* of the cotter. The cotter joint offers the following advantages:

(i) The assembly and dismantling of parts of the cotter joint is quick and simple. The assembly consists of inserting the spigot end into the socket end and putting the cotter into their common slot. When the cotter is hammered, the rods are drawn together and tightened. Dismantling consists of removing the cotter from the slot by means of a hammer.

(ii) The wedge action develops a very high tightening force, which prevents loosening of parts in service.

(iii) The joint is simple to design and manufacture.

Free body diagram of forces acting on three components of cotter joint, viz., socket, cotter and spigot is shown in Fig. 4.12. This diagram is constructed by using the principle that actions and reactions are equal and opposite. The forces are determined in the following way,

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(i) Consider rod–*A* with a socket end. The rod is subjected to a horizontal force *P* to the left. The sum of all horizontal forces acting on the rod *A* with socket must be equal to zero. Therefore, there should be a force *P* to the right acting on the socket. This force is shown by two parts, each equal to (*P*/2) on the socket end.

(ii) Consider rod-*B* with the spigot end. The rod is subjected to a force *P* to the right. The sum of all horizontal forces acting on rod-*B* must be equal to zero. Therefore, there should be a force *P* to the left on the spigot end.

(iii) The forces shown on the cotter are equal and opposite reactions of forces acting on the spigot end of rod-*B* and the socket end of rod-*A*.

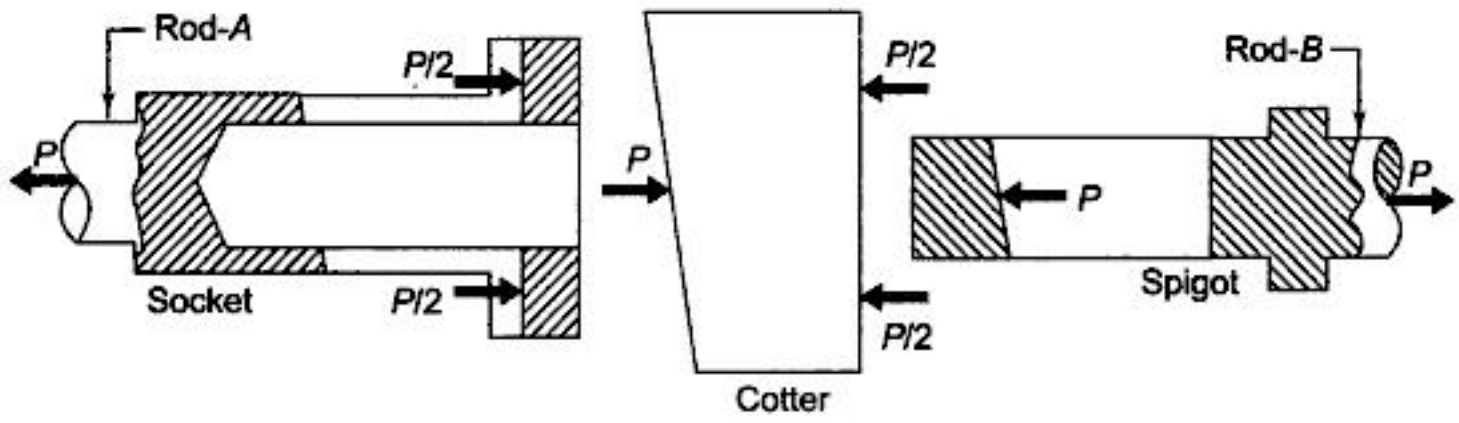

Fig. 4.12 *Free Body Diagram of Forces*

For the purpose of stress analysis, the following assumptions are made:

(i) The rods are subjected to axial tensile force.

(ii) The effect of stress concentration due to the slot is neglected.

(iii) The stresses due to initial tightening of the cotter are neglected.

In Fig. 4.11, the following notations are used

P = tensile force acting on rods (N)
d = diameter of each rod (mm)
d_1 = outside diameter of socket (mm)
d_2 = diameter of spigot or inside diameter of socket (mm)
d_3 = diameter of spigot-collar (mm)
d_4 = diameter of socket-collar (mm)
a = distance from end of slot to the end of spigot on rod-*B* (mm)
b = mean width of cotter (mm)
c = axial distance from slot to end of socket collar (mm)
t = thickness of cotter (mm)
t_1 = thickness of spigot-collar (mm)
l = length of cotter (mm)

In order to design the cotter joint and find out the above dimensions, failures in different parts and at different cross-sections are considered. Based on each type of failure, one strength equation is written. Finally, these strength equations are used to determine various dimensions of the cotter joint.

(i) Tensile Failure of Rods Each rod of diameter *d* is subjected to a tensile force *P*. The tensile stress in the rod is given by,

$$\sigma_t = \frac{P}{\left[\frac{\pi}{4} d^2\right]}$$

or
$$d = \sqrt{\frac{4P}{\pi \sigma_t}} \qquad (4.25a)$$

where σ_t is the permissible tensile stress for the rods.

(ii) Tensile Failure of Spigot Figure 4.13(a) shows the weakest cross-section at *XX* of the spigot end, which is subjected to tensile stress.

$$\text{Area of section at } XX = \left[\frac{\pi}{4} d_2^2 - d_2 t\right]$$

Therefore, tensile stress in the spigot is given by,

$$\sigma_t = \frac{P}{\left[\frac{\pi}{4} d_2^2 - d_2 t\right]}$$

or
$$P = \left[\frac{\pi}{4} d_2^2 - d_2 t\right] \sigma_t \qquad (4.25b)$$

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From the above equation, the diameter of spigot or inner diameter of socket (d_2) can be determined by assuming a suitable value of t. The thickness of the cotter is usually determined by the following empirical relationship,

$$t = 0.31d \qquad (4.25c)$$

(iii) Tensile Failure of Socket Figure 4.14(a) shows the weakest section at YY of the socket end, which is subjected to tensile stress. The area of this section is given by,

$$\text{area} = \left[\frac{\pi}{4}(d_1^2 - d_2^2) - (d_1 - d_2)t\right]$$

The tensile stress at section YY is given by,

$$\sigma_t = \frac{P}{\text{area}}$$

or $$P = \left[\frac{\pi}{4}(d_1^2 - d_2^2) - (d_1 - d_2)t\right]\sigma_t \qquad (4.25d)$$

From the above equation, the outside diameter of socket (d_1) can be determined.

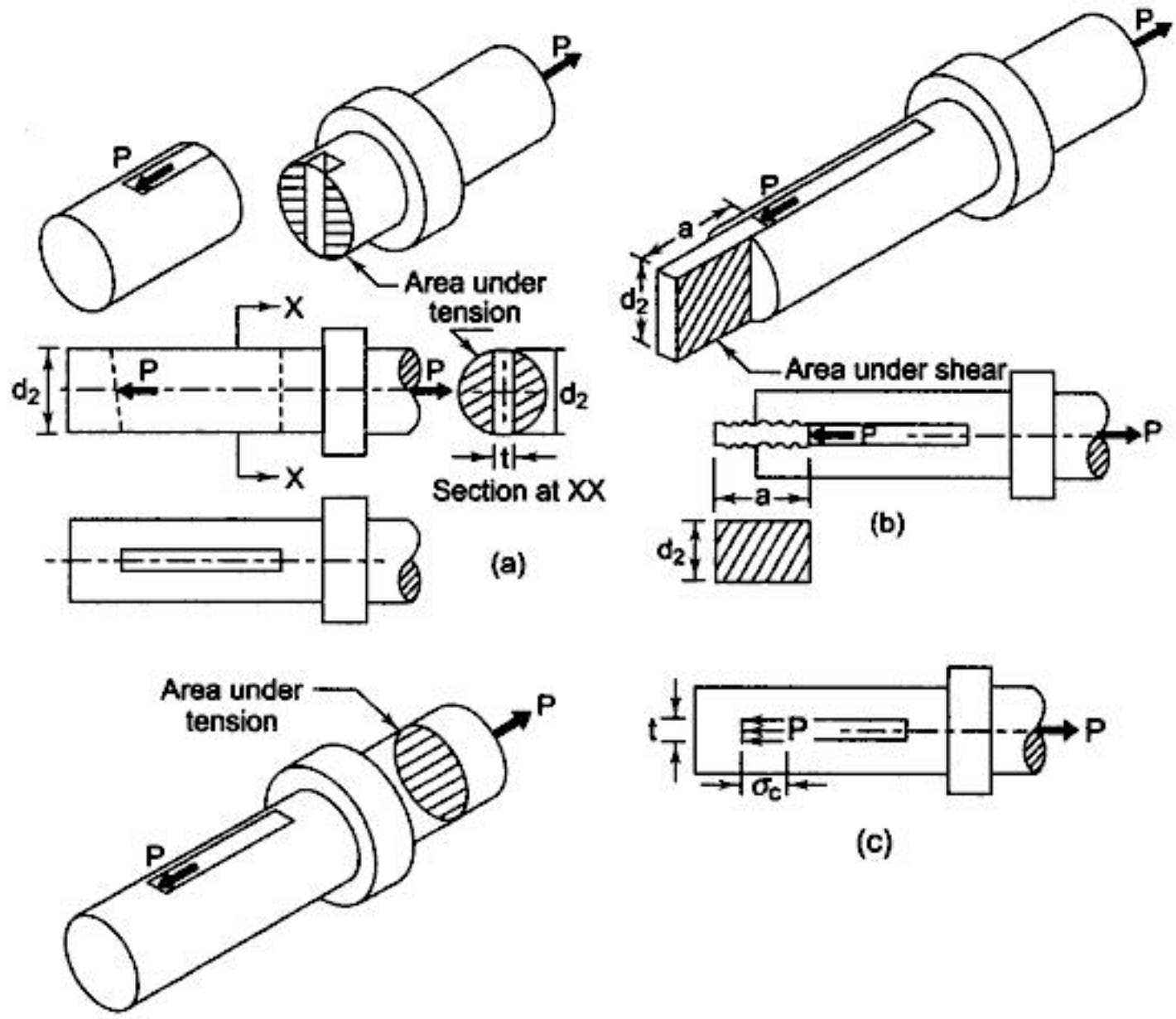

Fig. 4.13 *Stresses in Spigot End (a) Tensile Stress (b) Shear Stress (c) Compressive Stress*

(iv) Shear Failure of Cotter The cotter is subjected to double shear as illustrated in Fig. 4.15. The area of each of the two planes that resist shearing failure is (bt). Therefore, shear stress in the cotter is given by,

$$\tau = \frac{P}{2(bt)}$$

or $$P = 2\,bt\tau \qquad (4.25e)$$

where τ is permissible shear stress for the cotter. From Eq. (4.25e), the mean width of the cotter (b) can be determined.

(v) Shear Failure of Spigot End The spigot end is subjected to double shear as shown in Fig. 4.13(b). The area of each of the two planes that resist shear failure is (ad_2). Therefore, shear stress in the spigot end is given by,

$$\tau = \frac{P}{2(ad_2)}$$

or $$P = 2\,ad_2\tau \qquad (4.25f)$$

where τ is the permissible shear stress for the spigot. From Eq. (4.25f), the dimension a can be determined.

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(vi) Shear Failure of Socket End The socket end is also subjected to double shear as shown in Fig. 4.14(b). The area of each of the two planes that resist shear failure is given by,

$$\text{area} = (d_4 - d_2)\, c$$

Therefore, shear stress in the socket end is given by,

$$\tau = \frac{P}{2(d_4 - d_2)\, c}$$

or
$$P = 2(d_4 - d_2)\, c\, \tau \qquad (4.25g)$$

From the above equation, the dimension c can be determined.

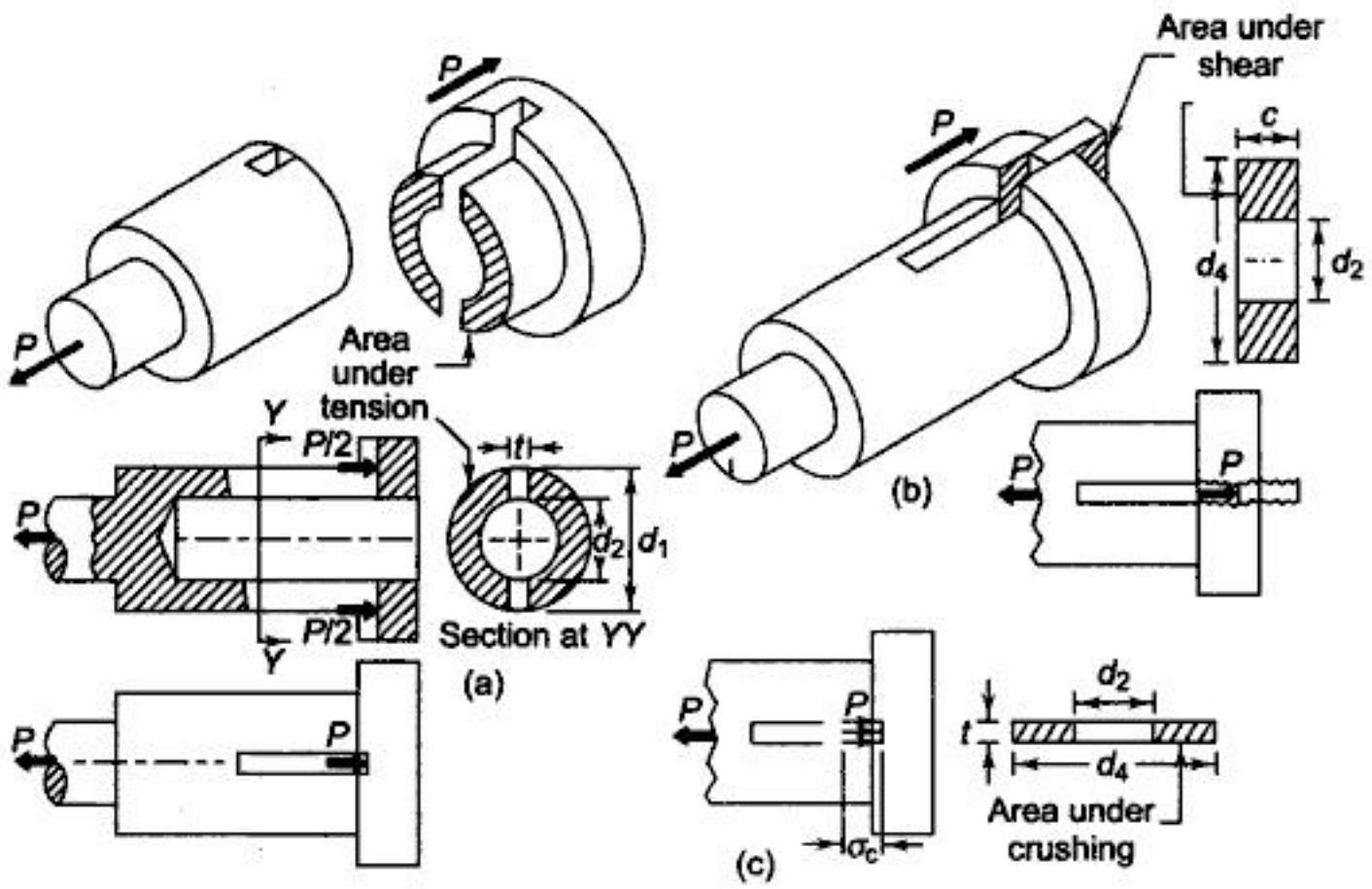

Fig. 4.14 *Stresses in Socket End (a) Tensile Stress (b) Shear stress (c) Compressive Stress*

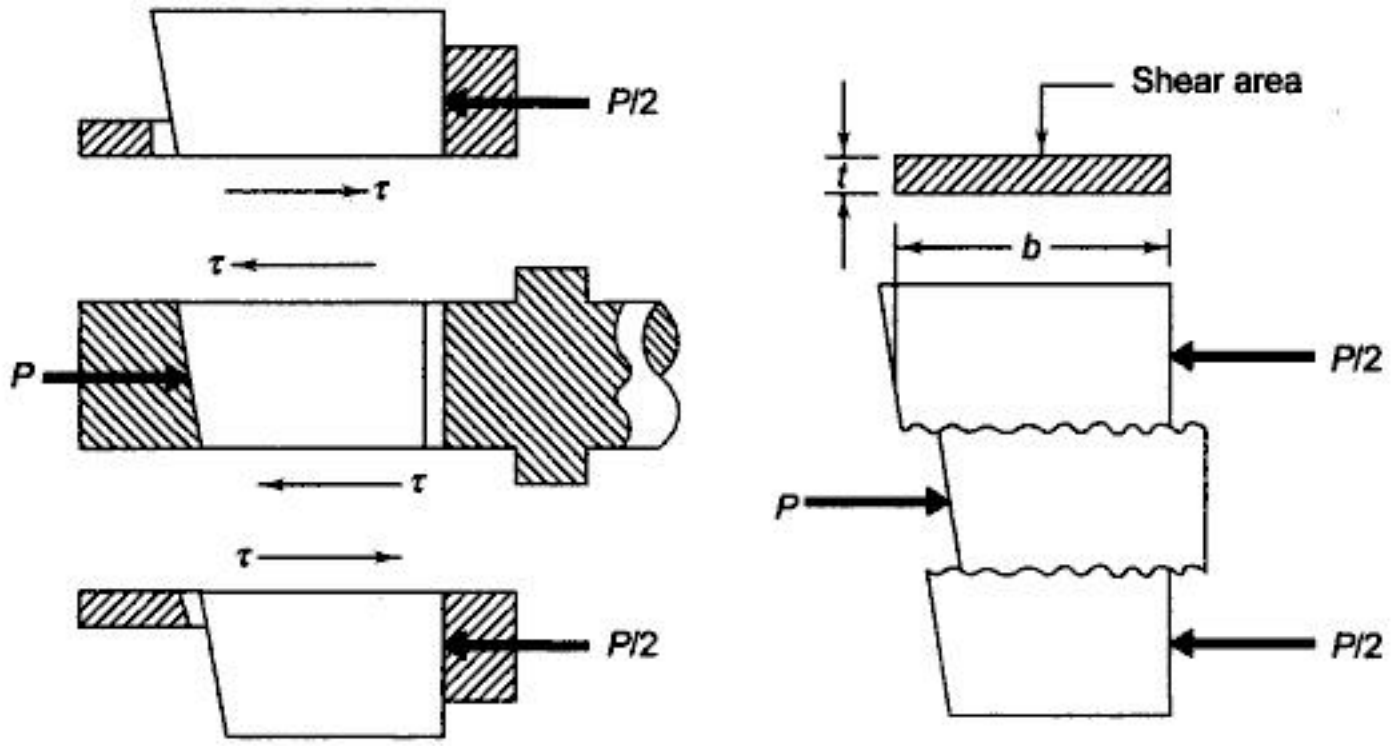

Fig. 4.15 *Shear Failure of Cotter*

(vii) Crushing Failure of Spigot End As shown in Fig. 4.13(c), the force P causes compressive stress on a narrow rectangular area of thickness t and width d_2 perpendicular to the plane of the paper. The compressive stress is given by,

$$\sigma_c = \frac{P}{td_2} \qquad (4.25h)$$

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(viii) Crushing Failure of Socket End As shown in Fig. 4.14(c), the force P causes compressive stress on a narrow rectangular area of thickness t. The other dimension of rectangle, perpendicular to the plane of paper is $(d_4 - d_2)$. Therefore, compressive stress in the socket end is given by,

$$\sigma_c = \frac{P}{(d_4 - d_2)t} \qquad (4.25i)$$

(ix) Bending Failure of Cotter When the cotter is tight in the socket and spigot, it is subjected to shear stresses. When it becomes loose, bending occurs. The forces acting on the cotter are shown in Fig. 4.16(a). The force P between the cotter and spigot end is assumed as uniformly distributed over the length d_2. The force between the socket end and cotter is assumed to be varying linearly from zero to maximum with triangular distribution. The cotter is treated as beam as shown in Fig. 4.16(b). For triangular distribution,

$$x = \frac{1}{3}y = \frac{1}{3}\left(\frac{d_4 - d_2}{2}\right) = \left(\frac{d_4 - d_2}{6}\right)$$

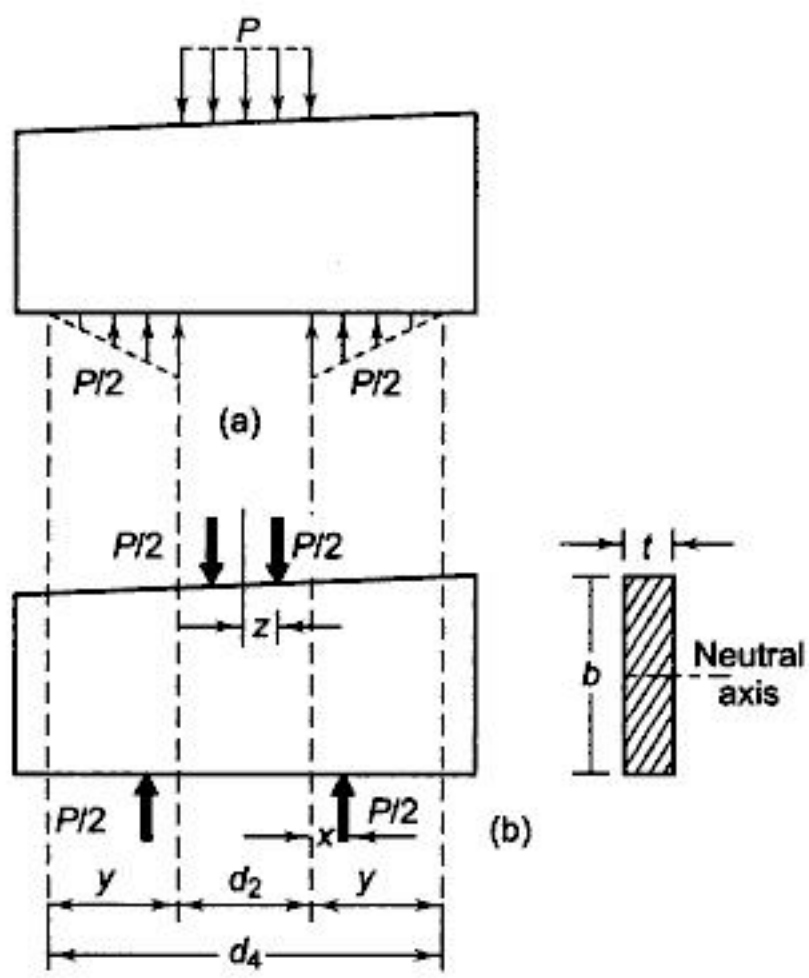

Fig. 4.16 *Cotter Treated as Beam (a) Actual Distribution of Forces (b) Simplified Diagram of Forces*

The bending moment is maximum at the centre. At the central section,

$$M_b = \frac{P}{2}\left[\frac{d_2}{2} + x\right] - \frac{P}{2}(z)$$

$$= \frac{P}{2}\left[\frac{d_2}{2} + \frac{d_4 - d_2}{6}\right] - \frac{P}{2}\left[\frac{d_2}{4}\right]$$

$$= \frac{P}{2}\left[\frac{d_2}{4} + \frac{d_4 - d_2}{6}\right]$$

Also, $I = \frac{tb^3}{12}$ $\qquad y = \frac{b}{2}$

and $\sigma_b = \frac{M_b y}{I}$

Therefore,

$$\sigma_b = \frac{\frac{P}{2}\left[\frac{d_2}{4} + \frac{d_4 - d_2}{6}\right]\frac{b}{2}}{\left(\frac{tb^3}{12}\right)} \qquad (4.25j)$$

The applications of strength equations from (4.25a) to (4.25j) in finding out the dimensions of the cotter joint are illustrated in the next example and the design project. In some cases, the dimensions of a cotter joint are calculated by using empirical relationships, without carrying out detail stress analysis. In such cases, following standard proportions can be used,

$d_1 = 1.75d$	$d_2 = 1.21d$
$d_3 = 1.5\,d$	$d_4 = 2.4\,d$
$a = c = 0.75\,d$	$b = 1.6\,d$
$t = 0.31\,d$	$t_1 = 0.45\,d$
Clearance = 1.5 to 3 mm	
Taper for cotter = 1 in 32	

4.10 DESIGN PROCEDURE FOR COTTER JOINT

The basic procedure to calculate the dimensions of the cotter joint consists of the following steps:

(i) Calculate the diameter of each rod by Eq. (4.25a),

$$d = \sqrt{\frac{4P}{\pi\sigma_t}}$$

(ii) Calculate the thickness of the cotter by the empirical relationship given in Eq. (4.25c),

$$t = 0.31\,d$$

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(iii) Calculate the diameter d_2 of the spigot on the basis of tensile stress. From Eq. (4.25b),

$$P = \left[\frac{\pi}{4} d_2^2 - d_2 t\right] \sigma_t$$

When the values of P, t and σ_t are substituted, the above expression becomes a quadratic equation.

(iv) Calculate the outside diameter d_1 of the socket on the basis of tensile stress in the socket, from Eq. (4.25d),

$$P = \left[\frac{\pi}{4}(d_1^2 - d_2^2) - (d_1 - d_2) t\right] \sigma_t$$

When values of P, d_2, t and σ_t are substituted, the above expression becomes a quadratic equation.

(v) The diameter of the spigot collar d_3 and the diameter of the socket collar d_4 are calculated by the following empirical relationships,

$$d_3 = 1.5\, d$$
$$d_4 = 2.4\, d$$

(vi) The dimensions a and c are calculated by the following empirical relationship,

$$a = c = 0.75\, d$$

(vii) Calculate the width b of the cotter by shear consideration using Eq. (4.25e) and bending consideration using Eq. (4.25j) and select the width, whichever is maximum between these two values.

$$b = \frac{P}{2\tau t} \quad \text{or} \quad b = \sqrt{\frac{3P}{t\sigma_b}\left[\frac{d_2}{4} + \frac{d_4 - d_2}{6}\right]}$$

(viii) Check the crushing and shear stresses in the spigot end by Eqs. (4.25h) and (4.25f) respectively.

$$\sigma_c = \frac{P}{t d_2}$$
$$\tau = \frac{P}{2 a d_2}$$

(ix) Check the crushing and shear stresses in the socket end by Eqs (4.25i) and (4.25g) respectively.

$$\sigma_c = \frac{P}{(d_4 - d_2) t}$$
$$\tau = \frac{P}{2 (d_4 - d_2) c}$$

(x) Calculate the thickness t_1 of the spigot collar by the following empirical relationship,

$$t_1 = 0.45\, d$$

The taper of the cotter is 1 in 32.

Example 4.2 *It is required to design a cotter joint to connect two steel rods of equal diameter. Each rod is subjected to an axial tensile force of 50 kN. Design the joint and specify its main dimensions.*

Solution

Given $P = (50 \times 10^3)$ N

Part I *Selection of material*

The rods are subjected to tensile force and strength is the criterion for the selection of the rod material. The cotter is subjected to direct shear stress and bending stresses. Therefore, strength is also the criterion of material selection for the cotter. On the basis of strength, the material of the two rods and the cotter is selected as plain carbon steel of Grade 30C8 ($S_{yt} = 400$ N/mm^2).

Part II *Selection of factor of safety*

In stress analysis of the cotter joint, the following factors are neglected:

(i) initial stresses due to tightening of the cotter; and

(ii) stress concentration due to slot in the socket and the spigot ends.

To account for these factors, a higher factor of safety is used in the present design. The factor of safety for the rods, spigot end and socket end is assumed as 6, while for the cotter, it is taken as 4. There are two reasons for assuming a lower factor of safety for the cotter. They are as follows:

(i) There is no stress concentration in the cotter.

(ii) The cost of the cotter is small compared with the socket end or spigot end. If at all, a failure is going to occur, it should occur in the cotter rather than in the spigot or socket end. This is ensured by assuming a higher factor of safety for the spigot and socket ends, compared with the cotter.

It is assumed that the yield strength in compression is twice the yield strength in tension.

Part III *Calculation of permissible stresses*

The permissible stresses for rods, spigot end and socket end are as follows:

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$$\sigma_t = \frac{S_{yt}}{(fs)} = \frac{400}{6} = 66.67 \text{ N/mm}^2$$

$$\sigma_c = \frac{S_{yc}}{(fs)} = \frac{2S_{yt}}{(fs)} = \frac{2(400)}{6} = 133.33 \text{ N/mm}^2$$

$$\tau = \frac{S_{sy}}{(fs)} = \frac{0.5S_{yt}}{(fs)} = \frac{0.5\,(400)}{6}$$

$$= 33.33 \text{ N/mm}^2$$

Permissible stresses for the cotter are as follows:

$$\sigma_t = \frac{S_{yt}}{(fs)} = \frac{400}{4} = 100 \text{ N/mm}^2$$

$$\tau = \frac{S_{sy}}{(fs)} = \frac{0.5S_{yt}}{(fs)} = \frac{0.5(400)}{4} = 50 \text{ N/mm}^2$$

Part IV *Calculation of dimensions*

The dimensions of the cotter joint are determined by the procedure outlined in Section 4.10.

Step I *Diameter of rods*

$$d = \sqrt{\frac{4P}{\pi\sigma_t}} = \sqrt{\frac{4(50 \times 10^3)}{\pi(66.67)}} = 30.90 \text{ or } 32 \text{ mm}$$

Step II *Thickness of cotter*

$$t = 0.31\,d = 0.31(32) = 9.92 \text{ or } 10 \text{ mm}$$

Step III *Diameter (d_2) of spigot*

$$P = \left[\frac{\pi}{4}d_2^2 - d_2 t\right]\sigma_t$$

$$50 \times 10^3 = \left[\frac{\pi}{4}d_2^2 - d_2(10)\right](66.67)$$

or $\quad d_2^2 - 12.73d_2 - 954.88 = 0$

Solving the above quadratic equation,

$$d_2 = \frac{12.73 \pm \sqrt{12.73^2 - 4(-954.88)}}{2}$$

$\therefore \quad d_2 = 37.91 \text{ or } 40 \text{ mm}$

Step IV *Outer diameter (d_1) of socket*

$$P = \left[\frac{\pi}{4}(d_1^2 - d_2^2) - (d_1 - d_2)t\right]\sigma_t$$

$$50 \times 10^3 = \left[\frac{\pi}{4}(d_1^2 - 40^2) - (d_1 - 40)\,(10)\right](66.67)$$

or $\quad d_1^2 - 12.73d_1 - 2045.59 = 0$

Solving the above quadratic equation,

$$d_1 = \frac{12.73 \pm \sqrt{12.73^2 - 4(-2045.59)}}{2}$$

$\therefore \quad d_1 = 52.04 \text{ or } 55 \text{ mm}$

Step V *Diameters of spigot collar (d_3) and socket collar (d_4)*

$$d_3 = 1.5d = 1.5(32) = 48 \text{ mm}$$

$$d_4 = 2.4d = 2.4(32) = 76.8 \text{ or } 80 \text{ mm}$$

Step VI *Dimensions a and c*

$$a = c = 0.75d = 0.75(32) = 24 \text{ mm}$$

Step VII *Width of cotter*

$$b = \frac{P}{2\tau t} = \frac{50 \times 10^3}{2(50)(10)} = 50 \text{ mm} \qquad \text{(a)}$$

or

$$b = \sqrt{\frac{3P}{t\sigma_b}\left[\frac{d_2}{4} + \frac{d_4 - d_2}{6}\right]}$$

$$= \sqrt{\frac{3\,(50 \times 10^3)}{(10)\,(100)}\left[\frac{40}{4} + \frac{80 - 40}{6}\right]}$$

$$= 50 \text{ mm} \qquad \text{(b)}$$

From (a) and (b),

$$b = 50 \text{ mm}$$

Step VIII *Check for crushing and shear stresses in spigot end*

$$\sigma_c = \frac{P}{td_2} = \frac{50 \times 10^3}{(10)(40)} = 125 \text{ N/mm}^2$$

$$\tau = \frac{P}{2\,ad_2} = \frac{50 \times 10^3}{2(24)(40)} = 26.04 \text{ N/mm}^2$$

$\therefore \quad \sigma_c < 133.33 \text{ N/mm}^2$ and $\tau < 33.33 \text{ N/mm}^2$

Step IX *Check for crushing and shear stresses in socket end*

$$\sigma_c = \frac{P}{(d_4 - d_2)\,t}$$

$$= \frac{50 \times 10^3}{(80 - 40)(10)} = 125 \text{ N/mm}^2$$

$$\tau = \frac{P}{2\,(d_4 - d_2)\,c}$$

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$$= \frac{50 \times 10^3}{2(80-40)(24)} = 26.04 \text{ N/mm}^2$$

$\therefore$ $\sigma_c < 133.33$ N/mm^2 and $\tau < 33.33$ N/mm^2

The stresses induced in the spigot and the socket ends are within limits.

Step X *Thickness of spigot collar*

$t_1 = 0.45d = 0.45(32) = 14.4$ or 15 mm

The taper for the cotter is 1 in 32.

Part V *Dimensioned sketch of cotter joint*

The dimensions of various components of the cotter joint are shown in Fig. 4.17.

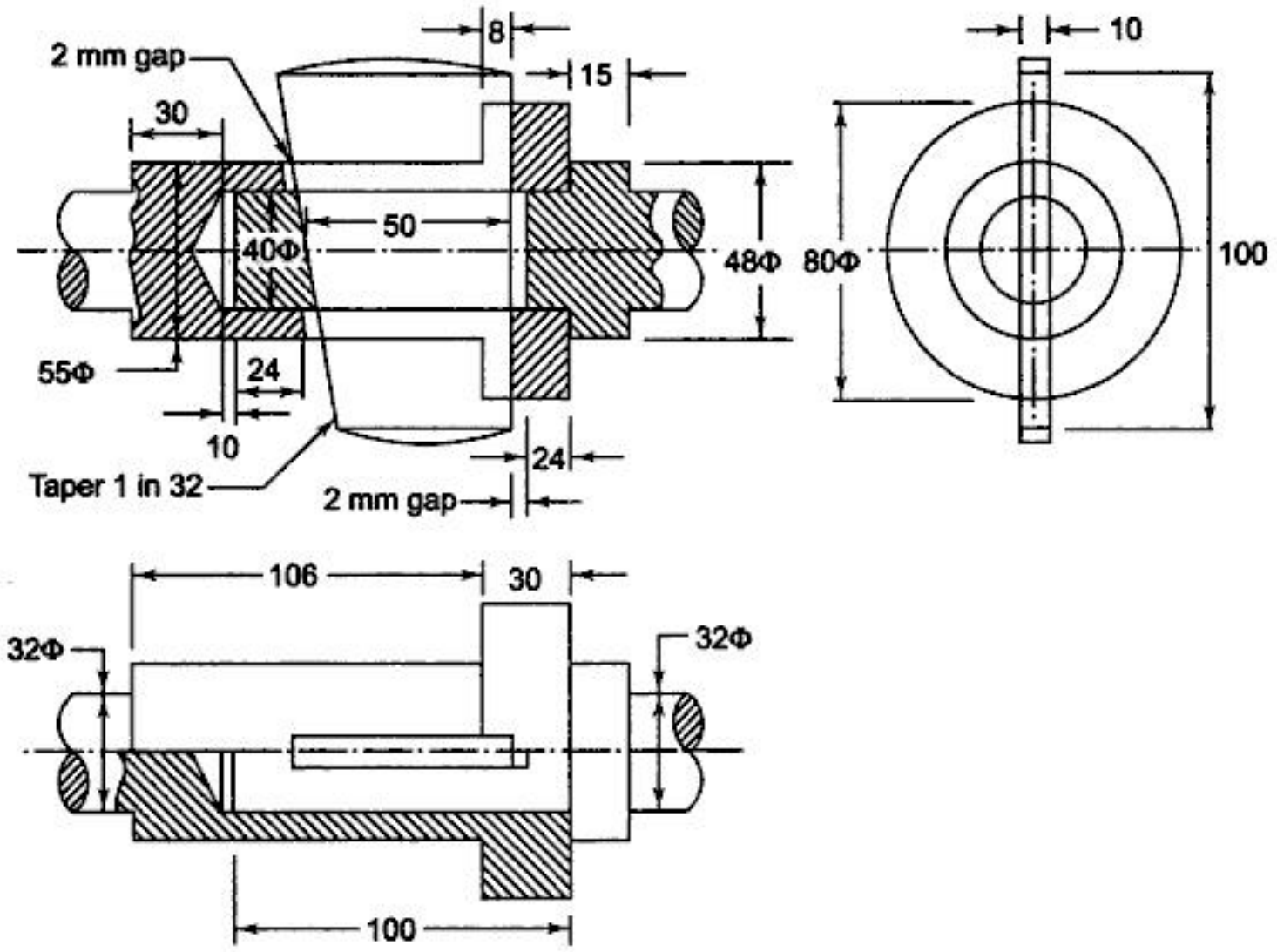

Fig. 4.17 *Dimensions of Cotter Joint*

Example 4.3 *Two rods are connected by means of a cotter joint. The inside diameter of the socket and outside diameter of the socket collar are 50 and 100 mm respectively. The rods are subjected to a tensile force of 50 kN. The cotter is made of steel 30C8 (S_{yt} = 400 N/mm^2) and the factor of safety is 4. The width of the cotter is five times of thickness. Calculate:*

(i) width and thickness of the cotter on the basis of shear failure; and

(ii) width and thickness of the cotter on the basis of bending failure.

Solution

Given $S_{yt} = 400$ N/mm^2 $(fs) = 4$

$P = (50 \times 10^3)$ N $d_4 = 100$ mm $d_2 = 50$ mm

Step I *Permissible stresses for cotter*

$$\sigma_t = \frac{S_{yt}}{(fs)} = \frac{400}{4} = 100 \text{ N/mm}^2$$

$$\tau = \frac{S_{sy}}{(fs)} = \frac{0.5S_{yt}}{(fs)} = \frac{0.5(400)}{4} = 50 \text{ N/mm}^2$$

Step II *Width and thickness on the basis of shear failure*

$$b = 5t$$

From Eq. (4.25e),

$P = 2bt\tau$ or $50 \times 10^3 = 2\,(5t)\,t\,(50)$

$\therefore$ $t = 10$ mm and $b = 5t = 50$ mm (i)

Step III *Width and thickness on the basis of bending failure*

$d_4 = 100$ mm $d_2 = 50$ mm

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From Eq. (4.25j),

$$\sigma_b = \frac{\frac{P}{2}\left[\frac{d_2}{4} + \frac{d_4 - d_2}{6}\right]\frac{b}{2}}{\left(\frac{tb^3}{12}\right)}$$

$$100 = \frac{\frac{50 \times 10^3}{2}\left[\frac{50}{4} + \frac{(100-50)}{6}\right]\frac{(5t)}{2}}{\left[\frac{t(5t)^3}{12}\right]}$$

$\therefore$ $t = 10.77$ or 12 mm and $b = 5t = 60$ mm (ii)

Example 4.4 *Two rods, made of plain carbon steel 40C8 (S_{yt} = 380 N/mm^2), are to be connected by means of a cotter joint. The diameter of each rod is 50 mm and the cotter is made from a steel plate of 15 mm thickness. Calculate the dimensions of the socket end making the following assumptions:*

(i) the yield strength in compression is twice of the tensile yield strength; and

(ii) the yield strength in shear is 50% of the tensile yield strength.

The factor of safety is 6.

Solution

Given $S_{yt} = 380$ N/mm^2 $(fs) = 6$ $t = 15$ mm
$d = 50$ mm

Step I *Permissible stresses*

$$\sigma_c = \frac{S_{yc}}{(fs)} = \frac{2S_{yt}}{(fs)} = \frac{2(380)}{6} = 126.67 \text{ N/mm}^2$$

$$\tau = \frac{S_{sy}}{(fs)} = \frac{0.5S_{yt}}{(fs)} = \frac{0.5(380)}{6} = 31.67 \text{ N/mm}^2$$

$$\sigma_t = \frac{S_{yt}}{(fs)} = \frac{380}{6} = 63.33 \text{ N/mm}^2$$

Step II *Load acting on rods*

$$P = \frac{\pi}{4} d^2 \sigma_t \quad \text{or} \quad P = \frac{\pi}{4}(50)^2 (63.33)$$
$$= 124\,348.16 \text{ N}$$

Step III *Inside diameter of socket (d_2)*

From Eq. (4.25b),

$$P = \left[\frac{\pi}{4} d_2^2 - d_2 t\right]\sigma_t$$

$$124\,348.16 = \left[\frac{\pi}{4} d_2^2 - d_2(15)\right](63.33)$$

or $d_2^2 - 19.1 d_2 - 2500 = 0$

Solving the above quadratic equation,

$$d_2 = \frac{19.1 \pm \sqrt{19.1^2 - 4(-2500)}}{2}$$

$\therefore$ $d_2 = 60.45$ or 65 mm (i)

Step IV *Outside diameter of socket (d_1)*

From Eq. (4.25d),

$$P = \left[\frac{\pi}{4}(d_1^2 - d_2^2) - (d_1 - d_2)t\right]\sigma_t$$

$$124\,348.16 = \left[\frac{\pi}{4}(d_1^2 - 65^2) - (d_1 - 65)(15)\right](63.33)$$

or $d_1^2 - 19.1 d_1 - 5483.59 = 0$

Solving the above quadratic equation,

$$d_1 = \frac{19.1 \pm \sqrt{19.1^2 - 4(-5483.59)}}{2}$$

$\therefore$ $d_1 = 84.21$ or 85 mm (ii)

Step V *Diameter of socket collar (d_4)*

From Eq. (4.25i),

$$\sigma_c = \frac{P}{(d_4 - d_2)t}$$

or $$126.67 = \frac{124\,348.16}{(d_4 - 65)(15)}$$

$\therefore$ $d_4 = 130.44$ or 135 mm (iii)

Step VI *Dimensions a and c*

From Eq. (4.25f),

$$a = \frac{P}{2 d_2 \tau} = \frac{124\,348.16}{2(65)(31.67)} = 30.20 \text{ or } 35 \text{ mm} \quad \text{(iv)}$$

From Eq. (4.25g),

$$c = \frac{P}{2(d_4 - d_2)\tau} = \frac{124\,348.16}{2(135-65)(31.67)}$$
$$= 28.04 \text{ or } 30 \text{ mm} \quad \text{(v)}$$

4.11 KNUCKLE JOINT

Knuckle joint is used to connect two rods whose axes either coincide or intersect and lie in one plane. The knuckle joint is used to transmit axial tensile force. The construction of this joint permits limited angular movement between rods, about the axis of the pin. This type of joint is popular in

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machines and structures. Typical applications of knuckle joints are as follows:

(i) Joints between the tie bars in roof trusses.
(ii) Joints between the links of a suspension bridge.
(iii) Joints in valve mechanism of a reciprocating engine.
(iv) Fulcrum for the levers.
(v) Joints between the links of a bicycle chain.

A knuckle joint is unsuitable to connect two rotating shafts, which transmit torque. The construction of a knuckle joint, used to connect two rods *A* and *B* subjected to tensile force *P*, is shown in Fig. 4.18. An eye is formed at the end of rod-*B*, while a fork is formed at the end of rod-*A*. The eye fits inside the fork and a pin passes through both the fork and the eye. This pin is secured in its place by means of a split-pin. Due to this type of construction, a knuckle joint is sometimes called a *forked-pin joint*. In rare applications, a knuckle joint is used to connect three rods—two with forks and a third with the eye.

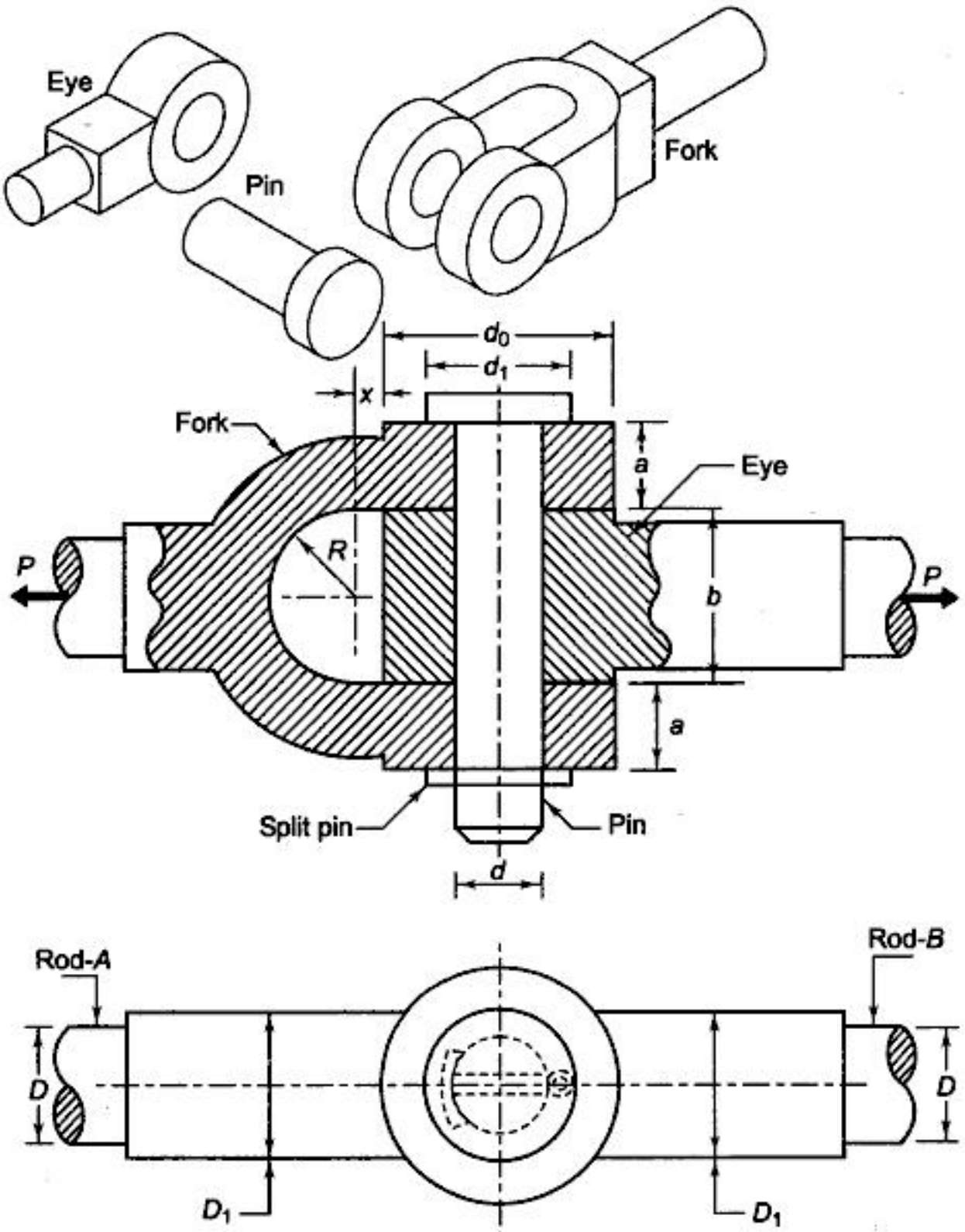

Fig. 4.18 *Knuckle Joint*

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The knuckle joint offers the following advantages:

(i) The joint is simple to design and manufacture.

(ii) There are a few parts in the knuckle joint, which reduces cost and improves reliability.

(iii) The assembly or dismantling of the parts of a knuckle joint is quick and simple. The assembly consists of inserting the eye of one rod inside the fork of the other rod and putting the pin in their common hole and finally putting the split-pin to hold the pin. Dismantling consists of removing the split-pin and taking the pin out of the eye and the fork.

In Fig. 4.18, the following notations are used.

D = diameter of each rod (mm)
D_1 = enlarged diameter of each rod (mm)
d = diameter of knuckle pin (mm)
d_0 = outside diameter of eye or fork (mm)
a = thickness of each eye of fork (mm)
b = thickness of eye end of rod-B (mm)
d_1 = diameter of pin head (mm)
x = distance of the centre of fork radius R from the eye (mm)

For the purpose of stress analysis of a knuckle joint, the following assumptions are made:

(i) The rods are subjected to axial tensile force.

(ii) The effect of stress concentration due to holes is neglected.

(iii) The force is uniformly distributed in various parts.

Free body diagram of forces acting on three components of the knuckle joint, viz., fork, pin and eye is shown in Fig. 4.19. This diagram is constructed by using the principle that actions and reactions are equal and opposite. The forces are determined in the following way,

(i) Consider rod-A with the fork end. The rod is subjected to a horizontal force P to the left. The sum of all horizontal forces acting on rod-A must be equal to zero. Therefore, there should be a force P to the right acting on the fork end. The force P is divided into two parts, each equal to $(P/2)$ on the fork end.

(ii) Consider rod-B with the eye end. The rod is subjected to a horizontal force P to the right side. The sum of all horizontal forces acting on rod-B must be equal to zero. Therefore, there should be a force P to the left acting on the eye end.

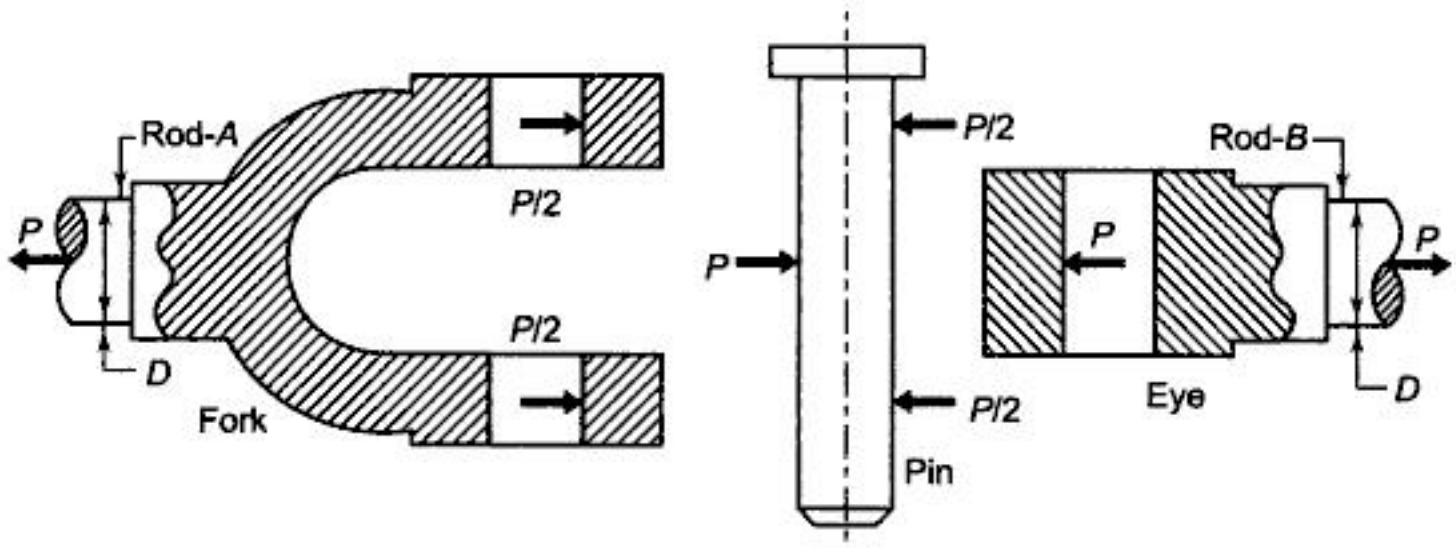

Fig. 4.19 *Free Body Diagram of Forces*

(iii) The forces shown on the pin are equal and opposite reactions of forces acting on the fork end of rod-A and the eye end of rod-B.

In order to find out various dimensions of the parts of a knuckle joint, failures in different parts and at different cross-sections are considered. For each type of failure, one strength equation is written. Finally, these strength equations are used to find out various dimensions of the knuckle joint.

(i) Tensile Failure of Rods Each rod is subjected to a tensile force P. The tensile stress in the rod is given by,

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$$\sigma_t = \frac{P}{\left(\frac{\pi}{4} D^2\right)} \quad \text{or} \quad D = \sqrt{\frac{4P}{\pi \sigma_t}} \qquad (4.26a)$$

where σ_t is the permissible tensile stress for the rods. The enlarged diameter D_1 of the rod near the joint is determined by the following empirical relationship,

$$D_1 = 1.1\, D \qquad (4.26b)$$

(ii) Shear Failure of Pin The pin is subjected to double shear as shown in Fig. 4.20. The area of each of the two planes that resist shear failure is $\left(\frac{\pi}{4} d^2\right)$. Therefore, shear stress in the pin is given by,

$$\tau = \frac{P}{2\left(\frac{\pi}{4} d^2\right)} \quad \text{or} \quad d = \sqrt{\frac{2P}{\pi \tau}} \qquad (4.26c)$$

where τ is the permissible shear stress for the pin. The standard proportion for the diameter of the pin is as follows,

$$d = D \qquad (4.26d)$$

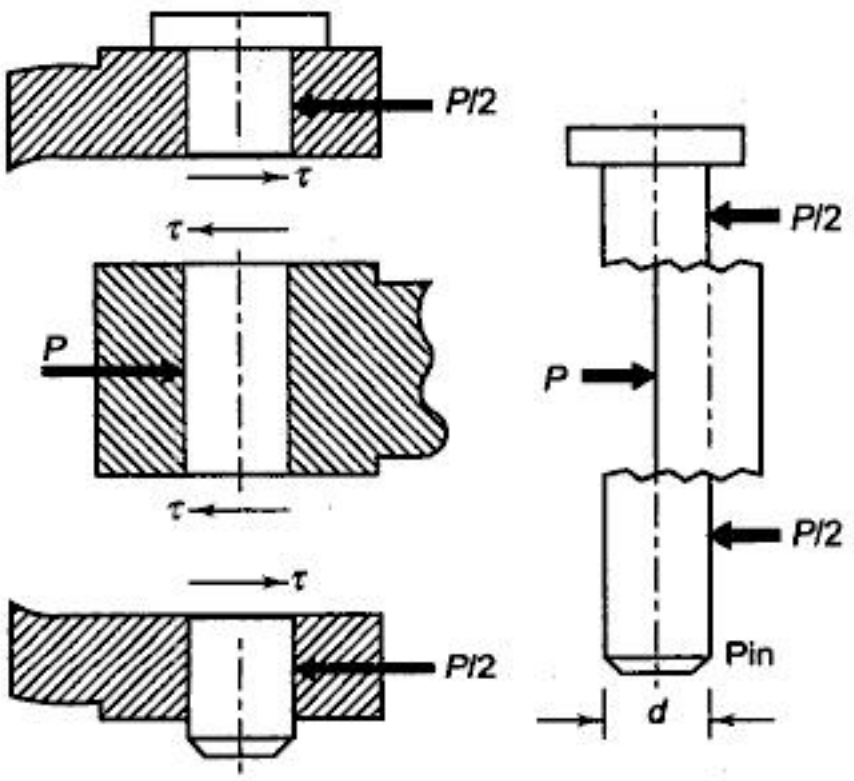

Fig. 4.20 *Shear Failure of Pin*

(iii) Crushing Failure of Pin in Eye When a cylindrical surface such as a pin is subjected to a force along its periphery, its projected area is taken into consideration to find out the stress. As shown in Fig. 4.21, the projected area of the cylindrical surface is $(l \times d)$ and the compressive stress is given by,

$$\sigma_c = \frac{\text{force}}{\text{projected area}} = \frac{P}{(l \times d)}$$

As shown in Fig. 4.18, the projected area of the pin in the eye is (bd) and the compressive stress between the pin and the eye is given by,

$$\sigma_c = \frac{P}{bd} \qquad (4.26e)$$

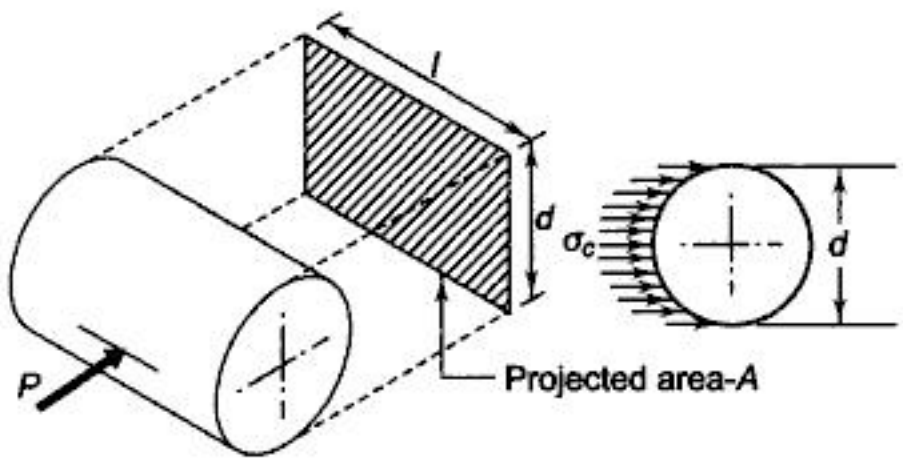

Fig. 4.21 *Projected Area of Cylindrical Surface*

(iv) Crushing Failure of Pin in Fork As shown in Fig. 4.18, the total projected area of the pin in the fork is $(2ad)$ and the compressive stress between the pin and the fork is given by,

$$\sigma_c = \frac{P}{2ad} \qquad (4.26f)$$

(v) Bending Failure of Pin When the pin is tight in the eye and the fork, failure occurs due to shear. On the other hand, when the pin is loose, it is subjected to bending moment as shown in Fig. 4.22. It is assumed that the load acting on the pin is uniformly distributed in the eye, but uniformly varying in two parts of the fork. For triangular distribution of load between the pin and the fork,

$$x = \frac{1}{3} a \qquad \text{also,} \qquad z = \frac{1}{2}\left(\frac{1}{2} b\right) = \frac{1}{4} b$$

The bending moment is maximum at the centre. It is given by,

$$M_b = \frac{P}{2}\left[\frac{b}{2} + x\right] - \frac{P}{2}(z)$$

$$= \frac{P}{2}\left[\frac{b}{2} + \frac{a}{3}\right] - \frac{P}{2}\left[\frac{b}{4}\right] = \frac{P}{2}\left[\frac{b}{4} + \frac{a}{3}\right]$$

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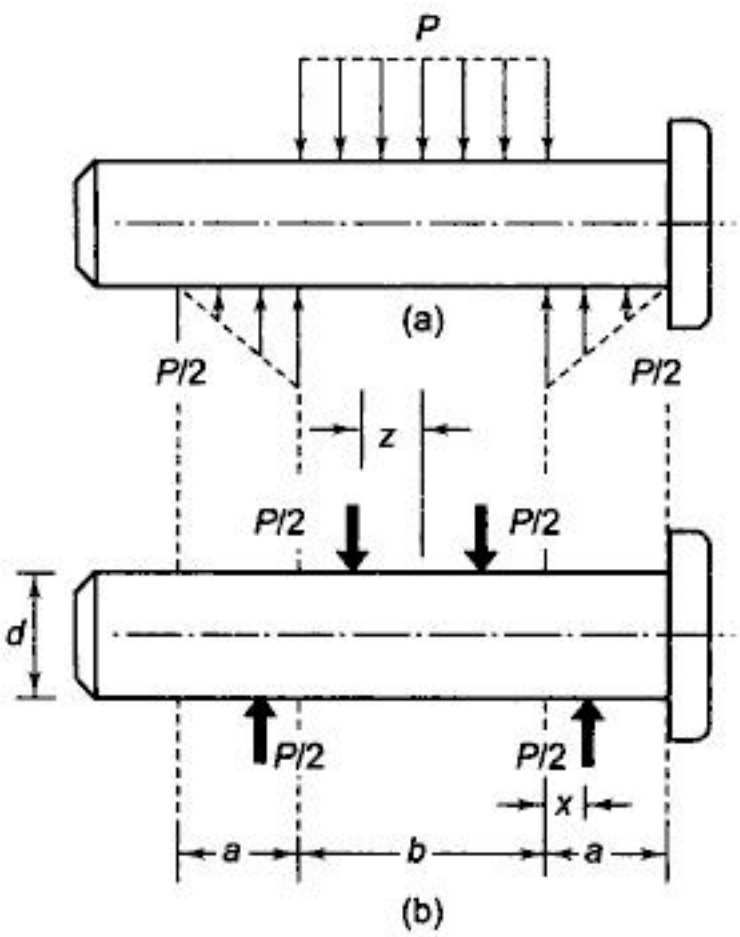

Fig. 4.22 *Pin Treated as Beam (a) Actual Distribution of Forces (b) Simplified Diagram of Forces*

Also, $I = \dfrac{\pi d^4}{64}$ and $y = \dfrac{d}{2}$

From Eq. (4.12),

$$\sigma_b = \frac{M_b y}{I} = \frac{\dfrac{P}{2}\left[\dfrac{b}{4} + \dfrac{a}{3}\right]\dfrac{d}{2}}{\dfrac{\pi d^4}{64}}$$

or $$\sigma_b = \frac{32}{\pi d^3} \times \frac{P}{2}\left[\frac{b}{4} + \frac{a}{3}\right] \qquad (4.26g)$$

(vi) Tensile Failure of Eye Section *XX* shown in Fig. 4.23(a) is the weakest section of the eye. The area of this section is given by,

$$\text{area} = b\,(d_0 - d)$$

The tensile stress at section *XX* is given by,

$$\sigma_t = \frac{P}{\text{area}} \quad \text{or} \quad \sigma_t = \frac{P}{b\,(d_0 - d)} \qquad (4.26h)$$

(vii) Shear Failure of Eye The eye is subjected to double shear as shown in Fig. 4.23(b). The area of each of the two planes resisting the shear failure is

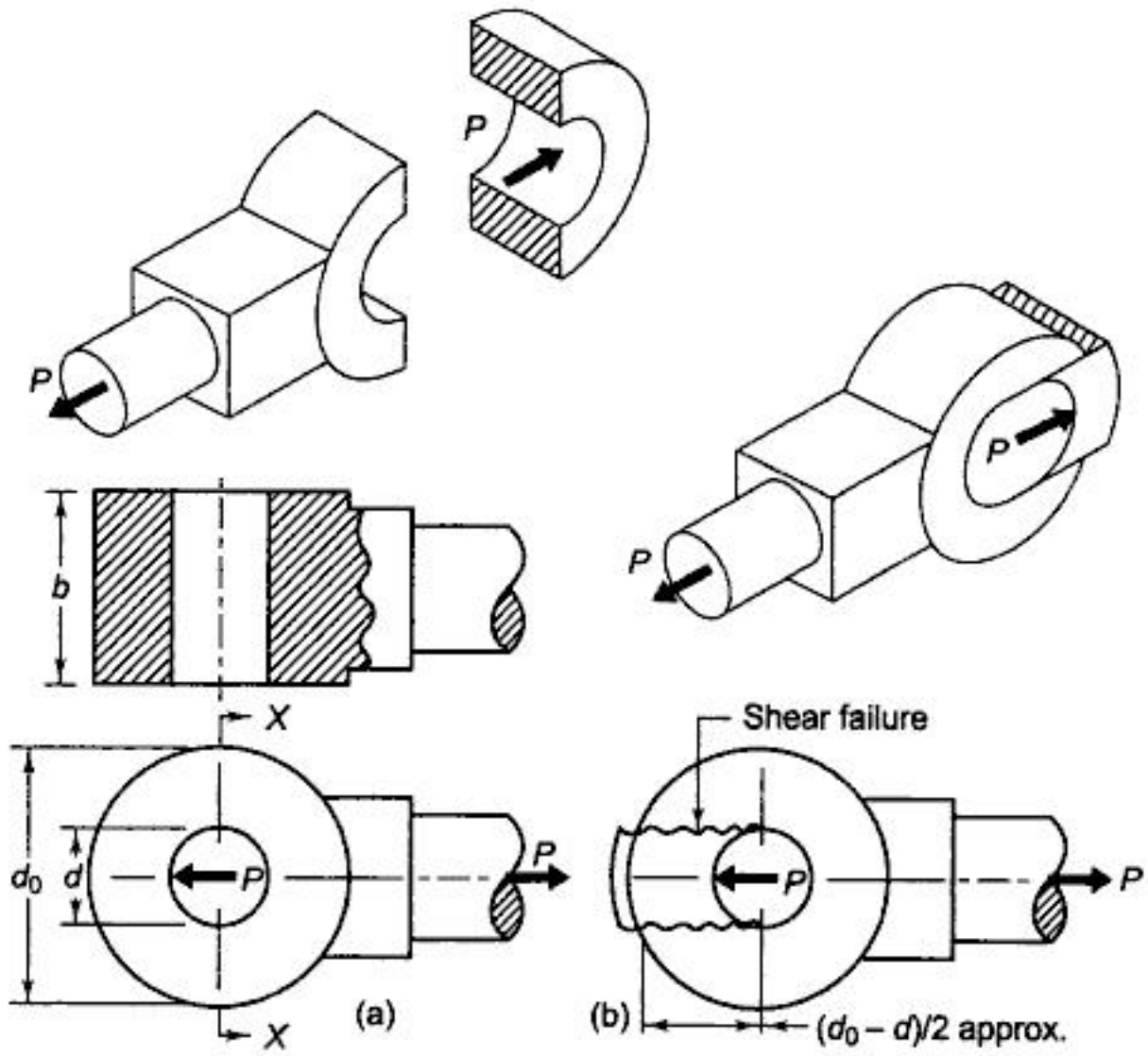

Fig. 4.23 *(a) Tensile Failure of Eye (b) Shear Failure of Eye*

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$[b\,(d_0 - d)/2]$ approximately. Therefore, shear stress is given by,

$$\tau = \frac{P}{2[b(d_0 - d)/2]}$$

or
$$\tau = \frac{P}{b\,(d_0 - d)} \qquad (4.26i)$$

Standard proportion for outside diameter of the eye or the fork is given by the following relationship,

$$d_0 = 2d \qquad (4.26j)$$

(viii) Tensile Failure of Fork Fork is a double eye and as such, Fig. 4.23 is applicable to a fork except for dimension b which can be modified as $2a$ in case of a fork. The area of the weakest section resisting tensile failure is given by

$$\text{area} = 2a\,(d_0 - d)$$

Tensile stress in the fork is given by

$$\sigma_t = \frac{P}{2a\,(d_0 - d)} \qquad (4.26k)$$

(ix) Shear Failure of Fork Each of the two parts of the fork is subjected to double shear. Modifying Eq. (4.26i),

$$\tau = \frac{P}{2a\,(d_0 - d)} \qquad (4.26l)$$

Standard proportions for the dimensions a and b are as follows,

$$a = 0.75\,D \qquad (4.26m)$$

$$b = 1.25\,D \qquad (4.26n)$$

The diameter of the pinhead is taken as,

$$d_1 = 1.5\,d \qquad (4.26o)$$

The gap x shown in Fig. 4.18 is usually taken as 10 mm.

$$\therefore \qquad x = 10 \text{ mm} \qquad (4.26p)$$

The applications of strength equations from (4.26a) to (4.26l) in finding out the dimensions of the knuckle joint are illustrated in the next example. The eye and the fork are usually made by the forging process and the pin is machined from rolled steel bars.

4.12 DESIGN PROCEDURE FOR KNUCKLE JOINT

The basic procedure to determine the dimensions of the knuckle joint consists of the following steps:

(i) Calculate the diameter of each rod by Eq. (4.26a).

$$D = \sqrt{\frac{4P}{\pi\sigma_t}}$$

(ii) Calculate the enlarged diameter of each rod by empirical relationship using Eq. (4.26b).

$$D_1 = 1.1\,D$$

(iii) Calculate the dimensions a and b by empirical relationship using Eqs (4.26m) and (4.26n).

$$a = 0.75\,D \qquad b = 1.25\,D$$

(iv) Calculate the diameters of the pin by shear consideration using Eq. (4.26c) and bending consideration using Eq. (4.26g) and select the diameter, whichever is maximum.

$$d = \sqrt{\frac{2P}{\pi\tau}} \quad \text{or} \quad d = \sqrt[3]{\frac{32}{\pi\sigma_b} \times \frac{P}{2}\left[\frac{b}{4} + \frac{a}{3}\right]}$$

(whichever is maximum)

(v) Calculate the dimensions d_o and d_1 by empirical relationships using Eqs (4.26j) and (4.26o) respectively.

$$d_o = 2d \qquad d_1 = 1.5d$$

(vi) Check the tensile, crushing and shear stresses in the eye by Eqs (4.26h), (4.26e) and (4.26i) respectively.

$$\sigma_t = \frac{P}{b\,(d_0 - d)}$$

$$\sigma_c = \frac{P}{bd}$$

$$\tau = \frac{P}{b\,(d_0 - d)}$$

(vii) Check the tensile, crushing and shear stresses in the fork by Eqs (4.26k), (4.26f) and (4.26l) respectively.

$$\sigma_t = \frac{P}{2a(d_0 - d)}$$

$$\sigma_c = \frac{P}{2ad}$$

$$\tau = \frac{P}{2a(d_0 - d)}$$

The application of the above mentioned procedure is illustrated in the next example.

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Example 4.5 *It is required to design a knuckle joint to connect two circular rods subjected to an axial tensile force of 50 kN. The rods are co-axial and a small amount of angular movement between their axes is permissible. Design the joint and specify the dimensions of its components. Select suitable materials for the parts.*

Solution

Given $P = (50 \times 10^3)$ N

Part I *Selection of material*

The rods are subjected to tensile force. Therefore, yield strength is the criterion for the selection of material for the rods. The pin is subjected to shear stress and bending stresses. Therefore, strength is also the criterion of material selection for the pin. On strength basis, the material for two rods and pin is selected as plain carbon steel of Grade 30C8 (S_{yt} = 400 N/mm^2). It is further assumed that the yield strength in compression is equal to yield strength in tension. In practice, the compressive strength of steel is much higher than its tensile strength.

Part II *Selection of factor of safety*

In stress analysis of knuckle joint, the effect of stress concentration is neglected. To account for this effect, a higher factor of safety of 5 is assumed in the present design.

Part III *Calculation of permissible stresses*

$$\sigma_t = \frac{S_{yt}}{(fs)} = \frac{400}{5} = 80 \text{ N/mm}^2$$

$$\sigma_c = \frac{S_{yc}}{(fs)} = \frac{S_{yt}}{(fs)} = \frac{400}{5} = 80 \text{ N/mm}^2$$

$$\tau = \frac{S_{sy}}{(fs)} = \frac{0.5\, S_{yt}}{(fs)} = \frac{0.5(400)}{5} = 40 \text{ N/mm}^2$$

Part IV *Calculation of dimensions*

The dimensions of the knuckle joint are calculated by the procedure outlined in Section 4.10.

Step I *Diameter of rods*

$$D = \sqrt{\frac{4P}{\pi \sigma_t}} = \sqrt{\frac{4(50 \times 10^3)}{\pi (80)}} = 28.21 \text{ or } 30 \text{ mm}$$

Step II *Enlarged diameter of rods (D_1)*

$$D_1 = 1.1\, D = 1.1(30) = 33 \text{ or } 35 \text{ mm}$$

Step III *Dimensions a and b*

$$a = 0.75\, D = 0.75(30) = 22.5 \text{ or } 25 \text{ mm}$$
$$b = 1.25\, D = 1.25(30) = 37.5 \text{ or } 40 \text{ mm}$$

Step IV *Diameter of pin*

$$d = \sqrt{\frac{2P}{\pi \tau}} = \sqrt{\frac{2(50 \times 10^3)}{\pi (40)}} = 28.21 \text{ or } 30 \text{ mm}$$

Also,

$$d = \sqrt[3]{\frac{32}{\pi \sigma_b} \times \frac{P}{2}\left[\frac{b}{4} + \frac{a}{3}\right]}$$

$$= \sqrt[3]{\frac{32}{\pi (80)} \times \frac{(50 \times 10^3)}{2}\left[\frac{40}{4} + \frac{25}{3}\right]}$$

$$= 38.79 \text{ or } 40 \text{ mm}$$

$\therefore \quad d = 40$ mm

Step V *Dimensions d_0 and d_1*

$$d_0 = 2d = 2(40) = 80 \text{ mm}$$
$$d_1 = 1.5d = 1.5(40) = 60 \text{ mm}$$

Step VI *Check for stresses in eye*

$$\sigma_t = \frac{P}{b(d_0 - d)} = \frac{(50 \times 10^3)}{40(80 - 40)} = 31.25 \text{ N/mm}^2$$

$\therefore \quad \sigma_t < 80$ N/mm^2

$$\sigma_c = \frac{P}{b\, d} = \frac{(50 \times 10^3)}{40(40)} = 31.25 \text{ N/mm}^2$$

$\therefore \quad \sigma_c < 80$ N/mm^2

$$\tau = \frac{P}{b(d_0 - d)} = \frac{(50 \times 10^3)}{40(80 - 40)} = 31.25 \text{ N/mm}^2$$

$\therefore \quad \tau < 40$ N/mm^2

Step VII *Check for stresses in fork*

$$\sigma_t = \frac{P}{2a(d_0 - d)} = \frac{(50 \times 10^3)}{2(25)(80 - 40)} = 25 \text{ N/mm}^2$$

$\therefore \quad \sigma_t < 80$ N/mm^2

$$\sigma_c = \frac{P}{2ad} = \frac{(50 \times 10^3)}{2(25)(40)} = 25 \text{N/mm}^2$$

$\therefore \quad \sigma_c < 80$ N/mm^2

$$\tau = \frac{P}{2a(d_0 - d)}$$

$$= \frac{(50 \times 10^3)}{2(25)(80 - 40)} = 25 \text{N/mm}^2$$

$\therefore \quad \tau < 40$ N/mm^2

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It is observed that stresses are within limits.

Part V *Dimensioned sketch of knuckle joint*

Main dimensions of the knuckle joint are shown in Fig. 4.24.

Example 4.6 *A wall-rack, used to store round steel bars, consists of two I-section cantilever beams fixed in the wall. The bars are stacked in a triangular fashion as shown in Fig. 4.25(a). The total weight of the bars is 75 kN. The permissible bending stress for the cantilevers is 165 N/mm².*

Select a standard rolled I-section beam from the following table:

Designation	*b (mm)*	*h (mm)*	I_{xx} *(mm⁴)*
ISLB 150	80	150	688.2×10^4
ISLB 175	90	175	1096.2×10^4
ISLB 200	100	200	1696.6×10^4
ISLB 225	100	225	2501.9×10^4
ISLB 250	125	250	3717.8×10^4

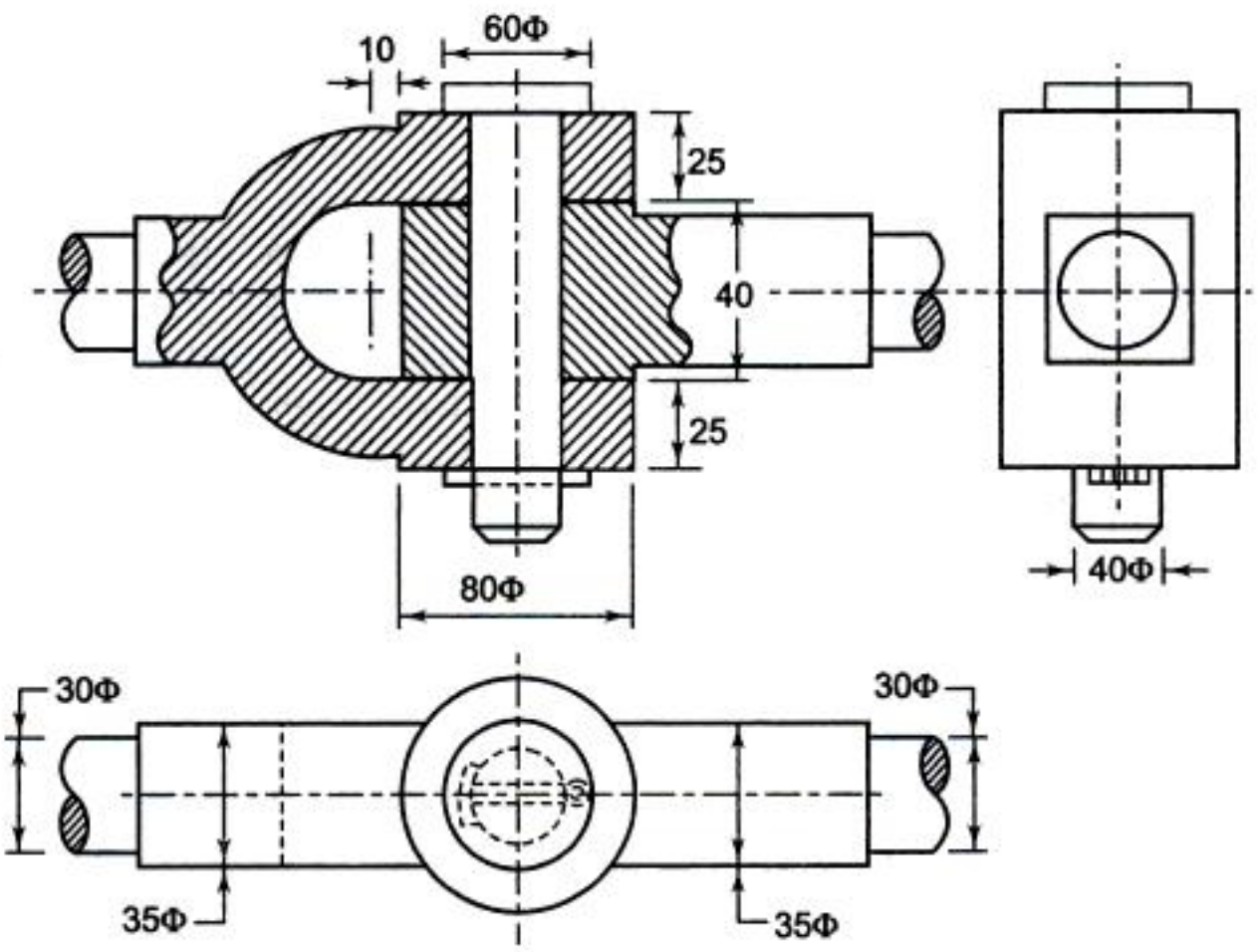

Fig 4.24 *Dimensions of Knuckle Joint*

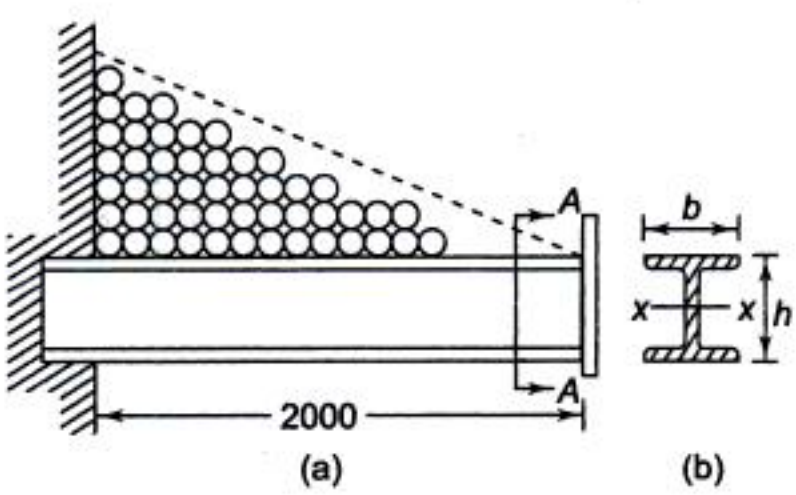

Fig 4.25

Solution

Given $W = 75$ kN $\quad \sigma_b = 165$ N/mm²

Step I *Calculation of bending moment*

There are two cantilever beams and the load supported by each beam is (75/2) or 37.5 kN. For a triangular load distribution, the centre of gravity of the resultant load is at a distance of (2000/3) mm from the wall. Therefore,

$$M_b = (37.5 \times 10^3)\left(\frac{2000}{3}\right) = 25 \times 10^6 \text{ N-mm}$$

Step II *Calculation of* (I_{xx}/y)

From Eq. (4.12),

$$\frac{I_{xx}}{y} = \frac{M_b}{\sigma_b} = \frac{25 \times 10^6}{165} = 151.51 \times 10^3 \text{ mm}^3$$

Step III *Selection of beam*

The cross-section of the beam is shown in Fig. 4.25 (b), $(y = h/2)$

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Trial I Suppose beam (ISLB 175) is suitable for the application. For this beam,

$$\frac{I_{xx}}{y} = \frac{1096.2 \times 10^4}{(175/2)} = 125.28 \times 10^3 \text{ mm}^3$$

Since the required (I_{xx}/y) is (151.51×10^3) mm^2, beam (ISLB 175) is not suitable.

Trial II Suppose beam (ISLB 200) is suitable for the application. For this beam,

$$\frac{I_{xx}}{y} = \frac{1696.6 \times 10^4}{(200/2)} = 169.66 \times 10^3 \text{ mm}^3$$

$(I_{xx}/y) > (151.51 \times 10^3)$ mm^3

Therefore, the cantilever beams of standard cross-section ISLB 200 are suitable for this application.

Example 4.7 *The frame of a hacksaw is shown in Fig. 4.26(a). The initial tension P in the blade should be 300 N. The frame is made of plain carbon steel 30C8 with a tensile yield strength of 400 N/mm^2 and the factor of safety is 2.5. The cross-section of the frame is rectangular with a ratio of depth to width as 3, as shown in Fig. 4.26(b). Determine the dimensions of the cross-section.*

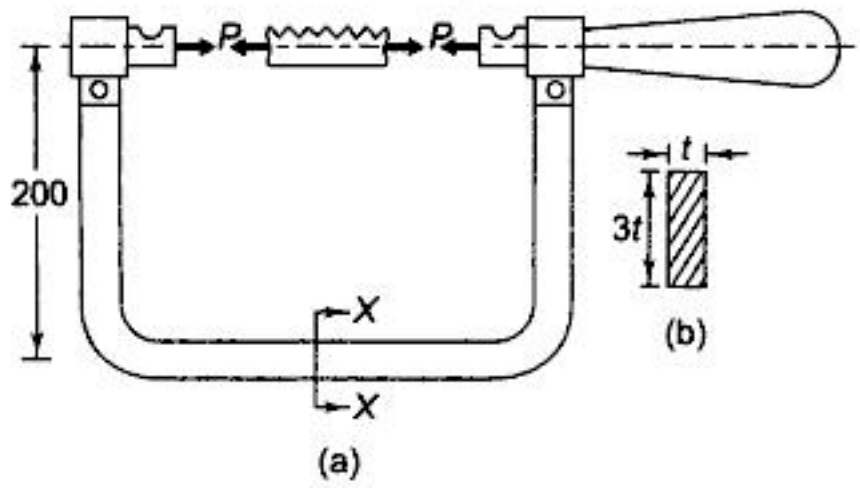

Fig. 4.26 *(a) Frame of Hacksaw (b) Section at XX*

Solution

Given $P = 300$ N $S_{yt} = 400$ N/mm^2 $(fs) = 2.5$
(depth/width) = 3

Step I *Calculation of permissible tensile stress*

$$\sigma_t = \frac{S_{yt}}{(fs)} = \frac{400}{2.5} = 160 \text{ N/mm}^2$$

Step II *Calculation of direct and bending stresses*
The stresses at section *XX* consist of direct compressive stress and bending stresses. The tensile stress is maximum at the lower fibre. At the lower fibre,

$$\sigma_c = \frac{P}{A} = \frac{300}{(t)(3t)} = \frac{100}{t^2} \text{ N/mm}^2 \quad \text{(i)}$$

$$\sigma_b = \frac{M_b\, y}{I}$$

$$= \frac{(300 \times 200)\ (1.5\, t)}{\left[\frac{1}{12}(t)(3t)^3\right]} = \frac{40000}{t^3} \text{ N/mm}^2 \quad \text{(ii)}$$

Step III *Calculation of dimensions of cross-section*
Superimposing the two stresses and equating it to permissible stress,

$$\frac{40000}{t^3} - \frac{100}{t^2} = 160$$

or $160\, t^3 + 100\, t - 40000 = 0$

Solving the above equation by trail and error method,

$$t \cong 6.3 \text{ mm}$$

Example 4.8 *An offset link subjected to a force of 25 kN is shown in Fig. 4.27. It is made of grey cast iron FG300 and the factor of safety is 3. Determine the dimensions of the cross-section of the link.*

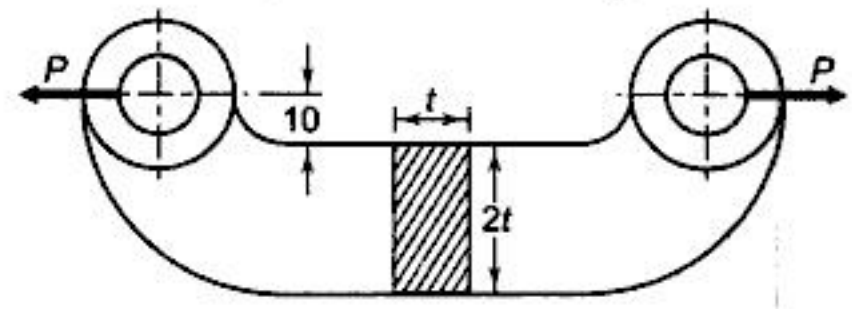

Fig. 4.27 *Offset Link*

Solution

Given $P = 25$ kN $S_{ut} = 300$ N/mm^2 $(fs) = 3$

Step I *Calculation of permissible tensile stress for the link*

$$\sigma_t = \frac{S_{ut}}{(fs)} = \frac{300}{3} = 100 \text{ N/mm}^2 \quad \text{(a)}$$

Step II *Calculation of direct tensile and bending stresses*
The cross-section is subjected to direct tensile stress and bending stresses. The stresses are maximum at the top fibre. At the top fibre,

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$$\sigma_t = \frac{P}{A} + \frac{M_b\, y}{I}$$

$$= \frac{25\times10^3}{t(2t)} + \frac{25\times10^3(10+t)(t)}{\left[\frac{1}{12}t(2t)^3\right]} \qquad \text{(b)}$$

Step III *Calculation of dimensions of cross-section*

Equating (a) and (b),

$$\frac{12500}{t^2} + \frac{37500\,(10+t)}{t^3} = 100$$

or, $t^3 - 500\,t - 3750 = 0$

Solving the above equation by trial and error method,

$$t \cong 25.5 \text{ mm}$$

Example 4.9 *The frame of a hydraulic press consisting of two identical steel plates is shown in Fig. 4.28. The maximum force P acting on the frame is 20 kN. The plates are made of steel 45C8 with tensile yield strength of 380 N/mm². The factor of safety is 2.5. Determine the plate thickness.*

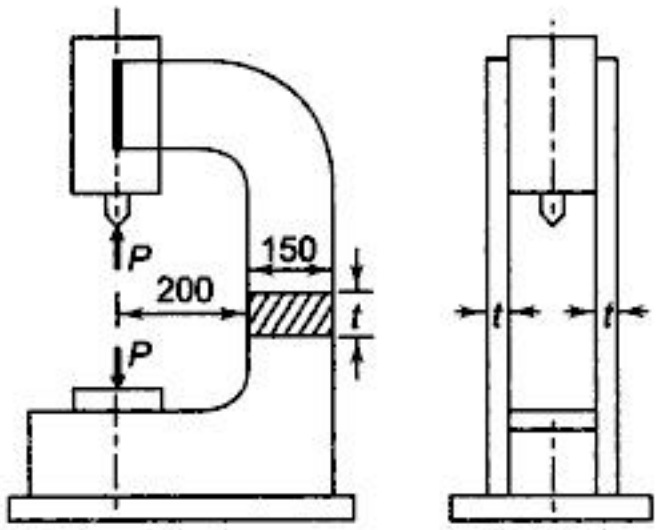

Fig. 4.28 *Frame of Hydraulic Press*

Solution

Given $P = 20$ kN $\quad S_{yt} = 380$ N/mm²
$(fs) = 2.5$

Step I *Calculation of permissible tensile stress for the plates*

$$\sigma_t = \frac{S_{yt}}{(fs)} = \frac{380}{2.5} = 152 \text{ N/mm}^2 \qquad \text{(i)}$$

Step II *Calculation of direct tensile and bending stresses in plates*

Since the plates are identical, the force acting on each plate is (20/2) or 10 kN. The plates are subjected to direct tensile stress and bending stresses. The stresses are maximum at the inner fibre.

At the inner fibre,

$$\sigma_t = \frac{P}{A} + \frac{M_b y}{I}$$

$$= \frac{10000}{(150\,t)} + \frac{10000\ (200+75)\,(75)}{\left[\frac{1}{12}t\,(150)^3\right]}$$

or $\qquad \sigma_t = \dfrac{800}{t} \qquad \text{(ii)}$

Step III *Calculation of plate thickness*

From (i) and (ii),

$$\frac{800}{t} = 152 \quad \text{or} \quad t = 5.26 \text{ mm}$$

Example 4.10 *A hollow circular column carries a projecting bracket, which supports a load of 25 kN as shown in Fig. 4.29. The distance between the axis of the column and the load is 500 mm. The inner diameter of the column is 0.8 times of the outer diameter. The column is made of steel FeE 200 (S_{yt} = 200 N/mm²) and the factor of safety is 4. The column is to be designed on the basis of maximum tensile stress and compression is not the criterion of failure. Determine the dimensions of the cross-section of the column.*

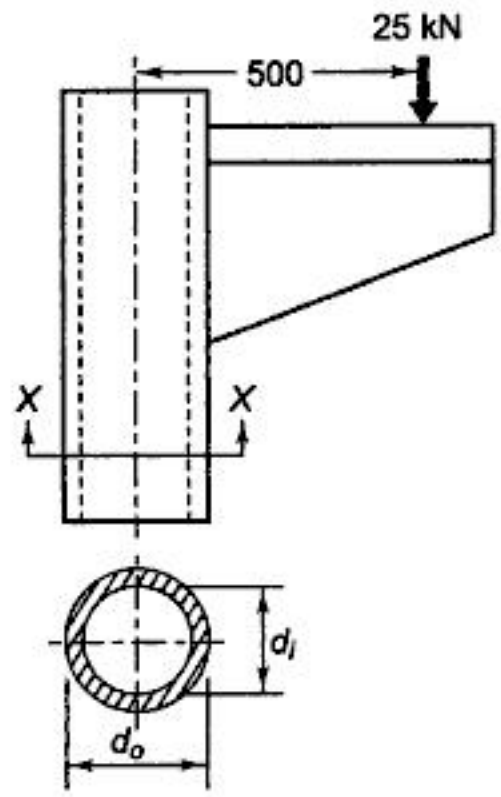

Fig. 4.29

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Solution

Given $P = 25$ kN $\quad e = 500$ mm

$S_{yt} = 200$ N/mm^2 $\quad (fs) = 4$ $\quad d_i = 0.8\, d_o$

Step I *Calculation of permissible tensile stress*

$$\sigma_t = \frac{S_{yt}}{(fs)} = \frac{200}{4} = 50 \text{ N/mm}^2$$

Step II *Calculation of direct compressive and bending stresses*

$$d_i = 0.8\, d_0$$

The direct compressive stress is given by,

$$\frac{P}{A} = \frac{25 \times 10^3}{\frac{\pi}{4}(d_o^2 - d_i^2)}$$

$$= \frac{25 \times 10^3}{\frac{\pi}{4}[d_0^2 - (0.8\, d_0)^2]} = \left(\frac{88419.41}{d_0^2}\right) \text{N/mm}^2 \qquad \text{(i)}$$

The bending stresses are tensile on the left side and compressive on the right side of the cross-section. The tensile stress due to bending moment is given by,

$$\frac{Pey}{I} = \frac{25 \times 10^3 (500)(0.5 d_0)}{\frac{\pi}{64}\left[d_0^4 - (0.8 d_0)^4\right]}$$

$$= \left(\frac{215\,657\,104.5}{d_0^3}\right) \text{N/mm}^2 \qquad \text{(ii)}$$

Step III *Calculation of outer and inner diameters*

The resultant tensile stress is obtained by subtracting (i) from (ii). Equating the resultant stress to permissible tensile stress,

$$\left(\frac{215\,657\,104{\cdot}5}{d_0^3}\right) - \left(\frac{88\,419.41}{d_0^2}\right) = 50$$

or, $\qquad (d_0^3 + 1768.39\, d_0) = 4\,313\,142{\cdot}09$

The above expression indicates cubic equation and it is solved by trial and error method. The trial values of d_o and corresponding values of left hand side are tabulated as follows:

d_0	$(d_0^3 + 1768.39\, d_0)$
150	3 640 258.5
160	4 378 942.4
158	4 223 717.6
159	4 300 853.0
160	4 378 942.4

From the above table, it is observed that the value of d_o is between 159 and 160 mm

$\therefore \quad d_0 = 160$ mm

$d_i = 0.8\, d_o = 0.8\,(160) = 128$ mm

4.13 PRINCIPAL STRESSES

In previous articles and examples, mechanical components, which are subjected to only one type of load, are considered. There are many components, which are subjected to several types of load simultaneously. A transmission shaft is subjected to bending as well as torsional moment at the same time. In design, it is necessary to determine the state of stresses under these conditions. An element of a plate subjected to two-dimensional stresses is shown is Fig. 4.30(a). In this analysis, the stresses are classified into two groups—normal stresses and shear stresses. The *normal stress* is perpendicular to the area under consideration, while the *shear stress* acts over the area.

There is a particular system of notation for these stresses. The normal stresses are denoted by σ_x, σ_y and σ_z in the X, Y and Z directions respectively. Tensile stresses are considered to be positive, while compressive stresses as negative. The shear stresses are denoted by two subscripts, viz. τ_{xy} or τ_{yz}, as shown in Fig. 4.30(a). The first subscript denotes the area over which it acts and the second indicates the direction of shear force. As an example, consider the shear stress denoted by τ_{xy}. The subscript x indicates that the shear stress is acting on the area, which is perpendicular to the X-axis. The subscript y indicates that the shear stress is acting in the Y-direction. The shear stresses are positive if they act in the positive direction of the reference axis. It can be proved that,

$$\tau_{xy} = \tau_{yx} \qquad (4.27)$$

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Figure 4.30(b) shows the stresses acting on an oblique plane. The normal to the plane makes an angle θ with the X-axis. σ and τ are normal

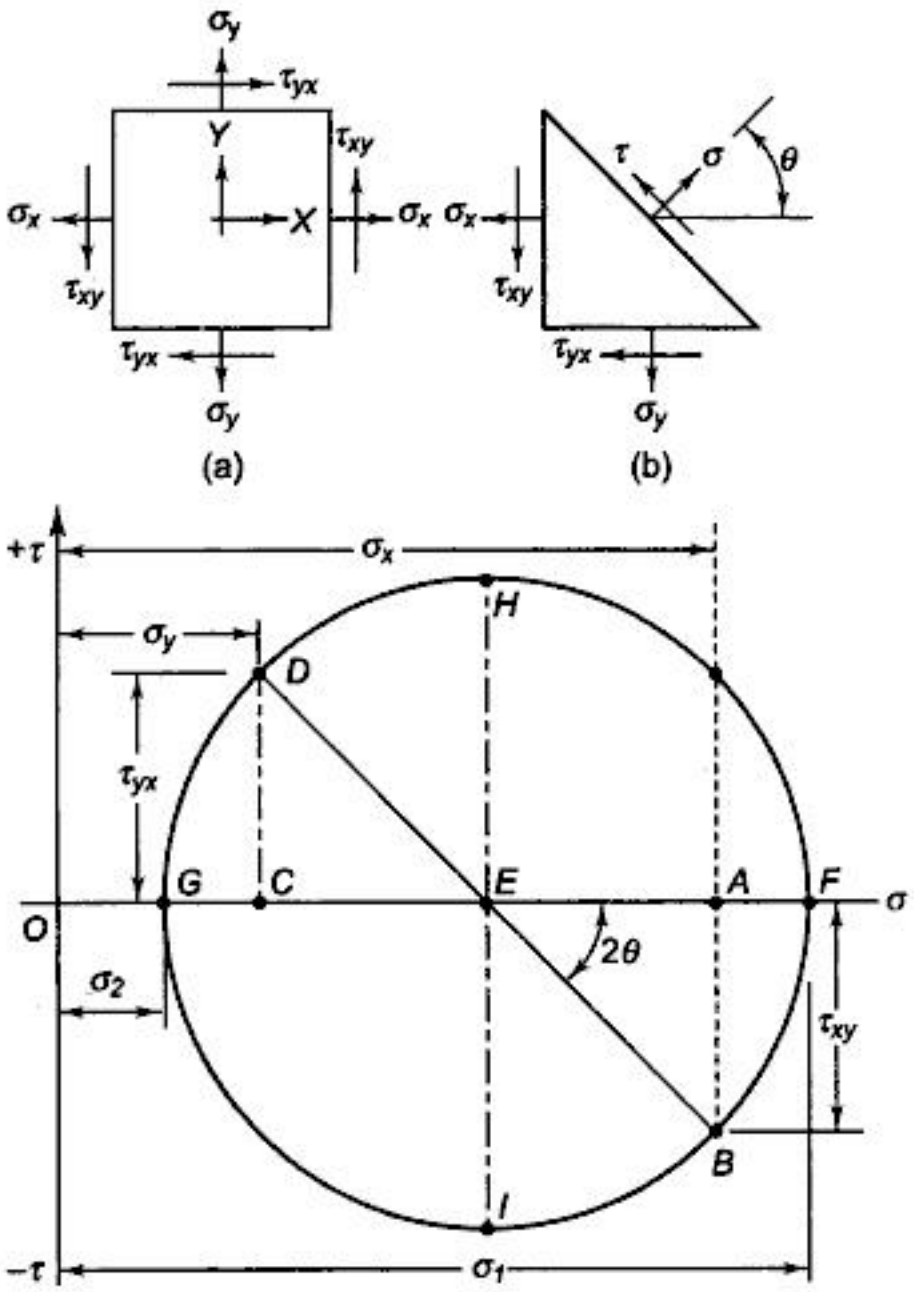

Fig. 4.30 *(a) Two-Dimensional State of Stress (b) Stresses on Oblique Plane (c) Mohr's Circle Diagram*

and shear stresses associated with this plane. Considering equilibrium of forces, it can be proved that,[2, 3]

$$\sigma = \left(\frac{\sigma_x + \sigma_y}{2}\right) + \left(\frac{\sigma_x - \sigma_y}{2}\right)\cos 2\theta + \tau_{xy} \sin 2\theta \quad (4.28)$$

and

$$\tau = -\left(\frac{\sigma_x - \sigma_y}{2}\right)\sin 2\theta + \tau_{xy} \cos 2\theta \quad (4.29)$$

Differentiating Eq. (4.28) with respect to θ and setting the result to zero, we have

$$\tan 2\theta = \frac{2\tau_{xy}}{\sigma_x - \sigma_y} \quad (4.30)$$

Equation (4.30) defines two values of (2θ), one giving the maximum value of normal stress and other the minimum value. If σ_1 and σ_2 are the maximum and minimum values of normal stress, then substituting Eq. (4.30) in Eq. (4.28), we get

$$\sigma_1 = \left(\frac{\sigma_x + \sigma_y}{2}\right) + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} \quad (4.31)$$

$$\sigma_2 = \left(\frac{\sigma_x + \sigma_y}{2}\right) - \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} \quad (4.32)$$

σ_1 and σ_2 are called the principal stresses.

Similarly, Eq. (4.29) is differentiated with respect to θ and the result is equated to zero. This gives the following condition:

$$\tan 2\theta = -\left(\frac{\sigma_x - \sigma_y}{2\tau_{xy}}\right) \quad (4.33)$$

Substituting Eq. (4.33) in Eq. (4.29),

$$\tau_{max.} = \pm\sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} \quad (4.34)$$

$\tau_{max.}$ is called the principal shear stress.

One of the most effective methods to determine the principal stresses and the principal shear stress is the construction of *Mohr's circle diagram* as shown in Fig. 4.30 (c). It is a graphical method for the representation of stresses. The following conventions are used to construct the *Mohr's circle*:

(i) The normal stresses σ_x, σ_y, and the principal stresses σ_1, σ_2 are plotted on the abscissa. The tensile stress, considered as positive, is plotted to the right of the origin and the compressive stress, considered as negative, to its left.

[2] JE Shigley —*Applied Mechanics of Materials*—McGraw-Hill Kogakusha Ltd.
[3] FR Shanley —*Strength of Materials*—McGraw-Hill Book Company Inc.

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(ii) The shear stresses τ_{xy}, τ_{yx} and the principal shear stress τ_{max} are plotted on the ordinate. A pair of shear stresses is considered as positive if they tend to rotate the element clockwise, and negative if they tend to rotate it anticlockwise.

The Mohr's circle in Fig. 4.30 (c) is constructed by the following method:

(i) Plot the following points:

$\overline{OA} = \sigma_x$ $\quad$ $\overline{OC} = \sigma_y$

$\overline{AB} = \tau_{xy}$ $\quad$ $\overline{CD} = \tau_{yx}$

(ii) Join $\overline{DB}$. The point of intersection of $\overline{DB}$ and $\overline{OA}$ is E.

(iii) Construct Mohr's circle with E as centre and $\overline{DB}$ as the diameter.

It can be proved that points F and G represent the maximum and minimum principal stresses σ_1 and σ_2 respectively. The two principal shear stresses $\pm\tau_{max}$ are denoted by points H and I respectively.

4.14 THEORIES OF ELASTIC FAILURE

There are number of machine components, which are subjected to several types of loads simultaneously. For example, a power screw is subjected to torsional moment as well as axial force. Similarly, an overhang crank is subjected to combined bending and torsional moments. The bolts of the bracket are subjected to forces that cause tensile stress and shear stress. Crankshaft, propeller shaft and connecting rod are examples of components subjected to complex loads. When the component is subjected to several types of loads, combined stresses are induced. For example, torsional moment induces torsional shear stress, while bending moment causes bending stresses in the transmission shaft. The failures of such components are broadly classified into two groups—elastic failure and yielding and fracture. Elastic failure results in excessive elastic deformation, which makes the machine component unfit to perform its function satisfactorily. Yielding results in excessive plastic deformation after the yield point stress is reached, while fracture results in breaking the component into two or more pieces. *Theories of failure discussed in this article are applicable to elastic failure of machine parts.*

The design of machine parts subjected to combined loads should be related to experimentally determined properties of material under 'similar' conditions. However, it is not possible to conduct such tests for different possible combinations of loads and obtain mechanical properties. In practice, the mechanical properties are obtained from a simple tension test. They include yield strength, ultimate tensile strength and percentage elongation. In the tension test, the specimen is axially loaded in tension. It is not subjected to either bending moment or torsional moment or a combination of loads. Theories of elastic failure provide a relationship between the strength of machine component subjected to complex state of stresses with the mechanical properties obtained in tension test. With the help of these theories, the data obtained in the tension test can be used to determine the dimensions of the component, irrespective of the nature of stresses induced in the component due to complex loads.

Several theories have been proposed, each assuming a different hypothesis of failure. The principal theories of elastic failure are as follows:

(i) Maximum principal stress theory (Rankine's theory)

(ii) Maximum shear stress theory (Coulomb, Tresca and Guest's theory)

(iii) Distortion energy theory (Huber von Mises and Hencky's theory)

(iv) Maximum strain theory (St. Venant's theory)

(v) Maximum total strain energy theory (Haigh's theory)

We will discuss the first three theories in this chapter. Let us assume σ_1, σ_2 and σ_3 as the principal stresses induced at a point on the machine part as a result of several types of loads. We will apply the theories of failure to obtain the relationship between σ_1, σ_2 and σ_3 on one hand and the properties of material such as S_{yt} or S_{ut} on the other.

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4.15 MAXIMUM PRINCIPAL STRESS THEORY

This criterion of failure is accredited to the British engineer WJM Rankine (1850). *The theory states that the failure of the mechanical component subjected to bi-axial or tri-axial stresses occurs when the maximum principal stress reaches the yield or ultimate strength of the material.*

If σ_1, σ_2 and σ_3 are the three principal stresses at a point on the component and

$$\sigma_1 > \sigma_2 > \sigma_3$$

then according to this theory, the failure occurs whenever

$$\sigma_1 = S_{yt} \quad \text{or} \quad \sigma_1 = S_{ut} \tag{4.35}$$

whichever is applicable.

The theory considers only the maximum of principal stresses and disregards the influence of the other principal stresses. The dimensions of the component are determined by using a factor of safety.

For tensile stresses,

$$\sigma_1 = \frac{s_{yt}}{(fs)} \quad \text{or} \quad \sigma_1 = \frac{s_{ut}}{(fs)} \tag{4.36}$$

For compressive stresses,

$$\sigma_1 = \frac{s_{yc}}{(fs)} \quad \text{or} \quad \sigma_1 = \frac{s_{uc}}{(fs)} \tag{4.37}$$

Region of Safety

The construction of region of safety for bi-axial stresses is illustrated in Fig. 4.31. The two principal stresses σ_1 and σ_2 are plotted on X and Y axes respectively. Tensile stresses are considered as positive,

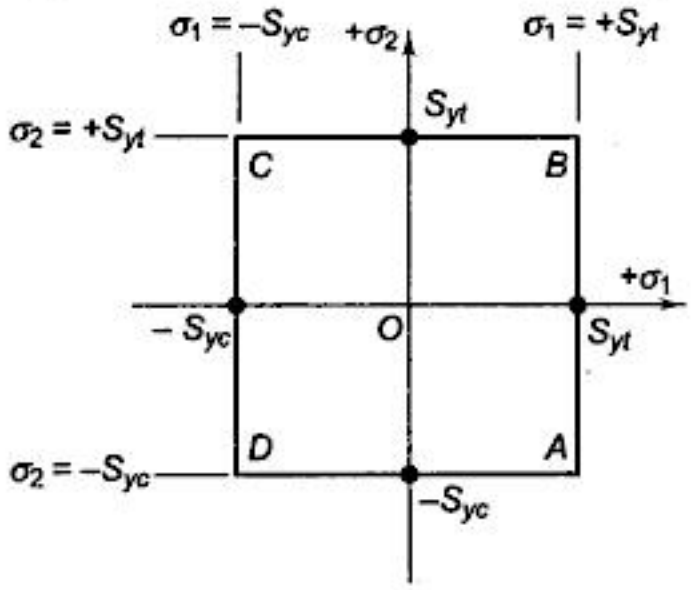

Fig. 4.31 *Boundary for Maximum Principal Stress Theory under Bi-axial Stresses*

while compressive stresses as negative. It is further assumed that

$$S_{yc} = S_{yt}$$

It should be noted that,

(i) The equation of vertical line to the positive side of X-axis is ($x = + a$)

(ii) The equation of vertical line to the negative side of X-axis is ($x = - a$)

(iii) The equation of horizontal line to the positive side of Y-axis is ($y = + b$)

(iv) The equation of horizontal line to the negative side of Y-axis is ($y = - b$)

The borderline for the region of safety for this theory can be constructed in the following way:

Step 1: Suppose $\sigma_1 > \sigma_2$ As per this theory, we will consider only the maximum of principal stresses (σ_1) and disregard the other principal stress (σ_2).

Suppose (σ_1) is the tensile stress. The limiting value of (σ_1) is yield stress (S_{yt}). Therefore, the boundary line will be,

$$\sigma_1 = + S_{yt}$$

A vertical line $\overline{AB}$ is constructed such that $\sigma_1 = + S_{yt}$.

Step 2: Suppose $\sigma_1 > \sigma_2$ and (σ_1) is compressive stress. The limiting value of (σ_1) is compressive yield stress ($- S_{yc}$). Therefore, the boundary line will be,

$$\sigma_1 = - S_{yc}$$

A vertical line $\overline{DC}$ is constructed such that $\sigma_1 = - S_{yc}$.

Step 3: Suppose $\sigma_2 > \sigma_1$ As per this theory, we will consider only the maximum of principal stresses (σ_2) and disregard the other principal stress (σ_1).

Suppose (σ_2) is the tensile stress. The limiting value of (σ_2) is the yield stress (S_{yt}). Therefore, the boundary line will be,

$$\sigma_2 = + S_{yt}$$

A horizontal line $\overline{CB}$ is constructed such that $\sigma_2 = + S_{yt}$

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Step 4: Suppose $\sigma_2 > \sigma_1$ and (σ_2) is the compressive stress. The limiting value of (σ_2) is compressive yield stress $(-S_{yc})$. Therefore, the boundary line will be,

$$\sigma_2 = -S_{yc}$$

A horizontal line $\overline{DA}$ is constructed such that $\sigma_2 = -S_{yc}$.

The complete region of safety is the area *ABCD*. Since we have assumed $(S_{yc} = S_{yt})$, *ABCD* is a square.

According to the maximum principal stress theory of failure, if a point with co-ordinates (σ_1, σ_2) falls outside this square then it indicates the failure condition. On the other hand, if the point falls inside the square, the design is safe and the failure may not occur.

Experimental investigations suggest that the maximum principal stress theory gives good predictions for brittle materials. However, it is not recommended for ductile materials.

4.16 MAXIMUM SHEAR STRESS THEORY

This criterion of failure is accredited to CA Coulomb, H Tresca and JJ Guest. *The theory states that the failure of a mechanical component subjected to bi-axial*

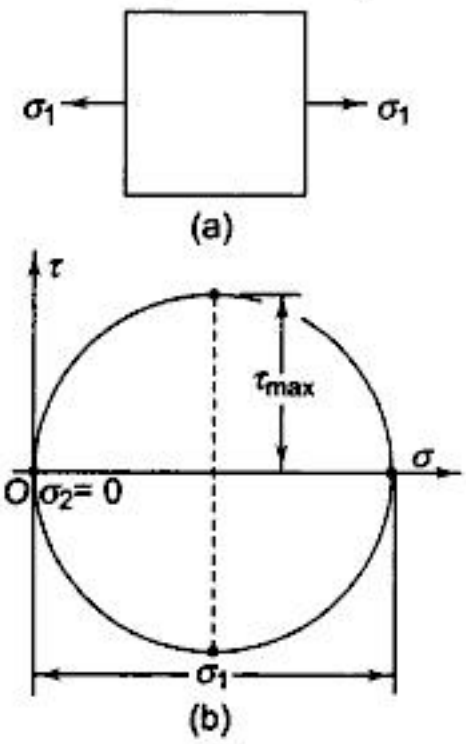

Fig. 4.32 *(a) Stresses in Simple Tension Test (b) Mohr's Circle for Stresses*

or tri-axial stresses occurs when the maximum shear stress at any point in the component becomes equal to the maximum shear stress in the standard specimen of the tension test, when yielding starts. In the tension test, the specimen is subjected to uni-axial stress (σ_1) and $(\sigma_2 = 0)$. The stress in the specimen of tension test and the corresponding Mohr's circle diagram are shown in Fig. 4.32. From the figure,

$$\tau_{max} = \frac{\sigma_1}{2}$$

When the specimen starts yielding $(\sigma_1 = S_{yt})$, the above equation is written as

$$\tau_{max} = \frac{S_{yt}}{2}$$

Therefore, the maximum shear stress theory predicts that the yield strength in shear is half of the yield strength in tension, i.e.,

$$S_{sy} = 0.5\, S_{yt} \qquad (4.38)$$

Suppose σ_1, σ_2 and σ_3 are the three principal stresses at a point on the component, the shear stresses on three different planes are given by,

$$\tau_{12} = \left(\frac{\sigma_1 - \sigma_2}{2}\right) \quad \tau_{23} = \left(\frac{\sigma_2 - \sigma_3}{2}\right)$$

$$\tau_{31} = \left(\frac{\sigma_3 - \sigma_1}{2}\right) \qquad (a)$$

The largest of these stresses is equated to (τ_{max}) or $(S_{yt}/2)$.

Considering factor of safety,

$$\left(\frac{\sigma_1 - \sigma_2}{2}\right) = \frac{S_{yt}}{2(fs)}$$

or
$$(\sigma_1 - \sigma_2) = \frac{S_{yt}}{(fs)}$$

$$(\sigma_2 - \sigma_3) = \frac{S_{yt}}{(fs)}$$

$$(\sigma_3 - \sigma_1) = \frac{S_{yt}}{(fs)} \qquad (4.39)$$

The above relationships are used to determine the dimensions of the component. Refer to expression (a) again and equating the largest shear stress $(\tau_{max.})$ to $(S_{yt}/2)$,

$$\left(\frac{\sigma_1 - \sigma_2}{2}\right) = \frac{S_{yt}}{2}$$

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or $\sigma_1 - \sigma_2 = S_{yt}$ (b)

Similarly,

$\sigma_2 - \sigma_3 = S_{yt}$ (c)

$\sigma_3 - \sigma_1 = S_{yt}$ (d)

For compressive stresses,

$\sigma_1 - \sigma_2 = -S_{yc}$ (e)

$\sigma_2 - \sigma_3 = -S_{yc}$ (f)

$\sigma_3 - \sigma_1 = -S_{yc}$ (g)

The above equations can be written as,

$\sigma_1 - \sigma_2 = \pm S_{yt}$ [Assuming $S_{yc} = S_{yt}$]

$\sigma_2 - \sigma_3 = \pm S_{yc}$

$\sigma_3 - \sigma_1 = \pm S_{yt}$

Region of Safety For bi-axial stresses,

$\sigma_3 = 0$

The above equations can be written as,

$\sigma_1 - \sigma_2 = \pm S_{yt}$ (h)

$\sigma_2 = \pm S_{yt}$ (i)

$\sigma_1 = \pm S_{yt}$ (j)

It will be observed at a later stage that Eq. (h) are applicable in second and fourth quadrants, while Eqs (i) and (j) are applicable in the first and third quadrants of the diagram.

The construction of region of safety is illustrated in Fig. 4.33. The two principal stresses σ_1 and σ_2 are plotted on the X and Y axes respectively. Tensile stresses are considered as positive, while compressive stresses as negative.

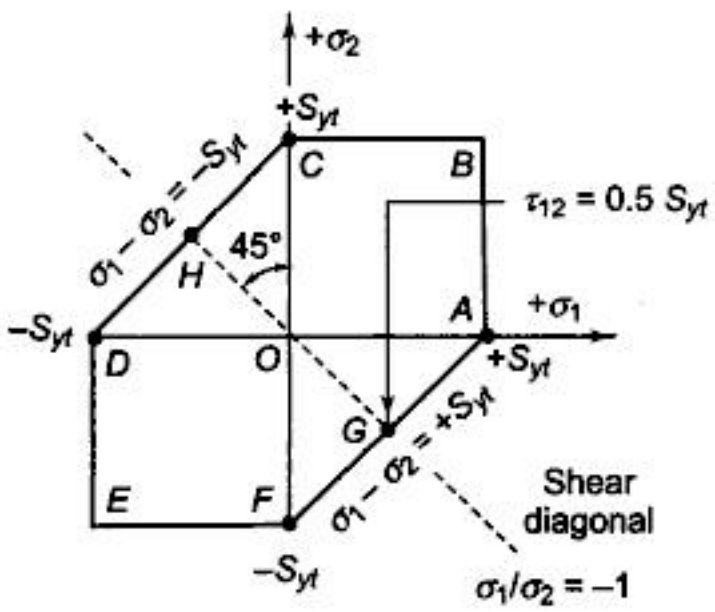

Fig. 4.33 *Boundary for Maximum Shear Stress Theory under Bi-axial Stresses*

It should be noted that,

(i) The equation $(x - y = -a)$ indicates a straight line in the second quadrant with $(-a)$ and $(+a)$ as intercepts on the X and Y axes respectively.

(ii) The equation $(x - y = +a)$ indicates a straight line in the fourth quadrant with $(+a)$ and $(-a)$ as intercepts on the X and Y axes respectively.

The borderline for the region of safety for this theory can be constructed in the following way:

Step 1: In the first quadrant, both (σ_1) and (σ_2) are positive or tensile stresses. The yielding will depend upon where (σ_1) or (σ_2) is greater in magnitude.

Suppose $\sigma_1 > \sigma_2$

The boundary line will be,

$\sigma_1 = +S_{yt}$

A vertical line $\overline{AB}$ is constructed such that

$\sigma_1 = +S_{yt}$.

Suppose $\sigma_2 > \sigma_1$

The boundary line will be,

$\sigma_2 = +S_{yt}$

A horizontal line $\overline{CB}$ is constructed such that $\sigma_2 = +S_{yt}$

Step 2: In the third quadrant, both (σ_1) and (σ_2) are negative or compressive stresses. The yielding will depend upon whether (σ_1) or (σ_2) is greater in magnitude.

Suppose $\sigma_1 > \sigma_2$

The boundary line will be,

$\sigma_1 = -S_{yt}$

A vertical line $\overline{DE}$ is constructed such that

$\sigma_1 = -S_{yt}$

Suppose $\sigma_2 > \sigma_1$

The boundary line will be,

$\sigma_2 = -S_{yt}$

A horizontal line $\overline{EF}$ is constructed such that

$\sigma_2 = -S_{yt}$

Step 3: In the second and fourth quadrants, (σ_1) and (σ_2) are of opposite sign. One stress is tensile while the other is compressive. The yielding will occur when,

$\sigma_1 - \sigma_2 = \pm S_{yt}$

In the second quadrant, line $\overline{DC}$ is constructed such that,

$\sigma_1 - \sigma_2 = -S_{yt}$

It is observed that the intercept of the above line on the X-axis $(\sigma_2 = 0)$ is $(-S_{yt})$ and intercept on the Y-axis $(\sigma_1 = 0)$ is $(+S_{yt})$.

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Step 4: In the fourth quadrant, line $\overline{FA}$ is constructed such that,

$$\sigma_1 - \sigma_2 = + S_{yt}$$

It is observed that the intercept of the above line on the X-axis ($\sigma_2 = 0$) is ($+ S_{yt}$) and intercept on the Y-axis ($\sigma_1 = 0$) is ($- S_{yt}$).

The complete region of safety is the hexagon *ABCDEFA*.

In case of bi-axial stress, if a point with coordinates (σ_1, σ_2) falls outside this hexagon region, then it indicates the failure condition. On the other hand, if the point falls inside the hexagon, the design is safe and the failure may not occur.

Shear Diagonal *Shear diagonal or line of pure shear is the locus of all points, corresponding to pure shear stress.* It will be proved at a later stage (Fig. 4.35) that for pure shear stress,

$$\sigma_1 = - \sigma_2 = \tau_{12}$$

The above equation can be written as,

$$\frac{\sigma_1}{\sigma_2} = -1 = - \tan (45°)$$

A line $\overline{GH}$ is constructed in such a way that it passes through the origin O and makes an angle of $-45°$ with the Y-axis. This line is called shear diagonal or line of pure shear. This line intersects the hexagon at two points G and H. The point of intersection of lines $\overline{FA}$ ($\sigma_1 - \sigma_2 = + S_{yt}$) and $\overline{GH}$ $\left[\frac{\sigma_1}{\sigma_2} = -1\right]$ is G.

Solving two equations simultaneously,

$$\sigma_1 = - \sigma_2 = + S_{yt}/2$$

Since $\sigma_1 = - \sigma_2 = \tau_{12}$

$$\tau_{12} = \frac{1}{2} S_{yt}$$

Since the point G is on the borderline, this is the limiting value for shear stress.

or $$S_{sy} = \frac{1}{2} S_{yt}$$

The maximum shear stress theory of failure is widely used by designers for predicting the failure of components, which are made of ductile materials, like transmission shaft.

4.17 DISTORTION-ENERGY THEORY

This theory was advanced by MT Huber in Poland (1904) and independently by R von Mises in Germany (1913) and H Hencky (1925). It is known as the *Huber von Mises and Hencky's theory. The theory states that the failure of the mechanical component subjected to bi-axial or tri-axial stresses occurs when the strain energy of distortion per unit volume at any point in the component, becomes equal to the strain energy of distortion per unit volume in the standard specimen of tension-test, when yielding starts.*

A unit cube subjected to the three principal stresses σ_1, σ_2 and σ_3 is shown in Fig. 4.34(a). The total strain energy U of the cube is given by,

$$U = \frac{1}{2}\sigma_1\varepsilon_1 + \frac{1}{2}\sigma_2\varepsilon_2 + \frac{1}{2}\sigma_3\varepsilon_3 \qquad \text{(a)}$$

where ε_1, ε_2 and ε_3 are strains in respective directions.

Also, $$\varepsilon_1 = \frac{1}{E}[\sigma_1 - \mu(\sigma_2 + \sigma_3)]$$

$$\varepsilon_2 = \frac{1}{E}[\sigma_2 - \mu(\sigma_1 + \sigma_3)]$$

$$\varepsilon_3 = \frac{1}{E}[\sigma_3 - \mu(\sigma_1 + \sigma_2)] \qquad \text{(b)}$$

Substituting the above expressions in Eq. (a),

$$U = \frac{1}{2E}[(\sigma_1^2 + \sigma_2^2 + \sigma_3^2) - 2\mu(\sigma_1\sigma_2 + \sigma_2\sigma_3 + \sigma_3\sigma_1)] \qquad \text{(c)}$$

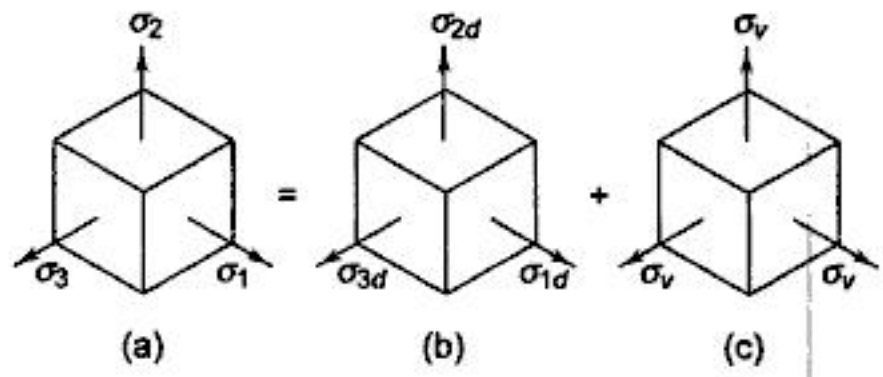

Fig. 4.34 *(a) Element with Tri-axial Stresses (b) Stress Components due to Distortion of Element (c) Stress Components due to Change of Volume*

The total strain energy U is resolved into two components—first U_v corresponding to the change of volume with no distortion of the element and the

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second U_d corresponding to the distortion of the element with no change of volume. Therefore,

$$U = U_v + U_d \qquad \text{(d)}$$

The corresponding stresses are also resolved into two components as shown in Fig. 4.34 (b) and (c). From the figure,

$$\sigma_1 = \sigma_{1d} + \sigma_v$$
$$\sigma_2 = \sigma_{2d} = \sigma_v$$
$$\sigma_3 = \sigma_{3d} = \sigma_v \qquad \text{(e)}$$

The components σ_{1d}, σ_{2d} and σ_{3d} cause distortion of the cube, while the component σ_v results in volumetric change. Since the components σ_{1d}, σ_{2d} and σ_{3d} do not change the volume of the cube,

$$\varepsilon_{1d} + \varepsilon_{2d} + \varepsilon_{3d} = 0 \qquad \text{(f)}$$

Also,

$$\varepsilon_{1d} = \frac{1}{E}[\sigma_{1d} - \mu(\sigma_{2d} + \sigma_{3d})]$$
$$\varepsilon_{2d} = \frac{1}{E}[\sigma_{2d} - \mu(\sigma_{1d} + \sigma_{3d})]$$
$$\varepsilon_{3d} = \frac{1}{E}[\sigma_{3d} - \mu(\sigma_{1d} + \sigma_{2d})] \qquad \text{(g)}$$

Substituting Eq. (g) in Eq. (f),

$$(1 - 2\mu)(\sigma_{1d} + \sigma_{2d} + \sigma_{3d}) = 0$$

Since $(1 - 2\mu) \neq 0$

$$\therefore \quad (\sigma_{1d} + \sigma_{2d} + \sigma_{3d}) = 0 \qquad \text{(h)}$$

From Eq. (h) in Eq. (e),

$$\sigma_v = \frac{1}{3}(\sigma_1 + \sigma_2 + \sigma_3) \qquad \text{(j)}$$

The strain energy U_v corresponding to the change of volume for the cube is given by,

$$U_v = 3\left[\frac{\sigma_v \varepsilon_v}{2}\right] \qquad \text{(k)}$$

Also $$\varepsilon_v = \frac{1}{E}[\sigma_v - \mu(\sigma_v + \sigma_v)]$$

or $$\varepsilon_v = \frac{(1 - 2\mu)\sigma_v}{E} \qquad \text{(l)}$$

From expressions (k) and (l),

$$U_v = \frac{3(1 - 2\mu)\sigma_v^2}{2E} \qquad \text{(m)}$$

Substituting expression (j) in the Eq. (m),

$$U_v = \frac{(1 - 2\mu)(\sigma_1 + \sigma_2 + \sigma_3)^2}{6E} \qquad \text{(n)}$$

From expressions (c) and (n),

$$U_d = U - U_v$$

or $$U_d = \left(\frac{1 + \mu}{6E}\right)[(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2] \qquad (4.40)$$

In simple tension test, when the specimen starts yielding,

$$\sigma_1 = S_{yt} \quad \text{and} \quad \sigma_2 = \sigma_3 = 0$$

Therefore, $$U_d = \left(\frac{1 + \mu}{3E}\right) S_{yt}^2 \qquad (4.41)$$

From Eqs (4.40) and (4.41), the criterion of failure for the distortion energy theory is expressed as

$$2S_{yt}^2 = \left[(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2\right]$$

or $$S_{yt} = \sqrt{\frac{1}{2}\left[(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2\right]} \qquad (4.42)$$

Considering the factor of safety,

$$\frac{S_{yt}}{(fs)} = \sqrt{\frac{1}{2}\left[(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2\right]} \qquad (4.43)$$

For bi-axial stresses ($\sigma_3 = 0$),

$$\frac{S_{yt}}{(fs)} = \sqrt{(\sigma_1^2 - \sigma_1\sigma_2 + \sigma_2^2)} \qquad (4.44)$$

A component subjected to pure shear stresses and the corresponding Mohr's circle diagram is shown in Fig. 4.35.

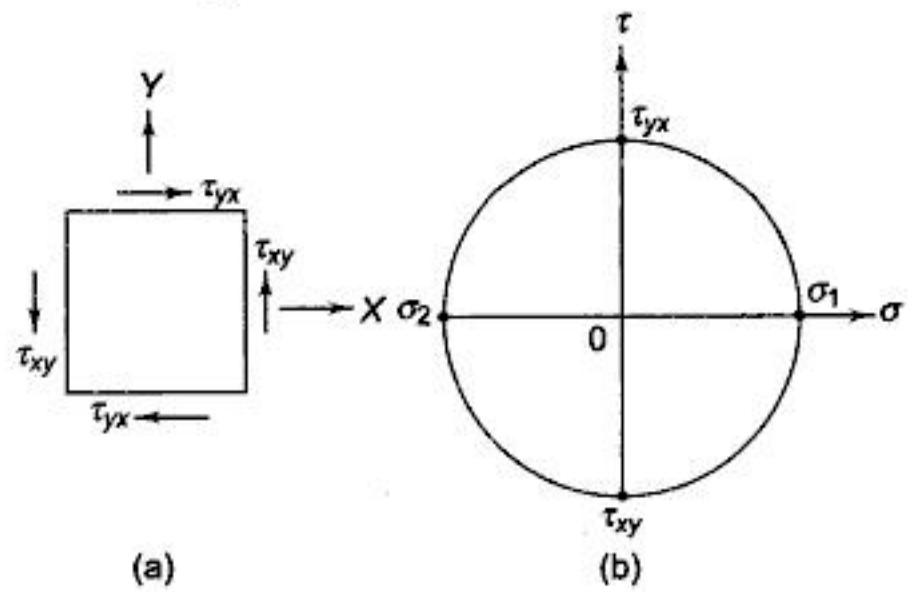

Fig. 4.35 *(a) Element subjected to Pure Shear Stresses (b) Mohr's Circle for Shear Stresses*

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From the figure,

$$\sigma_1 = -\sigma_2 = \tau_{xy} \quad \text{and} \quad \sigma_3 = 0$$

Substituting these values in Eq. (4.42),

$$S_{yt} = \sqrt{3}\,\tau_{xy}$$

Replacing (τ_{xy}) by S_{sy},

$$S_{sy} = \frac{S_{yt}}{\sqrt{3}} = 0.577\,S_{yt} \qquad (4.45)$$

Therefore, according to the distortion-energy theory, the yield strength in shear is 0.577 times the yield strength in tension.

Region of Safety The construction of the region of safety is illustrated in Fig. 4.36. The two principal stresses σ_1 and σ_2 are plotted on the X and Y axes respectively. Tensile stresses are considered as positive, while compressive stresses as negative.

It should be noted that,

$$x^2 - xy + y^2 = a^2$$

is an equation of an ellipse whose semi-major axis is $(\sqrt{2}a)$ and semi-minor axis is $(\sqrt{2/3}\,a)$.

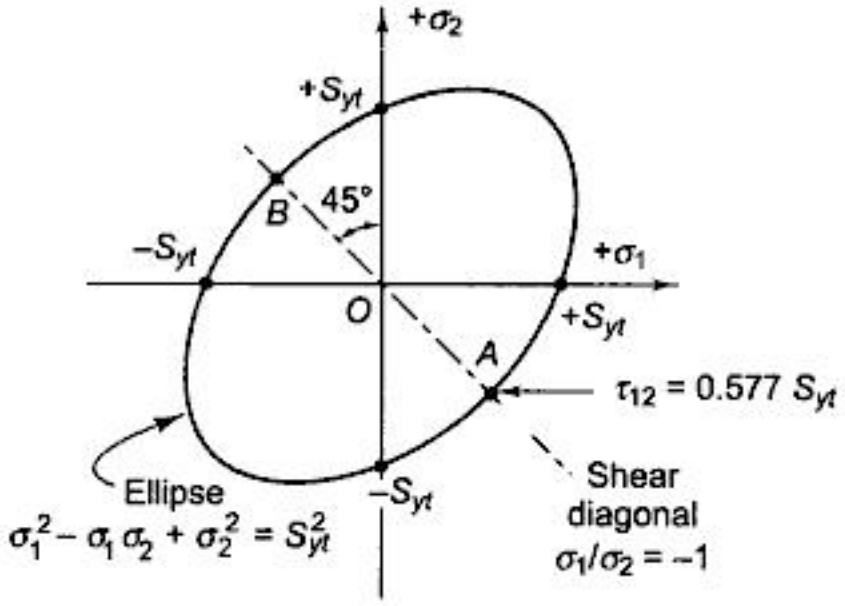

Fig. 4.36 *Boundary for Distortion Energy Theory under Bi-axial Stresses*

For bi-axial stresses,

$$\sigma_3 = 0$$

Substituting this value in Eq. (4.42),

$$\sigma_1^2 - \sigma_1\sigma_2 + \sigma_2^2 = S_{yt}^2$$

The above equation indicates an ellipse whose semi-major axis is $(\sqrt{2}\,S_{yt})$ and semi-minor axis is $\left[\sqrt{\frac{2}{3}}S_{yt}\right]$.

If a point with coordinates (σ_1, σ_2) falls outside this ellipse, then it indicates the failure condition. On the other hand, if the point falls inside the ellipse, the design is safe and the failure may not occur.

Shear Diagonal As mentioned in the previous section, shear diagonal or line of pure shear is the locus of all points, corresponding to pure shear stress. The condition for the line of shear is,

$$\sigma_1 = -\sigma_2 = \tau_{12}$$

The above equation can be written as

$$\frac{\sigma_1}{\sigma_2} = -1 = -\tan(45°)$$

A line $\overline{AB}$ is constructed in such a way that it passes through the origin O and makes an angle of $-45°$ with the Y-axis. This line is called shear diagonal or line of pure shear. This line intersects the ellipse at two points A and B.

A is the point of intersection of the ellipse and the line AB. The coordinates of the point A are obtained by solving the following two equations simultaneously,

$$\sigma_1^2 - \sigma_1\sigma_2 + \sigma_2^2 = S_{yt}^2$$

$$\sigma_1/\sigma_2 = -1$$

Solving two equations simultaneously,

$$\sigma_1 = -\sigma_2 = +\frac{1}{\sqrt{3}}S_{yt}$$

Since $\sigma_1 = -\sigma_2 = \tau_{12}$

$$\tau_{12} = \frac{1}{\sqrt{3}}S_{yt} = 0.577S_{yt}$$

Since the point A is on the borderline, this is the limiting value for shear stress.

or $$S_{sy} = 0.577\,S_{yt}$$

Experiments have shown that the distortion-energy theory is in better agreement for predicting the failure of a ductile component than any other theory of failure.

4.18 SELECTION AND USE OF FAILURE THEORIES

The plots of three theories of failure on the σ_1, σ_2 coordinate system are shown in Fig. 4.37. While selecting theories of failure, the following points should be noted.

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(i) Ductile materials typically have the same tensile strength and compressive strength. Also, yielding is the criterion of failure in ductile materials. In maximum shear stress theory and distortion energy theory, it is assumed that the yield strength in tension (S_{yt}) is equal to the yield strength in compression (S_{yc}). Also, the criterion of failure is yielding. Therefore, maximum shear stress theory and distortion energy theory are used for ductile materials.

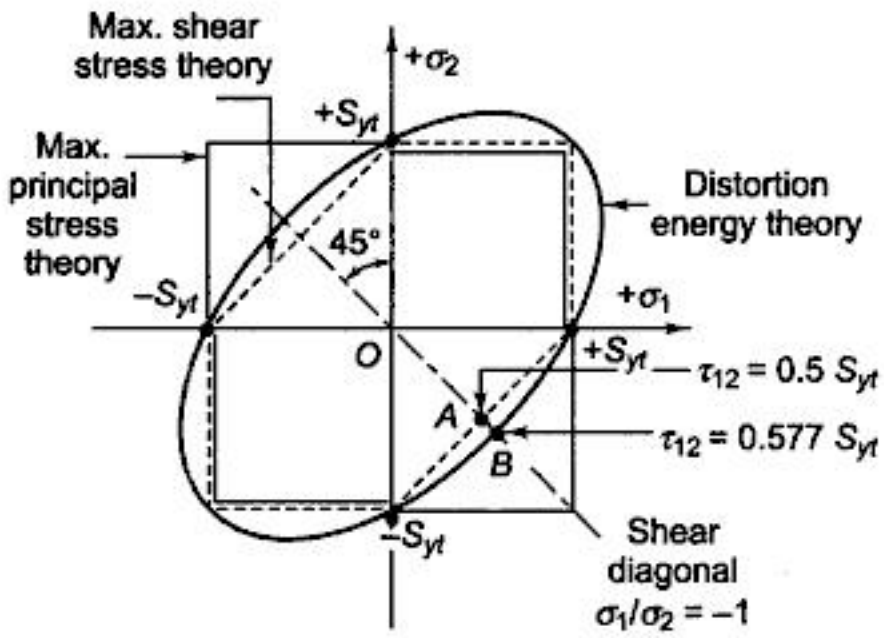

Fig. 4.37 *Comparison of Theories of Failure*

(ii) Distortion energy theory predicts yielding with precise accuracy in all four quadrants. The design calculations involved in this theory are slightly complicated as compared with those of maximum shear stress theory.

(iii) The hexagonal diagram of maximum shear stress theory is inside the ellipse of distortion energy theory. Therefore, maximum shear stress theory gives results on the conservative side. On the other hand, distortion energy theory is slightly liberal.

(iv) The graph of maximum principal stress theory is the same as that of maximum shear stress theory in the first and third quadrants. However, the graph of maximum principal stress theory is outside the ellipse of distortion energy theory in the second and fourth quadrants. Thus, it would be dangerous to apply maximum principal stress theory in these regions, since it might predict safety, when in fact no safety exists.

(v) Maximum shear stress theory is used for ductile materials, if dimensions need not be held too close and a generous factor of safety is used. The calculations involved in this theory are easier than those of distortion energy theory.

(vi) Distortion energy theory is used when the factor of safety is to be held in close limits and the cause of failure of the component is being investigated. This theory predicts the failure most accurately.

(vii) The compressive strength of brittle materials is much higher than their tensile strength. Therefore, the failure criterion should show a difference in tensile and compressive strength. On this account, maximum principal stress theory is used for brittle materials. Also, brittle materials do not yield and they fail by fracture.

To summarise, the maximum principal stress theory is the proper choice for brittle materials. For ductile materials, the choice of theory depends on the level of accuracy required and the degree of computational difficulty the designer is ready to face. For ductile materials, the most accurate way to design is to use distortion energy theory of failure and the easiest way to design is to apply maximum shear stress theory

Example 4.11 *A cantilever beam of rectangular cross-section is used to support a pulley as shown in Fig. 4.38 (a). The tension in the wire rope is 5 kN. The beam is made of cast iron FG 200 and the factor of safety is 2.5. The ratio of depth to width of the cross-section is 2. Determine the dimensions of the cross-section of the beam.*

Solution

Given $P = 5$ kN $\quad S_{ut} = 200$ N/mm^2
$(fs) = 2.5 \quad d/w = 2$

Step I *Calculation of permissible bending stress*

$$\sigma_b = \frac{S_{ut}}{(fs)} = \frac{200}{2.5} = 80 \text{ N/mm}^2$$

Step II *Calculation of bending moments*

The forces acting on the beam are shown in Fig. 4.38(b). Referring to the figure,

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$(M_b)_{\text{at B}} = 5000 \times 500 = 2500 \times 10^3$ N-mm

$(M_b)_{\text{at A}} = 5000 \times 500 + 5000 \times 1500$

$= 10000 \times 10^3$ N-mm

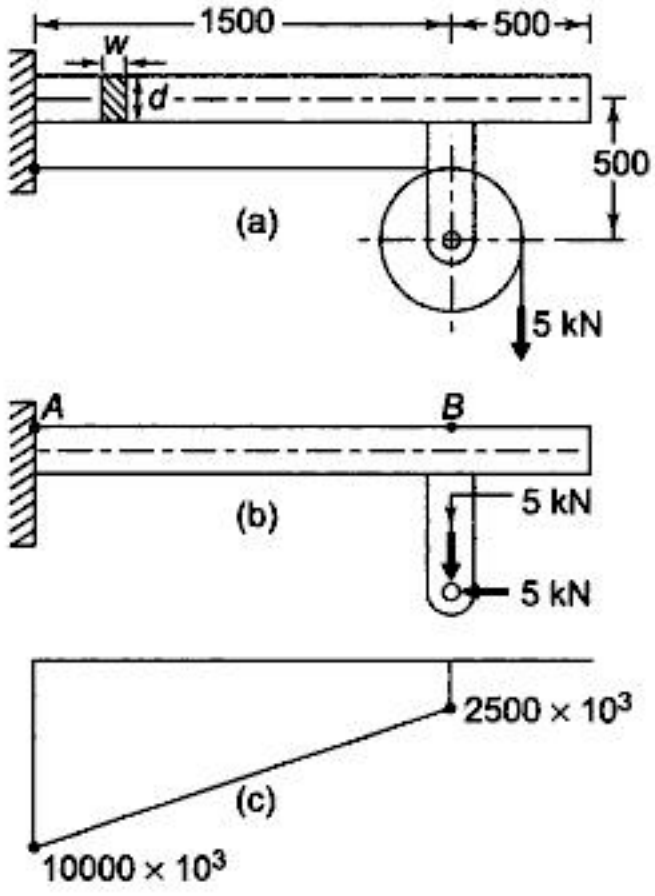

Fig. 4.38

Step III *Calculation of dimensions of cross-section*

The bending moment diagram is shown in Fig. 4.38(c). The cross-section at A is subjected to maximum bending stress. For this cross-section,

$$y = \frac{d}{2} = w \qquad I = \frac{1}{12}[(w)(2w)^3] = \frac{2}{3}w^4 \text{ mm}^4$$

$$\sigma_b = \frac{M_b y}{I} \quad \text{or} \quad 80 = \frac{(10000 \times 10^3)(w)}{\left(\frac{2}{3}w^4\right)}$$

Therefore,

$w = 57.24$ mm or 60 mm $\qquad d = 2w = 120$ mm

Example 4.12 *A wall bracket with a rectangular cross-section is shown in Fig. 4.39. The depth of the cross-section is twice of the width. The force P acting on the bracket at 60^0 to the vertical is 5 kN. The material of the bracket is grey cast iron FG 200 and the factor of safety is 3.5. Determine the dimensions of the cross-section of the bracket. Assume maximum normal stress theory of failure.*

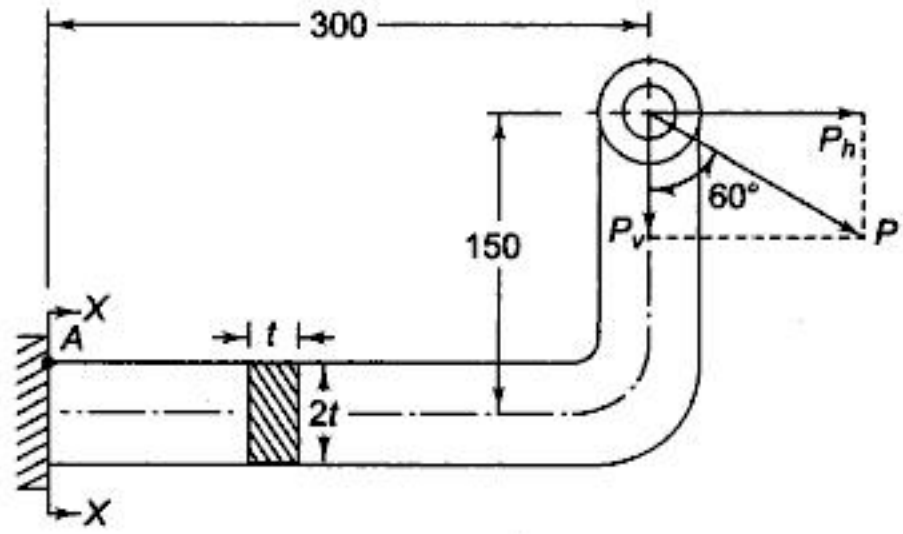

Fig. 4.39 *Wall Bracket*

Solution

Given $P = 5$ kN

$S_{ut} = 200$ N/ mm^2 $\quad (fs) = 3.5 \quad d/w = 2$

Step I *Calculation of permissible stress*

$$\sigma_{\max} = \frac{S_{ut}}{(fs)} = \frac{200}{3.5} = 57.14 \text{ N/mm}^2 \qquad \text{(i)}$$

Step II *Calculation of direct and bending tensile stresses*

The stress is maximum at the point A in the section XX. The point is subjected to combined bending and direct tensile stresses. The force P is resolved into two components—horizontal component P_h and vertical component P_v.

$P_h = P \sin 60° = 5000 \sin 60° = 4330.13$ N

$P_v = P \cos 60° = 5000 \cos 60° = 2500$ N

The bending moment at the section XX is given by

$$M_b = P_h \times 150 + P_v \times 300$$
$$= 4330.13 \times 150 + 2500 \times 300$$
$$= 1399.52 \times 10^3 \text{ N-mm}$$

$$\sigma_b = \frac{M_b y}{I}$$

$$= \frac{(1399.52 \times 10^3)(t)}{\left[\frac{1}{12}(t)(2t)^3\right]} = \frac{2099.28 \times 10^3}{t^3} \text{ N/mm}^2$$

The direct tensile stress due to component P_h is given by,

$$\sigma_t = \frac{P_h}{A} = \frac{4330.13}{2t^2} = \frac{2165.07}{t^2} \text{ N/mm}^2$$

The vertical component P_v induces shear stress at the section XX. It is however small and neglected.

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Step III *Calculation of dimensions of cross-section*
The resultant tensile stress $\sigma_{max.}$ at the point A is given by,

$$\sigma_{max.} = \sigma_b + \sigma_t = \frac{2099.28 \times 10^3}{t^3} + \frac{2165.07}{t^2} \quad \text{(ii)}$$

Equating (i) and (ii),

$$\frac{2099.28 \times 10^3}{t^3} + \frac{2165.07}{t^2} = 57.14$$

or

$$t^3 - 37.89\, t - 36739.24 = 0$$

Solving the above cubic equation by trial and error method,

$$t = 33.65 \text{ mm} \cong 35 \text{ mm}$$

The dimensions of the cross-section are 35×70 mm

Example 4.13 *The shaft of an overhang crank subjected to a force P of 1 kN is shown in Fig. 4.40(a). The shaft is made of plain carbon steel 45C8 and the tensile yield strength is 380 N/mm². The factor of safety is 2. Determine the diameter of the shaft using the maximum shear stress theory.*

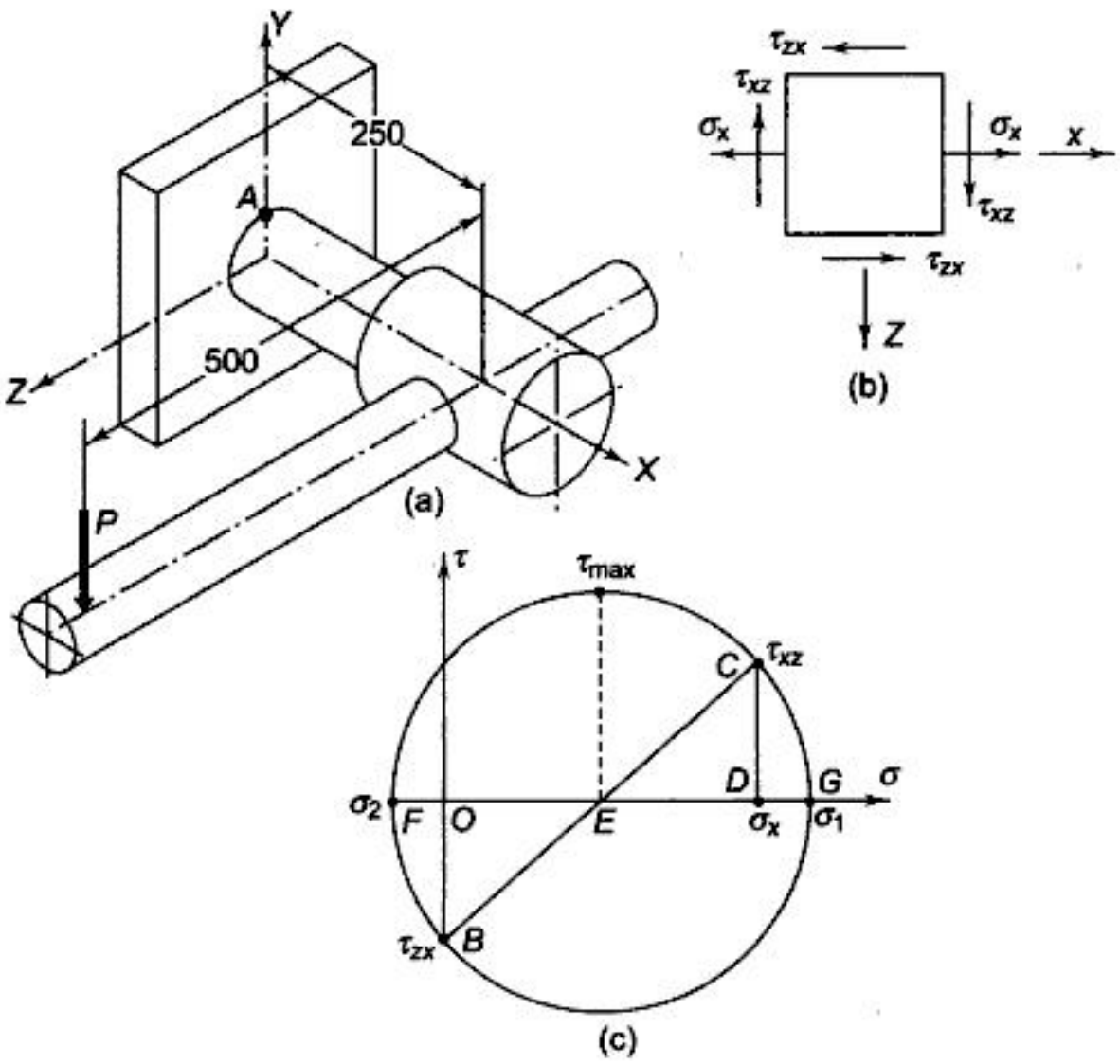

Fig. 4.40

Solution

Given $P = 1$ kN $\quad S_{yt} = 380$ N/mm² $\quad (fs) = 2$

Step I *Calculation of permissible shear stress*
According to maximum shear stress theory,

$$S_{sy} = 0.5\, S_{yt} = 0.5\,(380) = 190 \text{ N/mm}^2$$

The permissible shear stress is given by,

$$\tau_{max.} = \frac{S_{sy}}{(fs)} = \frac{190}{2} = 95 \text{ N/mm}^2 \quad \text{(i)}$$

Step II *Calculation of bending and torsional shear stresses*
The stresses are critical at the point A, which is subjected to combined bending and torsional moments.
At the point A,

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$M_b = P \times (250) = (1000)\ (250) = 250 \times 10^3$ N-mm
$M_t = P \times (500) = (1000)\ (500) = 500 \times 10^3$ N-mm

$$\sigma_b = \frac{M_b y}{I} = \frac{(250 \times 10^3)(d/2)}{(\pi d^4/64)}$$

$$= \left(\frac{2546.48 \times 10^3}{d^3}\right) \text{N/mm}^2$$

$$\tau = \frac{M_t r}{J} = \frac{(500 \times 10^3)(d/2)}{(\pi d^4/32)}$$

$$= \left(\frac{2546.48 \times 10^3}{d^3}\right) \text{N/mm}^2$$

Step III *Calculation of maximum shear stress*

The stresses at point A and corresponding Mohr's circle are shown in Fig. 4.40(b) and (c) respectively. In these figures,

$$\sigma_x = \sigma_b = \left(\frac{2546.48 \times 10^3}{d^3}\right) \text{N/mm}^2 \qquad \sigma_z = 0$$

$$\tau = \tau_{xz} = \tau_{zx} = \left(\frac{2546.48 \times 10^3}{d^3}\right) \text{N/mm}^2$$

From Mohr's circle,

$$\tau_{\max.} = \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + (\tau_{xz})^2}$$

$$= \left[\sqrt{\left(\frac{2546.48}{2d^3}\right)^2 + \left(\frac{2546.48}{d^3}\right)^2}\right] \times 10^3$$

$$= \frac{2847.05 \times 10^3}{d^3} \qquad \text{(ii)}$$

Step IV *Calculation of shaft diameter*

Equating (i) and (ii),

$$\frac{2847.05 \times 10^3}{d^3} = 95 \qquad \therefore\ d = 31.06 \text{ mm}$$

Example 4.14 *The dimensions of an overhang crank are given in Fig. 4.41. The force P acting at the crankpin is 1 kN. The crank is made of steel 30C8 (S_{yt} = 400 N/mm²) and the factor of safety is 2. Using maximum shear stress theory of failure, determine the diameter d at the section - XX.*

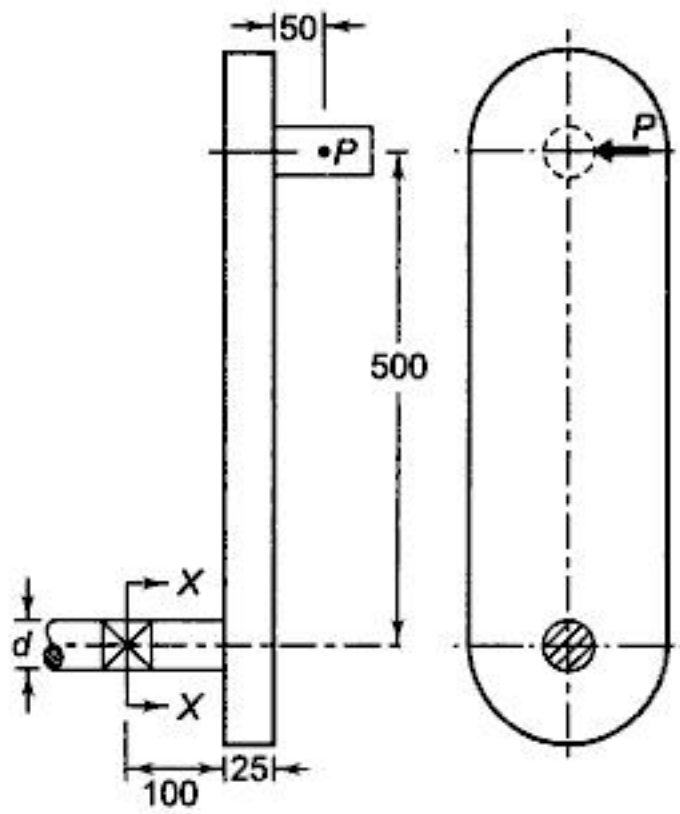

Fig. 4.41 *Overhang Crank*

Solution

Given $P = 1$ kN $\quad S_{yt} = 400$ N/mm² $\quad (fs) = 2$

Step I *Calculation of permissible shear stress*

According to maximum shear stress theory,

$$S_{sy} = 0.5\ S_{yt} = 0.5\ (400) = 200 \text{ N/mm}^2$$

The permissible shear stress is given by,

$$\tau_{\max.} = \frac{S_{sy}}{(fs)} = \frac{200}{2} = 100 \text{ N/mm}^2 \qquad \text{(i)}$$

Step II *Calculation of bending and torsional shear stresses*

The section of the crankpin at XX is subjected to combined bending and torsional moments. At the section XX,

$M_b = 1000 \times (50 + 25 + 100) = 175 \times 10^3$ N-mm
$M_t = 1000 \times 500 = 500 \times 10^3$ N-mm

$$\sigma_x = \sigma_b = \frac{M_b y}{I} = \frac{(175 \times 10^3)(d/2)}{(\pi d^4/64)}$$

$$= \left(\frac{1782.54 \times 10^3}{d^3}\right) \text{N/mm}^2$$

$$\sigma_y = 0$$

$$\tau = \frac{M_t r}{J} = \frac{(500 \times 10^3)(d/2)}{(\pi d^4/32)}$$

$$= \left(\frac{2546.48 \times 10^3}{d^3}\right) \text{N/mm}^2$$

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Step III *Calculation of maximum shear stress*
The problem is similar to the previous one and the maximum shear stress is given by,

$$\tau_{max.} = \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + (\tau)^2}$$

$$= \sqrt{\left(\frac{1782.54 \times 10^3}{2d^3}\right)^2 + \left(\frac{2546.48 \times 10^3}{d^3}\right)^2}$$

$$= \frac{2697.95 \times 10^3}{d^3} \text{ N/mm}^2 \qquad \text{(ii)}$$

Step IV *Calculation of diameter at section-XX*
Equating (i) and (ii),

$$\frac{2697.95 \times 10^3}{d^3} = 100 \quad \therefore \quad d = 29.99 \text{ or } 30 \text{ mm}$$

4.19 LEVERS

A lever is defined as a mechanical device in the form of a rigid bar pivoted about the fulcrum to multiply or transfer the force. The construction of a simple lever is shown in Fig. 4.42. F is the force produced by the lever and P is the effort required to produce that force. The force F is often called '*load*'. The perpendicular distance of the line of action of any force from the fulcrum is called the *arm* of the lever. Therefore l_1 and l_2 are the effort arm and load arm respectively. Taking moment of forces about the fulcrum,

$$F \times l_2 = P \times l_1$$

or

$$\frac{F}{P} = \frac{l_1}{l_2} \qquad \text{(a)}$$

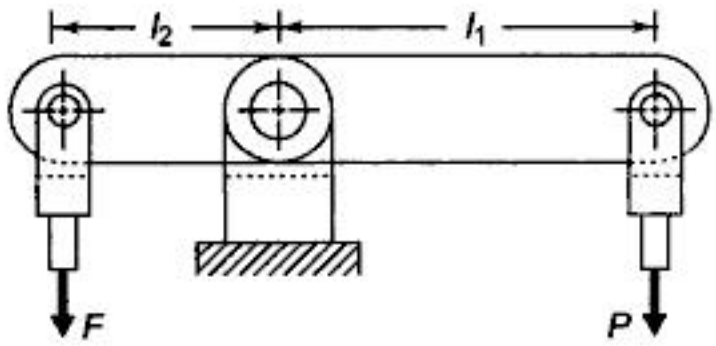

Fig. 4.42 *Construction of Lever*

The ratio of load to effort, i.e., (F/P) is called the '*mechanical advantage*' of the lever. The ratio of the effort arm to the load arm, i.e., (l_1/l_2) is called the '*leverage*'. Therefore, mechanical advantage is equal to the leverage. It is seen by Eq. (a), that a large force can be exerted by a small effort by increasing leverage, i.e., increasing l_1 and reducing l_2. In many applications, it is not possible to increase effort arm l_1 due to space restrictions. In such applications, compound levers are used to obtain more leverage.

There are three types of levers, based on the relative positions of the effort point, the load point and the fulcrum as illustrated in Fig. 4.43. They are as follows:

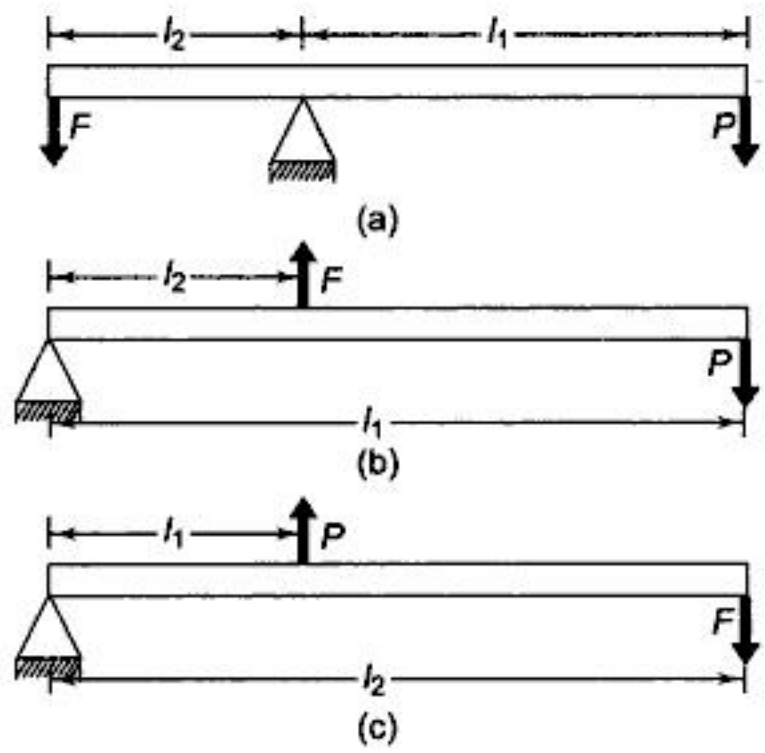

Fig. 4.43 *Types of Lever*

(i) In the '*first*' type of lever, the fulcrum is located between the load and the effort, as shown in Fig. 4.43(a). In this case, the effort arm can be kept less than the load arm or equal to the load arm or more than the load arm. Accordingly, the mechanical advantages will vary in the following way:
When $l_1 < l_2$,
mechanical advantage < 1
When $l_1 = l_2$,
mechanical advantage = 1
When $l_1 > l_2$,
mechanical advantage > 1
Usually, the effort arm is kept more than the load arm to get mechanical advantage. This type of lever is used in applications like the rocker arm for the overhead valves of internal combustion engine, bell crank levers

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in railway signal mechanisms and levers of hand pumps.

(ii) In the '*second*' type of lever, the load is located between the fulcrum and the effort, as shown in Fig. 4.43(b). In this case, the effort arm is always more than the load arm and the mechanical advantage is more than 1. This type of lever is used in lever-loaded safety valves mounted on the boilers.

(iii) In the '*third*' type of lever, the effort is located between the load and the fulcrum, as shown in Fig. 4.43(c). In this case, the load arm is always greater than the effort arm and the mechanical advantage is less than 1. This type of lever is not recommended in engineering applications. A picking fork is an example of this type of lever.

Levers have wide applications, ranging from simple nutcrackers and paper punching machines to complex lever systems in scales and weighing machines.

4.20 DESIGN OF LEVERS

Lever design is easy compared to design of other machine elements. The length of the lever is decided on the basis of leverage required to exert a given load F by means of an effort P. The cross-section of the lever is designed on the basis of bending stresses. The design of a lever consists of the following steps:

Step 1: Force Analysis

In any application, the load or the force F to be exerted by the lever is given. The effort required to produce this force is calculated by taking moments about the fulcrum. Therefore,

$$F \times l_2 = P \times l_1$$

or,

$$P = F\left(\frac{l_2}{l_1}\right) \qquad (4.46)$$

The free body diagram of forces acting on the 'first' type of lever is shown in Fig. 4.44. R is the reaction at the fulcrum pin. Since the sum of vertical forces acting on the lever must be equal to zero,

$$R = F + P \qquad (4.47)$$

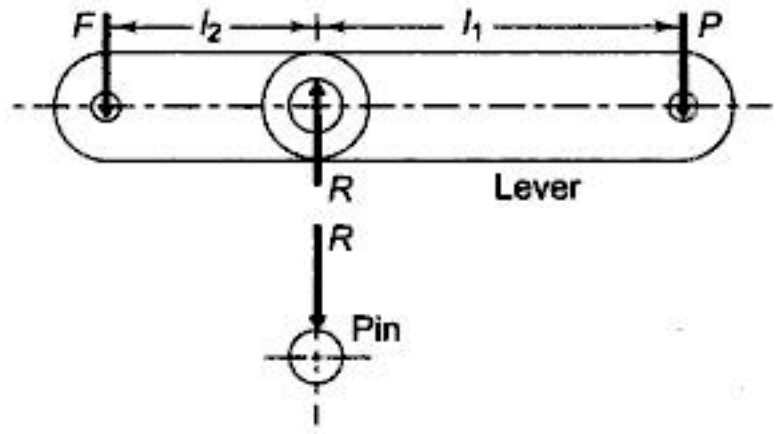

Fig. 4.44 *Free body Diagram of Forces acting on First type of Lever*

The free body diagram of forces acting on the 'second' type of lever is shown in Fig 4.45. In this case, the load and the effort act in opposite directions. Considering equilibrium of forces in a vertical direction,

$$F = R + P$$

or,

$$R = F - P \qquad (4.48)$$

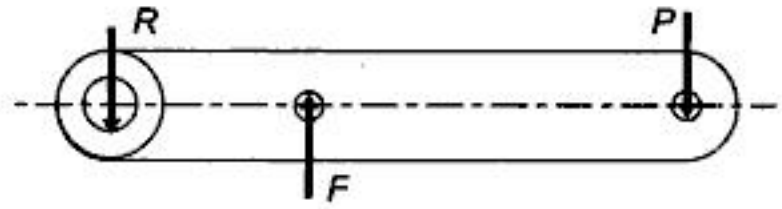

Fig. 4.45 *Free body Diagram of Forces acting on Second type of Lever*

In the above two cases, the forces are assumed to be parallel. Sometimes, the forces F and P act along lines that are inclined to one another as shown in Fig. 4.46. In such cases, l_1 is the perpendicular distance from the fulcrum to the line of action of the force P. Similarly, l_2 is the

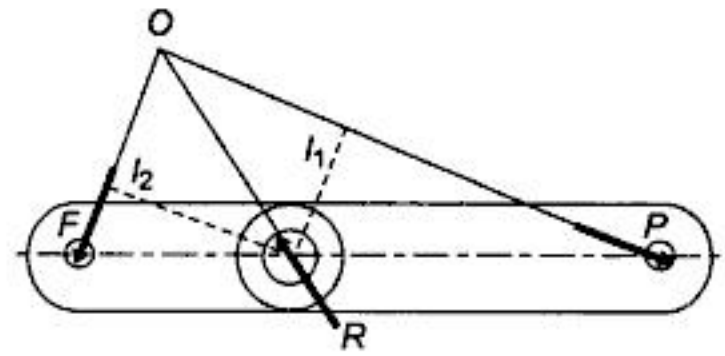

Fig. 4.46

perpendicular distance from the fulcrum to the line of action of the force F. The following rules from statics apply to the reaction R at the fulcrum:

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(i) The magnitude of the reaction R is equal to the resultant of the load F and the effort P. It is determined by the *parallelogram law* of forces.

(ii) The line of action of the reaction R passes through the intersection of P and F, i.e., the point O in Fig. 4.46 and also through the fulcrum.

Figure 4.47 illustrates a bell-crank lever with the arms that are inclined at angle θ with one another. The load F and the effort P act at right angles to their respective arms. The reaction R at the fulcrum is given by

$$R = \sqrt{F^2 + P^2 - 2FP\cos\theta} \qquad (4.49)$$

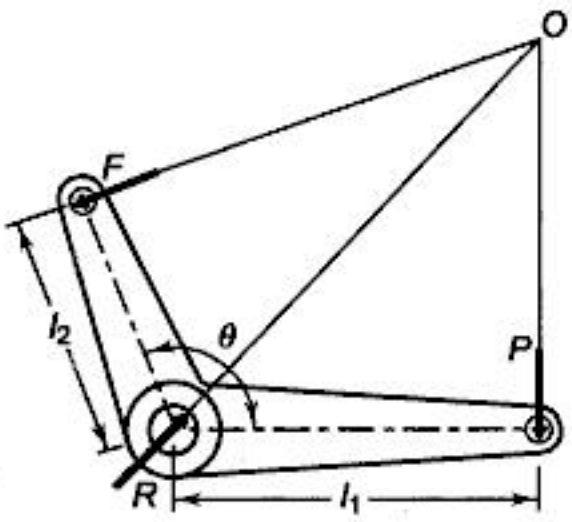

Fig. 4.47

When the arms of the bell-crank lever are at right angles to one another,

$$\theta = 90° \text{ and } \cos\theta = 0$$

Therefore,

$$R = \sqrt{F^2 + P^2} \qquad (4.50)$$

Step 2: Design of Lever Arm

When the forces acting on the lever are determined, the next step in lever design is to find out the dimensions of the cross-section of the lever. The cross-section of the lever is subjected to bending moment. In case of a two-arm lever, as shown in Fig. 4.48(a), the bending moment is zero at the point of application of P or F and maximum at the boss of the lever. The cross-section at which the bending moment is maximum can be determined by constructing a bending-moment diagram. In Fig. 4.48(b), the bending moment is maximum at section XX and it is given by,

$$M_b = P\,(l_1 - d_1)$$

The cross-section of the lever can be rectangular, elliptical or I-section.

For a rectangular cross-section,

$$I = \frac{bd^3}{12} \quad \text{and} \quad y = \frac{d}{2}$$

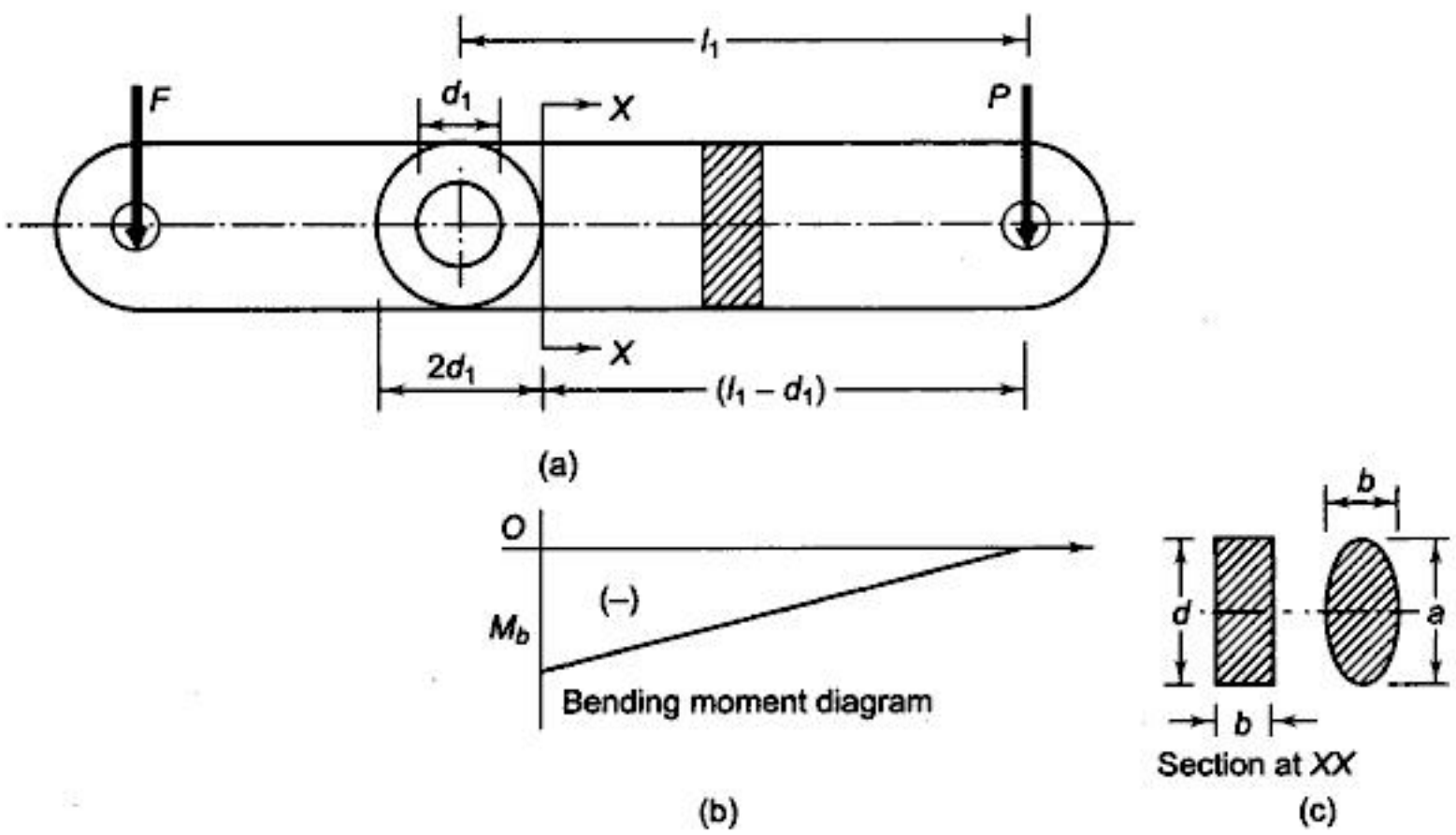

Fig. 4.48

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where b is the distance parallel to the neutral axis, and d is the distance perpendicular to the neutral axis. The dimension d is usually taken as twice of b.

or, $$d = 2b$$

For an elliptical cross-section,

$$I = \frac{\pi b a^3}{64} \quad \text{and} \quad y = \frac{a}{2}$$

where a and b are major and minor axes of the section. Usually, the major axis is taken as twice the minor axis.

or, $$a = 2b$$

Using the above mentioned proportions, the dimensions of the cross-section of the lever can be determined by,

$$\sigma_b = \frac{M_b y}{I}$$

Figure 4.48(b) shows the variation of bending moment. It varies from a maximum value M_b at the section XX to zero at the point of application of P. Therefore, the cross-section of the arm is usually tapered from the boss of the fulcrum to the end.

Step 3: Design of Fulcrum Pin

The fulcrum pin is subjected to reaction R as shown in Fig. 4.49. The forces acting on the boss of lever and the pin are equal and opposite. The dimensions of the pin, viz., diameter d_1 and length l_1 in lever boss are determined by bearing consideration and then checked for shear consideration. There is relative motion between the pin and the boss of lever and bearing pressure becomes the design criterion. The projected area of the pin is $(d_1 \times l_1)$. Therefore,

$$R = p\,(d_1 \times l_1) \qquad (4.51)$$

where p is the the permissible bearing pressure.

For the fulcrum pin, the ratio of length to diameter (l_1/d_1) is usually taken from 1 to 2. The outside diameter of the boss in the lever is taken as twice of the diameter of the pin, i.e., $(2d_1)$. A phosphor bronze bush, usually 3 mm thick, is fitted inside the boss to reduce the friction. The permissible bearing pressure for a phosphor bronze bush is 5 to 10 N/mm^2. A lubricant is provided between the pin and the bush to reduce the friction.

It can be observed that expressions for bearing pressure and compressive or crushing stress are same. Rearranging Eq. (4.51), the bearing pressure is given by,

$$p = \frac{R}{(d_1 \times l_1)} \qquad \text{(a)}$$

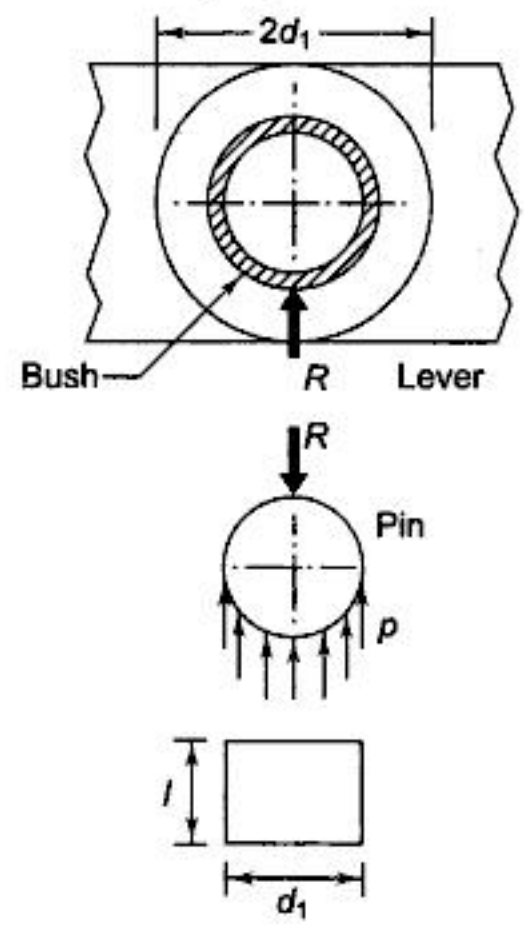

Fig. 4.49

There is a similar example of the pin in a knuckle joint illustrated in Fig. 4.21. For this pin, the compressive stress is given by,

$$\sigma_c = \frac{\text{force}}{\text{projected area}}$$

or

$$\sigma_c = \frac{P}{(d \times l)} \qquad \text{(b)}$$

Although the expressions (a) and (b) are same, there is a basic difference between bearing pressure and crushing or compressive stress. The bearing pressure is considered when there is relative motion between two surfaces such as surfaces of the pin and the bushing. On the other hand, crushing stress is considered when there is no relative motion between the surfaces under consideration. The bearing pressure is always low such as 10 N/mm^2, while the magnitude of compressive stress is high such as

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150 N/mm^2. A rotating shaft in the bearing, a fulcrum pin of an oscillating lever, a power screw rotating inside the nut are examples where bearing pressure is the design consideration. The contact area between a cotter and spigot end, and cotter and socket end; between knuckle pin and eye or knuckle pin and fork are the examples where crushing stress is the criterion of design.

Example 4.15 *A lever-loaded safety valve is mounted on the boiler to blow off at a pressure of 1.5 MPa gauge. The effective diameter of the opening of the valve is 50 mm. The distance between the fulcrum and the dead weights on the lever is 1000 mm. The distance between the fulcrum and the pin connecting the valve spindle to the lever is 100 mm. The lever and the pin are made of plain carbon steel 30C8 (S_{yt} = 400 N/mm^2) and the factor of safety is 5. The permissible bearing pressure at the pins in the lever is 25 N/mm^2. The lever has a rectangular cross-section and the ratio of width to thickness is 3:1. Design a suitable lever for the safety valve.*

Solution

Given $S_{yt} = 400$ N/mm^2 $(fs) = 5$
For valve, $d = 50$ mm $p = 1.5$ MPa
For lever, $l_1 = 1000$ mm $l_2 = 100$ mm $d/b = 3$
For pin, $p = 25$ N/mm^2

The construction of the lever-loaded safety valve is shown in Fig. 4.50. It is mounted on steam boilers to limit the maximum steam pressure. When the pressure inside the boiler exceeds this limiting value, the valve automatically opens due to excess of steam pressure and steam blows out through the valve. Consequently, the steam pressure inside the boiler is reduced.

Step I *Calculation of permissible stresses for lever and pin*

$$\sigma_t = \frac{S_{yt}}{(fs)} = \frac{400}{5} = 80 \text{ N/mm}^2$$

$$\tau = \frac{S_{sy}}{(fs)} = \frac{0.5S_{yt}}{(fs)} = \frac{0.5(400)}{5} = 40 \text{ N/mm}^2$$

Step II *Calculation of forces acting on lever*

The valve is held tight on the valve seat against the upward steam force F by the dead weights P attached at the end of the lever. The distance l_1 and the dead weights P are adjusted in such a way, that when the steam pressure inside the boiler reaches the limiting value, the moment $(F \times l_2)$ overcomes the moment $(P \times l_1)$. As a result, the valve opens and steam blows out until the pressure falls to the required limiting value and then the valve is automatically closed.

The maximum steam load F, at which the valve blows off is given by,

$$F = \frac{\pi}{4} d^2 p = \frac{\pi}{4} (50)^2 (1.5) = 2945.24 \text{ N} \qquad \text{(a)}$$

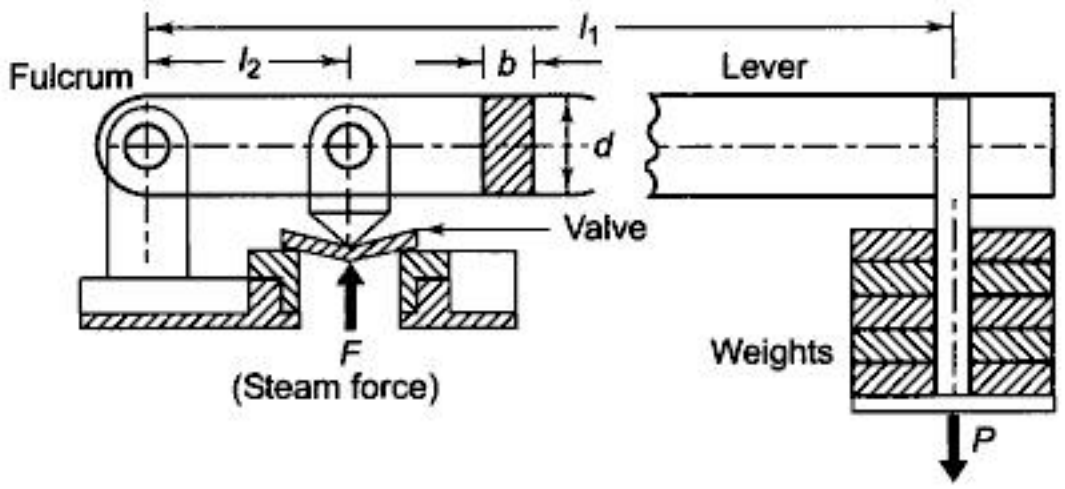

Fig. 4.50 *Lever Loaded Safety Valve*

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Taking moment of forces F and P about the fulcrum,

$$F \times l_2 = P \times l_1 \quad \text{or} \quad 2945.24 \times 100 = P \times 1000$$

$$\therefore \quad P = 294.52 \text{ N} \qquad \text{(b)}$$

The forces acting on the lever are shown in Fig. 4.51(a). Considering equilibrium of vertical forces,

$$F = R + P$$

$$\text{or} \quad R = F - P = 2945.24 - 294.52 = 2650.72 \text{ N} \qquad \text{(c)}$$

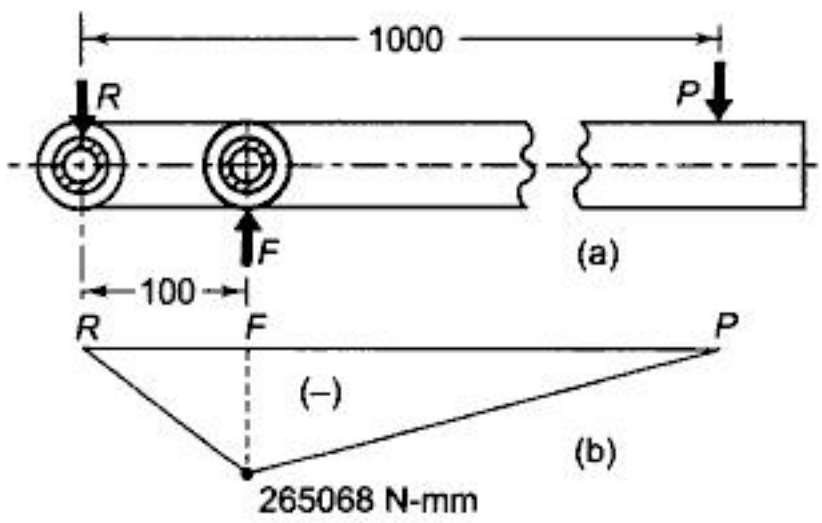

Fig. 4.51 *Bending Moment Diagram of Lever*

Step III *Diameter and length of pin*

From (a), (b) and (c), the pin at the point of application of the force F is subjected to maximum force and as such, it is to be designed from bearing consideration. Suppose, d_1 and l_1 are the diameter and the length of the pin at F and assume,

$$l_1 = d_1$$

From Eq. (4.51),

$$F = p\,(d_1 \times l_1) \quad \text{or} \quad 2945.24 = 25\,(d_1 \times d_1)$$

$$\therefore \quad d_1 = 10.85 \text{ or } 12 \text{ mm}$$

$$l_1 = d_1 = 12 \text{ mm} \qquad \text{(i)}$$

The pin is subjected to double shear stress, which is given by,

$$\tau = \frac{F}{2\left[\frac{\pi}{4} d_1^2\right]} = \frac{2945.24}{2\left[\frac{\pi}{4}(12)^2\right]} = 13.02 \text{ N/mm}^2$$

$$\therefore \quad \tau < 40 \text{ N/mm}^2$$

The force on the fulcrum pin (R) is comparatively less than the force acting on the spindle pin (F). Therefore, the dimensions d_1 and l_1 of the pin at the fulcrum will be slightly less. However, we will assume both pins of the same diameter and length to facilitate interchangeability of parts and variety reduction.

Step IV *Width and thickness of lever*

A gunmetal bush of 2-mm thickness is press fitted at both pin holes to reduce friction. Therefore, the inside diameter of the boss will be $(d_1 + 2 \times 2)$ or $(12 + 2 \times 2)$ or 16 mm. The outside diameter of the boss is kept twice of the inside diameter, i.e., 32 mm.

The bending moment diagram for the lever is shown in Fig. 4.51(b). The bending moment is maximum at the valve spindle axis. It is given by,

$$M_b = \text{P}\,(1000 - 100) = 294.52\,(1000 - 100)$$
$$= 265\,068 \text{ N-mm}$$

For a lever,

$$d = 3b$$

$$\sigma_b = \frac{M_b y}{I} \qquad 80 = \frac{(265\,068)\,(1.5b)}{\left[\frac{1}{12} b\,(3b)^3\right]}$$

$$\therefore \quad b = 13.02 \text{ or } 15 \text{ mm} \qquad d = 3b = 45 \text{ mm}$$

The lever becomes weak due to the pinhole at the valve spindle axis and it is necessary to check bending stresses at this critical section. The cross-section of the lever at the valve spindle axis is shown in Fig. 4.52. In this case, the length of the pin is increased from 12 mm to 20 mm to get practical proportions for the boss. For this cross-section,

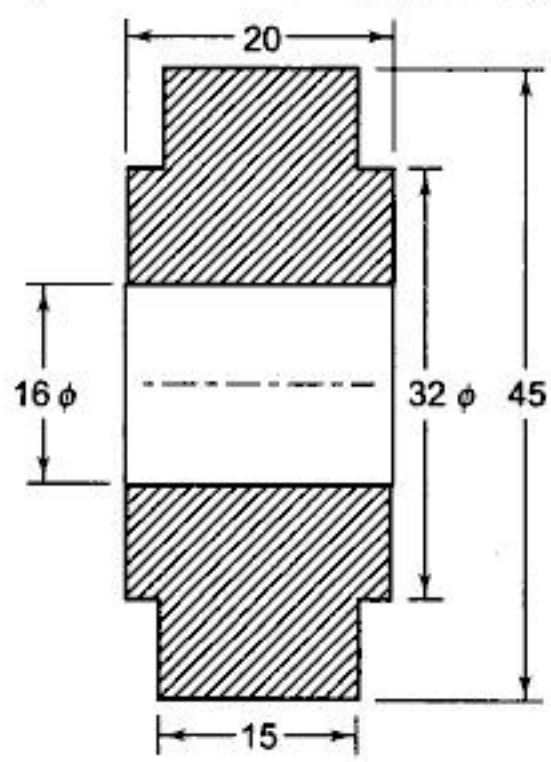

Fig. 4.52 *Cross-section of Lever*

$$M_b = 265\,068 \text{ N-mm} \quad y = 22.5 \text{ mm}$$

$$I = \frac{1}{12}\left[15(45)^3 + 5(32)^3 - 20(16)^3\right]$$

$$= 120\,732.92 \text{ mm}^4$$

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l_1 = length of the pin (mm)

p = permissible bearing pressure (N/mm^2)

Substituting values in Eq. (a),

$$2684.47 = 10\,(d_1 \times 1.25\,d_1)$$

$\therefore$ $d_1 = 14.65$ or 15 mm

$l_1 = 1.25\,d_1 = 1.25(15) = 18.75$ or 20 mm

The pin is subjected to double shear stress, which is given by,

$$\tau = \frac{R}{2\left[\frac{\pi}{4}d_1^2\right]} = \frac{2684.47}{2\left[\frac{\pi}{4}(15)^2\right]} = 7.60 \text{ N/mm}^2$$

$\therefore$ $\tau < 25$ N/mm^2

Step IV *Diameter of link*

The link is subjected to tensile stress as shown in Fig. 4.58(c). Therefore,

$$\sigma_t = \frac{P_1}{\left(\frac{\pi}{4}d^2\right)} \quad \text{or} \quad 50 = \frac{2684.47}{\left(\frac{\pi}{4}d^2\right)}$$

$\therefore$ $d = 8.27$ or 10 mm

Step V *Dimensions of lever*

As shown in Fig. 4.59, the lever is subjected to bending moment. The maximum bending moment occurs at C, which is given by,

$$M_b = P \times 900 = 268.45 \times 900 = 241\,605 \text{ N-mm}$$

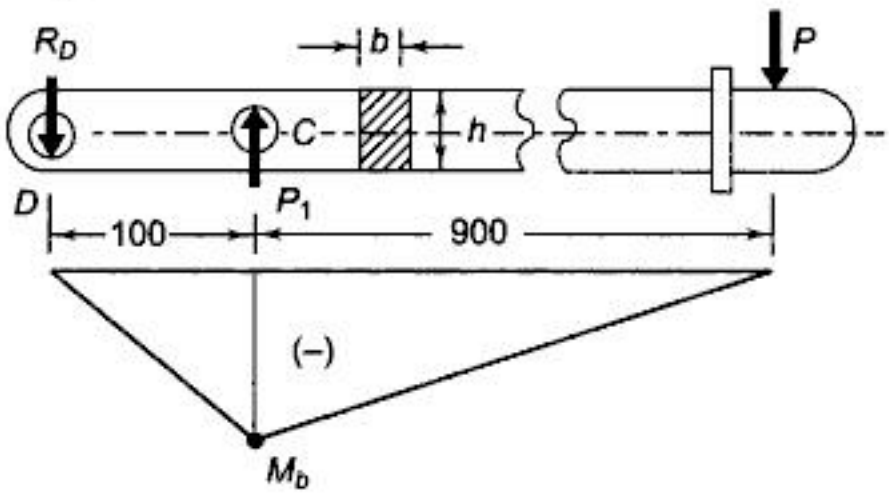

Fig. 4.59 *Bending Moment Diagram for Lever*

Since,

$$h = 2b \quad I = \frac{bh^3}{12} = \frac{b(2b)^3}{12} = \left(\frac{2b^4}{3}\right) \text{mm}^4$$

$$y = \frac{h}{2} = b \text{ mm} \qquad \sigma_b = \frac{M_b y}{I}$$

Substituting,

$$50 = \frac{(241\,605)(b)}{\left(\frac{2b^4}{3}\right)}$$

$\therefore$ $b = 19.35$ or 20 mm $\quad h = 2b = 2(20) = 40$ mm

The lever becomes weak due to the pinhole at C and it is necessary to check bending stresses at this critical cross-section, which is shown in Fig. 4.60.

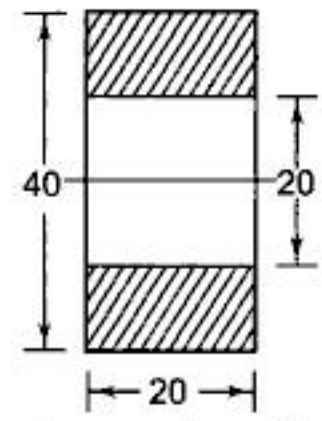

Fig. 4.60 *Cross-section of Lever at Pin-C*

The diameter of the pin is 15 mm, while the length is 20 mm. It is assumed that a gunmetal bush of 2.5 mm thickness is fitted in the hole to reduce friction and wear. Therefore, inner diameter of the boss will be $(15 + 2 \times 2.5)$ or 20 mm. The outside diameter of the boss is kept twice of the inner diameter, i.e., 40 mm. Therefore,

$$I = \frac{1}{12}[20(40)^3 - 20(20)^3] = 93\,333.33 \text{ mm}^4$$

$y = 20$ mm

$$\sigma_b = \frac{M_b y}{I} = \frac{(241\,605)(20)}{(93\,333.33)} = 51.77 \text{ N/mm}^2$$

There is slight difference between bending stress (51.77 N/mm^2) and permissible stress (50 N/mm^2). Since the difference is small, it is neglected and the dimensions of the boss of the lever are kept unchanged.

4.21 FRACTURE MECHANICS

There was a series of catastrophic failures of pressure vessels, ships and aircrafts in 10 years between 1940 and 1950. These accidents focused the attention of design engineers to the existence

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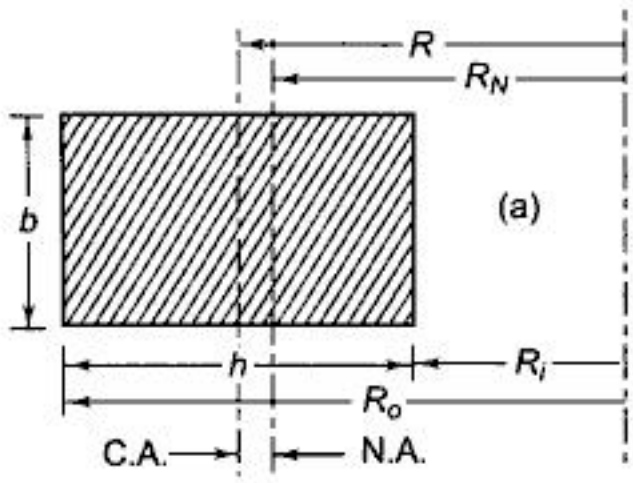

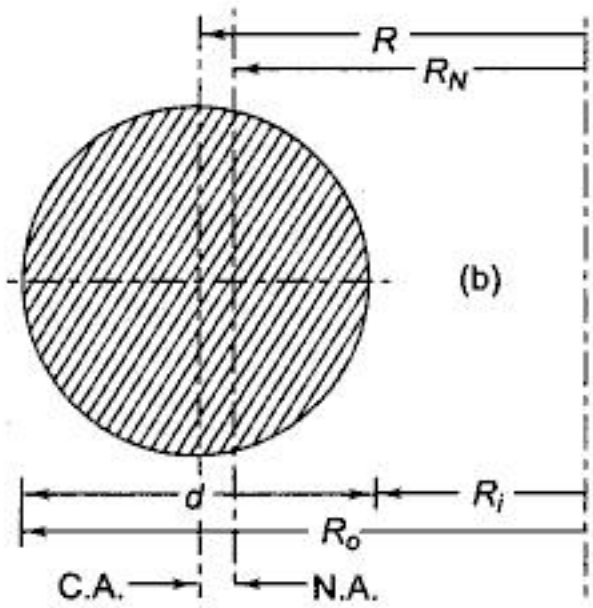

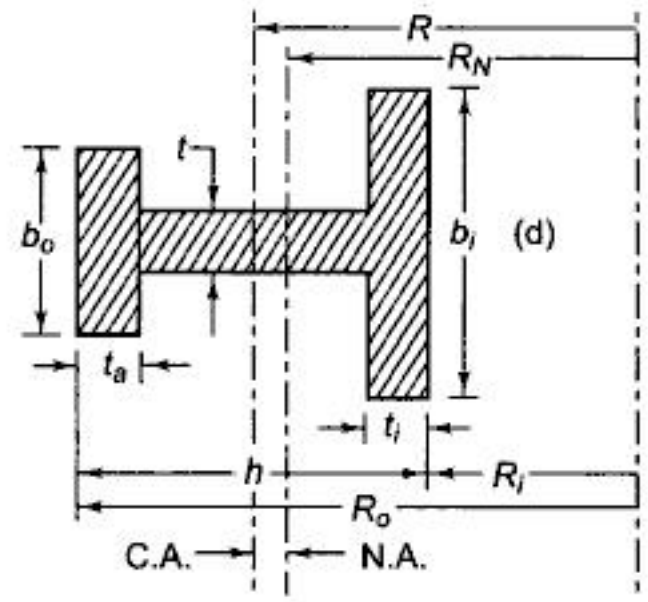

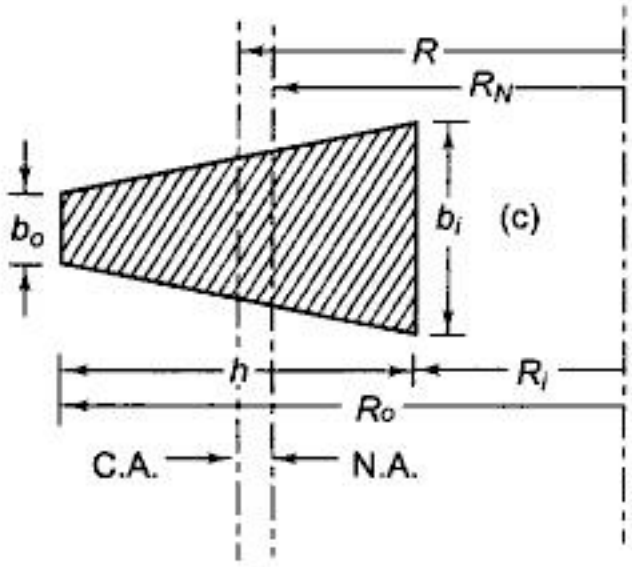

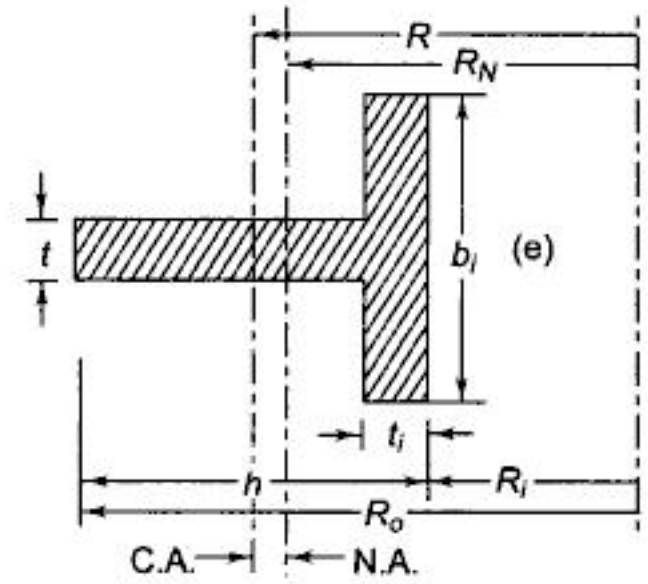

Fig. 4.65 *Nomenclature of Cross-section for Curved Beams*

$$R = R_i + \frac{\frac{1}{2}th^2 + \frac{1}{2}t_i^2(b_i - t)}{th + t_i(b_i - t)} \qquad (4.67)$$

Example 4.19 *A crane hook having an approximate trapezoidal cross-section is shown in Fig. 4.66. It is made of plain carbon steel 45C8 (S_{yt} = 380 N/mm^2) and the factor of safety is 3.5. Determine the load carrying capacity of the hook.*

Solution

Given $S_{yt} = 380$ N/mm^2 $(fs) = 3.5$
$b_i = 90$ mm $b_o = 30$ mm $h = 120$ mm
$R_o = 170$ mm $R_i = 50$ mm

Step 1 *Calculation of permissible tensile stress*

$$\sigma_{max.} = \frac{S_{yt}}{(fs)} = \frac{380}{3.5} = 108.57 \text{ N/mm}^2$$

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4.5 The principal stresses induced at a point in a machine component made of steel 50C4 ($S_{yt} = 460$ N/mm^2) are as follows:
$\sigma_1 = 200$N/mm^2 $\sigma_2 = 150$ N/mm^2 $\sigma_3 = 0$
Calculate the factor of safety by (i) the maximum shear stress theory, and (ii) the distortion energy theory.

[(i) 2.3 (ii) 2.55]

4.6 The stresses induced at a critical point in a machine component made of steel 45C8 (S_{yt} = 380 N/mm^2) are as follows:
$\sigma_x = 100$ N/mm^2 $\sigma_y = 40$ N/mm^2
$\tau_{xy} = 80$ N/mm^2
Calculate the factor of safety by (i) the maximum normal stress theory, (ii) the maximum shear stress theory, and (iii) the distortion energy theory.

[(i) 2.44 (ii) 2.22 (iii) 2.32]

4.7 A link of *S*-shape made of a round steel bar is shown in Fig. 4.73. It is made of plain carbon steel 45C8 (S_{yt} = 380 N/mm^2) and the factor of safety is 4.5. Calculate the dimensions of the link.

(d = 23.38 mm)

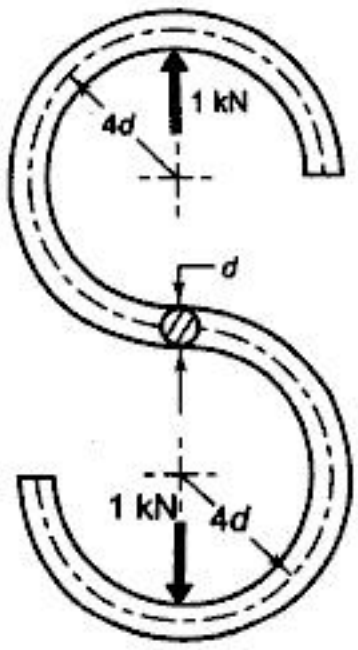

Fig. 4.73

4.8 The frame of a 100 kN capacity press is shown in Fig. 4.74. It is made of grey cast iron FG 300 and the factor of safety is 2.5. Determine the dimensions of the cross-section at *XX*.

(t = 26.62 mm)

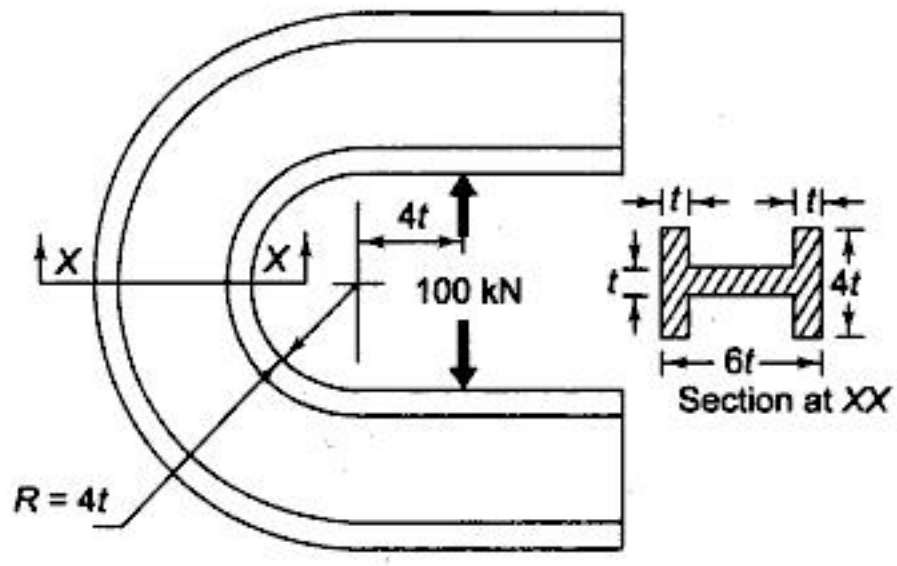

Fig. 4.74

4.9 A bell crank lever is subjected to a force of 7.5 kN at the short arm end. The lengths of the short and long arms are 100 and 500 mm respectively. The arms are at right angles to each other. The lever and the pins are made of steel FeE 300 (S_{yt} = 300 N/mm^2) and the factor of safety is 5. The permissible bearing pressure on the pin is 10 N/mm^2. The lever has a rectangular cross-section and the ratio of width to thickness is 4:1. The length to diameter ratio of the fulcrum pin is 1.5:1. Calculate:

(i) the diameter and the length of the fulcrum pin
(ii) the shear stress in the pin
(iii) the dimensions of the boss of the lever at the fulcrum pin
(iv) the dimensions of the cross-section of the lever

Assume that the arm of the bending moment on the lever extends up to the axis of the fulcrum.

[(i) 22.58 and 33.87 mm (ii) 9.55 N/mm^2 (iii) D_i = 23 mm D_0 = 46 mm length = 34 mm, (iv) 16.74 × 66.94 mm]

4.10 A bracket, made of steel FeE 200 (S_{yt} = 200 N/mm^2) and subjected to a force of 5 kN acting at an angle of 30° to the vertical, is shown in Fig. 4.75. The factor of safety is 4. Determine the dimensions of the cross-section of the bracket.

[t = 33.5 mm]

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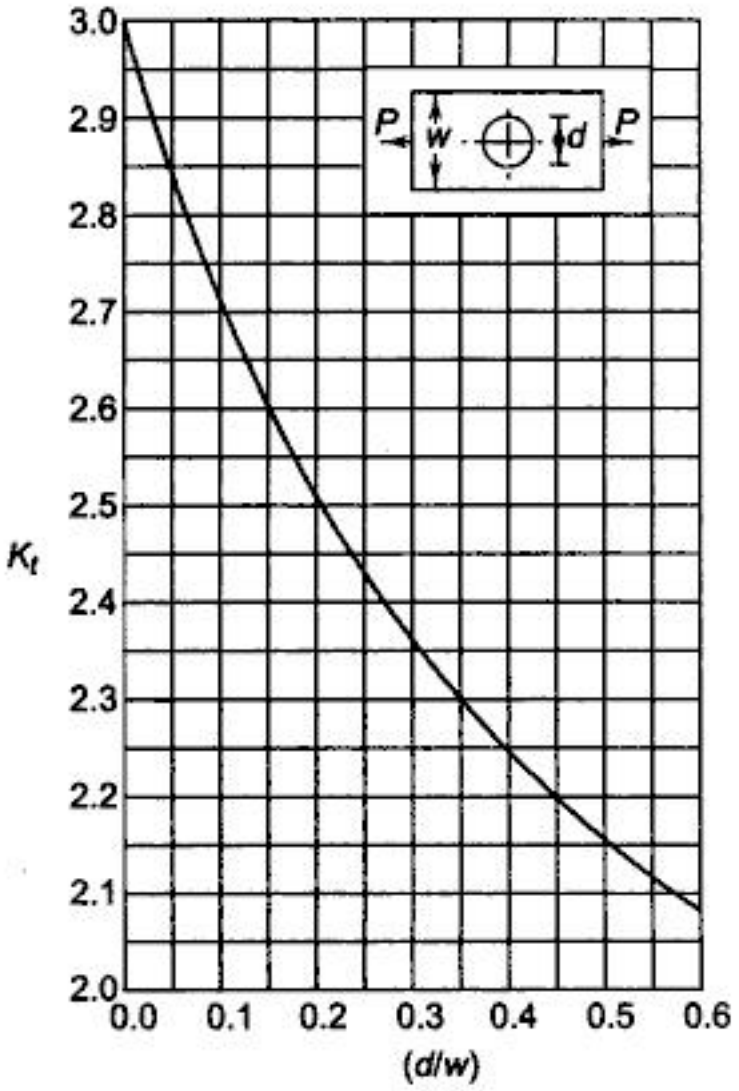

Fig. 5.2 *Stress Concentration Factor (Rectangular Plate with Transverse Hole in Tension or Compression)*

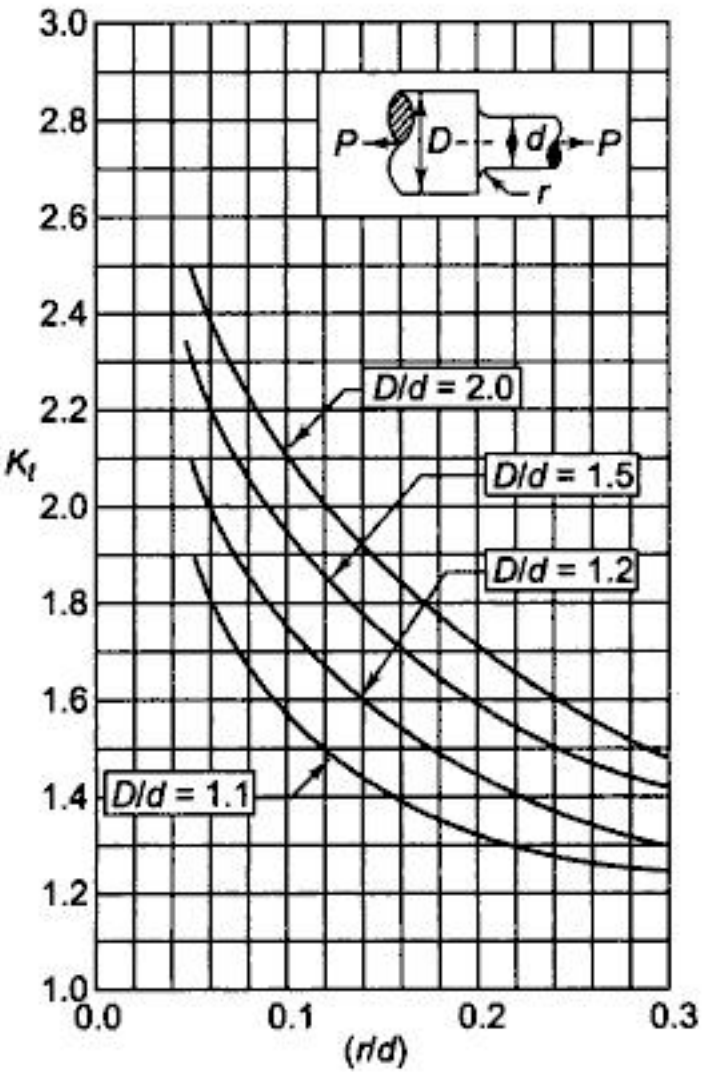

Fig. 5.4 *Stress Concentration Factor (Round Shaft with Shoulder Fillet in Tension)*

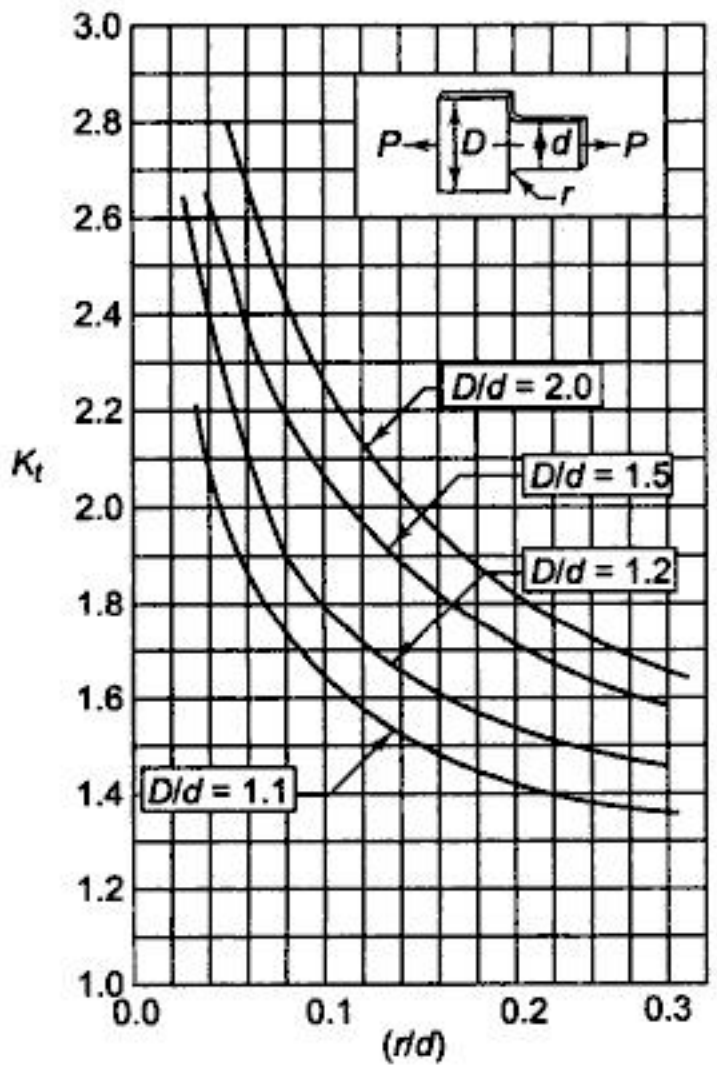

Fig. 5.3 *Stress Concentration Factor (Flat Plate with Shoulder Fillet in Tension or Compression)*

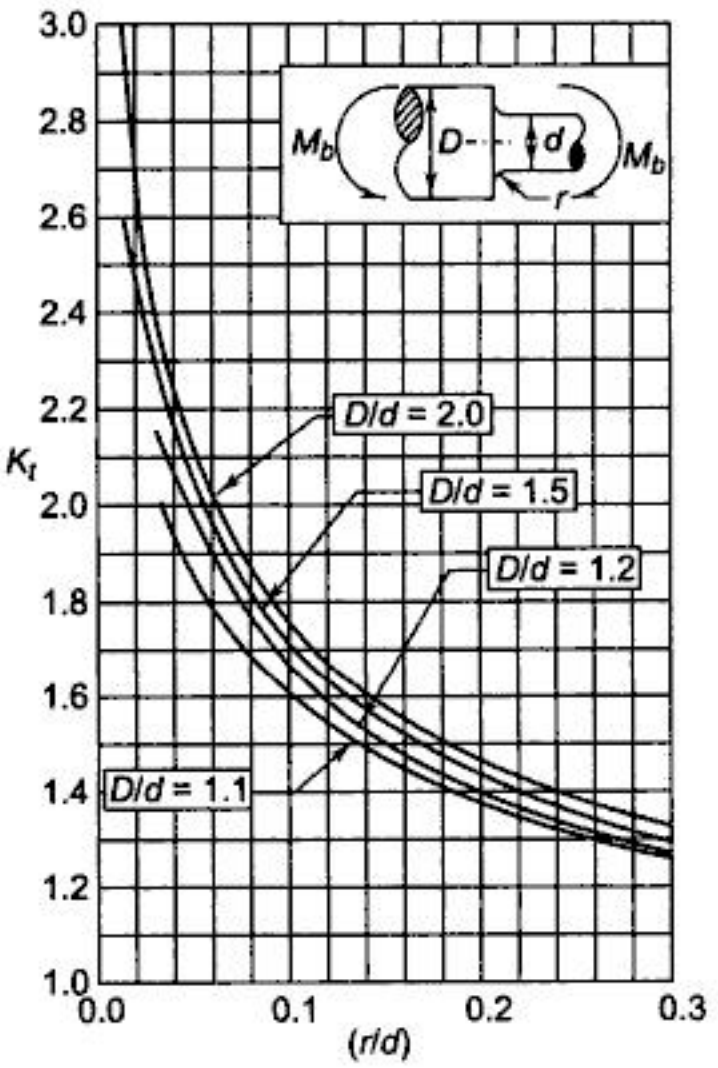

Fig. 5.5 *Stress Concentration Factor (Round Shaf with Shoulder Fillet in Bending)*

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The fillet radius can be increased by undercutting the shoulder as illustrated in Fig. 5.10(c). A notch results in stress concentration. Surprisingly, cutting an additional notch is an effective way to reduce stress concentration. This is illustrated in Fig. 5.10(d).

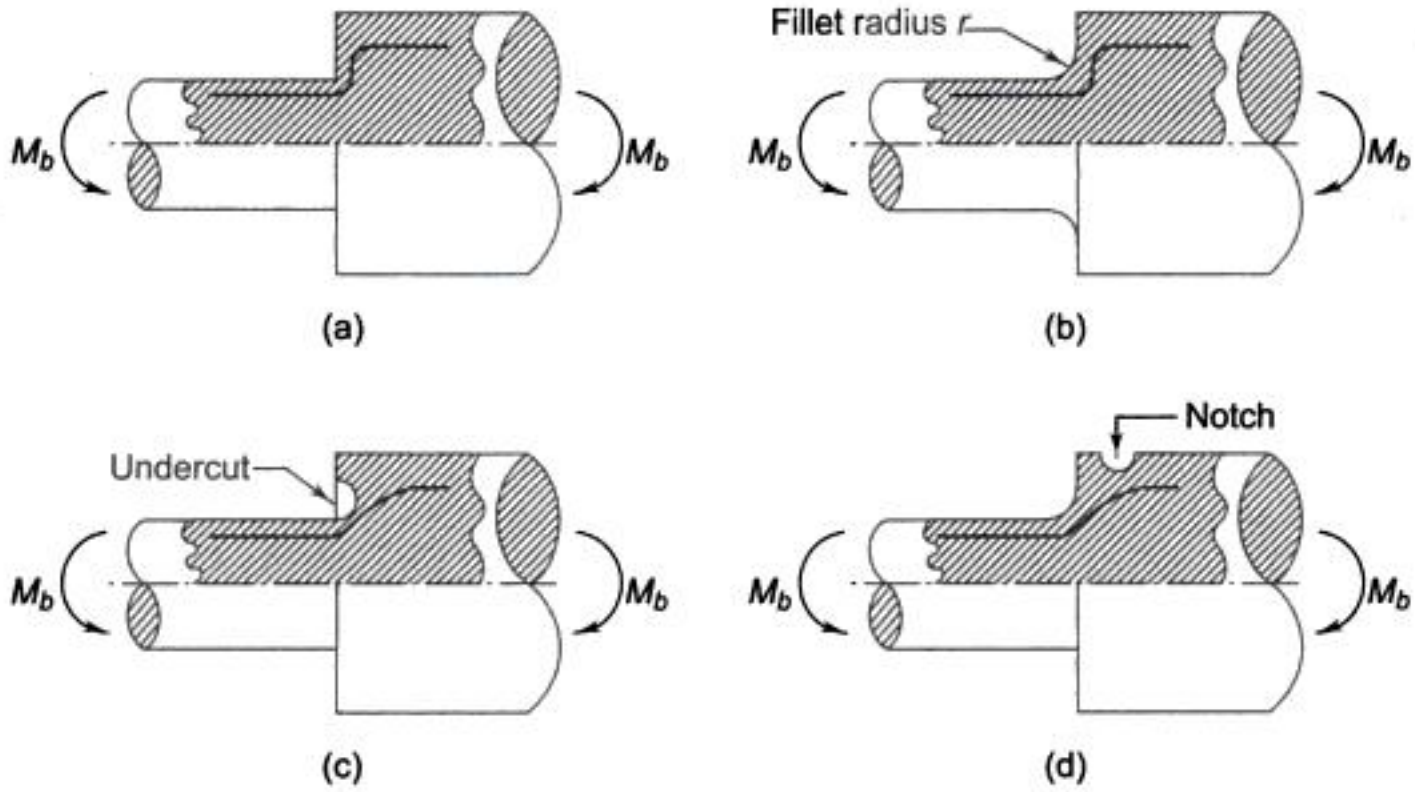

Fig. 5.10 *Reduction of Stress Concentration due to Abrupt Change in Cross-section: (a) Original Component (b) Fillet Radius (c) Undercutting (d) Addition of Notch*

(iii) Drilling Additional Holes for Shaft A transmission shaft with a keyway is shown in Fig. 5.11(a). The keyway is a discontinuity and results in stress concentration at the corners of the keyway and reduces torsional shear strength. An empirical relationship developed by HF Moore for the ratio C of torsional strength of a shaft having a keyway to torsional strength of a same sized shaft without a keyway is given by

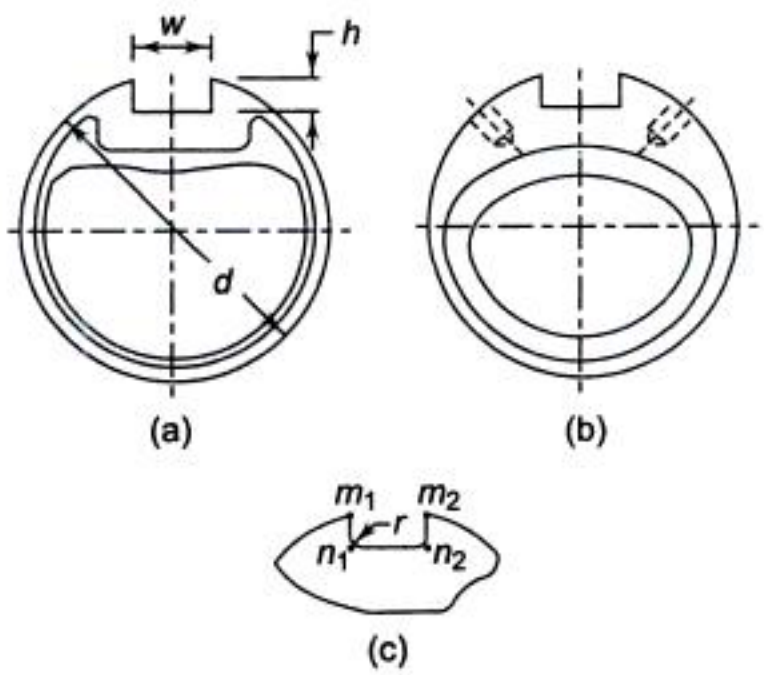

Fig. 5.11 *Reduction of Stress Concentration in Shaft with Keyway: (a) Original Shaft (b) Drilled Holes (c) Fillet Radius*

$$C = 1 - 0.2\left(\frac{w}{d}\right) - 1.1\left(\frac{h}{d}\right) \qquad (5.8)$$

where w and h are width and height dimensions of the keyway respectively and d is the shaft diameter. The four corners of the keyway, viz., m_1, m_2, n_1 and n_2 are shown in Fig. 5.11(c). It has been observed that torsional shear stresses at two points, viz. m_1 and m_2 are negligibly small in practice and theoretically equal to zero. On the other hand, the torsional shear stresses at two points, viz., n_1 and n_2 are excessive and theoretically infinite which means even a small torque will produce a permanent set at these points. Rounding corners at two points, viz., n_1 and n_2 by means of a fillet radius can reduce the stress concentration. A stress concentration factor $K_t = 3$ should be used when a shaft with a keyway is subjected to combined bending and torsional moments.

In addition to giving fillet radius at the inner corners of the keyway, there is another method of drilling two symmetrical holes on the sides of the keyway. These holes press the force flow lines and minimise their bending in the vicinity of the keyway. This method is illustrated in Fig. 5.11(b).

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5.5 FATIGUE FAILURE

It has been observed that materials fail under fluctuating stresses at a stress magnitude which is lower than the ultimate tensile strength of the material. Sometimes, the magnitude is even lower than the yield strength. Further, it has been found that the magnitude of the stress causing fatigue failure decreases as the number of stress cycles increase. This phenomenon of decreased resistance of the materials to fluctuating stresses is the main characteristic of fatigue failure.

Let us examine a phenomenon we have experienced in our childhood. Suppose, there is a wire of 2 to 3 mm diameter and we want to cut it into two pieces without any device like a hacksaw. One method is to shear the wire by applying equal and opposite forces P_1 and P_2 by left and right hands

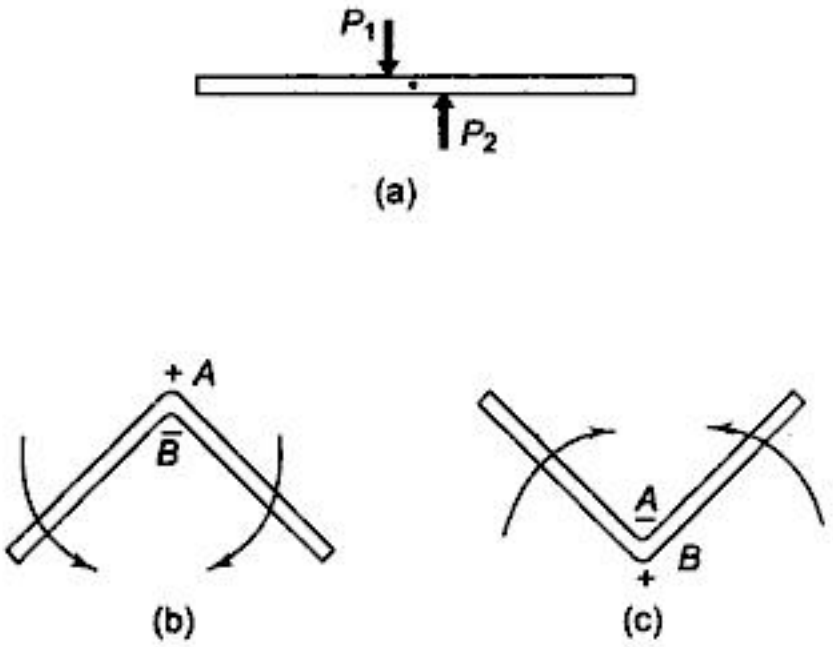

Fig. 5.16 *Shear and Fatigue Failure of Wire: (a) Shearing of Wire (b) Bending of Wire (c) Unbending of Wire*

as illustrated in Fig. 5.16(a). It is difficult to cut the wire by this method. The second method consists of alternatively bending and unbending the wire for few cycles. Let us consider two diametrically opposite points *A* and *B* on the surface of the wire. As shown in Fig. 5.16(b), when the wire is bent, *A* is subjected to tensile stress while *B* to compressive stress. When the wire is unbent, there is compressive stress at *A* and tensile stress at *B*, as shown in Fig. 5.16(c). Therefore, there is complete reversal of stress from tensile stress to compressive stress at the point *A* due to alternate bending and unbending. Similarly, the point *B* is subjected to reversal of stress from compressive stress to tensile stress during the same cycle. We have experienced that the wire can be cut very easily in few cycles of bending and unbending. This is a fatigue failure and the magnitude of stress required to fracture is very low. In other words, there is decreased resistance of material to cyclic stresses. *Fatigue failure is defined as time delayed fracture under cyclic loading*. Examples of parts in which fatigue failures are common are transmission shafts, connecting rods, gears, vehicle suspension springs and ball bearings.

There is a basic difference between failure due to static load and that due to fatigue. The failure due to static load is illustrated by the simple tension test. In this case, the load is gradually applied and there is sufficient time for the elongation of fibres. In ductile materials, there is considerable plastic flow prior to fracture. This results in a silky fibrous structure due to the stretching of crystals at the fractured surface. On the other hand, fatigue failure begins with a crack at some point in the material. The crack is more likely to occur in the following regions:

(i) Regions of discontinuity, such as oil holes, keyways, screw threads, etc.
(ii) Regions of irregularities in machining operations, such as scratches on the surface, stamp mark, inspection marks, etc.
(iii) Internal cracks due to defects in materials like blow holes

These regions are subjected to stress concentration due to the crack. The crack spreads due to fluctuating stresses, until the cross-section of the component is so reduced that the remaining portion is subjected to sudden fracture. There are two distinct areas of fatigue failure—(i) region indicating slow growth of crack with a fine fibrous appearance, and (ii) region of sudden fracture with a coarse granular appearance.

In case of failure under static load, there is sufficient plastic deformation prior to failure, which gives a warning well in advance. On the other hand, fatigue cracks are not visible till they reach the surface of the component and by that time, the failure has already taken place. The fatigue failure is sudden and total. It is relatively easy to design

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$\therefore$ actual stress $= K_f \sigma_o$

theoretical stress $= K_t \sigma_o$

increase of actual stress over nominal stress $= (K_f \sigma_o - \sigma_o)$

increase of theoretical stress over nominal stress $= (K_t \sigma_o - \sigma_o)$

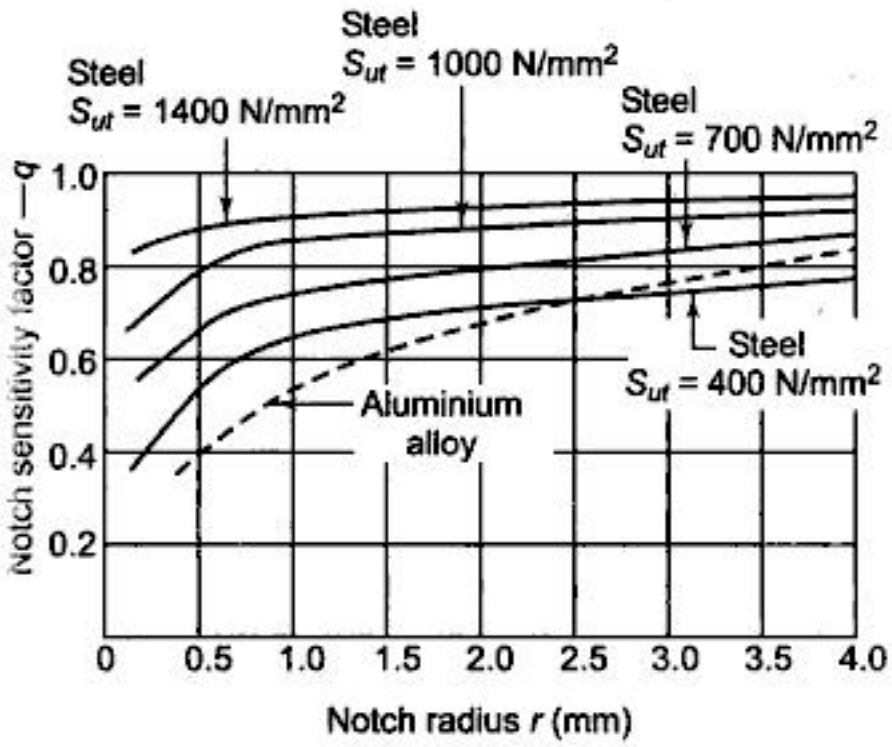

Fig. 5.22 *Notch Sensitivity Charts (for Reversed Bending and Reversed Axial Stresses)*

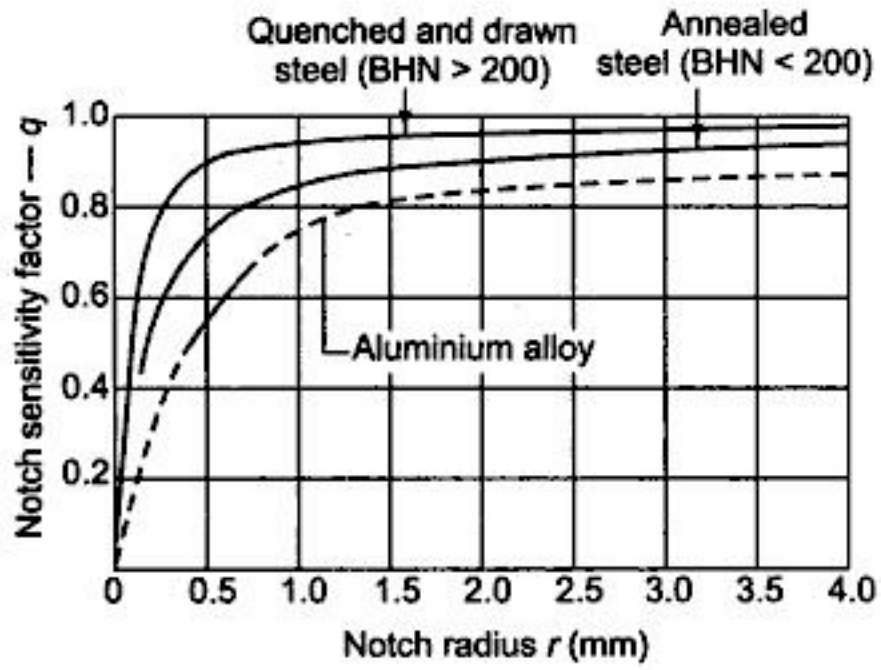

Fig. 5.23 *Notch Sensitivity Charts (for Reversed Torsional Shear Stresses)*

Therefore,

$$q = \frac{(K_f \sigma_o - \sigma_o)}{(K_t \sigma_o - \sigma_o)}$$

or
$$q = \frac{(K_f - 1)}{(K_t - 1)} \qquad (5.11)$$

The above equation can be rearranged in the following form:

$$K_f = 1 + q(K_t - 1) \qquad (5.12)$$

The following conclusions are drawn with the help of Eq. (5.12).

(i) When the material has no sensitivity to notches,

$q = 0$ and $K_f = 1$

(ii) When the material is fully sensitive to notches,

$q = 1$ and $K_f = K_t$

In general, the magnitude of the notch sensitivity factor q varies from 0 to 1. The notch sensitivity factors for various materials for reversed bending or axial stresses and reversed torsional shear stresses[2] are obtained from Fig. 5.22 and 5.23 respectively. In case of doubt, the designer should use ($q = 1$) or ($K_t = K_f$) and the design will be on the safe side.

5.9 ENDURANCE LIMIT – APPROXIMATE ESTIMATION

The laboratory method for determining the endurance limit of materials, although more precise, is laborious and time consuming. A number of tests are required to prepare one *S–N* curve and each test takes considerable time. It is, therefore, not possible to get the experimental data of each and every material. When the laboratory data regarding the endurance limit of the materials is not available, the procedure discussed in this article should be adopted.

Two separate notations are used for endurance limit, viz, (S'_e) and (S_e) where,

S'_e = endurance limit stress of a rotating beam specimen subjected to reversed bending stress (N/mm²)

[2] George Sines and J L Waisman, *Metal Fatigue*—McGraw-Hill Book Co., Inc., 1959 and University of California (Engineering Extension Series), Article No. 13, Notch Sensitivity by R E Peterson.

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considerable dispersion of the data when a number of tests are conducted even using the same material and same conditions. The standard deviation of endurance limit tests is 8% of the mean value. The reliability factor K_c depends upon the reliability that is used in the design of the component. The greater the likelihood that a part will survive, the more is the reliability and lower is the reliability factor. The reliability factor is one for 50% reliability. This means that 50% of the components will survive in the given set of conditions. To ensure that more than 50% of the parts will survive, the stress amplitude on the component should be lower than the tabulated value of the endurance limit. The reliability factor is used to achieve this reduction. The reliability factors based on a standard deviation of 8% are given in Table 5.3[3].

Table 5.3 *Reliability factor*

Reliability R (%)	K_c
50	1.000
90	0.897
95	0.868
99	0.814
99.9	0.753
99.99	0.702
99.999	0.659

(iv) Modifying Factor to Account for Stress Concentration The endurance limit is reduced due to stress concentration. The stress concentration factor used for cyclic loading is less than the theoretical stress concentration factor due to the notch sensitivity of the material. To apply the effect of stress concentration, the designer can either reduce the endurance limit by (K_d) or increase the stress amplitude by (K_f). We will use the first approach. The modifying factor K_d to account for the effect of stress concentration is defined as,

$$K_d = \frac{1}{K_f} \tag{5.27}$$

The above mentioned four factors are used to find out the endurance limit of the actual component.

The endurance limit (S_{se}) of a component subjected to fluctuating torsional shear stresses is obtained from the endurance limit in reversed bending (S_e) using theories of failures.

According to the maximum shear stress theory,

$$S_{se} = 0.5\, S_e \tag{5.28}$$

According to distortion energy theory,

$$S_{se} = 0.577\, S_e \tag{5.29}$$

When the component is subjected to an axial fluctuating load, the conditions are different. In axial loading, the entire cross-section is uniformly stressed to the maximum value. In the rotating beam test, the specimen is subjected to bending stress. The bending stress is zero at the centre of cross-section and negligible in the vicinity of centre. It is only the outer region near the surface, which is subjected to maximum stress. There is more likelihood of a microcrack being present in the much higher high-stress field of axial loading than in the smaller volume outer region of the rotating beam specimen. Therefore, endurance limit in axial loading is lower than the rotating beam test.

For axial loading,

$$(S_e)_a = 0.8\, S_e \tag{5.30}$$

5.10 REVERSED STRESSES—DESIGN FOR FINITE AND INFINITE LIFE

There are two types of problems in fatigue design—(i) components subjected to completely reversed stresses, and (ii) components subjected to fluctuating stresses. As shown in Fig. 5.15, the mean stress is zero in case of completely reversed stresses. The stress distribution consists of tensile stresses for the first half cycle and compressive stresses for the remaining half cycle and the stress cycle passes through zero. In case of fluctuating stresses, there is always a mean stress, and the stresses can be purely tensile, purely compressive or mixed depending upon the magnitude of the mean stress. Such problems are solved with the help of the modified Goodman diagram, which will be discussed in Section 5.13.

The design problems for completely reversed stresses are further divided into two groups—(i) design for infinite life, and (ii) design for finite life.

Case I: When the component is to be designed for infinite life, the endurance limit becomes the

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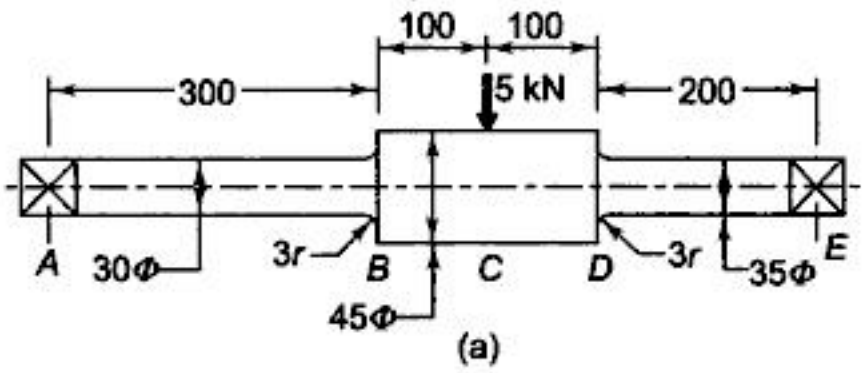
Therefore,

$\log_{10} N = 3 + \overline{EF} = 3 + 1.3754$

$\log_{10} N = 4.3754$

$N = 23\,736.2$ cycles

Example 5.8 *A rotating shaft, subjected to a non-rotating force of 5 kN and simply supported between two bearings A and E is shown in Fig. 5.32(a). The shaft is machined from plain carbon steel 30C8 (S_{ut} = 500 N/mm²) and the expected reliability is 90%. The equivalent notch radius at the fillet section can be taken as 3 mm. What is the life of the shaft?*



(a)

(b)

Fig. 5.32

Solution

Given $P = 5$ kN $S_{ut} = 500$ N/mm²

$R = 90\%$ $r = 3$ mm

Step I *Selection of failure-section*

Taking the moment of the forces about bearings *A* and *E*, the reactions at *A* and *E* are 2143 and 2857 N respectively. The bending moment diagram is shown in Fig. 5.32(b). The values of the bending moment shown in the figure are in N-m. The possibility of a failure will be at the three sections *B*, *C* and *D*. The failure will probably occur at the section *B* rather than at *C* or *D*. At the section *C*, although the bending moment is maximum, the diameter is more and there is no stress concentration. At the section *D*, the diameter is more and the bending moment is less compared with that of section *B*. Therefore, it is concluded that failure will occur at the section *B*.

Step II *Construction of S–N diagram*

At the section *B*,

$$S_f = \sigma_b = \frac{32M_b}{\pi d^3} = \frac{32(642.9 \times 10^3)}{\pi (30)^3}$$

$$= 242.54 \text{ N/mm}^2$$

$S'_e = 0.5S_{ut} = 0.5(500) = 250$ N/mm²

From Fig. 5.24 (machined surface and S_{ut} = 500 N/mm²),

$$K_a = 0.79$$

For 30 mm diameter, $K_b = 0.85$

For 90% reliability, $K_c = 0.897$

Since $\frac{r}{d} = \frac{3}{30} = 0.1$ and $\frac{D}{d} = \frac{45}{30} = 1.5$

From Fig. 5.5, $K_t = 1.72$

From Fig. 5.22 (r = 3 mm and S_{ut} = 500 N/mm²),

$$q = 0.78$$

From Eq. (5.12),

$K_f = 1 + q\,(K_t - 1) = 1 + 0.78\,(1.72 - 1)$

$= 1.5616$

$$K_d = \frac{1}{K_f} = \frac{1}{1.5616} = 0.64$$

$S_e = K_a K_b K_c K_d S'_e$

$= 0.79\,(0.85)(0.897)(0.64)(250) = 96.37$ N/mm²

$0.9S_{ut} = 0.9\,(500) = 450$ N/mm²

$\log_{10}(0.9S_{ut}) = \log_{10}(450) = 2.6532$

$\log_{10}(S_e) = \log_{10}(96.37) = 1.9839$

$\log_{10}(S_f) = \log_{10}(242.54) = 2.3848$

Also, $\log_{10}(10^3) = 3$ and $\log_{10}(10^6) = 6$

The *S–N* curve for the shaft is shown in Fig. 5.33.

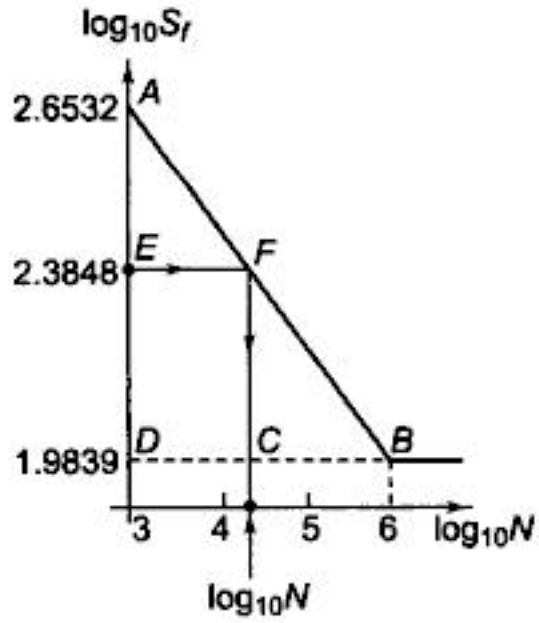


Fig. 5.33

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$$\overline{EF} = \frac{\overline{DB} \times \overline{AE}}{\overline{AD}} = \frac{(6-3)(2.7738 - \log_{10}\sigma)}{(2.7738 - 2.4472)} \quad \text{(a)}$$

and $$\log_{10} N = 3 + \overline{EF} \quad \text{(b)}$$

From (a) and (b),

$\log_{10} N = 3 + 9.1855\ (2.7738 - \log_{10}\sigma)$

Therefore,

$\log_{10}(N_1) = 3 + 9.1855\ (2.7738 - 2.5441)$

or $N_1 = 128\,798$

$\log_{10}(N_2) = 3 + 9.1855\ (2.7738 - 2.6021)$

or $N_2 = 37\,770$

$\log_{10}(N_3) = 3 + 9.1855\ (2.7738 - 2.6990)$

or $N_3 = 4865$

Step III *Fatigue life of component*

From Eq. 5.34,

$$\frac{\alpha_1}{N_1} + \frac{\alpha_2}{N_2} + \frac{\alpha_3}{N_3} = \frac{1}{N}$$

$$\frac{0.85}{128\,798} + \frac{0.12}{37\,770} + \frac{0.03}{4865} = \frac{1}{N}$$

or $N = 62\,723$ cycles

5.12 SODERBERG AND GOODMAN LINES

When a component is subjected to fluctuating stresses as shown in Fig. 5.15 (a), there is mean stress (σ_m) as well as stress amplitude (σ_a). It has been observed that the mean stress component has an effect on fatigue failure when it is present in combination with an alternating component. The fatigue diagram for this general case is shown in Fig. 5.39. In this diagram, the mean stress is plotted on the abscissa. The stress amplitude is plotted on the ordinate. The magnitudes of (σ_m) and (σ_a) depend upon the magnitudes of maximum and minimum force acting on the component. When stress amplitude (σ_a) is zero, the load is purely static and the criterion of failure is S_{ut} or S_{yt}. These limits are plotted on the abscissa. When the mean stress (σ_m) is zero, the stress is completely reversing and the criterion of failure is the endurance limit S_e that is plotted on the ordinate. When the component is subjected to both components of stress, viz., (σ_m) and (σ_a), the actual failure occurs at different scattered points shown in the figure. There exists a border, which divides safe region from unsafe region for various combinations of (σ_m) and (σ_a). Different criterions are proposed to construct the borderline dividing safe zone and failure zone. They include *Gerber line, Soderberg line* and *Goodman line.*

Gerber Line *A parabolic curve joining S_e on the ordinate to S_{ut} on the abscissa is called the Gerber line.*

Soderberg Line *A straight line joining S_e on the ordinate to S_{yt} on the abscissa is called the Soderberg line.*

Goodman Line *A straight line joining S_e on the ordinate to S_{ut} on the abscissa is called the Goodman line.*

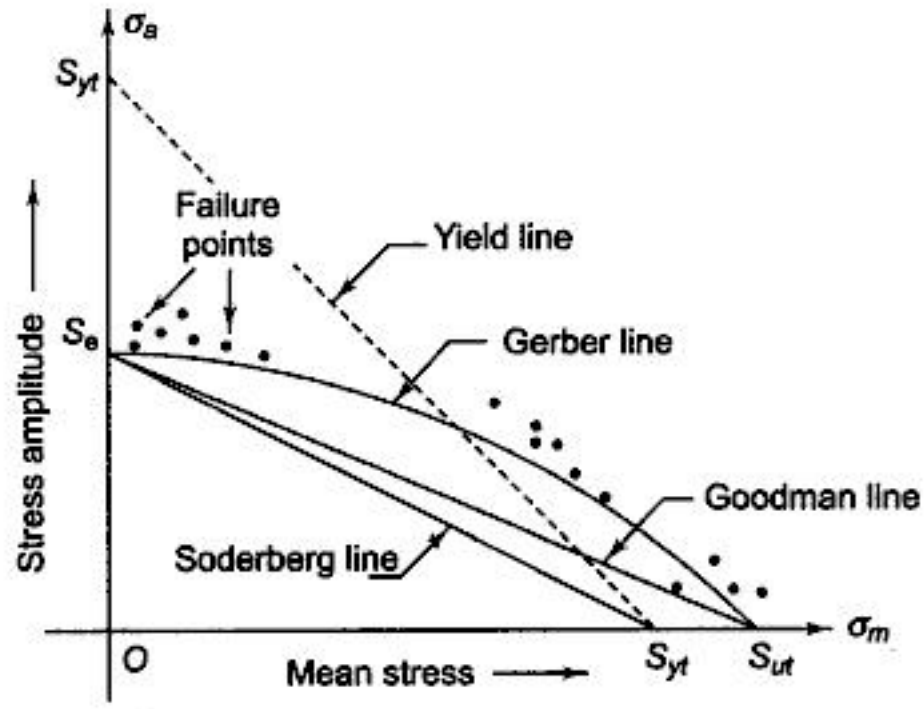

Fig. 5.39 *Soderberg and Goodman Lines*

The Gerber parabola fits the failure points of test data in the best possible way. The Goodman line fits beneath the scatter of this data. Both Gerber parabola and Goodman line intersect at (S_e) on the ordinate to (S_{ut}) on the abscissa. However, the Goodman line is more safe from design considerations because it is completely inside the Gerber parabola and inside the failure points. The Soderberg line is a more conservative failure criterion and there is no need to consider even yielding in this case. A yield line is constructed connecting (S_{yt}) on both axes. It is called the limit on '*first cycle*' of stress. This is because if a part yields, it has failed, regardless of its safety in fatigue.

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Step I *Endurance limit stress for shaft*

$S'_e = 0.5S_{ut} = 0.5\ (500) = 250\ \text{N/mm}^2$

From Fig. 5.24 (cold drawn steel and $S_{ut} = 500\ \text{N/mm}^2$),

$K_a = 0.79$

Assuming $7.5 < d < 50$ mm,

$K_b = 0.85$

For 90% reliability, $K_c = 0.897$

$S_e = K_a K_b K_c S'_e = 0.79\ (0.85)\ (0.897)\ (250)$
$= 150.58\ \text{N/mm}^2$

Step II *Construction of modified Goodman diagram*

Using the distortion energy theory,

$S_{se} = 0.577\ S_e = 0.577\ (150.58) = 86.88\ \text{N/mm}^2$

$S_{sy} = 0.577\ S_{yt} = 0.577\ (300) = 173.1\ \text{N/mm}^2$

$$(M_t)_m = \frac{1}{2}[(M_t)_{\max.} + (M_t)_{\min.}]$$

$$= \frac{1}{2}[400 - 100] = 150\ \text{N-m}$$

$$(M_t)_a = \frac{1}{2}[(M_t)_{\max.} - (M_t)_{\min.}]$$

$$= \frac{1}{2}[400 + 100] = 250\ \text{N-m}$$

$$\tan\theta = \frac{(M_t)_a}{(M_t)_m} = \frac{250}{150} = 1.67$$

$$\theta = 59.04^\circ$$

The modified Goodman diagram for this example is shown in Fig. 5.44.

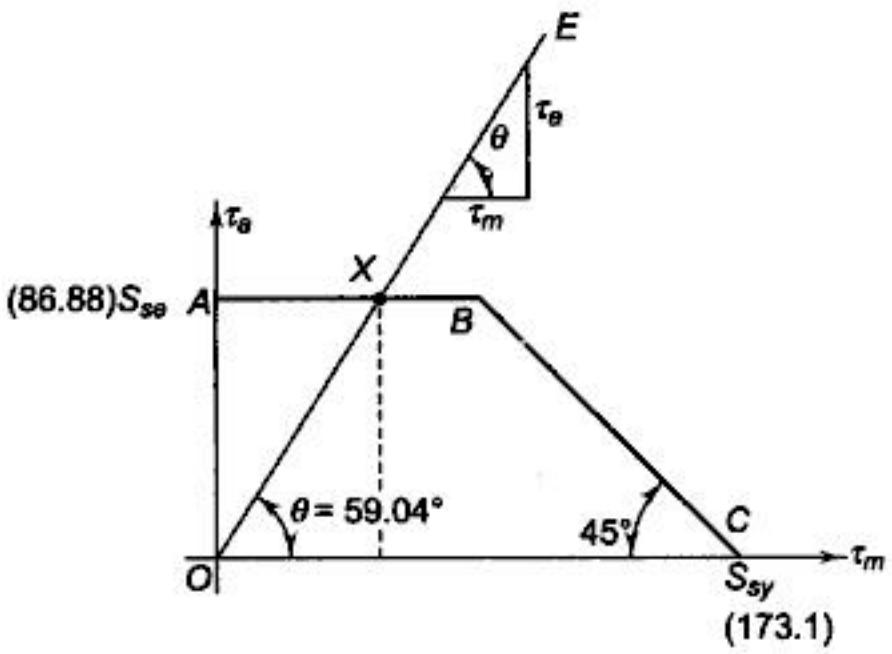

Fig. 5.44

Step III *Permissible shear stress amplitude*

Refer to Fig. 5.44. The ordinate of the point X is S_{se} or 86.88 N/mm^2.

$\therefore \quad S_{sa} = 86.88\ \text{N/mm}^2$

Step IV *Diameter of shaft*

$$\text{Since } \tau_a = \frac{S_{sa}}{(fs)} \qquad \therefore \frac{16(M_t)_a}{\pi d^3} = \frac{S_{sa}}{(fs)}$$

$$\frac{16(250 \times 10^3)}{\pi d^3} = \frac{86.88}{2}$$

$$d = 30.83\ \text{mm}$$

Example 5.14 *A spherical pressure vessel, with a 500 mm inner diameter, is welded from steel plates. The welded joints are sufficiently strong and do not weaken the vessel. The plates are made from cold drawn steel 20C8 (S_{ut} = 440 N/mm^2 and S_{yt} = 242 N/mm^2). The vessel is subjected to internal pressure, which varies from zero to 6 N/mm^2. The expected reliability is 50% and the factor of safety is 3.5. The size factor is 0.85. The vessel is expected to withstand infinite number of stress cycles. Calculate the thickness of the plates.*

Solution

Given For vessel $D_i = 500$ mm

p_i = zero to 6 N/mm^2 $\quad S_{ut} = 440\ \text{N/mm}^2$

$S_{yt} = 242\ \text{N/mm}^2$ $\quad R = 50\%$ $(fs) = 3.5$ $\quad K_b = 0.85$

Step I *Endurance limit stress for vessel*

$S'_e = 0.5S_{ut} = 0.5\ (440) = 220\ \text{N/mm}^2$

From Fig. 5.24 (cold drawn steel and $S_{ut} = 440\ \text{N/mm}^2$),

$K_a = 0.82$

$K_b = 0.85$

For 50% reliability, $K_c = 1.0$

$S_e = K_a K_b K_c S'_e = 0.82\ (0.85)\ (1.0)\ (220)$
$= 153.34\ \text{N/mm}^2$

Step II *Construction of modified Goodman diagram*

For a spherical pressure vessel,

$$\sigma_t = \frac{p_i D_i}{4t}$$

$$\sigma_{\max.} = \frac{p_{\max.} D_i}{4t} = \frac{6(500)}{4t} = \left(\frac{750}{t}\right)\text{N/mm}^2$$

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of the most popular non-linear theories is the Gerber theory that is based on parabolic curve. The Gerber curve is shown in Fig. 5.52. The equation for the Gerber curve is as follows:

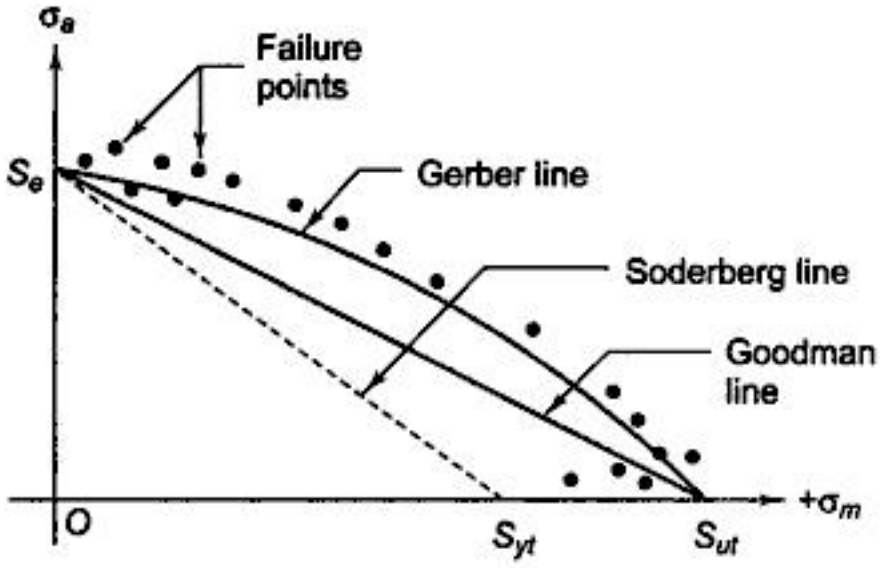

Fig. 5.52 *Gerber line*

$$\frac{S_a}{S_e}+\left(\frac{S_m}{S_{ut}}\right)^2=1 \qquad (5.46)$$

The above equation is called the *Gerber equation*. It can be also written in the following form:

$$S_a = S_e\left[1-\left(\frac{S_m}{S_{ut}}\right)^2\right] \qquad (5.47)$$

Theories based on the Soderberg line or the Goodman line, as failure criteria are conservative theories. This results in increased dimensions of the component. The Gerber curve takes the mean path through failure points. It is therefore more accurate in predicting fatigue failure.

Example 5.17 *A machine component is subjected to fluctuating stress that varies from 40 to 100 N/mm². The corrected endurance limit stress for the machine component is 270 N/mm². The ultimate tensile strength and yield strength of the material are 600 and 450 N/mm² respectively. Find the factor of safety using*

(i) Gerber theory
(ii) Soderberg line
(iii) Goodman line

Also, find the factor of safety against static failure.

Solution

Given $S_{ut} = 600$ N/mm² $S_{yt} = 450$ N/mm²
$S_e = 270$ N/mm² $\sigma_{max.} = 100$ N/mm²
$\sigma_{min.} = 40$ N/mm²

Step I *Permissible mean and amplitude stresses*

$$\sigma_a = \frac{1}{2}(100-40) = 30 \text{ N/mm}^2$$

$$\sigma_m = \frac{1}{2}(100+40) = 70 \text{ N/mm}^2$$

$$S_a = n\sigma_a = 30\,n$$
$$S_m = n\sigma_m = 70\,n$$

where n is the factor of safety.

Step II *Factor of safety using Gerber theory*

From Eq. (5.46),

$$\frac{S_a}{S_e}+\left(\frac{S_m}{S_{ut}}\right)^2=1$$

or
$$\left(\frac{30n}{270}\right)+\left(\frac{70n}{600}\right)^2=1$$

$$n^2 + 8.16\,n - 73.47 = 0$$

Solving the above quadratic equation,

$$n = 5.41 \qquad \text{(i)}$$

Step III *Factor of safety using Soderberg line*

The equation of the Soderberg line is as follows,

$$\frac{S_a}{S_e}+\frac{S_m}{S_{yt}}=1$$

$$\left(\frac{30n}{270}\right)+\left(\frac{70n}{450}\right)=1$$

$$n = 3.75 \qquad \text{(ii)}$$

Step IV *Factor of safety using Goodman line*

The equation of the Goodman line is as follows:

$$\frac{S_a}{S_e}+\frac{S_m}{S_{ut}}=1$$

$$\left(\frac{30n}{270}\right)+\left(\frac{70n}{600}\right)=1$$

$$n = 4.39 \qquad \text{(iii)}$$

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Step I *Mean and amplitude stresses*

$$(M_b)_m = \frac{1}{2}[(M_b)_{\max.} + (M_b)_{\min.}]$$

$$= \frac{1}{2}[600 + 200] = 400 \text{ N-m}$$

$$(M_b)_a = \frac{1}{2}[(M_b)_{\max.} - (M_b)_{\min.}]$$

$$= \frac{1}{2}[600 - 200] = 200 \text{ N-m}$$

$$(M_t)_m = \frac{1}{2}[(M_t)_{\max.} + (M_t)_{\min.}]$$

$$= \frac{1}{2}[200 + 70] = 135 \text{ N-m}$$

$$(M_t)_a = \frac{1}{2}[(M_t)_{\max.} - (M_t)_{\min.}]$$

$$= \frac{1}{2}[200 - 70] = 65 \text{ N-m}$$

$$\sigma_{xm} = \frac{32(M_b)_m}{\pi d^3} = \frac{32(400 \times 10^3)}{\pi d^3}$$

$$= \left(\frac{4074.37 \times 10^3}{d^3}\right) \text{N/mm}^2$$

$$\sigma_{xa} = \frac{32(M_b)_a}{\pi d^3} = \frac{32(200 \times 10^3)}{\pi d^3}$$

$$= \left(\frac{2037.18 \times 10^3}{d^3}\right) \text{N/mm}^2$$

$$\tau_{xym} = \frac{16(M_t)_m}{\pi d^3} = \frac{16(135 \times 10^3)}{\pi d^3}$$

$$= \left(\frac{687.55 \times 10^3}{d^3}\right) \text{N/mm}^2$$

$$\tau_{xya} = \frac{16(M_t)_a}{\pi d^3} = \frac{16(65 \times 10^3)}{\pi d^3}$$

$$= \left(\frac{331.04 \times 10^3}{d^3}\right) \text{N/mm}^2$$

$$\sigma_m = \sqrt{\sigma_{xm}^2 + 3\tau_{xym}^2}$$

$$= \sqrt{\left(\frac{4074.37 \times 10^3}{d^3}\right)^2 + 3\left(\frac{687.55 \times 10^3}{d^3}\right)^2}$$

$$= \left(\frac{4244.84 \times 10^3}{d^3}\right) \text{N/mm}^2$$

$$\sigma_a = \sqrt{\sigma_{xa}^2 + 3\tau_{xya}^2}$$

$$= \sqrt{\left(\frac{2037.18 \times 10^3}{d^3}\right)^2 + 3\left(\frac{331.04 \times 10^3}{d^3}\right)^2}$$

$$= \left(\frac{2116.33 \times 10^3}{d^3}\right) \text{N/mm}^2$$

Step II *Construction of modified Goodman diagram*

$$\tan\theta = \frac{\sigma_a}{\sigma_m} = \frac{2116.33}{4244.84} = 0.4986 \quad \text{or} \quad \theta = 26.5°$$

The modified Goodman diagram for this example is shown in Fig. 5.56.

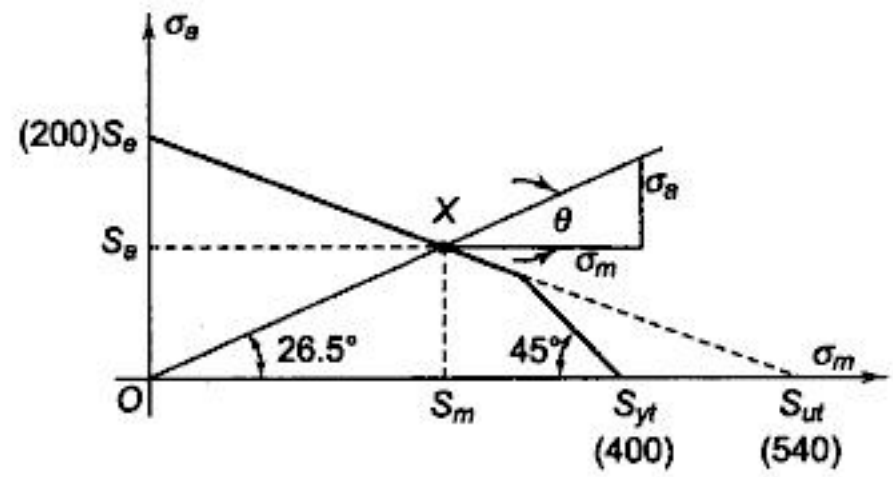

Fig. 5.56

Step III *Permissible stress amplitude*

Refer to Fig. 5.56. The co-ordinates of the point X are obtained by solving the following two equations simultaneously:

$$\frac{S_a}{200} + \frac{S_m}{540} = 1 \qquad \text{(a)}$$

$$\frac{S_a}{S_m} = \tan\theta = 0.4986 \qquad \text{(b)}$$

$\therefore$ $S_a = 114.76 \text{ N/mm}^2$

$S_m = 230.16 \text{ N/mm}^2$

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the fillet section. Assume the shaft material to be brittle. [*20.39 N/mm²*]

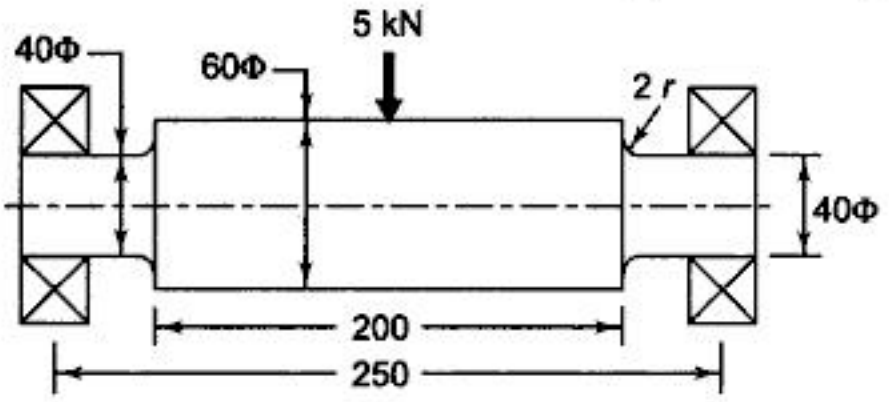

Fig. 5.60

5.4 A plate, 10 mm thick, subjected to a tensile load of 20 kN is shown in Fig. 5.61. The plate is made of cast iron (S_{ut} = 350 N/mm²) and the factor of safety is 2.5. Determine the fillet radius. [*2.85 or 3 mm*]

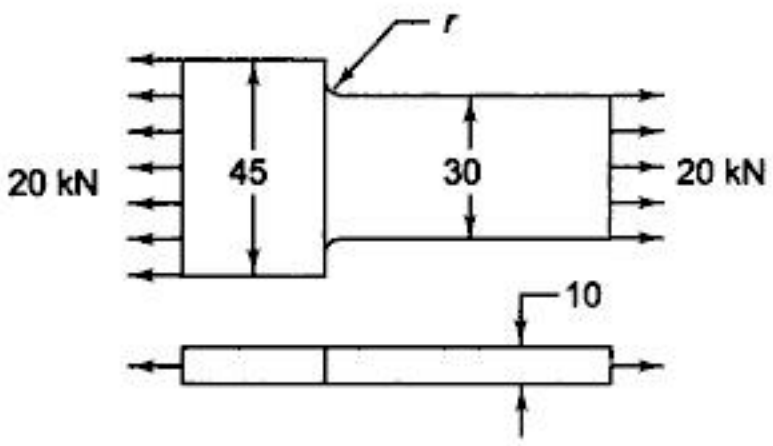

Fig. 5.61

5.5 A 25 mm diameter shaft is made of forged steel 30C8 (S_{ut} = 600 N/mm²). There is a step in the shaft and the theoretical stress concentration factor at the step is 2.1. The notch sensitivity factor is 0.84. Determine the endurance limit of the shaft if it is subjected to a reversed bending moment. [*59.67 N/mm²*]

5.6 A 40 mm diameter shaft is made of steel 50C4 (S_{ut} = 660 N/mm²) and has a machined surface. The expected reliability is 99%. The theoretical stress concentration factor for the shape of the shaft is 1.6 and the notch sensitivity factor is 0.9. Determine the endurance limit of the shaft. [*112.62 N/mm²*]

5.7 A cantilever beam made of steel Fe 540 (S_{ut} = 540 N/mm² and S_{yt} = 320 N/mm²) and subjected to a completely reversed load (P) of 5 kN is shown in Fig. 5.62. The beam is machined and the reliability is 50%. The factor of safety is 2 and the notch sensitivity factor is 0.9. Calculate

(i) endurance limit at the fillet section; and

(ii) diameter *d* of the beam for infinite life.

[*(i) 109.20 N/mm² (ii) 45.35 mm*]

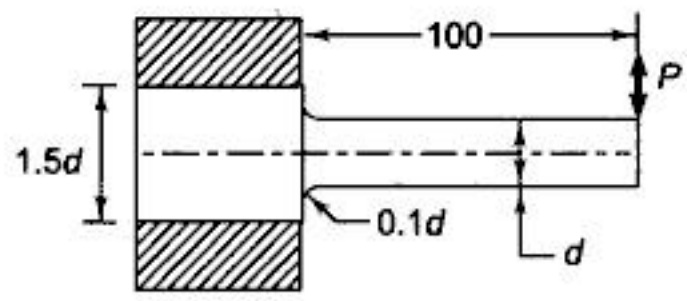

Fig. 5.62

5.8 A solid circular shaft made of steel Fe 620 (S_{ut} = 620 N/mm² and S_{yt} = 380 N/mm²) is subjected to an alternating torsional moment, which varies from –200 N-m to + 400 N-m. The shaft is ground and the expected reliability is 90%. Neglecting stress concentration, calculate the shaft diameter for infinite life. The factor of safety is 2. Use the distortion-energy theory of failure. [*29.31 mm*]

5.9 A solid circular shaft, 15 mm in diameter, is subjected to torsional shear stress, which varies from 0 to 35 N/mm² and at the same time, is subjected to an axial stress that varies from –15 to +30 N/mm². The frequency of variation of these stresses is equal to the shaft speed. The shaft is made of steel FeE 400 (S_{ut} = 540 N/mm² and S_{yt} = 400 N/mm²) and the corrected endurance limit of the shaft is 200 N/mm². Determine the factor of safety. [*4.05*]

5.10 A bar of steel has an ultimate tensile strength of 700 MPa, a yield point stress of 400 MPa and fully corrected endurance limit (S_e) of 220 MPa. The bar is subjected to a mean bending stress of 60 MPa and a stress amplitude of 80 MPa. Superimposed on it is a mean torsional stress and torsional stress amplitude of 70 and 35 MPa respectively. Find the factor of safety. [*1.54*]

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Table 6.2 *Proportions of ISO metric trapezoidal threads*

Nominal diameter, d (mm)	*Pitch, p (mm)*
24, 28	5
32, 36	6
40, 44	7
48, 52	8
60	9
70, 80	10
90, 100	12

There is a particular method of designation for square and trapezoidal threads. A power screw with single-start square threads is designated by the letters 'Sq' followed by the nominal diameter and the pitch expressed in millimetres and separated by the sign '×'. For example,

Sq 30 × 6

It indicates single-start square threads with 30 mm nominal diameter and 6 mm pitch.

Similarly, single-start ISO metric trapezoidal threads are designated by the letters 'Tr' followed by the nominal diameter and the pitch expressed in millimetres and separated by the sign '×'. For example,

Tr 40 × 7

It indicates single-start trapezoidal threads with 40 mm nominal diameter and 7 mm pitch.

Multiple-start trapezoidal threads are designated by the letters 'Tr' followed by the nominal diameter and the lead, separated by the sign '×' and in brackets the letter '*P*' followed by the pitch expressed in millimetres. For example,

Tr 40 × 14 (*P* 7)

In the above designation,

lead = 14 mm pitch = 7 mm

∴ No. of starts = 14/7 = 2

Therefore, the above designation indicates a two-start trapezoidal thread with 40 mm nominal diameter and a 7 mm pitch. In case of left-hand threads, the letter *LH* is added to the thread designation. For example,

Tr 40 × 14 (*P* 7) *LH*

6.3 MULTIPLE THREADED SCREWS

Multiple threaded power screws are used in certain applications where higher travelling speed is required. They are also called multiple start screws such as double-start or triple-start screws. These screws have two or more threads cut side by side, around the rod. Such a screw can be imagined by winding two or more coloured strings side by side around the pencil. A multiple threaded screw has the following advantages:

(i) It provides large axial motion per revolution of the screw. This increases the travelling speed of the sliding member. For a double-start screw, the travelling speed is twice that of the single-start screw.

(ii) The efficiency of a multi-threaded screw is more than a single-threaded screw due to increase in helix angle.

The disadvantages of a multiple threaded screw are as follows:

(i) The mechanical advantage obtained with multiple threaded screws is lower than that of a single-threaded screw. Therefore, the effort required to raise a particular load or apply a particular force is more.

(ii) It is likely that the self-locking property may be lost in a multi-threaded screw. The resulting condition is dangerous in certain applications like screw jacks, where the load may descend on its own.

Multiple threaded screws are used in high speed actuators and sluice valves.

6.4 TERMINOLOGY OF POWER SCREW

The main dimensions of a double-threaded screw are shown in Fig. 6.4. The terminology of the screw thread is as follows:

(i) Pitch The pitch is defined as the distance measured parallel to the axis of the screw from a point on one thread to the corresponding point on the adjacent thread. It is denoted by the letter *p*.

(ii) Lead The lead is defined as the distance measured parallel to the axis of the screw which the

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derived for a square threaded screw are used for trapezoidal or acme threads by making this change.

Case I *Lifting load*
Modifying Eq. (c) of Section 6.5,

$$P = \frac{W(\mu \sec\theta + \tan\alpha)}{(1 - \mu \sec\theta \tan\alpha)} \tag{6.12}$$

and $$M_t = P\left(\frac{d_m}{2}\right) = \frac{Wd_m}{2} \cdot \frac{(\mu \sec\theta + \tan\alpha)}{(1 - \mu \sec\theta \tan\alpha)} \tag{6.13}$$

Case II *Lowering load*
From Eq. (c) of Section 6.6

$$P = \frac{W(\mu \sec\theta - \tan\alpha)}{(1 + \mu \sec\theta \tan\alpha)} \tag{6.14}$$

and $$M_t = P\left(\frac{d_m}{2}\right) = \frac{Wd_m}{2} \cdot \frac{(\mu \sec\theta - \tan\alpha)}{(1 + \mu \sec\theta \tan\alpha)} \tag{6.15}$$

Efficiency of Screw From Eq. (a) of Section 6.8, the efficiency of a square threaded screw is given by,

$$\eta = \frac{W}{P} \tan\alpha$$

Substituting Eq. (c) of Section 6.5 in the above expression,

$$\eta = \frac{\tan\alpha(1 - \mu \tan\alpha)}{(\mu + \tan\alpha)}$$

Substituting ($\mu \sec\theta$) in place of (μ), the efficiency of trapezoidal threads is given by,

$$\eta = \frac{\tan\alpha(1 - \mu \sec\theta \tan\alpha)}{(\mu \sec\theta + \tan\alpha)} \tag{6.16}$$

6.11 COLLAR FRICTION TORQUE

In many applications of the power screw, there is collar friction in addition to the friction at the thread surface. The principle of collar friction can be explained with the help of Fig. 6.10(a). The cup remains stationary under the action of the load W, while the collar that is integral with the screw rotates when the load is being raised or lowered. Therefore, there is relative motion between the cup and the collar at the annular interface from diameter D_i to D_o. This relative motion results in friction called *collar friction*. The torque required to overcome this friction or collar friction torque $(M_t)_c$ can be determined by using uniform pressure theory or uniform wear theory.

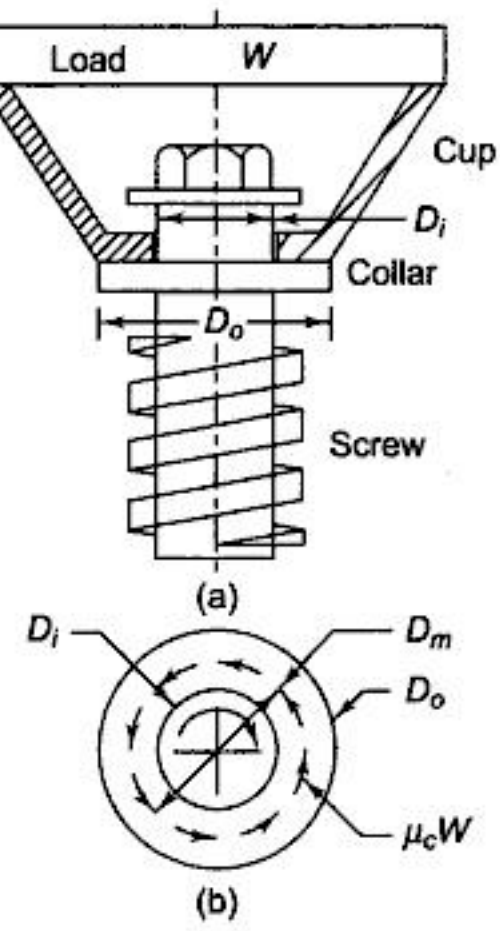

Fig. 6.10 *Collar Friction*

According to the uniform pressure theory,

$$(M_t)_c = \frac{\mu_c W}{3} \cdot \frac{D_0^3 - D_i^3}{D_0^2 - D_i^2} \tag{6.17}$$

According to the uniform wear theory,

$$(M_t)_c = \frac{\mu_c W}{4} \cdot (D_0 + D_i) \tag{6.18}$$

where

μ_c = coefficient of friction at the collar
D_o = outer diameter of the collar (mm)
D_i = inner diameter of the collar (mm)
$(M_t)_c$ = collar friction torque (N-mm)

Uniform pressure theory is applicable only when the collar surface is new. Uniform wear theory is applicable to the collar surface after the initial wear. Therefore, it is logical to use uniform wear theory for collar friction.

There is a simple method to derive Eq. (6.18) applicable to uniform wear condition. Since

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Step III *Force required to raise the load*

The force P_i at the radius of 500 mm is given by,

$$P_i = \frac{(M_t)_t}{\text{radius}} = \frac{204\,643.56}{500} = 409.3 \text{ N} \qquad \text{(iii)}$$

Example 6.2 *A double-threaded power screw, with ISO metric trapezoidal threads is used to raise a load of 300 kN. The nominal diameter is 100 mm and the pitch is 12 mm. The coefficient of friction at the screw threads is 0.15. Neglecting collar friction, calculate*

(i) torque required to raise the load;
(ii) torque required to lower the load; and
(iii) efficiency of the screw.

Solution

Given $W = 300$ kN

For screw, $d = 100$ mm $p = 12$ mm $\mu = 0.15$

No. of starts = 2

Step I *Torque required to raise the load*

For ISO metric trapezoidal threads,

$\theta = 15°$ $l = 2p = 2(12) = 24$ mm

$d_m = d - 0.5p = 100 - 0.5(12) = 94$ mm

$$\tan\alpha = \frac{l}{\pi d_m} = \frac{24}{\pi(94)} = 0.0813$$

$$\mu \sec\theta = \frac{\mu}{\cos\theta} = \frac{0.15}{\cos(15)} = 0.1553$$

$$M_t = \frac{Wd_m}{2} \frac{(\mu\sec\theta + \tan\alpha)}{(1 - \mu\sec\theta\tan\alpha)}$$

$$= \frac{(300\times10^3)(94)}{2} \times \frac{(0.1553 + 0.0813)}{(1 - 0.1553 + 0.0813)}$$

$$= 3378.72 \times 10^3 \text{ N-mm or } 3378.72 \text{ N-m} \quad \text{(i)}$$

Step II *Torque required to lower the load*

$$M_t = \frac{Wd_m}{2} \frac{(\mu\sec\theta - \tan\alpha)}{(1 + \sec\theta\tan\alpha)}$$

$$= \frac{(300\times10^3)(94)}{2} \frac{(0.1553 - 0.0813)}{(1 + 0.1553\times0.0813)}$$

$$= 1030.39 \times 10^3 \text{ N-mm or } 1030.39 \text{ N-m} \quad \text{(ii)}$$

Step III *Efficiency of s*

$$\eta = \frac{\tan\alpha(1 - \mu s \quad \tan\alpha)}{(\mu\sec\theta + \tan\alpha)}$$

$$= \frac{0.0813(1 - 0.1553\times0.0813)}{(0.1553 + 0.0813)}$$

$$= 0.3393 \quad \text{or} \quad 33.93\% \qquad \text{(iii)}$$

Example 6.3 *A machine vice, as shown in Fig. 6.12, has single-start, square threads with 22 mm nominal diameter and 5 mm pitch. The outer and inner diameters of the friction collar are 55 and 45 mm respectively. The coefficients of friction for thread and collar are 0.15 and 0.17 respectively. The machinist can comfortably exert a force of 125 N on the handle at a mean radius of 150 mm. Assuming uniform wear for the collar, calculate*

(i) the clamping force developed between the jaws; and
(ii) the overall efficiency of the clamp.

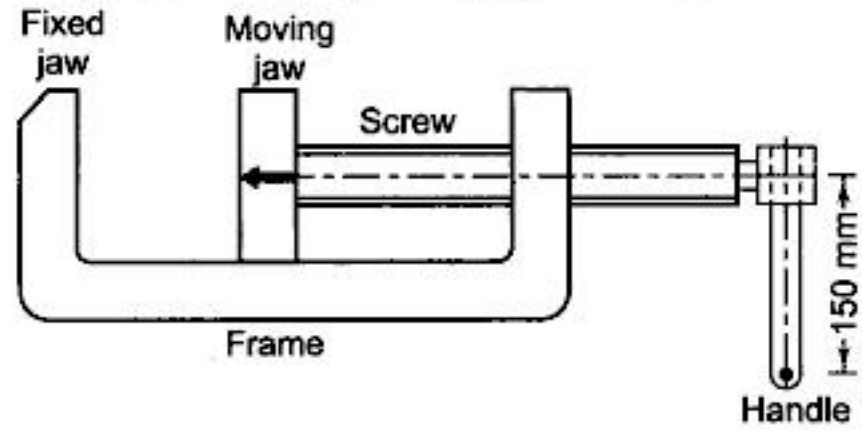

Fig. 6.12

Solution

Given For screw, $d = 22$ mm $l = p = 5$ mm

$\mu = 0.15$

For collar, $D_o = 55$ mm

$D_i = 45$ mm $\mu = 0.17$

For handle $P = 125$ N

length = 150 mm

Step I *Screw and collar friction torques*

$d_m = d - 0.5\,p = 22 - 0.5(5) = 19.5$ mm

$$\tan\alpha = \frac{l}{\pi d_m} = \frac{5}{\pi(19.5)} \quad \text{or} \quad \alpha = 4.666°$$

$\tan\phi = \mu = 0.15$ or $\phi = 8.531°$

From Eq. (6.6),

$$M_t = \frac{Wd_m}{2}\tan(\phi + \alpha)$$

$$= \frac{W(19.5)}{2}\tan(8.531 + 4.666)$$

$$= (2.286W) \text{ N-mm} \qquad \text{(a)}$$

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$$\tan\alpha = \frac{l}{\pi d_m} = \frac{8}{\pi(48)} = 0.0531$$

$$\mu \sec\theta = \frac{\mu}{\cos\theta} = \frac{0.15}{\cos(15)} = 0.1553$$

The torque required to overcome friction at the thread surface is given by,

$$M_t = \frac{Wd_m}{2} \cdot \frac{(\mu\sec\theta + \tan\alpha)}{(1 - \mu\sec\theta\tan\alpha)}$$

$$= \frac{(2\times10^3)(48)}{2} \frac{(0.1553 + 0.0531)}{(1 - 0.1553\times0.0531)}$$

$$= 10\,086.38 \text{ N-mm.}$$

From Eq. (6.18),

$$(M_t)_c = \frac{\mu_c W}{4} \cdot (D_0 + D_i)$$

$$= \frac{(0.12)(2000)(100 + 60)}{4}$$

$$= 9600 \text{ N-mm}$$

The total torque required to drive the lead screw is given by,

$$(M_t)_t = M_t + (M_t)_c = 10\,086.38 + 9600$$

$$= 19\,686.38 \text{ N-mm}$$

The power required to drive the lead screw is given by,

$$\text{kW} = \frac{2\pi n(M_t)_t}{60\times10^6} = \frac{2\pi(30)(19686.38)}{60\times10^6}$$

$$= 0.0618 \qquad \text{(i)}$$

Step II *Efficiency of screw*

From Eq. (6.20),

$$\eta_o = \frac{Wl}{2\pi(M_t)_t} = \frac{(2\times10^3)(8)}{2\pi(19686.38)}$$

$$= 0.1294 \quad \text{or} \quad 12.94\% \qquad \text{(ii)}$$

Example 6.7 *In a machine tool application, the tool holder is pulled by means of an operating nut mounted on a screw. The tool holder travels at a speed of 5 m/min. The screw has single-start square threads of 48 mm nominal diameter and 8 mm pitch. The operating nut exerts a force of 500 N to drive the tool holder. The mean radius of the friction collar is 40 mm. The coefficient of friction at thread and collar surfaces is 0.15. Calculate*

(i) power required to drive the screw; and

(ii) the efficiency of the mechanism.

Solution

Given $W = 500$ N

For screw, $d = 48$ mm $\quad l = p = 8$ mm $\quad \mu = 0.15$

For collar, $r_m = 40$ mm $\quad \mu = 0.15$

Tool holder speed = 5 m/min

Step I *Power required to drive the screw*

From Eq. (6.2),

$$d_m = d - 0.5p = 48 - 0.5(8) = 44 \text{ mm}$$

$$\tan\alpha = \frac{l}{\pi d_m} = \frac{8}{\pi(44)} = 0.0579 \text{ or } \alpha = 3.312°$$

$$\tan\phi = \mu = 0.15 \text{ or } \phi = 8.531°$$

From Eq. (6.6),

$$M_t = \frac{Wd_m}{2}\tan(\phi + \alpha)$$

$$= \frac{(500)(44)}{2}\tan(8.531 + 3.312)$$

$$= 2306.64 \text{ N-mm.}$$

From Eq. (6.18),

$$(M_t)_c = \frac{\mu_c W}{4} \cdot (D_0 + D_i) = \mu_c W r_m$$

or $(M_t)_c = \mu_c W r_m = (0.15)(500)(40) = 3000$ N-mm.

$$(M_t)_t = M_t + (M_t)_c = 2306.64 + 3000$$

$$= 5306.64 \text{ N-mm.}$$

The nut travels at a speed of 5 m/min along with the tool holder. If n is the rpm of the screw, the relationship between the pitch, speed and rpm can be written as,

$$n = \frac{(5\times10^3)}{p} = \frac{(5\times10^3)}{8} = 625 \text{ rpm}$$

Power required to drive the screw is given by,

$$kW = \frac{2\pi n(M_t)_t}{60\times10^6} = \frac{2\pi(625)(5306.64)}{60\times10^6} = 0.35 \qquad \text{(i)}$$

Step II *Efficiency of mechanism*

From Eq. (6.20),

$$\eta_o = \frac{Wl}{2\pi(M_t)_t} = \frac{(500)(8)}{2\pi(5306.64)} = 0.12 \text{ or } 12\% \qquad \text{(ii)}$$

Example 6.8 *It is required to design a double-start screw with square threads for the C-clamp shown in Fig. 6.15(a). The maxi-*

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(ii) the length of the nut; and
(iii) the size of the balls.

Solution

Given For screw, $S_{yt} = 400$ N/mm^2 $(fs) = 3$
For punched holes, $d = 50$ mm $t = 1.5$ mm
$S_{us} = 375$ N/mm^2
Mass density of balls = 7280 kg/m^3 $R_m = 500$ mm

Step I *Diameter, pitch and number of starts for screw*
The force W required to shear the sheet is given by

$$W = \pi dtS_{us} = \pi(50)\,(1.5)\,(375) = 88\,357.29 \text{ N}$$

The lower portion of the screw is subjected to compressive stress. Assuming,

$$S_{yc} = S_{yt} = 400 \text{ N/mm}^2$$

$$\sigma_c = \frac{S_{yc}}{(fs)} = \frac{400}{3} = 133.33 \text{ N/mm}^2$$

The core diameter d_c of the screw is given by

$$d_c = \sqrt{\frac{4W}{\pi\sigma_c}} = \sqrt{\frac{4(88357.29)}{\pi(133.33)}} = 29.05 \text{ mm}$$

In addition to compressive stress, the screw is subjected to torsional shear stress. Therefore, the core diameter is to be increased. As a first trial value, a nominal diameter of 50 mm with 8 mm pitch is recommended for the screw. Since the punch is to be raised by 5 mm during the return stroke of one-quarter revolution, the minimum lead should be 20 mm. The screw is therefore triple-threaded.

$$l = 3p = 3(8) = 24 \text{ mm}$$
$$d_m = d - 0.5p = 50 - 0.5(8) = 46 \text{ mm}$$

Step II *Check for stresses in screw*

$$\tan\alpha = \frac{l}{\pi d_m} = \frac{24}{\pi(46)} \quad \text{or} \quad \alpha = 9.429°$$

Assuming the coefficient of friction as 0.15,

$$\tan\phi = \mu = 0.15 \quad \text{or} \quad \phi = 8.531°$$

$$M_t = \frac{Wd_m}{2}\tan(\phi + \alpha)$$

$$= \frac{(88\,357.29)\,(46)}{2}\tan(8.531 + 9.429)$$

$$= 658\,739.29 \text{ N-mm}$$

Stresses in screw body

$$d_c = d - p = 50 - 8 = 42 \text{ mm}$$

$$\sigma_c = \frac{W}{\frac{\pi}{4}d_c^2} = \frac{88\,357.29}{\frac{\pi}{4}(42)^2} = 63.78 \text{ N/mm}^2$$

$$\tau = \frac{16\,M_t}{\pi d_c^3} = \frac{16(658\,739.29)}{\pi(42)^3} = 45.28 \text{ N/mm}^2$$

$$\tau_{\max.} = \sqrt{\left(\frac{\sigma_c}{2}\right)^2 + (\tau)^2}$$

$$= \sqrt{\left(\frac{63.78}{2}\right)^2 + (45.28)^2} = 55.38 \text{ N/mm}^2$$

According to the maximum shear stress theory,

$$S_{sy} = 0.5\,S_{yt} = 0.5\,(400) = 200 \text{ N/mm}^2$$

$$(fs) = \frac{S_{sy}}{\tau_{\max.}} = \frac{200}{55.38} = 3.61$$

Therefore, the factor of safety is more than 3. The efficiency of the screw is given by

$$\eta = \frac{\tan\alpha}{\tan(\phi + \alpha)} = \frac{\tan(9.429)}{\tan(8.531 + 9.429)}$$

$$= 0.5123 \quad \text{or} \quad 51.23\%$$

Step III *Length of nut*
For steel screw with a bronze nut, the unit bearing pressure (S_b) is taken as 18 N/mm^2.

From Eq. (6.23),

$$z = \frac{4W}{\pi S_b\,(d^2 - d_c^{\,2})} = \frac{4\,(88\,357.29)}{\pi\,(18)\,(50^2 - 42^2)}$$

$$= 8.49 \quad \text{or} \quad 9 \text{ threads}$$

Length of nut = $zp = 9\,(8) = 72$ mm

The shear stresses in the threads of the screw and the nut are as follows:

$$t = p/2 = 8/2 = 4 \text{ mm}$$

$$\tau_s = \frac{W}{\pi d_c tz} = \frac{88\,357.29}{\pi(42)(4)(9)} = 18.6 \text{ N/mm}^2$$

$$\tau_n = \frac{W}{\pi dtz} = \frac{88\,357.29}{\pi(50)(4)(9)} = 15.62 \text{ N/mm}^2$$

Step IV *Size of balls*
Since the process of shearing is complete when the punch penetrates through half the thickness of the sheet, the work done by the punch is given by,
work done = punching force × 0.5 (sheet thickness)

$$= 88\,357.29 \times 0.5\,(1.5) = 66\,268 \text{ N-mm}$$
$$= 66.268 \text{ N-m or J}$$

The efficiency of the screw is 51.23% and the collar friction is to be neglected. Therefore, the input or the work done by the balls is given by,

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There are additional stresses due to torsional and bending moments. The diameter should be increased to account for these stresses. As a first trial, a square threaded screw with 60 mm nominal diameter and 9 mm pitch (Table 6.1) is selected.

Trial No. 1

$d = 60$ mm $\quad p = 9$ mm

From Eqs (6.1) and (6.2),

$$d_c = d - p = 60 - 9 = 51 \text{ mm}$$

$$d_m = d - 0.5p = 60 - 0.5(9) = 55.5 \text{ mm}$$

It is assumed that the screw has single-start threads.

$$\therefore \quad l = p = 9 \text{ mm}$$

$$\tan \alpha = \frac{l}{\pi d_m} = \frac{9}{\pi (55.5)} \quad \text{or} \quad \alpha = 2.95°$$

The coefficient of friction between the steel screw and bronze nut is normally taken as 0.1. The maximum possible value of the coefficient of friction is 0.18. This occurs when the friction is maximum on account of poor lubrication. We will consider the worst case where the operator is careless about the lubrication of the screw.

$\tan \phi = \mu = 0.18 \quad$ or $\quad \phi = 10.20°$

Since $\phi > \alpha$ the screw is self locking.

From Eq. (6.6),

$$M_t = \frac{W d_m}{2} \tan (\phi + \alpha)$$

$$= \frac{(100 \times 10^3)\ (55.5)}{2} \tan (10.20 + 2.95)$$

$$= 648\ 316.03 \text{ N-mm}$$

The torque diagram for the screw is shown in Fig. 6.19(b). It is important to note the following points with respect to this diagram,

(i) The portion of the screw between the nut and the axis of the handle is subjected to torque M_t only.

(ii) The portion of the screw between the cup and the axis of the handle is subjected to torque $(M_t)_c$ only.

(iii) The external torque $(P \times l_h)$ exerted at the axis of the handle consists of addition of M_t plus $(M_t)_c$.

(iv) No cross-section of the screw is subjected to addition of M_t plus $(M_t)_c$.

At the section-*XX*,

$$\tau = \frac{16 M_t}{\pi d_c^3} = \frac{16 (648\ 316.03)}{\pi (51)^3}$$

$$= 24.89 \text{ N/mm}^2 \qquad \text{(i)}$$

$$\sigma_c = \frac{W}{\left(\frac{\pi}{4} d_c^2\right)} = \frac{100 \times 10^3}{\frac{\pi}{4} (51)^2} = 48.95 \text{ N/mm}^2 \qquad \text{(ii)}$$

The portion of the screw with the side view of the handle is shown in Fig. 6.19(d). The hand force P acting on the handle causes a bending moment at the section-*XX*. The bending moment is given by,

$$M_b = P \times l_1$$

The lifting height of the jack is 500 mm and the distance l_1 can be assumed as,

$$l_1 = 500 + 50 + 20 = 570 \text{ mm}$$

$$M_b = P \times l_1 = (0.9 \times 2 \times 400)\ (570)$$

$$= 410\ 400 \text{ N-mm}$$

Therefore,

$$\sigma_b = \frac{32 M_b}{\pi d_c^3} = \frac{32\ (410\ 400)}{\pi (51)^3} = 31.51 \text{ N/mm}^2 \qquad \text{(iii)}$$

Figure 6.20(c) shows the superimposition of direct compressive stress and bending stresses as determined by equations (ii) and (iii). In this figure, tensile and compressive stresses are shown as positive and negative respectively. The resultant stresses are compressive. A compressive stress

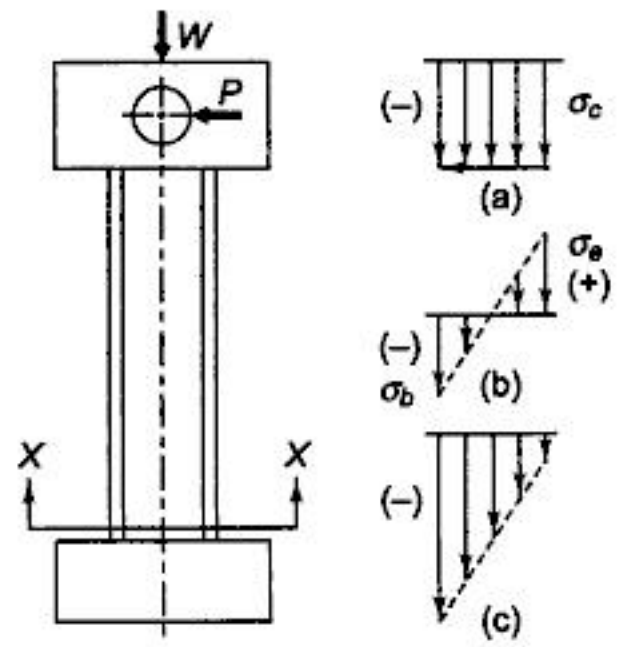

Fig. 6.20 *Stresses at Section XX: (a) Direct Compressive stress (b) Bending Stresses (c) Resultant Stresses*

closes the crack, while tensile stress opens the crack. In general, failure is associated with tensile

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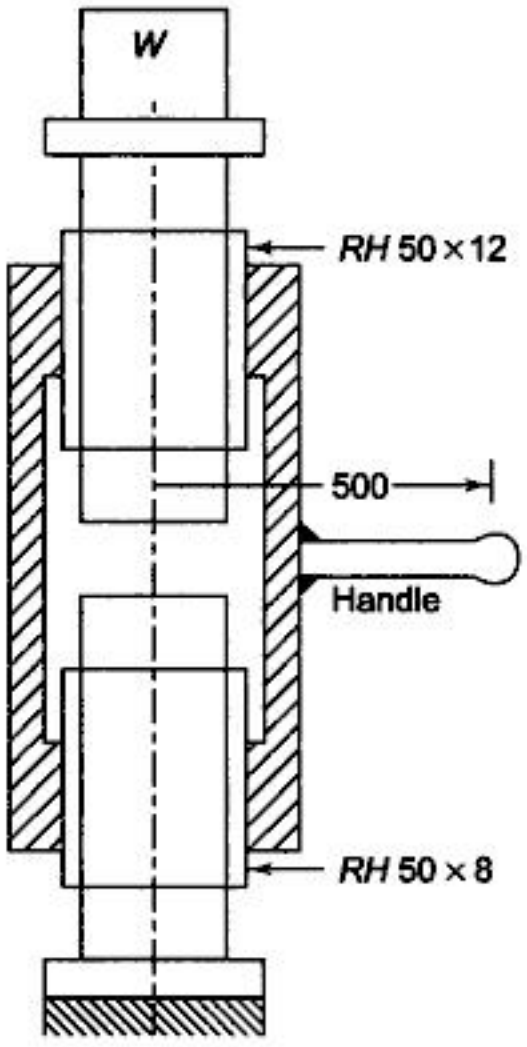

Fig. 6.27 *Differential Screw Jack*

Solution

Given For handle, $R = 500$ mm $P = 100$ N

$\mu = 0.15$

Upper screw = RH 50 × 12

Lower screw = RH 50 × 8

Step I *Load capacity of screw jack*

In one revolution of the nut, the load is raised through a distance equal to the difference in the pitch of the two screws.

Output = work done on $W = W(p_1 - p_2)$

$= W(12 - 8) = (4\ W)$ N-mm

For the upper screw,

$$d_m = d - 0.5p = 50 - 0.5(12) = 44 \text{ mm}$$

$$\tan\alpha = \frac{l}{\pi d_m} = \frac{12}{\pi(44)} \quad \text{or} \quad \alpha = 4.962°$$

$$\tan\phi = \mu = 0.15 \quad \text{or} \quad \phi = 8.531°$$

$$M_{t1} = \frac{Wd_m}{2}\tan(\phi + \alpha)$$

$$= \frac{W(44)}{2}\tan(8.531 + 4.962)$$

$$= (5.279\ W) \text{ N-mm}$$

For the lower screw,

$$d_m = d - 0.5\,p = 50 - 0.5(8) = 46 \text{ mm}$$

$$\tan\alpha = \frac{l}{\pi d_m} = \frac{8}{\pi(46)} \quad \text{or} \quad \alpha = 3.169°$$

When the nut is rotated, the upper screw moves in the upward direction. As far as the lower screw is concerned, it is the nut, which moves in the upward direction. In other words, the lower screw is moving in a downward direction relative to the nut and the load. Therefore,

$$M_{t2} = \frac{Wd_m}{2}\tan(\phi - \alpha)$$

$$= \frac{W(46)}{2}\tan(8.531 - 3.169)$$

$$= (2.159\ W) \text{ N-mm}$$

Adding the torques of upper and lower screws and equating with external torque,

$$100(500) = 5.279\ W + 2.159\ W$$

$$\therefore \quad W = 6722 \text{ N} \qquad \text{(i)}$$

Step II *Efficiency of the screw jack*

$$\eta_o = \frac{4W}{2\pi(100 \times 500)} = \frac{4(6722)}{2\pi(100 \times 500)}$$

$$= 0.0856 \quad \text{or} \quad 8.56\% \qquad \text{(ii)}$$

6.17 RECIRCULATING BALL SCREW

A recirculating ball screw, as shown in Fig. 6.28, consists of a screw and a nut, the surfaces of which are separated by a series of balls. The screw and the nut have approximately semi-circular thread profiles instead of conventional square or trapezoidal shape. As the screw is rotated, the balls advance in the grooves in the nut and the screw. They are collected at the end of the nut and returned back. The recirculating ball screw is also called *ball bearing screw* or simply *ball screw*. Such screws are preloaded and give accurate motion due to elimination of the backlash. There is no heat generation due to negligible friction. Recirculating ball screws can be used for high speeds even up to 10 m/min. The balls, screw and nut are subjected to contact stresses. They are usually made of nickel chromium steel and heat treated to a surface hardness of 58 to 65 HRC.

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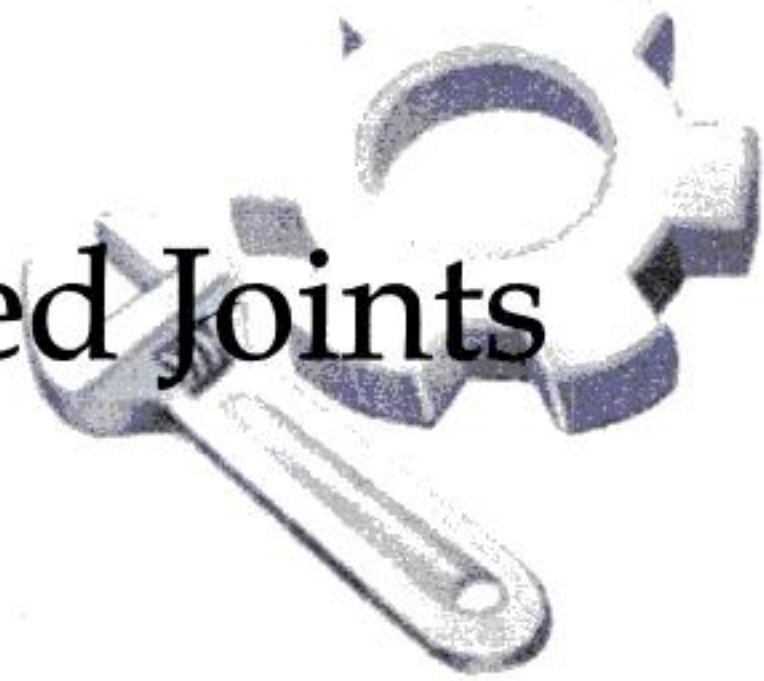

Threaded Joints

Chapter 7

7.1 THREADED JOINTS

Threaded joint is defined as a separable joint of two or more machine parts that are held together by means of a threaded fastening such as a bolt and a nut. The salient features of this definition are as follows:

(i) Threaded joints are used to hold two or more machine parts together. These parts can be dismantled, if required, without any damage to machine parts or fastening. Therefore, threaded joints are *detachable joints*, unlike welded joints.

(ii) Thread is the basic element of these joints. The thread is formed by cutting a helical groove on the surface of a cylindrical rod or cylindrical hole. The threaded element can take the shape of bolt and nut, screw or stud. Sometimes, threads are cut on the parts to be joined.

Threaded joints are extensively used in mechanical assemblies. It has been observed that over 60% of the parts have threads. The popularity of threaded joints is due to certain advantages offered by them. The advantages of threaded joints are as follows:

(i) The parts are held together by means of a large clamping force. There is wedge action at the threads, which increases the clamping force. There is no loosening of the parts. Therefore, threaded joints are '*reliable*' joints.

(ii) The parts are assembled by means of a spanner. The length of the spanner is large compared with the radius of the thread. Therefore, the mechanical advantage is more and force required to tighten the joint is small.

(iii) Threaded joints have small overall dimensions resulting in compact construction.

(iv) The threads are self-locking. Therefore, threaded joints can be placed in any position—vertical, horizontal or inclined.

(v) Threaded fasteners are economical to manufacture. Their manufacturing is simple. High accuracy can be maintained for the threaded components.

(vi) The parts joined together by threaded joints can be detached as and when required. This requirement is essential in certain applications for the purpose of inspection, repair or replacement.

(vii) Threaded fasteners are standardised and a wide variety is available for different operating conditions and applications.

There are certain disadvantages of threaded joints. They are as follows:

(i) Threaded joints require holes in the machine parts that are to be clamped. This results in

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(c), (d) and (e). They are called *fillister, button, flat,* and *hexagonal socket heads*. A fillister head cap screw has cylindrical head with a slot for the screwdriver. A button head cap screw has spherical head with similar slot for the screwdriver. A flat head cap screw has conical head with a slot. Heads with the slot for the screwdriver are used for cap screws of small sizes, which require small torque for tightening. Fillister heads are used for cap screws with the head sunk in a counterbore. If the head is to be flush with the surface and the part being fastened has insufficient thickness to accommodate a fillister head, a cap screw with flat head is employed. When it is not possible to sink the head and a better appearance is required, cap screws with button or round head are used.

Cap screw with hexagonal socket head is shown in Fig. 7.2(e). These cap screws are tightened by a simple wrench made of hexagonal bar bent at right angles. The important advantage of cap screw with hexagonal socket head is that the maximum tightening torque permitted by the socket wrench corresponds to the strength of the screw. Therefore, the head cannot be twisted off during tightening. Cap screws with heads engaged internally and from the end face have the following advantages:

(i) Such cap screws can be installed in confined spaces and in counter bored holes.
(ii) They result in better appearance for the product.
(iii) They reduce the overall dimensions of the product resulting in compact construction.
(iv) There is convenience in cleaning or wiping off the surface of the product.

7.4 SETSCREWS

Setscrew is used to prevent relative motion between two parts. The threaded portion of the setscrew passes through a tapped hole in one of the parts and the end of the screw presses against the other part. The end of the screw is called the *point* of the screw. The friction between the point and the part to be held prevents relative motion of that part with respect to the part through which the setscrew is screwed. Setscrew can be used instead of key to prevent relative motion between the hub and the shaft in small power transmission. Setscrews differ from cap screws in the following respects:

(i) Setscrews are subjected to compressive force only. Cap screws are subjected to tensile and shear forces.
(ii) Setscrew transmits force from threaded component to the other mating component by means of screw point. In cap screw, the force is transmitted by the head.
(iii) Setscrews are short and threaded over full length of the shank compared with the cap screws.

Setscrews with different types of points and heads are illustrated in Fig. 7.3. Some setscrews have hexagonal or square head as shown in Fig. 7.3(b) and (c), but the majority do not have heads. Usually, they have a slot for the screwdriver as shown in Fig. 7.3(a) and (e) or a hexagonal

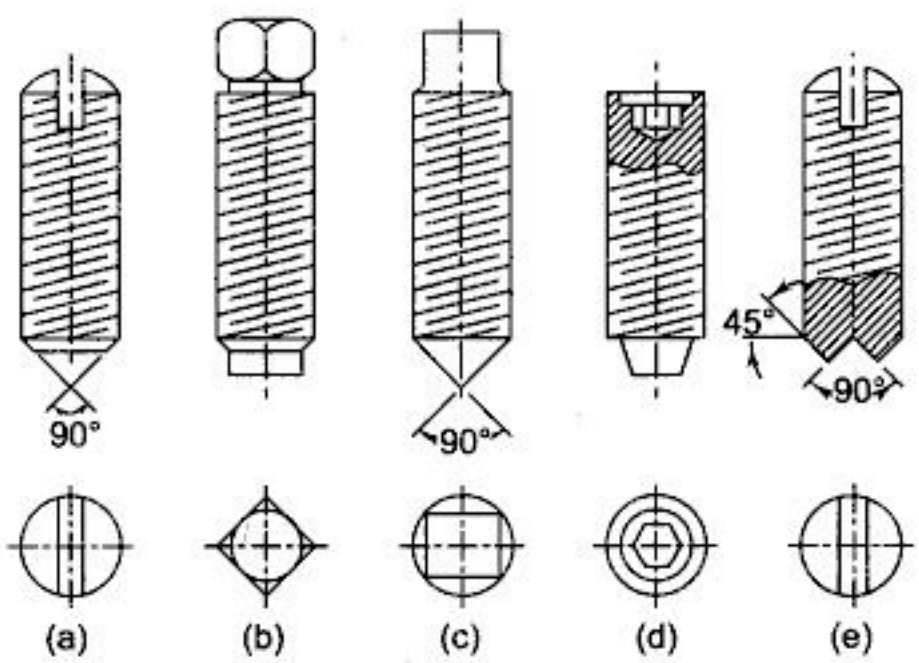

Fig. 7.3 *Types of Set Screws*

socket shown in Fig. 7.3(d). Setscrew used in rotating parts has no head. On the other hand, setscrew for stationary parts may have hexagonal or square heads. The types of point used in setscrew are as follows:

(i) Flat Point (Fig. 7.3 a) Flat point is used when the lateral force, which tends to displace one part with respect to another, is randomly applied. It is also used when the part with tapped hole into which

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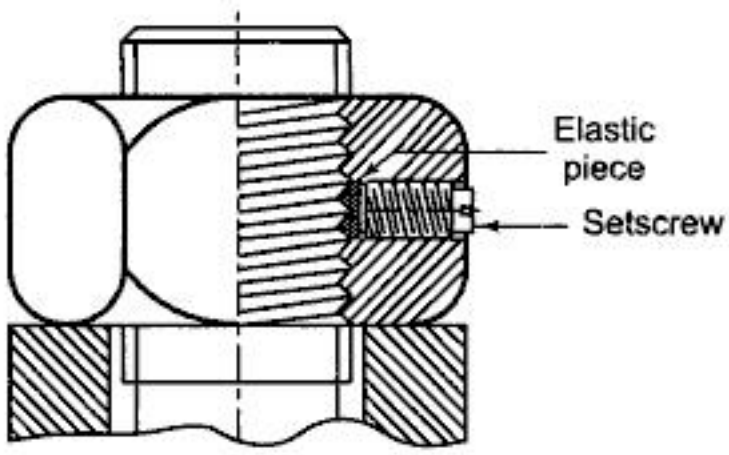

Fig. 7.8 *Locking with Setscrew*

Locking by spring washer is common method to prevent loosening of threads. The spring washer consists of a ring made of hardened steel with a cut at an angle of 15° with the ring axis as shown in Fig. 7.9(b). The ends of the washer at the cut are slightly separated and provided with sharp teeth. During tightening, the spring washer is compressed and its teeth bite into the contacting surface of the nut on one side and the base on the other. Thus, the nut gets fixed in the base. The principle of locking by means of spring washer is illustrated in Fig. 7.9(c). The action of the spring washer is more effective when the contacting surfaces are soft. The cut in the spring washer has '*left-hand*' inclination for right-hand threads and '*right-hand*' inclination for left-hand threads.

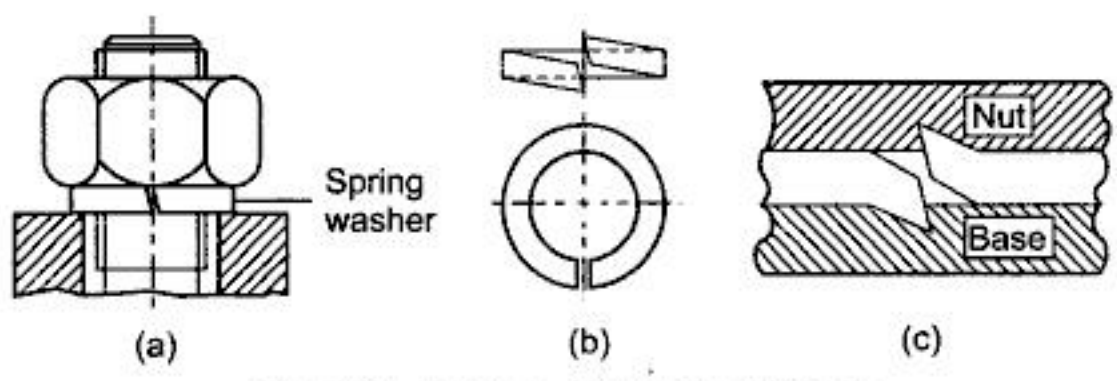

Fig. 7.9 *Locking with Spring Washer*

7.7 TERMINOLOGY OF SCREW THREADS

The right-hand threads are always used unless there is special reason for requiring left-hand thread. Unless and otherwise stated, specifications for threads imply right-hand threads. When the screw is vertical, the thread lines slope upward from left to right in case of right-hand threads. On the other hand, the thread lines slope downward from left to right in case of left-hand threads.

Various dimensions of external and internal threads are illustrated in Fig. 7.10. They are as follows:

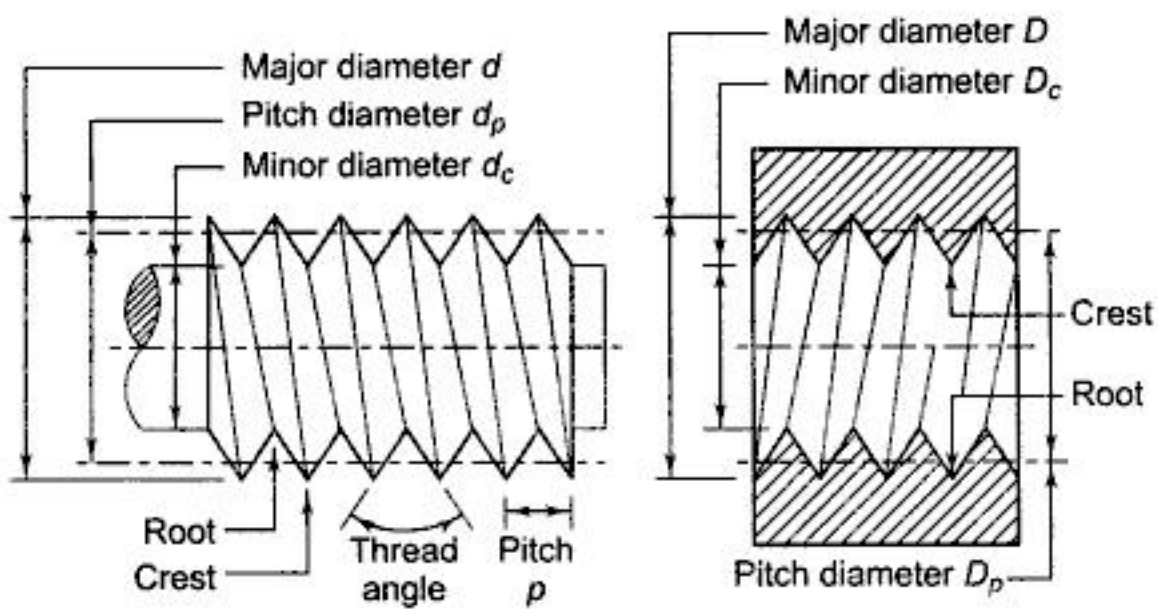

Fig. 7.10 *Terminology of Threads*

(i) Major Diameter The major diameter is the diameter of an imaginary cylinder that bounds the crest of an external thread (d) or the root of an internal thread (D). The major diameter is the largest diameter of the screw thread. It is also called the *nominal diameter* of the thread.

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There are two methods for making threads, viz., *thread cutting* and *thread rolling*. Thread cutting is done on automatic machines called '*screw*' machines. In the thread rolling method, threads are formed by rolling the bar stock between dies which depress part of the material to form the root of the thread and which force the remaining material up the top to form the crest of threads. Therefore, the outside diameter of the thread is more than the bar stock on which it was rolled. Thread rolling is a superior method of making threads. The advantages of thread rolling over thread cutting are as follows:

(i) Cold forming induces residual compressive stresses on the thread surface, which improve fatigue strength of the bolt.
(ii) Cold forming creates radii at the root and the crest and reduces stress concentration.
(iii) Thread cutting results in cutting the fibre lines of the original bar stock. In thread rolling, the fibre lines are rearranged to suit the thread shape.
(iv) Compared with cut threads, rolled threads have less waste as no material is removed.

All these factors contribute to increased popularity of rolled threads as compared to cut threads.

7.10 BOLTED JOINT – SIMPLE ANALYSIS

A bolted joint subjected to tensile force P is shown in Fig. 7.12. The cross-section at the core diameter d_c is the weakest section. The maximum tensile stress in the bolt at this cross-section is given by,

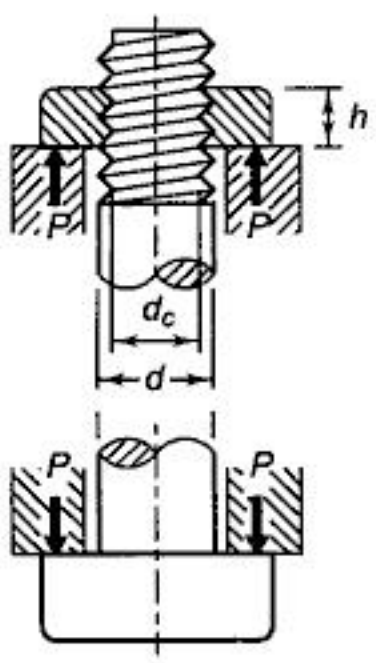

Fig. 7.12 *Bolt in Tension*

$$\sigma_t = \frac{P}{\left(\frac{\pi}{4} d_c^2\right)} \qquad (7.2)$$

The height of the nut h can be determined by equating the strength of the bolt in tension with the strength in shear. The procedure is based on the following assumptions:

(i) Each turn of the thread in contact with the nut supports an equal amount of load.
(ii) There is no stress concentration in the threads.
(iii) The yield strength in shear is equal to half of the yield strength in tension ($S_{sy} = 0.5 S_{yt}$).
(iv) Failure occurs in the threads of the bolt and not in the threads of the nut.

Substituting,

$$\sigma_t = \frac{S_{yt}}{(fs)}$$

in Eq. (7.2), the strength of the bolt in tension is given by,

$$P = \frac{\pi}{4} d_c^2 \left(\frac{S_{yt}}{fs}\right) \qquad \text{(a)}$$

The threads of the bolt in contact with the nut are sheared at the core diameter d_c. The shear area is equal to ($\pi d_c h$), where h is the height of the nut. The strength of the bolt in shear is given by,

$$P = (\pi d_c h)\left(\frac{S_{sy}}{fs}\right)$$

$$= (\pi d_c h)\left(\frac{S_{yt}}{2fs}\right) \qquad \text{(b)}$$

Equating (a) and (b),

$$h = 0.5 d_c$$

Assuming ($d_c = 0.8d$),

$$h = 0.4d$$

Therefore, for standard coarse threads, the threads are equally strong in failure by shear and failure by tension, if the height of the nut is approximately 0.4 times of the nominal diameter of the bolt. The height of the standard hexagonal nut is ($0.8d$). Hence, the threads of the bolt in the standard nut will not fail by shear.

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$$= 8139.41 \text{ N}$$

$$P_2 = \sqrt{(P_2'' \cos\theta + P_2')^2 + (P_2'' \sin\theta)^2}$$

$$= \sqrt{[10\,000 \cos(36.87) + 2500]^2 + [10\,000 \sin(36.87)]^2}$$

$$= 12093.38 \text{ N}$$

Bolts 2 and 4 are subjected to maximum shear forces.

Step IV *Size of bolts*

$$\tau = \frac{P_2}{A} \qquad 80 = \frac{12093.38}{\frac{\pi}{4} d_c^2}$$

$\therefore \quad d_c = 13.87$ mm

From Eq. (7.4)

$$d = \frac{d_c}{0.8} = \frac{13.87}{0.8} = 17.34 \text{ or } 18 \text{ mm}$$

From Table 7.1, the standard size of the bolts is M 20.

Example 7.4 *A steel plate subjected to a force of 5 kN and fixed to a channel by means of three identical bolts is shown in Fig. 7.19(a). The bolts are made from plain carbon steel 45C8 (S_{yt} = 380 N/mm^2) and the factor of safety is 3. Specify the size of bolts.*

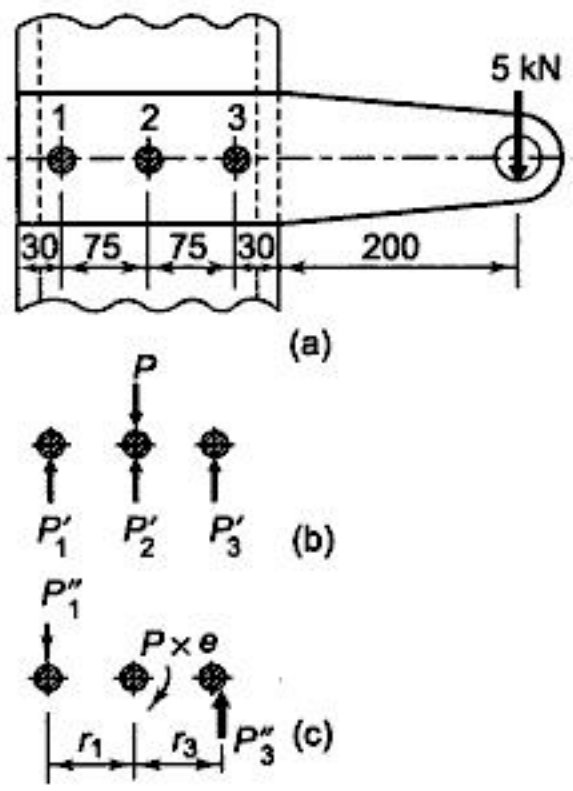

Fig. 7.19

Solution

Given $P = 5$ kN $\quad S_{yt} = 380$ N/mm^2 $\quad (fs) = 3$

Step I *Permissible shear stress*

$$\tau = \frac{S_{sy}}{(fs)} = \frac{0.5\ S_{yt}}{(fs)} = \frac{0.5\ (380)}{3} = 63.33 \text{ N/mm}^2$$

Step II *Primary and secondary shear forces*

The centre of gravity of three bolts will be at the centre of bolt-2.

The primary and secondary shear forces are shown in Fig. 7.19(b) and (c).

$$P_1' = P_2' = P_3' = \frac{P}{3} = \frac{5000}{3} = 1666.67 \text{ N}$$

$$P_1'' = P_3'' = \frac{(Pe)(r_1)}{(r_1^2 + r_3^2)} = \frac{(5000 \times 305)(75)}{(75^2 + 75^2)}$$

$$= 10166.67 \text{ N}$$

Step III *Resultant shear force*

The resultant shear force on the bolt 3 is maximum.

$$P_3 = P_3' + P_3'' = 1666.67 + 10166.67 = 11833.34 \text{ N}$$

Step IV *Size of bolts*

$$\tau = \frac{P_3}{A} \quad \text{or} \quad 63.33 = \frac{11833.34}{\frac{\pi}{4} d_c^2}$$

$\therefore \quad d_c = 15.42$ mm

From Eq. (7.4),

$$d = \frac{d_c}{0.8} = \frac{15.42}{0.8} = 19.28 \quad \text{or} \quad 20 \text{ mm}$$

The standard size of the bolts is M20.

7.12 ECCENTRIC LOAD PERPENDICULAR TO AXIS OF BOLT

A bracket, fixed to the steel structure by means of four bolts, is shown in Fig. 7.20(a). It is subjected to eccentric force P at a distance e from the structure. The force P is perpendicular to the axis of each bolt. The lower two bolts are denoted by 2, while the upper two bolts by 1. In this analysis, the following assumptions are made:

(i) The bracket and the steel structure are rigid.

(ii) The bolts are fitted in reamed and ground holes.

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Solution

Given $P = 50$ kN $l = 250$ mm
$(\sigma_t)_{max.} = 100$ N/mm^2

Step I *Maximum tensile force*

The force P tends to tilt the bracket about edge C. Each bolt is stretched by an amount (δ), which is proportional to its distance from the tilting edge as shown in Fig. 7.22 (b).

$\delta_1 \propto l_1 \qquad \delta_2 \propto l_2 \qquad \delta_3 \propto l_3$

Also,

force $\propto$ stress because $(P = \sigma A)$
stress $\propto$ strain because $(\sigma = E\epsilon)$
strain $\propto$ stretch because $(\epsilon = \delta/l)$

Therefore, it can be concluded that the force induced in any bolt due to the tendency of the bracket to tilt about the edge C is proportional to its distance from the tilting edge. Therefore,

$$P_1 = Cl_1 \qquad P_2 = Cl_2 \qquad P_3 = Cl_3 \qquad \text{(i)}$$

where C is the constant of proportionality. The bolts, denoted by 1, are subjected to maximum tensile force because of the farthest distance from the tilting edge. Equating the moment of resisting forces to the moment due to external force about C,

$$Pl = 2P_1 l_1 + 2P_2 l_2 + 2P_3 l_3 \qquad \text{(ii)}$$

Substituting (i) in (ii),

$$Pl = 2C\ (l_1^2 + l_2^2 + l_3^2)$$

$$C = \frac{Pl}{2(l_1^2 + l_2^2 + l_3^2)} \qquad \text{(iii)}$$

From (i) and (iii), $P_1 = \dfrac{Pl\ l_1}{2\,(l_1^2 + l_2^2 + l_3^2)}$

Substituting numerical values,

$$P_1 = \frac{(50 \times 10^3)\ (250)\ (300)}{2\ (300^2 + 200^2 + 100^2)} = 13\,392.86 \text{ N}$$

The shear stress is to be neglected.

Step II *Size of bolts*

$$P_1 = A\ (\sigma_t)_{max.} \quad \text{or} \quad 13\,392.86 = A\ (100)$$

$\therefore \quad A = 133.93$ mm^2

From Table 7.1, the standard size of the bolts is $M16$ ($A = 157$ mm^2).

Example 7.8 *A crane-runway bracket is fastened to the roof truss by means of two identical bolts as shown in Fig. 7.23. Determine the size of the bolts, if the permissible tensile stress in the bolts is limited to 75 N/mm^2.*

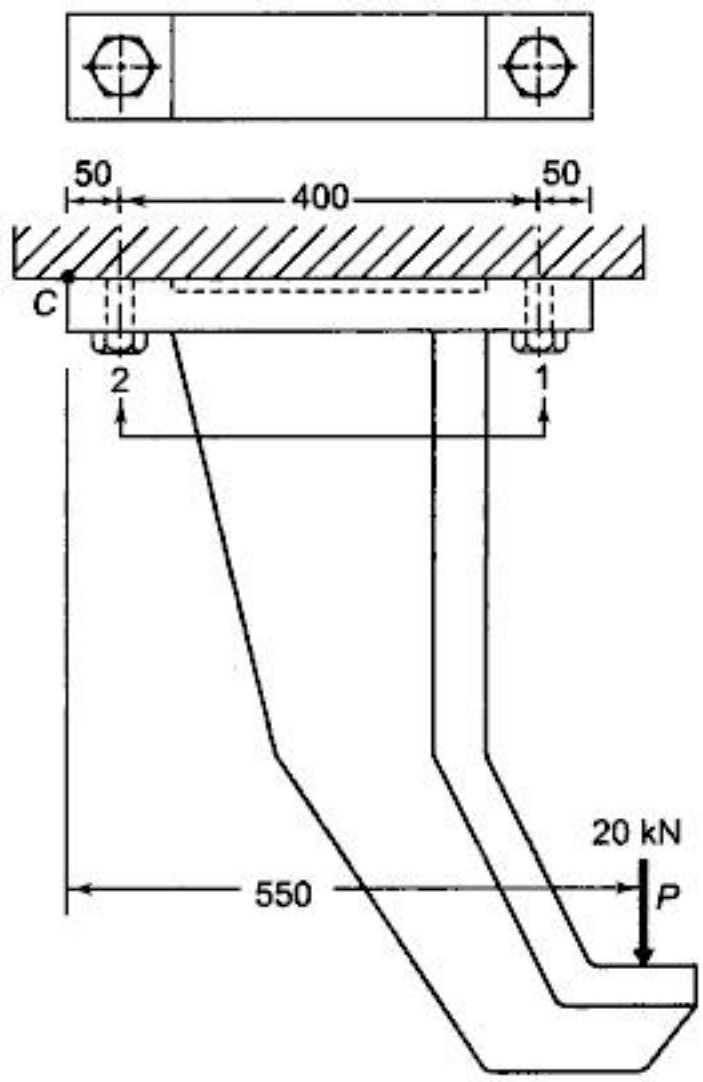

Fig. 7.23

Solution

Given $P = 20$ kN $e = 550$ mm
$(\sigma_t)_{max.} = 75$ N/mm^2

The bolts are subjected to tensile stress on account of the following two factors:

(i) direct tensile stress because the force P is parallel to the axis of the bolts; and
(ii) tensile stress due to the tendency of the bracket to tilt about the edge C due to eccentricity of the force P.

Step I *Direct tensile force*

$$P_1 = P_2 = \frac{P}{\text{(No. of bolts)}} = \frac{20 \times 10^3}{2} = 10\,000 \text{ N} \qquad \text{(i)}$$

Step II *Tensile force due to the tendency of bracket to tilt*

The moment ($P \times 550$) tends to tilt the bracket about the edge C. Suppose P_1'' and P_2'' are resisting forces set up in bolts 1 and 2 respectively. As we know,

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If $P_1\ P_2 \ldots$ are the resisting forces induced in the bolts,

$$P_1 \propto l_1$$

or,

$$P_1 = Cl_1 \quad P_2 = Cl_2 \quad P_3 = Cl_3 \quad P_4 = Cl_4 \qquad \text{(a)}$$

where C is the constant of proportionality. Equating the moment due to the external force P about C with the moments due to resisting forces,

$$Pl = P_1 l_1 + P_2 l_2 + P_3 l_3 + P_4 l_4 \qquad \text{(b)}$$

From (a) and (b),

$$Pl = C\ (l_1^2 + l_2^2 + l_3^2 + l_4^2)$$

$$C = \frac{Pl}{(l_1^2 + l_2^2 + l_3^2 + l_4^2)} \qquad \text{(c)}$$

From, (a) and (c), the force acting on the bolt 1 is given by,

$$P_1 = \frac{Pl\, l_1}{(l_1^2 + l_2^2 + l_3^2 + l_4^2)} \qquad \text{(d)}$$

Suppose,

a = radius of the flange

b = radius of pitch circle of the bolts.

From Fig. 7.26(b),

$$l_1 = a - b\cos\alpha$$

$$l_2 = a + b\sin\alpha$$

$$l_3 = a + b\cos\alpha$$

$$l_4 = a - b\sin\alpha$$

Therefore,

$$(l_1^2 + l_2^2 + l_3^2 + l_4^2) = 4a^2 + 2b^2 \qquad \text{(e)}$$

From (d) and (e),

$$P_1 = \frac{Pl(a - b\cos\alpha)}{2(2a^2 + b^2)} = \frac{2Pl(a - b\cos\alpha)}{4(2a^2 + b^2)} \qquad \text{(f)}$$

Four bolts are considered in the above analysis. If the procedure is repeated for n equally spaced bolts, we get the general expression in the following form:

$$P_1 = \frac{2\,Pl(a - b\cos\alpha)}{n(2a^2 + b^2)} \qquad (7.15)$$

The force P_1 has maximum value when the term (cos α) has minimum value. The minimum value of (cos α) is (–1), when ($\alpha = 180°$). With reference to Fig. 7.26 (b), the bolt 1 will occupy the topmost position, at the farthest distance form C, when $\alpha = 180°$, Substituting $\alpha = 180°$ in Eq. (7.15),

$$P_{max.} = \frac{2Pl(a+b)}{n(2a^2 + b^2)} \qquad (7.16)$$

Equation (7.16) gives absolute maximum value of the force acting on any of the bolts. It should be used for finding out the size of the bolts, when the direction of the external force P can change with respect to the bolts, as in case of the base of a vertical pillar crane. When the direction of the external force P is fixed and known, the maximum load on the bolts can be reduced, so that the two of them can be equally stressed as shown in Fig. 7.26(c). In this particular case, the number of bolts is 4 and the angle α made by the centre line of bolt 2 is 135°. For a general case with n as number of bolts,

$$\alpha = 180 - \beta \qquad \text{(i)}$$

$$\beta = \frac{1}{2}\,\frac{360}{n} \qquad \text{(ii)}$$

$$\therefore \quad \alpha = 180 - \frac{360}{2n} \qquad (7.17)$$

Therefore,

$$\cos\alpha = \cos\left[180 - \frac{360}{2n}\right] = -\cos\left(\frac{360}{2n}\right)$$

$$= -\cos\left(\frac{180}{n}\right)$$

Substituting the above value of (cos α) in Eq. (7.15),

$$P_1 = \frac{2Pl\left[a + b\cos\left(\frac{180}{n}\right)\right]}{n(2a^2 + b^2)} \qquad (7.18)$$

The above equation is applicable only when two bolts are equally stressed. This condition can be satisfied under the following circumstances:

(i) the direction of the external force P is fixed with respect to the bolts;

(ii) the number of bolts is even; and

(iii) two bolts at the top are symmetrically spaced with angle β on either side of the vertical line.

The bolts are usually relieved of shear stresses by using dowel pins as shown in Fig. 7.26(c).

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From Eq. (7.10),

$$P_1'' = \frac{(Pe)\,l_1}{2(l_1^2 + l_2^2)} = \frac{(M_h + M_v)\,l_1}{2(l_1^2 + l_2^2)}$$

$$= \frac{(173\,205.2 + 600\,000)(260)}{2(260^2 + 60^2)}$$

$$= 1411.75 \text{ N} \qquad \text{(v)}$$

From (i) and (v), the tensile force acting on each of the two bolts, denoted by 1, is given by,

$$P_t = P_1 + P_1'' = 1082.53 + 1411.75$$

$$= 2494.28 \text{ N} \qquad \text{(vi)}$$

The tensile stress in the bolt is given by,

$$\sigma_t = \frac{P_t}{A} = \frac{2494.28}{A} \text{ N/mm}^2 \qquad \text{(a)}$$

From (iii), the shear stress in the bolt is given by,

$$\tau = \frac{P_s}{A} = \frac{625}{A} \text{ N/mm}^2 \qquad \text{(b)}$$

Step III *Maximum shear stress in bolts*

From Eq. (7.14),

$$t_{\max.} = \sqrt{\left(\frac{\sigma_t}{2}\right)^2 + \tau^2} = \sqrt{\left(\frac{2494.28}{2A}\right)^2 + \left(\frac{625}{A}\right)^2}$$

$$= \left(\frac{1394.99}{A}\right) \text{N/mm}^2$$

Step IV *Size of bolts*

$$\tau_{\max.} = \frac{1394.99}{A} \quad \text{or} \quad 40 = \frac{1394.99}{A}$$

$$\therefore \quad A = 34.87 \text{ mm}^2$$

From Table 7.1, the standard size of the bolts is M 8 ($A = 36.6$ mm^2).

Example 7.15 *A rigid bracket subjected to a vertical force of 10 kN is shown in Fig. 7.30(a). It is fastened to a vertical stanchion by means of four identical bolts. Determine the size of the bolts by maximum shear stress theory. The maximum permissible shear stress in any bolt is limited to 50 N/mm^2.*

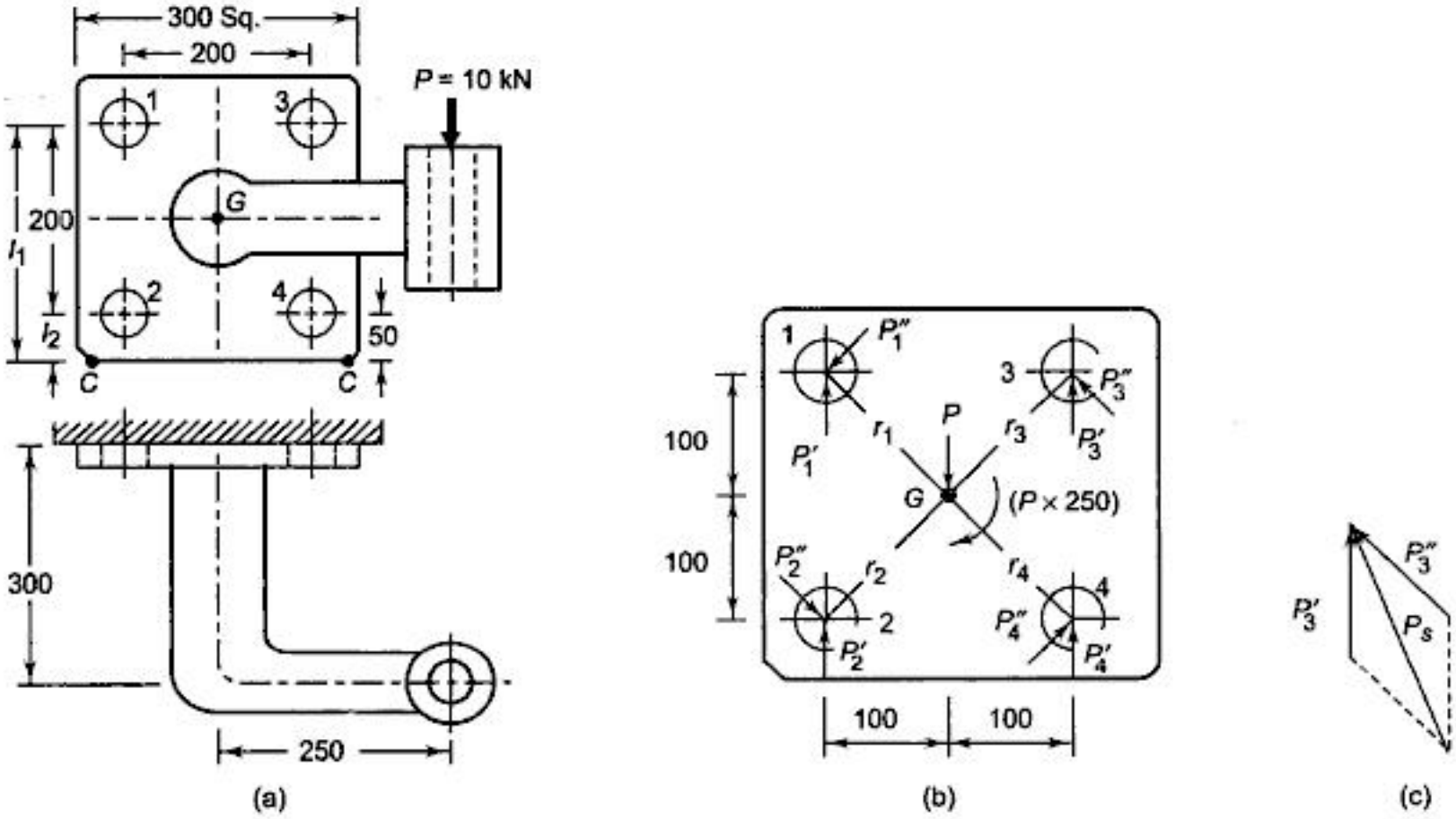

Fig. 7.30

Solution

Given $P = 10$ kN $\quad \tau_{\max.} = 50$ N/mm^2

Step I *Tensile and shear stresses in bolts*

The bolts are subjected to primary and secondary shear stresses and tensile stress due to the tendency of the bracket to tilt about the edge-*CC*.

As shown in Fig. 7.30(b), the external force P acting at a distance of 250 mm from the centre

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term [1/22)P]. The fluctuating term [(1/22)P] is small and neglected. The total load on the bolt will be almost static and the bolt can be designed on the basis of static failure.

(ii) If the initial pre-load is not high enough, the term [(1/22)P] is comparable with the term (P_i). The resultant load on the bolt will be affected by the external load, which is fluctuating and it is required to design the bolt on the basis of endurance limit to avoid fatigue failure.

It is desirable to design the bolts on the basis of static strength rather than on the basis of endurance limit.

Therefore, in automotive applications, the connecting rod bolts are tightened with very high pre-load with the stress approaching the yield point.

Example 7.18 *The assembly of two circular plates clamped together by means of a bolt, which is shown in Fig. 7.35, is subjected to a variable force P varying from 0 to 10 kN. The bolt is made of plain carbon steel 45C8 (S_{ut} = 630 N/mm^2; S_{yt} = 380 N/mm^2 and E = 207000 N/mm^2). The two circular plates are made of aluminium (E = 71000 N/mm^2). The fatigue stress concentration factor is 2.2 and the expected reliability is 90%. The initial pre-load in the bolt is 5 kN. Determine the size of the bolt if the factor of safety is 2.*

Solution

Given $P_i = 5$ kN $P = 0$ to 10 kN $(fs) = 2$

For bolts, $S_{ut} = 630$ N/mm^2 $S_{yt} = 380$ N/mm^2

$E = 207000$ N/mm^2 $K_f = 2.2$ $R = 90\%$

For plates, $E = 71000$ N/mm^2

Step I *Endurance limit stress for bolt*

$$S'_e = 0.5\ S_{ut} = 0.5(630) = 315 \text{ N/mm}^2$$

The surface finish factor is incorporated in the fatigue stress concentration factor. The size factor is 1 for axial load.

For 90% reliability,

$$K_c = 0.897$$

$$K_d = \frac{1}{K_f} = \frac{1}{2.2} = 0.4545$$

$$S_e = K_b K_c K_d S'_e = (1.0)(0.897)(0.4545)(315)$$
$$= 128.42 \text{ N/mm}^2$$

Step II *Construction of Goodman diagram*

As determined in Example 7.16,

$$\left(\frac{k'_b}{k'_b + k'_c}\right) = \frac{(3251.55d^2)}{[3251.55d^2 + 3345.52d^2]} = 0.4929$$

The maximum and minimum forces in the bolt are given by

$$P_{max.} = P_i + (0.4929)\ P$$
$$= 5000 + (0.4929)(10000) = 9929 \text{ N}$$
$$P_{min.} = P_i + (0.4929)\ P$$
$$= 5000 + (0.4929)(0) = 5000 \text{ N}$$
$$P_m = \tfrac{1}{2}\ (P_{max.} + P_{min.}) = \tfrac{1}{2}\ (9929 + 5000)$$
$$= 7464.5 \text{ N}$$
$$P_a = \tfrac{1}{2}\ (P_{max.} - P_{min.}) = \tfrac{1}{2}\ (9929 - 5000)$$
$$= 2464.5 \text{ N}$$

The Goodman diagram for this example is shown in Fig. 7.39.

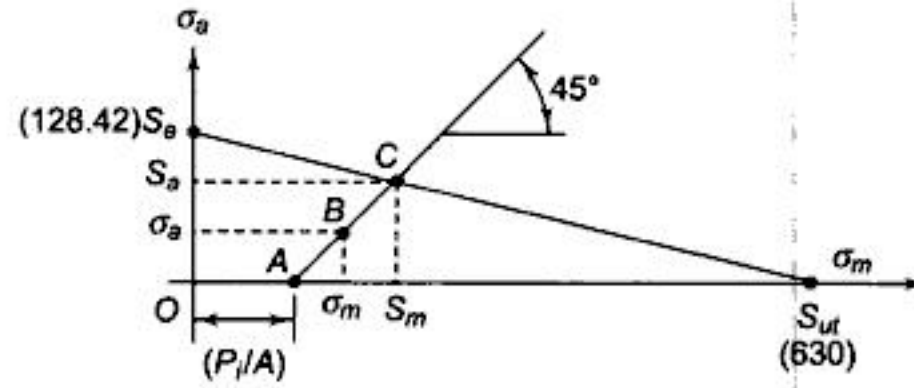

Fig. 7.39

Step III *Permissible stress amplitude*

The coordinates of the point C (S_m, S_a) are obtained by solving the following two equations simultaneously:

$$\frac{S_m}{630} + \frac{S_a}{128.42} = 1 \qquad \text{(a)}$$

$$S_m = \frac{P_i}{A} + S_a = \frac{5000}{A} + S_a \qquad \text{(b)}$$

where A is the tensile stress area of the bolt.

The solution is obtained by Eq. (7.30).

From Eq. (7.30),

$$S_a = \frac{S_{ut} - (P_i / A)}{1 + (S_{ut} / S_e)}$$

$$= \frac{630 - (5000 / A)}{1 + (630 / 128.42)} = \frac{630 - (5000 / A)}{5.9}$$

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welding and electric arc welding. In gas welding, the rate of heating is slow compared with electric arc welding. Therefore, the operator has more control over both heating and cooling rates. Gas welding is preferred for joining thin parts due to low rate of heating.

(iii) Electric arc Welding In this method, the heat required for the fusion is generated by an electric arc between the parts to be joined and an electrode. This electrode can be made of filler metal and consumed as the welding progresses or the electrode can be of non-consumable type. Filler metal is separately fed to the joint in case of non-consumable electrodes. In some cases, the consumable electrode is coated with a flux material that vaporizes to form a gaseous shield around the joint and prevents the molten metal from absorbing oxygen or nitrogen from the atmosphere. If not prevented, the absorption of oxygen or nitrogen adversely affects the mechanical properties of the weld. Electric arc welding is popular in machine building industries as well as for structural work because of the consistently high quality of welding.

Welding processes that use a combination of heat and pressure to join the two parts are classified into the following two groups:

(i) Forge welding
(ii) Electric resistance welding

(i) Forge Welding In forge welding, the parts are heated to reach the plastic stage and the joint is prepared by impact force. The impact force is produced either by a hand hammer or a press. Wrought iron or low carbon steel can be forge welded. This process has limited use in present times. It is used in the fabrication of wrought iron pipes.

(ii) Electric Resistance Welding In electric resistance welding, the parts to be connected are clamped together and a high amperage electric current is passed from one part to another. Resistance of metallic parts to electric current creates heat. This heat is utilized for melting the adjoining parts. No filler material is used in this process. There are two terms related to electric resistance welding, viz., spot welding and seam welding. When a lap joint is formed by applying pressure by two electrodes, one on each side of overlapped plates, a *spot weld* is produced. If two rollers are substituted for the point electrodes and the plates are pulled between the rollers, a '*seam*' *weld* is produced. Electric resistance welding can be easily automated and is often used as a mass production technique. Therefore, it is commonly used in automobile industries.

8.3 STRESS RELIEVING OF WELDED JOINTS

Welded joints are subjected to residual stresses due to non-uniform heating of the parts being joined. There is always a possibility that localised thermal stresses may result from uneven heating and cooling during fusion and subsequent cooling. This also results in distortion. The magnitude of residual stresses cannot be predicted with any degree of certainty. This is the major disadvantage of welded joints. The following two methods can reduce the residual stresses:

(i) Preheating of the weld area to retard cooling of the metal in the vicinity of the joint.
(ii) Stress relieving of weld area by using proper heat treatment such as normalising and annealing in temperature range of 550° to 675°.

One of the methods of stress relieving is *hand peening*. It consists of hammering the weld along the length with the peen of the hammer while the joint is hot. It reduces residual stresses and induces residual compressive stresses on the surfaces. This improves the fatigue strength of the joint.

8.4 BUTT JOINTS

Welded joints are divided into two groups—butt joints and fillet joints. *A butt joint can be defined as a joint between two components lying approximately in the same plane.* A butt joint connects the ends of the two plates. The types of butt joints are illustrated in Fig. 8.1. The selection of the types of butt joints depends upon the plate thickness and the reliability. Some guidelines are as follows:

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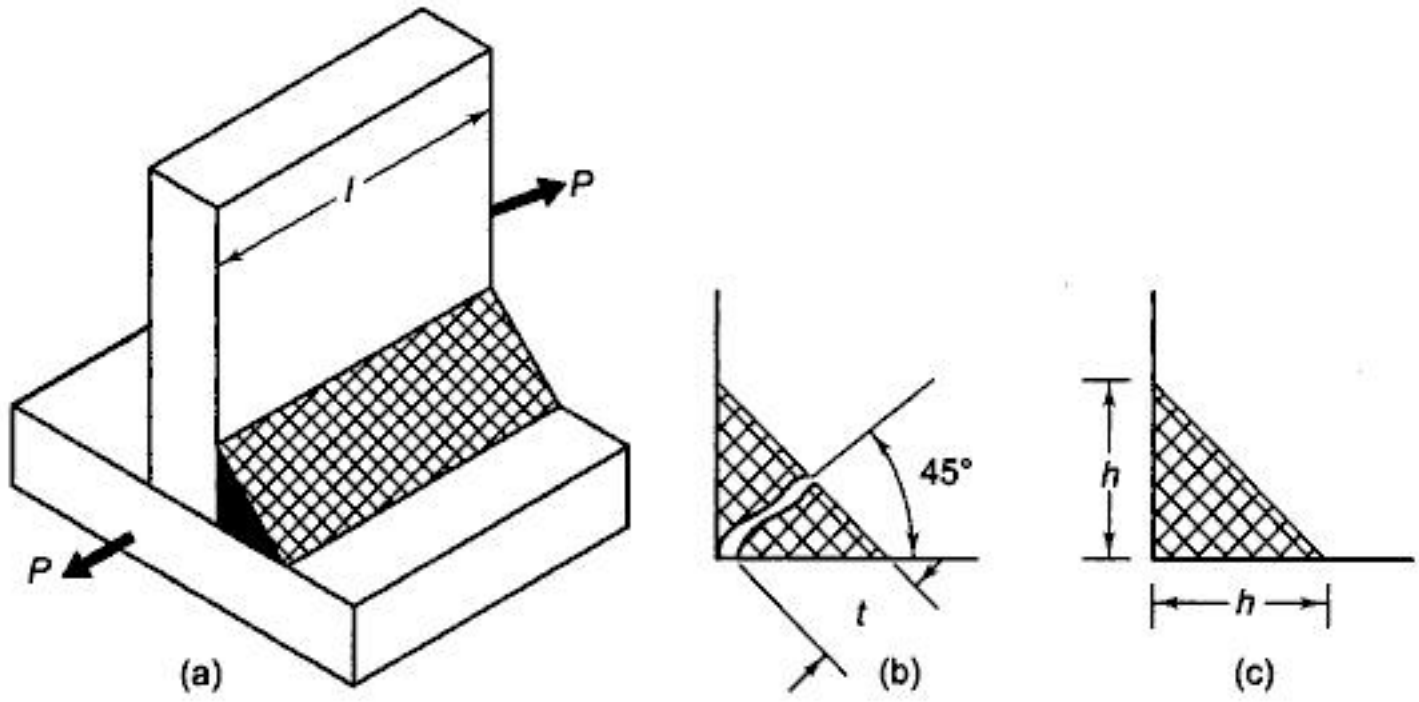

Fig. 8.7 *Parallel Fillet Weld in Shear*

Rearranging the terms of Eq. (8.5), the strength equation of the parallel fillet weld is written in the following form:

$$P = 0.707\ hl\tau \qquad (8.6)$$

where,

P = tensile force on plates (N)
h = leg of the weld (mm)
l = length of the weld (mm)
τ = permissible shear stress for the weld (N/mm^2)

Usually, there are two welds of equal length on two sides of the vertical plate. In that case,

$$P = 2\ (0.707\ hl\tau)$$

or

$$P = 1.414\ hl\tau \qquad (8.7)$$

In determining the required length of the weld, 15 mm should be added to the length of each weld calculated by Eqs (8.6) and (8.7) to allow for starting and stopping of the weld run. In case of a static load, the permissible shear stress for the fillet welds is taken as 94 N/mm^2 as per the code of American Welding Society (AWS).

8.8 STRENGTH OF TRANSVERSE FILLET WELDS

A transverse fillet weld subjected to a tensile force P is shown in Fig. 8.8(a). The transverse fillet welds are subjected to tensile stress. The minimum cross-section of the weld is at the throat. Therefore, the failure due to tensile stress will occur at the throat section. The cross-sectional area at the throat is (tl). The tensile stress in the transverse fillet weld is given by,

$$\sigma_t = \frac{P}{tl}$$

Substituting Eq. (8.4) in the above equation,

$$\sigma_t = \frac{P}{0.707\ hl} \qquad (8.8)$$

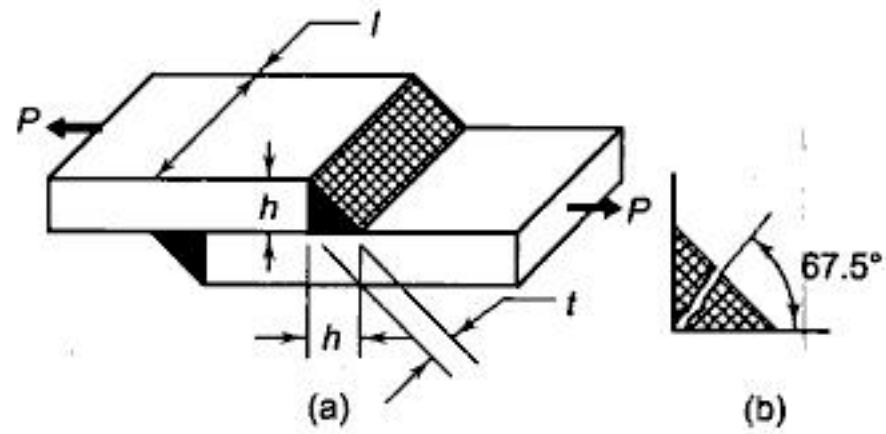

Fig. 8.8 *Failure of Fillet Weld*

Rearranging the terms of Eq. (8.8), the strength equation of the transverse fillet weld is written in the following form,

$$P = 0.707\ hl\sigma_t \qquad (8.9)$$

where,

σ_t = permissible tensile stress for the weld (N/mm^2)

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The area of the weld in the plane inclined at angle θ with horizontal is $(t'l)$. Therefore, the shear stress in this plane is given by,

$$\tau = \frac{P}{(t'l)}$$

Substituting Eq. (8.11) in the above expression,

$$\tau = \frac{P(\sin\theta + \cos\theta)}{hl} \qquad \text{(a)}$$

In order to find out the plane with maximum shear stress, differentiate τ with respect to θ and set the derivative equal to zero.

$$\frac{\partial\tau}{\partial\theta} = \frac{P}{hl}(\cos\theta - \sin\theta) = 0$$

$$\cos\theta - \sin\theta = 0$$

$$\sin\theta = \cos\theta$$

$$\tan\theta = 1$$

Therefore,

$$\theta = 45° \qquad (8.12)$$

The condition for plane with maximum shear stress is $(\theta = 45°)$.

Substituting this value of θ in Eq. (a), the maximum shear stress is given by,

$$\tau_{max.} = \frac{P(\sin 45° + \cos 45°)}{hl}$$

$$= \frac{1.414\,P}{hl} = \frac{P}{(1/1.414)\,hl} = \frac{P}{0.707hl}$$

or $$\tau_{max.} = \frac{P}{0.707hl} \qquad (8.13)$$

The above equation is same as Eq. (8.5). Substituting (l = 1 mm) in Eq. (8.13), the allowable load P_{all} per mm length of the weld is given by,

$$P_{all.} = 0.707h\tau_{max.} \qquad (8.14)$$

Suppose it is required to find out the allowable load per mm length of parallel fillet weld for the permissible shear stress of 94 N/mm^2 and the leg dimension of 8 mm. From Eq. (8.14),

$$P_{all.} = 0.707h\tau_{max.} = 0.707\,(8)\,(94)$$
$$= 531.66 \text{ N/mm}$$

8.10 MAXIMUM SHEAR STRESS IN TRANSVERSE FILLET WELD

A double transverse fillet weld of equal legs is subjected to a force $(2P)$ as shown in Fig. 8.15(a). It is required to find out the inclination (θ) of the plane in the weld where the maximum shear stress is induced and also, the magnitude of the maximum shear stress. The effect of bending is to be neglected.

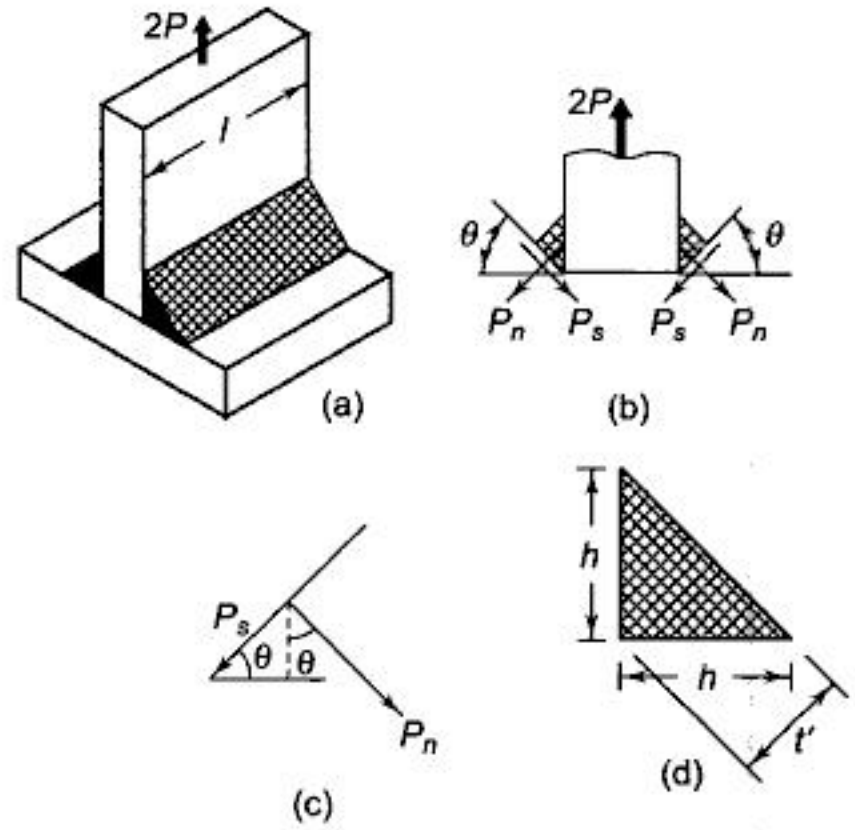

Fig. 8.15

The free body diagram of forces acting on the vertical plate with two symmetrically cut welds is shown in Fig. 8.15(b). The shear force is P_s and the normal force is P_n.

Considering equilibrium of vertical forces, [Fig. 8.15(b) and (c)],

$$2P = 2P_s \sin\theta + 2P_n \cos\theta$$

$$P = P_s \sin\theta + P_n \cos\theta \qquad \text{(a)}$$

Since the resultant of P_s and P_n is vertical, their horizontal components must be equal and opposite. Therefore,

$$P_s \cos\theta = P_n \sin\theta$$

or,

$$P_n = \frac{P_s \cos\theta}{\sin\theta} \qquad \text{(b)}$$

Substituting Eq. (b) in Eq. (a),

$$P = P_s \sin\theta + \frac{P_s \cos\theta \cos\theta}{\sin\theta} \qquad \text{(c)}$$

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The stresses in this welded joint are shown in Fig. 8.20. The force P acting through the centre of gravity causes direct shear stress in the welds [Fig. 8.20(a)]. It is called the *primary shear stress*. It is assumed that the primary shear stress is uniformly distributed over the throat area of all welds. Therefore,

$$\tau_1 = \frac{P}{A} \tag{8.21}$$

where A is the throat area of all welds.

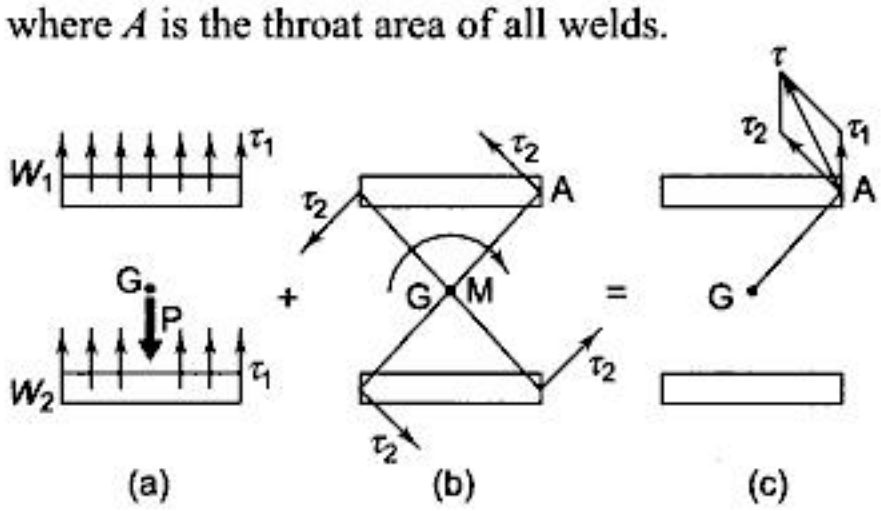

Fig. 8.20 *Primary and Secondary Shear Stresses*

The couple M causes torsional shear stresses in the throat area of welds [Fig. 8.20(b)]. They are called *secondary shear stresses* and given by

$$\tau_2 = \frac{Mr}{J} \tag{8.22}$$

where

r = distance of a point in the weld from G

J = polar moment of inertia of all welds about G

The secondary shear stress at any point in the weld is proportional to its distance from the centre of gravity. Obviously, it is maximum at the farthest point such as A. The resultant shear stress at any point is obtained by vector addition of primary and secondary shear stresses [Fig. 8.20(c)].

Figure 8.21 shows a weld of length l and throat t. G_1 is the centre of gravity of the weld, while G

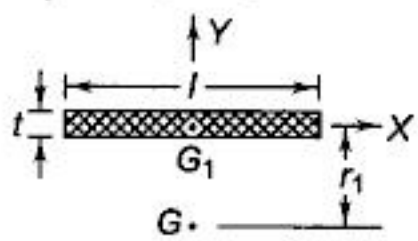

Fig. 8.21

is the centre of gravity of a group of welds. The moment of inertia of this weld about its centre of gravity G_1 is given by,

$$I_{xx} = \frac{lt^3}{12} \quad \text{and} \quad I_{yy} = \frac{tl^3}{12}$$

Since t is very small compared with l, I_{xx} is negligible compared with I_{yy}.

$$\therefore \quad J_{G_1} = I_{xx} + I_{yy} \cong I_{yy}$$

$$\text{or} \quad J_{G_1} = \frac{tl^3}{12} = \frac{tl(l^2)}{12} = \frac{Al^2}{12}$$

Therefore,

$$J_{G_1} = \frac{Al^2}{12} \tag{8.23}$$

where A is the throat area of the weld and J_{G_1} is the polar moment of inertia of the weld about its centre of gravity. The polar moment of inertia about an axis passing through G is determined by the parallel axis theorem. Thus,

$$J_G = J_{G_1} + Ar_1^2 \tag{8.24}$$

where r_1 is the distance between G and G_1. From Eqs (8.23) and (8.24),

$$J_G = A\left[\frac{l^2}{12} + r_1^2\right] \tag{8.25}$$

Where there are a number of welds, with polar moment of inertias J_1, J_2, J_3,..., etc., about the centre of gravity G, the resultant polar moment of inertia is given by,

$$J = J_1 + J_2 + J_3 + \cdots \tag{8.26}$$

The above value of J is to be used in Eq. (8.22) to determine secondary shear stresses.

Example 8.9 *A welded connection, as shown in Fig. 8.22 is subjected to an eccentric force of 7.5 kN. Determine the size of welds if the permissible shear stress for the weld is 100 N/mm². Assume static conditions.*

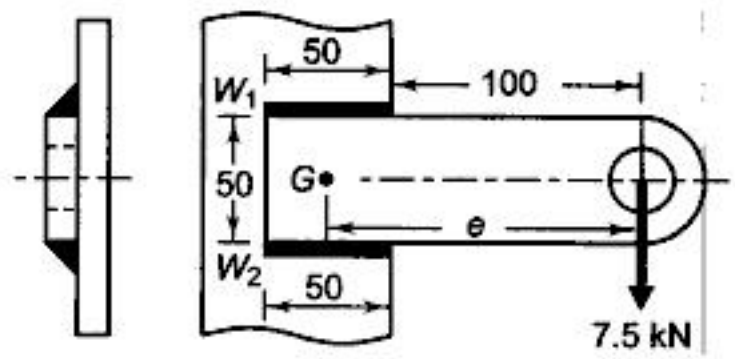

Fig. 8.22

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(iii) Punching injures the metal in the vicinity of the hole. The drilling operation does not injure metal.

(iv) Punching is feasible only for thin plates up to 25 mm thickness because fine cracks are formed around the periphery of the holes in case of punching of the thick plates. Drilling operation is feasible for any thickness of the plates.

Sometimes, the punching operation is followed by a *reaming* operation. In critical joints, this combines the advantages of punching and drilling.

Traditional mechanical structures involving riveted joints are classified into the following three groups:

(i) boilers, pressure vessels and tanks;

(ii) bridges, trusses, cranes and machinery in general; and

(iii) hulls of ships.

Fluid tightness is a desirable property of the joints in boilers, pressure vessels and ships. *Strength* and *rigidity* are desirable characteristics of joints in bridges, trusses and cranes. The joints in these applications are subjected to external load and strength is necessary to prevent failure of the joint. In application of a ship hull, strength, rigidity, durability and leakproofness are important criteria.

The scope of riveted joints in the above mentioned three groups of traditional applications is encroached upon by welded joints. However, riveted joint still remains the best and simplest type of permanent fastening in many applications. At present, riveted joints are mainly used in the following applications:

(i) In welding process, the parts are heated which deteriorate the metal structure and tamper the heat treated parts. It also results in warping of the components, which have been machined. Riveted joints are used where it is necessary to avoid the thermal after-effects of welding.

(ii) Riveted joints are used for metals with poor weldability like aluminium alloys.

(iii) When the joint is made of heterogeneous materials, such as the joint between steel plate and asbestos friction lining, riveted joints are preferred.

(iv) Welded joints have poor resistance to vibrations and impact load. A riveted joint is ideally suitable in such situations.

(v) Riveted joints are used where thin plates are to be assembled. They are popular especially for aircraft structures where light structures made of aluminium alloys are to be fastened.

A riveted joint has the following advantages over a welded joint:

(i) A riveted joint is more reliable than a welded joint in applications which are subjected to vibrations and impact forces.

(ii) Welded joints are, in general, restricted to steel parts. Riveted joints can be used for non-ferrous metals like aluminium alloy, copper, brass or even non-metals like asbestos or plastics.

(iii) The heat required for welding causes warping and affects the structure of heat treated components. The parts assembled by riveted joints are free from such thermal after-effects.

(iv) The quality of riveted joint can be easily checked, while inspection methods for welded joint, such as radiographic inspection of pressure vessels, are costly and time consuming.

(v) When the riveted joint is dismantled, the connected components are less damaged compared with those of welded joints.

The disadvantages of riveted joints compared with welded joints are as follows:

(i) The material cost of riveted joints is more than the corresponding material cost of welded joints due to high consumption of metal. There are two factors which account for more material consumption. The holes required for rivets weaken the working cross-section of the plate and it is necessary

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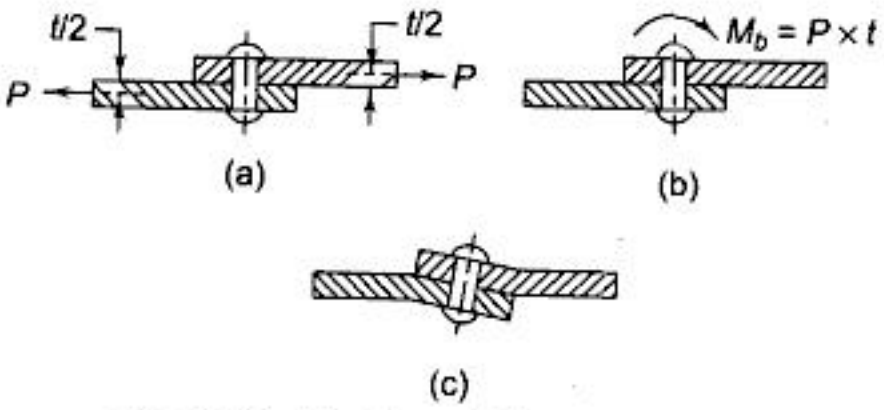

Fig. 8.50 *Bending of Plates in Lap Joint*

The construction of a *butt joint* is shown in Fig. 8.51. It consists of two plates, which are kept in alignment against each other in the same plane and a strap or cover plate is placed over these plates and riveted to each plate. Placing the two plates, which are to be fastened, against each other is called *butting*. Depending upon the number of rows of rivets in each plate, the butt joints are classified as *single-row butt joint* and *double-row butt joint*. Depending upon the number of straps, the butt joints are also classified into *single-strap butt joint* and *double-strap butt joint*. The types of butt joints are illustrated in Fig. 8.51 and 8.52. The line of action of the force acting on two plates, joined by butt joint, lies in the same plane. Therefore, there is no bending moment on the joint and no warping of the plates. This is the main advantage of butt joint compared with lap joint. The disadvantage of butt joint is the requirement of additional strap plates, which increases cost. Therefore, butt joint is costly compared with lap joint.

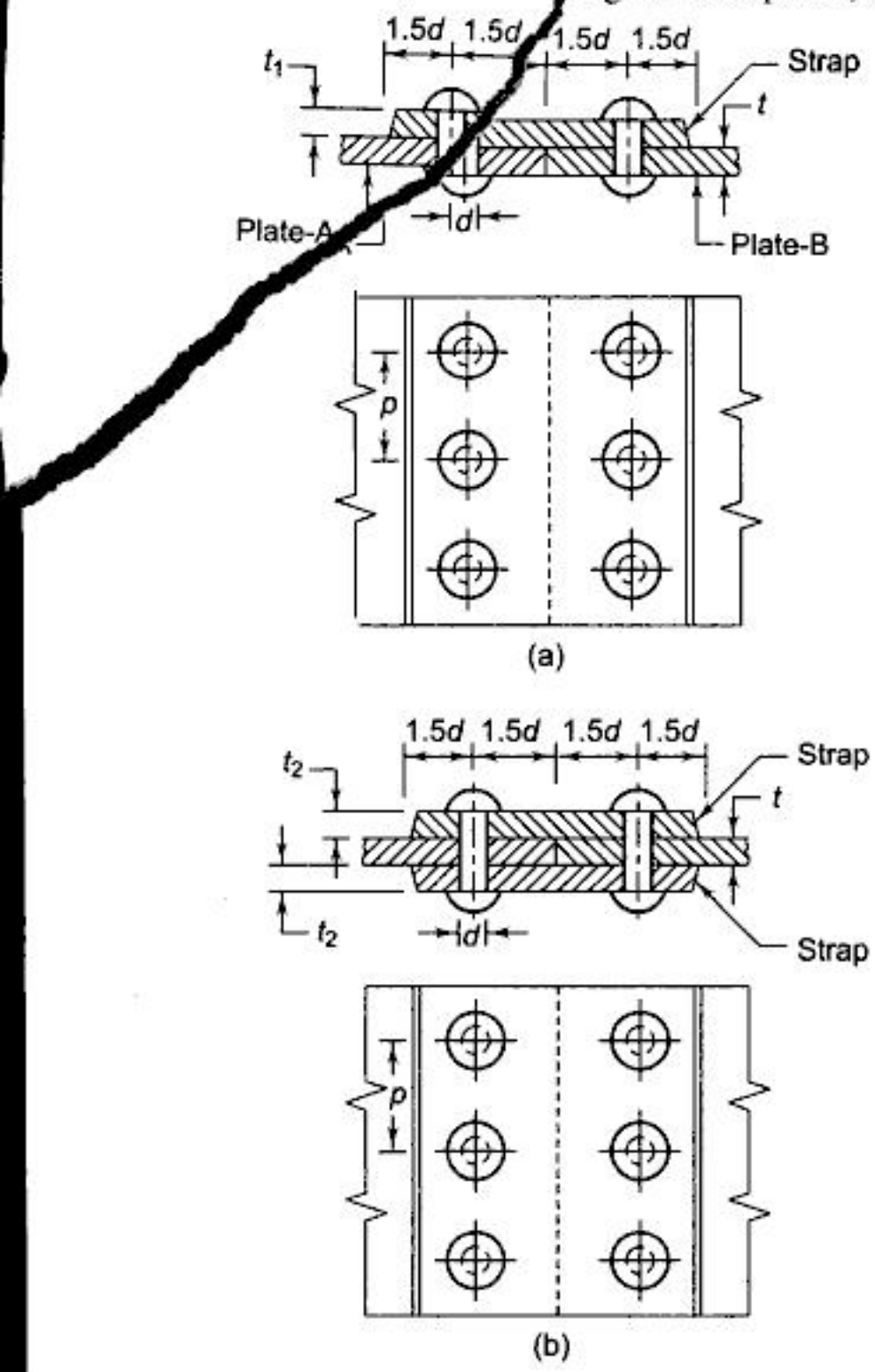

Fig. 8.51 *Types of Single-riveted Butt Joint (a) Single-strap Butt Joint (b) Double-strap Butt Joint*

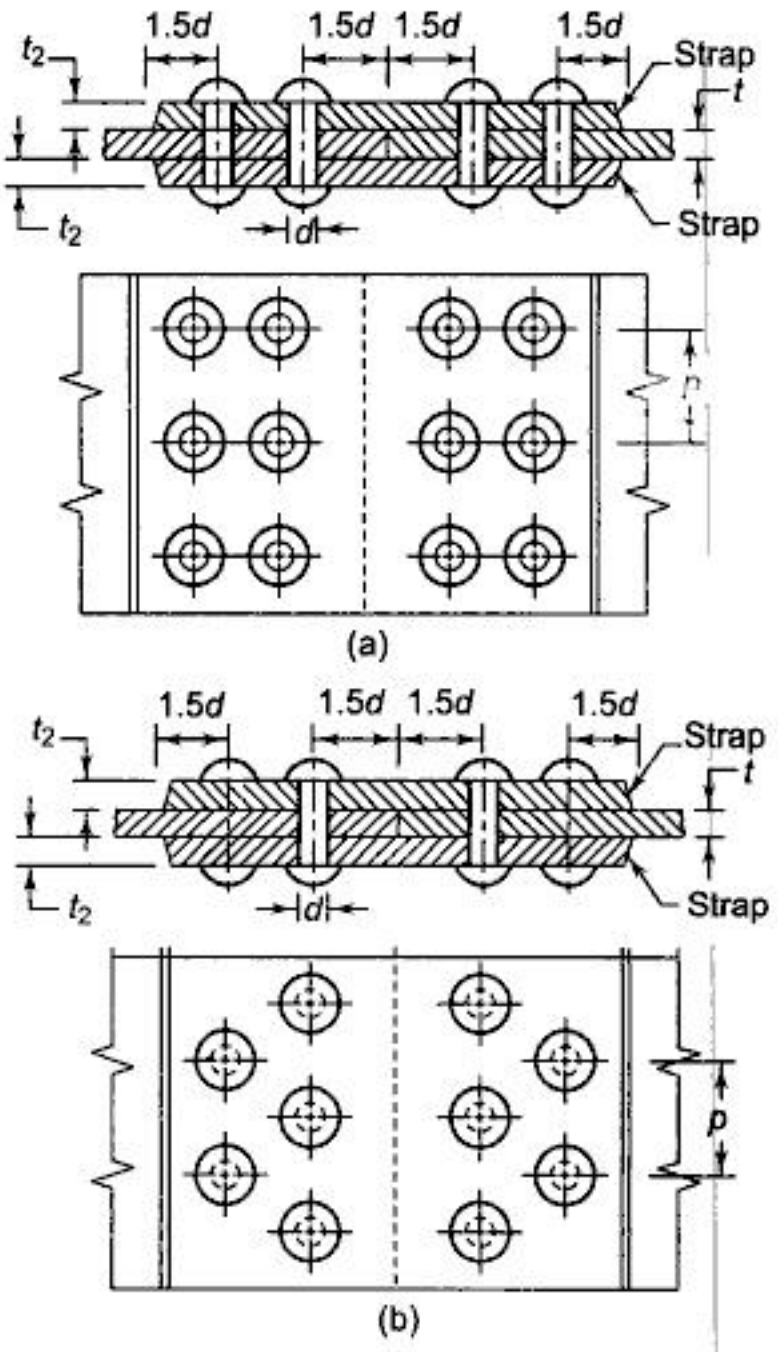

Fig. 8.52 *Types of Double-riveted Double-strap Butt Joint (a) Chain Pattern (b) Zig-zag Pattern*

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of caulking tool closes the surface asperities and cracks on the contacting surfaces between two plates and also between the rivet and the plates, resulting in leakproof joint. Great care must be exercised to prevent injury to the plate, otherwise caulking will result in opening of the joint instead of closing it. Caulking cannot be applied to plates with less than 6 mm thickness.

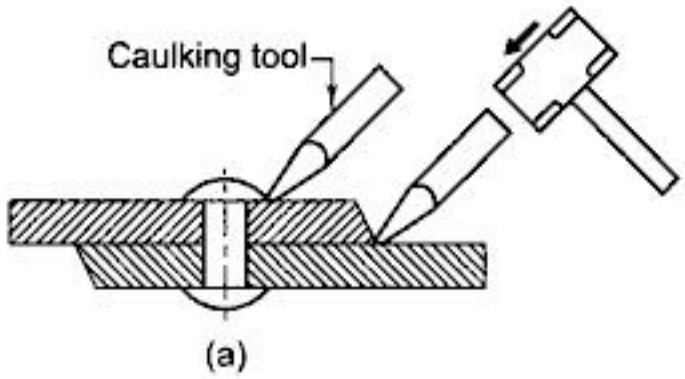

Fig. 8.58 *Caulking and Fullering*

Fullering, as shown in Fig. 8.58(b), is similar to the caulking process except for the shape of the tool. The width of the fullering tool is equal to the thickness of the plate being hammered. The blows of the fullering tool result in simultaneous pressure on the entire edge of the plate.

Example 8.18 *A brake band attached to the hinge by means of a riveted joint is shown in Fig. 8.59. Determine the size of the rivets needed for the load of 10 kN. Also, determine the width of the band. The permissible stresses for the band and rivets in tension, shear and compression are 80, 60 and 120 N/mm² respectively. Assume,*

margin (m) = 1.5d

transverse pitch (p_t) = p

Find the pitch of the rivets.

Solution

Given $P = 10$ kN $t = 3$ mm $\sigma_t = 80$ N/mm² $\tau = 60$ N/mm² $\sigma_c = 120$ N/mm²

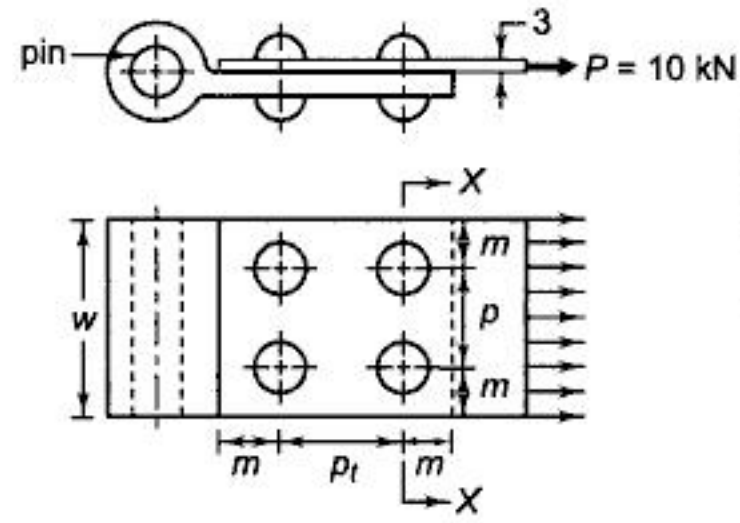

Fig. 8.59

Step I *Diameter of rivets*

There are four rivets in the lap joint, which are subjected to single shear. From shear consideration,

$$4\left[\frac{\pi}{4}d^2\tau\right] = P \quad \text{or} \quad 4\left[\frac{\pi}{4}d^2(60)\right] = 10\times10^3$$

$$\therefore \qquad d = 7.28 \quad \text{or} \quad 8 \text{ mm} \qquad \text{(a)}$$

From crushing consideration,

$$4dt\sigma_c = P \quad \text{or} \quad 4d(3)(120) = 10\times10^3$$

$$\therefore \qquad d = 6.94 \quad \text{or} \quad 7 \text{ mm} \qquad \text{(b)}$$

From (a) and (b), it is observed that shearing becomes the criterion of design. Therefore,

$$d = 8 \text{ mm}$$

Step II *Width of band*

Considering tensile strength of plate along the section *XX*,

$$(w - 2d)t\sigma_t = P \quad \text{or}$$

$$(w - 2\times8)(3)(80) = 10\times10^3$$

$$\therefore \qquad w = 57.67 \text{ or } 60 \text{ mm}$$

Step III *Pitch of rivets*

$$m = 1.5d = 1.5(8) = 12 \quad \text{or} \quad 15 \text{ mm}$$

$$p + 2m = w \quad \text{or} \quad p + 2(15) = 60$$

$$p = 30 \text{ mm}$$

$$p_t = p = 30 \text{ mm}$$

Example 8.19 *Two flat plates subjected to a tensile force P are connected together by means of double-strap butt joint as shown in Fig. 8.60. The force P is 250 kN and the width of the plate w is 200 mm. The rivets and plates are made of the same steel and the permissible stresses in tension, compression and shear are 70, 100 and 60 N/mm² respectively. Calculate:*

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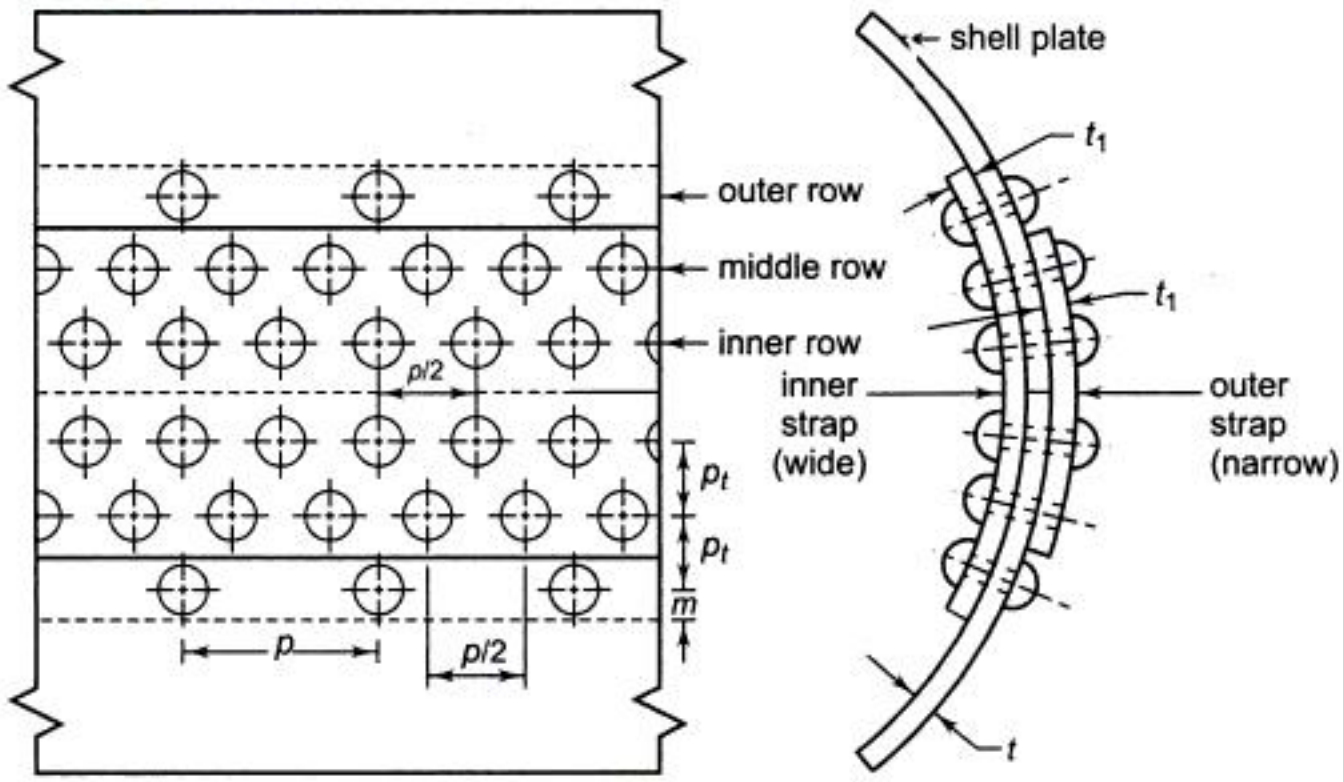

Fig. 8.62 *Triple-riveted Double-strap Butt Joint with Unequal Straps*

Corrosion allowance (CA) is additional metal thickness over and above that required to withstand internal pressure. A minimum corrosion allowance of 1.5 to 2 mm thickness is recommended unless a protective lining is employed. Introducing corrosion allowance in Eq. (8.42),

$$t = \frac{P_i D_i}{2\sigma_t \eta} + \text{CA} \qquad (8.43)$$

where,

CA = corrosion allowance (mm)

Equation (8.42) is used to find out the thickness of the boiler shell. The efficiencies of commercial boiler joints are given in Table 8.5.

Table 8.5 *Efficiencies of riveted boiler joints*

Type of joint	*Efficiency (per cent)*
Lap joint	
single-riveted	45–60
double-riveted	63–70
triple-riveted	72–80
Double-strap butt joint	
single-riveted	55–60
double-riveted	70–83
triple-riveted	80–90
quadruple-riveted	85–94

The permissible tensile stress in Eq. (8.43) is obtained by,

$$\sigma_t = \frac{S_{ut}}{(fs)} \qquad (8.44)$$

where,

S_{ut} = ultimate tensile strength of the plate material (N/mm^2)

(fs) = factor of safety

The factor of safety in boiler applications varies from 4.5 to 4.75. It is safe practice to assume the factor of safety as 5. There are two popular grades of steel used for boiler shells and boiler rivets. They are designated as Grade-St 37 BR and Grade-St 42 BR. Their ultimate tensile strengths are as follows[15]:

Grade	S_{ut} *(N/mm^2)*
St 37 BR	360–440
St 42 BR	410–500

(ii) Diameter of Rivet Indian Boiler Regulations do not specify any formula to calculate the rivet diameter. However, empirical relationships are suggested by design engineers. They are as follows:

(a) When the thickness of plate is more than 8 mm, the rivet diameter is calculated by following Unwin's formula,

[15] IS 1990–1973: Specification for steel rivet and stay bars for boilers.

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As shown in Fig. 8.63, the number of rivets per pitch length in various rows are as follows:

(*a*) *Outer row* There is one half-rivet on the left side and one half-rivet on the right side. These make one rivet per pitch length. It connects the inner strap, the shell plate and the outer strap. This results in double shear in the rivet in the outer row.

(*b*) *Middle row* There are two rivets per pitch length. They connect the inner strap, shell plate and outer strap. This results in double shear in the rivets in the middle row.

(*c*) *Inner row* There is one half-rivet on the left side, one complete rivet in the middle and one half-rivet on the right side. These make two rivets per pitch length. They connect the inner strap, shell plate and outer strap. This results in double shear in the rivets in the inner row.

Therefore, there are five rivets per pitch length for a triple-riveted butt joint. Since the straps are of equal width, all rivets are in double shear. Therefore,

$$n_1 = 0 \qquad n_2 = 5$$

Substituting values in Eq. (8.47),

$$p = \frac{(0 + 1.875 \times 5)\pi (27)^2 (60)}{4(20)(80)} + 27$$

$$= 228.29 \text{ or } 230 \text{ mm}$$

From Eqs (8.48) and (8.49),

$$p_{min.} = 2d = 2\ (27) = 54 \text{ mm}$$

$$p_{max.} = Ct + 41.28 = 6\ (20) + 41.28$$

$$= 161.28 \text{ mm}$$

Since, $p > p_{max.}$

it is assumed that,

$$p = p_{max.} = 161.28 \text{ or } 160 \text{ mm} \qquad \text{(iii)}$$

The pitch of rivets in the outer row is 160 mm. The pitch of rivets in the middle and inner rows is (160/2) or 80 mm.

Step IV *Distance between rows of rivets*

The number of rivets in the outer row is one-half of the number of rivets in the middle and inner rows. Also, the rivets have a zig-zag pattern in middle and inner rows. From Eq. (8.52), the distance between the outer and middle rows is given by,

$$p_t = 0.2p + 1.15d = 0.2\ (160) + 1.15\ (27)$$

$$= 63.05 \text{ or } 65 \text{ mm}$$

From Eq. (8.53), the distance between middle and inner rows is given by,

$$p_t = 0.165p + 0.67d = 0.165\ (160) + 0.67\ (27)$$

$$= 44.49 \text{ or } 50 \text{ mm} \qquad \text{(iv)}$$

Step V *Margin*

From Eq. (8.54),

$$m = 1.5d = 1.5\ (27) = 40.5 \text{ or } 45 \text{ mm} \qquad \text{(v)}$$

Step VI *Thickness of straps*

The straps are of equal width and every alternate rivet in the outer row is omitted. From Eq. (8.57),

$$t_1 = 0.625t\left[\frac{p-d}{p-2d}\right] = 0.625(20)\left[\frac{160-27}{160-2\times 27}\right]$$

$$= 15.68 \text{ or } 16 \text{ mm} \qquad \text{(vi)}$$

Step VII *Efficiency of joint*

The tensile strength of plate per pitch length in the outer row is given by,

$$P_t = (p-d)\, t\sigma_t = (160 - 27)(20)(80)$$

$$= 212\,800 \text{ N} \qquad \text{(a)}$$

The shear strength of rivets per pitch length is given by,

$$P_s = (n_1 + 1.875n_2)\left[\frac{\pi}{4} d^2 \tau\right]$$

$$= (0 + 1.875 \times 5)\left[\frac{\pi}{4}(27)^2 (60)\right]$$

$$= 322\,062.33 \text{ N} \qquad \text{(b)}$$

The crushing strength of the plate is given by,

$$P_c = (n_1 + n_2)\, dt\sigma_c$$

$$= (0 + 5)\ (27)\ (20)\ (120)$$

$$= 324\,000 \text{ N} \qquad \text{(c)}$$

The tensile strength of the solid plate per pitch length is given by,

$$P = pt\sigma_t = 160\ (20)(80) = 256\,000 \text{ N} \qquad \text{(d)}$$

From (a), (b), (c) and (d),

$$\eta = \frac{212\,800}{256\,000} = 0.8313 \quad \text{or} \quad 83.13\% \qquad \text{(vii)}$$

Figure 8.63 shows the sketch of longitudinal butt joint with calculated values of the dimensions.

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a single transverse and double parallel fillet welds, as shown in Fig. 8.73. The strength of the welded joint should be equal to the strength of the plates to be joined. The permissible tensile and shear stresses for the weld material and the plates are 100 and 70 N/mm^2 respectively. Find the length of each parallel fillet weld. Assume that the tensile force passes through the centre of gravity of three welds.

[23.68 mm]

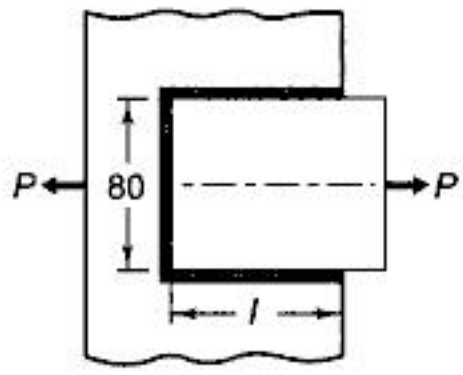

Fig. 8.73

8.3 A bracket, as shown in Fig. 8.74, is welded to a plate. The welds have the same size, and the permissible force per mm of the weld-length is 1 kN. Calculate the lengths l_1 and l_2.

[40 and 80 mm]

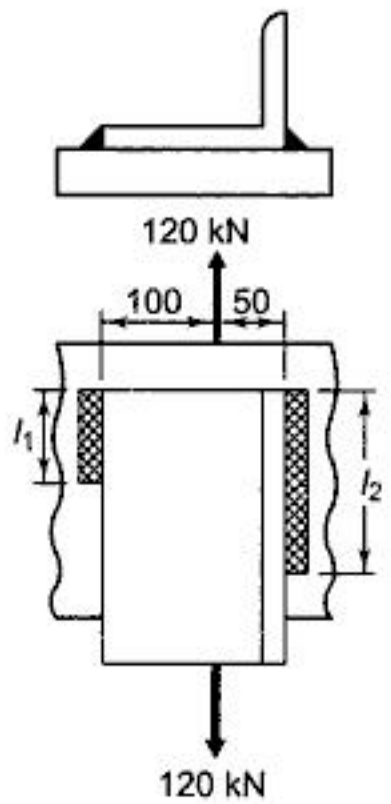

Fig. 8.74

8.4 A welded connection of steel plates, as shown in Fig. 8.75, is subjected to an eccentric force of 10 kN. Determine the throat dimension of the welds, if the permissible shear stress is limited to 95 N/mm^2. Assume static conditions.

[8.59 mm]

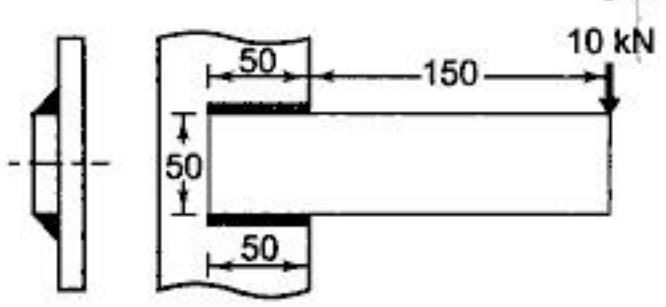

Fig. 8.75

8.5 A welded connection of steel plates is shown in Fig. 8.76. It is subjected to an eccentric force of 50 kN. Determine the size of the weld, if the permissible shear stress in the weld is not to exceed 70 N/mm^2.

[12.94 mm]

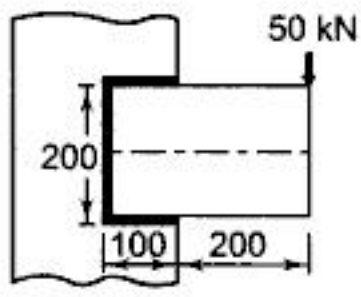

Fig. 8.76

8.6 A welded joint, as shown in Fig. 8.77, is subjected to an eccentric load of 2500 N. Find the size of the weld, if the maximum shear stress in the weld is not to exceed 50 N/mm^2.

[6.4 mm]

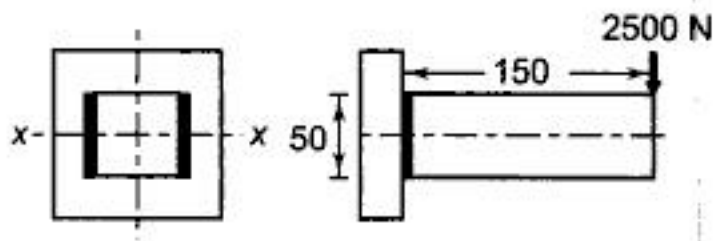

Fig. 8.77

8.7 A circular shaft, 75 mm in diameter, is welded to the support by means of a circumferential fillet weld. It is subjected to a torsional moment of 3000 N- m. Determine the size of weld, if the maximum shear stress in the weld is not to exceed 70 N/mm^2.

[6.86 mm]

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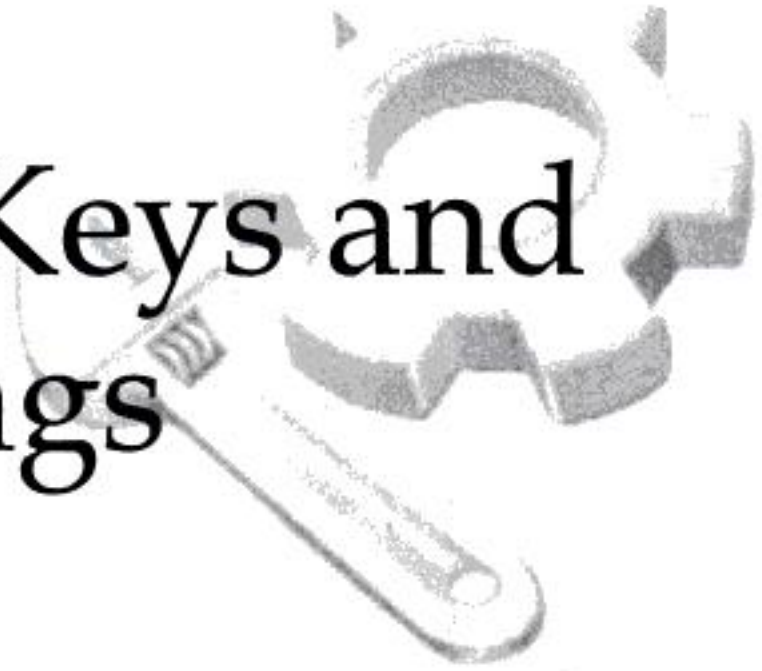
Shafts, Keys and Couplings



Chapter 9

9.1 TRANSMISSION SHAFTS

The term '*transmission shaft*' usually refers to a rotating machine element, circular in cross-section, which supports transmission elements like gears, pulleys and sprockets and transmits power. A transmission shaft supporting a gear in a speed reducer is shown in Fig. 9.1. The shaft is always stepped with maximum diameter in the middle portion and minimum diameter at the two ends, where bearings are mounted. The steps on the shaft provide shoulders for positioning transmission elements like gears, pulleys and bearings. The rounded-off portion between two cross-sections of different diameters is called *fillet*. The fillet radius is provided to reduce the effect of stress-concentration due to abrupt change in the cross-section.

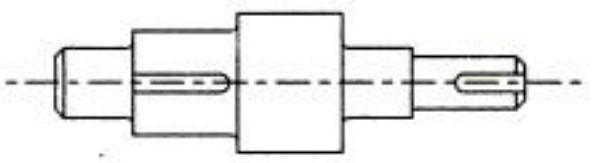


Fig. 9.1 *Transmission Shaft*

Shafts are given specific names in typical applications, although all applications involve transmission of power, motion and torque. Some of the specific categories of transmission shafts are as follows:

(i) Axle The term 'axle' is used for a shaft that supports rotating elements like wheels, hoisting drums or rope sheaves and which is fitted to the housing by means of bearings. In general, an axle is subjected to bending moment due to transverse loads like bearing reactions and does not transmit any useful torque, e.g., rear axle of a railway wagon. Occasionally, the axle also transmits torque, e.g., automobile rear axle. An axle may rotate with the wheel or simply support a rotating wheel.

(ii) Spindle A spindle is a short rotating shaft. The term 'spindle' originates from the round tapering stick on a spinning wheel, on which the thread is twisted. Spindles are used in all machine tools such as the small drive shaft of a lathe or the spindle of a drilling machine.

(iii) Countershaft It is a secondary shaft, which is driven by the main shaft and from which the power is supplied to a machine component. Often, the countershaft is driven from the main shaft by means of a pair of spur or helical gears and thus rotates '*counter*' to the direction of the main shaft. Countershafts are used in multi-stage gearboxes.

(iv) Jackshaft It is an auxiliary or intermediate shaft between two shafts that are used in transmission of power. Its function is same as that of the countershaft.

(v) Line shaft A line shaft consists of a number of shafts, which are connected in axial direction by means of couplings. Line shafts were popular

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$$\theta_r = \frac{M_t l}{JG}$$

Converting θ_r from radians to degrees (θ),

$$\theta = \frac{180}{\pi} \times \frac{M_t l}{JG} \qquad \text{(a)}$$

For solid circular shaft,

$$J = \frac{\pi d^4}{32} \qquad \text{(b)}$$

Combining Eqs (a) and (b),

$$\theta = \frac{584 M_t l}{G d^4} \qquad (9.13)$$

where,

θ = angle of twist (deg.)
l = length of shaft subjected to twisting moment (mm)
M_t = torsional moment (N-mm)
G = modulus of rigidity (N/mm^2)
d = shaft diameter (mm)

Equation (9.13) is used to design the shaft on the basis of torsional rigidity. The permissible angle of twist for machine tool applications is 0.25° per metre length. For line shafts, 3° per metre length is the limiting value. Modulus of rigidity for steel is 79 300 N/mm^2 or approximately 80 kN/mm^2.

9.4 ASME CODE FOR SHAFT DESIGN

One important approach of designing a transmission shaft is to use the ASME code. According to this code, the permissible shear stress $\tau_{max.}$ for the shaft without keyways is taken as 30% of yield strength in tension or 18% of the ultimate tensile strength of the material, whichever is minimum. Therefore,

$$\tau_{max.} = 0.30\, S_{yt}$$

or, $\tau_{max.} = 0.18\, S_{ut}$ (whichever is minimum) (9.14)

If keyways are present, the above values are to be reduced by 25 per cent. According to the ASME code, the bending and torsional moments are to be multiplied by factors k_b and k_t respectively, to account for shock and fatigue in operating condition. The ASME code is based on maximum shear stress theory of failure. Therefore, Eq. (9.10) is modified and rewritten as,

$$\tau_{max.} = \frac{16}{\pi d^3} \sqrt{(k_b M_b)^2 + (k_t M_t)^2} \qquad (9.15)$$

where,

k_b = combined shock and fatigue factor applied to bending moment
k_t = combined shock and fatigue factor applied to torsional moment

The values of k_b and k_t for rotating shafts are given in Table 9.2.

Table 9.2 *Values of shock and fatigue factors k_b and k_t*

	Application	k_b	k_t
(i)	Load gradually applied	1.5	1.0
(ii)	Load suddenly applied (minor shock)	1.5–2.0	1.0–1.5
(iii)	Load suddenly applied (heavy shock)	2.0–3.0	1.5–3.0

Equations (9.14) and (9.15) are used to design the shaft according to the ASME code.

We will rewrite Eq. (9.15) again.

$$\tau_{max.} = \frac{16}{\pi d^3} \sqrt{(k_b M_b)^2 + (k_t M_t)^2}$$

Similarly, we will write

$$\sigma_1 = \frac{16}{\pi d^3} \left[k_b M_b + \sqrt{(k_b M_b)^2 + (k_t M_t)^2} \right]$$

(i) Equivalent Torsional Moment The expression $\sqrt{(k_b M_b)^2 + (k_t M_t)^2}$ is called *'equivalent' torsional moment* when the shaft is subjected to fluctuating loads. The equivalent torsional moment is defined as the torsional moment, which when acting alone, will produce the same torsional shear stress in the shaft as under the combined action of bending moment (M_b) and torsional moment (M_t) under fluctuating loads.

(ii) Equivalent Bending Moment The expression $\left[k_b M_b + \sqrt{(k_b M_b)^2 + (k_t M_t)^2} \right]$ is called *'equivalent' bending moment* when the shaft is subjected to fluctuating loads. The equivalent bending moment is defined as the bending moment, which when acting alone, will produce the same bending stresses (tensile and compressive) in the shaft as under the combined action of bending moment (M_b) and torsional moment (M_t) under fluctuating loads.

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the diameter of the pulley, higher will be the belt tension for a given torque.

The torque transmitted by the shaft is given by,

$$M_t = (2700 - 900)\left(\frac{250}{2}\right) = 225\,000 \text{ N-mm}$$

Step III *Bending moment*

The forces and bending moments in vertical and horizontal planes are shown in Fig. 9.5(b). The maximum bending moment is at A. The resultant bending moment at A is given by,

$$M_b = \sqrt{(810\,000)^2 + (250\,000)^2} = 847\,703 \text{ N-mm}$$

Step IV *Shaft diameter*

From Eq. (9.15),

$$d^3 = \frac{16}{\pi \tau_{\max.}} \sqrt{(k_b M_b)^2 + (k_t M_t)^2}$$

$$= \frac{16}{\pi(85.5)} \sqrt{(1.5 \times 847\,703)^2 + (1.0 \times 225\,000)^2}$$

or $d = 42.53$ mm

Example 9.4 *The layout of an intermediate shaft of a gear box supporting two spur gears B and C is shown in Fig. 9.6. The shaft is mounted on two bearings A and D. The pitch circle diameters of gears B and C are 900 and 600 mm respectively. The material of the shaft is steel FeE 580 (S_{ut} = 770 and S_{yt} = 580 N/mm^2). The factors k_b and k_t of ASME code are 1.5 and 2.0 respectively. Determine the shaft diameter using the ASME code.*

Assume that the gears are connected to the shaft by means of keys.

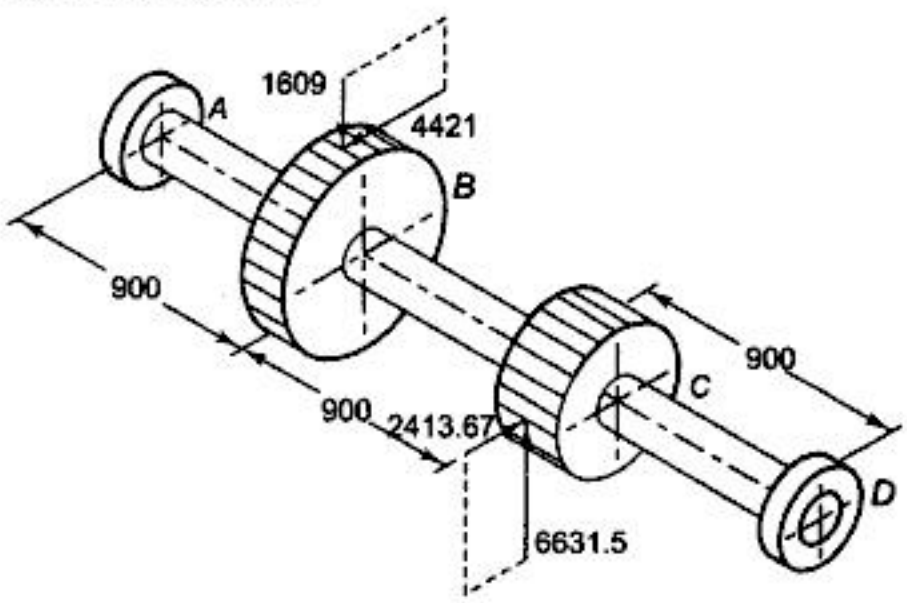

Fig. 9.6

Solution

Given $S_{ut} = 770$ N/mm^2 $S_{yt} = 580$ N/mm^2

$k_b = 1.5$ $k_t = 2.0$

For gears, $(d'_p)_B = 900$ mm $(d'_p)_C = 600$ mm

Step I *Permissible shear stress*

$$0.30\, S_{yt} = 0.30(580) = 174 \text{ N/mm}^2$$

$$0.18\, S_{ut} = 0.18(770) = 138.6 \text{ N/mm}^2$$

The lower of the two values is 138.6 N/mm^2 and there are keyways on the shaft.

$$\therefore \quad \tau_{\max.} = 0.75(138.6) = 103.95 \text{ N/mm}^2$$

Step II *Bending and torsional moment*

The forces and bending moments in vertical and horizontal planes are shown in Fig. 9.7. The maximum bending moment is at C. The resultant bending moment at C is given by,

$$M_b = \sqrt{(3\,496\,203)^2 + (121\,905)^2}$$

$$= 3\,498\,327.4 \text{ N-mm}$$

$$M_t = 4421(450) = 1\,989\,450 \text{ N-mm}$$

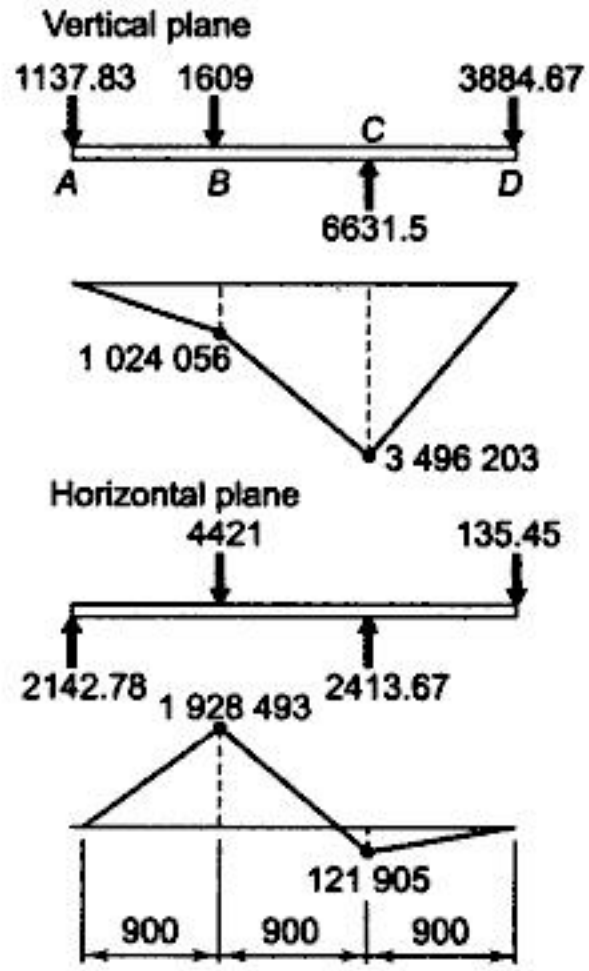

Fig. 9.7

Step III *Shaft diameter*

From Eq. (9.15),

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Solution

Given $G = 79\,300$ N/mm^2
$\theta = 3°$ per metre length

Step I *Permissible angle of twist for shaft*
Refer to Fig. 9.8. The portion of the shaft between the spur gear B and pulley D is subjected to twisting. The centre distance between the spur gear B and the pulley D is 800 mm (Fig. 9.8). Therefore,

$$\theta = \left(\frac{3}{1000}\right)(800) = 2.4° \qquad \text{(i)}$$

Step II *Shaft diameter*
From Eq. (9.13),

$$d^4 = \frac{584 M_t l}{G\theta} = \frac{584(381\,971.86)(800)}{(79\,300)(2.4)}$$

$$\therefore \qquad d = 31.12 \text{ mm} \qquad \text{(ii)}$$

Example 9.9 *Assume the data of the transmission shaft given in Example 9.6. For this shaft, the permissible angle of twist is 0.25° per metre length. The modulus of rigidity for the shaft material is 79 300 N/mm^2. Calculate:*

(i) the permissible angle of twist between helical and bevel gears; and
(ii) the shaft diameter on the basis of torsional rigidity.

Compare the results of these two examples and comment on the results.

Solution

Given $G = 79\,300$ N/mm^2 $\theta = 0.25°$ per metre

Step I *Permissible angle of twist for shaft*
Refer to Fig. 9.10. The portion of the shaft between the two gears is subjected to twisting. The distance between the helical gear B and the bevel gear D is 800 mm (Fig. 9.10). Therefore,

$$\theta = 0.25\left(\frac{800}{1000}\right) = 0.2° \qquad \text{(i)}$$

Step II *Shaft diameter*
From Eq. (9.13),

$$d^4 = \frac{584 M_t l}{G\theta} = \frac{584(144\,000)(800)}{(79\,300)(0.2)}$$

$$\therefore \qquad d = 45.38 \text{ mm} \qquad \text{(ii)}$$

Step III *Criterion of design*
In Example 9.6, the shaft was designed on the basis of its strength and its diameter was calculated as 32.95 mm. On the basis of torsional rigidity, the diameter comes out to be 45.38 mm. Therefore, it is necessary to design the shaft on the basis of strength as well as rigidity.

9.5 DESIGN OF HOLLOW SHAFT ON STRENGTH BASIS

The torsional shear stress and bending stresses in the shaft are given by,

$$\tau = \frac{M_t r}{J} \qquad \sigma_b = \frac{M_b y}{I}$$

The distribution of bending stresses and torsional shear stress is shown in Fig. 9.14. It is observed that the torsional shear stress as well as bending stresses are zero at the shaft centre (r = 0 and y = 0) and negligibly small in the vicinity of the shaft centre, where the radius is small. As the radius increases, the resisting stresses due to external bending and torsional moments increase. Therefore, outer fibres are more effective in resisting the applied moments. In hollow shafts, the material at the centre is removed and spread at large radius. Therefore, hollow shafts are stronger than solid shafts having the same weight.

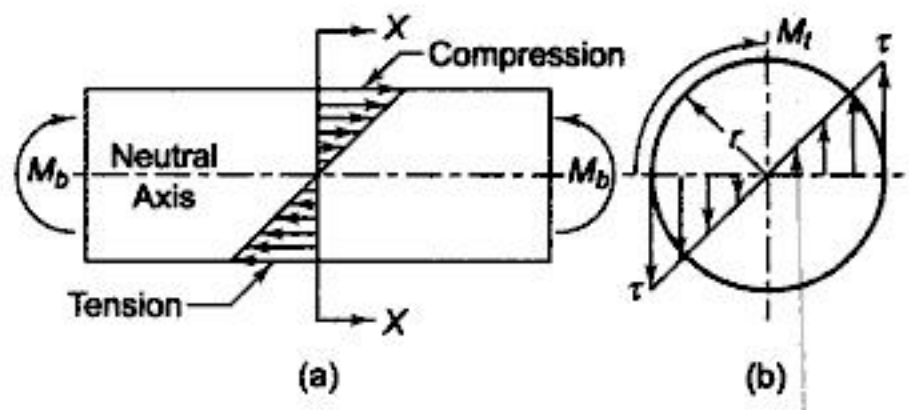

Fig. 9.14 *(a) Distribution of Bending Stresses (b) Distribution of Torsional Shear Stress*

Compared with solid shaft, hollow shaft offers following advantages:

(i) The stiffness of the hollow shaft is more than that of solid shaft with same weight.
(ii) The strength of hollow shaft is more than that of solid shaft with same weight.

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$$M_t = \left(\frac{\pi d^3}{16}\right)\tau \qquad \text{(a)}$$

Step II *Torque transmitted by hollow shaft*

$$M_t = \left(\frac{J}{r}\right)\tau \quad \text{where} \quad J = \frac{\pi(d_o^4 - d_i^4)}{32}$$

$$\therefore \qquad M_t = \frac{\pi(d_o^4 - d_i^4)}{16 d_o}\tau \qquad \text{(b)}$$

Step III *Outer diameter of hollow shaft*

The solid and hollow shafts are equally strong in torsion. Equating (a) and (b),

$$d^3 = \frac{d_o^4 - d_i^4}{d_o}$$

or
$$d_i^4 = d_o^4 - d^3 d_o \qquad \text{(c)}$$

Since the weight of hollow shaft per metre length is half of the solid shaft,

$$\frac{\pi(d_o^2 - d_i^2)}{4} = \frac{1}{2} \times \frac{\pi d^2}{4}$$

or
$$d_i^2 = d_o^2 - \frac{d^2}{2}$$

$$\therefore \qquad d_i^4 = \left(d_o^2 - \frac{d^2}{2}\right)^2 \qquad \text{(d)}$$

Eliminating the term (d_i^4) from Eqs (c) and (d),

$$d_o^4 - d^3 d_o = \left(d_o^2 - \frac{d^2}{2}\right)^2$$

$$\therefore \qquad d_o^4 - d^3 d_o = d_o^4 - d_o^2 d^2 + \frac{d^4}{4}$$

or
$$d_o^2 d^2 - d_o d^3 - \frac{d^4}{4} = 0 \qquad \text{(e)}$$

Equation (e) is a quadratic equation and the roots are given by,

$$d_o = \frac{d^3 \pm \sqrt{d^6 - 4(d^2)\left(-\frac{d^4}{4}\right)}}{2d^2} = \frac{d \pm \sqrt{2}\, d}{2}$$

Taking positive root,

$$d_o = \left(\frac{d + \sqrt{2}d}{2}\right)$$

9.7 FLEXIBLE SHAFTS

Flexible shafts are used to transmit torque between machine units, which may change their relative positions during operation. Their main applications include portable power driven tools, speedometer drives, positioning devices, portable grinders, concrete vibrators, remote control devices and servo drives. In these applications, flexibility is required to accommodate large angular and offset misalignment between driving and driven units. Flexible shafts have two important properties:

(i) They have low rigidity in bending, making them flexible.
(ii) They have high rigidity in torsion, making them capable to transmit torque.

A flexible shaft[2] consists of an inner core, outer casing and end attachments. The inner core consists of a wire coil. In effect, it is a helical torsion spring with closely wound coils. The direction of rotation of the shaft should be such that when the torque is transmitted, the spring is further twisted. The casing protects the shaft from dirt, retains the lubricant such as grease on the shaft surface, prevents the formation of loops in operation, and protects the operator from injury. The end fittings are intended to connect the flexible shaft to driving and driven members.

The applications of flexible shafts are restricted to low power drives.

9.8 KEYS

A key can be defined as a machine element which is used to connect the transmission shaft to rotating machine elements like pulleys, gears, sprockets or flywheels. A keyed joint consisting of shaft, hub and key is illustrated in Fig. 9.16. There are two basic functions of the key. They are as follows:

[2] IS 4600–1968: Specification for flexible shafts.

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relative axial movement between them. The feather key is a particular type of sunk key with uniform width and height. There are number of methods to fix the key to the shaft or hub. Figure 9.20 shows a feather key, which is fixed to the shaft by means of two cap screws, having countersunk-heads. There is a clearance fit between the key and the keyway in the hub. Therefore, the hub is free to slide over the key. At the same time, there is no relative rotational movement between the shaft and the hub. Therefore, the feather key transmits the torque and at the same time permits some axial movement of the hub. Feather keys are used where the parts mounted on the shaft are required to slide along the shaft such as clutches or gear shifting devices. It is an alternative to splined connection.

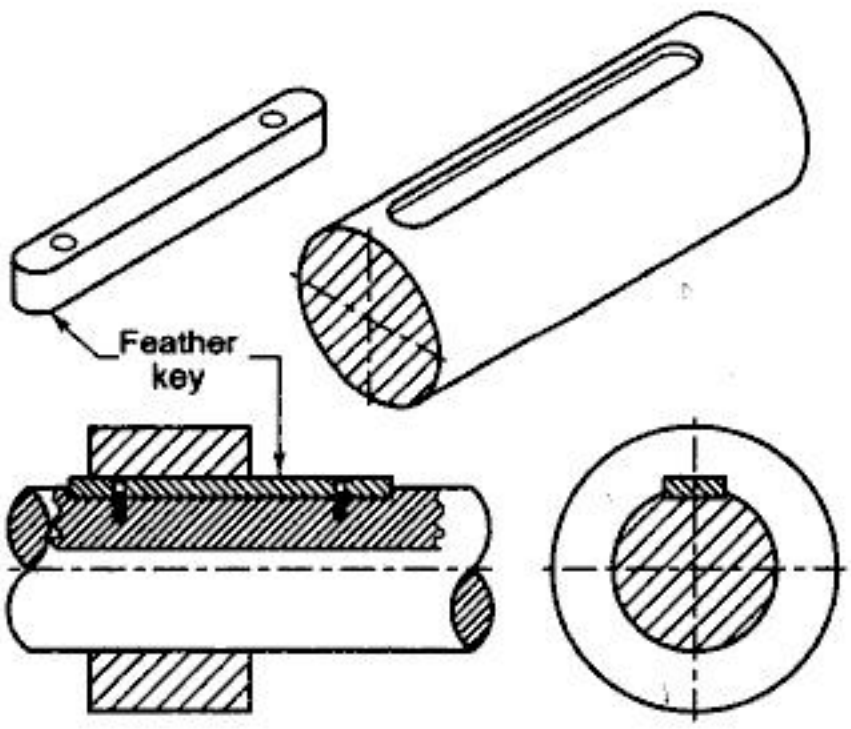

Fig. 9.20 *Feather Key*

9.12 WOODRUFF KEY

A Woodruff key is a sunk key in the form of an almost semicircular disk of uniform thickness as shown in Fig. 9.21. The keyway in the shaft is in the form of a semicircular recess with the same curvature as that of the key. The bottom portion of the Woodruff key fits into the circular keyway in the shaft. The keyway in the hub is made in the usual manner. The projecting part of Woodruff key fits in the keyway in the hub. Once placed in position, the Woodruff key tilts and aligns itself on the shaft. The advantages of Woodruff key are as follows:

(i) The Woodruff key can be used on tapered shaft because it can align by slight rotation in the seat.

(ii) The extra depth of key in the shaft prevents its tendency to slip over the shaft.

The disadvantages of Woodruff key are as follows:

(i) The extra depth of keyway in the shaft increase stress concentration and reduces its strength.

(ii) The key does not permit axial movement between the shaft and the hub.

Woodruff keys are used on tapered shafts in machine tools and automobiles.

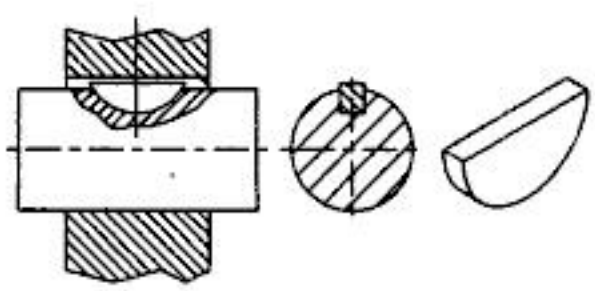

Fig. 9.21 *Woodruff Key*

9.13 DESIGN OF SQUARE AND FLAT KEYS

Although there are many types of keys, only square and flat keys are extensively used in practice. Therefore, the discussion in this chapter is restricted to square and flat keys. A square key is a particular type of flat key, in which the height is equal to the width of the cross-section. Therefore, for the purpose of analysis, a flat key is considered.

The forces acting on a flat key, with width as b and height as h, are shown in Fig. 9.22. The

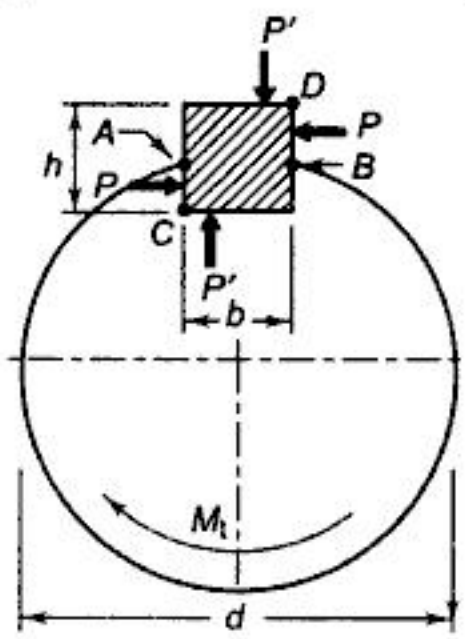

Fig. 9.22 *Forces Acting on Key*

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at 300 rpm The key is made of steel 50C4 ($S_{yt} = S_{yc}$ = 460 N/mm²). Calculate the factor of safety used in design.

Solution

Given kW = 5 n = 300 rpm

$S_{yt} = S_{yc}$ = 460 N/mm² d = 30 mm

Step I *Torque transmitted by the shaft*

The torque transmitted by the shaft is given by,

$$M_t = \frac{60 \times 10^6 (\text{kW})}{2\pi n} = \frac{60 \times 10^6 (5)}{2\pi (300)}$$

$$= 159\,154.94 \text{ N-mm}$$

Step II *Force induced by torque*

The torque is transmitted by two, equal and opposite, forces P. It is assumed that the force P acts at the shaft surface. Therefore,

$$P \times 15 = M_t$$

$$P = \frac{M_t}{15} = \frac{159\,154.94}{15} = 10\,610.33 \text{ N}$$

Step III *Key area*

Referring to Fig. 9.26(b),

$$\cos\theta = \frac{\overline{OE}}{\overline{OC}} = \frac{4}{10} \quad \text{or} \quad \theta = 66.42°$$

$$\overline{CD} = 2\overline{CE} = 2\overline{OC}\sin\theta$$

$$= 2(10)\sin(66.42°) = 18.33 \text{ mm}$$

Total area of the key

$$= \frac{1}{2}\left[\frac{\pi}{4}(20)^2\right] = 157.08 \text{ mm}^2$$

Area of sector $OCFD$

$$= \left(\frac{2\theta}{180}\right)(157.08) = \left(\frac{2(66.42)}{180}\right)(157.08)$$

$$= 115.93 \text{ mm}^2$$

$$\text{Area of } \Delta OCD = \frac{1}{2}\overline{CD} \times \overline{OE}$$

$$= \frac{1}{2}(18.33)(4) = 36.66 \text{ mm}^2$$

Area of key in contact with the shaft = 115.93 − 36.66 = 79.27 mm²

Area of key in contact with the hub = 157.08 − 79.27 = 77.81 mm²

Step IV *Factor of safety against compression failure*

The area of the key in contact with the hub is less and the compressive stress in the area is given by,

$$\sigma_c = \frac{P}{A} = \frac{10\,610.33}{77.81} = 136.36 \text{ N/mm}^2$$

$$(fs) = \frac{S_{yc}}{\sigma_c} = \frac{460}{136.36} = 3.37 \qquad \text{(i)}$$

Step V *Factor of safety against shear failure*

The shear failure will occur in the plane CD, and its area is ($\overline{CD} \times 6$ or (18.33×6) mm² Therefore,

$$\tau = \frac{P}{A} = \frac{10\,610.33}{(18.33 \times 6)} = 96.48 \text{ N/mm}^2$$

$$(fs) = \frac{S_{sy}}{\tau} = \frac{0.577\,S_{yt}}{\tau}$$

$$= \frac{0.577(460)}{96.48} = 2.75 \qquad \text{(ii)}$$

9.15 SPLINES

Splines are keys which are made integral with the shaft. They are used when there is a relative axial motion between the shaft and the hub. The gear shifting mechanism in automobile gearboxes requires such type of construction. Splines are cut on the shaft by milling and on the hub by broaching. A splined connection, with straight splines, is shown in Fig. 9.27. The following notations are used:

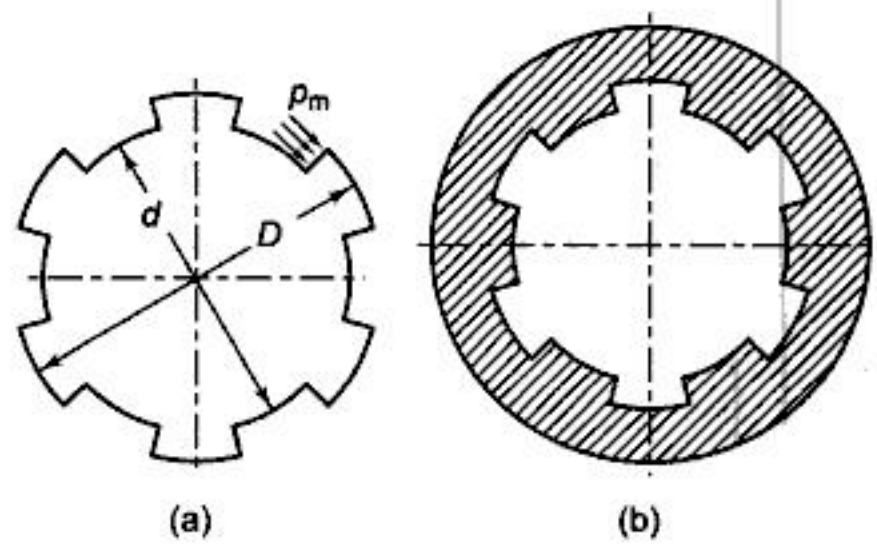

Fig. 9.27 *Splines: (a) Shaft (b) Hub*

D = major diameter of splines (mm)
d = minor diameter of splines (mm)
l = length of hub (mm)
n = number of splines

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(i) Calculate the diameter of each shaft by the following equations:

$$M_t = \frac{60 \times 10^6 (\text{kW})}{2\pi n} \quad \text{and} \quad \tau = \frac{16\, M_t}{\pi d^3}$$

(ii) Calculate the dimensions of the sleeve by the following empirical equations,

$$D = (2d + 13) \text{ mm} \quad \text{and} \quad L = 3.5\, d$$

Also, check the torsional shear stress induced in the sleeve by the following equations:

$$\tau = \frac{M_t r}{J} \quad J = \frac{\pi(D^4 - d^4)}{32} \quad r = \frac{D}{2}$$

(iii) Determine the standard cross-section of flat sunk key from Table 9.3. The length of the key in each shaft is one-half of the length of the sleeve. Therefore,

$$l = \frac{L}{2}$$

With these dimensions of the key, check the shear and compressive stresses in the key by Eqs (9.27) and (9.28) respectively.

$$\tau = \frac{2M_t}{dbl} \quad \text{and} \quad \sigma_c = \frac{4M_t}{dhl}$$

The shafts and key are made of plain carbon steel. The sleeve is usually made of grey cast iron of Grade FG 200.

Example 9.18 *Design a muff coupling to connect two steel shafts transmitting 25 kW power at 360 rpm. The shafts and key are made of plain carbon steel 30C8 ($S_{yt} = S_{yc} = 400$ N/mm^2). The sleeve is made of grey cast iron FG 200 ($S_{ut} = 200$ N/mm^2). The factor of safety for the shafts and key is 4. For the sleeve, the factor of safety is 6 based on ultimate strength.*

Solution

Given kW = 25 $n = 360$ rpm
For shafts and key, $S_{yt} = S_{yc} = 400$ N/mm^2 $(fs) = 4$
For sleeve, $S_{ut} = 200$ N/mm^2 $(fs) = 6$

Step I *Permissible stresses*
For the material of shafts and key,

$$\sigma_t = \frac{S_{yt}}{(fs)} = \frac{400}{4} = 100 \text{ N/mm}^2$$

$$\sigma_c = \frac{S_{yc}}{(fs)} = \frac{400}{4} = 100 \text{ N/mm}^2$$

$$\tau = \frac{S_{sy}}{(fs)} = \frac{0.5\, S_{yt}}{(fs)} = \frac{0.5(400)}{4} = 50 \text{ N/mm}^2$$

For sleeve material,

$$\tau = \frac{S_{su}}{(fs)} = \frac{0.5\, S_{ut}}{(fs)} = \frac{0.5(200)}{6} = 16.67 \text{ N/mm}^2$$

Step II *Diameter of each shaft*

$$M_t = \frac{60 \times 10^6 (\text{kW})}{2\pi n} = \frac{60 \times 10^6 (25)}{2\pi(360)}$$

$$= 663\,145.60 \text{ N-mm}$$

$$\tau = \frac{16\, M_t}{\pi d^3} \quad \text{or} \quad 50 = \frac{16(663\,145.60)}{\pi d^3}$$

$\therefore$ $d = 40.73$ or 45 mm

Step III *Dimensions of sleeve*

$D = (2d + 13) = 2 \times 45 + 13 = 103$ or 105 mm
$L = 3.5d = 3.5\,(45) = 157.5$ or 160 mm

The torsional shear stress in the sleeve is calculated by treating it as a hollow cylinder.

$$J = \frac{\pi(D^4 - d^4)}{32} = \frac{\pi(105^4 - 45^4)}{32}$$

$$= 11\,530\,626.79 \text{ mm}^4$$

$$r = \frac{D}{2} = \frac{105}{2} = 52.5 \text{ mm}$$

$$\tau = \frac{M_t r}{J} = \frac{(663\,145.60)(52.5)}{(11\,530\,626.79)}$$

$$= 3.02 \text{ N/mm}^2$$

$\therefore$ $\tau < 16.67$ N/mm^2

Step IV *Dimensions of key*
From Table 9.3, the standard cross-section of flat sunk key for a 45 mm diameter shaft is 14×9 mm. The length of key in each shaft is one-half of the length of sleeve. Therefore,

$$l = \frac{L}{2} = \frac{160}{2} = 80 \text{ mm}$$

The dimensions of the key are $14 \times 9 \times 80$ mm.

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The dimensions of the key are,

$$22 \times 14 \times 140 \text{ mm} \qquad \text{(v)}$$

Step VII *Check for stresses in key*

From Eq. (9.27),

$$\tau = \frac{2M_t}{d\,b\,l} = \frac{2\,(3\,978\,873.58)}{80\,(22)(140)}$$

$$= 32.3 \text{ N/mm}^2 \quad \therefore \tau < 40 \text{ N/mm}^2$$

From Eq. (9.28),

$$\sigma_c = \frac{4M_t}{dhl} = \frac{4(3\,978\,873.58)}{80(14)(140)}$$

$$= 101.5 \text{ N/mm}^2 \quad \therefore \sigma_c < 120 \text{ N/mm}^2$$

The design of key is safe from shear and compression considerations.

9.21 RIGID FLANGE COUPLINGS

A flange coupling consists of two flanges—one keyed to the driving shaft and the other to the driven shaft as shown in Fig. 9.33. The two flanges are connected together by means of four or six bolts arranged on a circle concentric with the axes of the shafts. Power is transmitted from the driving

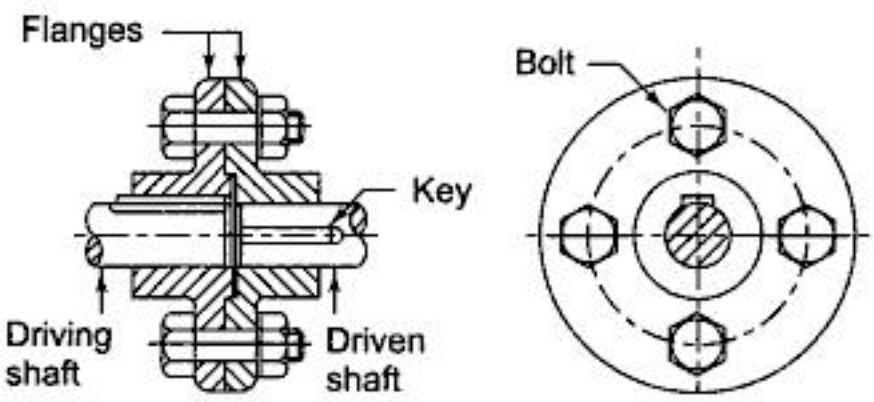

Fig. 9.33 *Unprotected Type Flange Coupling*

shaft to the left side flange through the key. It is then transmitted from the left side flange to the right side flange through the bolts. Finally, power is transmitted from the right side flange to the driven shaft through the key. Since flange coupling is rigid type of coupling, provision should be made for precise location of the axes of two shafts. The mating flanges have spigot and recess for precise location. The left side flange has a cylindrical projection called *spigot* while the right side flange has a corresponding *recess*. The diameters of the spigot and the recess are machined with more accuracy. During the assembly, the spigot fits into the recess and the two flanges are located precisely with respect to each other. This ensures alignment of the axes of the two shafts.

There are two types of rigid flange couplings—unprotected and protected. The flange coupling shown in Fig. 9.33 is the unprotected type of coupling. The revolving bolt heads and nuts are dangerous to the operator and may lead to accident. Protected type flange coupling is shown in Fig. 9.34. In this case, protecting circumferential rims cover the bolt heads and nuts. In case of failure of bolts while the machine is being run, the broken pieces will dash against this rim and eventually fall down. This protects the operator against injuries.

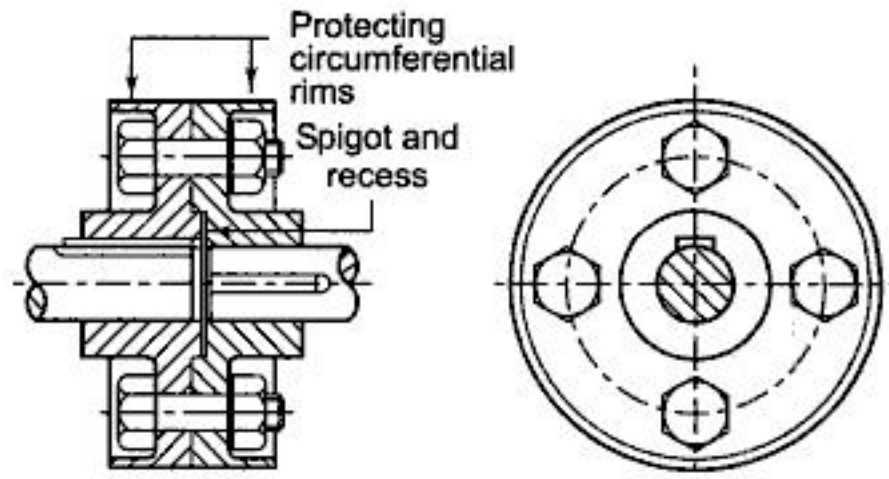

Fig. 9.34 *Protected Type Rigid Coupling*

The rigid flange couplings have the following advantages:

(i) Rigid coupling has high torque transmitting capacity.
(ii) Rigid coupling is easy to assemble and dismantle.
(iii) Rigid coupling has simple construction. It is easy to design and manufacture.

The rigid flange couplings have the following disadvantages:

(i) It is a rigid type of coupling. It cannot tolerate misalignment between the axes of two shafts.
(ii) It can be used only where the motion is free from shocks and vibrations.
(iii) It requires more radial space.

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Example 9.21 *A rigid coupling is used to transmit 50 kW power at 300 rpm. There are six bolts. The outer diameter of the flanges is 200 mm, while the recess diameter is 150 mm. The coefficient of friction between the flanges is 0.15. The bolts are made of steel 45C8 (S_{yt} = 380 N/mm^2) and the factor of safety is 3. Determine the diameter of the bolts.*

Assume that the bolts are fitted in large clearance holes.

Solution

Given kW = 50 n = 300 rpm μ = 0.15
For bolts, S_{yt} = 380 N/mm^2 (fs) = 3 N = 6
For flanges, D_o = 200 mm D_i = 150 mm

Step I *Permissible tensile stress*

$$\sigma_t = \frac{S_{yt}}{(fs)} = \frac{380}{3} = 126.67 \text{ N/mm}^2$$

Step II *Preload in bolts*

The torque transmitted by the shaft is given by,

$$M_t = \frac{60 \times 10^6 (\text{kW})}{2\pi n} = \frac{60 \times 10^6 (50)}{2\pi (300)}$$

$$= 1\,591\,549.4 \text{ N-mm}$$

$$R_f = \frac{2}{3}\frac{(R_o^3 - R_i^3)}{(R_o^2 - R_i^2)} = \frac{2}{3}\frac{(100^3 - 75^3)}{(100^2 - 75^2)} = 88.1 \text{ mm}$$

From Eq. (9.43),

$$P_i = \frac{M_t}{\mu N R_f} = \frac{1\,591\,549.4}{0.15(6)(88.1)} = 20\,072.51 \text{ N}$$

Step III *Diameter of bolts*

Due to pre-load of 20 072.51 N, the bolts are subjected to tensile stresses.

$$P_i = \left(\frac{\pi}{4}\right) d_1^2 \sigma_t$$

or $$d_1^2 = \frac{4P_i}{\pi\sigma_t} = \frac{4(20\,072.51)}{\pi(126.67)}$$

$\therefore$ $$d_1 = 14.2 \text{ mm}$$

Example 9.22 *It is required to design a rigid type of flange coupling to connect two shafts. The input shaft transmits 37.5 kW power at 180 rpm to the output shaft through the coupling. The service factor for the application is 1.5, i.e., the design torque is 1.5 times of the rated torque. Select suitable materials for various parts of the coupling, design the coupling and specify the dimensions of its components.*

Solution

Given kW = 37.5 n = 180 rpm
design torque = 1.5 (rated torque)

Step I *Selection of materials*

(i) The shafts are subjected to torsional shear stress. On the basis of strength, plain carbon steel of grade 40C8 (S_{yt} = 380 N/mm^2) is used for the shaft. The factor of safety for the shafts is assumed to be 2.5.

(ii) The keys and bolts are subjected to shear and compressive stresses. On the basis of strength criterion, plain carbon steel of grade 30C8 (S_{yt} = 400 N/mm^2) is selected for the keys and the bolts. It is assumed that the compressive yield strength is 150% of the tensile yield strength. The factor of safety for the keys and the bolts is taken as 2.5.

(iii) Flanges have complex shape and the easiest method to make the flanges is casting. Grey cast iron FG 200 (S_{ut} = 200 N/mm^2) is selected as the material for the flanges from manufacturing considerations. It is assumed that ultimate shear strength is one half of the ultimate tensile strength. The factor of safety for the flanges is assumed as 6, since the permissible stress is based on the ultimate strength and not on the yield strength.

Step II *Permissible stresses*

(i) Shaft

$$\tau = \frac{S_{sy}}{(fs)} = \frac{0.5\,S_{yt}}{(fs)} = \frac{0.5(380)}{(2.5)} = 76 \text{ N/mm}^2$$

(ii) Keys and bolts

$$\tau = \frac{S_{sy}}{(fs)} = \frac{0.5\,S_{yt}}{(fs)} = \frac{0.5(400)}{(2.5)} = 80 \text{ N/mm}^2$$

$$\sigma_c = \frac{S_{yc}}{(fs)} = \frac{1.5\,S_{yt}}{(fs)} = \frac{1.5(400)}{(2.5)} = 240 \text{ N/mm}^2$$

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