VIKRAM KAPSE

Process Equipment Design

M V Joshi

Ch. 8

M

chemical

First published 1976 by
The Macmillan Company of India Limited
Delhi Bombay Calcutta Madras
Associated companies throughout the world

SBN 33390 107 X

This book has been published with a subsidy under the Indo-American Textbook Programme operated by the National Book Trust, India

Rs. 30/-

Published by S G Wasani for The Macmillan Company of India Limited and printed at Rakesh Press, Naraina Industrial Area, New Delhi.

Preface

There are at present only a few books available on the subject of Process Equipment Design. The scope of these books is essentially limited to design of pressure vessels components and some machine elements. However, if the subject is to be made sufficiently comprehensive, much more emphasis is needed on design and construction features of different items of equipment. The present book is an attempt to fill this gap. It is expected to satisfy the long felt need for a more comprehensive book which sets out to embrace a wider field, viz., application of fundamental principles of machine design to specific items of equipment and elaborate details of construction illustrated through sketches and drawings. The book is intended both for under-graduate students in Chemical Engineering and for practising engineers in chemical industry. It is the outcome of experience gained by the author in class-rooms and industry.

Chemical industries involve problems in process design, unit operations, equipment design and overall plant design. In a design of a chemical plant these problems cannot be segregated. However, in spite of their interdependence, these problems may be advantageously segregated for study and development because of different principles involved in each. Process problems are primarily physico-chemical in nature. Unit operation problems are for most part physical, while equipment design problems are to a large extent mechanical. The fundamentals and theory of chemical engineering process design and unit operations are well covered in a number of books and handbooks. Overall sizing of equipment and its components is no doubt determined by the above considerations. Equipment design is therefore essentially limited to mechanical aspects of design and construction features of process equipment.

The first few chapters are devoted to a review of materials of construction, corrosion and protective coatings, stresses arising out of different loading conditions and factors which influence design. An outline of the design features of some machine elements, which form a part of chemical equipment, is presented in Chapter 5. Chemical equipment may be classified on the basis of certain common features with somewhat similar design procedures. Such classification leads to three groups, namely, pressure vessel group, structural group and rotational motion group. Chapters 6 to 16 deal with design and construction features of equipment from these three groups. Hazards and methods of protection relevant to equipment design are briefly discussed in Chapter 17. Use of computers in a design organisation is becoming an economical advantage. Fundamentals of the application of computers in design problems are presented in the last chapter, Numerical problems have been incorporated to illustrate the application of equations. At the end of each chapter references have been cited to enable the reader to locate the sources of information. The method of presentation is particularly suited to direct solutions of design problems. The theoretical development of equation is excluded. Details of constructional features are described and illustrated by a large number of sketches and drawings.

In view of the variety of equipment used in the chemical iadustry and its continuous development, it is difficult to state just what topics should be included in a book of this scope. The field is vast and ever-expanding. It is virtually impossible for one individual to be familiar with all the facets. It was therefore necessary to omit certain categories of equipment. The decision to include or omit is somewhat arbitrary based on practical considerations of time space and personal experience. However, an attempt has been made to include the important items of equipment.

Certain omissions or errors may have inevitably crept in the present volume. Readers are requested to bring these to the notice of the author and make suggestions for rendering the book more useful.

MVJ

Acknowledgements

The author wishes to acknowledge with sincere appreciation the encouragement received from Prof. N.R. Kamath and assistance given by friends and colleagues in particular Mr. C.S. Raje, Mr. R.S. Barve, Dr. A.S. Dighe, Mr. D.K. Korgonkar, Mr. L.V. Dixit and Mr. V.V. Mahajani. Thanks are due to Mr. D.V. Tamhankar who helped the author in checking the manuscript and press proofs and also to Messrs. V.C. Lad, A.N. Joshi and S.R. Kadam who prepared the sketches and drawings. Special thanks are due to Mr. K.B. Patil who typed the manuscript.

The author is particularly grateful to National Book Trust for granting financial subsidy, which helped a great deal in the publication of this book.

In the preparation of this book, information has been obtained from various sources such as text books, hand books, journals and catalogues. Indebtedness to these sources is freely acknowledged. Particular mention may be made of the following books whose publishers have permitted incorporation of figures, tables, etc.

1. *Unit Operations* by G.G. Brown, John Wiley & Sons, Figs. 264, 268 and 270.
2. *Process Equipment Design* by L.E. Brownell and E.H. Young ; John Wiley & Sons, Figs. 10.5, 11.19 and 11.20.
3. *Pressure Vessel Engineering Technology* by R. W. Nichols ; Applied Science Publishers Ltd., Tables 5 2a, 5.2b, 5.5a and 5.5b.
4. *Pressure Vessel Design and Analysis* by M.B. Bickell and C. Ruiz ; Macmillan and Co. Ltd., (London) Equations. 15 3 and 15.4.; Figs. 15.1 and 15.2; Table 1.1.

5. *Chemical Engineering Progress*; December 1954, American Institute of Chemical Engineers, Fig. 13.
6. *Indian Standard Specifications* ; Indian Standards Institutions, New Delhi. IS-803, 1962—Figs 6, 9, 10, 11 and 12 (Page Nos. 15, 18, 19 and 23) ; IS-4072, 1967 Tables 2 and 3 ; IS-5403, 1967 Table 1, 4 and 7 ; IS-2825, 1969 Tables 3.1, 3.2, 3.3, 3.4, 4.1, 4.3, 4.5, 6.3 and Appendix A-1; Figs. 3.22, 4.2, G.9, G.51, G.54, G.55, G.60, G.61.

(These standards are available from Indian Standards Institution, New Delhi and its branch offices at Ahmedabad, Bangalore, Bombay, Calcutta, Chandigarh, Hyderabad, Kanpur, Madras and Patna).

Contents

CHAPTER 1

Basic Considerations in Process Equipment Design

1.1 Introduction

In modern competitive chemical industry, new plants are being continuously set up and existing units modified and expanded. This involves both technical and economic evaluations. Knowledge of the various technical subjects such as thermodynamics, reaction kinetics, unit operations, process design, equipment design, etc., is a prerequisite to the establishment or development of any chemical plant. Of these the process along with the associated equipment, governs the shape and the size of the plant. The purpose of this book is to present the methods and the procedure adopted in designing process equipment. The emphasis here is not so much on the study of the actual process, but on specifying the function, operation and size of the equipment and also the choice of material of construction and strength considerations.

In early stages of the development of the chemical industry, the equipment was crude and the operation was essentially manual. During the nineteenth century, more elaborate processes such as the manufacture of sulphuric acid by chamber process or manufacture of alkali by the Leblanc process were developed. These processes called for more sophisticated equipment. A modern chemical process is, in general, even more complex and involves a series of operations which must be run continuously for many months or years. It demands equipment of exceptional robustness, ingenuity and reliability.

A variety of equipment is required for storage, handling and processing of chemicals. Each piece of equipment is expected to serve a specific function although in some cases it can be suitably modified to perform a different function. Conditions such as temperature, pressure, etc., under which the equipment is expected to perform are stipulated by the process requirements. Although the maximum capacity or size of the equipment may be specified, it is necessary to assure satisfactory performance even under certain amount of overloads. The overall satisfactory performance and reliability of the equipment will depend on the following factors :

(1) Optimum processing conditions
(2) Appropriate materials of construction
(3) Strength and rigidity of components
(4) Satisfactory performance of mechanisms
(5) Reliable methods of fabrication
(6) Ease of maintenance and repairs
(7) Ease of operation and control
(8) Safety requirements.

1.2 The General Design Procedure

It is difficult to suggest a standard design procedure. Such a procedure however would involve the following steps :

(*a*) Specifying the problem
(*b*) Analysing the probable solution
(*c*) Applying chemical process principles and theories of mechanics satisfying the conditions of the problem
(*d*) Selecting materials and stresses to suits processing conditions
(*e*) Evaluating and optimising the design
(*f*) Preparing the drawings and specifications

The first step is the recognition and understanding of the goal or objective. There is a need to specify the problem as precisely as possible. The solution of the problem may be based on a new method, scheme or idea or an old idea applied in a new way. Sometimes this requires a great deal of imagination, ingenuity and inventiveness. Sometimes it is quite a routine application or revision of an existing idea. The solution may involve a system, a process and an equipment to accomplish the specified task optimally, subject to certain

constraints. The method, scheme or idea must be analysed quantitatively to ensure that it can be made to work satisfactorily. After the analysis is made appropriate theories pertaining to process principles and mechanics must be applied. Use is made of the most basic principles available. Complex techniques or equations are used only when absolutely necessary. After selecting correct processing conditions, suitable materials and stresses, the problem requires a numerical answer. The calculations may require simple arithmetic, algebra or differential or integral calculus. In many cases exact solutions are not possible and various approximate techniques, such as graphical and numerical methods are necessary.

Sometimes assumptions are made because no absolutely accurate values or methods of calculation are available. Methods involving close approximations are used because exact treatments would require long and laborious calculations giving little gain in accuracy. Assumptions and approximations are made only when they are necessary and essentially correct. In some equipment, only a portion of the total number of parts are designed on the basis of analytical calculations. The form and the size of the remaining parts are then usually determined by practical conditions.

Every step is checked both in respect of mathematical calculations and engineering feasibility. It is necessary to ascertain whether the results are consistent with experience and are practical. In some problems the final result is obtained by optimisation. This is carried out by taking into account the influence of all the controllable parameters, that make the final result a maximum or minimum. It may take several iterations before the satisfactory solution is obtained.

Experience in designing similar machines or equipment will be helpful in choosing suitable sizes. A designer should realise the practical limitations involved in the application of his work. In arriving at the final dimensions of any component, based on optimum theoretical and economic considerations, the designer should ascertain the exact shape and the standard sizes in which materials are available and adopt these as far as possible. In the case of a new equipment, fabrication and testing of a prototype for satisfactory performance would be essential, before the design is finalised. Design calculations must be accompanied by sketches. Based on these sketches detailed and assembly drawings can be made, specifying properly all dimensions, materials selected, tolerances, fits and limits, required surface finish, etc.

Provision must be made during different stages in the development of the design to facilitate assembly and installation of machine or equipment. Similarly problems connected with dismantling and maintenance should be considered.

1.3 Fabrication Techniques

An economical and reliable fabrication technique must be adopted, to manufacture the various components of a machine or an equipment. In fact, the selection of materials and an appropriate choice of the method of fabrication do form an important aspect, influencing the design considerations. The fabrication techniques may be classified in two groups.

(*a*) Techniques adopted to give an approximate shape to the material. Such methods are casting, forging, rolling, extrusion, drawing, stamping and welding.

(*b*) Techniques adopted to impart the final precise dimensions and ensure the desired surface finish. Such methods are planing, shaping, turning, milling, drilling, boring, reaming, broaching, grinding, honing, polishing, electroplating, coating, etc.

1.4 Equipment Classification

Classification of chemical equipment is generally based on the particular type of unit operation. Each equipment is therefore designed for carrying out a specific unit operation such as distillation, evaporation, solid-liquid separation, etc. Equipment may also be classified to emphasise certain common features, which require similar design procedures. Such classification leads to three groups.

(A) *Pressure Vessel group*—This group of equipment has a cylindrical or spherical vessel as the main component, which

has to withstand variations in pressures and temperatures, in addition to other loading conditions. The design procedure therefore is based on satisfying a number of criteria, involving different loading conditions. Such equipment which covers a major part of this book are included in Chapters 6 to 12.

(B) *Structural group*—This group consists of equipment or components which are stationary and have to sustain only dead loads. They are generally made up of structural sections and must satisfy conditions of elastic and structural stability. Such items are covered in Chapter 13, and a few others in Chapters 15 and 16.

(C) *Group involving rotational motion*—This group covers equipment or components where a rotational motion is necessary to satisfy process requirements. A drive system and power supply are essential features. Considerations of torque, dynamic stresses, apart from other loading conditions form the basis of design. Such items are covered in Chapters 14 to 16.

Each equipment may not strictly belong to only one group, but may have features involving combination of the above groups. In such cases it may be possible to design components belonging to each group independently and then combine them to form a complete equipment.

1.5 Power for Rotational Motion

Rotational motion often involves assessment of power requirements. In arriving at the maximum horse power an analysis should be made of the mechanical operation of the equipment in terms of torque at the motor shaft, and also the speed requirements.

Torque—The value of the torque varies according to the stipulated conditions of working.

(*a*) Starting torque, i.e., torque required to overcome static friction and produce motion.

(*b*) Accelerating torque, i.e., torque required to accelerate the driven equipment to full speed.

(*c*) Running torque, i.e., torque required to drive the equipment or machine under normal conditions at a specified speed.

On the basis of the above torques, it is possible to classify different types of mechanical loads.

(*a*) *Friction loads*—In this type of loads, static or dynamic friction is the main component responsible for torque requirements. Equipment subject to such loads are conveyers, grinders, rotating filters, rotary driers, etc.

(*b*) *Fluid viscosity loads*—In this type of loads, viscous friction between layers of fluid is responsible for torque. Equipment with fluid viscosity loads are agitators, fans, pumps, etc.

(*c*) *Accelerating loads*—In certain loads accelerating motion forms a major portion of the torque apart from frictional torque. Such equipments are cranes, elevators, etc.

Speed—Equipment with rotary motion may be run (*a*) at constant speed (*b*) at two or three specified speeds or (*c*) at variable speeds.

Horse Power—The horse power can be calculated by the following equation:

$$\text{H.P.} = \frac{2\pi\ N\ T}{75 \times 60} \tag{1.1}$$

where T—Torque in kg-m
N—R.P.M.

Since during the operation of any equipment, both the torque and speed are likely to vary according to the loading conditions, it is necessary to draw a horse power duty cycle, showing the variation of horse power requirement with time of operation (Fig. 1.1). Based on this it is possible to determine the maximum power required as well as the rated power of the prime mover, which usually consists of an electric motor or a turbine.

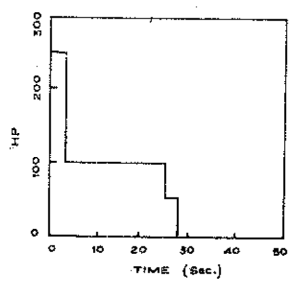


Fig. 1.1 Horse Power Duty Cycle

Reading References

Dixon J.R., *Design Engineering*, McGraw-Hill Book Co., New York, (1966).

Cremer H.W., *Chemical Engineering Practice* Vol. IX, Butterworths, London, (1965).

Schwartz M.M., *Modern Metal Joining Techniques*, John Wiley and Sons, New York, (1969).

CHAPTER 2

Materials and Protective Coatings

2.1 Properties

A designer of machines or equipment must be well acquainted with a large number of materials available. In choosing such materials, the most important criteria are obviously the various properties of the materials. The final choice will, however, depend on the ease of fabrication and the overall cost. Certain properties of materials can be well defined, others can be stated in relative terms, while a few may be rather vague. The basic properties are composition, structure, specific weight, thermal conductivity, expansibility and resistance to corrosion. Properties such as strength, elastic limit, modulii of elasticity, endurance limit, resilience, toughness, ductility, brittleness and hardness are termed as mechanical properties. From the point of view of fabrication, machinability, weldability and malleability might be considered as relevant properties. While designing equipment, apart from process requirements an appropriate choice of the materials of construction depends on the clear understanding of the mechanical properties of materials, which are therefore considered in some detail in the following paragraphs.

2.1.1 STRENGTH

Strength represents the capacity of the material to withstand external forces. Depending on the nature of the force, strength can be classified as tensile, compressive, shear and impact. External forces are resisted by the material, and therefore induce stresses and deformations. Stress may be simply defined as internal force of resistance per unit area, while deformation

in the original size and/or shape of the material is defined as strain.

In linear elastic deformation, stress and strain are related by Hooke's law

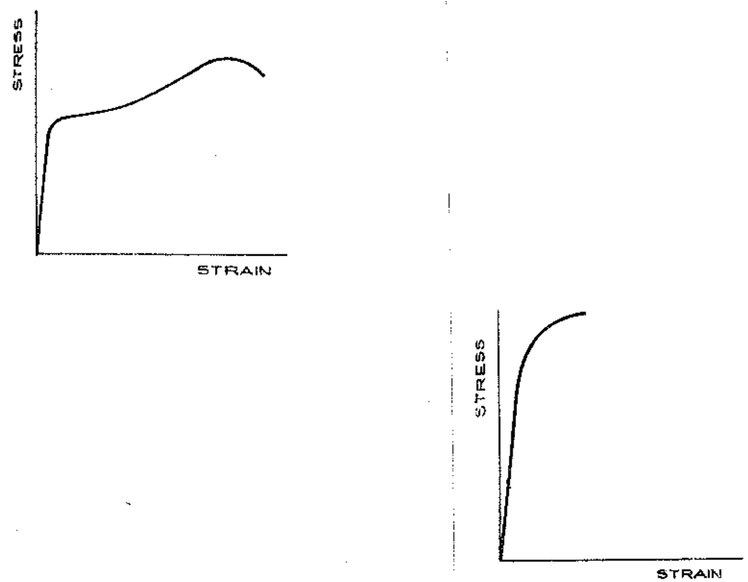
$$\text{Strain} = \frac{\text{Stress}}{\text{constant}} \quad \text{or} \quad \varepsilon = \frac{f}{E} \tag{2.1}$$

where E is known as Young's modulus of elasticity. In non linear elastic behaviour

$$\varepsilon = \left(\frac{f}{\lambda}\right)^n \tag{2.2}$$

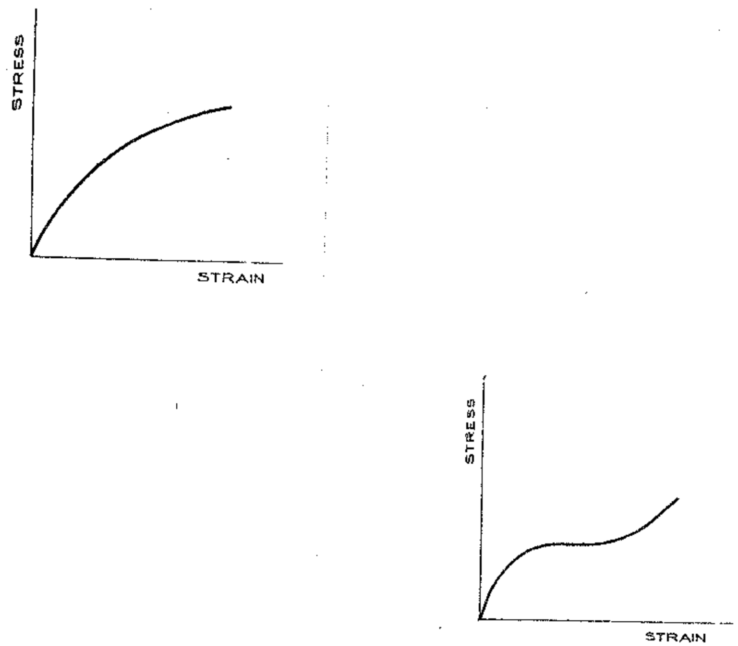
where, λ — pseudoplastic modulus
n — constant

The stress-strain relationship of different materials can be determined by uniaxial tension or compression tests. Some typical stress-strain diagrams are shown in Fig. 2.1.



Figs. 2.1 (a) and (b)

Fig. 2.1 (a) indicates the deformation produced in a ductile material, while (b) represents deformation of a brittle material,



Figs. 2.1 (c) and (d)

showing very limited deformation. Large nonlinear deformations, representing highly elastic materials are shown in (c) and (d).

2.1.2 Stiffness And Rigidity

It is a measure of the ability of the material to resist deformation.

2.1.3 Elasticity

Elasticity is the ability of the material to regain its original shape as soon as the load is removed. In the design of most of the components, permanent deformations are generally avoided, and the material is utilised with a view to retaining its elasticity.

2.1.4 Proportional Limit

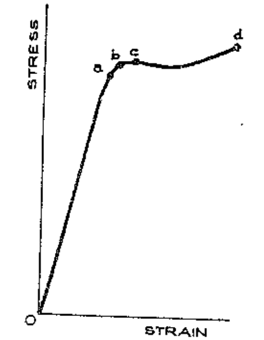


Fig. 2.2

Fig. 2.2 represents a stress-strain graph of an elastic material. Point (a) on the graph indicates the greatest stress upto which the Hooke's law of proportionality of stress to strain is observed. This point is therefore known as proportional limit.

2.1.5 Elastic Limit

It is the maximum stress, indicated by point (b) in Fig. 2.2 which the material can withstand without permanent deformation. It is generally very close to proportional limit.

2.1.6 Yield Stress

It is the stress at which the resistance of the molecules of the material begins to break down rapidly and a sudden large increase in strain occurs without an increase in stress. This is indicated by point (c) in Fig. 2.2.

2.1.7 Proof Stress

For certain materials there is no well defined stress at which yielding occurs, the stress-strain diagram being curved almost from origin (Fig. 2.3). If a tangent to the curve at the origin is drawn (*oh*) and a line (*fg*) cutting the curve at (*g*), such that (*of*) = 0.1% then the stress at (*g*) is called 0.1% proof stress. It is the stress at which the strain has departed 0.1% of the gauge length from the line of proportionality (*oh*).

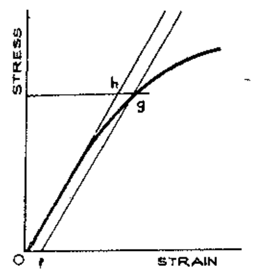


Fig. 2.3

2.1.8 Ultimate Stress

It is the greatest stress at which the failure of the material takes place. The portion of the stress-strain curve between the yield stress (c) and ultimate stress (d) in Fig. 2.2 is termed as plastic range. Hooke's law is not applicable to this range and the deformation produced is permanent.

2.1.9 Ductility

It is a measure of the deformability of the material, determined by percentage of elongation or reduction of area.

2.1.10 Resilience

It is the elastic energy released by the material, as a result of the stress removal. It is measured by finding out the area under the stress-strain curve within the elastic limit. The stress can be tensile, compressive or shear.

2.1.11 Toughness

It is the ability of the material to absorb energy in deformation in the plastic range and is measured by finding out the total area of the stress-strain curve.

2.1.12 Hardness

It represents the surface characteristics of a material and can

be assessed by the resistance it offers to indentation and scratching.

2.1.13 Fatigue

Fatigue represents the failure of the material when subjected to the application of a cyclic load, which causes a progressive enlargement of an initial small crack.

2.1.14 Creep

It is a slow and progressive deformation of a material with time under constant stress.

The mechanical properties stated above are affected by the temperature conditions under which the material has to function. In general an increase in temperature reduces the strength of the material. In addition, other conditions such as state of stress, strain rate, size and shape of the component and method of fabrication are likely to have a significant effect on the mechanical properties.

2.2 Resistance to Corrosion

Apart from the mechanical properties stated so far, another important property is corrosion resistance. Corrosion causes destruction of metals by chemical or electrochemical action. The majority of the corrosion failures can be classified according to the appearance of the corroded component.

Uniform corrosion—This involves general overall attack and the part fails due to thinning. The cause is the choice of wrong material.

Two-metal corrosion or galvanic corrosion—This is due to contact of two dissimilar metals. In such cases corrosion of the less resistant (anodic) metal is accelerated and the more resistant (cathodic) metal is protected.

Crevice corrosion—It is an intense, localised corrosion, which occurs within crevices and other shielded areas. This form of attack is associated with small volumes of stagnant solutions in holes, gasket surfaces, lap joints, bolt heads, etc.

Pitting—It is a form of extremely localised attack which results in holes and perforations.

Intergranular corrosion—It is an attack which occurs only on the grain boundaries and not the grains of metal. The structure disintegrates because the grains fall out.

Selective leaching—It is the removal of one element from a solid alloy by the corrosion process. This type of corrosion is not readily detected, since the dimensions of the component do not change, and breakage occurs suddenly without any indication.

Stress corrosion—It is the failure due to a combination of stress and corrosion. There is no overall corrosion and it is limited to a specific environment.

Fatigue corrosion—It is stress corrosion, where the stresses are cyclic instead of static. This contributes greatly to the initiation of corrosion fatigue, in that a surface pit acts as a stress raiser.

Erosion corrosion—It is an attack due to combination of corrosion and mechanical wear effect (erosion). This usually occurs when handling liquids at high velocities and/or with solids in suspension. Many metals and alloys depend on surface films for resistance to corrosion. When this film is worn off, the metal becomes active and rapid deterioration results.

2.3 Choice of Material

Various materials of construction used in chemical equipment, have to withstand high temperatures, pressures and flow rates. These severe operating conditions intensify corrosive action. A broad range of materials is now available for corrosive service. The task of selecting the best material is rather difficult. Final choice of the material cannot be made merely by choosing a suitable material having the requisite mechanical behaviour and anti-corrosive properties, but must be based on a sound economic analysis of competing materials. Many factors enter into economic considerations. A high materials cost may be accompanied by easier fabrication, with significant saving in labour, or it may be accompanied by long life owing to a

property such as corrosion resistance so that annual cost of replacement would be considerably lower. Sometimes a material must be selected, even though it has inferior properties, because the right type material is too expensive or is not available.

2.4 Materials

Materials of construction may be divided into two general categories viz., metals and non-metals. Table 2.1 gives physical constants of some common materials. For properties of materials see Appendices C, D and E.

2.4.1 Metals

2.4.1.1 Ferrous metals

These are the most important metals, which are known for their strength, wide availability and ease of forming and machining. A wide range of properties can be obtained by suitable alloying. According to the percentage of carbon these are classified as wrought iron, cast iron and steel.

Wrought iron—This is almost pure iron. It is used mainly for chain links, hooks and couplings, due to its malleable and ductile nature.

Cast iron—This has usually 2.5 to 4.5% carbon. In general it is cheap and easy to cast even in complicated shapes because of its fluidity in molten state. Cast iron can be classified as (i) grey cast iron (ii) white cast iron, depending on the form in which the carbon is distributed in iron. In grey cast iron the carbon is in the form of graphite, distributed as free crystals throughout the iron. In white cast iron there is a chemical combination of iron and carbon. The formation of the two types of cast irons depends on the rate at which it is cooled from the molten state.

Grey cast iron—The compressive strength of this is about three to four times its tensile strength, and hardness number between 180 to 240. Generally cast iron has no outstanding resistance to acids or acid solutions. It is not corroded by alkali solutions of less than 30% concentration at any temperature. Its applications are in pumps and valves, special piping, acid coolers, acid reaction and concentration vessels. Many reaction kettles and vats are made of enamelled cast iron. It is particularly used for those enamelled components which are too complex to be readily fabricated.

In general cast iron is used to produce components whose complicated shape introduces fabricating difficulties and in cases where resistance to abrasion and corrosion is desired. The components produced must have enough thickness and are therefore rigid.

White cast iron—Due to its highly brittle nature it is seldom used as such. Usually grey cast iron is chilled to produce a hard superficial depth of white cast iron. It is used in such parts as furnace bars, teeth and jaws of crushing and grinding machines. Such components have a good wear resistance. White cast iron can be made malleable by heat treatment. The resultant castings are able to withstand shocks better, and are used for pipe fittings, gear wheels, housings, pulleys, etc.

A high duty cast iron can be produced by the use of small quantities of calcium silicide. The resultant structure after heat treatment has better tensile and shear strength. It is used for fabricating pressure vessels and high pressure and temperature pipes and fittings. It has a higher rate of contraction than ordinary cast iron.

Alloy cast iron—It is made by adding such elements as silicon, copper, nickel, molybdenum and chromium in suitable proportions, to confer certain properties to cast iron.

High silicon iron is essentially an acid resistant iron but is very hard and brittle. Chromium gives great resistance to heat, wear and corrosion. Molybdenum produces a marked improvement in strength—especially with copper or nickel.

Steel—Steel is iron alloyed with carbon between 0.05 to 2.0%. In addition it contains smaller proportions of phosphorus, sulphur, silicon and manganese. These steels are known as plain carbon steels. In some case scertain other alloying elements are added; such steels are known as alloy steels.

Table 2.1

Physical constants of some common materials

Material	Modulus of Elasticity E kg/cm^2	Modulus of rigidity (G), kg/cm^2	Poisson's ratio, μ	Density gm/cm^3
Aluminium (alloys)	0.724×10^6	0.267×10^6	0.334	2.73
Beryllium	2.928×10^6	—	—	1.82
Beryllium copper	1.266×10^6	0.492×10^6	0.285	8.23
Brass	0.970×10^6	0.350×10^6	0.30—0.40	8.45
Bronze	1.110×10^6	—	—	8.73
Carbon steel	2.060×10^6	0.800×10^6	0.292	7.82
Cast iron, grey	1.020×10^6	0.422×10^6	0.211	7.20
Copper	1.230×10^6	0.390×10^6	0.260	8.96
Inconel	2.180×10^6	0.775×10^6	0.290	8.96
Lead	0.160×10^6	0.076×10^6	0.450	11.34
Magnesium	0.457×10^6	0.169×10^6	0.350	1.80
Molybdenum	3.375×10^6	1.195×10^6	0.307	10.20
Monel metal	1.828×10^6	0.668×10^6	0.320	8.83
Nickel silver	1.300×10^6	0.492×10^6	0.322	8.69
Nickel steel	2.000×10^6	0.780×10^6	0.291	7.75
Phosphor bronze	1.132×10^6	0.422×10^6	0.349	8.16
Stainless steel (18—8)	1.940×10^6	0.745×10^6	0 305	7.75
Titanium	1.055×10^6	—	—	4.48
Tungsten	4.153×10^6	1.770×10^6	0.170	19.30
Zirconium	$0,697 \times 10^6$	—	-	6.50

Plain carbon steels—According to the percentage of carbon these steels are classified as low or mild, medium and high carbon steels.

Low carbon steel or Mild steel—It has 0.05 to 0.3% carbon. It is undoubtedly one of the most versatile materials available. It has a good strength and ductility. It can be easily rolled, forged and drawn. Fabrication by welding and machining is also easily carried out. For chemical equipment it is therefore used on a large scale. Some of the applications are in pressure vessels, pipes and fittings, machine components and structural sections. The corrosion resistance of mild steel is low. If the rate of corrosion is slow it is certainly economical to use such steel. In cases, where contamination of the product with corrosion products cannot be accepted, the surface of steel may be protected by paint, metal spraying or by other protective linings. This type of protection is not satisfactory for severe corrosive conditions, because it is usually not possible to obtain a perfect coating.

Medium carbon steel—It has a carbon content between 0.3 and 0.5%, which helps to improve hardness, strength and fatigue resistance. The material is used for fabrication of shafts, springs, gears, bolts and certain structural sections.

High carbon steel—It has a carbon content of more than 0.5% up to a maximum of 2.0%. It is much harder and stronger than the two preceding grades, but much less ductile. The material is primarily used for cutting tools and dies.

Alloy steels—A number of elements, such as Nickel, Chromium, Silicon, Manganese, Molybdenum, Tungsten, Beryllium, Vanadium, Cobalt, and Titanium are used as alloying elements. Generally, certain properties are associated with certain elements.

Manganese—abrasion resistance and toughness
Nickel and chromium—corrosion and high temperature resistance.
Chromium, tungsten, vanadium, molybdenum, cobalt—cutting action
Nickel, chromium, titanium—creep resistance
Silicon, manganese—elasticity

A few of the important alloys used in the chemical industry are mentioned here.

Low alloy steels—These contain small percentages (below 10%) of nickel, chromium, molybdenum, manganese, etc., in different combinations. These are used primarily for machine components and to a limited extent for pressure vessels. In these steels the carbon content is usually of the order of 0.15%, manganese about 1.0% and silicon 0.3%. Steels with ½%, 1%, 2¼%, 3% and 5% chromium have better corrosion resistance than ordinary mild steel and are used for high temperature service under mildly corrosive conditions. For bolting, medium carbon steel with 1% Cr—Mo with or without vanadium is most suitable.

High alloy steels—These are of two types: Straight chromium steels, with chromium content ranging from 13 to 17% and chromium-nickel steels or stainless steels, with chromium contents between 18 and 25% and nickel contents between 8 and 20%. The carbon content varies between 0.04 and 0.25%. According to the carbon content, straight chromium steels with 0.08% are used for plates, 0.1% for forgings, 0.2% for coatings and 0.15% for bolting. Only those varieties with a carbon content higher than 0.1% are hardenable by heat treatment.

Special mention may be made of stainless steels. These are essentially high chromium or high nickel-chromium alloys of iron containing small amounts of other essential elements. They have excellent corrosion resistance and heat resistance properties. The most common variety of stainless steel contains approximately 18% chromium and 8% nickel. Addition of molybdenum to the alloy increases corrosion resistance and high temperature strength. Certain varieties of stainless steel are hardenable by cold working, certain others by heat-treatment.

Stainless steels exhibit the best resistance to corrosion when the surface is oxidized to a passive state. This condition can be obtained at least temporarily by a passivation operation in which the surface is treated with nitric acid and then rinsed with water. Localized corrosion can occur at places where foreign material collects, such as scratches, crevices and corners.

There are more than a hundred different types of stainless steels, which can be classified according to the alloy content, microstructure and major characteristics. According to microstructure and alloy content, they are classified as (i) austenitic (ii) martensitic and (iii) ferritic. The major applications in the chemical industry are

(*a*) Vessels such as storage tanks, reactors, absorption and distillation columns and heat exchangers.
(*b*) Machinery such as pumps, fans, compressors, centrifuges, dryers, coolers and filters.
(*c*) Materials handling equipment such as pipes, conveyers and tankers.

The American Iron and Steel Institute (AISI) numbering system to standardise stainless steels is widely accepted. These are indicated in Table 2.2. Table 2.3 gives a comparative study of Indian and foreign standard stainless steels. Amongst these the more important types used for process equipment are indicated in Table 2.4 based on AISI recommendations.

Stainless steel is used as a cladding material for plain carbon or low alloy steel. Economically it is important to choose a suitable thickness of cladding. Generally it will be about 8—20% of the total thickness, 15% for plates of thickness less than 20 mm and 10% for higher thickness.

2.4.1.2 NON-FERROUS METALS

Aluminium and alloys—These materials are light and easy to fabricate. Commercial pure grade aluminium is a relatively weak material. Alloys of aluminium have better mechanical properties. The most common alloying elements are copper, silicon, zinc, magnesium, nickel and manganese. Aluminium resists attack by acids because a surface film of inert hydrated aluminium oxide is formed. This film adheres to the surface and offers good protection. Aluminium alloys are satisfactory for low temperature service upto—150°C. Some of the more important alloys are duralumin with 4% copper, 0.6% magnesium, 0.5% manganese ; aluminium-silicon alloys ; and aluminium-zinc-magnesium alloys. The last ones are the strongest alloys of aluminium. These alloys are used for fabrications of storage vessels, reaction vessels, heat exchangers and absorption columns.

Copper and its alloys—Pure copper has good ductility, malleability, high electrical and thermal conductivity. It has fair mechanical strength and can be fabricated easily into a wide variety of shapes. It can be easily joined by brazing, welding and soldering. Copper exhibits good corrosion resistance to strong alkalies, and organic solvents. It is resistant to atmospheric moisture or oxygen because a protective coating of copper oxide is formed on the surface.

Copper is alloyed with zinc, tin, nickel, aluminium and lead. In special cases beryllium and manganese are also used. These alloys have greater strength and provide greater ease in

TABLE 2.2

Group	Structure	Hardenability	AISI Type	Analysis built up from basic types
Chromium Nickel	Austenitic	Hardenable by Cold Work	302	Cr 18%+Ni 8% basic type
			301	Cr and Ni lower for more work hardening
			304	C lower to avoid carbide precipitation
			305	Ni higher for less work hardening
			302B	Si higher for more scaling resistance
			303	S or Se added for easier machining
			316	Mo added for more corrosion resistance
			317	Mo higher for more corrosion resistance and strength at heat
			318	Cb added to avoid carbide precipitation
			321	Ti added to avoid carbide precipitation
			347	Cb added to avoid carbide precipitation
			347F	Se added to improve machinability
			308	Cr and Ni higher with C low for more corrosion and scaling resistance
			309	Cr and Ni still higher for more corrosion and scaling resistance
			309S	C lower to avoid carbide precipitation
			309C	Cb added to avoid carbide precipitation
			310	Cr and Ni highest to increase scaling resistance
Chromium	Martensitic	Hardenable by Heat Treatment	410	Cr 12% basic type
			403	Cr 12% adjusted for special physicals
			414	Ni added to increase corrosion resistance and physicals
			416	S or Se added for easier machining
			418	W added to improve high temperature properties
			420	C higher for cutting purposes
			420F	S or Se added for easier machining
			431	Cr higher and Ni added for better resistance and property
			440A	C higher for cutting applications
			440C	C still higher for wear resistance
			440F	S or Se added for easier machining
	Ferritic	Non-hardenable	405	Al added to Cr 12% to prevent hardening
			430	Cr 17% basic type
			430F	S or Se added for easier machining
			442	Cr higher to increase scaling, resistance
			446	Cr much higher for improved scaling resistance

Table 2.3

IS	BS	AISI	SIS	DIN
Grade 1	405 S 17	405	—	X7 Cr A113 (1.4002)
Grade 2	410 S 21	410	—	—
Grade 3	420 S 37	420	2303	X20 Cr 13 (1.4021)
Grade 4	420 S 45	—	—	—
Grade 5	—	—	—	—
Grade 6	—	430	2320	X8 Cr 17 (1.4016)
Grade 7	431 S 29	431	—	—
Grade 8	—	440 C	—	—
Grade 9	304 S 12	304 L	—	—
Grade 10	304 S 15	304	2332	X5 CrNi 18 9(1.4301)
Grade 11	302 S 25	305	2330	X12 CrNi 18 8(1.4300)
Grade 12	—	301	—	X12 CrNi 17 7(1.4310)
Grade 13	321 S 12	321	2337	X10 CrNi Ti 18 9 (1.4541)
Grade 14	347 S 17	347	—	X10 Cr Ni Nb 18 9 (1.4550)
Grade 15	316 S 16	316	—	—
Grade 16	316 S 12	316 L	—	—
Grade 17	320 S 17	—	—	—
Grade 18	—	201	—	—
Grade 19	—	446	—	X10 Cr 25 (1.3811)
Grade 20		309	—	—
Grade 21		310	—	—

machining and coating. Copper-zinc alloys, known as brasses, have upto 45% zinc and are suitable for tubes, wires and sheets, which can be cold worked. Copper-tin alloys, known as bronzes, contain tin upto 5% for wrought work and about 10% for castings. These alloys are used widely for pumps, valves, pipe fittings etc. Copper-aluminium alloys contain upto 14% aluminium, and are known as aluminium bronzes. They are highly resistant to oxidation and scaling and for that reason are among the best copper alloys for service at moderately elevated temperatures. They include some of the strongest copper alloys. These alloys are used for pump castings, condenser tubes, valve seats, etc. Copper-nickel alloys contain 20

Table 2.4

Stainless Steels for the Chemical Industry

Types 304, 304L, 321, 347	general corrosion resistance.
Types 308, 309, 309S, 310, and 310S	largely for high-temperature work, such as preheaters and vessel liners, as well as for welding rod
Types 316, 316L	general corrosion resistance especially when Types 304 and 304L are subject to pitting or excessive corrosion
Type 317	for applications where somewhat better corrosion resistance than can be obtained from Types 316 and 316L is needed.

Note. Type 304 is the *general purpose* grade for the Process Industries

to 30% copper. They have good strength, ductility and corrosion resistance. These alloys are extensively used for condenser fabrication. Copper and its alloys have been used in chemical industry over a long period. Copper itself is nearly always used in the wrought form as sheet or tube, while the fittings such as valves, pumps and other cast parts are fabricated from one of the copper alloys. They are used for evaporators, stills, condensers and heat exchangers.

Nickel and its alloys—Nickel exhibits high corrosion resistance to most alkalies. The strength and hardness of nickel is almost as great as that of carbon steel and the metal can be fabricated easily. For economical use of nickel, steel clad with nickel is used extensively in the production of caustic soda and alkalies. An alloy of nickel and copper, known as monel contains 67% nickel and 30% copper. The resistance of monel to molten salts and alkali metals is inferior to that of pure nickel, but the copper content imparts superior resistance to non-oxidising chloride solutions and to hot dilute solutions of non-oxidising acids. The alloy is used in food industries. In chemical industries it is used in evaporators, heat exchangers, etc. An alloy of nickel-chromium-iron, known as inconel, contains 76% nickel, 15% chromium and the balance mainly iron. The presence of chromium in this alloy increases its resistance to oxidizing conditions. It has also a good heat resistance. Applications of inconel include heat exchangers, digesters, etc. Nickel-molybdenum alloys, known as hastelloys, have high resistance to corrosion over wider ranges of materials with higher concentrations and temperatures. Hastelloy C has 56% nickel, 17% molybdenum, 16% chromium, 5% iron and 4% tungsten. It has excellent corrosion resistance even at high temperatures. It is used for valves, piping, heat exchangers and vessels. To meet very severely corrosive conditions nickel-molybdenum alloys are often the only alternatives to precious metals like platinum and silver.

Chromium and its alloys—Chromium is used either for plating of iron and steel, or as an alloying element in low and high alloy steels. It helps to improve corrosion resistance, creep resistance, wear resistance and the cutting action of steels. The above combination of properties makes chromium stubbornly resistant to abrasion, enabling it to preserve surfaces at exact dimension. Electroplated chromium can produce a surface which is either mirror-bright, dulled grain or an intermediate texture, which can be maintained for continuity of production.

Lead—It is one of the oldest metals, known for its resistance to corrosion. It has low creep and fatigue resistance. It is soft and malleable. Due to these properties lead is used for lining or cladding of metal equipment. The corrosion resistance of lead depends on the fact that a protective surface film is formed. If the surface film is one of highly insoluble lead salts, such as sulphate, carbonate or phosphate, good corrosion resistance is obtained. Lead sheets and pipes are used for fabrication of equipment.

Titanium—This material is available over a wide range of tensile strengths, ranging from that obtainable from carbon steel upto those of highly alloyed steels. However, its elastic modulus is only about half that of steel. It resists corrosion due to sea-water, brine, chlorides, and is unaffected by oxidants, such as nitric acid and many organic acids. The corrosion resistance of titanium depends on the formation and maintenance of a thin oxide film. The material can be used

for pumps, valves and heat exchangers. It can also be used for lining of mild steel vessels. In all such applications it is necessary to choose the correct grade of the metal to suit the particular component and conditions.

Beryllium—This has low ductility, but its modulus of elasticity is usually high. It is stronger than steel and retains its strength at high temperatures. Beryllium is used primarily as an alloying element due to its corrosion resistance, and excellent wear and fatigue properties. Due to its high strength/weight ratio combined with corrosion resistance, it is used in special applications, such as reactor components.

Zirconium—This has good mechanical properties at room temperature, but retains strength at elevated temperature only if it is highly alloyed. It is completely resistant to hydrochloric acid and oxidising media such as solutions of nitric and chromic acid. The material is used in special applications such as thermometer pockets, sheaths, etc.

Tantalum—This has good mechanical properties at room and elevated temperatures. The chemical behaviour of tantalum which is scarcely attacked by most acids, but is corroded by hot strong alkalies, is often compared with that of glass. It is resistant to both oxidising and non-oxidising media. The metal is used for construction of special heat exchangers.

2.4.2 NON-METALS

2.4.2.1 CARBON AND GRAPHITE

These comprise a family of materials. Each member is essentially pure carbon, but differs from others in such things as basic structure, orientation of the crystallites, size and number of pore spacings, degree of graphitization, etc. Consequently it is possible to manufacture many different grades of carbon and graphite, ranging from coarse-grained, relatively weak materials to the fine-grained, strongest and purest materials. Impervious graphite and carbon are made by impregnating them with a suitable resin.

These materials are anisotropic and therefore their properties vary with grain orientation. The corrosion resistance of both carbon and graphite is almost unsurpassed. Graphite is generally inert to all chemicals except in strongly oxidizing applications. Unlike metals, these materials do not depend for corrosion resistance on the formation of an inherent oxide or other film. This is particularly important in chemical process equipment in which erosion and corrosion can combine to destroy the metal that is protected by a film. Carbon and graphite have an unusual range of thermal conductivity. Carbon is a fairly good insulator, while graphite is a good conductor of heat. The low thermal conductivity of carbon is advantageous when carbon bricks are used to line vessels for high temperature reactions. On the other hand, the high coefficient of thermal conductivity of graphite makes it possible to manufacture efficient, compact heat exchangers.

2.4.2.2 Glass

Commonly available glass has a low tensile strength, but a high compressive strength. It is brittle and is damaged by thermal shock. Glass has excellent resistance to corrosion and is subject to attack only by hydrofluoric acid and hot alkaline solutions. Coating of steel by glass gives a material which has the corrosion resistance of glass and the strength of steel.

2.4.2.3 Rubber

Natural and synthetic rubbers are primarily used as linings for chemical equipment. Some other components such as gaskets and bushes are also made of rubber. The hardness of rubber can be improved by addition of various ingredients. Natural rubber is resistant to dilute acids, alkalies and salts, but is affected by oxidizing media, oils, benzene etc. Some of the important synthetic rubbers are polychlorophrene, or neoprene, nitrile, butyl, silicone, hypalene floroelastomer or viton and urethane. They are resistant to oils and solvents. Silicone and viton have excellent high temperature resistance, while urethane rubber has good abrasion resistance, hardness and resilience. Due to its resistance to acids and alkalies, rubber lining is used for stirrers, fans, centrifuge baskets, pumps, valves, storage vessels, pickling vats, etc.

2.4.2.4 Plastics

These materials are classified in two main categories : thermoplastics, which soften under the application of heat and thermosetting plastics, which under pressure and heat are cured and become hard and will not deform when subjected to heat again. The important properties of these materials are their low weight, good thermal and electrical insulation, ease of fabrication and pleasing appearance. However, they can be used only at moderate pressures and temperatures. Their strength is low and they have only a fair resistance to solvents. They have excellent resistance to weak mineral acids and salt solutions and are not corroded by atmospheric conditions.

Thermoplastics—The most common thermoplastics are cellulose acetate, polyethylene, polystyrene, polyvinyl chloride, polypropylene, nylon. methacrylate, etc. These materials are generally available in the form of films, sheets, rods and pipes. They can be moulded and extruded to various shapes and sizes. The main applications are in reaction vessels, storage tanks, gas washing towers, ducting, ventilating systems, fans, pipes, pipe fittings, gaskets, etc.

Thermosetting plastics—These comprise phenol formaldehyde, urea formaldehyde, melamine formaldehyde, polyester, epoxy and silicone resins. In general these materials are comparatively hard and brittle and are therefore added with fillers such as wood flour, cotton flock, asbestos, etc. Polyester and epoxy resins with glass fibre reinforcements are used for fabrication of pipes and fittings, ducts, vessels and tanks. They are also used as lining materials. Thermosetting plastics are moulded to give them specific shapes. Laminates can also be formed.

Mechanical properties of plastics—In respect of the mechanical properties, plastics differ considerably from metals. For instance, when a stress-strain diagram is made for a thermoplastic, it is found that the thermoplastic does not follow Hooke'slaw of elasticity. The modulus is not constant but varies with stress. The best measure of the stiffness in this case is the average stiffness at a given stress, which is given by the secant modulus (Fig. 2.4). Also the results of the tensile strength test are significantly affected by the rate of loading and the

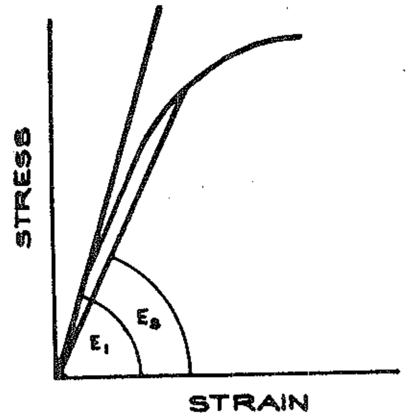


Fig. 2.4. Tangent and secant modulii E_1—Tangent modulus
E_s—Secant modulus

temperature and the elongation at rupture often varies considerably. Furthermore, at all practical temperatures a thermoplastic will continue to deform when a constant load is applied for some time, in other words thermoplastics are invariably subject to creep. (See Appendix C).

The basic creep data are obtained by determining the elongation as a function of time for a series of stress levels, each for a series of temperatures. These data are usually presented in the form of graphs (Fig. 2.5).

2.5 Protective Coatings

Quite often limitations are imposed by cost and fabrication requirements, on the choice of the most desirable material for resisting corrosion. Therefore, a protective coating on a less resistant metal or alloy represents a practical compromise. There are three general categories of coatings, metallic, inorganic and organic. A coating should be applied only on a clean surface. The presence of grease, oil dirt or scale adversely affects the adherence, continuity and durability of coatings.

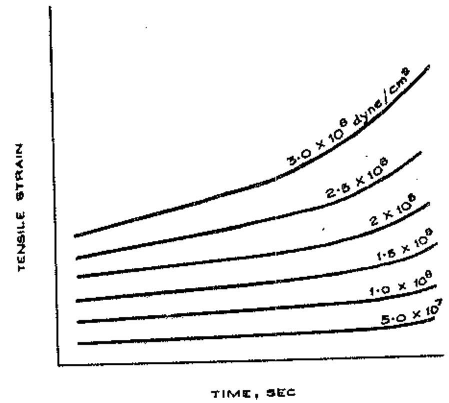


Fig. 2.5. Creep curve at different stress levels.

2.5.1 Metallic Coatings

Many different techniques are practised commercially for applying metallic coatings. The most common methods are hot-dipping, cementation, mechanical cladding, electroplating, metal spraying and condensation of metal vapours. Not all metals may be applied by all the methods, but usually two or more processes are practical and economically feasible for any metal. Table 2.5 indicates the principal methods, their characteristics and the metals used.

2.5.2 Inorganic Coatings

Commercial inorganic coatings for protection of metals fall into two general categories, namely chemical or electrochemical surface conversion treatments and vitreous enamels. Those in the first group often do not constitute the final protective layer but serve as undercoating for paint or other organic materials. Chemical dip methods are employed to create protective oxide films on iron, steel, stainless steel, aluminium, copper and some of their alloys. Such films are usually very thin and are frequently coloured. Electrolytic coatings may be made, as

Table 2.5

Method	Coating metal requirements	Coating characteristics	Metal used
Hot Dipping Short-time immersion into bath of coating metal	Low melting point alloying capability	Alloy layers Adherent, moderately uniform thickness difficult to control.	Al, Zn, Sn, Pb alloys
Cementation Powdered coating metal alloyed with base metal at temperature below melting point.	Alloying capability	Alloy throughout; hard, brittle porous; poor corrosion resistance.	Al, Zn, Cr, W, Si, Mo
Cladding Veneering and alloying of two or more metals under pressure; rolling of duplex ingots.	Rolling characteristics like base metal. alloying capability.	Pure metal surface; Impervious; controlled thickness.	Cu, Ni, stainless steel, noble metals
Electroplating Electrodeposition of coating metal at cathode from solution on fused salt bath.	Waterable deposition potential; Adhesion to base metal	Pure, uniform, impervious, No alloying unstressed, controlled thickness, hardness.	Ag, Au, Cd, Co, Cr, Ni, Pb, Sn, Zn, Cu alloys
Metal spraying Atomization of molten metal in hot gas stream. Wire or powder may be melted in gun by flame or arc.	Non refractory. Not easily oxidized.	Porous laminate structure; Harder; No alloying with surface.	Al, Cu, Fe, Pb, Sn, Zn, alloys
Vapour Plating Surface condensation of metal vapours by pyrolysis of metal compounds, cathode sputtering, vacuum evaporation.	Vapourisability	Costly, crystalline, fine grain, very thin; Readily applicable to non-metals.	Mo, W, Cr, Al, Au, Ag, Ni

in the case of anodising treatments for aluminium which produce a relatively thick, abrasion resistant coating.

Vitreous coatings although brittle, possess surface hardness and complete inertness to many corrosive environments. Enamelled or glass lined vessels and other equipment are available in a variety of shapes and sizes. These enamels are made from fused silicates of various compositions, containing colloidal suspensions of colouring materials. For metallic sheets, the enamel is applied as slurry. Coatings are enamelled by powder. Subsequent heating conditions are dependent on enamel composition and intended service.

2.5.3 Organic Coatings

These represent a large variety of materials and are the most widely used methods of protecting metals against corrosion. More than a thousand different synthetic resins as well as a wide variety of pigments, modifying oils and solvents are used in coating formulations. These coatings protect metals by interposing a continuous, adherent, inert film between the metal and its environment. They also markedly change appearance of the metal. These coatings can be divided into three classes. Paint is a dispersion of pigments in a vehicle which consists of drying oils modified with solvent or thinner to aid application. Enamel is a dispersion of pigments in varnish or resin vehicle which polymerises either by oxidation at room temperature or by application of heat. Lacquer is a pigmented natural or synthetic resin dissolved or suspended in solvents.

2.6 Linings for Chemical Plants and Equipment

Storage tanks, reaction vessels, pipes, ducting, etc. are covered with linings in order to (a) give the underlined structure, protection against chemical attack (b) prevent contamination of the materials being processed (c) minimise the effect of abrasion. The various methods commonly used for lining are as follows :

2.6.1 Rubber

This material is applied in the form of latex, lightly vulcanised rubber or fully vulcanised soft and hard rubber. The surface to be lined is suitably prepared and treated, and adhesives or bonding agents are applied to the cleaned surface. Vulcanisation is carried out, using steam, hot water or hot air.

Latex-cement based rubber linings do not require the application of a specific bonding cement and are normally applied directly to the prepared surfaces. Lightly vulcanised rubber sheets have been applied with great success as a lining material for general industrial purposes, and for chemical plants. However the most effective and the most commonly employed method of protecting corrodible surfaces with soft or hard rubber is to apply it over a bonding cement before vulcanisation.

After being rubber-lined, the equipment is usually allowed to remain for a day or two before being vulcanised. Any air entrapped between rubber and metal will then be diffused. Vulcanisation changes rubber from a plastic state to an elastic state. Tensile strength, resistance to abrasion, tearing and flexing increase due to vulcanisation.

2.6.2 Lead

Lead linings are principally used in two forms. In loose lead sheet lining, the lead sheet is applied to surfaces in thicknesses from 3 mm to 10 mm or more and the joints are lead-burnt in position. Proper anchoring of lead sheets is essential. Where process conditions are more rigorous a better form of homogeneous lead lining is prescribed. In this form lead is bonded to the cleaned surfaces with the aid of a flux and the thickness of lead lining required is thereafter deposited in drop-by-drop lead burning and peened after each run, which finally forms a perfectly fused homobonded mass. Homogeneous lead lining should have a minimum thickness of 4.5 mm upto a maximum of 8 mm. This type of lining is expensive and is therefore used only in those cases where temperatures above 150°C are obtained, where good heat

transfer properties are required, or where the operation is under vacuum.

2.6.3 Glass

The production of glass lined equipment requires suitable basic materials. In addition to chemical composition, surface quality and absence of internal flaws contribute a lot to the process of lining. There are three methods for lining glass on iron and steel, viz., Wet spray process, Hot dust process and Spray dust process.

(*a*) *Wet spray process*—After cleaning the metal surface the ground coat enamel is applied in slip form by spraying. The coat is dried and the equipment is then fired in the enamelling furnace after which it is allowed to cool. The process is repeated with subsequent coats till the desired thickness is obtained.

(*b*) *Hot dust process*—This process is generally applicable to cast iron components. After cleaning the metal surface the ground coat is applied in a manner similar to that in the wet spray process. When firing of the compound is completed, the equipment is removed from the furnace and a dry powder cover coat enamel is dusted over the surface while it is still at red heat. Several separate dustings of the cover coat may be applied.

(*c*) *Spray dust process*—This process is a combination of the two processes described above. A wet coat of enamel slip is sprayed on, while the equipment is cold and powdered enamel fit is dusted on this coat while it is still wet. In baking operations the lined equipment is air-dried and baked in a furnace at a temperature of 800-900°C. The firing time and temperature are suitably controlled depending on the size of equipment and the thickness of lining.

2.6.4. Plastics

These linings are made of different plastic materials such as polyethylene, polyvinyl chloride, fibre glass reinforced polyester (FRP) or epoxy. The process involves cleaning of the metal surface, application of bonding agent, and lining with requisite sheet. The joints between sheets may be either welded or filled up with resin. Apart from the lining of sheets there are other methods of depositing films of thermoplastic materials on metal surfaces. Three principal methods have been developed. These are described below: (a) the object to be coated is heated above the melting point of the thermoplastic and is immersed in a fluidized bed of finely divided polymer; (b) thermoplastic powder is sprayed under pressure, through a hot air or flame gun where the powder particles are softened and on impact with the surface to be coated, fuse together; (c) a suspension of the polymer in water or in a volatile organic medium is first deposited on the metal object which after evaporation of the liquid medium is fired for removal thus fusing together the deposited particles. In all cases the preparation of the surface plays an important part in the subsequent adhesion of the film to the metal.

Reading References

Samans, C.H., *Engineering Metals and Their Alloys*, Macmillan, New York, (1963).

Hepner, I.L., *Materials of Construction for Chemical Plants*, Leonard-Hill, London, (1962).

Mantell, C.L., *Engineering Materials Handbook*, McGraw-Hill, New York, (1958).

Richards, G.W., *Engineering Materials Science*, Prentice-Hall, New Delhi, (1963).

Cadwick, R., *Trends in Aluminium Fabrication*, Chem. and Process Eng. *48* (2), 55 (1967).

Cremer, H.W., *Chemical Engineering Practice*-Vol. IX, Butterworths, London, (1965).

Jelinek, R.V., *Design Factors in Corrosion Control*, Chem. Eng., *65* (23), 151 (1958).

Jelinek, R.V., *Protective Coatings Limit Corrosion*, Chem. Eng. *65* (21), 163 (1958).

Landrum, R.J., *Design for Corrosion Resistance*, Chem. Eng. *76* (4), 118 (1969), *76* (6), 172 (1969).

McDonald, J.P., *Corrosion Resistant Materials and Their Applications*, Brit. Chem. Eng., *16* (9), 801 (1971).

Special Feature on Plant Linings, Chem. Age of India, *22* (10), 701 (1971).

The Application of Stainless Steel in Design and Fabrication for the Chemical and Process Industries, Symposium Proceedings. Chem. Age of India, *22* (11), 777 (1971).

Chemical Resistant Linings for Pipes and Vessels. Handbook No. 6; The Engineering Equipment Users Association, Handbook.

Rabaces, E., *Corrosion Guide*, second edn., Elsevier, Amsterdam, (1968).

Perry, J.H., *Chemical Engineers' Handbook*, 5th edn. McGraw-Hill, New York (1973).

Khurana, K.L., *Designing of Rubber Lined Equipments*, Chem. Age of India, *25* (1), 27 (1974).

Morton, T.R., *Fibre Glass Reinforced Plastics for Corrosion Control*, Part I, Chem. Eng. *80* (Dec. 24) 70 (1973), Part II, *81* (2), 140 (1974).

Johnson, D.B., *Caring for Glass-Lined Equipment*, Chem. Eng., *71* (12), 244 (1964).

Landils, H.H., Stout, E., *Glassed Steel Equipment. A guide to current technology*, Brit. Chem. Eng., *15* (10), 1289 (1970).

Uhlig H.H., *Corrosion and Corrosion Control*, John Wiley and Sons, New York, (1963).

Henthorne M., *Fundamentals of Corrosion*, Chem. Eng., *78* (11), 127 (1971).

Henthorne M., *Cathodic and Anodic Protection for Corrosion Control*, Chem. Eng., *78* (29), 73 (1971).

Henthorne M., *Stress Corrosion*, Chem. Eng., *78* (21), 159 (1971).

CHAPTER 3

Stress Analysis

3.1 Introduction

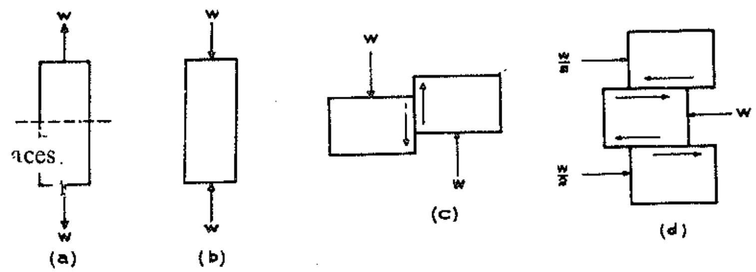
Different types of stresses are created in the different components of equipment due to application of forces. Classification and analysis of these stresses will form an essential background to any design procedure. Various theories and related applications are discussed at great length in books on 'Strength of Materials' and 'Theory of Machines'. With this knowledge, it is possible to analyse precisely the nature of forces and the resultant stresses. Besides the application of these relationships, a great deal of manipulation has to be made to satisfy the requirements in respect of appearance, function, method of production and economical considerations. Nevertheless no design problem can be satisfactorily attempted without a clear understanding of the stress pattern. An outline of some basic force systems and corresponding resultant stresses are presented in this chapter. The determination of these is based on the equilibrium conditions between external forces and internal stresses.

3.2 Stresses due to Static Loads

3.2.1 Direct Stresses

Direct loading creates tensile, compressive and shear stresses in the material. These stresses are obtained by dividing the load by the relevant area of cross section (Fig. 3.1).

For tension and compression $f = \dfrac{W}{A}$ (3.1)



(*a*) tension (*b*) compression (*c*) single shear (*d*) double shear

Fig. 3.1. Direct stresses

For shear in single plane $f_s = \dfrac{W}{A}$ (3.2)

For shear in two planes $f_s = \dfrac{W}{2A}$ (3.3)

where A—area of cross-section
f_s—shear stress
W—applied load
f—tensile or compressive stress

3.2.2 Bearing Stress or Stress in Crushing

A special case of compressive stress, is the stress caused at the surface of contact of two pieces that are relatively at rest. The load causing this type of stress does not cause a shortening of the material as in the case of a normal compressive stress. The material fails due to crushing of the surface area

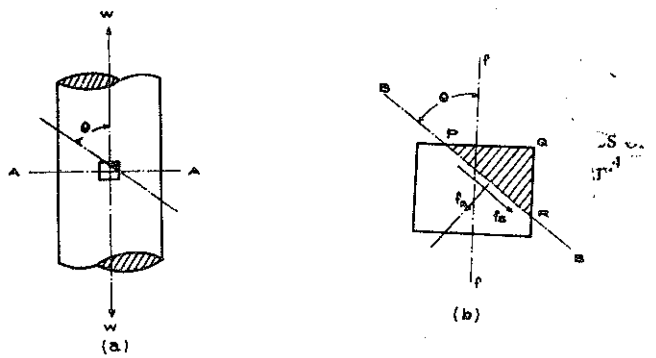
$$f_b = \frac{\text{force in crushing}}{\text{area of contact}} \quad (3.4)$$

3.2.3 Indirect Shear Stress

Fig. 3.2 (*a*) shows a rod subjected to a load W, which will produce a maximum tensile stress f across section AA. If a small cube [Fig. 3.2(*b*)] of the rod situated on AA is cut by a plane BB inclined at an angle θ to the axis of stress, then on the face PR the components of stress f, are f_n normal to PR, and f_s tangential to PR. Then

$$f_n = f \sin^2\theta \quad (3.5)$$

$$\text{and } f_s = \tfrac{1}{2} f \sin 2\theta \quad (3.6)$$



Figs. 3.2 (a) and (b) Indirect shear stress

f_n varies from zero when $\theta=0$ to a maximum value when $\theta=90°$ and f_s varies from zero when $\theta=0°$ to a maximum when $\theta=45°$, falling again to zero when $\theta=90°$; when $\theta=45°$ $f_s=\frac{1}{2}f$

Thus, although there is a direct load on the rod, it gives rise to a shear stress on planes inclined at 45° to the axis of the rod. In a similar manner a compressive load will also give rise to a shear stress.

3.2.4 Complementary Shear Stresses and Associated Normal Stresses

Fig. 3.3 shows an elemental prism subjected to shear stresses

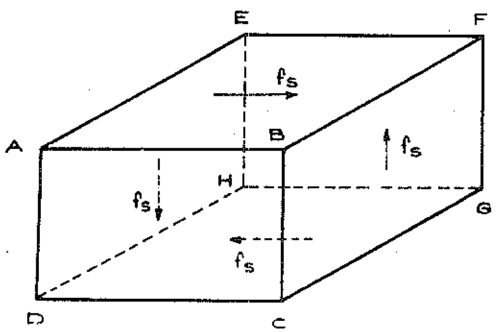


Fig. 3.3. Complementary shear stresses

f_s on the faces $ABFE$ and $DCGH$. For the prism to be in equilibrium, a clockwise couple set up by these horizontal forces must be balanced by an equal couple of opposite sense, which will prevent the rotation of the prism. The opposing couple is produced by the forces due to shear stresses on the faces $BCGF$ and $ADHE$. The shear stresses created on the four faces are complementary and have the same magnitude. Thus a shear stress can never exist singly, but is always accompanied by shear stresses on three other planes. Fig. 3.4 shows the face $ABCD$ of the prism, with the four shear stresses acting. If the shear stresses on the faces AB and BC are resolved

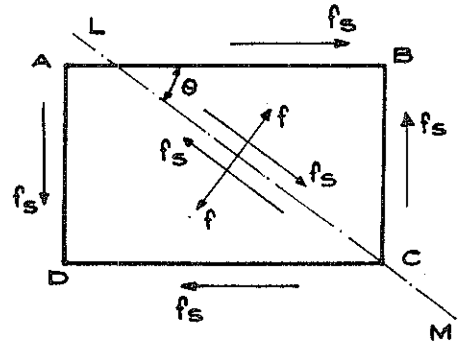


Fig. 3.4. Shear stresses and associated normal stresses.

normally and tangentially to plane LM, the stresses on this plane would be normal stress f_n and shear stress f_s.

$$f_n=f_s \sin 2\theta \qquad (3.7)$$

$$\text{when } \theta=45° \ f_n=f_s,$$

which is of maximum intensity and is tensile; When $\theta=135°$ $f_n=-f_s$, indicating that the normal stress is compressive.

$$f_{s_1}=f_s \cos 2\theta \text{ when } \theta=45°, \ f_{s_1}=0 \qquad (3.8)$$

$$\text{when } \theta=135°, f_{s_1}=0$$

Thus on planes 45° to the planes on which the shear stress acts, these are maximum tensile and compressive stresses equal in intensity to the shear stress.

3.3 Strains

When loads are applied to bodies they give rise to changes in shape. Strain is a measure of this change. Strain may be a change in length, angular change or change in volume. It is defined as a ratio of change in dimension to the original dimension. A load or force may give rise to strains in different directions. Such strains are related to each other. A force acting in longitudinal direction, would create a strain in the same direction as also in the transverse direction. The ratio of the strain in the transverse direction to that in the longitudinal direction is known as Poissons' Ratio (μ). Strains are tensile (e_t), compressive (e_c) or shear (e_s).

3.4. Elastic Constants

Within elastic limit, according to Hooke's law, stress is proportional to strain. The constant of proportionality is known as modulus of elasticity or elastic constant. When the stresses and strains are tensile or compressive, the modulus is known as modulus of direct elasticity (E). When these are in shear, it is known as modulus of rigidity (G) and when the strain is in volume, it is known as bulk modulus (k).

3.5 Thermal Stresses

A body when heated expands. If the expansion is prevented, a stress is set up in the body. In the case of a bar the free linear expansion is given by

$$\triangle l = L\,\alpha\,t \tag{3.9}$$

where $\triangle l$—increase in length of bar

L—length of bar

α—coefficient of linear expansion

t—increase in temperature.

If the expansion is prevented, the stress in the bar is given by

$$f = E\,\alpha\,t \tag{3.10}$$

where E—modulus of elasticity.

3.6 Stresses Caused by Bending

Bending is caused by forces acting normal to the axis of a beam. In such cases the distribution of the stress on the cross-section of the beam is not uniform. The maximum stresses are f_1 and f_2 at the outer surfaces of the beam at the maximum distances

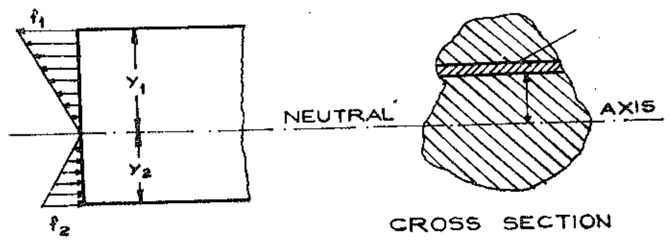


Fig. 3.5. Stresses due to bending.

y_1 and y_2 respectively from the neutral axis. As shown in Fig. 3.5, the stress goes on decreasing from the outer surface to the neutral axis. A relation between the moment of resistance and the stress can be written as

$$\frac{M}{I} = \frac{f}{y} \text{ or } M = \frac{fI}{y} = fZ \tag{3.11}$$

where

M—moment of resistance at the cross-section being considered (Appendix B)

I—second moment of area of cross-section about the neutral axis (Appendix A)

f—tensile or compressive bending stress at a distance y from the neutral axis.

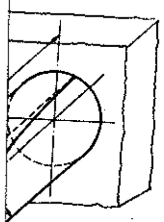
y—distance from the neutral axis to the point where the strees f is to be determined.

$Z - \dfrac{I}{y}$ is known as modulus of section

3.7 Deflection

When a beam is loaded, it is acted upon by a bending moment. The beam is stressed under this bending moment the effect of these actions ction can be defined as a n many problems it is neces- flection to a permissible limit. imating deflections in beams. with loads and the deflections B.

e, experience torsional shear over the cross-section. In hich the torque is applied is shaft. The relations between here are therefore limited to

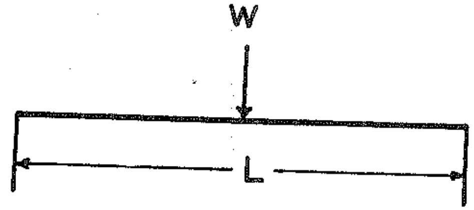
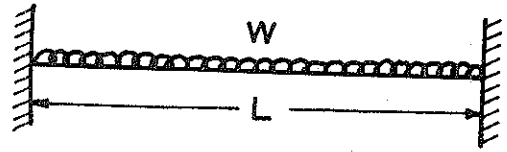
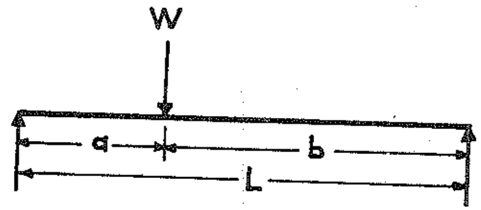
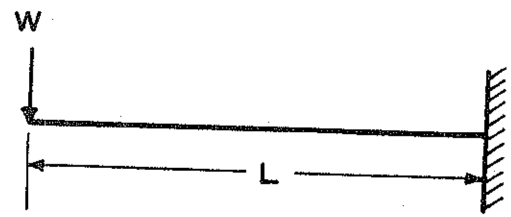
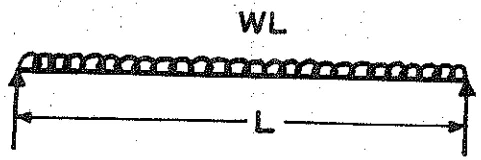
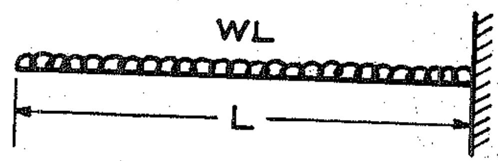
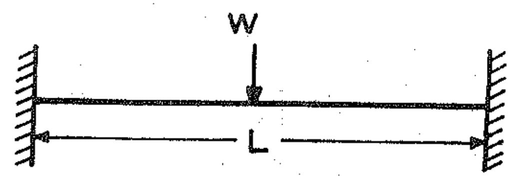


th

of shaft

th l, fixed at one end, to ue, which may be imposed applied at a distance R, the will be twisted and a shear material. The relation is as

Table 3.1

 Simply supported concentrated load at centre δ_{max} (at point of load) $= \frac{WL^3}{48\ EI}$	 Fixed at both ends—uniformly distributed load δ_{max} (at centre) $= \frac{WL^4}{384\ EI}$
 Simply supported—concentrated load at any point δ (at point of load) $= \frac{Wa^2b^2}{3EIL}$	 Cantilever—concentrated load at free end δ_{max} (at free end) $= \frac{WL^3}{3EI}$
 Simply supported—uniformly distributed load δ_{max} (at the centre) $= \frac{5WL^4}{384\ EI}$	 Cantilever—uniformly distributed load δ_{max} (at free end) $= \frac{WL^4}{8\ EI}$
 Fixed at both ends—concentrated load at centre δ_{max} (at the centre) $= \frac{WL^2}{192\ EI}$	

as shown above. In addition, the effect of these actions is to deflect the beam. Deflection can be defined as a displacement in any direction. In many problems it is necessary to restrict the maximum deflection to a permissible limit. There are many methods of estimating deflections in beams. Some of the cases of the beams with loads and the deflections caused are indicated in Appendix B.

3.8 Stresses Caused by Torsion

Components subjected to torque, experience torsional shear stresses, which act tangentially over the cross-section. In many cases the component to which the torque is applied is circular in cross section, such as a shaft. The relations between torque and shear stress derived here are therefore limited to such cross-sections.

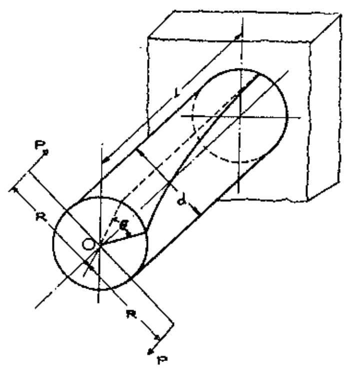


Fig. 3.6. Torsion of shaft

Fig. 3.6 shows a shaft of length l, fixed at one end, to illustrate the resistance to any torque, which may be imposed at the free end. If the force P is applied at a distance R, the torque is given by PR. The shaft will be twisted and a shear stress will be created in the shaft material. The relation is as

follows:
$$\frac{T}{I_p}=\frac{f_s}{r}=\frac{G\theta}{l} \tag{3.12}$$

$$T=\frac{f_s I_p}{r}=f_s\, Z_p \tag{3.13}$$

where T—torque applied

I_p—polar second moment of area

f_s—shear stress at any radius r

r—radius of the shaft at which the shear stress is to be determined.

d—diameter of the shaft

G—modulus of rigidity

θ—angle of twist over the length l

l—length of shaft under torsion

Z_p—polar modulus of section

$=\dfrac{\pi d^3}{16}$ for a circular section.

The shear stress is zero at the centre of the shaft and increases towards the outer diameter. The maximum shear stress is at the surface of the shaft.

3.9 Stresses in Struts

Components which are under longitudinal compression are known as struts. The load under which a strut fails is known as crippling or buckling load. Its magnitude depends on various factors, viz., the end conditions, the material of strut and slenderness ratio. Failure results partly from direct compression of the strut and partly from bending. Theoretically bending should not occur in a straight strut subjected to a load acting along its longitudinal axis. In practice no strut is ideally straight. The longer the strut the greater the tendency for failure to occur as a result of bending alone.

3.9.1 SHORT STRUTS

The failure in these struts will result from direct compression alone.

$$P=f_c A \tag{3.14}$$

where P—external load

f_c—compressive stress
A—area of cross-section normal to load.

In such struts the slenderness ratio is about 10. (In steel this value may be upto 40.)

Slenderness ratio $\frac{l_e}{k} < 40$

where l_e—effective length of strut
k—least radius of gyration of the cross-section.

3.9.2 Long Struts

In these struts failure occurs by bending of the strut i.e. by buckling.

In accordance with Euler's theory

$$P = a.\frac{\pi^2 EI}{l_e^2} \tag{3.15}$$

where a—a constant depending on the conditions of the restrain of column
P—external load
E—modulus of elasticity
I—second moment of area of cross-section

3.9.3 Intermediate Struts

In many practical cases struts have slenderness ratios (for steel these may be between 40 to 120) whose values lie below the minimum values at which Euler's formula is valid and yet are too high to accept simple compression. For such cases the following formulae may be used.

(*a*) Rankine-Gordon formula

$$P = \frac{f_c A}{l + a\left(\frac{l_e}{k}\right)^2} \tag{3.16}$$

where a—a constant
A—area of cross-section
k—radius of gyration
f_c—compressive stress
P—external load
l_e—effective length of strut.

(*b*) Johnson's parabolic formula

$$P = f_c A\left(1 - b\left(\frac{l_e}{k}\right)^2\right) \tag{3.17}$$

where b—constant depending on the conditions of the end.

(*c*) Straight line formula

$$P = f_c A\left(1 - n\left(\frac{l_e}{k}\right)\right) \tag{3.18}$$

where n—constant depending the conditions of the end.

3.10 Stresses in Flat Plates

3.10.1 Solid Circular Plate

(*a*) UNIFORMLY LOADED, EDGE FREELY SUPPORTED.

Two stresses both in the plane of the plate and perpendicular to each other are created due to the uniformly distributed load.

$$f_z = \frac{3\,py}{4mt^3}\left\{ r^2\,(3m+1) - x^2\,(m+3) \right\} \tag{3.19}$$

$$f_x = \frac{3\,py}{4mt^3}\left\{ (3m+1)\,(R^2 - x^2) \right\} \tag{3.20}$$

where
p—uniformly distributed load

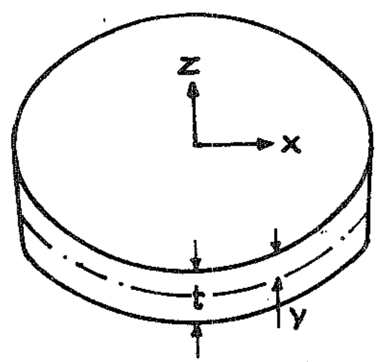


Fig. 3.7. Solid circular plate

y—distance of the plane from the neutral plane

μ—Poisson's ratio $= \dfrac{1}{m}$

R—radius of the plate

x—distance along the radius for which the stresses are determined

t—thickness of plate.

f_z and f_x are maximum, when y is maximum and x is minimum. At $x=0$ and at $y=\pm\dfrac{t}{2}$

(that is, at the surface)

$$f_z=f_x=\frac{3p\,R^2}{8\,t^2}\left(\frac{3\mu+1}{\mu}\right) \tag{3.21}$$

(*b*) UNIFORMLY LOADED AND FIXED AT THE EDGES

Similar to the first case, two types of stresses are created in this plate. The stresses at the centre of plate are,

$$f_z=f_x=\frac{3}{8}\,\frac{pR^2}{t^2}\left(\frac{\mu+1}{\mu}\right) \tag{3.22}$$

and the stresses at the circumference are

$$f_z=\frac{3}{4}\,\frac{pR^2}{t^2}\times\frac{1}{\mu} \tag{3.23}$$

$$f_x=\frac{3}{4}\,\frac{pR^2}{t^2} \tag{3.24}$$

(*c*) PERFORATED PLATE

The maximum stress in a perforated plate is given by

$$f_{max}=\frac{\text{maximum stress in a solid plate}}{\text{ligament efficiency}} \tag{3.25}$$

The ligament efficiency depends on the arrangement of holes. For example when the pitch (p) of the hole (d) on every row is equal, the

$$\text{ligament efficiency } \eta=\frac{p-d}{p} \tag{3.26}$$

3.10.2 Rectangular Plate

UNIFORMLY LOADED, AND SUPPORTED AT ITS PERIMETER

$$f_{max}=\frac{pb^2}{2t^2}\times\left(\frac{1}{1+\dfrac{b^2}{a^2}}\right) \tag{3.27}$$

where p—uniformly distributed load.

a, b, t—dimension of the plate as shown in Fig. 3.8

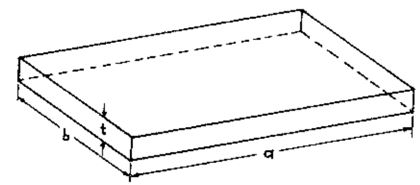


Fig. 3.8. Rectangular plate

3.11 Stresses in Cylinders and Spheres

3.11.1 Thin Cylinder under Internal Pressure

Stresses in a thin cylinder due to internal pressure are produced in three directions. These are the circumferential or hoop stress, the longitudinal stress and the radial stress (Fig. 3.9). If the ratio of thickness to internal diameter is less than about 1/20, it may be assumed with reasonable accuracy that the hoop and longitudinal stresses are constant over the thickness of the cylinder and that the radial stress is small and can be neglected (in fact it must have a value equal to the internal pressure at the inside surface and zero at the outside surface). These stresses are given by circumferential or hoop stress

$$f_p=\frac{pD}{2t} \tag{3.28}$$

Longitudinal stress of axial stress

$$f_a=\frac{pD}{4t} \tag{3.29}$$

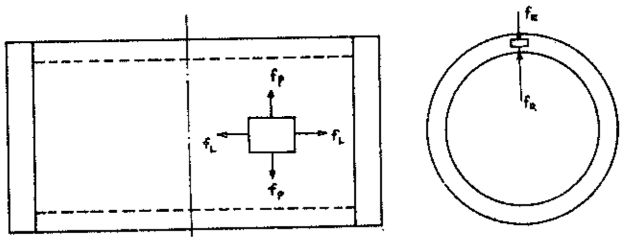


Fig. 3.9. Stresses in thin cylinder.

where p—internal pressure
D—internal diameter
t—thickness

3.11.2 Thin Spherical Shell under Internal Pressure

In this case both the stresses are equal.

$$f_p = f_a = \frac{pD}{4t} \qquad (3.30)$$

3.11.3 Thick Cylinder under Pressure

The stresses in the cylinder, due to pressure, are in three directions as in the case of thin cylinder. However, these stresses are expected to vary over the cross-section. The stress in the radial direction may be given by

$$f_R = A - \frac{B}{R^2} \qquad (3.31)$$

where A and B are constants
R—any radius

and the stress in the circumferential direction (hoop stress) is given by

$$f_p = A + \frac{B}{R^2} \qquad (3.32)$$

The stress in the longitudinal direction is constant.

$$f_a = \frac{pR_1^2}{(R_2^2 - R_1^2)} \qquad (3.33)$$

where R_1 and R_2 are the internal and external radii and p—pressure.

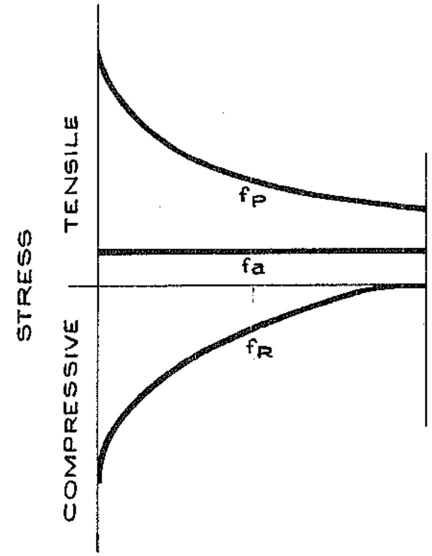


Fig. 3.10. Variation of stresses along thickness for a thick cylinder.

The variation of stresses along the thickness of the cylinder is indicated by Fig. (3.10). For details see 12.4.1.

3.11.4 Thick Spherical Shell

In this case the radial stress is given by

$$f_R = A - \frac{B}{D^3} \qquad (3.34)$$

where A, B—constants
D—diameter of shell.

Circumferential or hoop stress is given by

$$f_p = A + \frac{B}{2D^3} \qquad (3.35)$$

The radial stress f_R, at the surface of the thick cylinder or sphere is equal to the operating pressure.

4(54-24/1974)

3.12 Stress Concentration

The bar as shown in Fig. 3.11 is loaded under tension. Average stresses at the two cross-sections AA and BB can be calculated

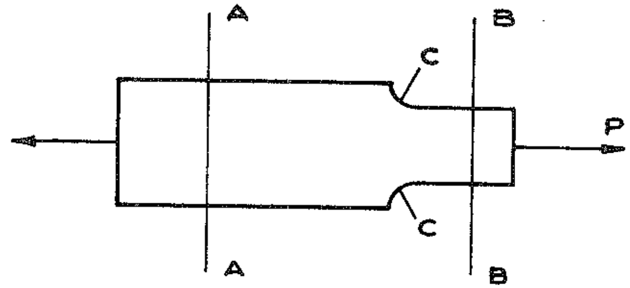


Fig. 3.11. A bar under tension.

by dividing the load p, by the appropriate cross-sectional area. However, in the region, where the width is changing, there is a redistribution of the stress. In this portion the load is no longer uniform at all points on the cross-section, but the material near the edges is stressed considerably higher than the average value. The maximum stress occurs at some point on the fillet, as at C. Similarly in the case of a bar with a hole there is high stress in the neighbourhood of the hole. High stresses such as the one at the fillet or at the hole cause a fatigue crack to start under fluctuating load.

This irregularity in the stress distribution caused by abrupt changes of form is called stress concentration. It occurs for all kinds of stresses, in the presence of fillets, holes, notches, keyways, splines, tool marks, etc. The maximum value of the stress at such points is given by a stress concentration factor k which is defined as

$$k = \frac{\text{highest value of the stress at fillet notch, hole, etc.}}{\text{nominal stress given by elementary equation for minimum cross-section.}}$$

3.13 Dynamic Stresses

Stresses produced by variable loads changing in magnitude and direction are termed as dynamic stresses. These may be divided into two groups: (1) those produced by external forces, (2) those resulting from the inertia of the mass of the component.

The variable applied load may set up (*a*) alternating stress, i.e., stress which varies between maxima of different signs (*b*) repetitive stress i.e., stress varying from zero to a maximum or (*c*) fluctuating stress in which the stress varies from a minimum to a maximum but retains the same sign. Such cyclic stresses are likely to cause a fatigue failure. If fatigue failure is to be prevented, the stress should be limited to a certain value called the fatigue or endurance limit. The material will last indefinitely if the stress is below the fatigue limit. However for slightly greater values of the stress, failure can be expected after a certain number of repetitions of the stress cycle. The fatigue or endurance limit might vary from 30% to 60% that of the ultimate strength, depending on the material.

In cases where abrupt changes in shape of the material occur a stress concentration is caused. Due to repeated or fatigue loading a breakage of the material takes place at this portion.

3.14 Stresses in Rotating Rims and Disks

3.14.1 Rotating Rim

A ring or a rim when rotating about its centre of gravity induces a hoop or circumferential stress, across the section, which is given by

$$f_y = \frac{\rho}{g}\omega^2 r^2 \qquad (3.36)$$

where ρ = density of the material of the rim

ω = angular velocity of the rim

r = radius at which the stress is determined.

The above equation is based on the centrifugal force produced due to rotation of the ring mass.

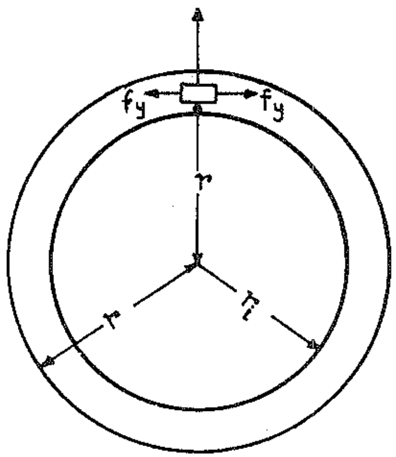


Fig. 3.12. Stresses in a rotating rim.

3.14.2 Rotating Disk

The stresses created due to the rotation of a flat circular disk are both circumferential and radial as shown in Fig. 3.13,

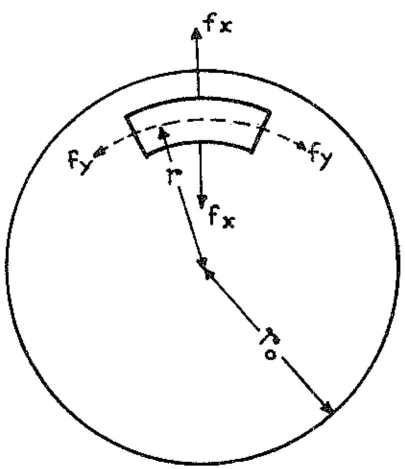


Fig. 3.13. Stresses in a rotating disk.

which are given by

$$f_y = \frac{\rho\omega^2}{8g\mu}\{(3\mu+1)\,r_0^2-(\mu+3)r^2\} \tag{3.37}$$

$$f_x = \frac{\rho\omega^2}{8g\mu}\{3\mu+1)\,(r_0^2-r^2)\} \tag{3.38}$$

where μ—Poisson's ratio
r_0—maximum radius
r—radius at which the stress is determined
ρ—density of the material
ω—angular velocity of the disk.

The maximum values of f_y and f_x will be when $r=0$; i.e., at the centre of the disk, and these will be equal in magnitude. The stress is due to centrifugal force of the disk mass.

3.15 Impact Stresses

If a moving body strikes another body the second body is subjected to an impact, which is equal to the kinetic energy of the moving body. The impact energy of a body falling from a height h is given by $W \times h$ where W is the weight of the body. If the weight strikes a rod, the work done is $W(h+e)$ where e is the compression of the rod. The stress in the rod is given by

$$f = \frac{W}{A}\left(1+\frac{2hEA}{WL}\right) \tag{3.39}$$

where A—area of cross-section of the rod
E—modulus of elasticity
L—length of the rod

If the weight W is applied suddenly, it does not have an appreciable velocity before it strikes the body, then h is zero and the stress produced is double that produced by the same weight applied statically.

3.16 Compound Stresses

Several cases, where forces acting in different directions, and thus giving rise to tensile, compressive and shear stresses have been considered so far. In a number of cases these forces induce stresses of different natures simultaneously. The stress that results from the combined action of several stresses acting simultaneously is called the resultant stress. It is possible to resolve this stress acting at a point in the material, in three mutually perpendicular planes, called principal planes across each of which only one single normal stress, either tensile or compressive will act. These normal stresses are known as principal stresses. Of the three principal stresses one or two may have zero value, giving two-dimensional or one-dimensional forms of stresses respectively. In many important problems, the stresses are two-dimensional, in which two principal stresses acting normally to two principal planes lie at right angle to each other. One of these normal stresses will usually be greater than the other and will be the maximum principal stress set up in the material. On planes at 45° to the principal planes, will occur maximum shear stresses which may or may not be accompanied by normal stresses.

If at any point in the material with a two-dimensional stress pattern [Fig. 3.14(a)] two perpendicular normal stresses f_x and f_y and a shear stress f_s are acting, then the principal stresses are given by

$$f_{RN}=\tfrac{1}{2}\,(f_x+f_y) \pm \tfrac{1}{2}\,[(f_x-f_y)^2+4f_s^2)] \qquad (3.40)$$

the +ve sign gives one principal stress and the −ve sign gives the second principal stress. In the above equation f_x and f_y are assumed to be tensile and are taken as positive. If either or both are compressive there will be a change of sign.

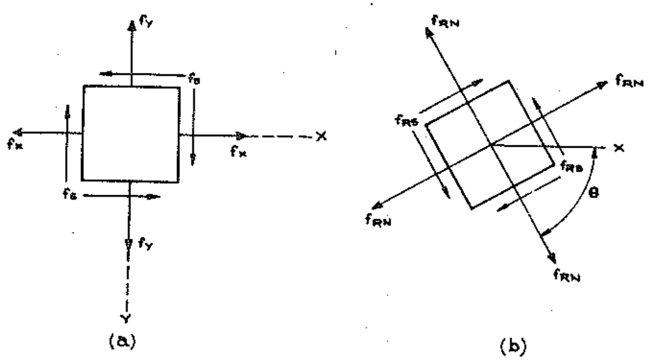


Fig. 3.14 (a) Normal and shear stress (b) Resultant normal and shear stresses.

The maximum shear stress is given by

$$f_{Rs}=\tfrac{1}{2}\{(f_x-f_y)^2+4f_s^2\} \qquad (3.41)$$

Fig. 3.14 (b) indicates the normal stress f_{RN} and maximum shear stresses f_{Rs}.

Reading References

Barve, R.S., *Strength of Materials*, B.I., Bombay, (1970).

Ryder, G.H., *Strength of Materials*, Macmillan Co., London, (1973).

Marin, J., *Strength of Materials*, Macmillan Co., London (1948).

Timoshenko, S., *Strength of Materials*, Vol. I, Vol. II, McGraw-Hill, New York, (1951).

Huckert, J., *Engineering Tables* (ASME Handbook) McGraw-Hill, New York, (1956).

Brooks W.H., *Strength and Elasticity of Materials*, Vol. I to Vol. IV Asia Publishing House, Bombay, (1963).

Mahadevan K. et al, *Design Data Handbook*, Rangs, Bombay, (1972).

CHAPTER 4

Design Considerations

4.1 Introduction

A large number of factors, which influence the design of components can now be considered in sufficient detail. It is necessary to consider the effect and influence of each factor individually, and also in combination with other factors so that the final design will be based on optimum considerations. In Chapter 1, general principles involved in designing of machine parts and equipment have been discussed and certain guidelines have been indicated. For a satisfactory design of any component, various factors which must be considered are

(1) materials of construction and corrosion
(2) stresses created due to static and dynamic loads
(3) elastic instability
(4) combined stresses and theories of failure
(5) fatigue
(6) brittle fact fracture.
(7) creep
(8) temperature effects
(9) radiation effects
(10) effects of fabrication methods
(11) economic considerations.

4.2 Materials of Construction and Corrosion

Different materials of construction and their properties have been discussed in Chapter 2. Different modes of failure due to corrosion and methods used in protective coatings have also been indicated in the same chapter. In choosing a suitable material of

construction, it is necessary to consider the relative merits of the various materials available. Mechanical and anti-corrosive properties, effective application of protective coatings, ease of fabrication and economic considerations are some of the important factors, which influence the final choice.

Corrosion of materials should be avoided wherever possible. Complete corrosion resistance is most desirable but it may be too expensive. As a general guideline, metals, with corrosion rates upto 0.125 mm per year are acceptable. In some cases rate of 1.25 mm per year may be acceptable but only if the alternative corrosion-resistant material is a very expensive material like platinum. Protective coatings must be considered as an important alternative to expensive materials. Quite often it may be possible to prevent or reduce corrosion by certain modifications in design. Similarly poor workmanship in fabrication processes such as welding, machining, assembly and heat treatment, which cause corrosion failures can be avoided by adhering to strict fabrication standards and carrying out frequent inspection during construction. The designer should observe a few simple rules based on corrosion principles and sound engineering practice.

(*a*) *Avoid galvanic couples*—A galvanic couple is formed between dissimilar metals (for example, use of sheet aluminium over steel structures, or brass fittings in steel piping) and causes severe corrosive attack. In such cases, electrical insulation should be provided at contact points by using non-metallic washers, gaskets and nipples. The designer should try to select materials which are close together in the galvanic series. Exposed area of the less noble metal should be kept large relative to more noble metal.

(*b*) *Avoid opportunities for concentration cells*—Many jobs normally required in all equipment and structures provide ready opportunities for concentration cells and crevice corrosion. The designer should avoid crevices by using welded joints in preference to bolted or riveted joints wherever practical. Unavoidable crevices as shown in Fig. 4.1 may sometimes be caulked or sealed with an organic compound. Whenever possible horizontal surfaces exposed to the atmosphere should be avoided, since these tend to hold moisture and dirt.

For vessels, dish-shaped heads are preferable to flat heads. Tanks and vessels must be provided for complete drainage of

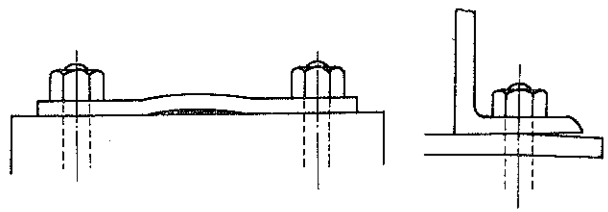


Fig. 4.1. Crevices

liquid (Fig. 4.2). Pockets in which stagnant liquids can accumulate should be avoided. Connecting nozzles must not project

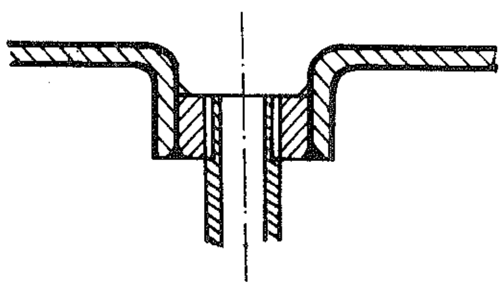


Fig. 4.2. Drainage from a tank

into vessels. Crevice corrosion can be particularly troublesome in heat transfer equipment. As such, all welded heat exchangers are preferable.

(*c*) *Avoid localized stresses*—The designer should take every precaution to avoid uneven stress distribution in equipment exposed to corrosive conditions where these are likely to arise in welding, machining or other fabrication processes. Stress relief through heat treatment should be specified.

(*d*) *Keep surfaces smooth and streamlined*—Rough surfaces are more susceptible to corrosion than smooth surfaces.

Similarly projections and sharp corners provide good starting points for corrosion-erosion failures. The designer should provide for streamlining of channels, and also for a generous radius at fillets and rounds. Avoid notches. Joints and welds should be well finished.

(*e*) *Specify fabrication and inspection standards*—The designer should specify inspection procedures and ensure satisfactory fabrication.

4.3 Stresses Created due to Static and Dynamic Loads

Chapter 3 deals with several cases, in which different types of stresses are created due to static and dynamic loads. The nature of these loads is constant, impact or alternating. To ensure adequacy of design it is necessary to assess the correct stress distribution in various components. These stresses result in deformations which may interfere with the functional operation of the component. Sufficient rigidity must be incorporated into the design of the part to restrict the amount of deformation to a permissible value. The most widely used criterion to restrict deformation is to maintain the induced stresses within the elastic region of the material of construction in order to avoid plastic deformation resulting from exceeding the yield point. However in special cases where plastic deformation is limited to small regions in the neighbourhood of structural discontinuities or stress raisers, failure will not occur provided that the material remains sufficiently ductile to accommodate these deformations without rupture. In such cases it is, therefore, of utmost importance to ensure that the ductility of the original material is not lost due to fabrication process or during service. The stresses created must be limited to a permissible value that is accepted as safe for a particular material and its applications. This value of the stress is known as a design stress or permissible or allowable stress. It is controlled by a number of factors such as, the accuracy with which the loads can be estimated, the reliability of the stresses computed, the uniformity of the material, the hazard if failure occurs, and other considerations like local stress concentrations, fatigue etc.

In the case of ductile materials, where failure can be expected to occur as a result of plastic deformation, the design stress is usually obtained by dividing the yield stress of the material by a factor of safety.

Table 4.1

Kind of load	Factor of safety		
	Ductile meterial F.S. based on ultimate strength.	Ductile material F.S. based on yield strength	Brittle material F.S. based on ultimate strength
1. Dead or steady lead	3 to 4	1.5 to 2	5 to 6
2. Repeated load applied gradually but not reversed	6	3	7 to 8
3. Repeated load applied gradually and reversed	8	4	10 to 12
4. Load applied with medium or heavy shock	10 to 15	5 to 7	15 to 20

$$\text{Design stress} = \frac{\text{Yield stress}}{\text{Factor of safety}}$$

Brittle materials are characterised by remaining elastic upto fracture. Such materials fail by fracture with little or no deformation. The design stress, therefore, is to be based on the ultimate strength.

$$\text{Design stress} = \frac{\text{Ultimate stress}}{\text{Factor of safety}}$$

Some values of the factor of safety for ductile and brittle materials are given in Table 4.1.

In the case of thermoplastics, as has been pointed out under material properties, the behaviour of such materials is not elastic and the elongations are relatively large. The design stress in these cases is based on creep data, and the deformations are

to be less than 3 per cent. The design stress should be determined by taking the lesser of the following two criteria :

(1) Average stress required to produce rupture in 10^5 hours at design temperature, divided by a factor of safety (usually between 1.5 to 2).

(2) Average stress required to produce a total creep strain of 2 to 3 per cent in 10^5 hours at design temperature.

4.4 Elastic Instability

This is caused due to insufficient stiffness or rigidity in a component or structure which is subjected to compression, bending, torsion or a combination of such loading conditions. In the case of bending and torsion, stiffness is determined by the extent of deflection and the angle of twist. Design in such cases is based on limiting the stresses, as well as limiting the deflections and twists.

In cases of columns, vessels under axial load and vessels under external pressure, and design is based both on the critical buckling load and the permissible or design stress. Buckling may be (1) in the form of bending and deflection of the component as in the case of medium and long columns, (2) local buckling or wrinkling as in the case of a vessel with axial load and (3) deformation of shape or collapse as in the case of vessel with external pressure.

The critical buckling load may be obtained by multiplying the actual or the working load by a factor of safety. Similarly the permissible or the design stress is obtained from the yield stress, by dividing it by a factor of safety. For instance in a thin cylinder under axial compression, failure may take place by one of the following ways :

(a) buckling of the complete cylinder as a long column or strut

$$f_c = \frac{K_1 E}{2} \cdot \frac{\pi D^2}{4 l^2} \text{ (Euler buckling)} \qquad (4.1)$$

(b) local buckling with formation of axial or circumferential corrugations on the surface, also known as wrinkling.

$$f_c = \frac{K_2 E}{\sqrt{3(1-\mu^2)}} \cdot \frac{t}{\left(\frac{D}{2}\right)} \qquad (4.2)$$

(c) plastic yielding when the stress in the material reaches the yield stress

$$f_c = f_y \tag{4.3}$$

To avoid failure and assure safe design, a factor of safety must be used for the value of f_c in all cases. In the equations above

f_c —compressive stress
K_1—end condition factor
D —diameter of thin cylinder
l —length of cylinder
f_y —yield stress
μ —Poisson's ratio
t —thickness of cylinder
K_2—factor to account for initial imperfections of the cylinder.

4.5 Combined Stresses and Theories of Failure

Section 3.16 deals with combined action of several stresses acting simultaneously, from which it is possible to determine resultant stresses or principal stresses. For components which are subjected to combined stresses, a procedure will have to be established which will relate the resultant stress or principal stress to the yield stress obtained from a test of the material. This procedure stipulates yielding of the material and consequent failure, if the maximum value of the stress or strain created by the combination of stresses, exceeds the conditions at the yield point. Various theories for the failure of the material have been suggested and are made applicable according to the nature of the stresses and the type of material (ductile or brittle). Some of these theories are described below.

4.5.1 The Maximum Normal Stress Theory or Rankine Theory

According to this theory, failure of the component will take place if any of the resultant normal stresses reaches the value of the stress at the yield point, in simple tension. For ductile materials, the tension and compression properties are the same and the direction of the stresses (i.e. tensile or compressive) will, therefore, not affect the above condition. The condition of failure is

$$f_{RN} = f_y \tag{4.4}$$

where f_{RN} —Maximum resultant normal stress (as per equation 3.40)
f_y—yield stress in tension test

4.5.2 The Maximum Shear Stress Theory (Guest Theory)

According to this theory failure will take place, if the maximum resultant shear stress in a component, subjected to combined stresses, reaches the value of the shear stress in simple tension at the yield point. The condition of failure is

$$f_{RS} = \tfrac{1}{2} f_y \tag{4.5}$$

f_{RS} =resultant shear stress (as per equation 3.41)

4.5.3 The Maximum Strain Theory (Saint-Venant's Theory)

According to this theory failure will occur if the maximum strain created due to combined stresses, reaches the strain created in simple tension at yield point. The condition of failure for a three dimensional stress pattern is

$$\frac{f_{RN_1}}{E} - \frac{\mu}{E}\left(f_{RN_2} + f_{RN_3}\right) = \frac{f_y}{E} \tag{4.6}$$

f_{RN_1}, f_{RN_2}, f_{RN_3}, —Resultant normal stresses in three directions.

where μ—Poisson's ratio
E—Modulus of elasticity.

4.5.4 Strain Energy Theory

This theory stipulates that failure will occur when the strain energy stored, under the action of the combined stresses, reaches the strain energy at yield point in simple tension or compression i.e.,

$$\frac{1}{2E}\left(f^2_{RN_1} + f^2_{RN_2} + f^2_{RN_3}\right) - \frac{\mu}{E}\left(f_{RN_1} f_{RN_2} + f_{RN_2} f_{RN_3} + f_{RN_3} f_{RN_1}\right) = \frac{f_y}{E} \tag{4.7}$$

4.5.5 Distorsion Energy Theory or Shear Energy Theory (Hencky-Von Mises Theory)

According to this theory failure of the material, subjected to combined stresses, will take place when the elastic strain energy required for distorsion of the material reaches the energy required to produce yielding under simple tension or compression. This gives for a three dimensional stress pattern.

$$\frac{1}{\sqrt{2}}\{(f_{RN_1}-f_{RN_2})^2+f(_{RN_2}-f_{RN_3})^2+f_{RN_1}-f_{RN_3})^2\}^{\frac{1}{2}}=f_y \quad (4.8)$$

For a two dimensional stress pattern f_{RN_3} will be zero. The equation for two dimensional stress pattern can also be written in terms of the actual stresses.

$$\{(f_x)^2 + (f_y)^2-f_x f_y + 3f_s^2)\}^{\frac{1}{2}}= f_y \quad (4.9)$$

In making an appropriate choice from among the above theories it is necessary to consider the relative magnitude of the resultant normal and shear stresses in the components as also the properties of the material. When the resultant normal stresses are either all positive (tensile) or all negative (compressive) the application of maximum normal stress theory is satisfactory. In cases where some stresses are positive and some negative the maximum shear stress theory or shear energy theory should be used.

In all the above theories, the deformation and stress have to be restricted to elastic region, and hence the value of yield stress f_y, will have to be divided by a factor of safety.

4.6 Fatigue

In arriving at a safe value of the design stress one of the factors, which need consideration is the nature of any cyclic load, which induces alternating stresses or fluctuating stresses. Failure due to fatigue is due to a slow but progressive enlargement of an initial crack subjected to cyclic load. It is therefore necessary to ascertain the greatest stress or range of stress which can be applied to a material an unlimited number of

times without causing failure. This is known as fatigue limit or endurance limit. The ratio of the endurance limit for reversed stresses to ultimate static stress is known as endurance ratio. The usual values of the endurance ratio are:

steels with hardness less than 400 BHN (at 10^6 cycle). (0.5)
Cast iron and cast steel (at 10^6 cycles) (0.4)
Aluminium and magnesium (at 5×10^8 cycles) (0.3)

Various components or machine elements have usually grooves, holes, fillets and notches. Such changes in shape interrupt the lines of stress and create concentrations of stress which cause reductions in the fatigue strengths of these components. These effects have to be taken into account by using appropriate stress concentration factors. Fatigue failures can therefore be avoided by providing smooth and gradual changes in shape, and eliminating sharp corners in a groove.

Some factors which influence fatigue are (a) material and material factors (b) type of loading (c) size of the member (d) surface finish (e) stress raisers (f) surface stressing (g) corrosion (h) temperature.

4.7 Brittle Fracture

Ductile materials are likely to fail due to brittle fracture under the following conditions:

(*a*) Presence of a defect of sufficient size or notch.
(*b*) High localised stresses in the vicinity of the notch.
(*c*) Operation at sufficiently low temperature.
(*d*) Wrong selection or treatment of material.

Due to a notch in a ductile material the material surrounding the notch becomes brittle. The phenomenon is known as 'notch brittleness'. Similarly due to low temperatures the material becomes brittle. In all such cases, these materials should be treated as brittle materials and the design stress should be based on a higher factor of safety.

Brittle failure can occur at levels of nominal stress below yield point, or even at ordinary permissible levels of stress. It can occur suddenly without prior indication of any deformation.

4.8 Creep

Certain materials like thermoplastics and lead at room temperature and metals at high temperature, exhibit a slow but continuous plastic deformation which eventually results in failure. This process is termed as creep. It is a function of stress, temperature and time. With metals, creep is generally associated with high temperatures. It is essential to know, whether the rate of creep is compatible with the working life of the component. A maximum plastic deformation at the end of a certain period is selected to satisfy the working conditions and the corresponding stress may be considered as safe. Assuming a safety factor, the design stress may be found. In carbon steels upto 350°C, creep is not significant. Above 400°C temperature the design stress is reduced. In the creep range even a small increase in temperature often results in a large reduction in creep strength and consequently in the design stress.

4.9 Temperature Effects

The effect of temperature on continuous deformation in the form of creep, has already been referred to. In general with increase in temperature, there is a reduction in ultimate strength, modulus of elasticity, and hardness. Materials expand with increase in temperature. These changes must be taken into account in designing components at higher temperatures. Stresses are also created, if the expansion due to temperature is prevented, or in case the temperatures at different points in a component are not the same.

4.10 Radiation Effects

Nuclear reactor vessels, are subject to material irradiation, due to neutron bombardment from the core. Neutrons are classified as fast neutrons and thermal neutrons. Fast neutrons cause damage by dislocation or displacement of the atomic structure of the metal, whereas the effect of thermal neutrons is one of transmutation of trace impurities that can substantially

change the properties of the material. Typical changes in such cases are a marked increase in yield point, a smaller increase in tensile strength, and decrease in ductility. These factors are taken into account by a suitable factor of safety.

4.11 Effects of Fabrication Methods

Fabrication processes such as casting, hot and cold rolling, sheet forming and welding give rise to stresses or changes in properties of materials. For instance in cold forming operations, due to stretching of the material beyond its yield point the material hardens and suffers a loss of ductility and toughness. This phenomenon is called *strain* or *work hardening*. The work of stretching is not entirely transformed into heat, but part of it is retained in the form of strain energy. In joints made by welding certain amount of stress concentration takes place, due to uneven temperature rise, structure of weld metal, and welding defects such as porosity, slag inclusions and shrinkage cracks. Work hardening and residual stresses may be relieved by heat treatments like annealing and normalising.

4.12 Economic Considerations

In many components, the designer has a good deal of choice in the materials to be selected and the methods of fabrication. A preliminary estimate regarding the cost of the material chosen and cost of fabrication may be made. An estimate of cost of fabrication should be generally based on the material cost multiplied by a factor ranging from 1.5 to 10. The cost data thus obtained, although rather crude, should be helpful in making the initial selection of the materials.

The designer should set up a systematic file both for material costs and typical part costs. The cost of fabrication may be composed of costs from several operations, such as material preparation, machining or processing operations, jointing operations, heat treatment, finishing operations, etc. Each step must be analysed for cost contribution. Volume of production is an important factor in cost determination, since it will usually dictate what method of production will have to be used. Custom built equipment will be more expensive than mass produced equipment. A more accurate and reliable cost appraisal may be possible with systematic filing of costs. Accordingly material cost factors or multipliers can be evolved on the basis of lower or higher volumes of production. Other cost factors, which contribute to the total cost are inspection, packaging and transport.

Reading References

Redford, G.D., *Mechanical Engineering Design*, Macmillan Co., London, (1966).

Faupel, J.H., *Engineering Design*, John Wiley & Sons, New York, (1964).

Ramamohana Rao A., *Principles of Engineering Design Strength Considerations* (Vol. I), Indian Institute of Technology, Madras.

Henthorne M., *Good Engineering Design Minimizes Corrosion*, Chem. Eng. 78 (26), 163 (1971).

CHAPTER 5

Design of Machine Elements

5.1 Introduction

A variety of machine elements usually form part of the equipment used in the chemical industry. Such elements are shafts, keys, couplings, bearings, rivets, bolts, etc. A detailed account of the type, capacity and design of these elements is available in standard books on the subject. It is proposed to give here an outline of the design procedure adopted for each element. In the overall design of chemical equipment, discussed in subsequent chapters, the design procedure for machine elements suggested here may be referred to wherever such machine elements are provided.

5.2 Shafts

These are used for transmitting mechanical power with the help of rotary motion. They form either a direct component of an equipment or are merely used for transmitting rotary motion from a prime mover to an equipment. The first type of shaft is known as a machine shaft, while the second type is known as a counter shaft or a line shaft. The majority of rotating shafts carry a fairly steady torque and bending moment, the loads remaining fixed in space in both direction and magnitude.

5.2.1 Strength Under Steady Loads

The average torque transmitted by the shaft is assessed from the horse power

$$T_{av}=\frac{\text{h.p.}\times 75\times 60}{2\pi N} \tag{5.1}$$

where h.p.—horse power

N—speed in rpm

The maximum torque (T_{max}) is taken as 20 to 25 per cent more than the average torque.

The maximum value of the bending moment is determined from the loads normal to the axis of the shaft, by drawing a bending moment diagram. The resultant normal and shear stresses are determined by combining the maximum torque and bending moment, on the basis of the maximum normal stress and maximum shear stress (see equations 3.40, 3.41).

$$f_N=\frac{1}{2Z}\{M+(M^2+T^2)^{\frac{1}{2}}\} \tag{5.2}$$

$$f_S=\frac{1}{Z_p}(M^2+T^2)^{\frac{1}{2}} \tag{5.3}$$

where f_N—resultant normal stress

f_S—resultant shear stress

Z—modulus of section of the shaft.

Z_p—polar modulus of section of the shaft

M and T—maximum bending moment and torque respectively.

Shafts are usually circular in cross-section. In some cases hollow shafts are used. The diameters of shafts are determined by choosing permissible values for shear stresses, for materials which are ductile, in equation 5.3, and permissible values of maximum normal stresses, for brittle materials, in equation 5.2.

5.2.2 Strength Under Suddenly Applied Loads And Fluctuating Loads

When the shafts have to withstand suddenly applied loads, or fluctuating loads the values of the maximum torque and bending moment are increased by applying combined shock and fatigue factors; namely K_m and K_t. Values of these factors vary between 1.5 and 3 (Table 5.1).

Table 5.1

Nature of Loading	Value for K_m	Value for K_t
Stationary Shafts		
Gradually applied load	1.0	1.0
Rotating Shafts		
Suddenly applied load	1.5 to 2.0	1.5 to 2.0
Steady or gradually applied loads	1.5	1.0
Suddenly applied loads, minor shocks only	1.5 to 2.0	1.0 to 1.5
Suddenly applied loads heavy shocks	2.0 to 3.0	1.5 to 3.0

5.2.3 Deflection

It is necessary to control deflection of the shaft due to loads acting normal to the shaft. Determination of deflection in some standard cases of beams is indicated in Appendix B. The deflection in shafts should be limited to a maximum value of 0.02 mm at important positions along the shaft.

5.2.4 Slope and Twisting of Shaft

The deflection of the shaft creates a slope, while the torque creates an angular twist. The slope of the shaft at the supporting bearing should be limited to 4 minutes and the angular twist should be limited to 1° in a shaft length of 20 times the diameter.

5.2.5 Shaft Speed

There are certain limitations on the speed range in which the shaft should rotate. At certain speeds, the shaft tends to whirl or vibrate violently. Such vibrations are in resonance with the natural vibration frequency of the elastic system. At these speeds, known as critical speeds, the vibrations can cause complete and rapid failure of the shaft and equipment. The critical speeds correspond to the speed at which the centrifugal force of the displaced centre of mass of the shaft just equals the deflecting forces on it.

The determination of critical speed is based on the static deflections of the shaft, created due to the positions of loads and supporting bearings. The critical speed is given by (Fig. 5.1)

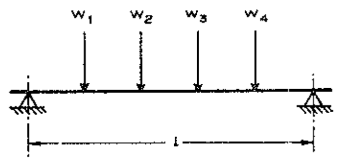


Fig. 5.1 Shaft freely supported

$$N_c = \frac{30}{\pi}\left\{\frac{g(\Sigma W\delta^2)}{\Sigma W(\delta)}\right\}^{\frac{1}{2}} \tag{5.4}$$

where

N_c—shaft speed rpm

g—gravity constant (981 cm/sec)

w_1, w_2, w_3—loads on shaft in kg

δ_1, δ_2, δ_3—static deflections under loads w_1, w_2, w_3 etc. (in cm)

or by Dunkerlays method

$$N_c = \frac{60\times 4{\cdot}987}{\left\{\delta_1+\delta_2+\delta_3+\dots\frac{\delta_s}{1{\cdot}27}\right\}^{\frac{1}{2}}} \tag{5.5}$$

where

δ_s—maximum deflection (in cm) due to weight of shaft

The operating speeds of the shaft should not be greater than 60 per cent of the critical speed.

5.2.6 Effect of Keyway

A shaft is rigidly attached to a hub or a machine part, by means of a key (Fig. 5.2). A keyway, in the form of an axial groove is made on the surface of the shaft, to accommodate the key. A keyway reduces both the strength and the rigidity of a shaft. This is taken into account by introducing a factor, similar to a stress concentration factor. The design of the shaft in such cases is based on a reduced value of the permissible

stress, which is usually taken as 75 per cent of the normal permissible stress.

5.2.7 Materials

Ordinary shafts are fabricated from medium carbon steel having

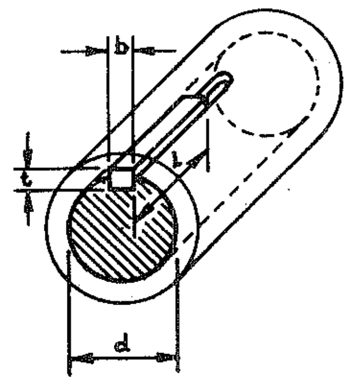


Fig. 5.2 Rectangular key fixed in shaft and hub

0.15 to 0.4 per cent carbon. When greater strength is required shafts are made of alloy steels, the most common alloys being

Table 5.2

Material (Steel)	Percentage carbon	Ultimate strength kg/cm²			Elastic limit, kg/cm²			Percentage Elongation
		Tension	Compression	Shear	Tension	Compression	Shear	
Commercial Cold rolled	0.10—0.25	4920	4920	2460	2460	2460	1250	35
Commercial turned	0.10—0.25	4200	4200	2100	2100	2100	1050	35
	0.15—0.25	4600	4600	2300	2500	2500	1150	26
Hot rolled or forged	0.25—0.35	4920	4920	2460	2800	2800	1230	24
	0.34—0.45	5300	5300	2650	3200	3200	1325	22
	0.45—0.55	5640	5640	2820	3520	3520	1410	20
3½% Nickel	0.15—0.25	6000	6000	3000	3900	3900	1500	26
Chrome Vanadium	0.25—0.35	6320	6320	3160	4220	4220	1580	25

nickel, nickel-chromium and chrome-vanadium steels. When resistance to corrosion is required shafts may be made of copper alloys or steel shafts may be lined with lead, glass or other materials.

Steel shafts are cold rolled or hot rolled. Most commercial shafting, after being hot rolled is turned and polished. Alloy steel shafts are always heat treated. Properties of shafting materials are given in Table 5.2

5.3 Keys and Pins

Keys are used to attach a machine part to a shaft and therefore transmit the torques preventing relative motion (both rotary and axial) between the two components. Some type of keys allow axial motion. According to the shape and function, keys are classified as square, rectangular, dovetailed, woodruff, round, flat, saddle, jib headed, etc. Pins are cylindrical in shape and are either straight or tapered. They may be used either for locating the relative position of two parts or for fixing two parts together.

Square or rectangular keys—The cross-section of the key is square or rectangular, with a straight length. When the torque is transmitted by the key, the tangential force acts as a shearing and crushing force on the key. This can be expressed as (Fig. 5.2)

$$\frac{T_{max}}{\frac{d}{2}} = lbf_s = \frac{lt}{2} f_c \tag{5.6}$$

For the shaft

$$T_{max} = \frac{\pi d^3}{16} f_s' \tag{5.7}$$

where

T_{max}—maximum torque transmitted
d —diameter of shaft
l —length of key
t —thickness of key
f_s —shear stress in the key
f_c —stress in crushing of key
f_s'—shear stress in the shaft.

If the material for shaft and key is similar then

$$f_s = f_s'$$

$$lb = \frac{\pi d^2}{8} \tag{5.8}$$

If l is taken as equal to 1.5 d, then

$$b \approx d/4$$

For a rectangular key, the value of t may be taken as $\frac{1}{16}$ d.

5.4 Couplings

These are used to connect two shafts which are in line. Couplings are mainly of two types namely rigid and flexible. According to their construction they are known as sleeve, butt-muff, flange, clamp, oldham, flexible bush or flexible disc couplings. Some of the more common types are considered here.

5.4.1 Rigid Flange Coupling

It consists of two flanges, one for each shaft, fixed by keys. Certain dimensions are based on practical and safety considera-

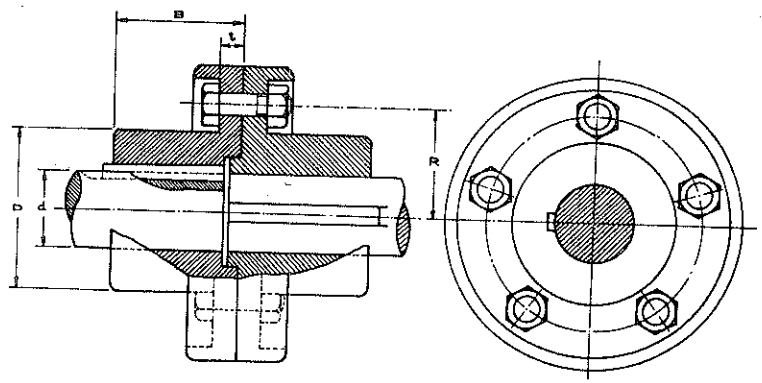


Fig. 5.3 Rigid coupling

tions. Fig. 5.3 shows a rigid coupling in which

$R = 1.5\ d$; $D = 2d$; $B = 1.5$ to $2d$

Number of bolts (n) $= 3 + 0.2d$ (d in cm)

Choose nearest whole even number.

The bolts are under shearing and crushing stresses

Force on the bolt $(F) = \frac{T_{max}}{R}$ (5.10)

$$F = \frac{\pi\delta^2}{4} f_s \times n = \delta t f_c \times n \quad (5.11)$$

where

δ—mean diameter of bolt

t—thickness of flange

f_s, f_c—shearing and crushing stresses.

Assuming permissible values of the stresses, the bolt diameter δ and the thickness of the flange t, can be calculated. It is usual to assume higher factor of safety for permissible stresses (f_s and f_c) to account for uneven loading, sudden application of load and bending moment.

The flange is usually made of cast iron, the stress in the hub and the portion of the flange can be checked by the following equations.

$$f_s\,(\text{hub}) = \frac{T_{max}}{\pi\,(D-d^4)} \times 16D \quad (5.12)$$

$$f_s\,(\text{flange}) = \frac{T_{max}}{\pi\,Dt \times D/2} \quad (5.13)$$

5.4.2 Split-Muff or Clamp Coupling

It consists of a sleeve generally made of cast iron split into two halves, which are clamped over the shaft by bolts (Fig. 5.4). This coupling is more suitable for vertical shifts, and is easy to dismantle. The transmission of torque is by pure friction which can be written in terms of shaft diameter and length, assuming uniform contact between shaft and sleeve.

$$T_{max} = \mu\,\pi\,d_p \frac{l}{2} \times \frac{d}{2} \quad (5.14)$$

where

μ—coefficient of friction between shaft and sleeve

d—diameter of shaft

p—contact pressure

l—length of sleeve.

$$\text{force per bolt } (P) = \frac{p.d}{\left(\frac{n}{2}\right)} \frac{1}{2} = \frac{2T_{max}}{\pi\,\mu\,d\left(\frac{n}{2}\right)} \quad (5.15)$$

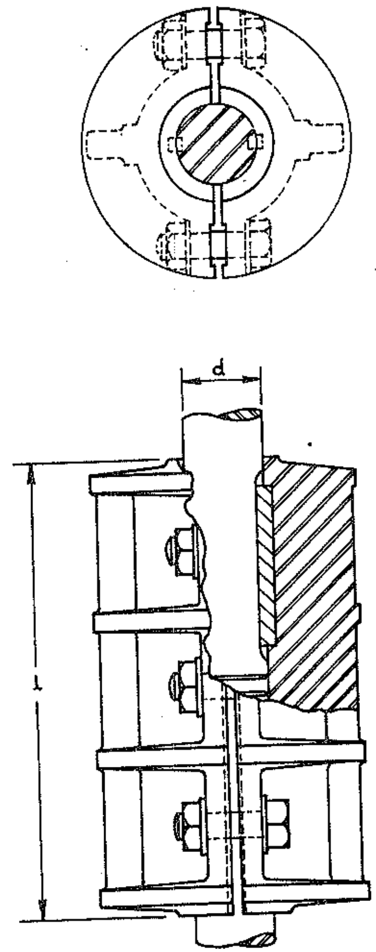


Fig. 5.4 Split-muff coupling

where

n—total number of bolts.

The number of bolts used are six upto 50 mm shaft diameter, and eight for larger diameters.

5.4.3 Flexible Coupling

This is essentially similar to a flange coupling with some modification to create flexibility. It is applicable particularly for connecting shafts which have a lack of alignment or shafts which are exerted with sudden changes of load. As shown in Fig. 5.5, the holes for the flange connecting pins are provided with rubber bushings cemented into the holes. These rubber

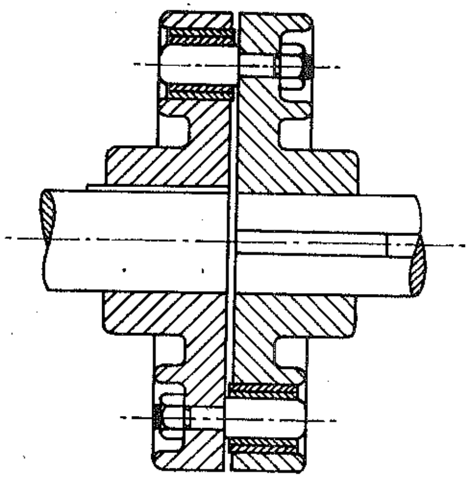


Fig. 5.5 Flexible coupling

bushings have a bronze bush cemented inside to reduce wear between the rubber bush and the pins. The thickness of the rubber bush is usually between 6 to 9 mm and that of the bronze bush about 2 mm.

The design of this coupling is somewhat similar to that of a rigid flange coupling. The pins are however located at a radius of 2.5 to 3 times the shaft diameter. The bearing pressure on the rubber bush is given by

$$p_b = \frac{T_{max}}{R \times n} \times \frac{1}{d_B \times t_b} \tag{5.6}$$

where

T_{max}—maximum torque
R—Radius at which the pins are located
n—number of pins
d_B—outsid diameter of bush
t_b—length of bush

5.5 Bearings

A bearing is a machine component, which acts as a support for a moving part, having rotary, oscillating or sliding motion. Its function is mainly to support the load of the moving part, with minimum loss due to friction. A bearing which supports a load normal to the longitudinal axis is known as radial bearing or journal bearing. A portion of a rotating shaft which is supported by the bearing is known as a journal. If the load on the bearing acts in the axial direction, the bearing is known as thrust bearing.

If the contacting surfaces of the rotating member and the bearing, cause a sliding action, the bearing is known as sliding contact bearing. If the action is of a rolling type the bearing is known as rolling contact bearing. In the first case, supporting contact is over a surface. In the latter case the supporting contact is on rollers or balls. In either case the friction between the rotating part and the bearing must be reduced by proper lubrication.

5.5.1 Sliding Contact Bearings

The bearing is lined with a relatively softer material than the shaft material. Such materials are brass, bronze, white metal, aluminium alloys, plastics, rubber, etc. (Table 5.3). The design is based on several considerations, such as

(*a*) diameter, length and clearance
(*b*) provision of sufficient lubrication
(*c*) dissipation of heat generated by friction
(*d*) strength and rigidity of shell, cap and bolts.
(*e*) minimum wear and provision for taking up wear.

Table 5.3

Material	Characteristics	Applications
1. Babbitt or white metal alloys (tin base)	Very expensive	Used for general purpose, resistant to corrosive effects of acids. I.C. engine bearings

Material	Characteristics	Applications
2. Lead-base white-metals	Low cost, lower strength and susceptibility, high coefficient of friction	General machinery purpose. Used at moderately high temperature. Load ranging from 10 kg/cm^2 to 22 kg/cm^2 at 60 m/min
3. Plastic bronze	Alloy of copper and lead, resistance to seizure, conformability and embedability. Less corrodible metal. Low coefficient of friction	Used in automobiles, locomotive and rolling mills where there are heavy loads, max. load carrying capacity is approx. 105 kg/cm^2 at 900 m/min
4. Phosphor bronze	Good mechanical and antifriction properties	All machines. The permissible load is 700 kg/cm^2. Max. rubbing speed 300 m/min
5. Gun metal	Chill casting gives a finer structure and is preferable to sand casting	Light loads
6. Brass	Lower cost than babbitt	At high pressures
7. Aluminium	Good corrosion, fatigue resistance and thermal conductivity, low embedability, high thermal expansion	Suitable for heavy load up to about 245 kg/cm^2 peak load and 140 kg/cm^2 continuous load at moderate speeds of 240 to 360 m/min
8. Cast Iron	Low friction, must have good lubrication	Cam shaft, light transmission. Max. speed 40 m per min., max. load 35 kg/cm^2
9. Cadmium base bearing metals	Allow much higher operating temperatures, than are safe with babbitt	Used for severe service in internal combustion engines
10. Porous bearings	Oil impregnated bearings.	Medium duty application in small size bearings. Where supply of lubricant is difficult, inadequate or infrequent. Rubbing speed from 22 to 450 m/min. with light load.
11. Wood bearing. (Lignum vitae, rock maple or oak)	Low cost, self lubricating	Conveyors
12. Rubber	Low coefficient of rubbing friction	Hydraulic turbines, centrifugal and deep well pumps
13. Nylon or metal filled Teflon	Extremely low coefficient of friction	Resistance to corrosion, light and medium load.

5.5.1.1 RADIAL BEARING OR JOURNAL BEARING

(*a*) The load in a radial bearing (Fig. 5.6) is supported on the projected bearing area given by $l \times d$

$$\text{Bearing pressure } p_b = \frac{P}{l \times d} \tag{5.17}$$

where

P—radial load

l—length of bearing

d—diameter of shaft

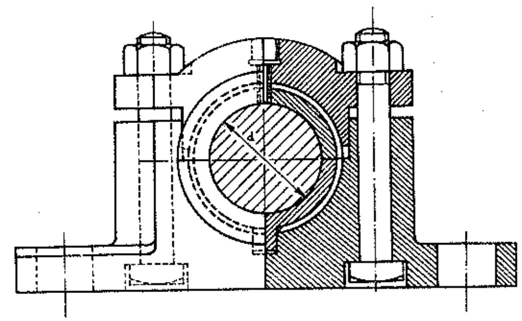


Fig. 5.6 Radial pedestal bearing

The ratio of $\frac{l}{d}$ varies between 1 and 3, and the permissible bearing pressure depends on the material and the type of bearing. Longer bearings have a lesser bearing pressure, but friction, torque and deflection increase. The clearance between the bearing and the journal (shaft) depends on the bearing pressure, rubbing velocity and materials of construction. In general, higher the rubbing velocity greater can be the clearance.

(*b*) *Lubrication*—Oil admission to a bearing must be in the region of low pressure. Oil grooves cut on the surface of the lined material, help to distribute the oil evenly over the entire surface.

(c) *Dissipation of heat generation*—The heat generated due ɔ the friction between the journal (shaft) and the bearing epends on the coefficient of friction, load and the speed. The oefficient of friction is a function of several factors such as iscosity of the lubricant, bearing pressure, speed, clearance, tc. Heat generation can be written as—

$$H/\text{minute} = \mu \times p_b \times d \times l \times \pi d N \qquad (5.18)$$

here

μ—coefficient of friction

p_b—bearing pressure

d—diameter of shaft

l—length of bearing

N—rpm of shaft

[eat is dissipated by flow of oil and conduction to other parts f the machine. The temperature of the oil film should be etween 28°C and 60°C. Lower temperature results in high perating viscosity and consequently high power loss. Higher mperatures may cause oil film breakdown, rapid vaporisation nd oxidation of the lubricant.

(d) *Strength and rigidity*—(i) The thickness of the brass or ronze shell is taken as $0.1d+3$ mm (where d is the diameter f the shaft in mm. For larger diameter bearings, the shells re made of cast iron with a layer of white metal or other soft aterial. (ii) *Cap*—The covering cap of the bearing is not sually subjected to a heavy load. However in some cases the ad acts on the cap. In such cases, the cap may be regarded a simple beam loaded at the centre and supported by the olding down bolts. It is also desirable to check the deflection f the cap. (iii) *Holding-down bolts*—The bolts used for holdig down the cap have to withstand a total load due to tighning of the cap in addition to the bearing load. Each bolt therefore, designed for a tension load which is 33 per cent eater than the normal load on each bolt. This also takes to account the uneven distribution of the load due to iction.

(e) *Wear in bearing*—With perfect lubrication there should no wear of the bearing surface. However with imperfect brication metal to metal contact causes a wear of the rubbing

surfaces. Different methods have been suggested for take-up of wear.

5.5.1.2 Thrust Bearing

Such bearings (Fig. 5.7) operate without clearance between the adjacent parts of the bearing and therefore, adequate supply of oil between the rubbing surfaces is extremely important. The

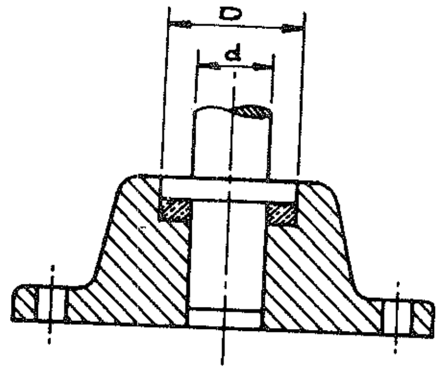


Fig. 5.7 Thrust bearing

area of the bearing should be adequate to take the thrust load. Oil grooves reduce the actual area of contact.

$$\text{Load } W = \frac{\pi}{4}(D^2 - d^2)p_b \qquad (5.19)$$

where

p_b—bearing pressure

D—diameter of thrust collar

d—diameter of shaft

W—thrust load.

Heat generation in thrust bearing is given by

$$H/\text{minute} = \tfrac{2}{3}\,\mu W\left(\frac{D+d}{2}\right) \times 2\pi N \qquad (5.20)$$

where

N—shaft speed in rpm

μ=coefficient of friction

5.5.2 Bearings with Rolling Contact—Anti-friction Bearings

In sliding contact bearings there is considerable loss of friction, particularly when the oil film lubrication is not perfectly maintained. In rolling contact bearings friction is reduced. Ball bearings [Fig. 5.8 (a)] have a lesser area of contact, and have therefore, limited load carrying capacity. For higher loads, roller bearings [Fig. 5.8 (*b*)] with a roller length of 4 to 6 times

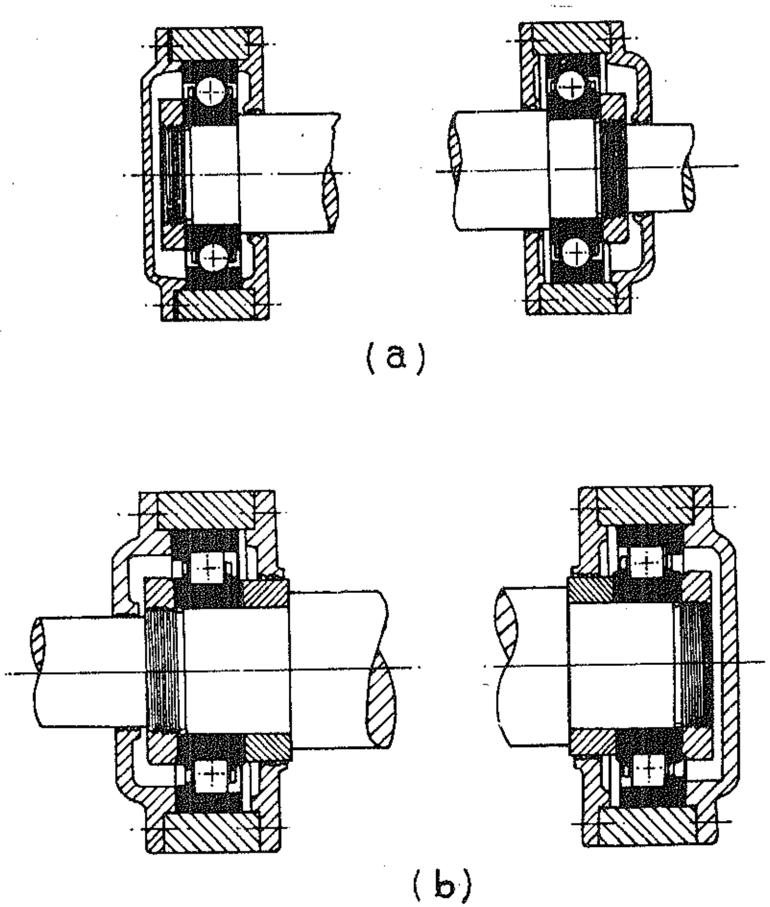


Fig. 5.8 Rolling contact bearings (a) Ball bearing (b) Roller bearing

the diameter are used. Such bearings may have cylindrical, conical or concave rollers.

Ball and roller bearings are standardised in respect of their capacity, dimensions, type, service conditions and life. In the selection of the bearing it is usual to choose a bearing from standard tables available from the manufacturers. The equivalent load p_e, which can be sustained by the bearing is given by the following equation:

$$p_e = XVP + YW \tag{5.21}$$

where

X—radial factor
V—rotation factor
Y—thrust factor
p—radial load
W—axial load

Values of factors X, V and Y are available in the Ball and Roller Bearing catalogues (*see* also IS 3824 Part II).

5.6 Belts and Pulleys

Belts and pulleys are used for transmitting power between two parallel shafts. In special cases these elements can transmit power in shafts which are perpendicular. Pulleys are fixed to the shafts and a flexible belt is made to envelope these pulleys. Transfer of power is due to the friction between the pulley and the belt. Two types of pulleys and belts are generally used. Pulley with flat rims and a belt of rectangular section are used for shafts with large centre distance. For short centre distance pulleys with V-grooves on their rim surfaces and with belts of trapezoidal section are used. [Fig. 5.9 (*a*) and (*b*)].

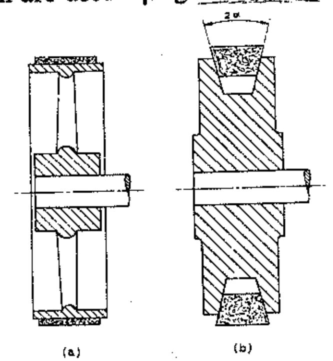


Fig. 5.9 Pulley with belt (a) Flat belt (b) V-belt

The power transmitted between pulleys by use of a belt depends on the friction and the area of contact. Tensions on the two sides of the belt are related as follows (Fig. 5.10).

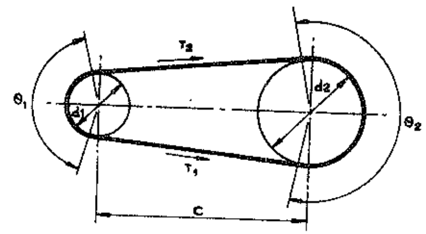


Fig. 5.10 Belt Tensions

For flat belt

$$\frac{T_1}{T_2}=e^{\mu\theta} \tag{5.22}$$

where

T_1—tension on the tight side
T_2—tension on the slack side
θ—angle of lap for the smaller pulley
μ—coefficient of friction between belt and pulley

For V-belt

$$\frac{T_1}{T_2}=e^{\mu_1\theta} \tag{5.23}$$

where

μ_1—μ cosec α and 2α—angle of the V-groove [Fig. 5.9(b)].

The angle of lap between the belt and the pulley may be found from the following:

$$\theta=2\pi\pm2\sin^{-1}\left(\frac{D_2-D_1}{2C}\right) \tag{5.24}$$

$$\simeq 2\pi\pm\frac{D_2-D_1}{C}$$

where

C—centre distance between pulleys
θ—angle of contact in the open belt drive in radians

The +ve sign gives the angle for larger pulley, while the -ve sign gives that for the smaller pulley.

Power transmitted is given by

$$hp=\frac{2\pi N_1(T_1-T_2)\frac{D_1}{2}}{60\times75} \tag{5.25}$$

Where

D_1—diameter of the smaller pulley
N_1—rpm of the smaller pulley

Speed ratio between the pulleys is

$$\frac{N_1}{N_2}=\frac{D_2}{D_1} \tag{5.26}$$

N_1, N_2 and D_1, D_2 are rpm and diameters of the two pulleys respectively.

The cross-section of the belt can be determined by the capacity of the belt to withstand tension T_1 on the tight side

$$T_1=f_t A \tag{5.27}$$

where

f_t—permissible tensile stress in the belt
A—area of cross-section of the belt

Belts are made of leather, canvas or rubberised canvas. Flat belts of rectangular section, are fabricated from plies of leather, with thickness of 4 to 6 mm in single ply and 7.5 to 10 mm in double ply. V-belts are fabricated from textile cord covered with rubber. The angle between the sides is 40° and the other dimensions are standardised.

According to the size, V-belts are classified as A, B, C, D and E.

Pulleys are made of cast iron, stamped sheet steel, aluminium, wood, and plastics. The hub of the pulley is attached to the shaft by a key. The rim of the pulley is made sufficiently wide to accommodate the belt on its surface. The hub and the rim are held in position by a series of arms, radiating out from the hub. The arms are circular or elliptical in cross-section. The belt tension creates a bending movement on the arms. The magnitude of the bending moment determines the size of the arm. V-pulleys are generally smaller in size and are fabricated from a solid piece of the material by providing a hole for inserting the shaft and grooves on the pheripheral surface for accommodating the belts.

5.7 Chain Drive

Chains are used for power transmission, hoisting and conveying. The system consists of two sprocket wheels attached to the shafts and a chain meshing with the teeth on the sprocket wheels. The arrangement is essentially a belt built up of rigid links which are hinged together in order to provide the necessary flexibility for the wrapping action round the driving and the driven wheels. These wheels have projecting teeth, which fit into suitable recesses in the links of the chain and thus enable a positive drive to be obtained. Various types of chains are used. The roller chain consists of a series of rollers freely mounted on and connected by thin metallic links (Fig. 5.11).

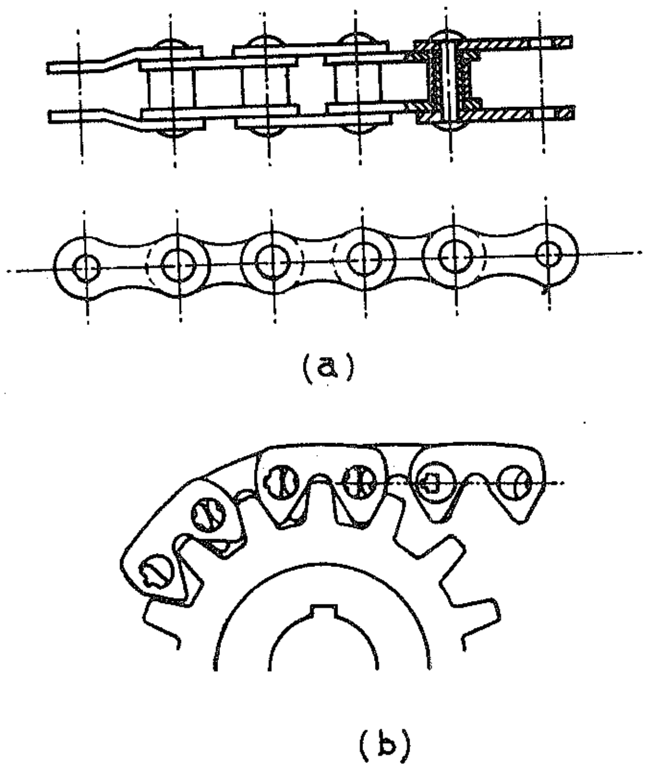


Fig. 5.11 Chain drive (a) Roller chain (b) Link chain

The chains are made of medium carbon or alloy steel. The speed ratio of the chain drive depends on the number of teeth in the driving and the driven sprockets. Short centre drives with high speed ratio require a fine pitch chain, while narrow large pitch chain is better adapted to low ratio long centre drives. When the speed ratio is less than 2.5, the minimum centre distance may be equal to one-half the sum of the sprocket diameters plus the pitch; for speed ratios greater than 2.5, the minimum centre distance should be equal to the sum of the sprocket diameter. The maximum centre distance should not exceed eight times the pitch.

Chains and sprocket wheels are standardised, according to the pitch, number of teeth, power transmission capacity, strength, etc. The maximum allowable pitch for a given speed may be found.

$$p_m \leqslant 10\left[\frac{3640}{N_1}\right]^{\frac{2}{3}} \qquad (5.28)$$

where

p_m—pitch in mm
N_1—speed of smaller sprocket in rpm

5.8 Gear Drives

Gear drives are used for transmission of power between parallel, intersecting or skew shafts. Two gear wheels are fixed to two corresponding shafts and remain in mesh throughout the

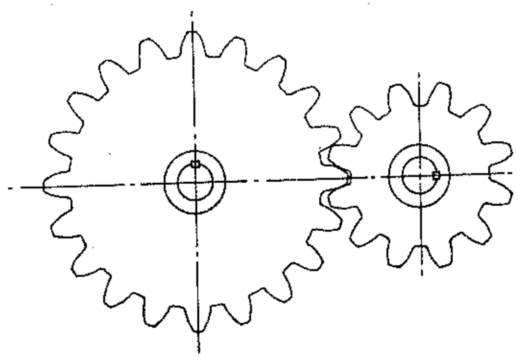


Fig. 5.12 Spur Gears

rotations of the shafts. Gear drives are preferred to belt drives for transmitting medium or large amounts of power at constant speed ratios.

Three general types of gears are classified with respect to relative position of the axes of the shafts on which the gears are mounted.

(a) Spur gearing (Figs. 5.12 and 5.13) is used for connecting shafts whose axes are parallel, which include external spur

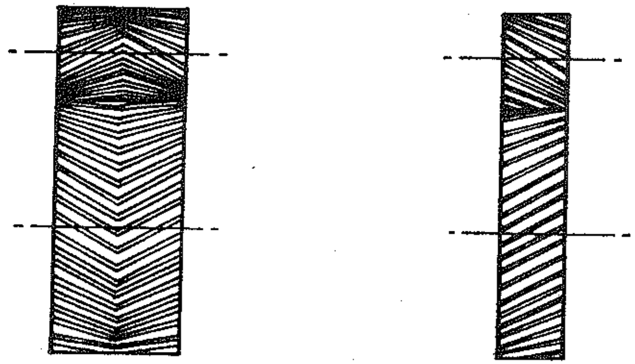


Fig. 5.13 Helical Gears

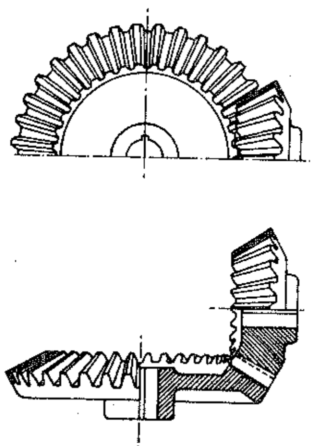


Fig. 5.14 Bevel Gears

gearing, internal gearing, rack and pinion, helical gearing and herringbone gearing.

(b) Bevel gearing (Fig. 5.14) is used for connecting intersecting shafts which include straight bevel gears, mitter gears and crown gears.

(c) The third type is used for connecting shafts whose axes are neither parallel nor intersecting, which include worm and worm wheels (Fig. 5.15) hypoid gearing and spiral gearing.

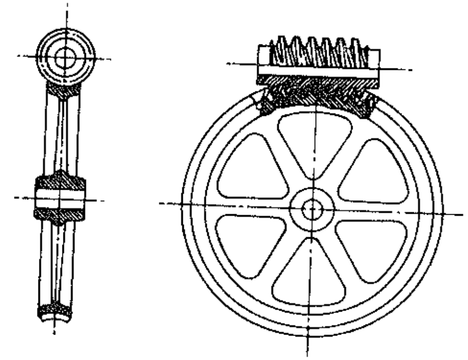


Fig. 5.15 Worm and Worm Gear

5.8.1 Spur Gear Terminology

Fig. 5.16 shows details of spur gear teeth. The profile of the

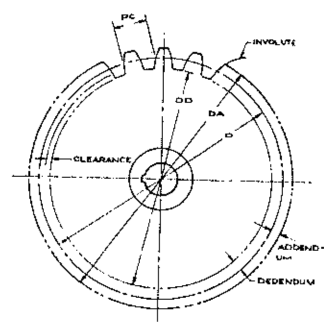


Fig. 5.16 Gear Terminology

tooth is an involute curve. Some of the terms used are indicated in the figure.

(a) *Pitch circle*—It is the circle of a cylindrical body, which by pure rolling action transmits motion to another cylindrical body, same as the motion of two gears.

(b) *Pitch diameter* (D)—It is the diameter of the pitch circle.

(c) *Circular pitch* (p_c)—It is the distance between similar points of adjacent teeth measured along the pitch circle.

(d) *Diametral pitch* (p_d)—It is the number of teeth per unit diameter.

(e) *Module*—It is the length of diameter per tooth. Teeth are standardised according to module size in mm.

(*f*) *Pressure angle* (α)—It is the angle which the line of action of the force or pressure makes with the common tangent to the pitch circle at the pitch point. Commonly used pressure angles are $14\frac{1}{2}°$ and 20°.

The above terms give the following relations:

$$p_c = \frac{\pi D}{T} \quad \text{where } T\text{—number of teeth.}$$

$$p_d = \frac{T}{D}\,;\; m = \frac{D}{T}$$

$$p_c \times p_d = \pi\,;\; p_c = \pi m$$

5.8.2 Spur Gear Tooth Design

$$\text{Force on the gear tooth } F = \frac{hp \times 450}{2\pi N(D/2)} \tag{5.29}$$

where

N=rpm

D=pitch circle diameter, in metres

The stress under static loading conditions is given by

$$F = fb\,p_c\,Y \tag{5.30}$$

where

f—permissible stress due to bending

b—face width which is

$2.5 \times p_c$—for cast teeth

$3.5 \times p_c$—for finished steel teeth

Y—Lewis factor

$$Y = 0.124 - \frac{0{\cdot}684}{T} \quad \text{for } 14\tfrac{1}{2}° \text{ pressure angle}$$

$$Y = 0.154 - \frac{0.912}{T} \quad \text{for } 20° \text{ pressure angle}$$

Under dynamic loading, which is actually the type of loading expected, equation (5.29) has to be multiplied by a velocity factor (k_v) and a service factor (k_s) (Table 5.4).

$$\text{where } k_v = \frac{10}{10+v}$$

v—pitch line velocity in metres/sec.

TABLE 5.4

Service factors (Ks)

Type of load	Intermittent	Continuous
Steady load	1	0.8
Light and Medium shock	0.8	0.65
Heavy shock	0.6	0.5

Out of the two wheels, which remain in mesh, while transmitting motion, the smaller wheel is known as pinion and the larger is known as gear wheel. The minimum number of teeth for the pinion should be sufficiently large to reduce the specific loading per tooth to a minimum value, so that the wheels will work smoothly without excessive wear and heating. In general, the pinion should have a pitch circle diameter of $2.2d$ for steel, $2.5d$ for cast iron and $2.8d$ for plastics, where d is the diameter of the shaft. A velocity ratio of about 4 to 6 is usually adopted. Standard modules vary from 0.3 mm to higher values. Gears are made of cast iron, steel, brass, bronze, plastics and other non-metallic materials.

5.9 Riveted Joints

Rivets are used for connecting permanently structural sections, plates and machine components. The joints made between structural sections or between machine components must have enough strength and rigidity. Joints for storage tanks, drums and pressure vessels must have enough strength, rigidity and must also be leakproof. In recent years riveted joints are being replaced by welded joints in structural and pressure vessel work. However, riveted joints are still in use to a limited extent, in pressure vessels fabricated from non-ferrous, ductile materials such as copper, brass, aluminium etc., and for fabrication of certain machine components such as frames, bases, etc. Various forms of rivet heads are shown in Fig. 5.17.

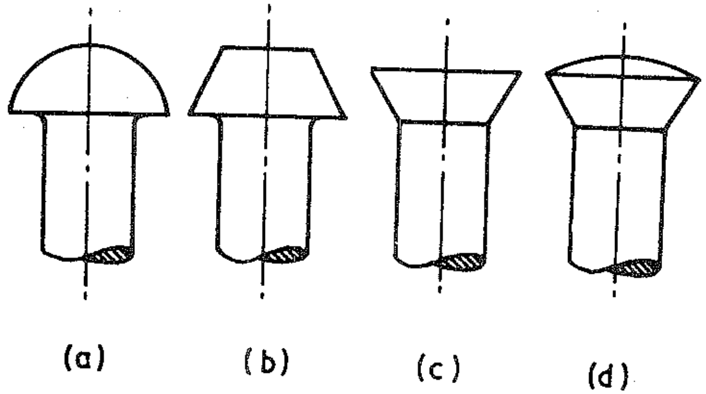


Fig. 5.17 Rivet heads (a) Snap (b) Pan (c) Countersunk (d) Rounded countersunk

Rivetted joints may be classified as follows :

Riveted Joints
- Lap joint
  - ngle
  - Double
  - Triple
- Butt joint
  - Single butt Strap
    - Single
    - Double
    - Triple
    - Quadruple
  - Double butt strap

n lap riveting, one plate overlaps the other and the rivets pass hrough drilled plates. In butt riveting the plates are kept in

alignment and a butt strap or a cover plate is placed over the joints and riveted to each plate. Frequently two cover straps are used. These straps are of equal width. In some cases the outer strap may be lesser in width than the inner one. Depending on the rows of rivets the joint is termed as single, double, triple riveted, etc. If the rivets are spaced opposite to each

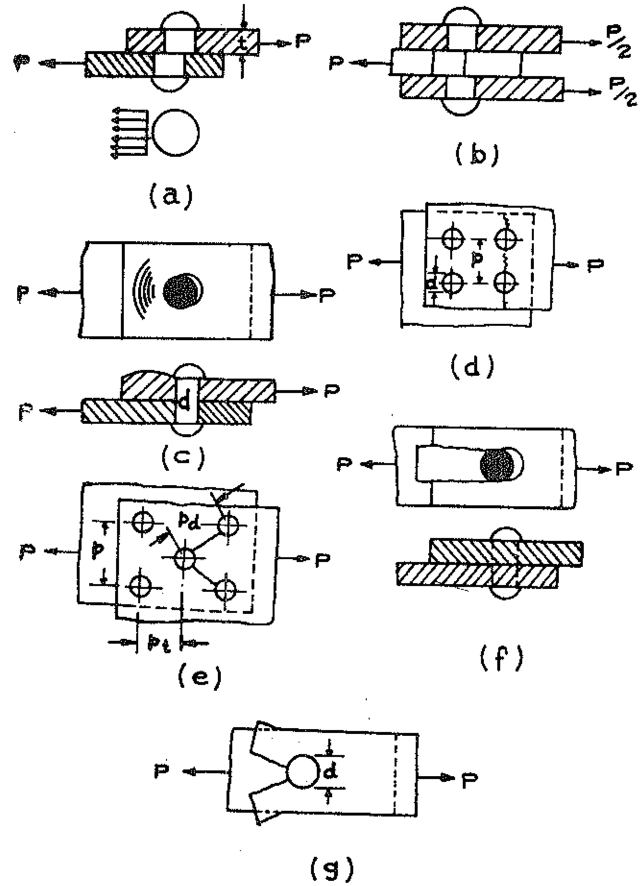


Fig. 5.18 Modes of failures of riveted joints
(a) Single shear of rivet, (b) double shear of rivet, (c) crushing of plate or rivet, (d) tearing of plate across rivets, (e) tearing of rivet diagonally between rivet holes, (f) shearing of margin (g) end tearing of plate

other in adjacent rows, the joint is said to be chain riveted and if the rivets are staggered, it is known as zig zag riveted. Fig. 5.19 (see section 5.9.6) shows different types of lap as well as butt joints.

5.9.1 Terminology

Pitch (p)—The distance between centres of adjacent rivets on the same line or row is known as pitch.

Transverse pitch (p_t)—The distance between two rows of rivets is known as transverse pitch.

Diagonal pitch (p_d)—The distance between centres of rivets in adjacent rows of staggered riveting is known as diagonal pitch.

Margin—The distance between the edge of the plate and the nearest row of rivet is known as margin.

5.9.2 Modes of Failure

Failure of a riveted joint (Fig. 5.18) may take place due to any one of the following modes.

(a) Shearing of a rivet along one cross-section, known as single shear.

(b) Shearing of a rivet simultaneously along two cross-sections known as double shearing.

(c) Crushing of plate or rivet.

(d) Tearing of plate across rivets

(e) Tearing of plate in a zig-zag line diagonally between the rivet holes in staggered riveting.

(f) Shearing of margin.

(g) Tearing of the end of the plate.

Failure due to the last three modes can be prevented if the following rule is observed.

The minimum distance between the centre of a rivet and the edge of the plate to be $1.5d$ and the transverse pitch to be $2d$ to $2.5d$, where d is the diameter of the rivet.

5.9.3 Design of Joints

When the load is applied to a riveted joint, it is necessary t investigate whether the joint is capable of withstanding the loa without failure. This can be checked by considering th strength of a pitch length of the joint, under each mode o failure.

Shearing of rivets (Fig. 5.18) (a) and (b)—The resistance t shearing of the rivets will depend on whether the rivet is i single shear or double shear, and the number of rivets to b sheared over the length of a pitch. In case of lap joints an butt joints with single cover strap, the rivets will be in singl shear, while in case of butt joint with two butt straps the rive will be in double shear. Theoretically the strength of the rive in double shear will be double that of rivet in single shear But in actual practice, it is taken as 1.875 times single shear.

Strength of one rivet in single shear $=\frac{\pi}{4}\, d^2 f_s$.

(lap joint)

Strength of one rivet in double shear $=1.875\times\frac{\pi}{4}\, d^2 f_s$.

(butt joint)

From the number of rivets to be sheared in a pitch, the total resistance to shearing is calculated.

Crushing of rivet (Fig. 5.18) (c)—The resistance to crushing can be determined from the area of the rivet under crushing action.

Strength under crushing—$dt\, f_c$

From the number of rivets to be crushed in a pitch, the total resistance to crushing is calculated.

Tearing of plate (Fig. 5.18) (d)—The resistance of the plate to tearing along the weakest section, is determined for a pitch.

Tearing strength of the plate section $=(p-d)\, t\, f_t$.

5.9.4 Efficiency of the Joint

The efficiency of the joint is the percentage ratio of the least strength of a joint to that of the solid plate. This is determined by first calculating the strength of the joint for a pitch under different modes of failure. The ratio of the lowest strength to the strength of the original plate is then determined.

For example

$$\text{Efficiency in tearing of plate} = \frac{(p-d)\,t\,f_t}{pt\,f_t} \tag{5.30 a}$$

$$\left.\begin{aligned} \text{Efficiency in shearing of rivet} &= \frac{\frac{\pi\,d^2}{4}\,f_s}{pt\,f_t} \\ \text{or } &\frac{1.875\left(\frac{\pi}{4}\,d^2\,f_s\right)}{pt\,f_t} \end{aligned}\right\} \tag{5.30 b}$$

$$\text{Efficiency in crushing of rivet} = \frac{dt\,f_c}{ptf_t} \tag{5.30 c}$$

5.9.5 Design of Joints Made in Structural Sections

The diameter of the rivet is assumed as follows:

For lap joint $d=2t$ for plates of 1 cm thickness or less.

For butt joint $d=t+0.6$ cm

where d and t are in cm

The thickness of butt strap

$t_s=1.1\ t$ for single strap

$t_s=0.6$ to $0.8t$ for double strap.

The pitch is determined to obtain as high an efficiency as possible, consistent with the size of the rivet head, area for accommodation and fabrication requirements. A suitable pitch may be between 3 to 3.5 times the diameter of the rivet.

5.9.6 Design of Joints for Pressure Vessels

In forming pressure vessels, joints (Fig. 5.19) are made for the following:

(a) A longitudinal joint to form a cylinder from a plate.

(b) A circumferential joint to connect two cylindrical pieces and to connect a cover at either end of the cylinder.

(c) A circumferential joint of two hemispherical pieces to form a spherical cylinder.

(d) A joint to connect a nozzle or manhole.

Table 5.5 gives various proportions of the longitudinal joints for a cylindrical vessel. The type of joint and the probable efficiency are chosen from the table. The efficiency is finally checked by considering all possible modes of failures [Equations 5.30 (a), (b), (c)]. For details of pressure vessel design see Chapter 6.

Table 5.5

Type Ref. (Fig. 5.19)	$D \times P$ kg/cm Longitudinal Joint	Efficiency	dia of rivet cm	pitch p cm	transverse p_t cm	Cover plate thickness cm
(*a*)	up to 1000	0.56	$\sqrt{5t}-0.4$	$2d+0.8$	—	—
(*b*)	800 to 1900	0.69	,,	$2.6d+1.5$	$0.6p$	—
(*c*)	,,	0.68	,,	$2.6d+1$	$0.8p$	—
(*d*)	1400 to 2700	0.75	,,	$3d+2.2$	$0.5p$	—
(*e*)	700 to 1700	0.68	$\sqrt{5t}-0.5$	$2.6d+1$	—	$0.75t$
(*f*)	1600 to 3200	0.80	$\sqrt{5t}-0.6$	$5d+1.5$	$0.4p$	$0.8t$
(*g*)	1300 to 2700	0.75	$\sqrt{5t}-0.6$	$3.5d+1.5$	$0.5p$	$0.75t$
(*h*)	2600 to 4600	0.85	$\sqrt{5t}-0.7$	$6d+2$	$0.3p$	$0.8t$
(*i*)	200 to 4800	0.85	,,	,,	$0.3p$	$0.8t$
(*j*)	2500 to 5000	0.85	,,	,,	$0.3p$	$0.8t$

5.10 Welded Joints

This method of joining components of machines or structural sections is now being used extensively in place of riveting or bolting. In many cases it has replaced even castings, since parts such as machinery bases, frames, brackets are now being fabricated by welding from standard sections. This method produces parts which are much greater in strength and rigidity than parts made of cast iron.

In welding no holes are required to be made as in the case of riveting. The entire cross-section of the component is therefore, effective. The weight of the material is therefore, less by as much as 25 per cent, thus effecting a saving not only in cost of material but also in the cost of handling. In cases where the thickness of the plates to be joined is large, this method is the only solution. Pressure vessels, tanks, and other similar equipment are most conveniently and efficiently fabricated by welding.

5.10.1 Type of Joints

There are two types of joints made by the method of welding:

(A) *Butt weld*, which is made by placing two pieces to be welded, close to each other and filling the gap by welding material.

(B) *Fillet weld*, which is made by overlapping two pieces to be welded and filling the corners formed between them by suitable welding material.

By use of the above methods it is possible to make joints by placing the components to be jointed in different relative positions as shown in Fig. 5.20. The figure shows butt welded plates (*a* to *b* and *k* to *n*) and fillet welded plates (*c* to *j* and *p* to *s*).

5.10.2 Design of Welded Joints

Butt welds—When the load is applied to the plates or machine components which are connected by butt welds, it is necessary to ensure that the joint will withstand the load without failure. The nature of the stresses, namely tensile, compressive, shear etc., created in the weld are similar to those in the original plate or component to which the load is applied. The magnitude of the stress however, depends on the size of the welded metal. In Fig. 5.20 (a) a single V-butt weld is loaded by a tensile force F, which will create a tensile stress in the weld which is given by

$$F = f_{wt}\, bL \tag{5.32}$$

where b—throat of weld

L—length of weld

and f_{wt}—tensile stress in the weld material.

Similar cases can be considered for the loads, which give rise to compression, bending and torsion Fig. 5.20(*b*) and (*k*) to (*n*).

Fillet welds—These welds are of two types: (i) transverse fillet welds which are under tension and (ii) parallel fillet welds which are under shearing action. Fig. 5.20 (*c*) and (*e*) show a transverse fillet weld subjected to a load F, which creates a tensile stress in the weld

$$F = 2\times 0.707bL\times f_{wt} \tag{5.33}$$

where $0.707bL$ represents the throat area of the weld

f_{wt}— tensile stress in the weld material.

Fig. 5.20(f) shows a parallel fillet weld subjected to a load F which creates a shear stress in the weld

$$P = 2\times 0.707bL\times f_{ws} \tag{5.34}$$

where

$0.707bL$—represents the throat area of the weld

f_{ws}—shear stress in the weld material.

Since $f_{wt} > f_{ws}$ for the same weld material, a transverse fillet weld is stronger than a parallel fillet weld. But in design practice both the welds are assumed to have equal strength. Similar cases of fillet welds are shown in Fig. 5.20 for loads which give rise to bending and torsion.

5.10.3 Joints for Pressure Vessels

These are similar to those indicated under riveted pressure vessel joints, except that the riveting being replaced by welded joints. Table 5.6 shows the efficiency of the welded joints and their range of application. For details of pressure vessel components and welded connections applicable to pressure vessels, see Chapter 6.

5.11 Threaded Fasteners

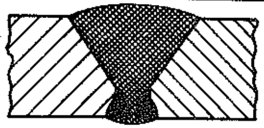
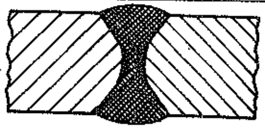
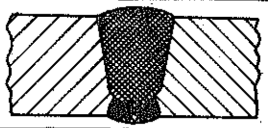
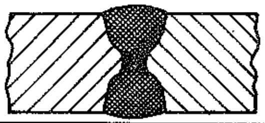
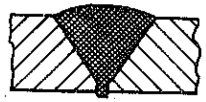
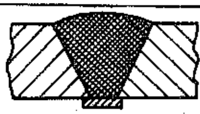
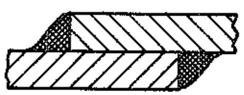
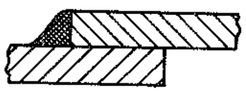
Bolts, studs and screws are used for fastening and making detachable joints between plates, sections, machine components etc. The standard V-threads formed on such elements are either fine or coarse. The fine threads are used where greater strength is required or where the connected parts are subjected to vibrations or for fine adjustments. Coarse threads are used where threads are to be formed in weaker materials. The following threaded fastenings are generally made (Fig. 5.21).

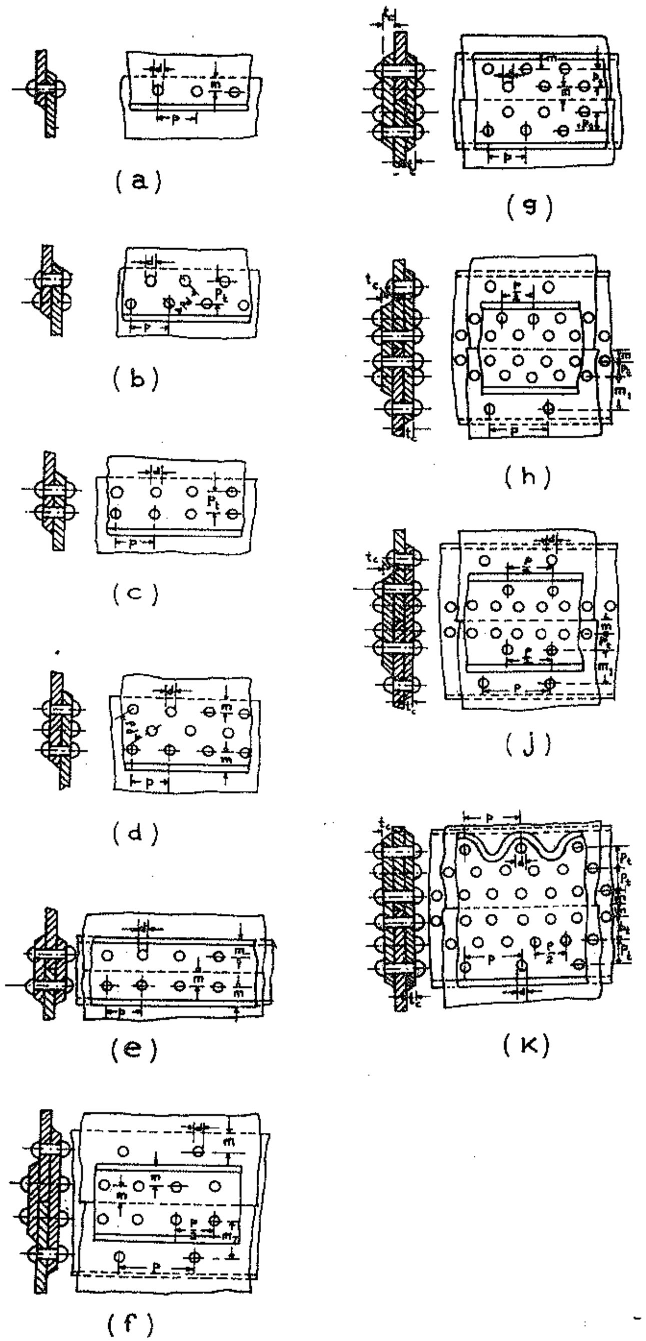
5.11.1 Bolts and Nuts

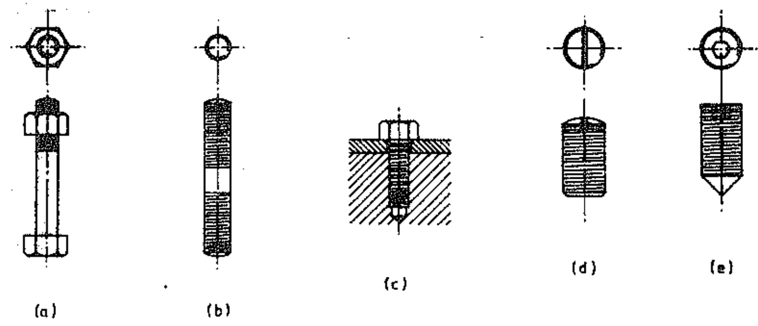
The size of the bolt is specified by the external diameter of the thread and the length is measured under the head. The height of the nut is generally 0.8 times diameter of the bolt for steel, 1.5 times diameter for bronze and 1.5 to 2 times diameter for cast iron.

5.11.2 Tap Bolts or Cap Screws

These devices do not require a nut, but are screwed directly into one of the components to be connected. These are generally used for joints which are not to be detached frequently and where through bolts are not possible because of lack

Joint	Name	Joint efficiency	*Application* Longitudinal	Circumferential
	Double-welded butt joint with single 'V'	0.75—0.95	All thicknesses	All thicknesses
	Double-welded butt joint with double 'V'	0.75—0.95	All thicknesses	All thicknesses
	Double-welded butt joint with single 'U'	0.75—0.95	All thicknesses	All thicknesses
	Double-welded butt joint with double 'U'	0.75—0.95	All thicknesses	All thicknesses
	Single-welded butt joint with 'V' groove without backing strip	No value given because this joint may only be used for circumferential joints	Not allowed	Not exceeding 15 mm plate
	Single-welded butt joint with 'V' groove with backing strip	0.65—0.85	Not exceeding 30 mm plate	Not exceeding 30 mm plate
	Single-welded butt joint with 'U' groove without backing strip	No value given because this joint may only be used for circumferential joints	Not allowed	Not exceeding 15 mm plate
	Single-welded square butt joint without backing strip	No value given because this joint may only be used for circumferential joints	Not allowed	Not exceeding 6 mm plate
	Single-welded square butt joint with backing strip	0.60—0.85	Not exceeding 6 mm plate	Not exceeding 6 mm plate
	Double full-fillet lap joint	0.45—0.70	Not exceeding 8 mm plate	Not exceeding 15 mm plate
	Single full-fillet lap joint with plug weld	No value given because this joint may only be used for circumferential joints	Not allowed	Not exceeding 14 mm plate. Plugs shall be proportioned to take 20 per cent of total load
	Single full-fillet lap joint	No value given because this joint may only be used for circumferential joints	Not allowed	Not exceeding 12 mm plate





5.21 Threaded fasteners
(a) bolt and nut
(b) Stud
(c) Tap bolt
(d) and (e) Set screws

of space. The threaded length is equal to *d* for steel, 1.5 *d* for cast iron and 3 *d* for aluminium.

5.11.3 Set Screws

These have a primary function to prevent relative motion.

5.11.4 Studs

These have threads at both the ends. They are prefered when the joint is to be detached or disassembled

5.11.5 Special Bolts: Swing Bolts, Anchor Bolts, Eye Bolts and T-head Bolts

These are used in special applications such as pressure vessel covers, foundations, lifting devices etc.

5.11.6 Locking Devices

In parts subjected to varying and vibrating loads, the nuts used for tightening the bolts or studs work loose. This tendency is prevented by using various locking devices such as lock nuts, slotted nuts, castle nuts, nuts with pins, and nuts with different types of washers.

5.11.7 Stresses in Screw Fastenings

In order to determine the size of the screwed fastenings, it is necessary to determine stresses under static loading as well as dynamic loading. Under static conditions of loading the following stresses should be considered.

(a) Initial stresses due to tightening
(b) Stresses due to external forces
(c) Stresses due to combination of (a) and (b).

5.11.7.1 INITIAL STRESSES

When a bolt, stud or screw is tightened the stresses created are

(a) tensile stress due to stretching of the bolt
(b) compression or bearing stresses on the thread
(c) shearing stresses across threads
(d) torsional shear stresses caused by the frictional resistance of the threads
(e) bending stresses if the surfaces under the head or nut are not perfectly normal to the bolt axis.

None of the above mentioned stresses can be accurately determined. So the bolts or screws are designed on the basis of direct tensile stress induced in the bolt or screw by a reasonable tightening action with a standard wrench. This initial tightening force is given by [3150 d] in kg, where d is the diameter of bolt in cm. This value should be used only for leak proof joints. It may be reduced for fastenings not made as tight as leak proof joints.

$$\text{Stress due to tightening } f = \frac{3150d}{A} \tag{5.35}$$

where A—mean area of cross-section of bolt. This value should be less than the permissible stress, determined with a high factor of safety.

It is difficult to tighten a bolt by hand wrenches to develop full strength of the bolt, if it is larger than 5 cm diameter. Beyond this size, three other methods are used namely, (1) bolt tensioners (2) bolt heaters (3) impact wrenches.

5.11.7.2 STRESSES DUE TO EXTERNAL FORCES

Bolts, studs, screws are subjected to tensile or shear or bearing stresses due to external forces. These can be calculated as follows:

$$\text{Axial tensile force} = f_t \times A$$

$$\text{shear force} = f_t \times A$$

$$\text{bearing or crushing force} = f_c \times d_c \times t$$

A—area of cross-section of the bolt

$$= \frac{\pi}{4}\,\frac{(d_o^2 + d_i^2)}{2}$$

t—length of bolt under bearing or crushing load

d_o—outside diameter

d_i—core diameter.

5.11.7.3 STRESSES DUE TO COMBINATION OF (a) AND (b)

In a number of applications, where the tightening of the bolt is limited, the stresses induced are mainly due to external load. In such cases the practice is to design the bolt for 1.25 to 1.75 times the external load or use a lower value of the permissible stress. In some applications design of the bolts based on initial stresses is desirable; in others, design for a combination of initial and load stresses is necessary. Such applications are cases where bolts and studs are used for tightening two flanges with gaskets, made to ensure leakproof joints. If the gasket is soft, the resultant load on the bolt will be approximately the sum of the external load and initial tightening load. In cases where the gasket is hard or when there is a metal to metal contact between the flanges the resultant load is either initial tightening load or external load whichever is greater. Detailed design procedure for flanged joints is given in Chapter 6.

5.12 Packing and Gaskets

Packings are used for dynamic sealing of moving parts, which have reciprocating, rotary or helical motion. The packing is installed in a stuffing box, surrounding the moving part and is then compressed by an adjustable component called a gland. Gaskets are static seals, which are used for making a pressure-tight joint between two rigid elements, usually in the form of flanges. Various gasket types and their applications are considered in Chapter 6.

The general arrangement of the stuffing boxes with packing is shown in Fig. 5.22. These indicate the shape of the stuffing

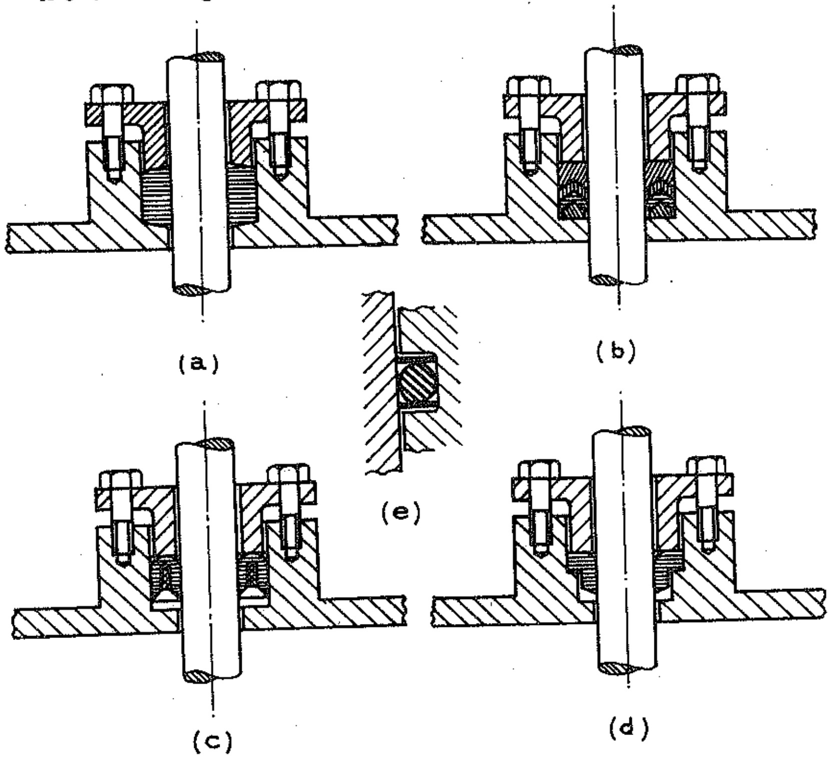


Fig. 5.22. Arrangement of packing in a stuffing box
(a) Ring packing
(b) V-ring or Chevron packing
(c) U-cup packing
(d) Hat packing
(e) O-ring packing

box, the type of packings and the gland used for compressing the packing close to the moving part, so as to prevent leakage. Packing throttles leakage between a moving and stationary part. However, it does not stop leak completely. If it runs dry, the packing will get heated and will wear out fast. It will also create considerable friction. A leakage rate of 10 to 12 drops per minute, which flows through the clearance between the moving shaft and packing acts as a lubricant. Packings are made of five basic materials namely, fibres, metals, plastics, rubber and leather, which are combined with various impregnants and lubricants, depending on the application. Asbestos is the most widely used material because of its strength, temperature resistance and versatility. Synthetic fibres such as rayon and nylon are good for water service at temperature below 120°C. Nylon has a low coefficient of friction. Compression metal packings are usually in foil form, although sometimes in braided and shredded forms. Lead, copper and aluminium are common metals for packings that serve as end rings. The most common plastics for packings are teflon and kel-F both of which are highly resistant to chemical attack. They can withstand temperatures upto 250°C. Natural rubber, neoprene, buna-N, butyle and silicone are some of the synthetic rubbers used for packings. Leather packings have high tensile strength. Plastics, rubber and leather are moulded and machined to form shapes like, V-, U- and O-rings.

Impregnated packings cannot provide enough lubricant when high temperatures are involved and when conditions create high frictional heat. External lubrication must be supplied through a lantern ring, which is usually located near the middle of the stuffing box. The sealing liquid may be injected into the packing at a pressure higher than that of the fluid being contained. They are also required for packings to prevent leakage of gases, which leak easily and are poor lubricants (refer to Figs. 14.14, 14.15, 14.16).

5.13 Design of Stuffing Box and Gland

Fig. 5.23 shows the details of the arrangement with a shaft either rotating or reciprocating. Some of the important

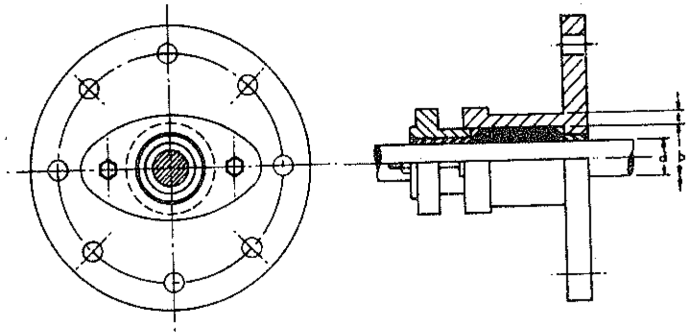


Fig. 5.23 Stuffing box and gland

dimensions may be determined as follows:

$$b = d + \sqrt{d} \text{ (cm)} \tag{5.36}$$

$$t = \left(\frac{pb}{2f}\right)10 + 6 \text{ mm} \tag{5.37}$$

where d—diameter of shaft (cm)

p—internal pressure (kg/cm^2)

b—internal diameter of stuffing box (cm)

t—thickness in (mm)

f—permissible stress in the material of stuffing box (kg/cm^2).

Load on the gland $F = \frac{\pi}{4} p\,(b^2 - d^2)$ (5.38)

If the shaft is reciprocating, the load is increased by 30%

The studs take the load F

$$F = \frac{\pi d_i^2}{4} \times n \times f_t \tag{5.39}$$

where n—number of studs

f_t—permissible stress in studs

d_i—mean diameter of stud.

The diameter of stud is increased to allow for initial tightening. A minimum diameter may be 16 mm. The flange thickness may be about 1.75 times the stud diameter.

The stuffing box may be made of cast iron or mild steel while the gland may be made of brass or gun metal. The

studs are of steel. For corrosive conditions special materials may be selected as lining materials (see Figs. 14.14, 14.15, 14.16).

5.14 Mechanical Seals

For dynamic sealing of rotating parts, such as pump shafts, agitator shafts, the conventional method of packed stuffing box and gland is unsatisfactory, particularly when pressure exceeds 10 kg/cm^2 or temperature exceeds 120°C or speed of rotation of shaft exceeds 300 rpm. In such applications a mechanical seal works efficiently. It consists of two rings one of which is stationary and other rotates with the shaft. Sealing action is obtained by intimate contact between opposing faces of the two rings.

Fig. 5.24 indicates details of a mechanical seal, which consists of a rotary ring held on the shaft by a flexible O-ring. The stationary ring is also placed in a box through another O-ring. The flexible O-rings prevent leakage between rotary seal ring and the shaft as also between the stationary ring and the box, while allowing the rings sufficient freedom of movement to maintain full free contact. The O-rings also compensate for the lack of alignment, runout, thermal expansion and shaft vibration. O-rings are made of natural synthetic or silicone rubbers and teflon. The scaling faces of the ring are finely lapped, so that they maintain close contact. The rotary ring is usually metallic, while the stationary ring is made of non-metallic material. Due to low coefficients of friction, carbon or graphite is used for this purposes. A spring gripping the shaft at one end and the rotary seal at the other end provides initial pressure on the sealing faces.

5.14.1 Design of Mechanical Seal

The design of the seal may be considered under three different headings.

5.14.1.1 Mechanical Design Features

It is necessary to provide two optically flat surfaces, which will maintain intimate contact and have low coefficient of

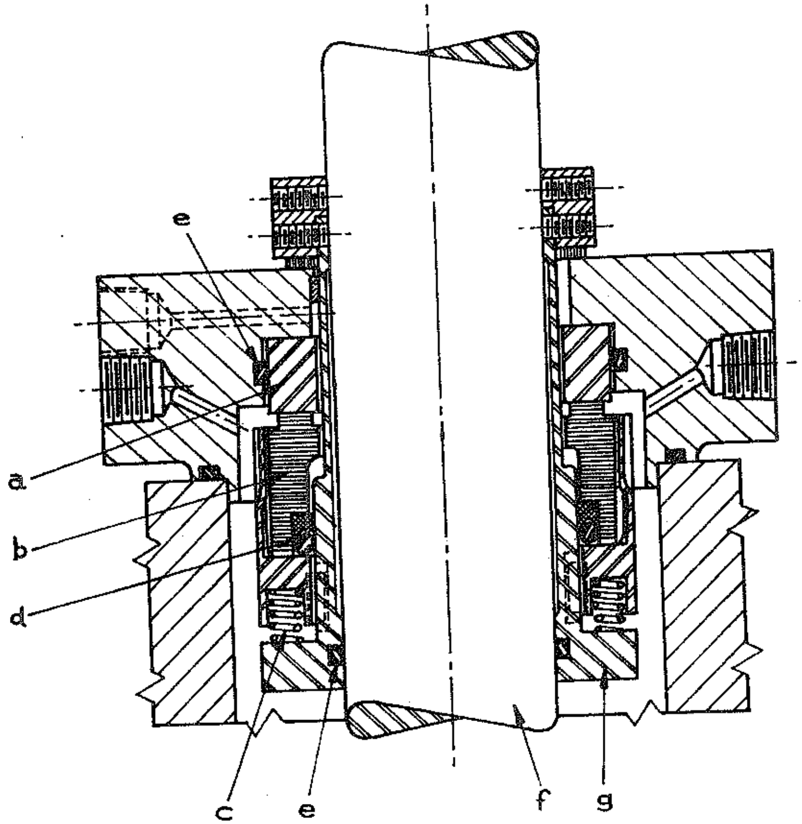


Fig. 5.24. Mechanical seal
(a) fixed ring
(b) rotating ring
(c) spring
(d) and (e) O-rings
(f) shaft
(g) shaft sleeve

friction under all conditions. The parallel lapped surfaces must not be disturbed by distortion due to mechanical,

hydraulic or thermal effects. Mountings for the seal and face should prevent distortion while allowing for flexibility of shaft movement relative to the casing.

5.14.1.2 HEAT DISSIPATION AND POWER ABSORPTION

The operating condition in many seals, may be light and power consumption due to friction between ring faces may be ignored. However, under severe conditions, there is a likelihood of overheating. In such cases a circulation of coolant is indicated. This is effected by circulating the liquid itself, from a point of high pressure and back to the line at the point of low pressure. When it is essential that the fluid handled should not come in contact with the seal faces, because of temperature or possibility of corrosion, a separate fluid for cooling is used.

5.14.1.3 LEAKAGE CONTROL

In order to reduce power losses and excessive temperature rise by rubbing of seal faces, it is essential that a thin film of liquid be present between the faces for lubrication purposes. The loss of liquid through this gap constitutes leakage. The seal design must aim at keeping both leakage and loss to a minimum. There are three major factors which influence the formation of a fluid film between seal faces.

(i) Wetting properties of the face and seal materials.
(ii) Vapour pressure and boiling point of coolant.
(iii) Fluid pressure relative to face pressure loading.

Reading References

Vallance, A., and Doughtie, V.L., *Design of Machine Elements*, McGraw-Hill, New York, (1951).

Joshi, T.N. and Shah, R.T., *Machine Elements*, Vol. I, Kothari, Baroda, (1964).

Joshi, T.N. and Shah, R.T., *Machine Elements*, Vol. II, Kothari, Baroda, (1965).

Mallev V.L. and Hartman, J.B., *Machine Design*, Scranton International, (1946).

Spotts, M.F., *Design of Machine Elements*, Prentice-Hall, New Delhi. (1969).

Fairs, V.M., *Design of Machine Elements*, Macmillan, New York, (1965).
Redford, G.D., *Mechanical Engineering Design*, Macmillan, London, (1966).
Dobrovolsky V., et al., *Machine Elements*, Foreign Languages Publishing House, Moscow.
Mahadevan K., et al, *Design Data Handbook*, Rangs, Bombay, (1972).
Hesse, H. G. and Rushton, J.H., *Process Equipment Design*, Affiliated East-West Press, New Delhi, (1967).

CHAPTER 6

Pressure Vessels

6.1 Introduction

Several types of equipment which are used in the chemical industry have an unfired pressure vessel as a basic component. Such units are storage vessels, kettles, distillation columns, heat exchangers, evaporators, autoclaves, etc. Each one of these is covered in detail in subsequent chapters. The general design procedure applicable to the main parts of the pressure vessel is considered in this chapter.

6.2 Operating Conditions

Pressure vessels are usually spherical or cylindrical with domed ends. They are provided with openings or nozzles with facilities for making threaded or flanged joints. Various methods are used for supporting the vessel. The operating conditions, may be specified as those resulting from the operation during maximum or normal conditions, as well as those that exist during starting up or shutting down or during change in loading. The latter are known as transient conditions.

6.2.1 Normal Conditions

These include the following (a) Operating pressure, (internal or external) existing during normal operation. The maximum pressure is generally not more than 10 per cent in excess of the normal value.

(b) Operating temperature is decided by the contained fluid. The maximum and the minimum temperature have to be specified.

(c) Influence of environment, including possible corrosion or chemical attack from the fluid contained and from the atmosphere. Similarly effects of erosion caused by high velocity of flow and effects of irradiation have to be considered.

(d) External loading such as wind and snow. Other external loadings are those resulting from the reaction of piping systems attached to pressure vessels, dead weight of agitator system, pumps, valves, etc., supported by the vessel and in general all forms of local loading imposed during service.

6.2.2 Transient Conditions

These may be repetitive, for example, those occurring during starting up and shutting down. It is necessary to know the anticipated modes of operation, including rates of change of fluid temperature, procedure for starting up and shutting down and finally possible emergency operation, and loads due to earthquakes.

6.3 Pressure Vessel Code

A number of National Codes which specify requirements of design, fabrication, inspection and testing of unfired pressure vessels are available (Table 6.1). The Indian Standards Institute has prepared a similar code I.S—2825. In most countries the National Codes have the force of law and strict adherence to their rules is required. It is therefore, desirable that any design, regardless of the method adopted, should be checked with standard code.

6.4 Selection of Material

Pressure vessels form a major part of the equipment used in the chemical industry. It is therefore, desirable to consider the suitability of the different materials for construction of pressure vessels operating under different conditions. Such conditions are temperatures in the range of 600°C to −200°C, pressures in range of vacuum conditions to as high as 3,000 kg/cm^2, corrosive effects due to acid and alkalis, steady or cyclic

Table 6.1

Principal National and International Codes

Country	Code Title	Scope
Australia	Standards Association of Australia Boiler Code, Parts I-V	B UFPV
Austria	Dampfkessel Veror dnung (DKV) RGBI No. 83/1948	B
	Werkstoff und Bauvorschriften (WBV) RGBI No. 264/1949	PV
Canada	C.S.A. Standard B 51-1957 incorporating A.S.M.E. Rules	B UFPV
Finland	Dimensioning, Materials and Welding of Steel Pressure Vessels	B UFPV
France	SNCT No. 1	UFPV
	Reglementation des appareils a vapeur et a pression de gaz	Government Rules not strictly forming a Design Code
Germany	Workstoff und Bauvorschriften fur Dampfkessel und Dampfkessel Bestimmungen AD—Merkblatter	B UFPV
	DIN-2413	Pipes
Holland	Grondslagen waarop de becoordeling van de constructie en het materiaal van stoom-toestelen, damptoestelen en druckhoudersberust	B UFPV
India	Indian Boiler Regulations 1950	B
	Code for Unfired Pressure Vessels IS-2825-1969	UFPV
	Specification for formed ends for tanks and pressure vessels IS—4049 1971	UFPV
	Specifications for shell flange for Vessels & Equipment IS—4864-4870—1968	UFPV
	Manholes & inspection openings for chemical equipment IS—3133	UFPV
	Code of Practice for Design Fabrication & Erection of vertical mild steel cylindrical welded oil storage tanks IS—803—1962	Tanks
	Specification for Shell & Tube type Heat Exchangers IS—4503	Heat Exchanger
	Specification for mild steel Tube IS-1239—(1968)	Tube

Country	Code Title	Scope
Italy	Controllo della combustione	B
	Aparecchi a Pressione	UFPV
New Zealand	N.Z. Boiler Code	B
	N.Z. Pressure Vessel Code	UFPV
Sweden	Tryckkarlsnormer	B UFPV
	Angpanneformer Pannsvetnormer	B Weld Code
Switzerland	Regulations of the Swiss Association of Boiler Properietors	B UFPV
Britain	Lloyd's Rules	B UFPV piping
	Rules of the Associated Office Technical Committee (AOTC)	B UFPV
	B.S. 1500 : 1958 Pt. I	UFPV
	B.S. 1515 : 1965 Pt. I	
	B.S. 1113 : 1958	B
	B.S. 806 : 1954	Pipes
	B.S. 1306 : 1955	Power
	B.S. 3351 : 1961	Pipes (oil)
	B.S. 2971 : 1961	
	B.S. 2654 : 1956 Pt. I : 1962 Pt. II	Vertical tanks
	B.S. 3274 : 1960	Heat exchangers
	B.S. 3915 : 1965	Nuclear vessels
U.S.A.	A.S.M.E. Codes :	B
	Pt. I—Boilers	UFPV
	Pt. II—Materials	
	Pt. III—Nuclear vessels	
	Pt. VIII	UFPV
	Pt. IX	Welding
	Tentative Structural Basis for Reactor Pressure Vessels and Directly Associated Components	UFPV Nuclear
	ASA— B31, 1-8-63	Piping
	API—A.S.M.E. (similar to A.S.M.E. Codes)	
	TEMA, Tubular Heat Exchangers Manufacturers Association, 1959	Heat exchangers

B—Boiler UFPV—Unified pressure vessel.

loading etc. (See Appendix E for strength properties). Apart from the mechanical properties and corrosion resistance of the material, fabrication problems, commercial availability of the material and the cost will have to be critically assessed in the final selection of the material. A survey of the materials of construction and the methods of protective coatings are covered in Chapter 2. A review of the important materials accepted for construction of pressure vessels is indicated here. Metallic materials may be divided into three groups:

(a) *Low cost*—Cast iron, cast carbon and low alloy steel, wrought carbon and low alloy steel.

(b) *Medium cost*—High alloy steel (12% Cr and upwards) aluminium, copper, nickel and their alloys, lead.

(c) *High cost*—Platinum, silver, tantalum, titanium, zirconium.

The materials in the second and third groups can be used in the form of cladding or bonding for materials in the first group. Similarly non-metallic linings such as rubber, plastic, etc., may also be used.

6.4.1 STEEL

This is the most versatile and most widely used material of construction in the pressure vessel industry. The carbon content in pressure vessel steels is usually limited to 0.3%. Low carbon or mild steel is the cheapest and most commonly used amongst the pressure vessel steels. Depending on the degree of deoxidation, a steel may be rimmed, semi-killed or killed. Rimmed steels are seldom used in pressure vessel construction, due to their lack of chemical homogeneity. They may be used for light duty vessels. Semi-killed steels are the cheapest steels used for general purpose light duty service. Almost all the plates used in the pressure vessels, up to 2.5 cm thickness for this type of service are semi-killed. Fully deoxidised silicon killed steels are more homogeneous. They are more expensive and are used for thicker vessels, or in all thicknesses for severe duty. These steels have 0.2% carbon and 0.7 to 0.9% manganese. Aluminium is usually added as grain refiner, to improve notch toughness of the material. Mild steel is generally used in normalised condition. This condition may be achieved by selecting the appropriate temperature and cooling rate during rolling and forming. Mild steel is readily attacked by most fluids and atmospheric environments. The corrosion resistance of the material depends to a large extent on the cleanliness and homogeneity of the steel and the fabrication process.

Low alloy steels with a carbon content of the order of 0.15%, manganese 1.0%, silicon about 0.3% and other alloy elements such as chromium (0.5 to 5%), molybdenum upto 0.5% are used for high temperature service and under mild conditions.

Carbon-molybdenum steel is widely used in the petroleum and petrochemical industry for hydrogen resistant applications. The most important pressure service is for steam drums particularly for large diameters and high pressure. The steels have 1 to 1.5% manganese with small additions of Cr, Mo and/or Ni. High strength low alloy steels are also used for thick wall pressure vessels in the chemical and petroleum industry when the wall thickness is in the region of 7.5 cm or more. Quenched and tempered carbon and low alloy steels are used for pressure vessel construction when high strength is required. In the case of carbon steel the brittle transition temperature is reduced due to quenching and tempering.

Low alloy steels, developed for high temperature application have a lower elongation to rupture than ordinary mild steels. This means that cracks may initiate during pressure test at the points where there are high, local stresses. It is important to ensure that the maximum stress in low alloy pressure vessels do not exceed the yield point during the pressure test to any significant extent.

High alloy steels are used when it is necessary to have a corrosion resistance material and whenever contamination to the fluid by dissolved corrosion products or by the material itself has to be avoided. High alloy vessels are mainly used for severe duty. Such vessels can only be designed on the basis of a comprehensive knowledge of the material properties and the behaviour of the material during fabrication.

For pressure vessel construction high alloy steels used, include ferrite steels containing 13, 17 and 27% chromium and

the austenitic steels of the 18/8, 25/12, 25/20 and 32/22 cromium-nickel composition. The straight cromium ferrite steels are generally specified for corrosion resistant duties in the form of clad plate. Only 13% Cr steel is employed for refinery vessels, while 17% Cr and 27% Cr steel are rarely employed. Austenitic chromium nickel steels are used for subzero temperatures, for corrosion resistance and for high temperature operation.

6.4.2 Aluminium and Aluminium Alloys

Aluminium alloys have been developed with mechanical properties comparable to those of mild steel. They retain their ductility at sub-zero temperatures, and are easy to fabricate. Aluminium and aluminium alloys are used for handling most organic solvents, foodstuffs, hydrogen peroxide, nitric acid, sea water and inorganic salts. This material is seldom used for vessels operating above 150°C. In all cases care must be taken to prevent any contact between aluminium and nobler metals such as copper and iron. The major disadvantages of aluminium are low strength and low melting point.

6.4.3 Copper and Copper Alloys

These are used in food processing plants, in the manufacture and recovery of organic solvents and in heat exchanger and evaporators for general purposes. Selection of various alloys is based on corrosion requirements, strength and weldability.

6.4.4 Nickel and Nickel Alloys

These are used for handling a variety of corrosive fluids at low or elevated temperatures and where it is essential to prevent the contamination of the contained fluid. These materials are very expensive. They are therefore, used primarily as cladding materials in pressure vessel construction. The only exceptions are heat-exchanger tubes and tube plates and small thin-walled vessels.

6.4.5 Titanium

The material is used as bonded titanium clad steel for pressure vessels.

6.5 Vessels Operating at Low Temperatures

The ductility of some metals, including carbon and low alloy steels is significantly diminished when the operating temperature is reduced below some critical value. The critical temperature, commonly described as the transition temperature, depends upon the material, method of manufacture, previous treatment and kind of stress system present. Fracture occurs at temperatures above the transition temperature only after considerable plastic strain or deformation, while below transition temperature, fracture may take place in a brittle manner with little or no deformation. Brittle fractures are likely to be extensive and may lead to catastrophic fragmentation of a vessel.

Construction features producing a notch effect or sudden change of section are particularly objectionable in vessels designed for low temperature operation, since they may create a state of stress such that the material will be incapable of relaxing high localised stresses by plastic deformation. For this reason materials for low temperature service are tested for notch ductility.

Carbon steels are used down to −60°C. Control of notch ductility in such materials is obtained through proper composition, steel making practice, heat treatment and fabrication practice. They have low carbon content with increased manganese carbon ratio. Various additions are made to promote fine grain size and improve notch ductility, the most effective of which is aluminium. Notch ductility can also be improved by normalising. Between −50°C to −100°C low alloy steels are used. 1% Cr, 0.25% Mo steel and nickel steels containing between 1% to 5% have been specified for this range. Below −100°C, the choice is made from among austenitic chromium-nickel steel, 9% Ni steel, aluminium or copper. All these materials are notch tough down to the lowest temperature. Tables 6.2 and 6.3 give the materials recommended at different temperatures.

Table 6.2

Selection of Material for Non-corrosive Service at Sub-zero and Atmospheric Temperature : Low Values of Pressure × Diameter (other than Refrigeration Equipment)

Temperature	Material selection	Alternatives
Below −100°C	Aluminium, 9% Ni steel Austenitic chromium-nickel steel	
−100°C to −50°C	3.5% Ni steel	5% Ni is usable down to −120°C 2.5% Ni down to −60°C
−50°C to 0°C	Impact tested carbon steel	Carbon steel may be acceptable without impact test : e.g., down to −20°F In such cases use finegrain steel as a minimum. It is normal practice to impact test at or below the design temperature
0°C to 20°C	Thin plate (say below 12 mm): normal boiler quality. Thicker plate: killed steel.	More conservatively, use fine grain steel for the thicker plate.

Table 6.3

Selection of Material for Non-corrosive Service at Sub-zero and Atmospheric Temperature : Higher Values of Pressure × Diameter

Temperature	Material selection	Alternatives
Below −100°C	9% Ni steel	Substitution of austenitic Cr-Ni steel may be necessary for some items
−100°C to −50°C	3.5% Ni steel	9% Ni steel may be considered.
−50°C to 0°C	Impact tested, fine grain carbon-manganese steel	
0°C to 20°C	Fine grain killed steel	Impact tested semi-killed fine grain steel

6.6 Vessels Operating at Elevated Temperature

There are two main criteria in selecting a steel for elevated temperature use, namely, strength and metallurgical stability. All carbon steels are reduced in their strength properties due to rise in temperature and are liable to creep. Their use is therefore, generally limited to 500°C. High alloy steels can be used for higher temperature upto 1000°C with proper preconditioning. The other major requirement for satisfactory use at elevated temperature is that the metal should not deteriorate significantly in service. Metallurgical faults must be avoided and corrosion rates must be controlled. Above 600°C, carbon steel is oxidised at a very high rate both in air and steam. Chromium in amounts of 3% and over significantly reduces corrosion rates and all heat resisting steels contain chromium for this reason. It is better to use austenitic chromium-nickel steel for unstressed parts operating in the range 500 to 600°C. For higher stressed parts stabilised grades of this steel are preferable due to their high tensile properties at higher temperature.

Embrittlement of carbon and alloy steel, may occur due to service at elevated temperature. In most instances brittleness is manifest only when the material is cooled to room temperature. Embrittlement is inhibited by addition of molybdenum and also improves tensile and creep properties. Tables 6.4 and 6.5 give materials recommended at elevated temperatures.

Table 6.4

Selection of Material for Non-corrosive Service at Elevated Temperature: Low Values of Pressure × Diameter

Temperature	Material selection	Alternatives
20°—450°C	Semi-killed carbon steel	Silicon-killed steel
450°—500°C	Silicon-killed steel	Low alloy steel, 1 Cr 0.5 Mo
500°—600°C	1 Cr 0.5 Mo	2.25 Cr 1 Mo
Above 600°C	Refractory-lined carbon or low alloy steel	Austenitic chromium-nickel steels

Table 6.5

Selection of Material for Non-corrosive Service at Elevated Temperature: Higher Values of Pressure × Diameter

Temperature	Material selection	Alternatives
20°—450°C	Silicon-killed carbon-manganese steel For thick plate (say over 7.5 cm) specify impact testing	Proprietary high tensile alloy steel
450°—600°C	2.25 Cr 1 Mo steel	1 Cr 0.5 Mo. Proprietary steels
Above 600°C	Refractory-lined carbon or low alloy steel	Austenitic chromium nickel steels

6.7 Design Conditions and Stresses

The design conditions for pressure vessels are specified primarily in terms of pressure and temperature. Vessels are protected with safety devices such as safety valves or bursting discs (see 17.5). The design pressure is therefore, based on the setting pressure of the safety devices, which is generally taken as equal to the maximum operating pressure or working pressure, including the static head plus 10%. Vessels subjected to vacuum and not provided with vacuum breaker valves are designed for 1 atmosphere gauge external pressure. When both internal and external pressures are operating, design is based on each pressure separately and choosing the safe value. The design temperature is generally taken as equal to the actual component wall temperature.

6.7.1 Nominal Design Stresses

These are based on the ultimate tensile stress, the yield stress or the 0.2% proof stress or creep properties. The design stress is finally obtained by using a factor of safety. According to the basicsaccepted by different codes the design stress might vary. This may be observed from Table 6.6.

Table 6.6

Safety Factor by Different Codes for Allowable Design Stresses

National Code	Country	a (UTS)	b (Y.S.)	c S_R	d S_C
ASME VIII Div. 1—'68	USA	4.0	1.6	...	1.0
ASME VIII Div. 2—'68	USA	3.0	1.5	...	...
BS—1500	UK	4.0	...	...	...
BS—1515 Part I—'65	UK	2.35	1.5	1.5	1.0
BS—1515 Part II—'65	UK	2.5	1.5	1.5	1.0
AD Merkblatter (ADM)	FR of Germany		1.5	...	1.5
IS-2825 : 1969	India CS and Low alloy	Not governing for standard material 3.0	1.5	1.5	1.0
	Non-ferrous	4.0		1.0	
RAP de Gaz	France	3.0	...	...	...
ANCC code	Italy	...	1.5	1.5	...
Pressure Vessel code	Japan		1.7	...	...
ISO/TC 11	International	2.4	1.5	1.6	1.0

Key
a = factor of safety on UTS at room temperature or design temperature
b = factor of safety on YS
c = factor of safety on S_R
d = factor of safety on S_C
S_R = Average stress of rupture after 100,000 hours at operating temperature (kg/cm^2)
S_C = Average stress for 1% creep in 100,000 hours at operating temperature (kg/cm^2)

Design stresses, values at different temperatures for various ferrous and non-ferrous metals and alloys as suggested by I.S. are given in the Appendix

Design codes usually give rules for the design of vessel shell areas and components under steady pressure. In addition it is necessary to assess the effects of other loads, such as support forces and temperature differences, some of which will be constant in the service life and others which may be cyclic. With increase in vessel size and duty these loads may be more significant. In many cases the codes recommend relatively low design stress to cover approximately the effectso f additional loads thus simplifying the design procedure. The present trend is to improve and rationalise component design and the codes are being revised accordingly.

6.7.2 Design Criteria

The methods of design are primarily based on elastic analysis. There are however other criteria such as stresses in the plastic region, fatigue, creep, etc., which sometimes need consideration. Elastic analysis has been developed on the assumption that the material is isotropic and homogeneous and that it is loaded in the elastic range. In the plastic range elastic analysis ceases to be applicable. However, elastic analysis can still be used if the plastic flow is limited to small regions in the neighbourhood of structural discontinuities or other stress raisers, since the deformation of such regions is forced by that of adjoining elastic material. Similarly under cyclic variations of load, causing plastic flow the material hardens and the behaviour of the material becomes purely elastic. This is the phenomenon of shakedown or cessation of plastic deformation under cyclic loading. Elastic analysis therefore, is the most important method of designing pressure vessel shells and components. Beyond the elastic limit, the material yields and the plastic region spreads without increased value of load. The value of the load for which this occurs is called collapse load or bursting pressure. Limit analysis is concerned with calculating the load or pressure at which flow of structure material occurs due to yielding. This method of analysis is seldom applied to the design of pressure vessels. When vessels are subject to cyclic loading, it is necessary to consider the requirements for elastic cycling of the material and the implications of this on component behaviour. In the case of a discontinuity of shape the load may give rise to a plastic yield limited to a small area. This will be under elastic and plastic cycling. Under these conditions a shakedown will occur. The maximum shakedown load is twice the first yield load. Elastic analysis is therefore, valid upto this range of load, under cyclic loading conditions. The fatigue strength is assessed by means of curves representing the stress range *verses* full reversal cycles to failure. A factor of safety of two is applied on the stress or a factor of safety of twenty is applied on the number of cycles. A design stress is accepted as the lower value.

A low-stress brittle fracture of a pressure vessel is possible as a result of the following factors:

(a) presence of a notch
(b) high localized stresses in the vicinity of a notch
(c) operation at a sufficiently low temperature
(d) wrong selection or treatment of material.

To guard against this contingency, the presence of a notch acting as a severe stress raiser or in a highly stressed region, at low temperatures, must be avoided. In addition an appropriate material must be selected for the specified service conditions and must not suffer any unexpected damage during fabrication or in operation. At elevated temperatures, in the so-called creep range for metals, creep may precipitate plastic instability or it may cause unacceptable deformations or ruptures. In such cases it is necessary to introduce new design stresses based on creep rather than on short time properties. Two criteria are usually specified, one for rupture and the other for deformation. According to the rupture criterion, the design stress is defined as 0.6 to 1.0 times of minimum or average stress for rupture after 100,000 hours at the design temperature. In accordance with the deformation criterion, the design stress is usually defined as the average stress for 1% creep strain after 100,000 hours, or 0.01% per 1000 hour creep strain rate. In case of carbon steels the design stresses can be based solely on short time properties upto a design temperature of 350°C,

while a knowledge of the creep behaviour of the material is required for design temperature above 400°C. These temperature limits are raised by about 75°C for alloy steels.

6.7.3 Corrosion Allowance

Corrosion of chemical equipment due to environmental conditions is a common feature. Various factors causing corrosion and the methods used to prevent it are indicated in Chapters 2 and 4. Every attempt should be made to avoid corrosion. However, this may not be always possible. In pressure vessels corrosion rates may be predictable, unpredictable or negligible. An allowance is therefore, made to allow for expected corrosion, by providing additional material thickness over and above the thickness determined from design conditions. For carbon steel and cast iron parts, the corrosion allowance is 1.5 mm on all parts except tubes. In the case of those chemical industries where severe conditions are expected the corrosion allowance may be 3 mm. For high alloy steels and non-ferrous parts under pressure no corrosion allowance is necessary. Similarly when the thickness is more than 30 mm corrosion allowance is not necessary.

6.8 Design of Shell and its Components

The simplest pressure vessel considered here is a single unit when fabricated. However, for convenience of design it is divided into following parts: (See Plate 1)

(1) Shell (Plate 1)
(2) Head or cover
(3) Nozzle
(4) Flanged joint
(5) Support.

Most of the above components are fabricated from sheets and plates. Seamless or welded pipes can also be used. Parts of vessel are formed and connected by welded or riveted joints. In designing these parts and connections between them, it is essential to take into account the efficiency of the joints. For welded joints, the efficiency may be taken as 100% if the joint is fully checked by radiograph. It is taken as 85% if it is checked at only a few spots. Efficiencies between 50 to 85% are taken when radiographic test is not carried out. In the case of riveted joints efficiencies vary between 70 to 85% for butt joints and 50 to 70% for lap joints. (For details see 5.9.6, 5.10.3 and tables 5.4 and 5.5). The pressure vessels considered here are those operating at pressures less than 200 kg/cm². Vessels for higher pressures are considered in Chapter 12. The design procedure is primarily based on fabrication by welding. However, specific features of riveted construction are indicated in 5.9.4.

6.8.1 CYLINDRICAL AND SPHERICAL SHELLS

The following design procedure is applicable for shells having the ratio of outside diameter to inside diameter not exceeding 1.5.

6.8.1.1 SHELLS SUBJECT TO INTERNAL PRESSURE

(a) *Cylindrical shell*—The internal pressure in the shell gives rise to stresses in the shell thickness, one in the circumferential and the other in the longitudinal direction. These are given by (Equations 3.28 and 3.29).

$$\text{Circumferential stress, } f_p = \frac{pD}{2t} \qquad (6.1)$$

$$\text{Longitudinal or Axial stress } f_a = \frac{pD}{4t} \qquad (6.2)$$

where

p—internal pressure

D—mean diameter of the shell.

Both the above stresses are tensile. Since the circumferential stress is greater, this is taken as the design stress. The shell is generally formed by a joint in the longitudinal direction, which is considered in terms of joint efficiency. The thickness of the shell is therefore, given by

$$t = \frac{pD}{2fJ} = \frac{pD_i}{2fJ-p} = \frac{pD_o}{2fJ+p} \qquad (6.3)$$

where

p—design pressure

D—mean diameter

J—joint efficiency

f—design or permissible stress at design temperature

D_i—internal diameter

D_o—outside diameter.

Values of the design or permissible stresses for different materials are given in Appendix E.

(b) *Spherical shell*—In this case both stresses are equal to the longitudinal stress indicated above. The thickness is therefore, given by equation (3.30)

$$t = \frac{pD}{4fJ} = \frac{pD_i}{4fJ-p} = \frac{pD_o}{4fJ+p} \qquad (6.4)$$

6.8.1.2 CYLINDRICAL VESSEL UNDER COMBINED LOADINGS

In addition to the internal pressure, the other loadings are the weight of the vessel with its contents and the wind. The effect of offset piping can also be taken into account. The stresses created due to each of the above loading can be stated as :

(1) Stress in the circumferential direction due to internal pressure as per equation (6.1). This is also known as tangential or hoop stress.

$$f_t = \frac{p(D_i+t)}{2t} \text{ (tensile)} \qquad (6.5)$$

where t—thickness of shell calculated from equation (6.3).

(2) Stresses in the longitudinal or axial direction

(a) due to internal pressure as per equation (6.2)

$$f_1 = \frac{pD_i}{4t} \text{ (tensile)} \qquad (6.6)$$

(b) due to weight of vessel and contents (vertical vessels only)

$$f_2 = \frac{W}{\pi t(D_i+t)} \text{ (compressive)} \qquad (6.7)$$

where

W—weight of vessel and contents

(c) due to wind or piping in the case of vertical vessels or due to weight of vessel in the case of horizontal vessels

$$f_3 = \pm \frac{M}{Z} = \frac{M}{\pi D_i^2 t} \quad \text{(Tensile or compressive)} \tag{6.8}$$

where

M—bending moment, due to loads normal to the vessel axis (calculations of bending due to wind load are given in Chapter 13).

Z—modulus of section of the cylindrical vessel.

Total stress in the longitudinal or axial direction

$$f_a = f_1 + f_2 + f_3 \quad \text{(Tensile or compressive)}, \tag{6.9}$$

(3) stress due to offset piping or wind

$$f_s = \frac{T}{\pi t D_i (D_i + t)} \tag{6.10}$$

where

T—torque about the vessel axis.

Combining the above stresses on the basis of *shear strain* energy theory criterion (see equation 4.9)

The equivalent stress

$$f_R = [(f_t^2 - f_t f_a + f_a^2 + 3 f_s^2)]^{\frac{1}{2}} \tag{6.11}$$

For satisfactory design, the following conditions must be satisfied.

f_R (tensile) $\leqslant f_t$ (permissible)
f_a (tensile) $\leqslant f_t$ (permissible)
f_a (compressive) $\leqslant f_c$ (permissible)

The axial compressive stress may cause wrinkling of the shell. The safe compressive stress which can be imposed, without failure by wrinkling is given by (Section 4.4.)

$$f_c \text{ (permissible)} = \frac{1}{12} \cdot \frac{E}{\sqrt{3(1-\mu^2)}} \cdot \frac{t}{(D_o/2)} \tag{6.12}$$

where

E — modulus of elasticity
μ — Poisson's ratio
D_o — outside diameter of vessel.

If these conditions are not satisfied, the thickness t, will have to be increased, and the solution by trial and error is necessary.

The final thickness shall be determined by addition of corrosion allowance.

In the case of tall vertical shells, all the above stresses have to be considered. But in addition stresses resulting from seismic forces are also to be considered. The design procedure is indicated in Chapters 11 and 13.

The design procedure outlined above does not take into consideration certain loads such as those caused due to supports, eccentricity of loading, temperature, etc.

6.8.1.3 SHELLS SUBJECTED TO EXTERNAL PRESSURE

(a) *Cylindrical shell*—Elastic buckling (Section 4.4) is usually the decisive criterion in the design of vessels operating under external pressure or vacuum. The critical buckling pressure for cylindrical vessels under external pressure is given by

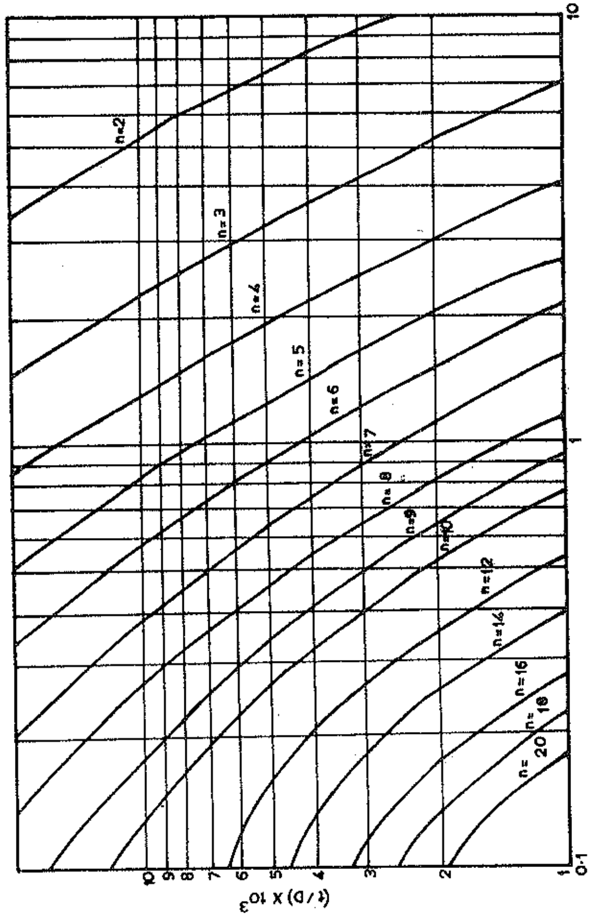
$$p_c = \frac{1}{3} \quad n^2 - 1 + \frac{2n^2 - 1 - \mu}{n^2 \left(\frac{2t}{\pi D_o}\right)^2 - 1} \quad \frac{2E}{(1-\mu^2)} \left(\frac{t}{D_o}\right)^3 + \frac{2E \frac{t}{D_o}}{(n^2-1)\left[n^2 \left(\frac{2L}{\pi D_o}\right)^2 + 1\right]^2} \tag{6.13}$$

where

L — unsupported length of the vessel
D_o— outside diameter
t — thickness
E — modulus of elasticity
μ — Poisson's ratio
n — number of lobes formed at buckling (from Fig. 6.1)

In cases of shells, where the ends are closed the critical buckling pressure in given by

$$p_c = \left\{ \frac{1}{3} \left[n^2 + \left(\frac{\pi D_o}{2L}\right)^2 \right]^2 \frac{2E}{1-\mu^2} \left(\frac{t}{D_o}\right)^3 + \frac{2E\left(\frac{t}{D_o}\right)}{\left[n^2 \left(\frac{2L}{\pi D_o}\right)^2 + 1\right]^2} \right\} \times \frac{1}{(n^2 + \frac{1}{2})\left(\frac{\pi D_o}{2L}\right)^2} \tag{6.14 a}$$



6.1 Number of lobes formed at buckling

An approximate equation is given by

$$p_c = \frac{2.42\,E}{(1-\mu^2)^{\frac{3}{4}}}\;\frac{\left(\dfrac{t}{D_o}\right)^{5/2}}{\left(\dfrac{L}{D_o}\right) - 0.45\left(\dfrac{t}{D_o}\right)^{1/2}} \tag{6.14b}$$

The distance L is measured as shown in Fig. (6.2a). The other terms have the same meanings as in equation (6.13)

In practice, cylindrical vessels do not have a perfectly circular cross-section and it becomes necessary to calculate the reduction in the critical buckling pressure caused by this out of roundness in order to specify manufacturing tolerances and choose a safety factor. As a rough guide, if the maximum deviation from the true circular shape is equal to shell thickness, the

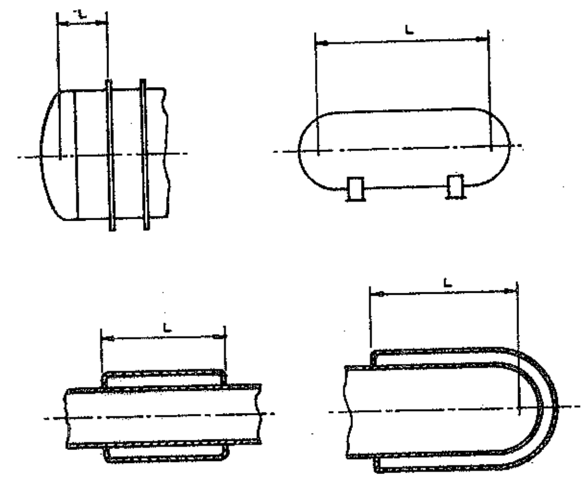


Fig. 6.2(a) Unsupported length of cylindrical vessel

critical buckling pressure will be less than 50% of the calculated value for the perfect shell. If it is one-tenth of the shell thickness, the reduction of critical buckling pressure is not more than 25%.

Considerable savings in weight and material can be made by use of reinforcing rings or stiffening rings attached to the inside or outside surface of the shell. These rings extend over the whole circumference and serve the same purpose as end supports. T-beams, flat plate rings welded with a continuous or intermittant seam are usually preferred as being simple, light and effective. I-beams, U-channels or angles bolted, riveted or welded to shell may also be used (Fig. 6.2b). In these

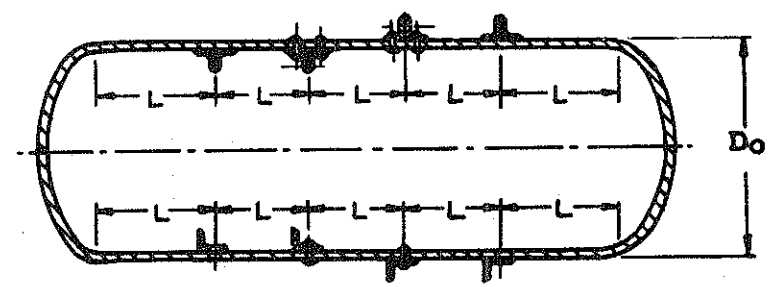


Fig. 6.2(b) Attachment of different types of reinforcing rings

cases the length L, is taken as distance between stiffening rings. The required moment of inertia of the ring is given by

$$I = \frac{p_c D_o^3 L}{24E} \tag{6.15}$$

where

E — modulus of elasticity

D_o— outside diameter of shell.

The value of I, may be reduced by about 30% to take into account the resistance of the shell. From the value of the critical pressure p_c, the critical compressive stress can be determined as

$$f_c = \frac{p_c D}{2t} \tag{6.16}$$

The permissible values of p_c and f_c can be worked out by using a factor of safety of 4.

It may be seen from equation (6.13) or (6.14) that to calculate the value of p_c, it is necessary to know the thickness 't'. To compute the critical pressure it is therefore, necessary to select values of t/D and L/D. The working pressure is calculated with a factor of safety of 4. A trial and error procedure is essential to arrive at the requisite working pressure. Finally, it is necessary to check the maximum stress which should not exceed the permissible value based on the ultimate stress or yield stress.

Indian Standard (IS-2825 Appendix F) gives a procedure for calculating the allowable working external pressure based on the ratios $\frac{L}{D_o}$ and $\frac{D_o}{t}$. Graphs have been supplied for different materials.

In addition to external pressure the cylindrical shell may have to withstand axial loads, as well as wind loads. These may be considered as combined loadings and an equivalent stress may be determined from equation (6.11). The thickness 't' of the shell may be finally calculated by trial and error to satisfy all the stress conditions. A suitable corrosion allowance may be added if required.

(b) *Spherical shell*—The critical buckling pressure is given by

$$p_c = \frac{8E}{\sqrt{3(1-\mu^2)}} \left(\frac{t}{D}\right)^2 \tag{6.17}$$

and the corresponding critical stress is

$$f_c = \frac{2E}{\sqrt{3(1-\mu^2)}} \left(\frac{t}{D}\right) \tag{6.18}$$

The permissible value is obtained by assuming a factor of safety of 6.

The design procedure is to select a value of $\frac{t}{D}$ and obtain the critical pressure and stress. These must satisfy the requisite external pressure and the permissible stress. A corrosion allowance may be added if necessary.

Indian Standard (IS 2825—Appendix F) gives a procedure for calculating the allowable external working pressure.

6.8.2. Head or Cover

To close either end of the cylindrical shell a cover or a closure is essential. This can be attached to the shell by welded riveted or bolted construction. The simplest cover is a flat plate of the same diameter as the shell. This plate can be firmly secured to the shell by a suitable method and must be capable of withstanding the working pressure. The more common types of closures are formed to a specific shape and are generally known as formed heads. The most economically formed head is that produced by simply forming a straight flange with a radius on a flat plate [Fig. 6.3(a)]. These heads are mainly used for horizontal cylindrical storage vessels at atmospheric pressure. The flat portion of such heads can be dished to form what are known as flanged and dished heads (Fig. 6.3). Depending on the depth of dishing and its shape these are classified as shallow dished, flared and dished, torispherical, elliptical, hemispherical and conical. Flared and shallow dished heads are used for low pressure vessels. Torispherical heads are used in the pressure range between 1 to 15 kg per cm² gauge. Elliptical heads are preferable for pressures over 15 kg per cm². Hemispherical heads are the strongest of the formed heads, but are expensive and are therefore, used only in few cases. Conical heads are widely used as bottom heads for a variety of equipment, where removal or draining of the material is facilitated. Cones having an angle of apex of 60° are commonly used for the removal of solids.

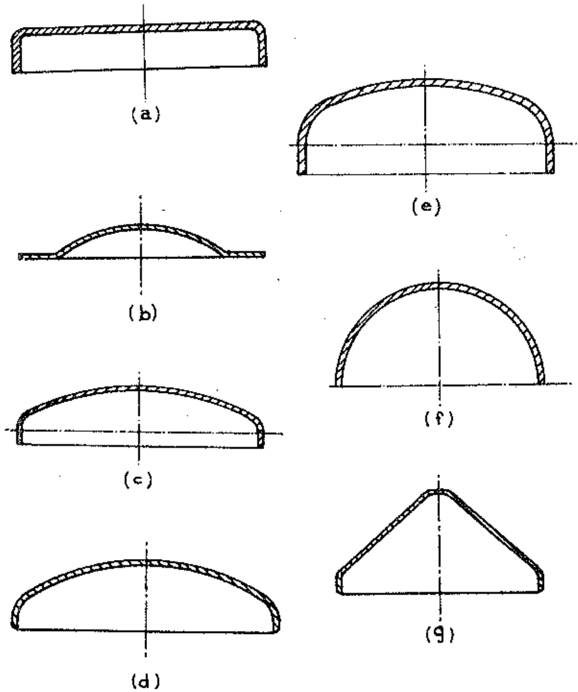


Fig. 6.3 Formed heads
(a) Plain formed head
(b) Flared and dished head
(c) Shallow dished head
(d) Torispherical dished head
(e) Elliptical head
(f) Hemispherical head
(g) Conical head

6.8.2.1 FLAT HEADS AND COVERS

Flat closures are often used as manhole covers in low-pressure vessels and small bore openings. Their use is generally limited to light duty service, since they are uneconomical for other applications. The stresses created in a flat plate, due to pressure, acting as a uniformly distributed load and with fixed edges are indicated under Section 3.9. The maximum stress is at the circumference and is given by equation (3.22).

$$f=\frac{3}{4}\,p\left(\frac{R}{t}\right)^2=\frac{3}{16}\,p\left(\frac{D}{t}\right)^2 \tag{6.19}$$

The thickness of the plate can therefore, be calculated as

$$t_h=CD\sqrt{\frac{p}{f}} \tag{6.20}$$

where

C—edge fixity constant
D—diameter of the plate which is actually under operating pressure
p—working pressure
f—design stress at operating temperature
R—$D/2$

According to the method of attachment of the flat head to the shell the edge fixity changes. This is taken into account by the factor C, which varies from 0.4 to 0.7 (Fig. 6.4).

For the same working pressure and design stress, the thickness of a flat plate is several times the thickness of a shell. Further due to the joint or junction between the shell and the plate, severe localised stresses are created in the wall of the shell surrounding the junction. These are due to a sharp discontinuity in the shape at the junction. The substantial difference between the magnitude and direction of the deformation or dilations between the flat plate and the shell under pressure, and which are contained by the rigid junction results in localised shear and bending stresses. In low pressure service, (below approximately 1.5 kg per cm^2) the magnitude of these stresses is small and can therefore, be ignored. For small diameter vessels slightly higher pressures may be possible. In general for higher pressures it is preferable to use flanged and dished heads. Such heads are formed with a gradual discontinuity in shape, to facilitate a smooth junction with the shell, resulting in a reduction of localised stresses at or near the junction.

6.8.2.2 FORMED AND DISHED HEADS

For a large number of cylindrical shells, dished heads with torispherical or elliptical shapes are commonly used. The hemispherical dished head is used in a few cases, due to the excessive dish forming required. The thickness of these heads is calculated on the basis of the circumferential stresses, created either due to internal pressure (i.e., acting on the

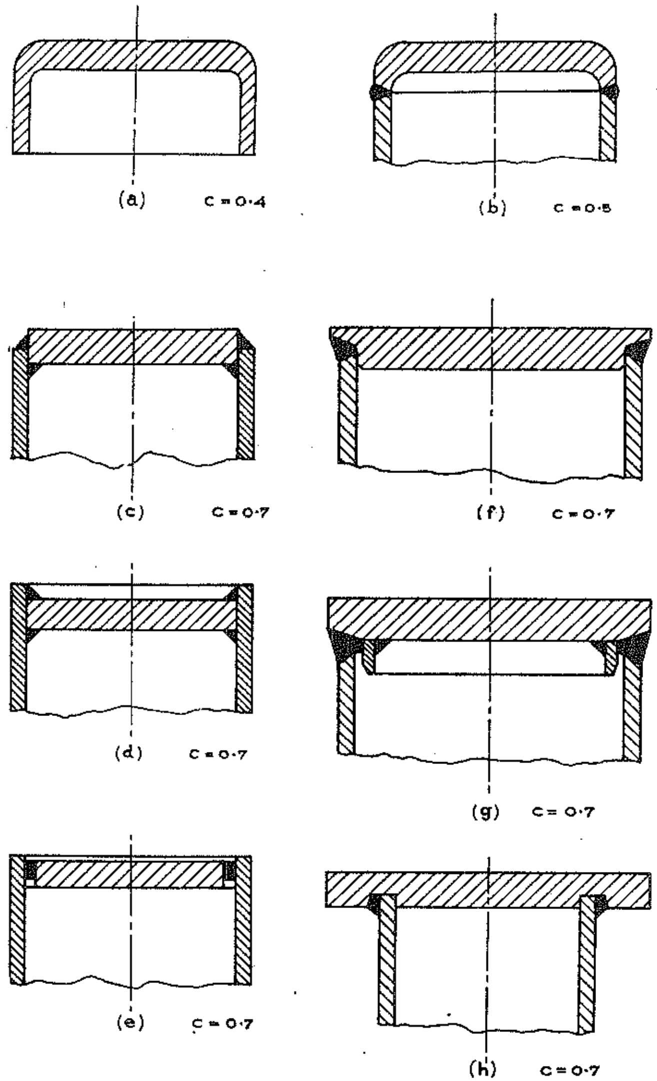


Fig. 6.4 Attachment of flat heads or plates to shell
(a) forged head
(b) welded head
(c), (d), (e) internal plates
(f), (g), (h) external plates

concave side) or external pressure (i.e., acting on the convex side).

Indian Standard 4049 gives specification for formed heads for tanks and pressure vessels. These include different sizes of semi-ellipsoidal and torispherical heads for pressure vessels, tanks and non-pressure vessels. The length of the straight flange (Sf) on the head shall not be less than three times the head thickness with a minimum of 20 mm.

(a) *Internal pressure*

(a-i) *Conical Head* (Fig. 6.3g) : The circumferential stress in this type of formed head is given by

$$f = \frac{p\,D}{2t\,\cos\alpha} \tag{6.21}$$

where

α—half apex angle of the cone
p—design pressure

the thickness t_h is therefore, given by

$$t_h = \frac{pD}{2fJ\cos\alpha} \tag{6.22}$$

At the junction of the conical head and the cylindrical shell, localised stresses are produced, which are likely to exceed the circumferential stress. In order to reduce the intensity of such localised stresses in the case of conical heads where angle exceeds 30°, provision has to be made to attach it to the shell, by a knuckle having an inside radius preferably not less than 10 % of the internal diameter of the shell. In some cases instead of providing a knuckle a compression ring welded to the shell at the junction may be acceptable.

(a-ii) *Shallow dished and torispherical head* (Fig. 6.3c and d) : This head has the dish radius or crown radius equal to or less than the diameter of the head. To reduce the stresses at corner of the head, a knuckle is formed, with a radius of at least 6% of the inside diameter. The thickness of such heads is given by

$$t_h = \frac{pR_cW}{2fJ} \tag{6.23}$$

where

p—internal design pressure
R_c—crown radius

W—stress intensification factor

$$= \frac{1}{4}\left(3+\sqrt{\frac{R_c}{R_1}}\right)$$

R_1—knuckle radius

(a-iii) *Elliptical head* (*semi-ellipsoidal*) (Fig. 6.3c): The thickness of such heads is given by

$$t_h = \frac{p\,D\,V}{2\,f\,J} \tag{6.24}$$

where

p—internal design pressure
D—major axis of ellipse
V—stress intensification factor$=\frac{1}{6}(2-k^2)$

$$k = \frac{\text{major axis}}{\text{minor axis}}$$

A common value of k is 2. This should not be greater than 2.6.

(a-iv) *Hemispherical head* (Fig. 6.3(f)): The thickness of a hemispherical head is given by

$$t_h = \frac{p\,D}{4\,f\,J} \tag{6.25}$$

(b) *External pressure*

(b-i) *Conical head*: Depending on the apex angle these heads are designed either as a cylindrical shell under external pressure or a flat plate. If the apex angle is less than 45°, the conical head is designed as a cylindrical shell with the same diameter as the large end of the cone and a length equal to the axial length of the cone. Heads with apex angle between 45° to 120° are designed, as shell except that the diameter at large end of the cone is also taken as length of an equivalent cylinder. Heads with apex angles greater than 120°, are designed as a flat plate having a diameter equal to the largest diameter of the cone.

(b-ii) *Torispherical, elliptical and hemispherical heads*: These heads are designed for external pressure, with the application of the same equations, as those indicated for internal pressure, except that the internal pressure in these equations, is to be taken as 1.67 times the external pressure.

In addition these heads are designed on the basis of prevention of collapse due to buckling. They should have sufficient elastic stability. Assuming a suitable factor of safety for the buckling pressure, the thickness is given by

$$t_h = 4.4\,R_c\,\sqrt[4]{3(1-\mu^2)}\;\sqrt{p/2E} \tag{6.26}$$

where

p—design external pressure
R_c—crown radius for torispherical and hemispherical heads and equivalent crown radius for elliptical head
E—modulus of elasticity
μ—Poisson's ratio

In all the above cases of heads or covers, a corrosion allowance may be added to the thickness as required by the rate of corrosion.

6.8.2.3 ATTACHMENT OF HEAD AND SHELL

Heads are attached to the shell either by a riveted, welded or flanged joint. In the case of riveted or welded joint, either a lap or butt construction is made according to the thickness of the shell and head. Fig. 6.5 shows welded joints. Flanged

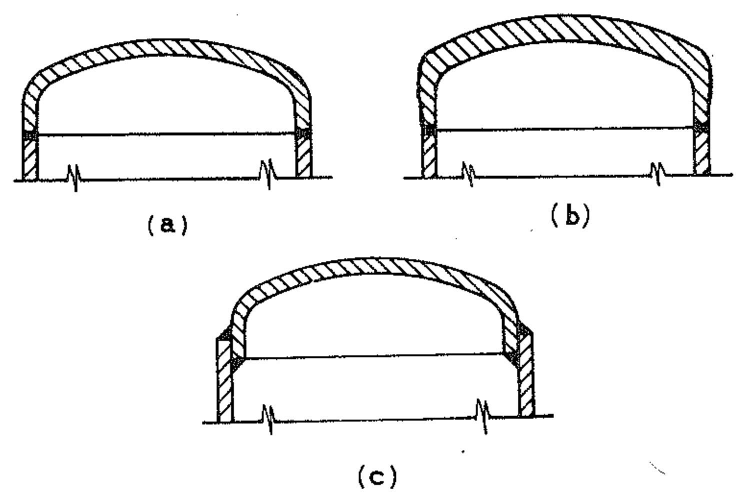


Fig. 6.5 Attachment of formed heads to shell
(a) butt welded
(b) butt welded with unequal thickness of shell and head
(c) lap welded

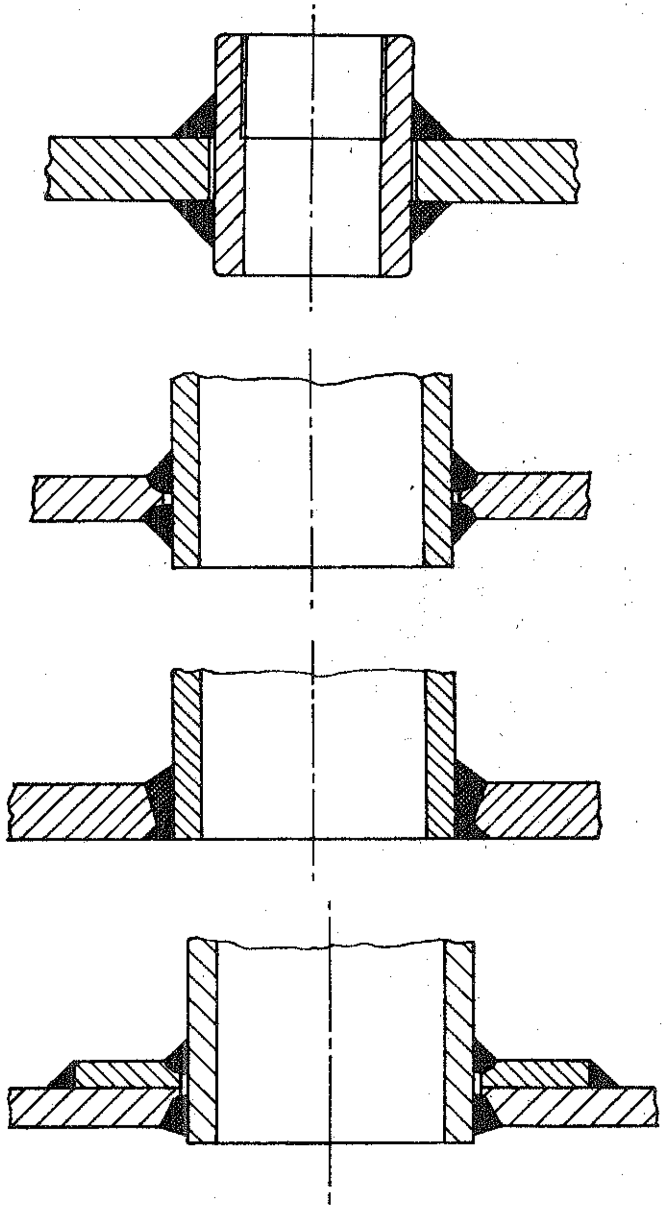


Fig. 6.6 Attachment of pipes to vessel wall

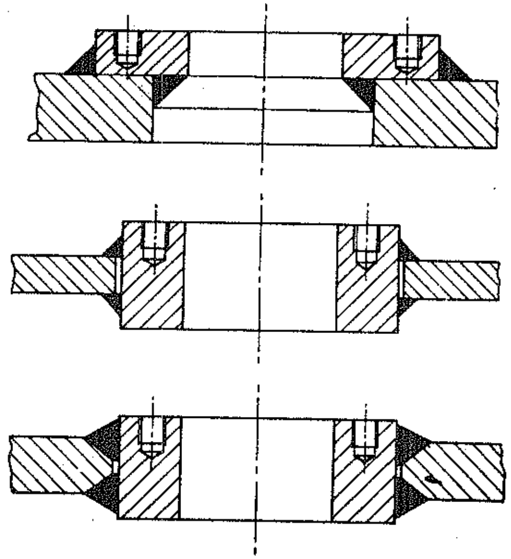


Fig. 6.7 Attachment of pads to vessel wall

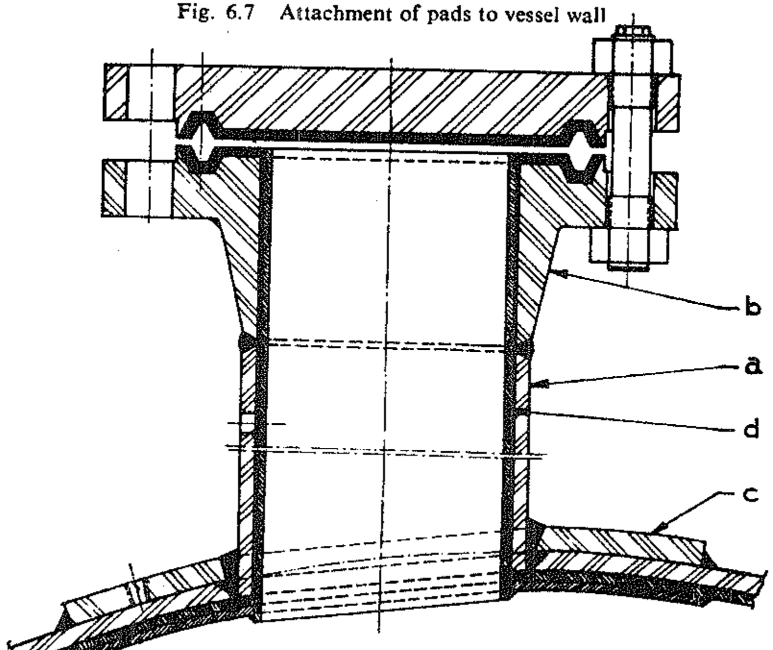


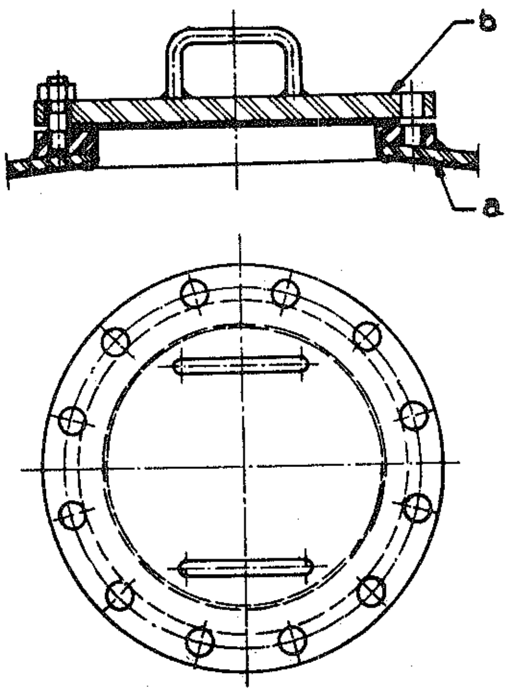
Fig. 6.8 Extended nozzle with flanged joint
(a) pipe (b) welding neck flange (c) reinforcement (d) lining

joints are discussed under 6.8.4. These are detachable unlike the welded or riveted joints which are permanent.

6.8.3 NOZZLES

Nozzles or openings are provided in the pressure vessels, to satisfy certain requirements such as inlet or outlet connections, manholes, handholes, vents and drains, etc. These may be located on the shell or head according to functional requirements, and are circular, elliptical or obround (with a ratio of major to minor inside diameter not exceeding 2). On the basis of the method of forming and attachment, nozzles can be classified as (a) integral, (b) fabricated and (c) formed.

(a) Integral nozzles are fabricated from a portion of the shell or head, by cutting and shaping the material to obtain the contour of the nozzle. This operation can be satisfactorily



6.9 Manhole
(a) vessel wall (b) cover

carried out only in cases where the material has sufficient ductility. These nozzles are therefore, adopted only in a few cases.

(b) Fabricated nozzles are essentially short pieces of pipes, tubes and plates. They are cut to a specific length from

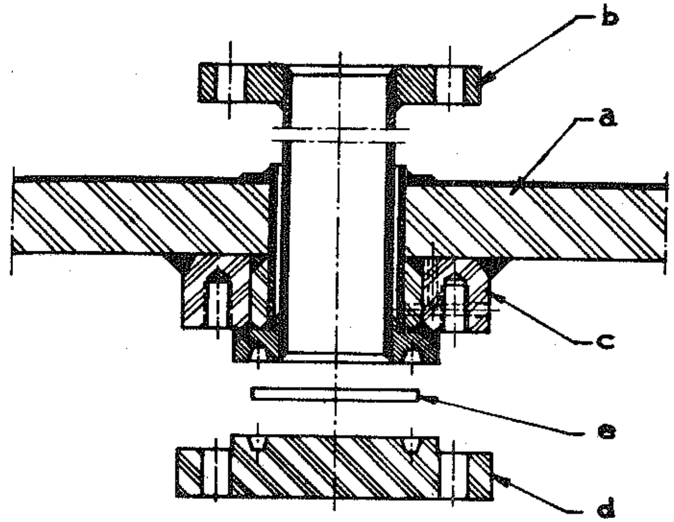


Fig. 6.10 Nozzle with flanges at either end
(a) vessel wall (b) flange (c) reinforcement (d) blank flange (e) gasket

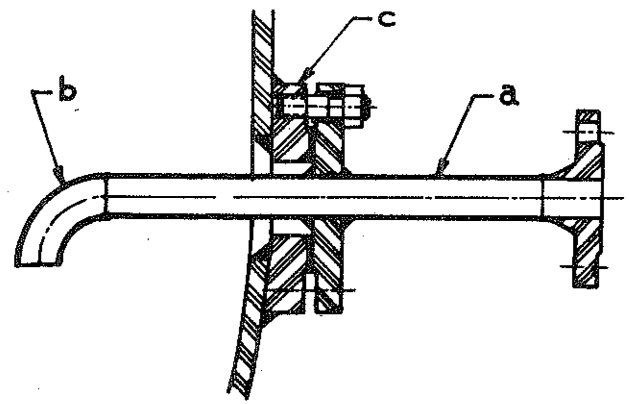


Fig. 6.11 An extended pipe through a nozzle
(a) pipe (b) bend (c) pad for nozzle

standard fittings and are welded to the vessel wall at an opening made in the wall. Such nozzles can be conveniently and satisfactorily used for relatively small openings and for low and medium pressures. Figs. 6.9 to 6.15 show different types of fabricated nozzles welded to shell or head. Fig. 6.6 shows a few methods of welding a piece of pipe to the vessel wall, while Fig. 6.7 shows three methods of welding a pad to the vessel wall. The pads are provided with tapped holes for attachment to other components. Fig. 6.8 shows an extended nozzle welded to flanges and lined with a corrosion resistant material. Fig. 6.9 shows a manhole formed out of a pad with a cover attached by studs. Fig. 6.10 shows a nozzle with flanges at either end. The nozzle is free from the vessel wall. A cover is also shown. Fig. 6.11 shows a pipe length extending inside a vessel through a nozzle. At one end of the pipe a bend is welded while at the other end a flange is welded. The arrangement facilitates the removal of the pipe from the vessel. Fig. 6.12

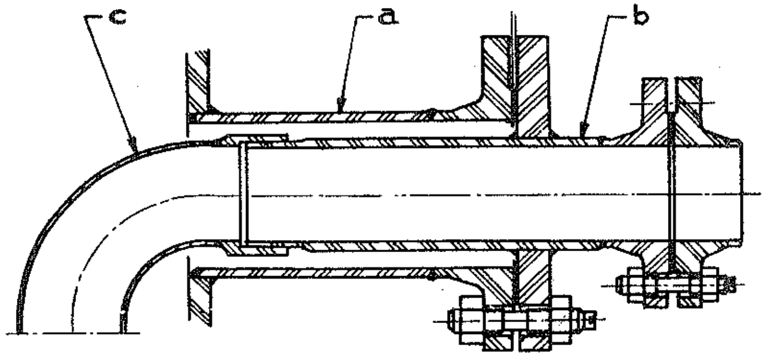


Fig. 6.12 A pipe through an extended nozzle
(a) extended nozzle (b) pipe (c) bend

shows a similar arrangement with an extended nozzle and the bend which is screwed. Figs. 6.13 and 6.14 show arrangements of fixing a sight glass. To prevent the fog formation on the surface of the glass, two glass pieces are provided as shown in Fig. 6.14. Between the two glass pieces electrical heating is provided. Fig. 6.15 shows a light glass. As may be seen from the above figures, it is possible to provide nozzles in various ways to satisfy certain functional requirements. These nozzles

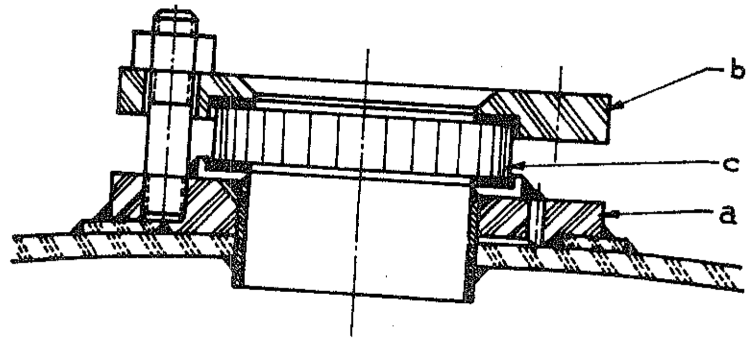


Fig. 6.13 Sight glass
(a) pad (b) cover (c) glass

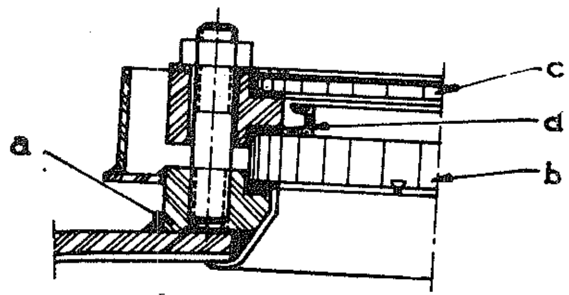


Fig. 6.14 Heated sight glass
(a) pad (b) sight glass (c) auxiliary glass (d) heater

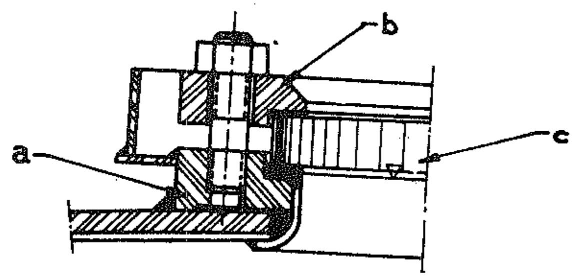


Fig. 6.15 Light glass
(a) pad (b) cover (c) light glass

may be connected to a pipe or valve by use of threads or flanges and bolts.

(c) Formed nozzles are fabricated to a specific shape and size, preferably with flanges. These are formed by rolling or forging and are attached to the vessel at an opening made in the wall. Standard formed nozzles of various sizes are manufactured to withstand different temperatures and pressures ratings. IS 3133 gives specification for manhole and inspection openings for chemical equipment. Fig. 6.16 shows different types of formed nozzles welded to shell or head.

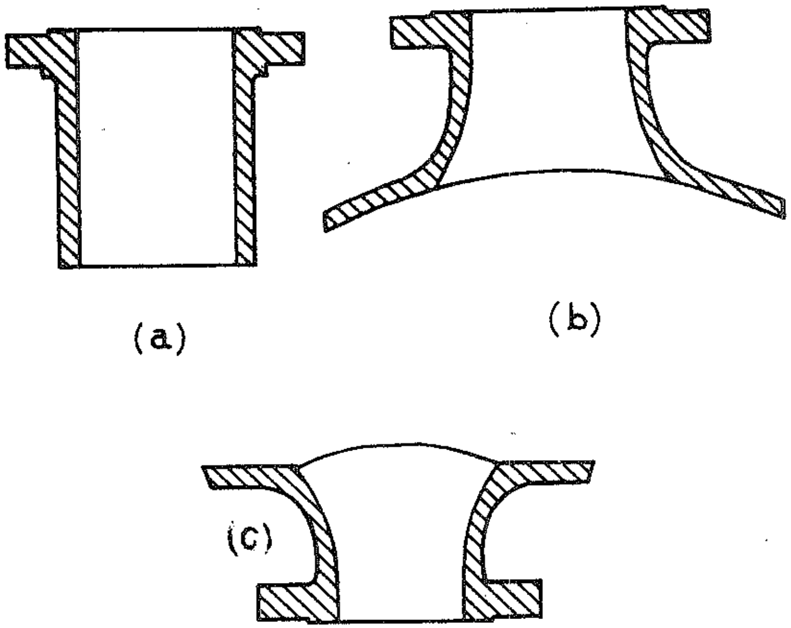


Fig. 6.16 Formed nozzles
(a) welding neck (b) and (c) sweep type

6.8.3.1 STRESSES AND REINFORCEMENT

Openings in vessels are made first by making holes in the wall of the vessel. Nozzles are then formed or welded around these holes. Holes cause a discontinuity in the vessel wall. Due to the pressure or other loads a stress concentration is created

near the holes the maximum value of the stress being at the edge of the holes. In the case of a circular hole in a cylinder with internal or external pressure, the stress at the hole is about 2.5 times the circumferential stress developed in the shell. In the case of elliptical holes, the maximum stress varies according to the direction of the axis, and also the ratio of the major to minor axis of the ellipse.

The stress concentration at the hole can be reduced by increasing the thickness of the vessel in the vicinity of the nozzle. This can be done either by providing additional thickness to the vessel wall itself near the nozzle or by use of a separate reinforcing plate attached to the vessel wall covering an area surrounding the hole. Sometimes the nozzle wall at the base can be made sufficiently thick to act as a reinforcement. In all such cases the reinforcement is provided with a view to satisfy the following requirements:

(a) The weakening due to the hole made for the nozzle has to be compensated by sufficient additional material. However, this should not disturb the general strain pattern prevailing in the wall of the vessel.

(b) The reinforcing material should be placed immediately adjacent to the hole, but suitably disposed in profile and contour so as not to introduce any stress concentration.

For a circular hole, considering the variation of stress in an area surrounding the hole, and the deflection characteristics of the nozzle, it has been observed that the satisfactory extent of reinforcement is upto a radius of four times the radius of the hole.

Fig. 6.17 shows the reinforcement boundary limits viz., *ABCD*. The hole area *EFGH* removed from the vessel wall is compensated partly by shaded nozzle area and additionally by compensating reinforcing material, located within the boundary *ABCD*. When the reinforcing material is placed within the above boundary, maximum reinforcement is obtained with minimum material.

The reinforcing material can be placed (a) on both sides of the vessel wall (balanced) [Fig. 6.18 (a)]. (b) on the inside or

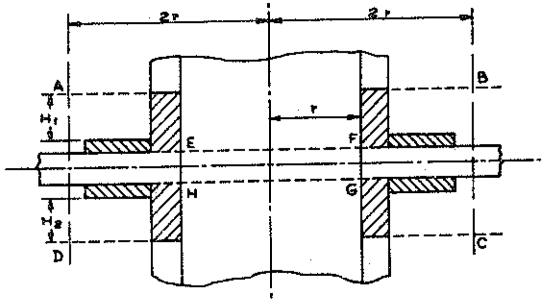


Fig. 6.17 Reinforcing pad limits for nozzle

outside surfaces of the vessel wall (unbalanced) [Fig. 6.18 (b) and (c)]. In the first case the reinforcement is balanced and therefore, prevents creation of local bending moments and stresses. In the second case the reinforcement is unbalanced.

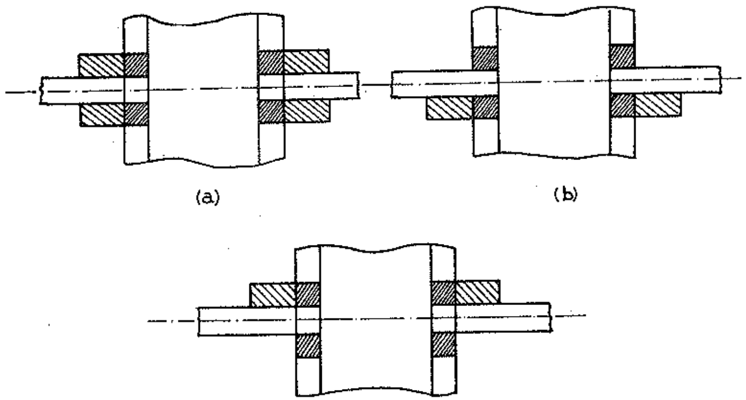


Fig. 6.18 Balanced and unbalanced reinforcement
(a) internal and external reinforcement
(b) internal reinforcement
(c) external reinforcement

Reinforcement on the inside surface is preferable to the outside surface, in the second case.

6.8.3.2 DESIGN OF REINFORCEMENT

There are three methods of designing reinforcement for a nozzle.

(a) *Area for area method.*—This method has been suggested in codes. It has proved both simple in application and essentially reliable in service.

(b) *Controlled maximum stress method.*—In this method the maximum stress in the vessel adjacent to the nozzle is restricted to a fixed multiple of the design stress. This multiple is usually 2.25.

(c) *Experimental yield method.*—This method is based on pressure tests for a series of cylinder/cylinder junctions, and the pressure ($P_{0\cdot2}$) to produce 0.2% residual strain at the junctions are used to derive 'weakening factors'. The design pressure equals $P_{0\cdot2}$. 1.5, so that the nozzle has the same reserve on yield as the membrane areas where the design stress equals the yield stress divided by 1.5.

(a) *Area for area method of compensation (Internal Pressure)*

The area to be replaced and the effective area of compensation are shown in Fig. 6.17 by the limits of *ABCD*.

The maximum horizontal distance $AB=4r=2d$

The maximum vertical distance $AD=6t_s$

or $(3.5t_s+2.5\ t_n)$

whichever is smaller.

If the compensation is only provided by the nozzle then

$$H_1=H_2=2.5\ t_s$$

If the compensation is to be provided by a combination of nozzle and a compensation ring, then

$$H_1=2.5t_n$$

The area for which compensation is required is given by

$$A=d\ t_s$$

Area available for compensation

(*i*) The portion of the shell or head as excess thickness

$$A_s=d\ (t_s-t'_s-C)$$

(*ii*) The portion of the nozzle external to the vessel

$$A_0=2H_1(f_n-t'_n-C)$$

(*iii*) The portion of the nozzle inside the vessel

$$A_1=2H_2\ (t_n-2C)$$

Where the calculated area A is greater than $(A_s+A_o+A_i)$, additional compensation equal to

$$A-(A_s+A_o+A_i),$$

is to be provided by use of a compensation ring welded to the nozzle and shell as shown in Fig. 6.17.

where

r=inner radius of the nozzle

d=inner diameter of nozzle+2 corrosion allowance

t_s=actual thickness of the shell or head

t_s'=theoretical minimum thickness of the shell or head calculated with suitable, joint efficiency (equations 6.3, 6.23, 6.24)

C=corrosion allowance

t_n=actual thickness of the nozzle

t_n'=theoretical minimum thickness of the nozzle calculated with suitable joint efficiency (equation 6.3)

H_1=height of effective compensation in branch wall external to vessel measured from outside surface of compensation ring or vessel wall

H_2=height of any portion of the branch pipe projecting inside vessel and effective as compensation, measured from the inside surface of vessel or compensation ring.

If the permissible stress for the compensating ring material is different from that of the shell material, the compensation provided by the ring is to be modified accordingly.

Area of compensation (External pressure)

This is only 50% of that required with internal pressure, where t_s is the required minimum thickness under external pressure.

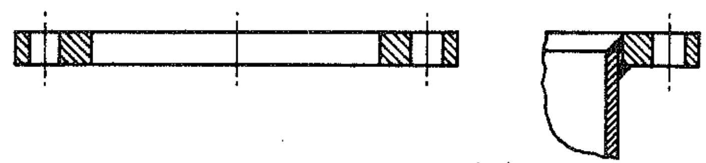
6.8.4 Flange Joints

When heads of pressure vessels are to be of the detachable type, they must be provided with flanges, with arrangement for bolting. Similarly, nozzles and piping systems may also be provided with flanges. The use of such bolted-flanged connections which are readily removable, facilitate disassembly of different components.

A flanged joint consists of a pair of flanges, one each attached to the two components to be joined, held securely together by a series of bolts or studs. A gasket is interposed between the two adjoining flange faces. A variety of flanges, gaskets and bolts are available for this purpose. The joint must have structural integrity with negligible leakage during service.

6.8.4.1 Flanges

Different types and sizes of flanges are fabricated either by casting, forging or formed from structural sections and plates. Flanges used for connections between shell sections and between the shell and the head are standardised (IS-4864 to 4869). These are made of carbon steel and are welded to carbon steel or stainless steel pressure vessel shells. According to the type of flange and working temperature, these flanges can be used upto 25 kg/cm^2 pressure. Steel and cast iron flanges used for piping joints are also standardised (IS-6392 and IS-6418). Pipe flanges are subject to bending and other stresses due to piping connection, unlike the shell flanges which are free from such external bending moments. Fig. 6.19 shows different types of standard flanges. Non-standard shell flanges can be formed from a structural section such as an angle iron or from a plate. These can be strengthened by gusset plates and may



(a)

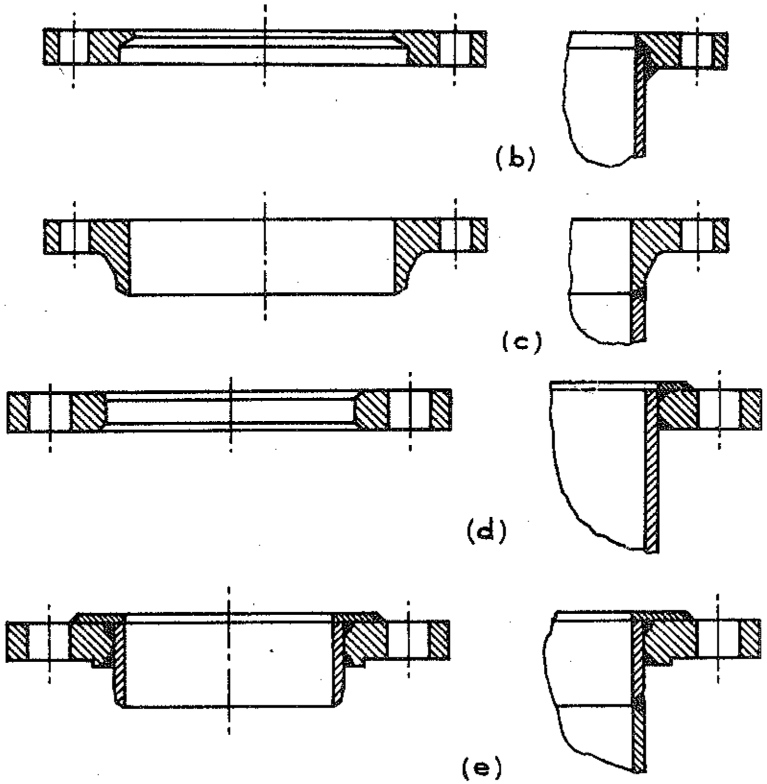
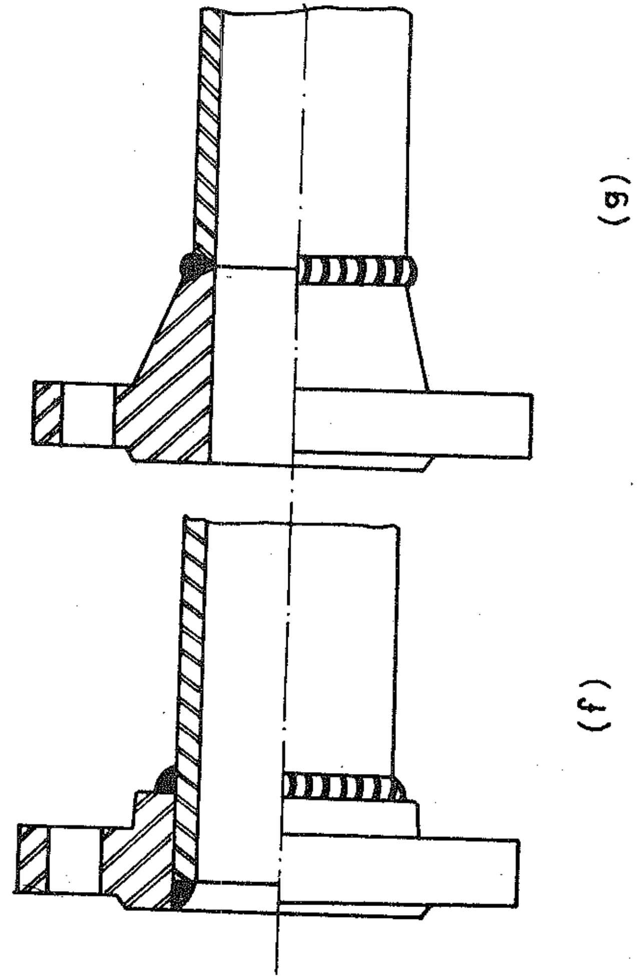
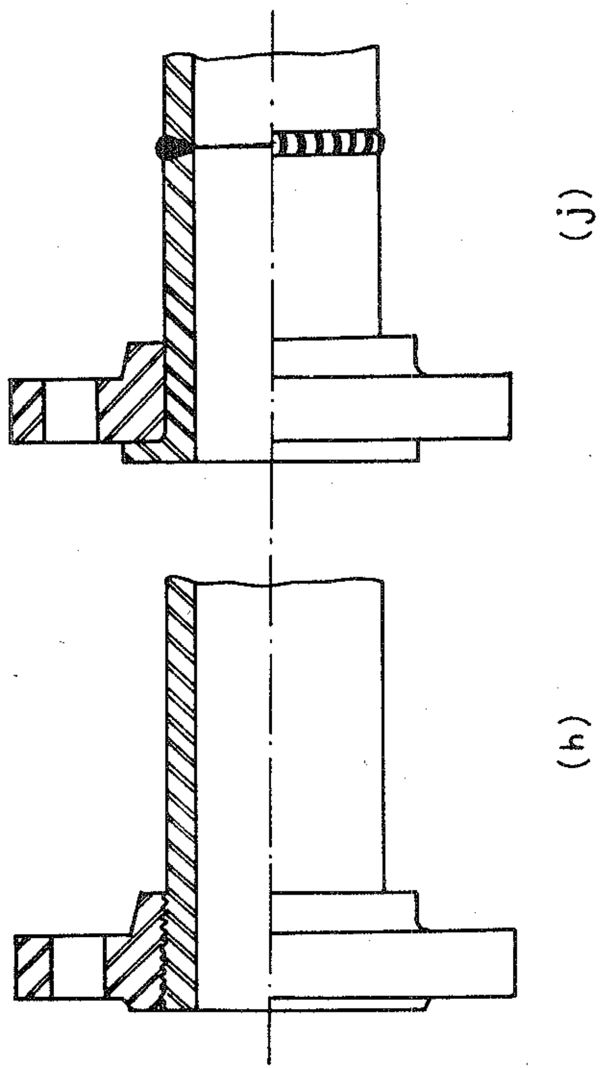


Fig. 6.19 Standard flanges Shell flanges
(a) ring flange for non-pressure service
(b) ring flange for carbon steel pressure vessels
(c) welded neck flange for carbon steel pressure vessels
(d) ring flange (with stainless steel lining) for stainless steel pressure vessels
(e) hub type flange (with stainless steel lining) for stainless steel pressure vessels

be used for low and medium pressure applications. Fig. 6.20 shows two flanges one fabricated out of an angle iron (shell flange) and the other from a plate (cover flange). The vessel shell and its conical head are made of special material such as copper, while the flanges are made of carbon steel. The arrangement is used for low pressure. A nozzle is also shown at the top of the head. Fig. 6.21 shows a flared and





Pipe flanges : (f) hub type flange ; (g) welding neck flange ;
(h) screwed flange ; (j) lap type flange

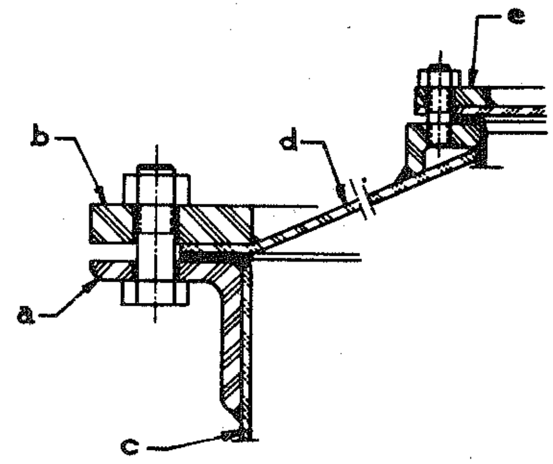


Fig. 6.20 Non-standard flanges
(a) angle ring flange for shell (b) ring flange for head (c) shell
(d) conical head (e) nozzle

dished head attached to the shell. The flange of the shell is fabricated from a plate and is strengthened by gusset plates.

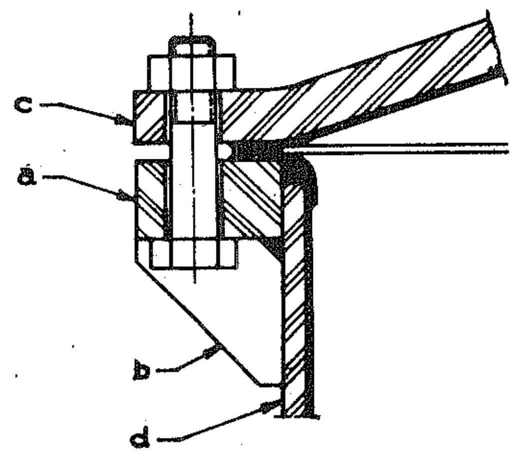


Fig. 6.21 Non-standard flanges
(a) flange for shell (b) gusset plate (c) flared and dished head (d) shell

Fig. 6.22 shows a similar arrangement except that the flange portion of the head is strengthened by an additional ring. A

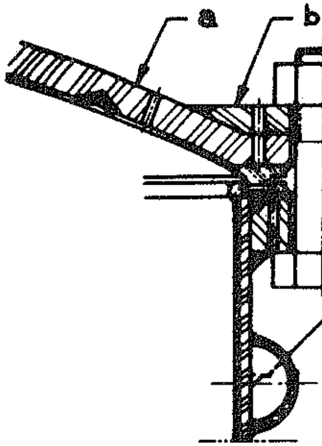


Fig. 6.22 Non-standard flanges
(a) flared and dished head (b) stiffening ring

tongue and groove facing is provided. Fig. 6.23 shows flanges made out of carbon steel plates, but lined with special steel for

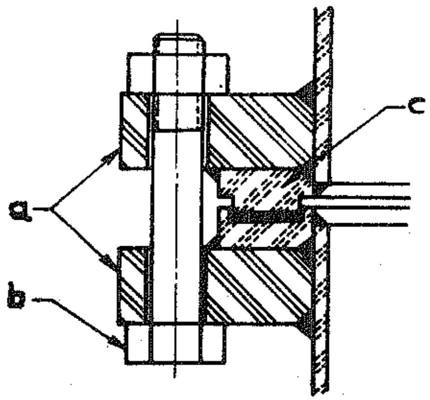


Fig. 6.23 Ring flanges lined with special steel
(a) flanges (b) bolt (c) linging

the facing. The vessel is also made of special steel. Fig. 6.24 shows a similar arrangement of flanges, except that the shell and the head are provided with an outside ring, thereby increasing the thickness of the portion adjacent to the flange.

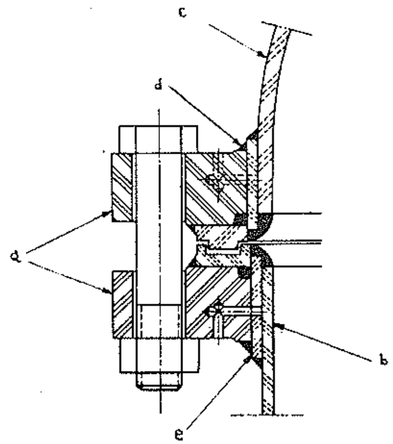


Fig. 6.24 Ring flanges lined with special steel
(a) flanges (b) shell (c) head (d) ring for head (e) ring for shell

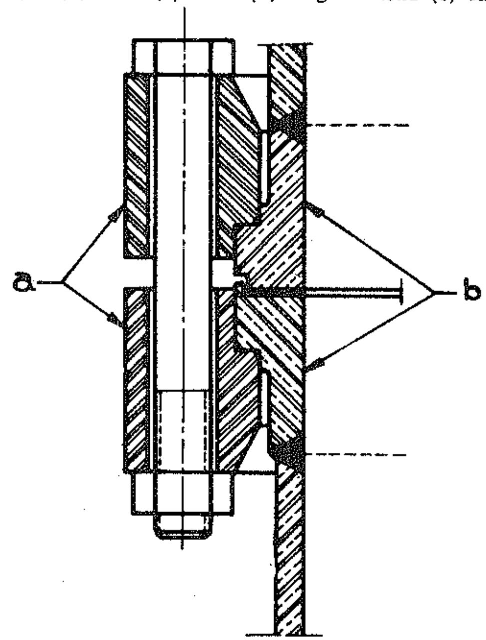


Fig. 6.25 Loose flanges
(a) flanges (b) shell and head rings

This method is adopted when the vessel thickness is not enough to carry out a satisfactory welding of the carbon steel flanges to the shell or head. Fig. 6.25 shows loose flanges held tight by bolts. These flanges are made to rest on the rings welded to shell and head. The flange facing for the gasket is provided on the ring. Fig. 6.26 shows special hub type flanges and stud

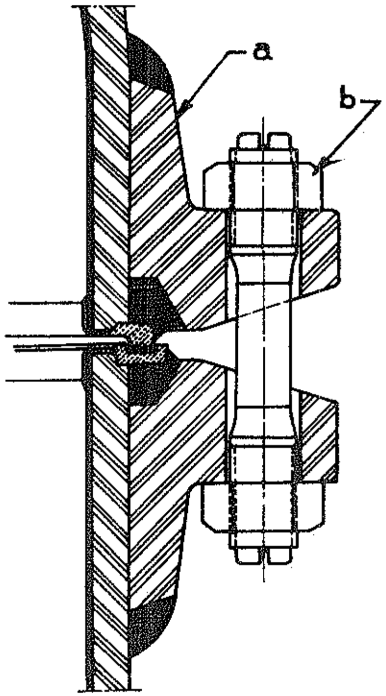


Fig. 6.26 Special flanges with studs
(a) welding neck flange (b) stud with nut

with locking arrangement. This arrangement is particularly used in cases where the vessel has to sustain continuous vibrations, probably from certain rotating elements. Fig. 6.27 is an arrangement which combines the essential features of the earlier figures, but in addition it provides a jacket with a flange. Figs. 6.28 and 6.29 show ring type flanges with eye bolts, which are hinged. This arrangement is used where the cover

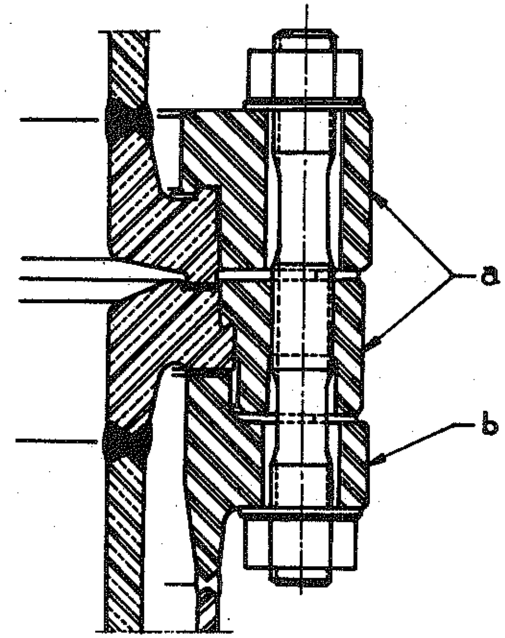


Fig. 6.27 Loose flanges with jacket flange
(a) loose flanges (b) jacket flange

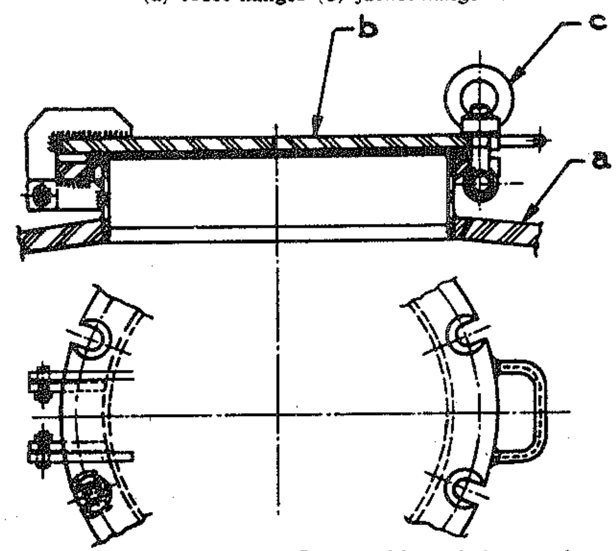


Fig. 6.28 Ring flanges with eye bolts
(a) vessel wall (b) hinged cover (c) eye bolt

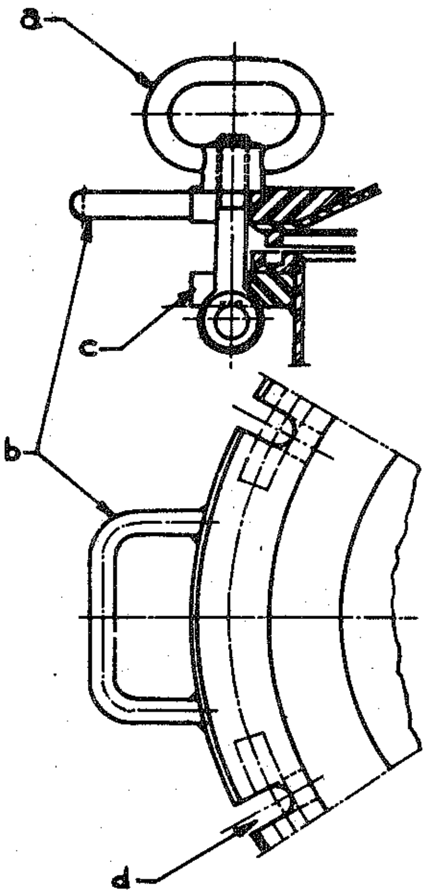


Fig. 6.29 Ring flanges with lining and with eye bolts
(a) eye bolt (b) handle for cover flange (c) shell flange (d) slot for eye bolt

is to be dismantled frequently. Fig. 6.30 shows special flanges connected by swing bolts and clamps, instead of normal bolts and nuts. No holes are made in the flange. The clamp can be either hinged or free. The arrangement facilitates dismantling.

6.8.4.2 FLANGE FACINGS

In order to locate the gasket in an appropriate position on the flange faces, these faces are formed to specific shapes and

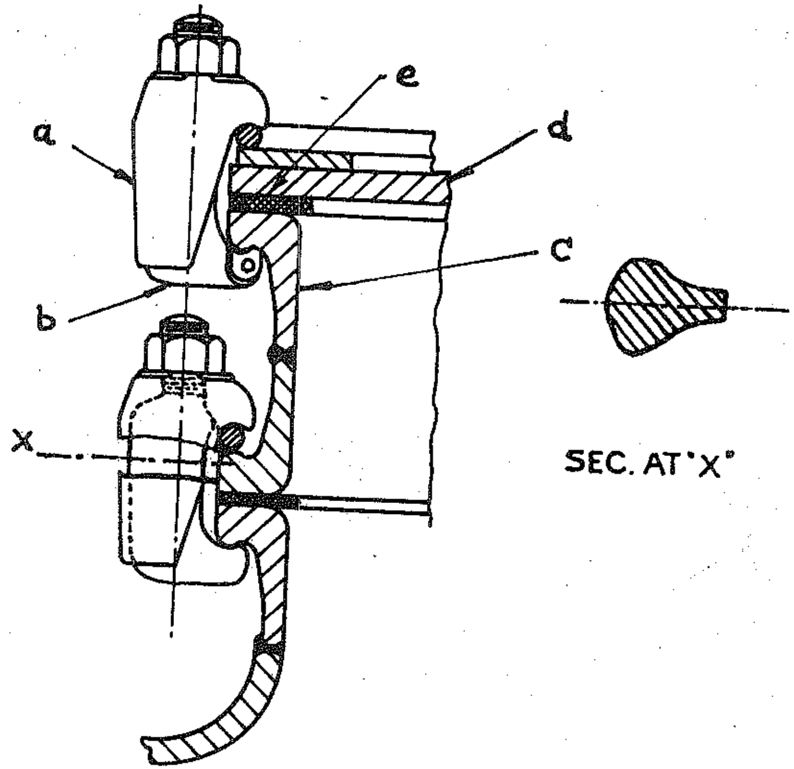


Fig 6.30 Special flange with swing clamps
(a) clamp (b) swing bolt (c) nozzle wall (d) cover (e) gasket

finish (Fig. 6.31). According to the width of gasket covering the flange facing, these can be divided into two broad types.

(A) *Wide-face flange*

(a) The gasket in this case extends over the full width of the flange face. These are suitable only when used with comparatively soft gaskets and should not be used for pressures exceeding 20 kg/cm^2 or temperatures exceeding 220°C.

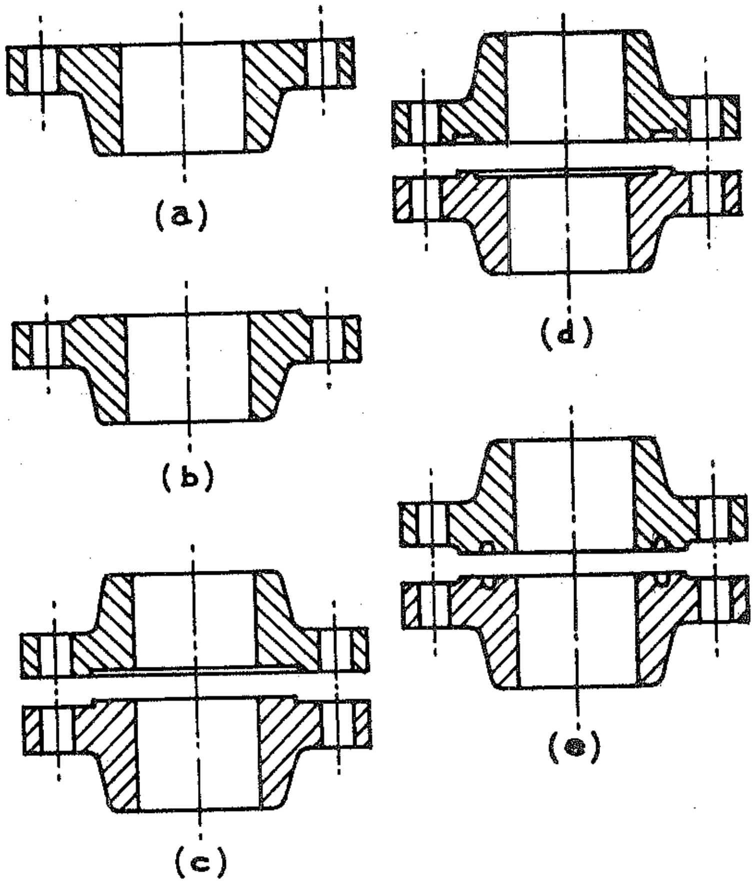


Fig. 6.31 Flange facings
(a) plain face (b) raised face (c) male and female facing (d) tongue and groove facing (e) ring facing

(B) *Narrow-face flanges*

(a) The gasket in these cases does not extend beyond the inside of bolt holes. The facings are as follows:

(b) A portion of the flange face is raised to locate the gasket. The raised face is finished with spiral or concentric grooves to hold the gasket. This type of facing is commonly employed. The gasket used is in the form of a flat ring of some soft material and is usually less in width than the width of the raised face.

(c) Male and female facings are formed with one flange facing recessed and the other flange face having a corresponding raised face portion. The surface of the faces is smooth, since the outer diameter of the female face acts to locate and retain the gasket.

(d) Tongue and groove facings have a groove formed on one flange facing and a raised ring on the other known as tongue. The gasket is held in position in the groove on both the inner and the outer diameter. The small tongue and groove construction requires only a narrow flat gasket.

(e) Ring facing is formed by making identical grooves usually trapezoidal in shape on both the flange facings. The gasket in the form of a ring fits into the grooves. The ring is well protected, since it is retained on the grooves.

6.8.4.3 GASKETS

Gaskets are interposed between two adjacent flange faces and are held tight by a series of bolts, which clamp the two flanges together. The gasket is therefore compressed, which causes a yielding of its surface, thus scaling the irregular surface of the flange faces. According to the properties and the shape different types of gaskets can be made.

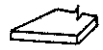
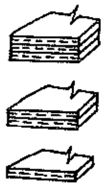
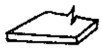
(a) *Flat ring gaskets*

These may be cut from sheets of different materials such as paper, cloth, rubber, compressed asbestos, soft iron, nickel, copper, aluminium, etc. Combinations of metals and non-metals can also be used. Their thicknesses normally range from 1/2 mm to 3 mm. Widths are usually 6 mm upwards. Narrow gaskets are preferable since they require less force for tightening, but they must not be too narrow otherwise they are likely to be crushed under the tightening force. Paper, cloth and rubber should not be used above 120°C. Asbestos may be employed up to 350°C, while ferrous or nickel base metal gaskets are satisfactory for higher temperatures. IS 4870 gives dimensions of flat gaskets for shell flanges.

(b) *Serrated gaskets*

These are flat metal gaskets having concentric grooves or serrations machined on to their faces. With serrations the

Table 6.7

Sketch	Dimension N (Min) mm	Gasket Material		Gasket Factor	Minimum Design Seating Stress kgf/mm²	Refer to Table 6.8 Use Facing Sketch	Refer to Table 6.8 Use Column
	10	Rubber without fabric or a high percentage of asbestos fibre : Below 70 IRHD 70 IRHD or higher		0.50 1.00	0 0.14	1 (a,b,c,d) 4,5	II
		Asbestos with a suitable binder for the operating conditions	3.2 mm thick 1.6 mm thick 0.8 mm thick	2.00 2.75 3.50	1.12 2.60 4.57		
		Rubber with cotton fabric insertion		1.25	0.28		
		Rubber with asbestos fabric insertion, with or without wire reinforcement	3-ply 2-ply 1-ply	2.25 2.50 2.75	1.55 2·04 2.60	1 (a,b,c,d), 4, 5	
		Vegetable fibre		1.75	0.77		
		Spiral-wound metal, asbestos filled	Carbon steel Stainless steel or monel metal	2.50 3.00	2.04 3.16		
		Corrugated metal, asbestos inserted or Corrugated metal, jacketed asbestos filled	Soft aluminium Soft copper or brass Iron or soft steel Monel metal or 4-6 per cent chrome steel Stainless steel	2.50 2.75 3.00 3.25 3.50	2.04 2.60 3.16 3.87 4.57	1, a, b)	II
		Corrugated metal	Soft aluminium Soft copper or brass Iron or soft steel Monel metal or 4-6 per cent chrome steel Stainless steel	2.75 3.00 3.25 3.50 3.75	2.60 3.1y 3.87 4.57 5.34	I (a, b, c, d)	
		Flat metal jacketed asbestos filled	Soft aluminium Soft copper or brass Iron or soft steel Monel metal or 4-6 per cent chrome steel	3.25 3.50 3.75 3.50	3.87 4.57 5.34 5.62	1a, 1b, 1c, 1d, 2	
		Grooved metal	Stainless steel Soft aluminium Soft copper or brass Iron or soft steel Monel metal or 4-6 per cent chrome steel	3.75 3.25 3.50 3.75 3.75	6.33 3.87 4.57 5.34 6.33	1 (a, b, c, d), 2, 3	II
	6	Solid flat metal	Stainless steels Soft aluminium Soft copper or brass Iron or soft steel Monel metal or 4-6 per cent chrome steel	4.25 4.00 4.75 5.50 6.00	7.10 6.19 9.14 12.66 15.33	1 (a, b, c, d), 2, 3, 4, 5	I
		Ring joint	Stainless steels Iron or soft steel Monel metal or 4-6 per cent chrome steel Stainless steels	6.50 5.50 6.00 6.50	18.28 12.66 15.33 18.28	6	
		Rubber 0-rings Below 75 IRHD Between 75 and 85 IRHD		3 6	0.0, 0.15	7 only	
		Rubber square section rings : Below 75 IRHD Between 75 and 85 IRHD		4 9	0.10 0.28	8 only	II

contact area is reduced to a few concentric lines, and the tightening force is greatly reduced and an efficient joint is obtained in cases where soft gaskets are unsuitable.

(c) *Laminated gaskets*

These are made of metal and a soft filler, usually soft asbestos sheet. In some gaskets the laminations are in the plane of the flange face, in others the laminations are formed in an axial or edgewise direction. These gaskets are considered more suitable for high temperature than plain asbestos gaskets and require less tightening load than solid metal gaskets.

(d) *Corrugated gaskets*

These are formed by inserting asbestos between the corrugations of a metallic sheet. They are intermediate in stiffness between flat non-metallic and metallic gaskets.

(e) *Ring gaskets*

These are in the form of continuous rings with usually octagonal or oval cross-sections. Such rings are normally made of metals for severe service conditions but for relatively low temperature, rings made of plastic or rubber may be employed.

(f) *Self-sealing gaskets*

Rings of different cross-section such as Lens, Delta, Grayloc wave are formed for use at high pressures. They are so shaped that a rise in pressure makes the sealing action more effective and hence the word self-sealing (for details see Chapter 12). (See further Table 6.7)

6.8.4.4 DESIGN OF FLANGED JOINT

As indicated earlier, standard steel flanges (IS-4864-4867) for shell sections, heads and pipes are selected according to the pressure and temperature requirement. In such cases no design calculations are necessary. However, where standard flanges cannot be used due to non-availability of specified material, special shape or size of flange, more elaborate bolting or clamping arrangements etc., it becomes necessary to follow a design procedure which involves selection of a gasket, flange facing, bolting, hub proportions, flange width and flange thickness.

Table 6.7

(1)	(2)	(3)	(4)	(5)	(6)
Corrugated metal	Monel metal or 4-6 % chrome steel	3.50	4.57		
	Stainless steel	3 75	5.34		
	Soft aluminium	3.25	3.87		
	Soft copper or brass	3.50	4.57		
	Iron or soft steel	3.75	5.34	1a, 1b, 1c, 1d, 2	
Flat metal jacketed asbestos filled	Monel metal or 4-6% chrome steel	3.50	5.62		
	Stainless steel	3.75	6.33		
	Soft aluminium	3.25	3.87		
	Soft copper or brass	3.50	4.57		
	Iron or soft steel	3.75	5.34	1(a, b, c, d), 2,3	II
Grooved metal	Monel metal or 4-6% chrome steel	3.75	6.33		
	Stainless steel	4.25	7.10		
	Soft aluminium	4.00	6.19		
	Soft copper or brass	4.75	9.14		
	Iron or soft steel	5.50	12.66	1, (a, b, c, d), 2, 3, 4, 5	I
Solid flat Metal	Monel metal or 4-6% chrome steel	6.00	15.33		
	Stainless steel	6.50	18.28		
	Iron or soft steel	5.50	12.66		
Ring joint	Monel metal or 4-6% chrome steel	6.00	15.33	6	
	Stainless steel	6.50	18.28		

(1)	(2)	(3)	(4)	(5)	(6)
	Rubber O-rings				
	Below 75 IRHD	3	0.07	7 only	
	Between 75 and 85 IRHD	6	0.15		
	Rubber square section rings:				
	Below 75 IRHD	4	0.10	8 only	
	Between 75 and 85 IRHD	9	0.28		
	Rubber T-section rings :				
	Below 75 IRHD	4	0.10	9 only	
	Between 75 and 85 IRHD	9	0.28		

Table 6.8

Basic Gasket Seating Width b_0

Column I	Column II
$\frac{N}{2}$	$\frac{N}{2}$
$\frac{w+25\,T}{2}$; $\frac{w+N}{4}$ Max	$\frac{w+25\,T}{2}$; $\frac{w+N}{4}$ Max
$\frac{w+N}{4}$	$\frac{w+3N}{8}$
$\frac{w}{2}$; $\frac{N}{4}$ Min	$\frac{w+N}{4}$; $\frac{3N}{8}$ Min
3	
$\frac{3N}{8}$	$\frac{7N}{16}$
$\frac{N}{4}$	$\frac{3N}{8}$
$\frac{w}{8}$	—
—	$\frac{N}{2}$
—	$\frac{N}{2}$
—	$\frac{N}{2}$

In designing a satisfactory flanged joint, the main considerations are (1) to insure a positive contact pressure at the gasket—flange interfaces under all service conditions and prevent leakage. The sealing force or the tightening force must satisfy this requirement. (2) to create tightening force by use of bolts, without over-stressing the bolts. (3) to ensure structural integrity of the flange faces, and to minimise deflections of the flanges.

Under the action of the tightening force, the gasket is compressed and stressed. It therefore, gives rise to a reaction. Three types of reactions are likely to be created.

(a) *Elastic reaction*—when the gasket is not over-stressed by the tightening force and its mass remains within elastic limit. Only the surface of the gasket yields, filling the irregularities of the flange faces, but the thickness or the mass of the gasket remains practically undeformed, i.e., it is stressed within elastic limit.

(b) *Plastic reaction*—when the gasket is over-stressed and its mass becomes plastic.

(c) *Springy reaction*—when the gasket is fully enclosed and resists compression like a confined fluid. Under varying pressure conditions plastic reaction is unsatisfactory, since once the mass of the gasket becomes plastic, it is unable to adjust itself to the required thickness and leakage starts.

The sizing of the gasket, and for the determination of the number and diameter of bolts necessary to create the tightening action, it is essential to evaluate the forces due to gasket reaction, both under atmospheric condition and also under the operating pressure condition. These forces are resisted by the bolt, creating a load on the bolt.

Under atmospheric condition, the bolt load due to gasket reaction is given by

$$W_a = A_g Y_a \tag{6.27}$$

where

A_g—area of gasket under compression

Y_a—stress in the mass of the gasket (seating stress)

Y_a should be less than the yield stress of gasket if elastic reaction is expected.

After the internal pressure is applied, the gasket which is compressed earlier, is released to some extent and the bolt load is given by Fig. 6.32.

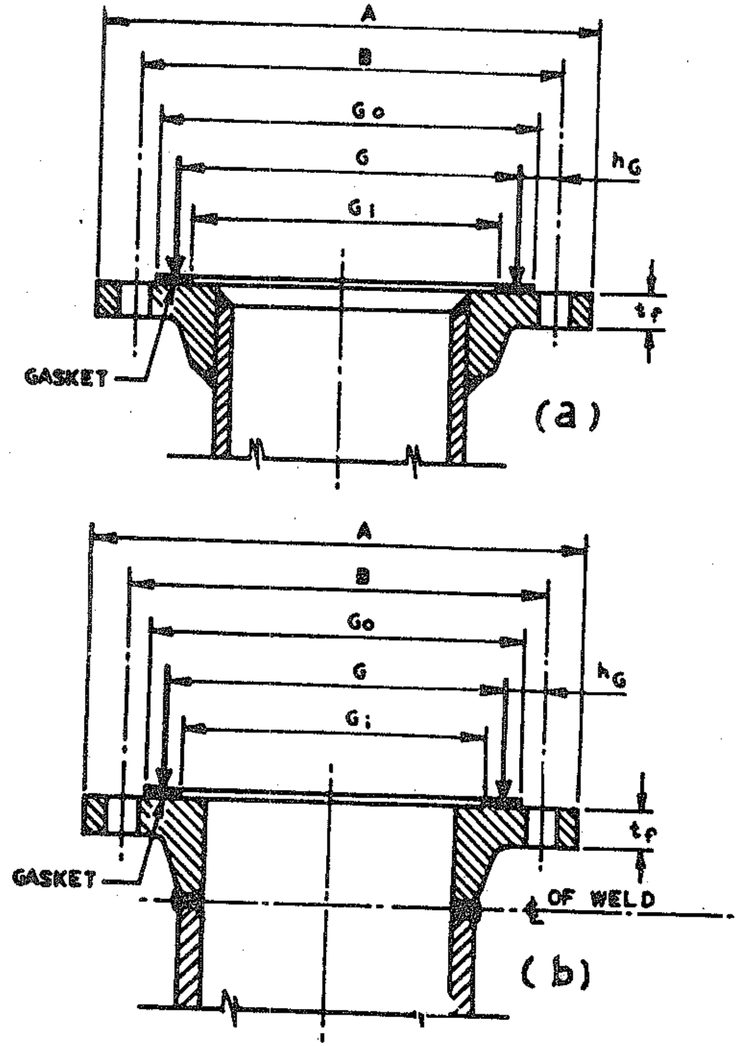
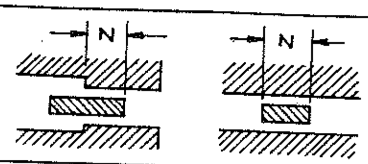
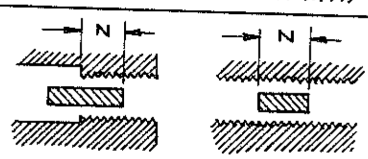
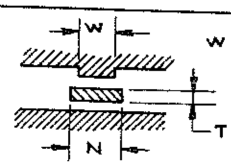
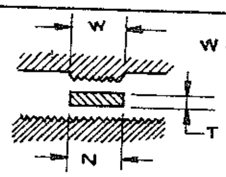
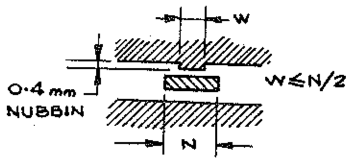
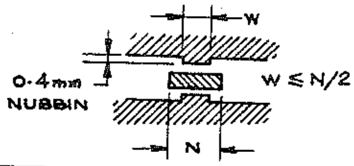
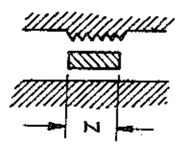
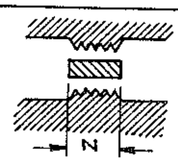
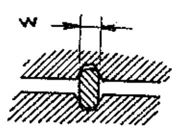
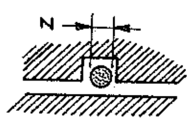
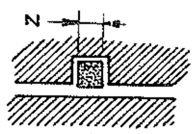
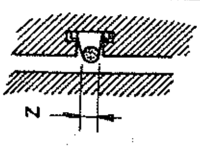


Fig. 6.32 Flange dimensions (a) lap welded (b) butt welded

Table 6.8

Basic Gasket Seating Width b_o

Facing Sketch — Exaggerated	Column I	Column II
	$\frac{N}{2}$	$\frac{N}{2}$
	—	—
	$\frac{w+25\,T}{2}$; $\frac{w+N}{4}$ Max	$\frac{w+25\,T}{2}$; $\frac{w+N}{4}$ Max
	—	—
	$\frac{w+N}{4}$	$\frac{w+3N}{8}$
	$\frac{w}{2}$; $\frac{N}{4}$ Min	$\frac{w+N}{4}$; $\frac{3N}{8}$ Min
	$\frac{3\,N}{8}$	$\frac{7\,N}{16}$
	$\frac{N}{4}$	$\frac{3\,N}{8}$
	$\frac{w}{8}$	—
	—	$\frac{N}{2}$
	—	$\frac{N}{2}$
	—	$\frac{N}{2}$

$$W_p = A_g\, Y_p + A_h\, p \qquad (6.28)$$

where

Y_p—residual stress in the mass of the gasket

A_h—area of the flange on which the pressure is exerted

p —operating pressure.

The change in the gasket mass stress is

$$\triangle Y = Y_a - Y_p \qquad (6.29)$$

The initial tightening action causes a compression of the thickness of the gasket, extension of the bolt length and deflection of both the flanges. Under pressure the gasket is released and its thickness changes, which is equal to the change in bolt length plus the change in the deflection of both the flanges. This is given by

$$\triangle T = \triangle L + 2\triangle f \qquad (6.30)$$

where

$\triangle T$—change in thickness of gasket

$\triangle L$—change in bolt length

f—change in deflection of each flange.

Applying Hooke's law

$$\frac{T\triangle Y}{E_g} = \frac{L\triangle W}{E_b\, A_b} + 2C\triangle W \qquad (6.31)$$

E_g ; E_b—modulus of elasticity of the gasket and bolt material respectively

T—thickness of gasket

L—length of bolt

$\triangle W$—$(W_p - W_a)$—change in bolt load

C—deflection factor of the flange.

From equations 6.27, 6.28 and 6.29

$$\triangle W = (W_p - W_a) = A_h p + \triangle Y A_g \qquad (6.32)$$

From equations 6.31 and 6.32

$$\triangle Y = \frac{B\, A_h\, p}{\frac{T}{E_g} + BA_g} \qquad (6.32a)$$

where

$$B = \frac{L}{E_b\, A_b} + 2C.$$

From equations 6.32 and 6.32a, change in bolt load

$$\triangle W = W_p - W_a = A_h\, p \left\{ 1 - \frac{BA_g}{\frac{T}{E_g} + BA_g} \right\} \qquad (6.33)$$

From this equation it may be observed that in case of metallic gaskets, whose modulus of elasticity (E_g) is very large the term $\frac{T}{E_g}$ is negligible. The value of the change in bolt load $\triangle W$ is almost zero. In such gaskets the bolt load during operating pressure conditions as well as atmospheric conditions remains practically constant. In the case of rubber or plastic materials, where E_g is less or in the case of thick gaskets, the term $\frac{T}{E_g}$ is significant and therefore, the bolt loads under both conditions must be assessed.

It is difficult to ascertain the value of residual stress (Y_p) in the mass of the gasket, under pressure conditions. However, it must be of sufficient magnitude to ensure that the joint will remain tight, without leakage. In effect, minimum compression of the gasket is necessary to seat the gasket properly. Tests on actual gaskets have shown that, if the joint is to remain pressure tight, when subjected to internal pressure, it is necessary to ensure that the residual gasket stress (Y_p), should be greater than m times the pressure (p). The term m (which is greater than 1) is known as gasket factor, which is dependent on the frictional resistance of the flange and the gasket contact surfaces. It is therefore, a property of the gasket material and its construction.

Tables 6.7, 6.8 show gaskets, materials with their properties, namely secting stress (Y_p), gasket factor (m) and basic secting widths (b_o). Gasket selection can be made from this table, depending upon temperature pressure and corrosion properties of the confined material. For low temperature and low pressure service, rubber, fibre and asbestos sheet material are commonly used. For pressures upto 20 kg/cm^2, and temperature upto 250°C the most common materials used are compressed asbestos sheet, and metallic laminated asbestos sheet. For higher temperatures, corrugated metallic gaskets and plain iron, aluminium, copper and monel sheet gaskets are used. When pressure exceeds 20 kg/cm^2 and temperature exceeds 400°C plane metal gaskets are preferred.

Gaskets may be either wide, covering almost the whole face of the flange or may be narrow. In most application, a narrow gasket is preferred. An approximate estimate of proportions of the gasket may be made as follows:

Gasket seating force—Hydrostatic pressure force
= Residual gasket force

$$\left[\frac{\pi}{4}(G_o^2-G_i^2)\right]-\left[\frac{\pi G_o^2}{4}p\right]$$
$$=\frac{\pi}{4}(G_o^2-G_i^2)\,Y_p$$
$$=\frac{\pi}{4}(G_o^2-G_i^2)\,mp \qquad (6.34)$$

G_o and G_i are the outside and the inside diameter of the gasket ring (Fig. 6.32)

$$\frac{G_o}{G_i}=\sqrt{\frac{Y_a-pm}{Y_a-p(m-1)}} \qquad (6.35)$$

Based on the above equations an appropriate size of the gasket, suitable for the requisite flange facing may be chosen. Gaskets having different values of the seating stress (Y_a) and the gasket factor (m) may be tried to arrive at a suitable width, and to satisfy the operating conditions.

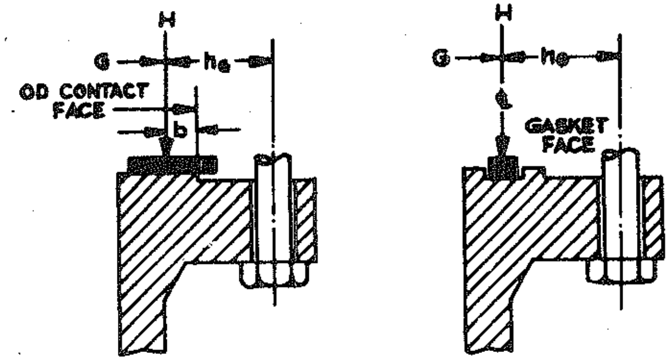
The minimum bolt load and the number and size of the bolts can be determined from equations 6.27 and 6.28, which can now be written in terms of the size of the gasket (Figs. 6.32 and 6.33).

Atmospheric condition

$$W_{m_1}=A_g\,Y_a=\pi bGY_a \qquad (6.36)$$

where

b—effective gasket or joint contact surface width (from Table 6.8 and Fig. 6.33)



For $b_o > 6{\cdot}3$ mm.   For $b_o \leqslant 6{\cdot}3$ mm.

b = Effective gasket seating width.

bo = Basic gasket seating width (From Table 6·7)

$b = bo$ when $bo \leqslant 6{\cdot}3$ mm.

$b = 2{\cdot}5\sqrt{bo}$ when $bo > 6{\cdot}3$ mm.

Fig 6.33 Effective gasket seating width

G—diameter of gasket load reaction

Y_a—gasket seating stress.

Operating condition

$$W_{m_2}=A_gY_p+A_hp=\pi(2b)\,Gm_p+\pi/4\,G^2p \qquad (6.37)$$

m—gasket factor (Table 6.7)

$2b$—effective gasket or joint contact surface width under pressure

p—design pressure.

The bolt loads either W_{m_1} or W_{m_2} wi ensile stress in the cross-section of the bolt

$$A_{m_1}=\frac{W_{m_1}}{f_a}\ \text{and}\ A_{m_2}=\frac{W_{m_2}}{f_b} \qquad (6.38)$$

where

A_{m_1} and A_{m2}—cross-section of bolt

f_a—permissible tensile stress in bolts under atmospheric conditions

f_b—permissible stress in bolts under operating conditions.

Further

$$A_{m_1} \text{ or } A_{m_2} = A_m \times \text{Number of bolts} \tag{6.39}$$

where

A_m=area of bolt.

The actual bolt area provided (A_b) shall not exceed the value given by

$$A_b = \frac{2\pi\, Y_a G N}{f_a} \tag{6.40}$$

where N—width of gasket (Table 6.7).

The actual bolt area (A_b) shall be greater than A_m or A_{m2}. To determine the number and size of bolts, the larger of the above two areas (A_{m1} or A_{m2}) should be considered. A large number of small sized bolts or a small number of large sized bolts can be selected. The bolt spacing must be enough to facilitate the use of wrenches. However, a large bolt spacing would cause a deflection of the flange. If this deflection is excessive, the joint will leak. The pitch between two bolts should be decreased with pressure. It may be generally between 3.5 to 5 times the diameter of the bolt. The total number of bolts may be approximately equal to mean diameter of gasket in cm divided by 2.5 and should be a multiple of four. The final choice of the number of bolts will depend on the proper bolt spacing. Bolts smaller than 15 mm should only be used when, considerable care is taken during tightening up to prevent yielding.

The next step is to determine the bolt circle diameter, consistent with the gasket diameter and bolt spacing. The final choice will be made only when the diameter of the flange hub is determined. The hub can be short and straight as in the case of a hub flange or tapered as in the case of a welding neck flange. The slope of the tapered hub may be equal to or less than 1 : 4. Assuming a suitable outside diameter of the hub at the base, the bolt circle diameter and the outside diameter of the flange can be determined. Within the practical limitations, the bolt circle, the gasket (centre line) circle, and the radial distance between them should be kept as small as possible.

6.8.4.5 FLANGE THICKNESS

An approximate value of the flange thickness may be given by Fig. 6.32.

$$t_f = G\sqrt{\frac{p}{kf}} + c \tag{6.41}$$

where

$$k = \frac{1}{\left[0.3 + \dfrac{1.5 W_m h_G}{H G}\right]}$$

G—diameter of gasket load reaction

p—design pressure

f—permissible stress

B—bolt circle diameter

c—corrosion allowance

W_m—total bolt load

h_G—radial distance from gasket load reaction to bolt circle $\left(\dfrac{B-G}{2}\right)$

H—total hydrostatic end force $= \dfrac{w}{4} G^2 p$

With low pressures the thickness of flange may work out to be negligible. It is necessary to provide a minimum thickness. The minimum flange thickness for carbon steel is between 20 mm to 40 mm depending on the diameter of the flange.

A rigorous approach to the determination of stresses created in the flange and the calculation of thickness is given in Indian Standard 2825 (paras 4.6, 4.7 and 4.8).

6.9 Supports

Details and design considerations of supports used for pressure vessels are indicated in chapter 13. These are non-pressure parts, and are therefore, designed essentially as structural members. They are attached to the vessel by welding, and produce local moments and stresses in the vessel wall area, surrounding the attachment.

6.10 Stresses from Local Loads and Thermal Gradient

As has been pointed out earlier under sections 6.2.1 and 6.8.1.2 apart from the three important loads namely, pressure, weight of vessel and wind load, there are a number of other forces which give rise to localised stresses in the vessel parts. These forces are due to

(*a*) local loads arising from
- (1) supports fer the vessel
- (2) structures supported by the vessel
- (3) loads imposed on branches by piping systems etc.

(*b*) Steady and transient thermal gradients

(*c*) local areas and lines of stiffening or thickening of the shell.

It is difficult to determine the effect of each one of the forces. Most of the above forces give rise to bending stresses in the shell or head, which decrease rapidly with distance from the area of application of load. Thermal gradients however, give rise not only to bending stresses, but longitudinal thermal gradients give rise to a combination of direct stresses and bending stresses.

The design criteria depend on whether the local bending stress system extends over a small or a large proportion of the circumference of the vessel. In the former case the criterion adopted is that the bending stresses are limited to the value which will just cause a plastic hinge to develop i.e., the localised material looses elasticity; the criterion is considered safe because a very small amount of distortion consequent upon the formation of a plastic hinge will cause a local increase in direct stresses which will prevent further distortion. The stress intensification factor corresponding to this condition is of the same order as that occurring around nozzles under the action of

pressure. If, however, the highly stressed area extends over considerable fraction of the circumference of vessel, loads approaching the plastic limit are liable to cause a kink to form in the shell. Under these conditions special precautions should be taken to reduce the bending stresses.

6.11 Thermal Stresses in Cylindrical Shell

In addition to the stresses created in the wall of a vessel, due to pressure and other loadings, stresses are also produced due to a temperature variation, through the wall. Thermal gradients through the wall are encountered when the contents of the vessels are heated from the outside or when heat from a reaction in the vessel is removed through the wall. Temperature gradient will cause expansion at any radius in three directions, namely, radial, circumferential and longitudinal. Portions of the wall having lower temperature will expand less and will resist expansion of the high temperature regions. Thermal stresses are set up due to uneven expansion.

Thermal stresses in the three directions can be determined, assuming flow of heat through the wall in a steady state and with a logarithmic thermal gradient. These stresses for a cylindrical shell are given by :

Radial direction

$$f_r = \frac{\alpha E(t_2 - t_1)}{2(1-\mu)\log_e\left(\frac{R_2}{R_1}\right)}\left[-\log_e\left(\frac{R_2}{r}\right) - \frac{R_1^2}{(R_2^2 - R_1^2)} \times \left(1 - \frac{R_2^2}{R_1^2}\right)\log_e\left(\frac{R_2}{R_1}\right)\right] \quad (6.46)$$

Circumferential direction

$$f_t = \frac{\alpha E\,(t_2 - t_1)}{2(1-\mu)\log_e\left(\frac{R_2}{R_1}\right)}\left[1 - \log_e\left(\frac{R_2}{R_1}\right) - \frac{R_1^2}{R_2^2 - R_1^2} \times \left(1 + \frac{R_2^2}{r^2}\right)\log_e\left(\frac{R_2}{R_1}\right)\right] \quad (6.47)$$

Axial direction

$$f_a=\frac{\alpha E(t_2-t_1)}{2(1-\mu)\log_e\left(\frac{R_2}{R_1}\right)}\left[1-2\log_e\left(\frac{R_2}{r}\right)-\frac{2R_1^2}{R_2^2-R_1^2}\log_e\left(\frac{R_2}{R_1}\right)\right] \quad (6.48)$$

where

α—coefficient of thermal expansion

E—modulus of elasticity

μ—Poisson's ratio

R_2, R_1—outer and inner radii

t_2, t_1—temperature at inner and outer radii

r—any radius

At the inner and outer surfaces, the radial stress f_r becomes zero. The circumferential or tangential stress (f_t) and the longitudinal or axial stress f_a have their largest numerical values at the inner and outer surfaces of the shell, which can be found by substituting $r=R_1$ and $r=R_2$.

Inner surfaces

$$f_t=f_a=\frac{\alpha E(t_2-t_1)}{2(1-\mu)\log_e\left(\frac{R_2}{R_1}\right)}\left[1-\frac{2R_2^2}{(R_2^2-R_1^2)}\log\left(\frac{R_2}{R_1}\right)\right] \quad (6.49)$$

Outer surface

$$f_t=f_a=\frac{\alpha E(t_2-t_1)}{2(1-\mu)\log_e\left(\frac{R_2}{R_1}\right)}\left[1-\frac{2R_1^2}{(R_2^2-R_1^2)}\log\left(\frac{R_2}{R_1}\right)\right] \quad (6.50)$$

If the heat flows from the inner to the outer wall of the shell compressive tangential stresses are set up. When the pressure is applied to the inside of this shell, tensile stresses are superimposed on the compressive thermal stresses. Thus in a shell subjected to internal pressure a temperature gradient with high temperature inside equalises the stresses throughout the wall. When the shell is heated from the outside the inside layer is put under tension and these thermal stresses are added to the tensile stresses due to internal pressure. At extreme temperature gradient thermal stresses may reach the yield point of the material, even in the absence of an internal pressure.

In the case of thin cylinders, the determination of thermal stresses is simplified and they are given by

$$f_t=f_a=\frac{\alpha E(t_2-t_1)}{2(1-\mu)} \quad (6.51)$$

Stresses determined so far are due to steady static temperature distribution throughout the wall. While warming up and cooling down additional stresses are induced in the wall of the vessel due to the transient temperature gradient. Theories in respect of these have not been worked out. However, in order to reduce these stresses, it is necessary to heat or cool vessels gradually.

6.12 Fabrication

The methods used for fabrication of the components of vessel may differ to some extent depending on the materials of construction. However, in a majority of the ferrous and nonferrous vessels, the methods of fabrication are essentially the same.

The most common methods used are welding, brazing, forging, flame cutting, sheet metal forming and other machining operations. Fusion welding forms a very important part of these methods. The various methods of fusion welding can be classified as (a) oxy-acetylene, (b) carbon arc, (c) metal arc, (d) atomic hydrogen, (e) inert gas, (f) submerged arc, (g) thermit (h) resistance welding.

Plates in various sizes and thicknesses are available for forming the shell and the heads. These plates are cut to suitable sizes by oxy-acetylene flame or by shearing in the case of thin plates. In the case of stainless steel, carbon arc (reversed polarity) or hydrogen arc may be used for cutting operations. The edges of these plates are then bevelled by an oxy-acetylene torch or can be machined to make them square.

The plates are then rolled to form a shell. In the case of thick plates the edges need crimpling, prior to the rolling operation, since the edges cannot be formed to the deserved curvature by the method of rolling alone. It is advisable to clamp together the two adjoining edges of the shell. The clamps can be removed as soon as these edges are tack welded. Finally the longitudinal seam of the shell is welded by any one of the fusion welding methods. For large diameter vessels and heavy sections, the submerged arc method is most suitable. This method is automatic, and gives a neat and regular weld which is free from any defects. In the case of copper and some special steel shells, after the initial rolling and welding, the shells are rolled again to give a uniform circular shape. The rollers on such rolling machines are removable for insertion of the original shell. Vessels having wall thickness above 5 cm are fabricated by forging.

The various types of dished heads are produced from blanks by either pressing them in a hydraulic press or by spinning them to the desired shape. In the second method the blank is made to rotate, while a pair of rollers are made to exert pressure on it. The blank is gradually moved in the radial direction simultaneously. The pressure of the rollers is therefore, gradually transferred to concentric areas on the blank. This pressure gives rise to the dished shape as required. It may be necessary to form a bevel on the edge of the head, for welding it to the shell. This can be produced by oxy-acetylene torch, which can be attached to the tool post of a vertical boring machine. The head is slowly rotated with the table of the machine, while the torch is being operated. The head is connected to the shell by welding and the assembly may be heat treated depending on the thickness.

Holes in the shell or the head can be made by an oxy-acetylene torch which is attached to a compass or a pantograph mechanism. The edges of the holes may be upset by a hydraulic press, by heating the area around the hole locally. The nozzles are welded to these openings as required. It is desirable to tack weld these nozzles and check their correct position and alignments prior to the final welding.

Nozzles may be fabricated from plates by the method of rolling and welding or may be formed to the desired shape by

forging. Flanges are produced by forging with the help of a steam hydraulic press. After suitable heat treatment, these are machined to give the final shape and size.

Post weld heat treatment is carried out in the case of carbon and low alloy ferritic steels either by normalising or by stress relieving. Special heat treatment is necessary in case of alloy steels and non-ferrous metals. Thermal stress relieving is necessary when the vessel is (a) intended for containing toxic or inflammable material ; (b) intended for operation below 0°C (c) intended for use with media liable to cause stress corrosion cracking (d) subject to excessive stress concentration or risk of fatigue due to changing loads (e) under risk of brittle fracture (f) to have dimensional accuracy and shape in service (g) to be made of a plate thickness (including corrosion allowance) at any welded joint in the vessel shell or head, which exceeds the values given in Table 6.9.

The vessel is stress relieved as a whole in an enclosed furnace wherever practicable. Where it is not practicable to do so, special precautions will have to be taken to heat the vessel in sections. To obtain good finish the vessel is sand blasted, ground and polished.

6.13 Numerical Problem

Data (Plate I)

(a) *Shell*

Internal diameter (Approx.)	1200 mm
Material—stainless steel	(0.5 Cr 18 Ni 11 Mo 3)
Permissible stress at 150°C	13.00 kg/mm^2
Internal pressure	3.00 kg/cm^2

(b) *Head—Flanged and shallow dished*

External diameter	1200 mm
Crown radius	1200 mm
Knuckle radius	72 mm
Material—Same as shell	

(Continued on page 185)

Table 6.9

Maximum Plate Thickness above which Post Weld Heat Treatment is Required

Percentage Chemical Composition and Mechanical Properties	Plate Thickness
Carbon 0.20 Max Manganese 1.00 Max Residual elements 0.80 Max Specified minimum yield strength 26 kgf/mm² Max	30 mm
Carbon 0.25 Max Manganese 1.60 Max Residual elements 0.80 Max Specified minimum yield strength 38 kgf/mm² Max	However, for fine grained steels this may be increased to 38 mm subject to preheating over 30 mm
Carbon 0.30 Max Manganese 1.20 Max Residual element 0.40 Max Specified minimum yield strength 38 kgf/mm² Max	
Carbon 0.25 Max Manganese 1.60 Max Chromium 0.60 Max Molybdenum 0.60 Max Vanadium 0.12 Max Residual or other elements 0.80 Max Specified minimum yield strength 44 kgf/mm² Max	20 mm
Carbon 0.35 Max Manganese 1.20 Max Residual or other element 0.40 Max Specified minimum yield strength 44 kgf/mm² Max	
Carbon 0.25 Max Manganese 1.60 Max Chromium 1.50 Max Molybdenum 1.50 Max Vanadium 0.16 Max Residual or other element 0.80 Specified minimum yield strength 44 kgf/mm² Max	All thicknesses
All other ferritic steels	All thicknesses

(c) *Flanges*

Material—carbon steel (IS—2002) grade I

Permissible stress (upto 250°C)—9.5 kg/mm²

Gasket asbestos

(d) *Bolts*

Material—Hot rolled carbon steel

Permissible stress—upto 50°C 5.87 kg/mm²

Permissible stress—upto 200°C 5.45 kg/mm²

(e) *Nozzle*—(*welded to head*)

Internal diameter—150 mm

Thickness — 3 mm

Material — same as shell

6.13.1 Design of Shell

From equation 6.3 (Internal design pressure is 3+0.3= 3.3 kg/cm²)

$$t=\frac{p\,D_i}{2fJ-p}=\frac{3.3\times1200}{2\times13.0\times10^2\times0.85-3.3}$$

$$=1.80 \text{ mm.}$$

2 mm thickness will be satisfactory. However, for rigidity a minimum thickness of 4 mm is commonly used. No corrosion allowance is necessary for stainless steel.

Combined loadings (based on 2 mm thickness):

From equation 6.5

$$f_t=\frac{p(D_i+t)}{2t}=\frac{3.3\,(1200)}{4}=993 \text{ kg/cm}^2 \text{ (tensile)} \simeq 0.973\times10^8$$

From equation 6.6

$$f_l=\frac{p\,D_i}{4t}=\frac{3.3\times1196}{8}=497 \text{ kg/cm}^2 \text{ (tensile)} \simeq 0.48\times10^8$$

From equation 6.7

$$f_2=\frac{W}{\pi t(D_i+t)}=\frac{3200\times10^2}{\pi\times2(1200)}=42.4 \text{ kg/cm}^2 \text{ (ccompressive)}$$

No wind load $f_3=0$

$$f_a=f_1+f_2=497-42.4=454.6 \text{ kg/cm}^2 \text{ (tensile)}$$

Torque due to offset piping etc.—50 kg m

From equation 6.10

$$f_s=\frac{T}{\pi t\ D_i\ (D_i+t)}=\frac{50\times 10^3\times 10^2}{\pi\times 2\times 1198\ (1200)}=0.55\ \text{kg/cm}^2$$

f_s is negligible as compared to other stresses

From equation 6.11

$$f_R=\left[(993)^2-(993\times 454.6)+(454.6)^2\right]^{\frac{1}{2}}$$

$=860$ kg/cm² (tensile)

Both f_R and f_a are less than the permissible tensile stress (1300 kg/cm²)

6.13.2 Head Design—Flanged and Shallow Dished Head

From equation 6.23 (Internal design pressure 3.3 kg/cm²)

$$t_h=\frac{3.3\times 1200\times 1.75}{2\times 1300\times 1}\ ; \qquad J=1$$

$=2.66$ mm

$$W=\frac{1}{4}\left(3+\sqrt{\frac{1200}{72}}\right)=1.75$$

Use 4 mm thickness (No corrosion allowance is necessary)
Straight flange length $3\times 4=12$ mm

Minimum specified 20 mm

6.13.3 Flanges for Top Head and Shell

The shell and top head are connected by a flanged joint. The vessel is made of stainless steel for which the flange may be selected from Indian Standard specification (IS-4868—Welded shell flanges for stainless steel pressure vessels and equipment) Table 1 of the specification gives a flange of 1200 mm nominal diameter. This requires a minimum shell thickness of 6 mm. The shell thickness as per design is only 4 mm. It is therefore, necessary to weld an additional ring between the shell and the flange (see Fig. 6.24).

The particulars of flange and bolts are as follows—

Nominal diameter of flange	—	1200 mm
Inside diameter of flange	—	1202 mm
Outside diameter of flange	—	1315 mm
Outside diameter of stainless steel backing ring	—	1240 mm
Bolt circle diameter	—	1270 mm
Thickness of flange	—	45 mm
Number of bolts	—	48 ~~mm~~
Nominal diameter		M20

If a flange is not of the standard size, it is necessary to carry out a design procedure as follows:

(a) ***Design of gasket and bolt size***

The flange is made of carbon steel with a stainless steel lining (raised face) in the form of a ring. A flat asbestos gasket of 1200 mm internal diameter and 1240 mm external diameter and 3 mm thickness is used, to cover the raised face.

From Tables 6.7 and 6.8

Gasket factor m—2.00

Minimum design seating stress—112 kg/cm²

Basic gasket seating width $b_0=\frac{1}{2}\ \frac{(1240-1200)}{2}=10$mm

Effective gasket seating width $b=2.5\sqrt{b_0}=2.5\sqrt{10}=7.9$ mm

From equation 6.36

$$W_{m_1}=\pi\times\frac{7.9}{10}\times\frac{1240+1200}{2\times 10}\times 112 \qquad (G=1220\ \text{mm})$$

$=34000$ kg

From equation 6.37

$$W_{m_2}=\pi\times 2\times\frac{7.9}{10}\times\frac{1240+1200}{2\times 10}\times 2\times 3.3 + \pi/4\times\left(\frac{1240+1200}{2\times 10}\right)^2\times 3.3$$

$= 4000+38400=42400$ kg

From equation 6.38

$$A_{m_2}=\frac{34000}{587}=57.9\ \text{cm}^2$$

$$A_{m_2} = \frac{42400}{545} = 77.8 \text{ cm}^2$$

$$\text{Number of bolts} = \frac{1220}{10 \times 2.5} = 49$$

Use 48 bolts

From equation 6.39

$$\text{Diameter of bolt} = \sqrt{\frac{77.8}{48} \times \frac{4}{\pi}}$$
$$= 1.435 \text{ cm}$$

Use M 18 bolts

Pitch diameter—16.376 mm

Minor diameter—14.480 mm

Actual bolt area

$$= \frac{\pi}{4}\left[\frac{16.375^2 + 14.48^2}{2 \times 100}\right] \times 48 = 90 \text{ cm}^2$$

From equation 6.40

$$A_b = \frac{2\pi}{587} \times \frac{1220}{10} \times 112 \times \frac{20}{10}$$
$$= 292 \text{ cm}^2$$

The bolt area suggested is satisfactory.

Pitch of bolts— $4.75 \times 18 = 85.6$ mm

Pitch circle diameter (B)

$$= \frac{85.6 \times 48}{\pi} = 1310 \text{ mm}$$

B=outside diameter of gasket $+ 2 \times$ diameter of bolt $+ 12$ mm

$= 1240 + 2 \times 18 + 12 = 1278$ mm

Outside diameter of flange

Pitch circle diameter $+ 2 \times$ diameter of the bolt

$= 1278 + 2 \times 18$

$= 1314$ mm

(b) *Flange thickness*

From equation 6.41

$$K = \frac{1}{0.3 + \dfrac{1.5 \times 42400 \times \dfrac{(1310-1220)}{2}}{\pi/4 \times (1220)^2 \times 3.3 \times 1220}}$$

$$= \frac{1}{0.3 + .0625} = 2.74$$

$$t_f = 1220 \sqrt{\frac{3.3}{2.74 \times 950}}$$
$$= 48 \text{ mm}$$

6.13.4 Nozzle Reinforcement Design

Nozzle is provided on the head (Fig. 6.17)

Internal design pressure—3.3 kg/cm^2

From equation 6.3 minimum nozzle thickness

$$t_n = \frac{3.3 \times 150}{2 \times 1300 \times 1 - 3.3} = 0.189 \text{ mm}$$

No corrosion allowance, since the material is stainless steel. Actual thickness $t_n = 3$ mm

Maximum horizontal distance for compensation

$(AB) - 2 \times 150 = 300$ mm

Maximum vertical distance for compensation

$(AD) - 6 \times 2.66 = 15.99$ mm

or $(3.5 \times 2.66 + 2.5 \times 3) = 16.8$ mm

Area of compensation

$150 \times 2.66 = 400$ mm^2

(*i*) Area of compensation provided by the head

$A_s = 150\,(3 - 2.66) = 51$ mm^2

(*ii*) Area of compensation provided by the nozzle

$A_o = 2 \times 2.5 \times 3\,(3 - 0.189) = 42$ mm^2

(*iii*) The nozzle does not project inside the vessel $H_2 = 0$

$A_1 = 0$

$A_s + A_o + A_1 = 51 + 42 + 0 = 93$ mm^2

Area of compensation required $= 400 - 93 = 307$ mm^2.

This can be provided by a ring of 260 mm outside diameter, 156 mm inside diameter and 3 mm thick. Area of compensation by ring $= (260 - 156)\ 3 = 312$ mm^2. This is satisfactory.

Reading References

Harvey, J.F., *Pressure Vessel Design*, Van Nostrand, Princeton, (1967).

Brownell, L.E. and Young, E.H., *Process Equipment Design*, John Wiley, New York, (1968).

Bickel, M.B. and Ruiz, C., *Pressure Vessel Design and Analysis*, Macmillan, London, (1967).

Spring, H.M., *Pressure Vessels for Industry*, McGraw-Hill, New York, (1947).

Gill, S.S., *The Stress Analysis of Pressure Vessels and Pressure Vessel Components*, Pergamon, Oxford, (1970).

Backhurst, J.R. and Harket, J.M., *Process Plant Design*, Heinemann, London, (1973).

Proceedings of a Symposium on *Pressure Vessel Technology*, Chem. Age of India, **23** (5), 329 (1972).

Jethandan, H.H. and Sarma, K.S., *Spotlight on Pressure Vessel Industry*, Chem. Age of India, **22** (6), 351 (1971).

Krishnamurthi, S., *Application of Stainless Steel in Design and Fabrication of Chemical and Process Industries*, Chem. Age of India, **22** (11), 794 (1971).

Pressure Vessel Guide, Brit. Chem. Eng. **15** (7), 869 (1970).

Laithwate, J., *Pressure Vessel Standardisation and Design*, Chem. Process Eng., **51** (10), 61 (1970).

Mead, W. J., *Encyclopedia of Chemical Process Equipment*, Reinhold, New York, (1964).

Riegel, E.R., *Chemical Process Machinery*, Reinhold, New York, (1953).

Dimoplon, N. Jr., *Geometry of Pressure Vessels Heads*, Hydrocarbon Processing, **53** (8), 71 (1974).

Raghavan, K.V., *Pressure Vessel Design*, Chem. Process. Eng., **51** (10), 81 (1970).

Fort, H., Bett, K.E., Rogeen, J., and Gardner H.F., *Design Criteria for Pressure Vessels : Safety in High Pressure Polyethylene Plants*, CEP Technical Mannual, **15** (1973).

Markovitze, R.E., *Choosing the Most Economical Vessel Head*, Chem. Eng., **78** (16), 102, (1971).

Lancaster, J.F., *Recent Developments in Pressure Vessel Steel—U.S. vs. Europe*, Hydrocarbon Processing, **43** (10), 161 (1964).

Thandavan, N. and Fogelman, J.A., *Fabrication of Stainless Steel Pressure Vessels*, Chem. Age of India, **22** (11), 886 (1971).

Pipe Jointing Methods for Metal Piping, *Handbook* No. 23, The Engineering Equipment Users Association Constable and Co., London. (1968).

Thielsh, H., *Defects and Failures in Pressure Vessels and Piping*, Reinhold Publishing Corporation, (1965).

CHAPTER 7

Storage Vessels

7.1 Introduction

Liquid and gaseous products must be stored during intervals between production, transportation, refining, blending and marketing. The object of storage at each of these stages is firstly to supply a sufficient balance of each stock to ensure continuity of operations, secondly to ensure that the product is conserved and maintained at an acceptable level of quality.

In general, the storage of liquid or gaseous products is done in large-sized vessels. These consist of units either completely fabricated in the shop or vessels so large that they must be manufactured on site. Various parts of the vessels are made in sizes which can be conveniently transported. The most common material used is mild steel. Vessels are also made of stainless steel, aluminium and clad steel, where the cladding may be either stainless steel, nickel, monel or inconel.

7.2 Storage of Fluids

Fluids normally stored in bulk can be classified in three groups, namely, non-volatile liquids, volatile liquids and gases.

7.3 Storage of Non-volatile Liquids

These liquids are usually stored in standard cylindrical tanks in sizes in excess of 60 metre diameter and with heights upto 30 metres. Very small tanks may be rectangular, made of flat plates. These tanks require stiffening arrangements, since flat plates are structurally weak, and are unable to withstand pressures. Small size tanks may also be horizontal cylindrical vessels with flat or slightly dished ends, supported on two saddles.

7.3.1 Standard Free Roof Cylindrical Tank

Vertical cylindrical tanks consist of a vertical shell usually butt welded with a flat bottom and roof sheets, supported on an umbrella type roof truss. The bottom and the roof are fabricated from sheets by welding. The roof sheets are not usually attached to the roof truss, and the tank is thus free to breathe with the changes in temperature and pressure, the roof sheet acting like a membrane leaves the truss when subject to internal pressure.

7.4 Storage of Volatile Liquids

These liquids are stored in specially designed tanks which are capable of conserving the contents. If these are stored in normal fixed volume tanks, a loss of liquid takes place due to the following causes:

(a) *Breathing losses*—A space above the liquid level is filled with air and vapour mixture. Vaporisation increases with rise in temperature during the day. Increase in temperature will therefore, result in the expansion of the mixture and the loss of certain volume through the vent. The temperature drops during the night and some condensation takes place, the air-vapour mixture contracts and fresh air is sucked into the tank.

(b) *Filling losses*—When the liquid is discharged from a fixed volume tank, the tank will be filled with air vapour mixture. If the tank is then filled again with a fresh batch of liquid, certain volume of air-vapour mixture will be vented and lost. This loss can be substantial if such fillings take place very frequently.

(c) *Boiling losses*—If the vapour pressure of the liquid stored is higher than the atmospheric pressure it boils and the vapour thus produced is lost. In such cases the liquid should be stored in a vessel operating under a pressure not less than the vapour pressure of the liquid at the maximum temperature of storage.

7.4.1 Standard Fixed-roof Storage Tank

Depending on the volatility of the liquid, the tank is designed to have pressure-tight roof. Normally the pressure variation in the vapour space of the tank roof may be from +200 mm to −6.25 mm w.g. In some cases the tanks are built with higher pressure upto 625 mm. A vent is fitted on such tanks with a pressure relief valve set to open at an internal pressure of 200 mm or 625 mm as the case may be. An anti-vacuum valve should also be provided and also connected separately to the vent, so that air can enter the tank whenever the tank develops a vacuum of 6.25 mm or higher as specified. When the temperature of air-vapour mixture in the tank drops, or when the liquid is discharged from the tank a vacuum higher than 6.25 mm is developed on account of the fact that the vent is connected to the atmosphere through a pressure relief valve and not directly. Fixed roof tanks (Fig 7.1) or any closed tank

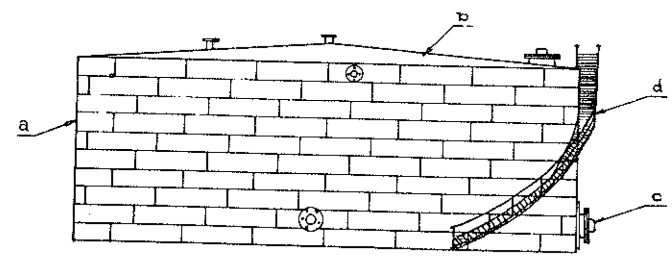


Fig. 7.1 Standard fixed-roof storage tank
(a) shell (b) roof (c) manhole (d) ladder

should be designed to withstand the vacuum conditions. With proper adjustment of the relief valve breathing losses can be completely eliminated and the filling losses reduced to some extent.

7.4.2 Variable Volume Tanks

7.4.2.1 Vapour-lift roof type

In this construction, the roof normally rests on the usual roof supports as in a fixed volume cone-roof tank. When the volume of the vapour increases due to rise in temperature or when the liquid is pumped, the roof rises vertically increasing the vapour space (Fig. 7.2). A liquid seal prevents the loss of

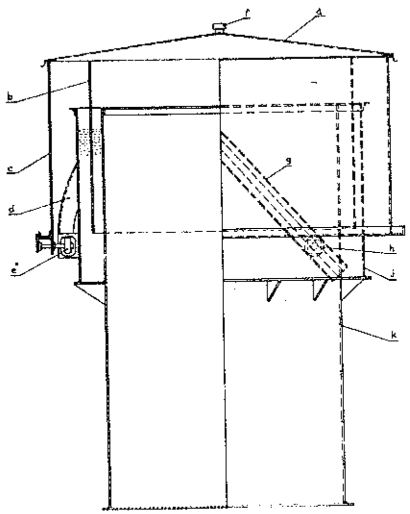


Fig. 7.2 Vapour-lift roof storage tank
(a) vapour lift roof (b) moving shell (c) wind apron
(d) guide track (e) guide roller (f) emergency valve
(g) guide track (h) guide roller (j) seal tank
(k) fixed tank shell

vapour at the junction of the roof and shell. A simple mechanism, which directs and stabilizes the upward and the downward movement of the roof, consists of tracks placed at regular intervals at an angle of 45°, roller guides ride in these tracks and other roller guides placed at 90° to the rollers in the tracks, ride against the side of the tank and take any lateral load such as from winds. The liquid seal and stabilizers are fully protected from the weather by a wind apron. Suitable pressure or vacuum relief valves are used to prevent accidental excess pressure or vacuum. The liquid seal is designed to relieve excess pressure in case the relief valve fails to function. These tanks will eliminate all breathing losses and also filling losses to a certain extent.

7.4.2.2 FLOATING ROOF TYPE

In this type of variable volume tanks, the roof (Fig. 7.3) is not supported either by the shell or by columns, but is made to rest on the stored liquid and is free to move with the level of

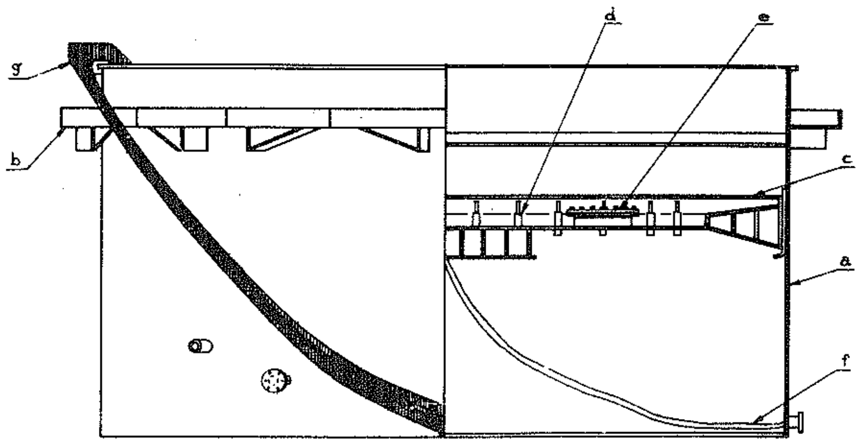


Fig. 7.3 Floating roof storage tank
(a) shell (b) shell reinforcement (c) pontoon type floating roof
(d) adjustable pipe support (e) manhole (f) hose drain (g) ladder

the liquid, the vapour space is thereby practically eliminated. No breathing losses are possible, since there is little space in which the air vapour mixture can form. It is difficult to create a perfect seal between the floating roof and the tank shell. Therefore, certain amount of liquid is expected to be lost through the imperfect seal. These tanks are therefore, most economical when used as working tanks which are filled and emptied at frequent intervals, than merely as storage tanks.

Three types of floating roofs are generally used. They are (a) Pan (b) Pontoon (c) Double-deck.

The pan type of roof is the cheapest. The double deck type will give the greatest structural protection as well as hold the evaporation losses to a minimum.

(a) *Pan roof*

This type of roof has an upstanding rim which gives the roof floating characteristics. Various types of pan roofs are shown in Fig. 7.4. Since the pan is a single-deck roof, an unchecked

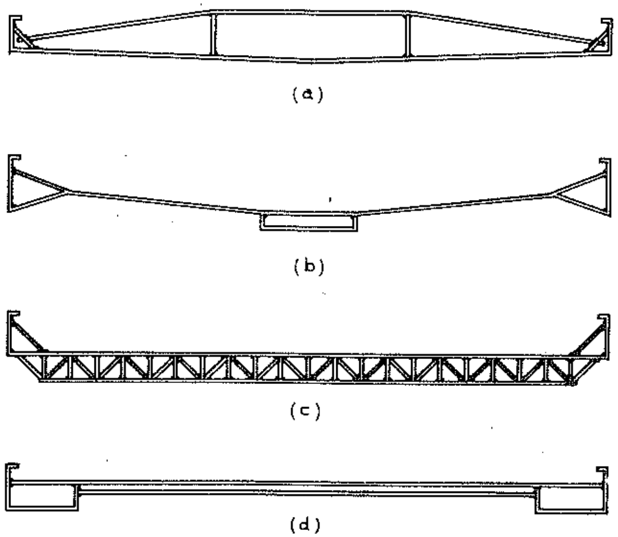


Fig. 7.4 Pan type roofs
(a) trussed pan roof (b) pan roof with centre weight
(c) Pan roof with underdeck truss (d) pan roof with underdeck girder

leak through this deck will eventually cause it to sink. It has no other buoyancy other than that provided by the deck and rim. If the roof is loaded by rain or snow the rim gets compressed and begins to deform. Excessive distortion can cause an imperfect seal.

(b) *Pontoon roof*

This represents a significant improvement over the pan roof. Buoyancy and stability are increased, as is the insulation of the product from the sun. As a result, these roofs have been widely used in both cold and hot climates and on corrosive and non-corrosive products with vapour pressures upto 1 kg/cm^2.

Some of the pontoon roofs are shown in Fig. 7.5. It is common to use a pontoon with 35 cm to 45 cm depth, and a pontoon area of about 20 to 40 per cent. These roofs are able to carry considerable water load.

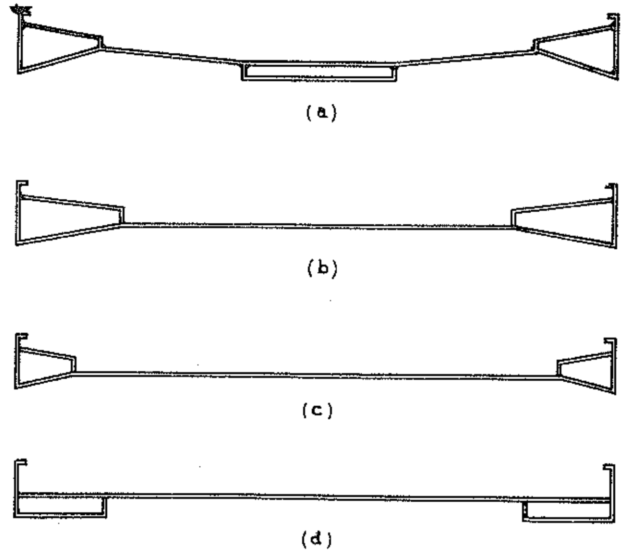


Fig. 7.5 Pontoon type roofs
(a) pontoon roof with centre weight (b) pontoon covering 30 per cent of roof area (c) standard pontoon roof (d) pontoon roof with deck raised off product

(c) *Double-deck roof*

This roof comprises of upper and lower decks separated by bulkheads and trusses (Fig. 7.6). Circular bulkheads divide

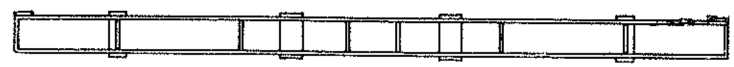


Fig. 7.6 Double-deck roof

entire roof into annular liquid tight compartments and some of these compartments are in turn divided into liquid tight sectors by means of radial bulkheads. Radial stresses are dispersed throughout to provide the required strength and stability.

Because of its obviously superior load capacity and insulating characteristics, the double-deck roof is used extensively for all types of products in all climates.

7.4.3.1 ACCESSORIES OF FLOATING ROOF TANK

(A) *Seal* : A flexible seal between the floating roof and the tank shell is an essential feature. Two types of seals are shown in Fig. 7.7.

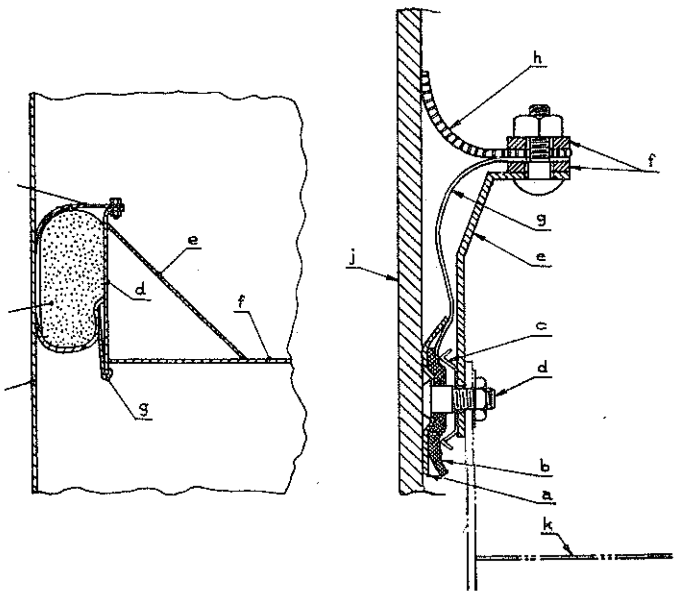


Fig. 7.7 Seals for floating roof tank
(A) (a) tank shell (b) endless fabric (c) flexible foam ring (d) rim (e) stiffener (f) deck plate (g) hold down bar
(B) (a) sliding shoe (b) continuous seal (c) clamp (d) bolt (e) apron plate (f) washer bar (g) curtain seal (h) secondary wiper seal (j) shell (k) floating roof

(i) The space between the pan type of roof and the shell, is filled by a seal consisting of a resilient foam doughnut covered by a polyurethene-coated nylon seal envelope (Fig. 7.7 A).

(ii) Fig. 7.7(B)—This consists of a sliding shoe (a) of extruded section formed so as to contact the tank shell along

two horizontal sections. It is covered on the inside by a continuous seal (b) of flexible material, which is held in position by clamps (c) and large number of bolts (d) A series of segmental apron plates (e) are held in position by bolts (d) at one end and washer bars (f) at the other end. The gaps at the top, bottom and sides of the apron are sealed by narrow strips of flexible material, known as curtain seals (g). A secondary wiper seal (h) consisting of flexible material is provided to exclude air currents. The assembly is connected to the floating roof.

(B) *Drains* : Drains are necessary to remove accumulated water from the floating roof. Open drains are needed on double-deck roof. They allow the water to flow into the stored product, where it settles to the bottom to be drawn off later. A siphon drain is an open pipe with screened top, having a bottom reservoir. The water reservoir seals against the product and prevents its flow on to the deck. A hose drain conducts water from a central sump on the roof to the outside of the tank through a shell nozzle.

In addition to drain, it is necessary to provide certain other facilities for the floating roofs such as adjustable pipe supports (c) to keep the roof from resting on the tank bottom when the tank is empty or nearly so. These pipe supports hold the roof about one metre off the bottom during normal operation, and two metres for cleaning and inspection. Vents (D) are required to provide air under the roof as the tank is emptied or to relieve pressure as the tank is filled. Manholes or openings (E) are placed in the deck or pontoons for inspection and cleaning operations and a rolling ladder (F) for getting on top of roof. Anti-rotation device (G) is needed for keeping the roof from rotating during product level changes and thereby prevent damage to rolling ladder, pipe drain, swing line and gauging equipment. A gauge platform (H) is usually designed at the top of the tank.

7.5 Storage of Gases

7.5.1 SPHERICAL VESSELS OR HORTONSPHERES

These types of vessels (Fig. 7.8) are extensively used for storage of gases and volatile liquids in the pressure range of 1 to 10 atmospheres. With higher pressures the increased plate thicknesses necessitate stress relieving of the vessel and create difficulties in construction and erection. Therefore, hortonspheres are rarely used for higher pressure.

Under similar operating pressure conditions, the thickness of the plates required to form a spherical vessel is about half that required for a cylindrical shell. Moreover, the ratio of surface area to volume is less for a sphere than any other shape of a vessel. This is an advantage and reduces the plates required. In case of insulated vessels, this reduces the amount of insulation required.

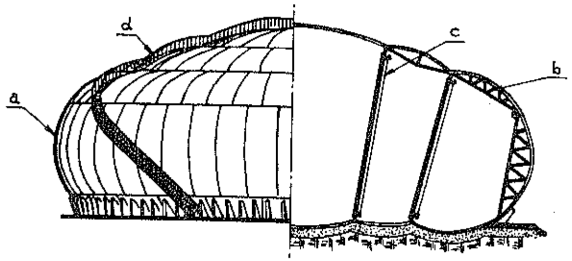


Fig. 7.8 Spherical storage vessel (hortonsphere)
(a) shell (b) truss (c) column (d) ladder

Hortonspheres usually range from 6 metres diameter upto about 25 metres diameter. They are supported by 6 to 12 tubular or rolled section columns attached to the shell at the equator. The upper section of each column is welded to the shell. The columns are braced with rods and turnbuckles to carry the wind load. Each column rests on a concrete pier. The upper half of the sphere acts like a hemispherical dome roof and the lower half of the sphere may be considered as a hemispherical suspended bottom.

7.6 Design of tanks

The design and construction features of the tanks are given in details in the following codes :

(1) Indian standard—803
(2) British standard—2654
(3) API standard—650

The above codes cover requirements for mild steel, vertical cylindrical tanks with conical or domed covers and flat bottoms, in various sizes, for erection above ground. Some of important design features are indicated below. For design purposes, these tanks can be divided into three categories.

(a) Fixed roof tanks designed to operate only at nominal pressure/vacuum, designated briefly as non-pressure tanks (Internal pressure 75 mm water gauge and vacuum of 25 mm water gauge).

(b) Fixed roof tanks designed to withstand small pressures/ vacuum.

(i) Class 'A' tanks—For an internal pressure of 200 mm and vacuum of 65 mm water gauge.

(ii) Class 'B' tanks—For an internal pressure of 550 mm and a vacuum of 65 mm water gauge for tanks normally not exceeding 20 metres in diameters.

(c) Open-top tanks.

7.6.1 MATERIALS

Structural mild steel plates and sections are used for tank construction. These steels are based on IS—2062 specification for structural steel. Suitable sizes of plates or strips are indicated in IS—1730.

7.6.2 BOTTOM DESIGN

The flat bottom of the tank is constructed of rectangular plates as shown in Fig. 7.9. For tanks over 12 metre diameter an annular ring of segmental plates is provided. All joints in the plates are lap welded, with an overlap not less than 5 times the thickness of the thinner plate.

It is assumed that the bottom plates are fully supported, either on a concrete slab, or on a well consolidated and levelled ground. The bottom plates are therefore, subjected to direct pressure of liquid and not to any bending moment. However the bottom plates near the shell to bottom joint are highly stressed due to the bending moment caused by the pressure of liquid on shell. Further, although the bottom plates are supposed to be uniformly supported by the ground, it is possible, particularly in the case of large tanks that certain portions of the ground may settle unevenly and the support may not be uniform. All bottom plates shall have, therefore, a minimum nominal thickness of 6 mm. All segmental plates for tanks greater than 12 metres diameter shall have a minimum thickness of 8 mm.

In the case of tanks which are supported on beams for providing a clearance under the tank, the bottom plate is subjected to a bending moment by the liquid pressure. The unsupported part of the bottom can be considered as a rectangular plate, uniformly loaded with pressure 'p' and supported at its perimeter. The maximum stress is given by

$$f_{max} = \frac{Pb^2}{2t^2} \times \frac{1}{1+b^2/a^2} \tag{7.1}$$

where

a—longer side of plate

b—shorter side of plate

t—thickness of plates

p—pressure of liquid on the bottom pH

H—maximum liquid height in the tank.

7.6.3 Shell Design

7.6.3.1 INTERNAL LOADING

The internal pressure in the tank-shell is computed on the assumption that the tank is filled to the full height by the liquid stored. The pressure varies from maximum at the lowest point to minimum at top. The thickness of the tank has to be determined according to the pressure variation. The shell is constructed by welding different courses with suitable thicknesses. It is assumed that, the bottom plate gives some support to the lowest course, and for other courses the lower courses would give some restrain to the top courses. Therefore, the tension in each course is computed at 30 cm above the centre line of the lower horizontal joint in question.

7.6.3.2 EXTERNAL LOADING

(a) Where tanks are subject to high wind velocities, the shell may be stiffened to prevent failure by buckling when empty.

(b) Heavy loads from platforms and elevated walkways shall be distributed appropriately.

7.6.3.3 SHELL THICKNESS

The minimum thickness of the shell plate is determined as follows:

The internal pressure is calculated from the liquid height

$$p = \rho\,(H-0.3)\times 10^{2}) \tag{7.2}$$

where p—pressure in kg/cm^2

ρ—density in kg/cm^3

H—height from bottom of the course under consideration to the top of the roof-curb angle in metres.

$$t = \frac{pD}{2fJ} \times 10^3 + C \tag{7.3}$$

where t—thickness of shell in mm

D—nominal diameter of tank in metres

J—joint efficiency usually taken as 85 per cent

C—corrosion allowance in mm

f—permissible stress in kg/cm^2.

The above equation is applicable to non-pressure and class 'A' tanks only. For class 'B' tanks, the pressure p, is calculated by an addition of the internal pressure to the pressure due to liquid height as calculated from equation 7.2.

In no case shall the thickness of shell plates be less than given in Table 7.1.

Table 7.1

Nominal Tank Diameter (metres)	Minimum Nominal Thickness (mm)
Smaller than 15	5.0
Over 15 upto and including 36	6.0
Over 36 upto and including 60	8.0
Over 60	10.0

The above thicknesses do not include corrosion allowance. Maximum plate thickness is 40 mm.

7.6.3.4 SHELL JOINTS

Horizontal and vertical joints between shell plates are of butt type. The type of butt welds are square butt, single bevel or double bevel for horizontal joints according to thickness of

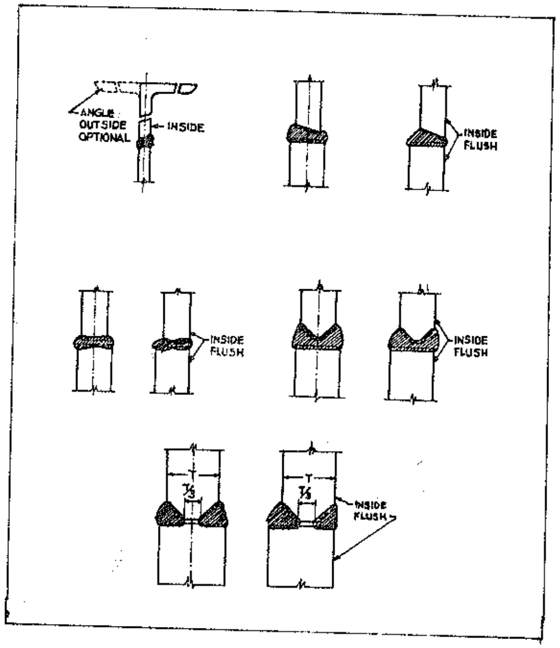


Fig. 7.10 Horizontal joints for shell plates

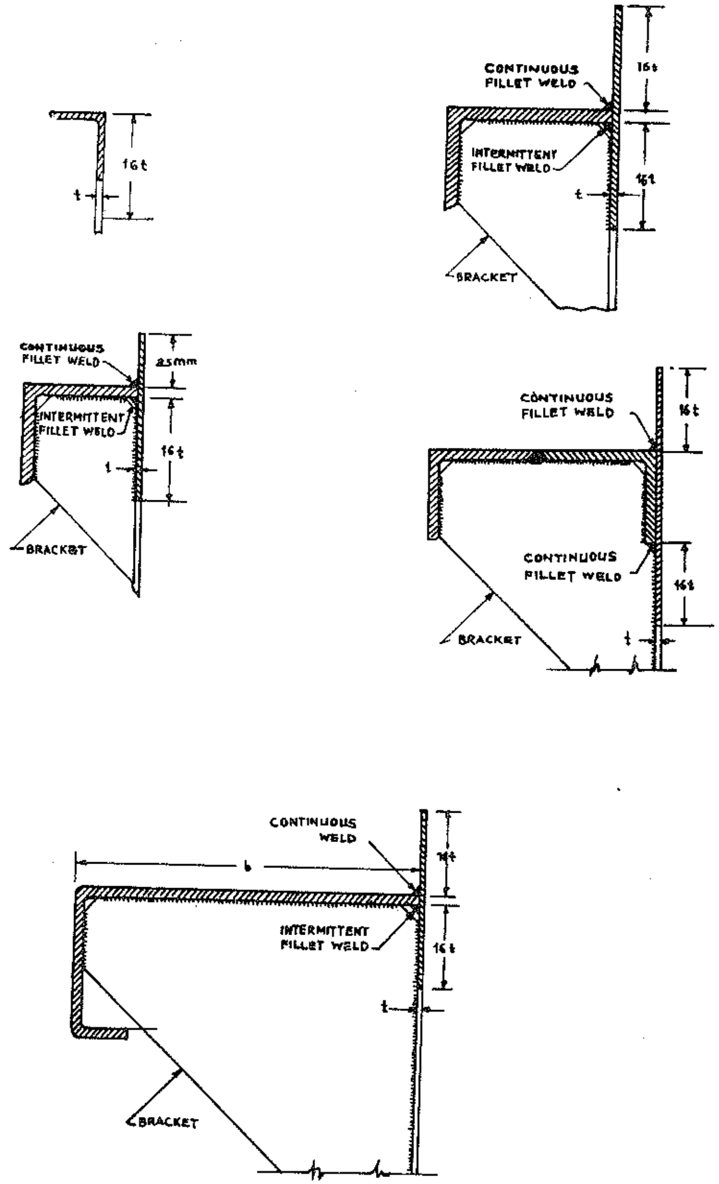


Fig. 7.11 Wind girder attached to shell top

plates (Fig. 7.10). Vertical joints are square butt, single V or U-butt or double V or U-butt type according to thicknesses of plates.

7.6.4 Wind Girders for Open-Top Tanks

These tanks are provided with stiffening rings to maintain roundness when the tank is subjected to wind load. Such rings, known as wind girders, are located near the top course, preferably on the outside of the tank-shell. The required minimum section modulus of the stiffening ring is given by

$$Z = 0.059\ D^2 H \tag{7.4}$$

where Z—section modulus in cubic centimetres

D—normal diameter of tank in metres

H—height of tank in metres.

The section modulus of the stiffening ring of the shell is based upon the properties of the applied members and may include a portion of the tank-shell for a distance of 16 plate thicknesses below, and if applicable, above the ring shell attachment. Stiffening rings are made of either structural sections, formed plate sections, or sections built up by welding. Attachment of wind girder sections to shell are shown in Fig. 7.11.

7.6.5 Roof-Curb Angles

For closed top tanks, the tank-shells are provided with top-curb angles, which are attached to the upper edge of the shell plate by a continuous double welded square butt joint or continuous double-fillet lap joint (Fig. 7.12).

(a) *Non-pressure tanks*—For these tanks, the top-curb angles are of the following minimum sizes:

(1) Tanks upto 10 metre diameter
$65 \times 65 \times 6.0$ mm

(2) Tanks over 10 metre diameter and upto 18 metre diameter
$65 \times 65 \times 8.0$ mm

(3) Tanks over 18 metres diameter and upto 36 metre
$75 \times 75 \times 10.0$ mm

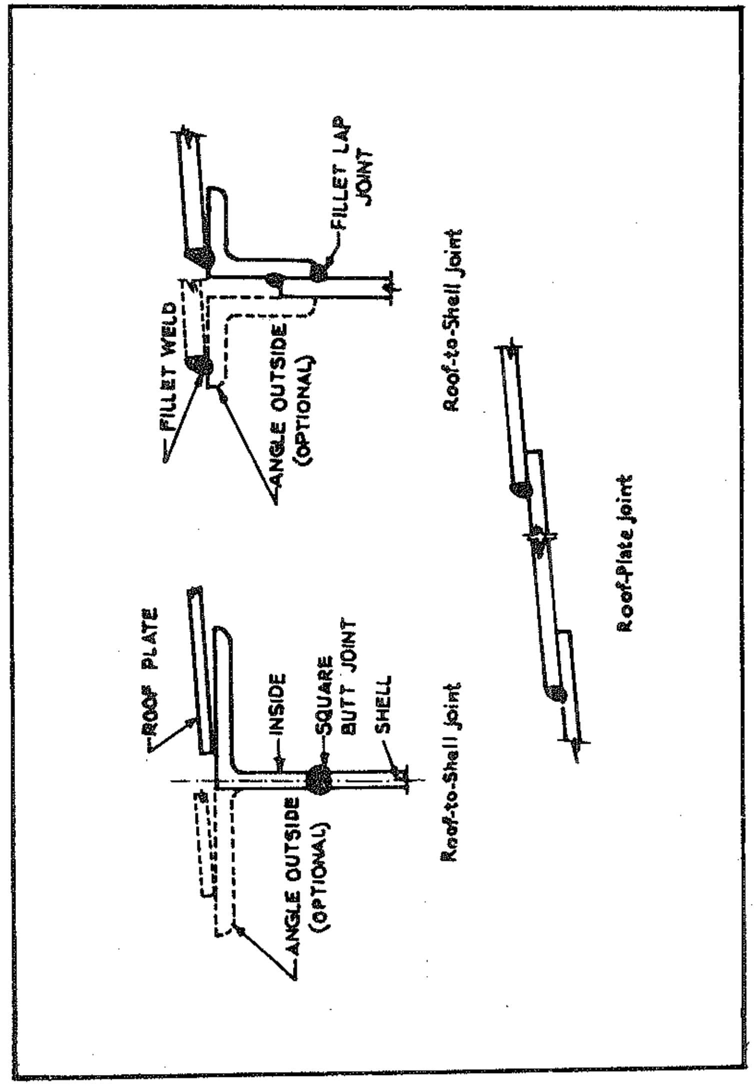


Fig. 7.12 Roof curb angles

(4) Tanks over 36 metre diameter

$$100 \times 100 \times 10.0 \text{ mm}$$

(b) *Pressure tanks* (*class A and class B*)—The top-curb member in such tanks will be required to take roof load, if the roof is of the self-supporting type. In such cases the top-curb angle has to be increased in size. The cross-sectional area required is indicated later under self-supporting roof design (equation 7.10).

7.6.6 Self-Supporting Roof Design

In this type of roof, the entire roof load is supported by the tank periphery. The roof shape may have the following forms:

(a) *Cone roof*—appropriate to the surface of a right cone. The slope of the cone is 1 in 5 or 1 in 6.

(b) *Dome roof*—The radius of curvature is a spherical radius.

(c) *Umbrella roof*—A modified dome roof, so formed that any horizontal section is a regular polygon with as many sides as there are plates.

7.6.6.1 ROOF LOADING

Roofs shall be designed to support the following loads and pressures.

(a) A superimposed load of not less than 125 kg/m² measured on the horizontal plane in addition to dead load of roof sheets and supporting structure. This load usually consists of snow, wind and men walking on the roof. This may be reduced if there is absolutely no possibility of snowfall in the area.

(b) An internal pressure equivalent to

(i) 75 mm water gauge or 75 kg/m² for non-pressure tanks

(ii) 200 mm water gauge or 200 kg/m² for class 'A' tanks

(iii) 550 mm water gauge or 530 kg/m² for class 'B' tanks

7.6.6.2 STRESSES IN CONE ROOF

The stresses in the cone roof plates are determined by considering the above loads. The stress in the cone roof plates under either external or internal pressure is given by (Fig. 7.13)

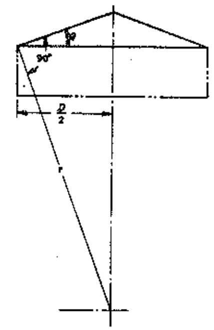


Fig. 7.13 Conical roof geometry

$$f = \frac{pD}{2t \sin \theta} \qquad (7.5)$$

where f —stress in roof plate

p —pressure due to roof loadings

D—diameter of cone roof

t —thickness of cone roof plates

θ —angle between cone roof and horizontal.

In the case of large diameter conical roofs, with external load the wrinkling of the roof or elastic instability may cause failure of the plates. The theoretical critical compressive stress that can cause failure of a curved plate due to wrinkling is given by

$$f_c \text{ (critical)} = \frac{E}{\sqrt{3(1-\mu^2)}} \frac{t}{(r)} \qquad (7.6)$$

where

E—modulus of elasticity

μ—Poisson's ratio

r—radius of curvature of roof $= \dfrac{D/2}{\sin \theta}$ (Fig. 7.13)

The safe compressive stress is taken with a factor of safety of 12 and with Poisson's ratio of 0.33 for steel.

$$f_c \text{ (permissible)} = \frac{1}{12} \frac{E}{\sqrt{3(1-0.33^2)}} \times \frac{t}{D/2} \sin\theta \qquad (7.7)$$

$$f_c \text{ (permissible)} = 0.102 \ E \left(t \right) \qquad (7.8)$$

Considering external load on the r f, the stress in equation (7.5) is compressive.

Equating stresses in equation (7.5) and (7.8)

$$(\sin\theta)^2 = \frac{pD^2}{0.202 \ E \ t^2} \qquad (7.9)$$

$$\sin\theta = \left(\frac{D}{t}\right)\sqrt{\frac{p}{0.202 \ E}} \qquad (7.10)$$

The slope of the cone is limited to 1 in 5 or 1 in 6, i.e. $\tan\theta < \frac{1}{5}$ or $\frac{1}{6}$, from which the plate thickness may be determined.

The minimum thickness of the steel roof plates is 5 mm. The roof plates are lapped and welded with continuous fillet weld on the top side only. They are supported on purlins. The spacing of roof purlins shall be such that the distance between them shall not exceed 1.6 metres. In addition the purlin frame is provided with bracings.

7.6.6.3 AREA OF CURB ANGLE

In a self-supporting roof, the load on the roof is transferred to the top of the shell and top curb angle. The curb angle must have enough cross-sectional area to withstand this compressive load. The total area required should not be less than

$$A = \frac{W \cot\theta}{2\pi f} \qquad (7.11)$$

where W —total load on roof in kg

θ —cone roof angle (Fig. 7.13)

f —allowable stress kg/cm^2.

The area A consists of

$$A = A_c + A_s + A_r$$

where

A_c—area of the curb member

A_s—area of shell plates effective $= 1.5 \ t_s\sqrt{Rt_s}$ (cm^2)

A_r—area of roof plates effective$= 0.75 \ t_r \ \sqrt{R_1 t_r}$ (em^2)

t_s—thickness of shell plate in cm

t_r—thickness of roof plate in cm

R—radius of tank (in cm)

R_1—radius of curvature of roof in cm.

7.6.7 COLUMN SUPPORTED ROOF

This type of roof (Fig. 7.14) has a column or columns transmitting the roof load to the tank bottom. The roof plates are placed on a structural framework and are designed to prevent

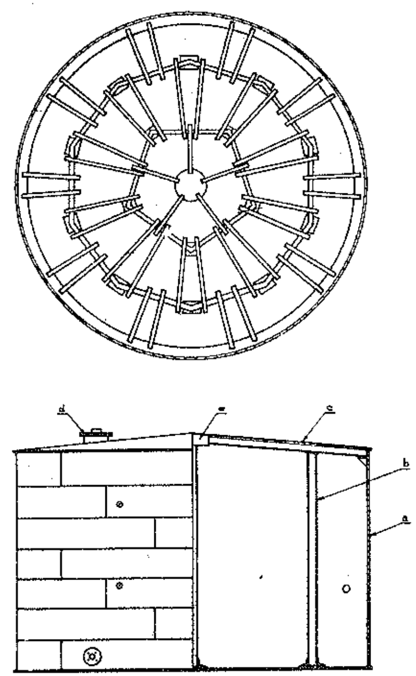


Fig. 7.14 Column supported roof.
(a) shell (b) column (c) roof (d) manhole (e) centre column cap

the plates from bending. The roof plates are assumed to act as flat continuous plates with uniform roof load. The rafters and girders are assumed to act as uniformly loaded beams with freely supported ends. The plates are designed as rectangular plates, uniformly loaded and freely supported at the edges (equation 7.1). If 5 mm thick plates are used the rafter spacing has to be limited to 2 metres.

Rafters and girders are designed as uniformly loaded beams with freely supported ends. Each rafter is considered to support the roof plates and roof load over an area extending on either side of the rafter and bounded by the centre line of the adjacent rafter. The length of rafters is limited to about 6 metres.

Column supports are used at the centre of the tank along with each ring of girders. Usually 5 or more straight girders are joined end-to-end to form a polygonal support for the ends of rafters. The girders are designed in the same manner as the rafters. Column supports are designed on the basis of Rankine-Gordon formula (equation 3.16).

7.7 Nozzles and Mountings

The following nozzles are provided on tanks :

(1) Shell manholes (50, 60, 75 and 90 cm diameters)
(2) Inlet and outlet connections
(3) Level gauge connection
(4) Overflow connection
(5) Drain
(6) Roof manhole (50 cm and 60 cm diameter)
(7) Vent connection
    (a) Breather vent
    (b) Emergency vent
(8) Combined water draw-off and clean out sump.

Details of the nozzles are shown in Figs. 7.15 and 7.16. Openings in tank-shells larger than 60 mm in diameter shall be reinforced.

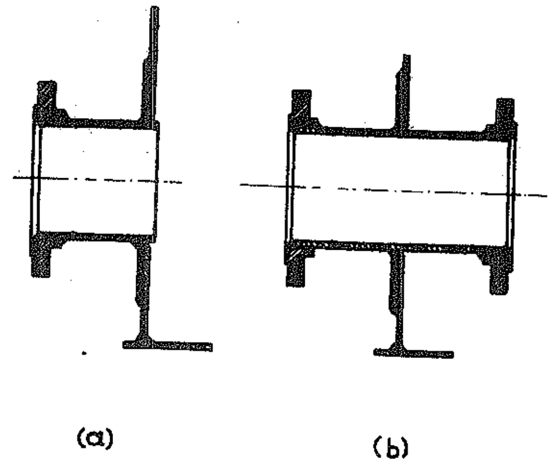


Fig. 7.16 Nozzles

Mountings are provided as (1) Stairways (2) Gangways (3) Handrailing and (4) Ladders

7.7.1 Breather Vents

The size of the vent (Table 7.2) should be adequate to prevent a pressure or vacuum to build up beyond the specified range.

Table 7.2

Capacity of tank (litres)	Breather vent size (cm)
4000—8000	3.5
8000—40,000	5
40,000—200,000	7.5
600,000—1600,000	10 and 7.5 used together

7.8 Numerical Problem

Fixed conical roof cylindrical tank

Data

Tank diamater	20 m (app.)
Tank height	12 m (app.)
Sp. gr. of liquid	1

Conical roof
Slope—permissible 1 in 5
Superimposed load—125 kg/cm²
Material—carbon steel (structural)
Permissible stress—1420 kg/cm²
Density—7.7
Modulus of elasticity—2×10^6 kg/cm²

7.8.1. Shell

From equation (7.2), pressure at the bottom of tank

$p=0.001(12-0.3)\times10^2$

$=1.17$ kg/cm²

From equation 7.3, thickness of the shell at the bottom

$$t=\frac{1.17\times20\times10^3}{2\times1420\times0.85}+C$$

$=9.7+C$

$=10$ mm

Butt joints are provided for welding the vertical joints between plates to form the lowest layer of the shell. An allowance of 2 mm is made between two adjacent plates to facilitate welding.

Total circumference with 10 plates

$=(\pi\times20)-(10\times2\times10^{-3})=62.8$ m

$$\text{Length of each plate}=\frac{62.8}{10}=6.28 \text{ m}$$

The size of the plate selected from IS-1730 is 6300 mm (length)×1800 mm (width).

To determine the number of layers to be used with 10 mm thick plate, it is necessary to determine the height upto which a plate of lesser thickness can be used.

Assuming a thickness of 8 mm, from equations (7.2) and (7.3)

$$8=\frac{0.001\ (H-0.3)\times10^2\times(20\times10^3)}{2\times1430\times0.85}$$

$H=9.65+0.3=9.75$ m

Since the width of the plate selected is 1800 mm it is necessary to use 10 mm thick plate to form the two lowermost layers. This will cover a height of (2×1.8) i.e., 3.6 m.

The remaining height of tank $=12-3.6=8.4$ m

The third layer will be of 8 mm thickness

The remaining height of tank $=8.4-1.8=6.6$ m

Assuming a plate thickness of 6 mm from equations (7.2) and (7.3)

$$6=\frac{0.001\ (H-0.3)\times10^2\times(20\times10^3)}{2\times1420\times0.85}$$

The remaining four layers will be of 6 mm thickness

Height of four layers $=4\times1.8=7.2$ m

Total height of tank with 7 layers

$=7\times1.8=12.6$ m

Tank height as per data—12 m

Excess height $=12.6-12=0.6$ m

This extra height is useful for overlap between layers to form horizontal fillet welding.

$$\text{Overlap per joint}=\frac{0.6}{7-1}\times100=10 \text{ cm}$$

The overlap may be reduced by cutting the width of the top plate.

7.8.2 Bottom

Diameter of the bottom of tank extends beyond the shell by 65 mm.

$$\text{Bottom diameter } D_b=20+\left(\frac{2\times10}{10^3}\right)+\left(\frac{2\times65}{10^3}\right)$$

$=20.15$ m

Thickness of plates to be used to form the bottom

Annular plates $=8$ mm

Other plates $=6$ mm

The bottom plates will be lap (fillet) welded. The overlap between the sketch plates and the annular plate is 65 mm. and between the sketch plates (5×6), i.e., 30 mm. The joints between plates will be staggered so that no more than three plates are overlapped within 30 cm of each other or of the shell.

Annular plates

Minimum width of annular plate

$$=65+380+10+65$$
$$=520 \text{ mm}$$

Circumference of bottom $=\pi \times D_b = \pi \times 20.15$
$$=54.6 \text{ m}$$

From IS-1730, a suitable plate is selected with 5600 mm length and 1100 mm width. Since the minimum width required is 520 mm to obtain 10 annular plates, 5 plates of the selected size may be cut in two parts each.

Sketch plates—Selected size 5000 m×2500 m

7.8.3 Self-Supporting Conical Roof

Assume plate thickness of 6 mm for roof plates

Weight of the roof plates

=area of plates×thickness×density

Weight per unit area

=thickness×density

$$=\frac{6}{10}\times 7.7\times 10=46.2 \text{ kg/m}^2$$

Total pressure on the roof

=superimposed load+weight of roof plate

$$=125+46.2$$
$$=171.2 \text{ kg/cm}^2$$

The slope of roof from equation 7.10

$$\sin\theta = \frac{20\times 10^3}{6}\times\left(\frac{171.2\times 10^{-4}}{0.202\times 2\times 10^6}\right)^{\frac{1}{2}}$$
$$=0.686$$

$$\tan\theta = 0.949$$
$$\text{slope} = 1 \text{ in } 1.06$$

The slope is far greater than the permissible values. Assuming a plate thickness of 18 mm

$$\sin\theta = \frac{20\times 10^3}{18}\times\frac{263.6\times 10^{-4}}{0.202\times 2\times 10^6}$$
$$=0.284$$
$$\tan\theta = 0.296$$
$$\text{slope} = 1 \text{ in } 3.38$$

Even with a thickness of 18 mm the slope is high. It would be therefore not desirable to use a self-supporting roof. In case a column supported type roof is used, it is possible to use a plate thickness of 6 mm and provide for rafters under the roof plates. The rafters are to be supported on columns placed within the tank (Fig. 7.14).

Reading References

Blenkin, B. and Wardale, J.K.S., *Storage Vessel—Design Fabrication and Use*, Chem. Process Eng., **44**(7), 344 (1963).

Bodley, R.W., *Choose a Right Floating Roof Tank*, Petrol. Refiner, **40**(3), 185 (1961).

Perry J.H., *Chemical Engineers' Handbook*, 5th Edn., McGraw-Hill New York, (1973).

Boberg, I.E., *Choosing Tank Designs*, Chem. Eng., **77** (17), 134 (1970).

Forbes K., McGregor A. and Turner S., *Design of Fluid Storage Tanks from Polypropylene*, Brit. Chem. Eng., **15** (10), 1333 (1970).

Binder, G. and Sutton, N., *Large Plastic Tanks*, Brit. Chem. Eng., **15** (10) 1332 (1970).

Riegel E.R., *Chemical Process Machinery*, Reinhold, New York, (1953).

Mead W.J., *Encyclopedia of Chemical Process Equipment*, Reinhold, New York, (1964).

Brownell L.E. and Young E.H., *Process Equipment Design*, John Wiley, New York, (1968).

CHAPTER 8

Reaction Vessels

8.1 Introduction

Pressure vessels are used as reaction vessels or kettles for carrying out operations such as blending, dispersion, gas absorption, dissolution, batch distillation, etc., under controlled conditions. The vessel may be open or closed. If open it may have no cover at all, or simply have a loose-fitting flat lid that prevents splashing or escape of fumes. Alternatively the reaction vessel may require a cover or head that can be secured tightly to the shell so that the reaction can be carried out under controlled pressure and temperature. If the closed vessel is capable of withstanding moderate pressures it is called a 'reaction kettle', but if the vessel is required to withstand high pressures and temperatures, that must be maintained at constant values during the reaction process the vessel is called an 'autoclave'.

Depending on the processing operation, the vessel may require heating, cooling and agitation of the contents. The rated capacities of reaction vessels normally vary between 100 litres to as large as 1500 litres, with the shell diameter varying between 50 cm and 250 cm. The type of heads used for such vessels are usually shallow-dished, torispherical or elliptical. When it is not necessary to remove the cover frequently, a normal flanged head secured by stud or nut and bolt connections is very suitable. If however the cover is to be opened frequently a special type flange is provided and swing type bolts are necessary (for details see 6.8.4). A common weakness of the covers of this kind is the wear and tear of the gasket sealing the cover. These gaskets must be protected and either tongue and groove or male and female facings should be provided. Several

nozzles, such as inlet, outlet, sight and light glasses, thermowell, handholes and manholes are provided to satisfy the process requirements. Bracket or column supports are generally satisfactory.

8.2 Materials of Construction

A number of metals can be used for construction of reaction vessels. The more common metals are low carbon steel, stainless steel and other alloy steels such as pastelloy. In special cases non-ferrous metals such as copper, nickel, aluminium, titanium are also used. At moderate pressures and temperatures, vessels can be made of glass reinforced polyesters, glass filled furons, phenolics and polyvinyl chloride. A saving in cost can be achieved by fabricating vessels from a low cost metal, with a cladding of corrosion resistant metals like stainless steel, nickel, inconel, monel, copper, etc. Vessels can also be lined with lead, rubber, glass and plastics to prevent corrosion. (For details see 6.4).

8.3 Agitation

Agitation of the reaction vessel contents is a requirement in a number of processing operations. Both heat and mass transfer are greatly influenced by agitation or mixing. An agitator, also known as a stirrer, produces high velocity liquid streams, which move through the vessel. As the high velocity streams come into contact with stagnant or slower moving liquid momentum transfer occurs. The classification of the agitation or mixing equipment is usually made on the basis of liquid viscosity, since viscosity is a major contribution to the forces tending to dampen flow through a mixing system. Details regarding agitation equipment and their applications are given in Chapter 14.

In a vertical cylindrical reactor the ratio of liquid depth to tank diameter (filling ratio) is normally between 0.5 and 1.5 and a value approximately equal to 1.0 is recommended for most purposes. When dispersing gas in a liquid, a filling ratio of about 2.0 is recommended in order to maintain a sufficiently long period of contact between the gas and the liquid. Flat-bottomed and cone-bottomed vessels have the disadvantage of low agitation efficiency in the corners formed between the walls and the bottom. This is particularly significant when mixing involves the suspension of heavy solids in the liquid. A vessel with dished bottom is always preferable, as its power consumption is low.

8.4 Classification of Reaction Vessels

Agitated reaction vessels may be divided into three main classes: batch reactors, continuous flow reactors and semi-batch reactors.

(a) *Batch reactor*—These are almost exclusively used for liquid phase reactions. The reactants are added to the empty vessel and the contents are removed after completion of the reaction. In this system, temperature and pressure as well as composition may vary with time.

(b) *Continuous flow reactor*—In these reactors the reactants flow continuously into the reactor and the products flow continuously out. Under ideal conditions, in a well-agitated system, a uniform concentration is maintained throughout the vessel.

(c) *Semi-batch reactor*—In these reactors one of the reactants is initially charged batchwise, while the other reactant is fed into the reactor continuously.

8.5 Heating Systems

Chemical reactions are accompanied by the absorption or liberation of heat. The reaction vessel must therefore be provided with the means of supplying or removing heat of reaction. The rate of heat transfer is a function of the physical properties of the agitated liquid and the heating and cooling medium, the vessel geometry, the material and the thickness of the vessel wall and the degree of agitation.

Heating systems for reaction vessels need special consideration. The devices used are either the direct or the indirect type. There are various electrical methods, which heat the vessel directly. The two most common methods are resistance heating and induction heating. These systems have a low overall efficiency and high operating costs. Indirect heat transfer systems are the most widely used. The heat is received from fluids such as steam, hot oil, hot water or air, molten salt mixtures, mercury and special organic compounds such as Dowtherm and Therminol. These organic compounds are high boiling materials, so that the heat transfer systems can be operated at low pressures. Both liquid and vapour phase heat transfer systems are used. In liquid systems, the heat transferred is from sensible heat. In vapour systems, the heat transferred is from latent heat; the entire heat transfer surface is therefore at a uniform temperature. In general, heat transfer coefficients are higher for condensing vapour systems than for liquid systems.

In indirect cooling systems, fluids employed are air, an evaporant such as liquid ammonia, water or brine, oil and organic compounds.

In the indirect heat transfer system, the fluid is supplied in either a jacket, which surrounds the vessels wall, or a coil which is placed inside the vessel. Heat transfer coefficients can be increased by increasing circulating fluid velocities. Similarly heat transfer coefficients can be increased by creating an agitation of the contents of the vessel.

There is no specific choice between a jacket or coil for a vessel carrying out an exothermic or endothermic reaction; although generally a jacket is installed when it is necessary to supply heat and a coil to remove heat. The reasons for this selection are that in the majority of cases heat is supplied by condensation of some vapour, and for a given heat transfer area there is a greater space for condensation in a jacket than in a coil. The greater the space the jacket provides greater is the ease of drainage of condensate. On the other hand a cooling coil is generally more suitable than a cooling jacket because the rate of heat transfer is greater under forced convectional conditions and greater turbulence can be achieved in the coolant liquid when it is pumped through a coil than when pumped through a jacket. The jacket can be constructed out of cheaper material like mild steel even when the vessel has to be of an expensive alloy.

8.5.1 Jackets

A plain jacket (Fig. 8.1 (a)) is used for steam. The space between the vessel wall and the jacket shell should be narrow, for

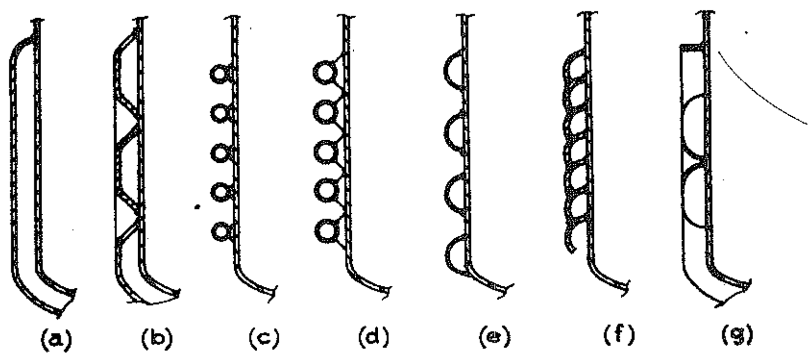


Fig. 8.1 Types of jackets (a) plain (b) channel (c) tube (d) tube with copper strip support (e) half coil (f) overlapped cut coil (g) dimpled construction

obtaining good heat transfer rates. The dimpled construction shown in Fig. 8.1 (g) helps to reduce the thickness of the shell. This construction is particularly suitable for large vessels, because of the reduction in weight. The dimpled jacket is generally used for condensing vapours such as steam or Dowtherm. High velocities of circulating fluids can be obtained by use of different types of jacket constructions, such as channel or coiled jackets [Fig. 8.1 (b) (c) (d) (e) (f)].

8.5.2 Coils

Coils are used for heating or cooling by immersing them in the contents of the vessel (Fig. 8.2). Such coils are formed from a tubing by shaping them in the form of a helical or double helical coil. A pancake type of coil, which is a spiral rolled in a single plane so as to lie horizontally near the bottom of the vessel is also used to transfer heat by free convection. Tubes can also be arranged vertically in the vessel, which serve the dual purpose of heat transfer and baffling.

8.6 Design Considerations

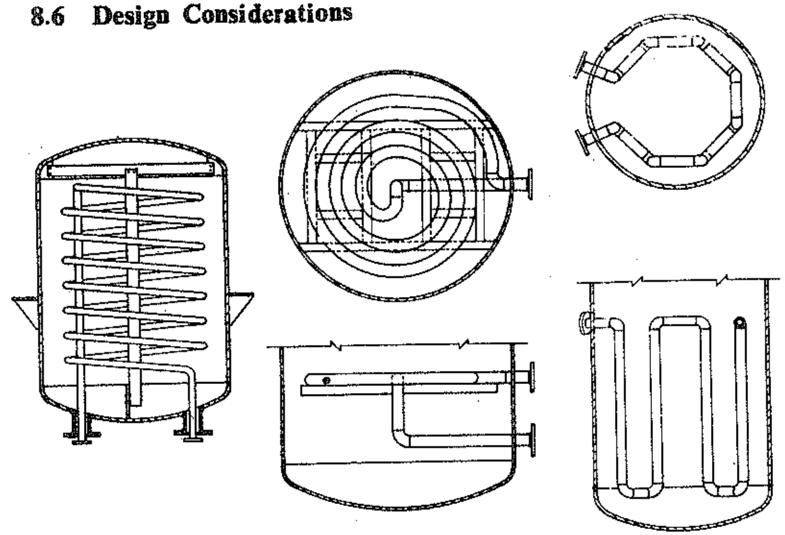


Fig. 8.2 Coils (a) helical coil (b) spiral coil (c) vertical tubes

Design of the reaction vessel is based on pressure and temperature conditions. It is essentially a pressure vessel with heads and nozzles. A jacket construction is an additional feature, involving operation of external pressure on the vessel wall. The reaction vessel is therefore, designed for both internal and external pressures operating independently and the higher value of the wall thickness is accepted as satisfactory. The vessel shell is subjected to external pressure and care must be taken in the design to prevent collapse (see para 6. 8.1.3.). The methods adopted are :

(a) Making the shell thick enough in proportion to its diameter and length so that it is self-supporting.

(b) Using stay bolts for attaching the inner shell to the outer jacket.

(c) Using stiffening rings or corrugations in the shell of the vessel.

The procedure indicated in Chapter 6 is followed to size the various components of the vessel. The overall dimensions of the vessel, the size and position of nozzles are determined by the process conditions.

8.6.1 Jacket Design

A plain jacket is the simplest arrangement for heating or cooling. The jacket is generally made of low carbon steel and is designed for internal operating pressure of the heating fluid at the appropriate temperature. Various methods are adopted to attach the jacket to the vessel wall. A common method is to use two rings of square or rectangular section (Fig. 8.3), one at

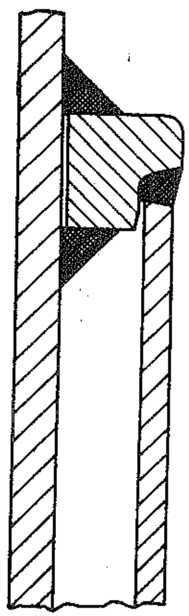


Fig. 8.3 Plain jacket welded to shell by ring

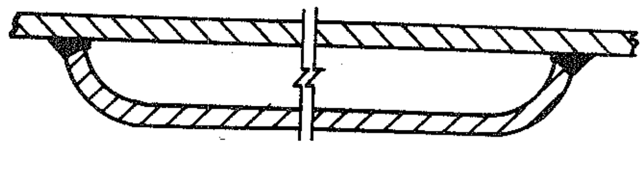


Fig. 8.4 Plain jacket welded to shell

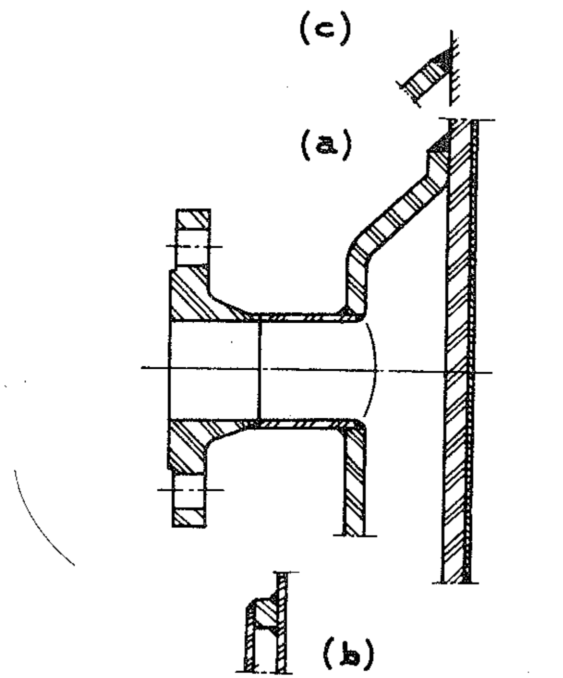


Fig. 8.5 Jacket welded to pressure vessel shell
(a) for low fluid pressure (b) for high fluid pressure
(c) for medium fluid pressure

the top and the other at the bottom to which the jacket and the vessel are securely welded. Figs. 8.4, 8.5 show an alternate method of attachment of jacket to shell. Figs. 8.6, 8.7 show attachment of jacket for the top head. Fig. 8.8 shows a detachable jacket. Most frequently the jacket extends over the base of the reactor and in such cases provision must be made for emptying the vessel as well as for removing the heating fluid from the

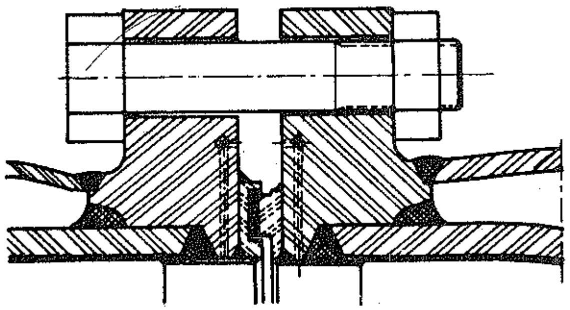


Fig. 8.6 Stainless steel lined flanged joint of a reaction vessel with jacket for shell and head, welded to flanges

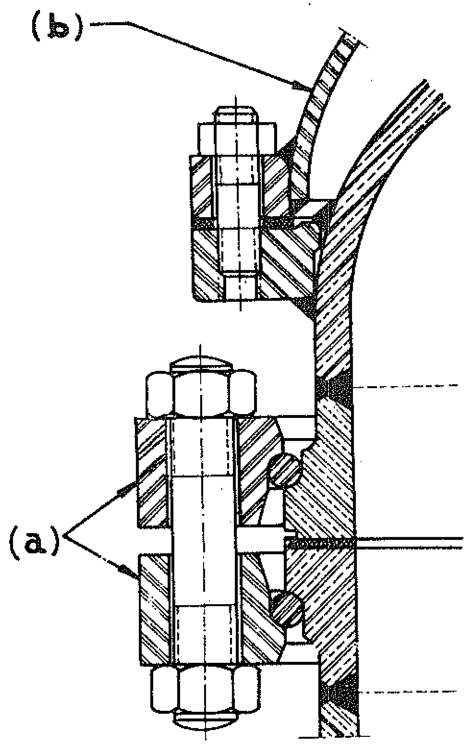


Fig. 8.7 Detachable joint for head jacket (a) loose flanges (b) head jacket

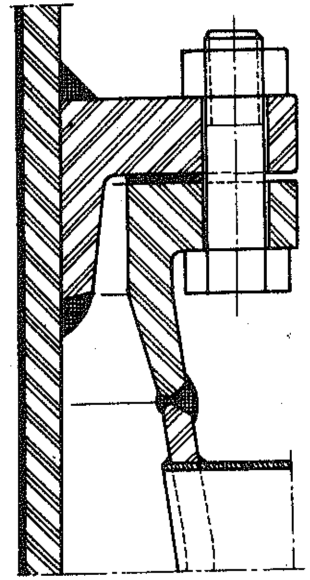


Fig. 8.8 Detachable jacket for vessel shell

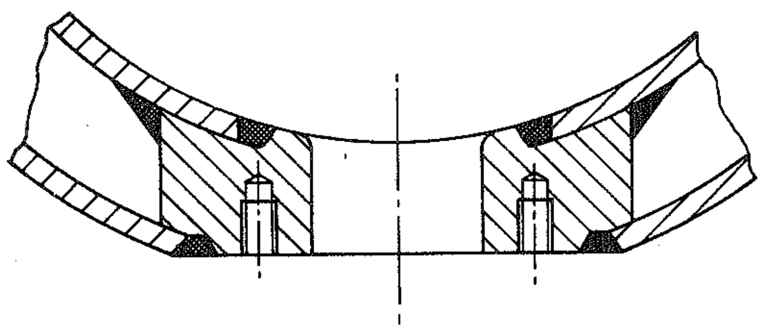


Fig. 8.9 Bottom outlet in a pad welded to head and jacket

jacket. The design of the drainage connection of the vessel is more complicated than the jacket because there will be differential expansion between jacket and vessel which, if not provided for, could lead to rupture of the vessel. Figs. 8.9, 8.10, 8.11,

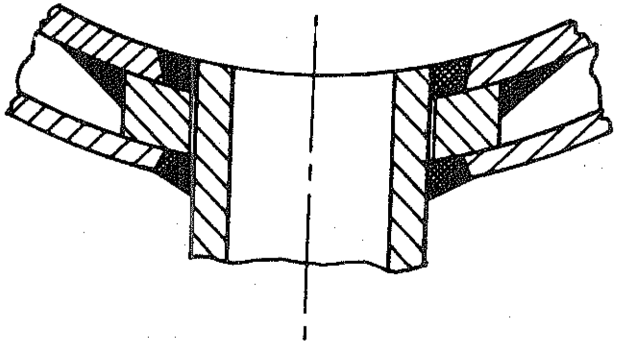


Fig. 8.10 Bottom outlet pipe with reinforcing ring welded to head and jacket

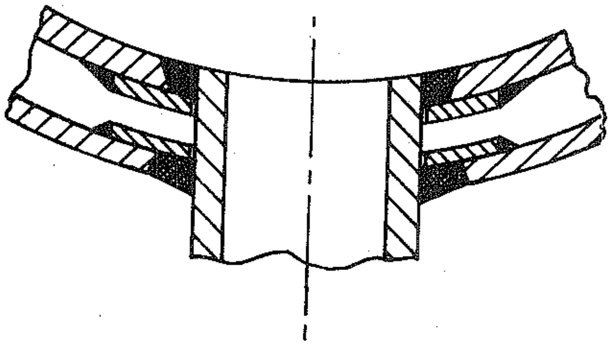


Fig. 8.11 Bottom outlet pipe with separate reinforcements for head and jacket

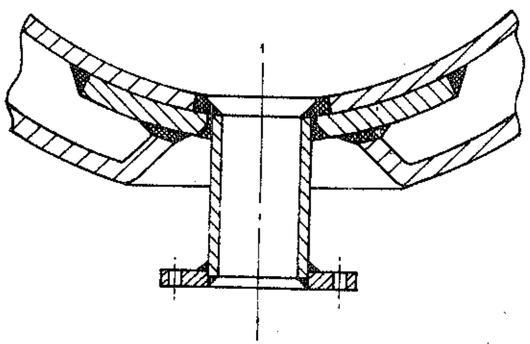


Fig. 8.12 Bottom outlet pipe with reinforcing ring welded to head and jacket welded to ring

8.12 and 8.13 show a few examples of the different methods suggested for jacket connections for the bottom head of the reactor. The thick vessel wall required for high jacket pressure makes the vessel expensive. The heat transfer through the thick wall will also be less. For higher jacket pressures it is therefore preferable to use other jacket designs. The dimpled jacket construction is formed by plug welds, which act as 'stays' and help to reduce the vessel wall thickness. Special precautions will have to be taken to prevent fouling, stress and cell corrosion at the welds.

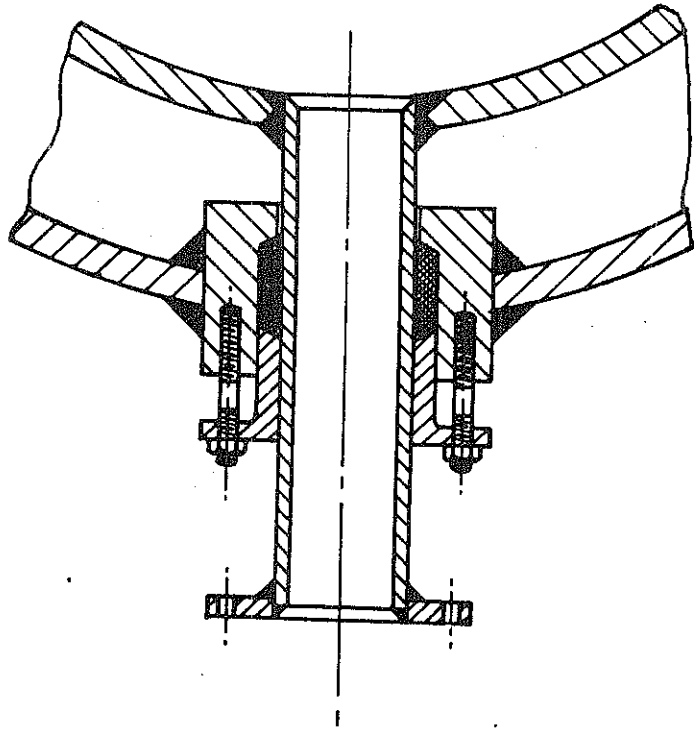
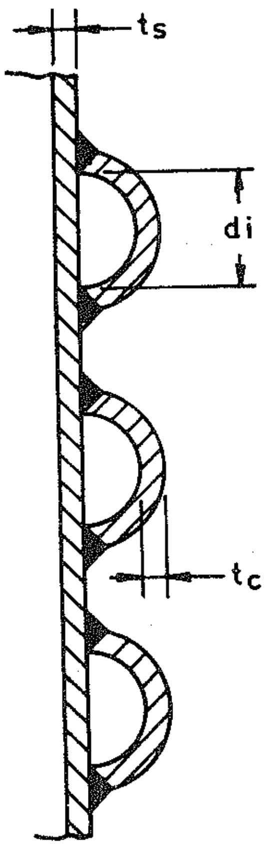


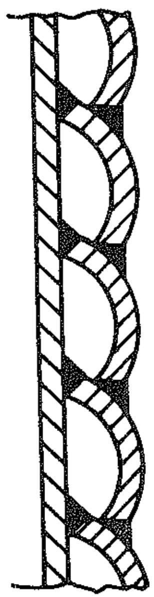
Fig. 8.13 Bottom outlet pipe with stuffing box type of packing gland for differential expansion between head and jacket

8.6.2 Coil and Channel Design

The half coil or channel construction shown in Fig. 8.1 (b) and (c) is formed by a continuous spiral of half pipe or channel section, attached to the vessel wall by continuous fillet welding with full penetration. Fig. 8.14 shows details of a portion of the vessel shell with half coil attached to it; while Fig. 8.15 shows a channel jacket.



(a)



(b)

Fig. 8.14 (a) Half coil jacket welded to shell
(b) overlapped cut coil welded to shell

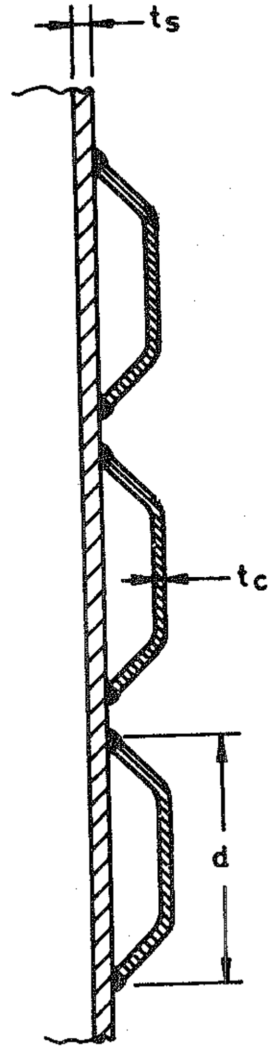


Fig. 8.15 Channel jacket welded to shell

8.6.2.1 DESIGN OF VESSEL SHELL WITH HALF COIL

A half coil is a part of a taurus, for which the circumferential stress which occurs at the junction with the shell is given by

$$f_{pc}=\frac{pd_i}{2t_c} \tag{8.1}$$

and the longitudinal (axial) stress is given by

$$fa_c=\frac{pd_i}{4t_c+2.5t_s} \tag{8.2}$$

where

- p—design pressure inside the half coil
- d_i—internal diameter of the half coil
- t_c—thickness of half coil
- t_s—thickness of shell

the thickness of half coil is calculated from equation 8.1 above and by addition of usual corrosion allowance.

The total circumferential stress f_{ps} in the shell is the sum of the circumferential stress f_p due to pressure in the vessel and the longitudinal stress fa_c in the coil caused by coil pressure.

$$f_{ps}=f_p+fa_c$$

$$=\frac{PD_i}{2t_s}+\frac{pd_i}{4t_c+2.5t_s} \tag{8.3}$$

The total longitudinal (axial) stress fa_s, in the vessel shell is made up of

(1) longitudinal stress due to pressure in the vessel

(2) longitudinal stress fa_c due to coil pressure

(3) the bending stress caused by the distortion of the shell at the junction with the coil.

$$fa_s=f_a+fa_c+f_b$$

$$=\frac{PD_i}{4t_s}+\frac{pd_i}{2t_s}+\frac{2\triangle Pd_o^2}{3t_s^2} \tag{8.4}$$

where

- p—internal pressure in the vessel
- D_i—internal diameter of shell
- $\triangle p$—maximum differential pressure between coil and shell
- d_o—external diameter of half coil.

The vessel shell thickness can be calculated from equation 6.3 and by the addition of corrosion allowance. The longitudinal

may then be checked. It is further necessary to check the buckling of the shell portion closed off by a coil with maximum differential pressure $\triangle p$.

8.6.2.2 DESIGN OF VESSEL SHELL WITH CHANNEL JACKET

Fig. 8.15 shows a part of a channel jacket. The thickness t_s of the vessel shell wall or t_c of the channel can be determined on the basis of uniformly loaded flat plate restrained at the ends. The thickness may be given as

$$t_s=d\sqrt{\frac{k_1p}{f_1}}+c \tag{8.5}$$

$$t_c=d\sqrt{\frac{k_2p}{f_2}}+c \tag{8.6}$$

where d— as shown in Fig. 8.15

- p— design jacket pressure
- f_1, f_2— stresses in the materials at the appropriate temperature
- k_1— 0.167 ; k_2=0.12
- c— corrosion allowance

For higher jacket pressures the width 'd' can be reduced to obtain a reduction in thickness 't'.

It is desirable to ensure a sound weld between the shell and the surrounding half coil or channel. A minimum thickness (t_s or t_c) of at least 2 mm is recommended for making a satisfactory joint.

It is extremely difficult to cut a half coil of channel to suit the vessel bottom. It may be easier to arrange a pattern of coils radially like the spokes of a wheel.

8.7 Numerical Problem

Reaction Vessel

Data

Vessel shell internal diamater	— 2130 mm
Jacket internal diameter	— 2260 mm
Jacket length	— 2500 mm
Diameter of half coil or width of channel jacket	— 100 mm

Flanged and Dished head

internal diameter	— 2130 mm
crown radius	— 2130 mm
knuckle radius	— 128 mm
straight flange length	— 60 mm
Internal pressure (shell)	— 5.5 kg/cm²
Internal pressure (jacket)	— 3.5 kg/cm²
Temperature	150°C
Material—open hearth steel (IS—2002)	
Allowable stress	— 9.8 kg/mm²
Modulus of elasticity (at 200°C)	—19.00×10³ kg/mm²
Poisson's Ratio	— 0.3

8.7.1 Shell with Plain Jacket

Thickness of shell

(i) internal pressure

Design pressure—5.5+10%=6.05 kg/cm²

From equation (6.3) $t_s=\dfrac{pD_i}{2fJ-p}=\dfrac{6.05\times2130}{2\times9.8\times10^2\times0.85-6.05}$

$=7.8$ mm

Use 9 mm thickness including corrosion allowance.

(ii) External pressure. Design pressure—3.5+10%=3.85 kg/cm²

To calculate the external critical pressure use a thickness of shell of 7.8 mm already calculated above. From equation 6.14 (b) the critical pressure is given by (the effective jacket length is taken as 2500+100=2600 mm)

$$p_c=\frac{2.42\times1950\times10^3}{(1-0.3^2)^{\frac{3}{4}}}\times\frac{\left(\dfrac{7.8}{2196}\right)^{\frac{5}{2}}}{\left(\dfrac{2600}{2146}\right)-0.45\left(\dfrac{7.8}{2148}\right)^{\frac{1}{2}}}$$

$=3.4$ kg/cm²

p_a (Allowable)$=\dfrac{3.4}{4}=0.85$ kg/cm²

Calculation of allowable external working pressure from IS-2825 Pressure vessel code

Appendix F

$$\frac{L}{D_o}=\frac{2600}{(2130+2\times7.8)}=\frac{2600}{2146}=1.21$$

$$\frac{D_o}{t}=\frac{2146}{7.6}=282$$

Factor B—3200 (at 150°C)

$$p_a=\frac{B}{14.22\dfrac{D_o}{t}}=\frac{3200}{14.22\times282}=0.8\text{ kg/cm}^2$$

The values of allowable external pressures obtained by both the methods are less than the design pressure. It is therefore, proposed to increase the thickness from 7.8 mm to 9 mm and use 6 stiffening rings each at a distance of 450 mm.

From equation 6.14 (b)

$$p_c=\frac{2.42\times1900\times10^3}{(10-.3^2)^{\frac{3}{4}}}\times\frac{\left(\dfrac{9}{2148}\right)^{\frac{5}{2}}}{\left(\dfrac{450}{2148}\right)-0.45\left(\dfrac{9}{2148}\right)^{\frac{1}{2}}}$$

$=31.09$ kg/cm²

$$p_a=\frac{31.09}{4}=7.77\text{ kg/cm}^2.$$

Referring to IS—2825 Appendix F

$$\frac{L}{D_o}=\frac{450}{2148}=0.209$$

$$\frac{D_o}{t}=\frac{2148}{9}=239$$

Factor $B=13000$

$$p_a=\frac{13000}{14.22\times239}=3.85\text{ kg/cm}^2$$

which is satisfactory. Use 10 mm thick shell. The equation 6.14 (b) gives a much higher value of the allowable pressure. The equation is therefore only used for approximate calculation.

Stiffening Ring

Required movement of inertia of the stiffening ring as per equation 6.15.

$$I = \frac{p_c D_o^3 L}{24 E} \qquad p_c = 4 \times p_a$$

$$= 4 \times 3.85$$

$$= 15.40$$

$$= \frac{15.4 \times 2148^3 \times 450}{24 \times 1900 \times 10^3}$$

$$= 1.51 \times 10^6 \text{ mm}^4.$$

The value of I is reduced by 30% to take into account the resistance of the shell

$$I = 1.053 \times 10^6 \text{ mm}^4$$

Use equal angle IS-A 2020—(size 20×20 mm thickness —3 mm)

$$I_{xx} = I_{yy} = 0.4 \text{ cm}^4$$

which is the smallest size of angle available.

8.7.2 Thickness of Jacket

Internal design pressure— 3.5+10%

$$= 3.85 \text{ kg/cm}^2$$

$$t_j = \frac{pD_i}{2fJ - p} = \frac{3.85 \times 226D}{2 \times 9.8 \times 10^2 \times 0} 85 - 3.85$$

$$= 5.25 \text{ mm.}$$

Use 6 mm thickness including corrosion allowance.

8.7.3 Head Thickness

(i) Internal pressure

From equation 6.23

$$t_h = \frac{p\, R_c\, W}{2\, f\, J}$$

$$= \frac{6.05 \times 2130 \times 1.77}{2 \times 9.8 \times 10^2 \times 0.85}$$

$$= 13.6 \text{ mm}$$

where, $$W = \frac{1}{4}\left(3 + \sqrt{\frac{R_c}{R_1}}\right)$$

$$= \frac{1}{4}\left(3 + \sqrt{\frac{2130}{128}}\right)$$

$$= 1.77$$

Use 15 mm thickness including corrosion allowance.

(ii) External pressure

From equation 6.26

$$t_h = 4.4 \times 2130 \times [3(1 - 0.3^2)]^{\frac{1}{4}} \times \left(\frac{3.85}{2 \times 1900 \times 10^3}\right)^{\frac{1}{2}}$$

$$= 12.1 \text{ mm}$$

According to IS-2825, para 3.4.6.1, the value obtained for head thickness under external pressure is less than 13.6 mm. Therefore a thickness of 15 mm is satisfactory.

8.7.4 Vessel Shell with Half Coil Jacket

Diameter of half coil (dia.)— 100 mm (Fig. 8.14)

From equation 8.2

(Design pressure—3.5+1%= 3.85 kg/cm²)

$$t_c = \frac{3.85 \times 100}{2 \times 9.8 \times 10^2} = 0.196 \text{ mm}$$

Use minimum 2 mm thickness.

Circumferential stress in the shell (equation 8.3)—

(shell thickness 7.8 mm)

$$f_{ps} = \frac{6.05 \times 2130}{2 \times 7.8} = \frac{3.85 \times 100}{4 \times 0.147 + 2.5 \times 7.8}$$

$$= 828 + 18.4$$

$$= 846.4 \text{ kg/cm}^2$$

which is less than the allowable stress value of 980 kg/cm². The longitudinal stress (equation 8.4)

$$f_{as} = \frac{6.05\times2130}{4\times7.8} + \frac{3.85\times10}{2\times7.8} + \frac{2\times3.85\times(104)^2}{3\times(7.8)^2}$$

$$= 414 + 24.7 + 456$$

$$= 894.7 \text{ kg/cm}^2$$

which is less than the allowable stress value of 980 kg/cm². Therefore use 9 mm shell thickness as required to withstand internal pressure and allowance for corrosion.

8.7.5 Vessel Shell with Channel Type Jacket

$$d = 100 \text{ mm} \qquad \text{(Fig. 8.15)}$$

From equation 8.5 (Design pressure—3.85 kg/cm²)

$$t_s = d\sqrt{\frac{k_1 p}{f}} + c$$

$$= 100\ \frac{0.167\times3.85}{980} + c$$

$$= 100\times2.56\times10^{-2} + c$$

$$= 2.56 \text{ mm} + c$$

$$= 4 \text{ mm}$$

Use 9 mm shell thickness required to withstand internal pressure and allowance for corrosion.

From equation 8.6, channel jacket thickness

$$t_c = d\sqrt{\frac{k_2 p}{f}} + c$$

$$= 100\ \frac{0.12\times3.85}{980}$$

$$= 2.17 + c$$

$$= 3 \text{ mm}$$

Reading References

Perry, J.H., *Chemical Engineers' Handbook*, 5th Edn., McGraw-Hill, New York (1973).

Feichtinger C.A., *New Idea in Jacketed Vessels*, Chem. Engg., 67 (18), 127 (1960).

Kneale, M., *Design of Vessels with Half Coils*, Trans. Inst. Chem. Eng., 47, 7279 (1969).

Markovitz, R.E., *Selecting Best Vessel Jacket*, Chem. Eng., 78 (26), 156 (1971).

Kern, D.Q., *Process Heat Transfer*, McGraw-Hill, New York (1950).

Riegel, E.R., *Chemical Process Machinery*, Reinhold, New York (1953).

Barona, N. and Prengle, H.W., *Special Report, Reactor Design*, Hydrocarbon Processsing 63 (1973) 52 (3), 63 (1973).

CHAPTER 9

Heat Exchangers

9.1 Introduction

A heat exchanger is a device, which is used for transferring heat from one fluid to another through a separating wall. These can be classified according to the process of heat transfer, mechanical construction and principal material of construction. The classification is not rigid, and some of the heat exchangers may fall under more than one category. The following are some of the more common types of heat exchangers.

9.2 Types of Heat Exchangers

9.2.1 Double Pipe Heat Exchanger

This (Fig. 9.1) is used particularly when the flow rates are low, and when the temperature range is relatively high. In this type one fluid flows inside a pipe, while a second fluid flows either co-or-counter currently in the annulus between a large pipe and the outerside of the inner pipe carrying the first fluid. The components of the heat exchanger unit consist of concentric pipes, connecting tees and return bends.

9.2.2 Shell and Tube Heat Exchangers

These are most widely used types of heat exchangers (Fig. 9.2). The equipment consists of a number of parallel tubes enclosed in a relatively close fitting cylindrical shell. One fluid flows inside the tube and is called the tube side fluid, the other flows outside the tubes and is called the shell side fluid. If none of the fluids condenses or evaporates the unit is known as a 'heat exchanger'. If one of the fluids, either

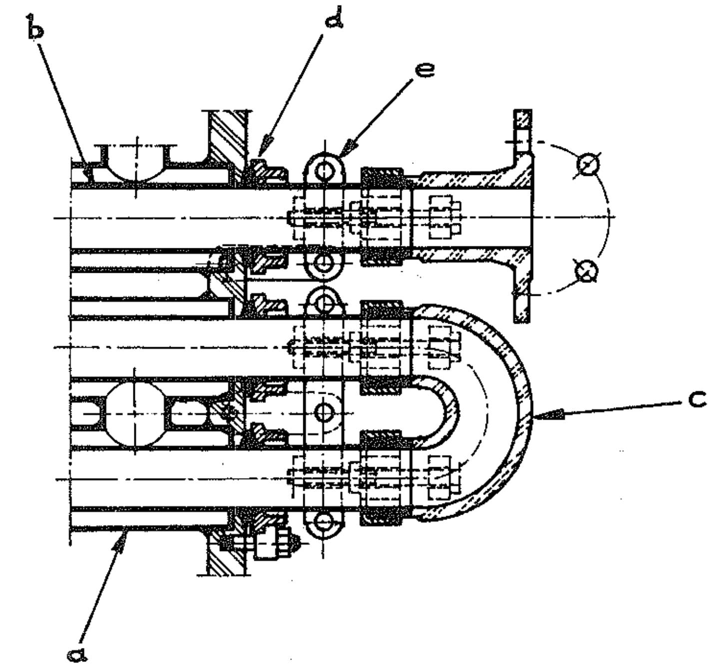
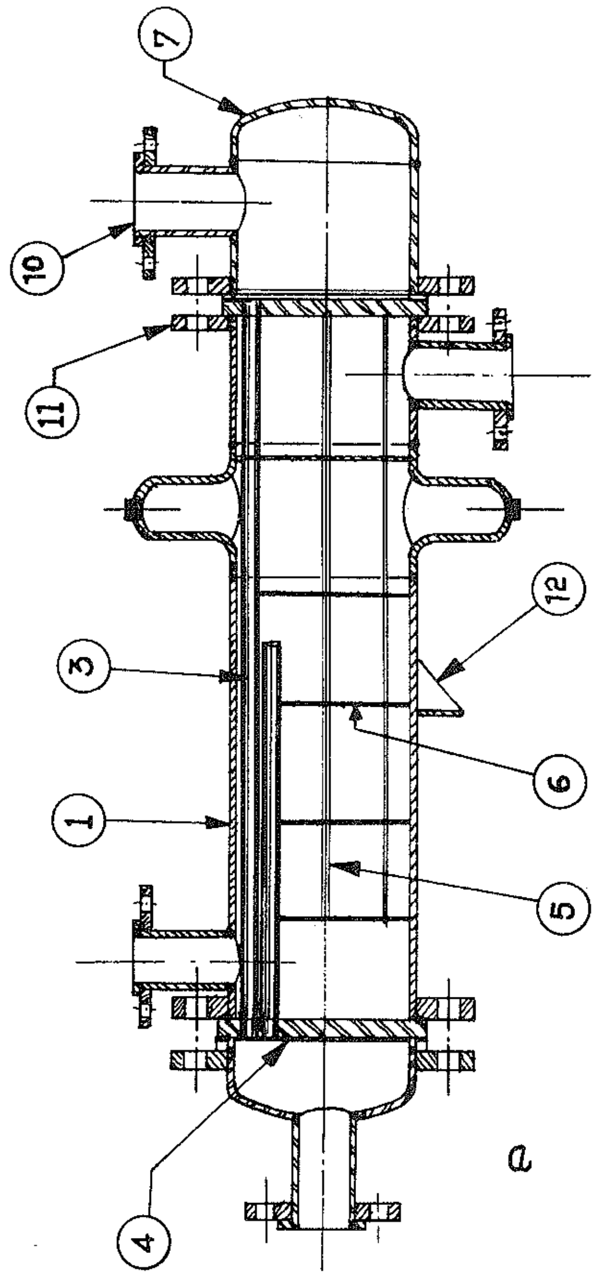
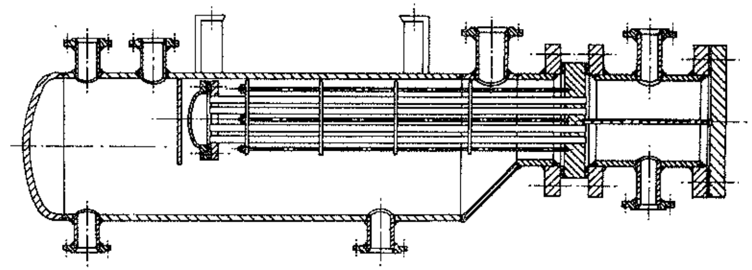


Fig. 9.1 Double pipe heat exchanger
(a) outer pipe (b) inner pipe (c) return bend (d) gland with packing (e) support bracket

in the tube or the shell condenses, the unit is known as a 'condenser' or as a 'heater', depending on whether the primary purpose of the unit is to condense one fluid or to heat the other. Similarly, if one of the fluids evaporates, the unit is designated as an evaporator or as a cooler, depending on whether the primary purpose is to evaporate one fluid or to cool the other. In the case of vapour-in-tube condensers, the vapour distribution is uniform, pressure drop is reasonable and vapour sub-cooling is readily accomplished. For service at high vacuum conditions, horizontal vapour-in-shell condenser is a preferred design, because at high vacuum the vapour lines are large, vapour temperatures are usually high and condensate subcooling is not too important.



a



e

Fig. 9.2 Types of shell and tube heat exchangers
(a) fixed tube sheet (b) outside packed floating head
(c) internal floating head (d) U-tube type
(e) reboiler (with internal floating head)

9.2.3 Special Type of Heat Exchangers

These are used in special cases where the standard forms of tubular exchangers are unsatisfactory.

(a) *Pipe coils*—These coils of spiral or helical shape are used for cooling purposes, especially at high temperature and pressure applications. Such coils are immersed in a tank of water or sprayed with water. The pressure drop in the coils is high, but the construction is simple and less expensive.

(b) *Spiral heat exchanger* (i) (Fig. 9.3). It is constructed by winding spirally, two plates one inside the other, so that two separate, rectangular narrow passages are formed through which the two fluids flow counter-currently. The end walls of the passages are formed by U-shaped packing strips of rubber, kept in position by means of flat steel covers. The hot liquid enters at the centre, flows through the inner passage, and leaves at the periphery. The cold liquid enters at the periphery flows through the outer passage, and leaves out at the centre. The advantages are high turbulance, compactness, no expansion problems and ease of cleaning.

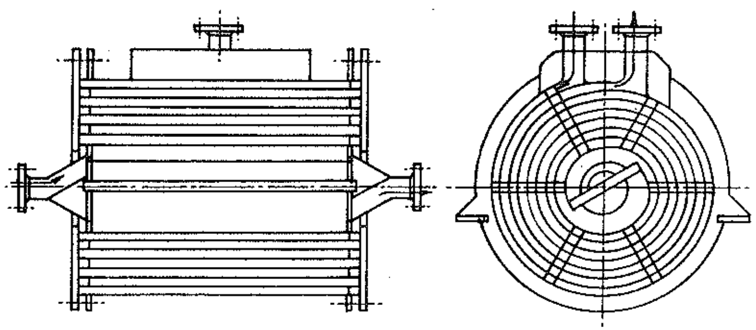


Fig. 9.3 Spiral heat exchanger

(c) *Plate type exchanger.*—This is constructed by a series of corrugated parallel plates held firmly together between frames. The heat transfer surfaces consist of the adjacent plates. Gaskets cemented to the plates help to form separate channels for the two fluids. The two liquids travel in counter-current directions, and the heat transfer takes place through the plate. Headers connecting the two liquids connect the alternate plate compartments through corner plates. The whole assembly is bolted together with stay bolts and end plates. This has the advantage of being able to increase the area by adding extra plates. The advantages are high rates of heat transfer, low pressure drop and easy cleaning and replacement of plates. This type of exchanger is used in food, dairy and other industries where even the slightest contamination of the products due to leak cannot be tolerated.

(d) *Finned tube exchanger.*—In case of heating air or gas, where the heat transfer coefficients are low, the surface area can be increased by use of fins either in transverse or longitudinal form. Transverse finned tubes are used for cooling process streams by air. Longitudinal finned tubes units are used at high temperature and high pressure service, where heat transfer rates are low.

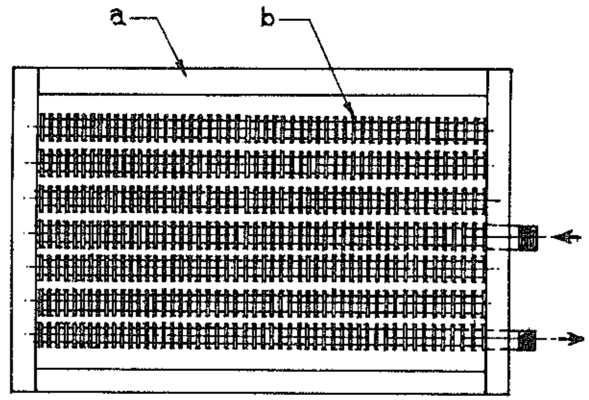


Fig. 9.4 Finned tube heat exchanger (a) box (b) finned tubes

9.3 Design of Shell and Tube Heat Exchangers

Among the various types of heat exchangers mentioned above the shell and tube type of heat exchangers are considered in detail. Due to the wide usage, their design and construction features are standardised. A number of codes such as Indian Standard 4503, standards of Tubular Exchanger Manufacturers Association, B.S. 3274 specify the design procedure and details of construction. It is expected that the design of the heat exchanger will satisfy the code requirements.

9.3.1 Fluid Flow Arrangement

One of the methods of classification of heat exchangers is on the basis of the configuration of the fluid flow paths through the heat exchangers. The four most common type of flow paths are:

(a) co-current or parallel flow units, where the two fluid streams enter together at one end, flow through in the same direction and leave together at the other end.

(b) Counter current or counter flow units in which the two fluid streams move in opposite directions.

(c) Single path cross flow units in which one fluid moves through the heat exchanger matrix at right angles to the flow path of the other fluid.

(d) Multipass cross flow units in which one fluid stream shuffles back and forth across the flow path of the other fluid stream usually giving a cross flow approximating to counter-flow.

9.3.2 Classification of Shell and Tube Type Heat Exchangers

These types of heat exchangers are built of a shell in which a number of round tubes are mounted by means of tube plates. Many variations of this basic type are available. The difference lies mainly in the detailed features of construction and provisions for differential thermal expansion between the tubes and the shells. According to the mechanical configuration these exchangers can be classified as (Fig. 9.2)

(a) Fixed tube sheet
(b) Outside packed floating head
(c) Internal floating head
(d) U-tube type
(e) Reboiler (with internal floating head or U-tube type).

The main components of the above exchangers are (1) shell (2) shell cover (3) tubes (4) tube sheet (5) tie rods and spacers (6) baffles (7) channel (8) channel cover (9) pass partitions (10) nozzles (11) flanges (12) supports.

9.3.3 Materials of Construction

Exchangers are generally constructed of carbon and low alloy steels. Carbon steels are recommended upto 540°C, while low alloy steels upto 590°C. In such exchangers the tubes may be of non-ferrous metals such as brass, copper, nickel, etc. For handling corrosive fluids non-ferrous alloys or high alloy steels are used. Such metals are copper, cupro-nickel, nickel, monel, inconel, aluminium, stainless steel, etc. In special cases heat exchangers are made of graphite, teflon and glass. For low temperature service, austenitic stainless steel or non-ferrous

alloys are used. For high temperature service, stabilized grade of low carbon steel or alloys of steel are used. Table 9.1 gives fluid temperature limitations for pressure parts of different materials.

Table 9.1
Fluid Temperature Limitations for Pressure Parts

Material	Maximum Permissible Temperature °C
Carbon steel	540
C-M_o steel	590
Cr-M_o steel	650
Low alloy steel (less than 6 per cent chromium)	590
Alloy steel (less than 17 per cent chromium)	590
Austenitic Cr-Ni steel	650
Cast iron	200
Brass	200

9.3.4 Design Pressure

For determining the minimum thickness of various components, the design pressure is obtained normally by adding a minimum of 5% to the maximum working pressure. This is finally decided by the severity of working conditions.

9.3.5 Design Temperature

It is taken as 10°C higher than the maximum temperature that any part of the exchanger is likely to attain in course of operation.

9.3.6 Corrosion Allowance

For carbon and cast iron pressure parts the corrosion allowance is 1.5 mm except for tubes. For severe conditions the allowance is 3 mm. For non-ferrous and stainless steel parts no allowance may be required. For internal covers and tube sheets, corrosion allowance is provided on both sides. Non-pressure parts such as tie rods, spacers, baffles, supports need not have any allowance for corrosion.

9.3.7 SHELL

The internal diameter of the shell is determined from the total area occupied by tubes. The procedure for calculation of shell diameter is indicated later. The ratio of tube length to shell

Table 9.2

Minimum Shell Thicknesses where Severe Conditions are not Expected

	Minimum Thickness in mm						
Nominal Diameter	Cast Iron	Carbon Steel (Including Corrosion Allowance)	Copper and Copper Alloys	Aluminium and Aluminium Alloys	Austenitic Stainless Steel	Nickel	Monel Inconel
150	10	5	3.2	5	3.2	3.2	3.2
200	10	6.3	3.2	5	3.2	3.2	3.2
250	10	6.3	3.2	5	3.2	3.2	3.2
300	13	6.3	3.2	5	3.2	3.2	3.2
350	13	6.3	5	5	3.2	5	3.2
400	13	6.3	5	6.3	3.2	5	3.2
500	13	8	6.3	8	3.2	6.3	3.2
600	16	8	6.3	8	5	6.3	5
700	16	10	8.3	10	5	8	5
800	16	10	10	11.2	6.3	8	6.3
900	19	10	10	11.2	6.3	10	6.3
1000	19	10	11.2	12.5	6.3	11.2	6.3
1100	22	11.2	11.2	14	6.3	11.2	6.3

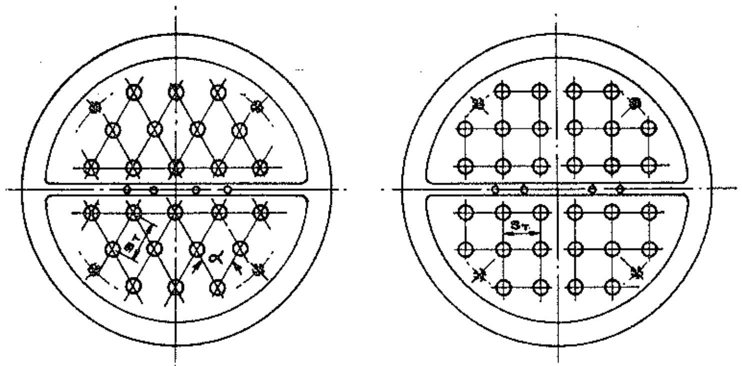
Note : The thickness values are exclusive of the corrosion allowance.

diameter for liquid-liquid exchangers is in the range of 4 : 1 to 8 : 1. For gas to gas exchangers this ratio may be even less than 4 : 1. The shell may be cut to the required length from a standard pipe upto 60 cm diameter or fabricated from a plate. The thickness of the shell may be calculated as a cylinder with internal pressure or vacuum depending on the working pressure in the shell (equation 6.3). The minimum thickness including corrosion allowance of the shell, made of carbon steel varies from 5 mm to 11 mm depending on the diameter. Table 9.2 gives minimum thickness for different materials.

For determination of shell thickness, nozzle thickness and flanges equations applicable to pressure vessel design are used. Joint efficiency is also to be taken into account on the same basis (see chapter 6).

9.3.8 TUBES

The two basic types of tubes are plain and finned, external or internal. The plain tube is used in the normal heat exchanger units. Finned tubes give an increased heat transfer surface. The duplex tube is a tube within a tube snugly fitted by drawing the outer tube on to the inner tube. This tube is useful in conditions where the shell side fluid is not compatible with the



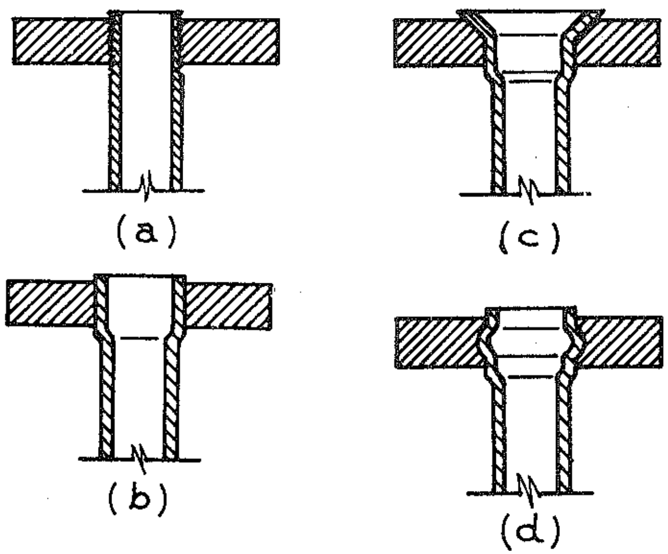
(a) (b)

Fig. 9.5 Tube sheet
(a) triangular pitch of tubes (b) square pitch of tube

material satisfactory for the tube side fluid. The outside diameter of the tubes varies from 6 mm to 40 mm, with thickness depending on the materials of construction and diameters. Tube lengths used are 0.5, 2.5, 3, 4, 5 and 6 metres. The tubes are laid out either on an equilateral triangular pitch or square pitch. The minimum pitch is 1.25 times the outside diameter of tubes. When tubes are on a square pitch a minimum cleaning lane of 6.5 mm is provided. (Fig. 9.5).

9.3.9 Tube Sheet

It is essentially a flat plate with a provision for making a gasketed joint, around the periphery. A large number of holes are drilled in the tube sheet according to the pitch requirements of the tubes (Fig. 9.5). The common methods of fixing the tubes in these holes consist of expanding the ends of the tubes, located in tube sheet holes by means of a rotating, roller expanding tool. The holes in the tube sheet are drilled undersize and are then reamed to a diameter slightly larger than the outside diameter of the tube. One or two grooves are cut inside the holes in order to prevent leakage and to increase the strength of the joints (Fig. 9.6).



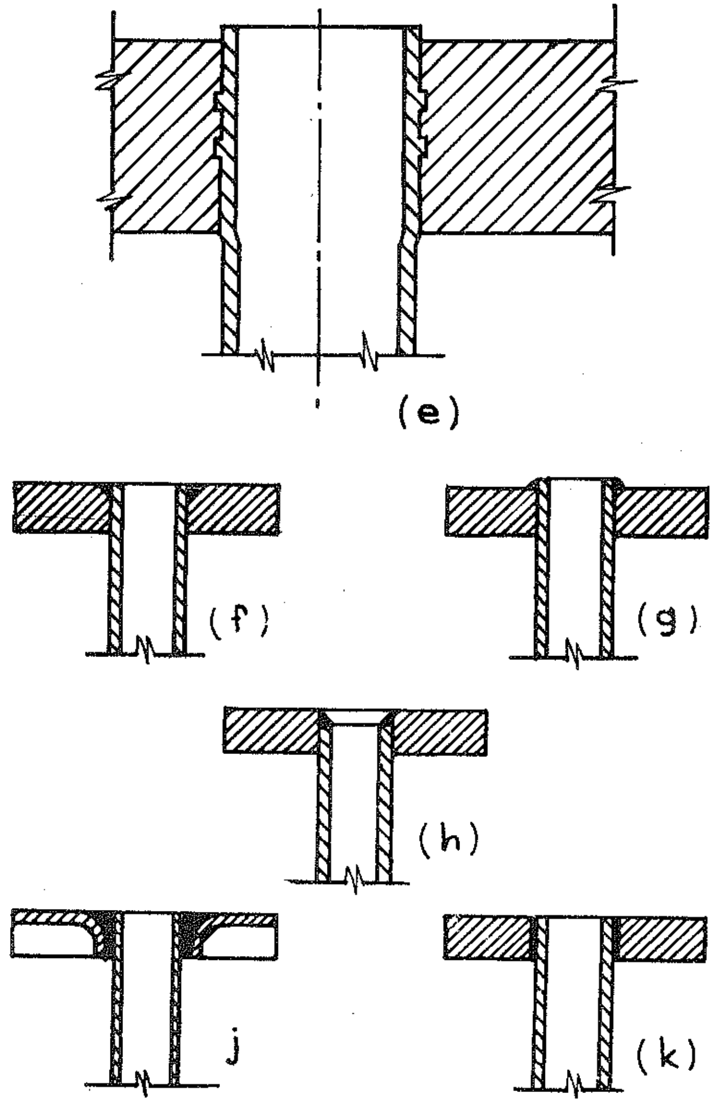


Fig. 9.6 Methods of fixing tubes to tube sheet
(a) screwed tube (b), (c), (d) and (e) rolled tubes (f), (g) and
(h) welded tubes (j) and (k) brazed tubes

Tubes are sometimes soldered or brazed into the tube sheet holes, particularly when the tubes have a small diameter. Tubes can also be welded in tube sheet holes but this method makes the tube replacements very difficult. A minimum tube sheet thickness is necessary in order to fix the tubes to the tube sheet.

The effective thickness of the tube sheet shall be the thickness measured at the bottom of the pass partition groove minus shell side corrosion allowance and the corrosion allowance on the tube side in excess of the groove depth. It is determined by taking into consideration the effect of working pressure, on a flat plate with holes as also the effect of expansion of tube and shell due to a rise in temperature. The detailed calculations are rather complex. A simpler equation for the effective tube sheet thickness is as follows.

$$t_{ts}=FG\sqrt{\frac{0.25\,p}{f}} \tag{9.1}$$

t_{ts}—effective thickness of tube sheet

p—design pressure

f—allowable stress at appropriate temperature

G—mean diameter of gasket

The value of F varies according to the type of heat exchanger. In most cases it is taken as 1. For a U-tube exchanger it is taken as 1.25.

For fixed tube sheet exchanger, the thickness is calculated for both tube side and shell side conditions, using whichever is greater.

For the tube side pressure

$$F=\sqrt{\frac{2+k}{2+3k}} \tag{9.2}$$

and for shell side pressure

$$F=\sqrt{\frac{k}{2+3k}}$$

where

$$k=\frac{E_s t_s\,(D_o-t_s)}{E_t N t_t\,(d_o-t_t)} \tag{9.3}$$

E_s—elastic modulus of shell

E_t— ,, ,, ,, tube

D_o—outside diameter of shell

d_o— ,, ,, ,, tube

t_s—shell thickness

t_t—tube wall thickness

N—number of tubes in shell

G in equation 9.1 depends on the fixity of the tube.

(a) *For tube side pressure—G*—Mean diameter of channel tube sheet gasket or if integral with channel, inside diameter of channel.

(b) *For shell side pressure—G*—Mean diameter of shell-tube sheet gasket or if integral with shell, inside diameter of shell.

Fig. 9.7 shows a fixed tube sheet arrangement with an increased shell thickness to facilitate welding of tube sheet to shell.

A minimum tube sheet thickness must be provided. It is usually equal to the outside diameter of the tube upto 15 mm tube diameter. For higher diameter tubes, the thickness is smaller than the tube diameter. Table 9.3 gives minimum tube sheet thicknesses.

Table 9.3

Minimum Tube Sheet Thicknesses

Tube Outer Diameter mm	Thickness mm
6	6
10	10
12	12
16	13
18, 19, 20	15
25, 25.4	19
31.8, 32	22.4
38, 40	25.4

On the basis of the type of tube arrangement and pitch, it is possible to determine the tube sheet diameter and also the

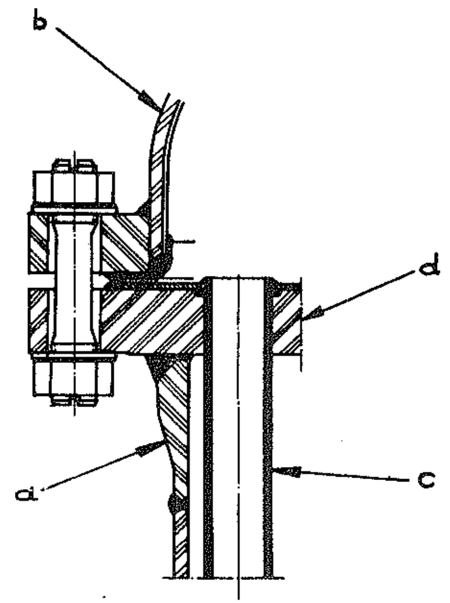


Fig. 9.7 Lined fixed tube sheet
(a) shell (b) channel (c) tube (d) lined tube sheet

inside shell diameter. Assuming equilateral spacing, the area occupied by each tube is given by (Fig. 9.5)

$$a = S_T^2 \tan \alpha = 0.866\, S_T^2 \tag{9.4}$$

where

S_T—pitch of tube

α—angle, formed by triangular spacing which is equal to 60°.

Area for n number of tubes is

$$a_n = n \times 0.866\, S_T^2 \tag{9.5}$$

To provide for pass partitions, tie rods etc., the actual area of the tube sheet, for locating the tubes will be greater. This is assessed by dividing the area a_n by a proportionality factor β, whose value varies from 0.8 to 1 for single tube side pass, 0.7 to 0.85 for double pass and 0.6 to 0.8 for multipass heat

exchangers. This area also corresponds to the area of the shell

$$A_s = \frac{\pi D^2}{4} = \frac{n \times 0.866\, S_T^2}{\beta} \tag{9.6}$$

where

A_s—area of shell

D—inside diameter of shell.

The overall tube sheet diameter can be determined by providing additional area required to position the gasket. Similar procedure may be evolved for square pitch.

9.3.10 Shell Side and Tube Side Passes

The direction of flow in the shell and tubes can be changed by use of different flow paths, known as passes. Such arangements are used to obtain higher velocities and longer paths for the fluid to travel without increasing the length of the exchanger. The passes on the shell side are single pass, two pass, single split pass, and double split pass (Fig. 9.8). Passes on

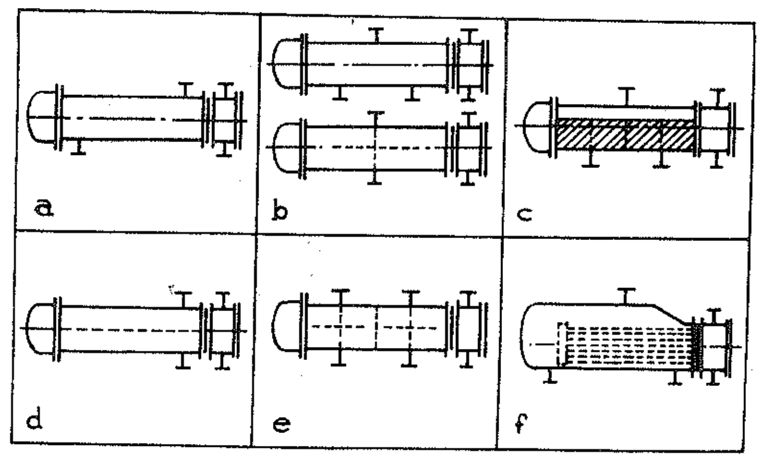


Fig. 9.8 Shell side passes
(a) single pass (b) single split pass (c) evaporator (d) two pass (e) double split pass (f) kettle type

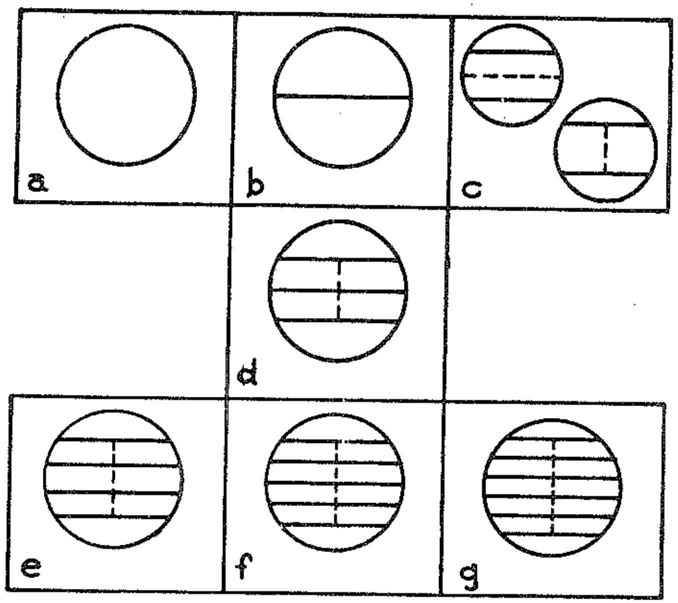


Fig. 9.9 Tube side passes
(a) single pass (b) two passes (c) four passes (d) six passes (e) eight passes (f) ten passes (g) twelve passes

the tube side are one, two, four, six upto twelve (Fig. 9.9). Shell side passes are formed by use of baffles placed in the shell, while those on the tube side are formed by partitions placed in the shell cover and channels. These partitions are fixed between the channel covers and tube sheets by providing approximately 5 mm deep grooves.

9.3.11 Baffles and Tie Rods

Baffles are of two types, namely, transverse and longitudinal. These are used to increase the rate of heat transfer by increasing the velocity and turbulance of the shell side fluid. The clearance between the shell and the baffles and between the tube and baffles must be minimum to avoid bypassing of the fluid. However, the clearance should be enough to permit the removal of the tube bundle. The baffles are supported independently of the tubes by tie rods and positioned by

spacers. The tie rods are fixed at one end in the tube sheet, by making blind holes. The minimum number of tie rods is four with at least 10 mm diameter. Transverse baffles are of different shapes, such as segmental, disk and ring, orifice, vane and helical types (Fig. 9.10). The common baffle shape

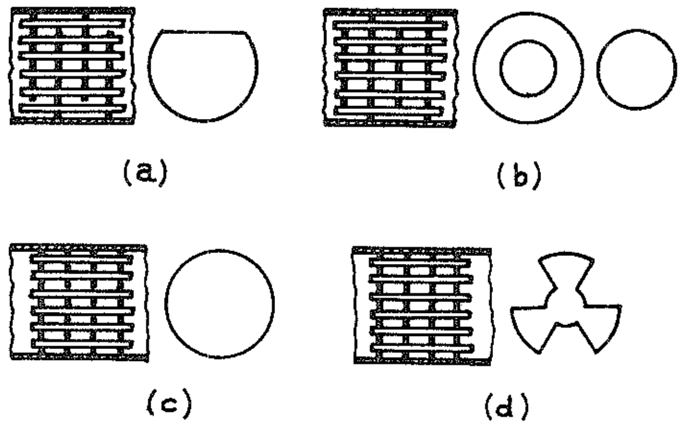


Fig. 9.10 Transverse baffles
(a) segmental (b) disc and ring (c) orifice (d) vane

is the segmental type. Longitudinal baffles consist of flat plates, which extend across the shell. These are used to obtain counter flow of the tube side and the shell side fluids.

9.3.12 Channel and Channel Cover

Channels may either be fabricated or be of cast construction and may be one piece 'bonnet' type or 'straight' type with a separate bolted cover. The thickness of the channel shall be greater of the following two values: (a) shell thickness (b) thickness calculated on the basis of pressure in a cylindrical channel. Under severe operating conditions, the depth in the case of multipass channels shall be such that the minimum cross over area for flow between successive tube passes is at least 1.3 times the flow area through the tubes in one pass. Where severe conditions are not expected the minimum cross over area shall be at least equal to the flow area through the tubes in one pass.

The effective thickness of the flat channel cover is calculated from the formula

$$t = G_c \sqrt{\frac{kp}{f}} \tag{9.7}$$

where

G_c—Mean gasket diameter for cover

p—design pressure

f—permissible stress at design temperature

k—a factor which is 0.25 when the cover is bolted with full faced gaskets and 0.3 when bolted with narrow faced or ring type gaskets. Dished heads can also be used as channel covers. The final thickness is determined by addition of corrosion allowance as required. A special type of channel is shown in Fig. 9.11. It shows a vertical exchanger with a

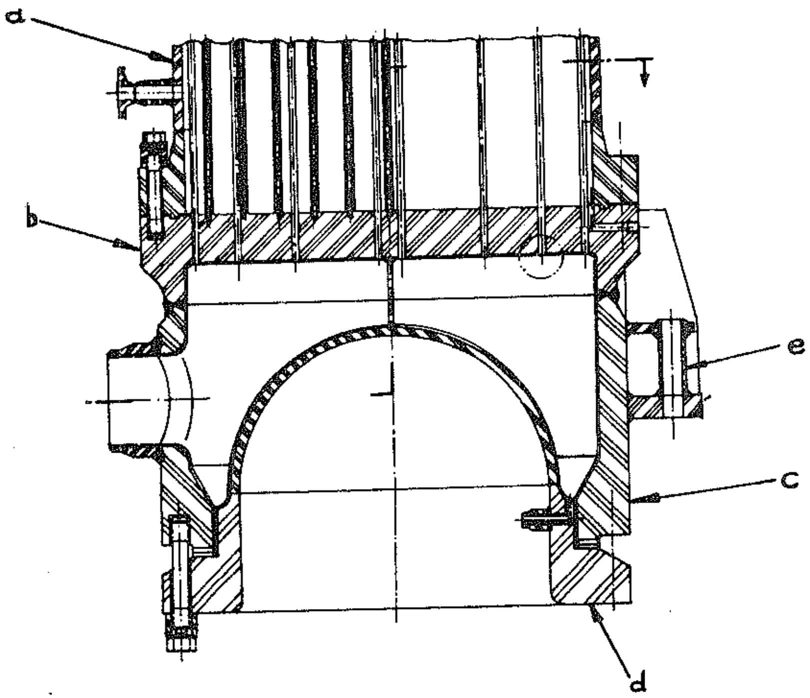


Fig. 9.11 Channel with tube sheet
(a) shell (b) tube sheet (c) channel (d) domed channel cover (e) support

dome shaped channel cover with tube sheet fixed to the channel. This arrangement facilitates complete draining of tube side liquid.

9.3.13 Tube Sheet, Channel and Shell Joints

Flanged joints are used to hold the tube sheet between shell and channel, as also for channel covers. These flanges are of ring type or welding neck type. Ring flanges thicker than 100 mm are not recommended. Above this thickness welding neck flanges are preferred. Flange facings may be raised face, male and female or tongue and groove (Fig. 9.12). A double tube sheet arrangement is also shown in the figure.

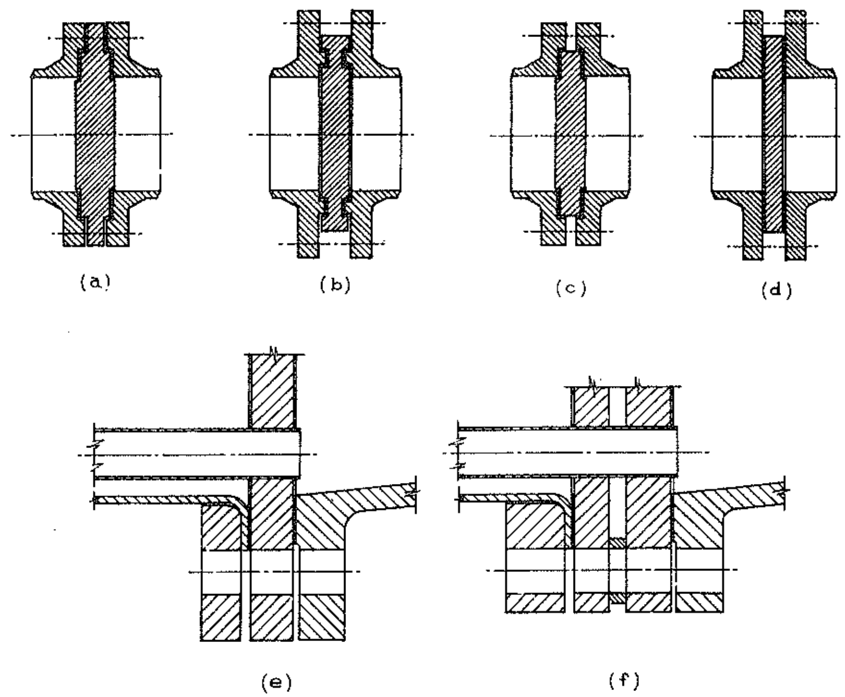


Fig. 9.12 Flanged joint with tube sheet
(a) plain face (b) male and female (c) tongue and groove (d) male and female (with full tube sheet) (e) single tube sheet (removable) (f) double tube sheet (removable)

Metal jacketed or solid metal gaskets are used for all joints in contact with oil or oil vapour and also for pressure above 16 kgf/cm^2. For pressure below this value and when there is no contact with oil and oil vapour, compressed asbestos, fibre, natural or synthetic rubber or other suitable gasket and packing materials having appropriate mechanical and corrosion resisting properties may be used.

Detailed information on the gaskets, bolt loads type of flanges and the flange thickness is indicated under chapter 6.

9.3.14 Expansion Provisions

In heat exchangers it is necessary to make provision to allow for differences in the expansion of the tube and shell due to rise in temperature. If the shell and the tubes are fixed together, as in the case of fixed tube sheet exchangers, the two components will not be free to expand. Lack of provision for free expansion is likely to induce considerable stresses in the shell and tubes. There is also a likelihood of damage to the joints made for holding the tube sheet.

If free expansion due to temperature rise is prevented, the stresses in the shell and the tubes can be assessed as follows:

Elongation of tubes due to rise in temperature

$$\triangle l_t = \alpha_t L_t T_t \tag{9.8}$$

Elongation of shell due to rise in temperature

$$\triangle l_s = \alpha_s L_s T_s \tag{9.9}$$

where

α—coefficient of liner expansion

L—length of tube or shell

T—rise in temperature above atmospheric temperature.

If $T_t > T_s$, the tubes will elongate more than the shell. (Assuming same material of construction. In the case of different materials relative elongation will depend on coefficient of linear expansion for each material and temperature rise). The shell will therefore resist the free expansion of the tubes, while the

tubes will create a greater expansion of the shell. If the mean elongation is $\triangle l$

$$\triangle l_s < \triangle l < \triangle_t$$

The force on the shell will be tensile, and is given by

$$p = \frac{\triangle l - \triangle l_s}{L_s}(A_s E_s) \tag{9.10}$$

The force on the tubes will be compressive, given by

$$p = \frac{\triangle l - \triangle l_t}{L_t}(A_t E_t) \tag{9.11}$$

$$\text{Stress in the shell} = \frac{p}{A_s} \tag{9.12}$$

$$\text{Stress in the tubes} = \frac{p}{A_t} \tag{9.13}$$

where

A—area of cross-section of shell or tubes

E—modulus of elasticity

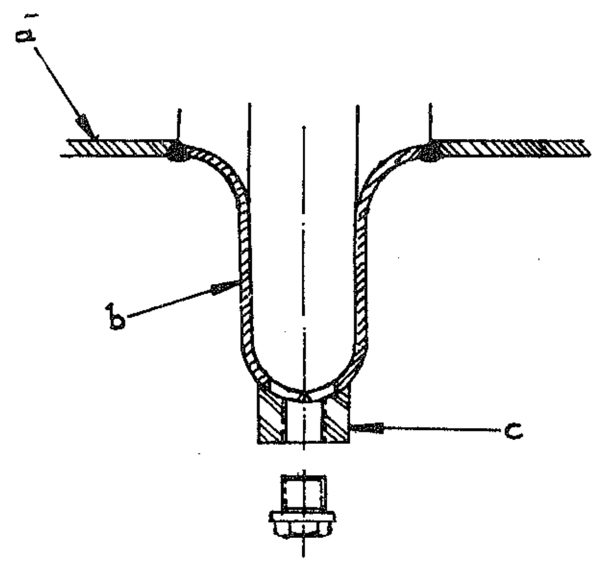


Fig. 9.13 Expansion diaphragm
(a) shell (b) diaphragm (c) drian

(*a*) *Expansion bellow or diaphragm*

One of the methods of reducing the thermal stresses in fixed tube sheet heat exchangers is the interposition of a fairly flexible element in the comparatively stiff shell of the heat exchanger. This flexible element called expansion bellow or diaphragm has sufficient flexibility and permits constrained expansion or contruction of the shell. However it has a certain amount of rigidity and thus the heat exchanger shell behaves as a partially constrained member. An expansion bellow can thus never reduce the thermal stresses to zero. Fig. 9.13 shows an expansion diaphragm, consisting of two dish-shaped disks each welded to the shell. The disks are welded together to form a diaphragm. Fig. 9.14 shows the attachment of the diaphragm to the shell.

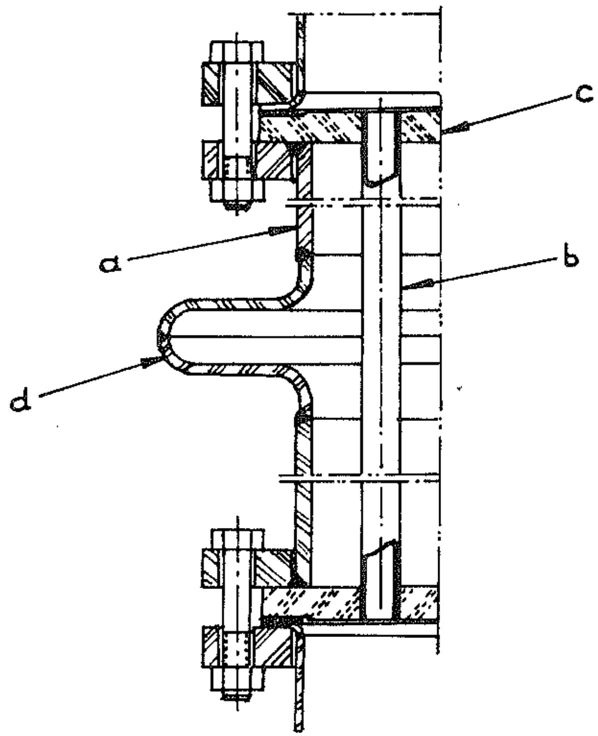


Fig. 9.14 Fixed tube sheet heat exchanger with diaphragm
(a) shell (b) tube (c) tube sheet (d) diaphragm

(*b*) *Outside packed floating head*

One end of the shell is formed into a stuffing box. An extension piece of the shell is provided with a gland. The two components of the shell are assembled together with a packing in between. The shell is therefore free to expand. It will thus compress the gasket and allow the extension piece to adjust its position accordingly. The tube sheet forms a part of the extension piece, which is closed by a separate cover.

In this arrangement the entire tube bundle can be withdrawn to allow access to the outside surfaces of the tubes. Fig. 9.15 shows details of outside packing arrangement.

(*c*) *Internal floating head*

In this method, one of the tube sheets is held between a floating head cover and a backing ring by use of a flanged joint. This assembly is placed inside a fixed head attached to the shell. In some cases the floating head cover is attached to the tube sheet without any ring. This arrangement permits free expansion of the tubes. The tube bundle can be removed readily for cleaning the outside surface of the tubes or for making repairs.

In cases where, either of the fluids passing through the exchanger undergoes a large change in temperature, a split floating head is used. The tube-side fluid passes successively through different sections of the split head, which are free to expand according to the temperature difference in each section. Figs. 9.16 and 9.17 show details of two arrangements of floating heads.

(*d*) *U-tube arrangement*

In this arrangement there is only one tube sheet to which both the ends of the tubes are attached. All the tubes are bent to form a U. This is a relatively inexpensive means of providing for expansion. The tubes are free to expand independently of each other. Internal cleaning of the tubes is difficult, which prohibits the use of this arrangement, when the tube-side fluid is dirty or scale forming.

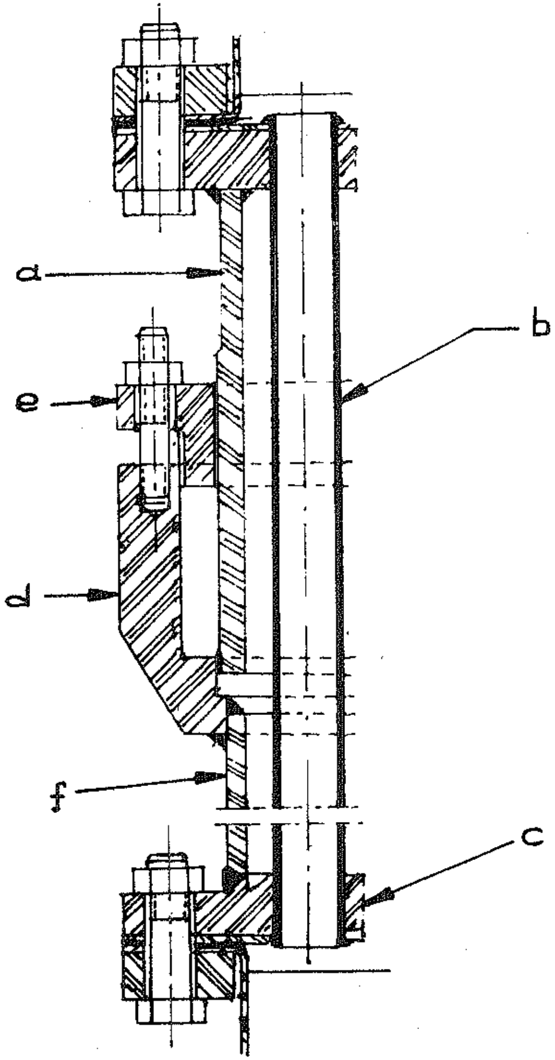


Fig. 9.15 Outside packed floating head
(a) shell (b) tube (c) tube sheet (d) stuffing box (e) gland
(f) shell extension piece

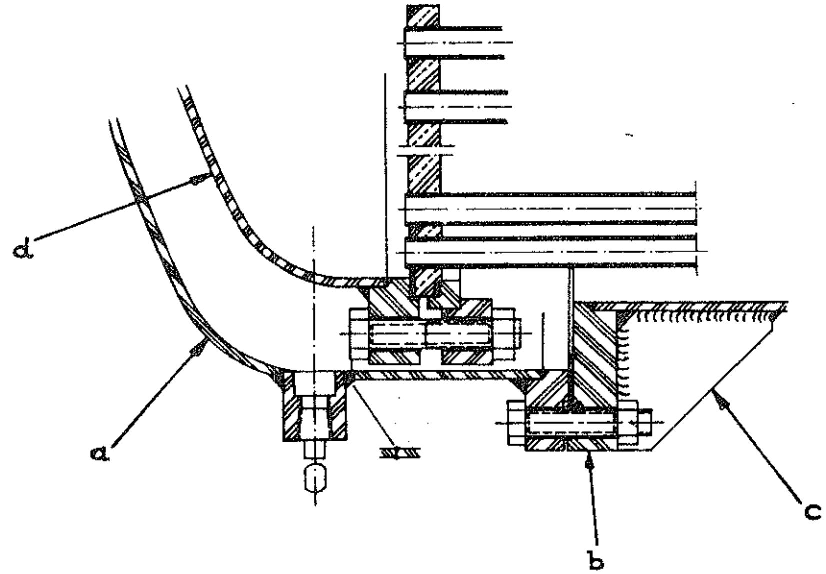


Fig. 9.16 Floating head details
(a) head (b) shell flange (c) gusset plate (d) floating head

(e) *Bowed tube arrangement*

Instead of using straight tubes, a short length of the tube near the tube sheet is bent. This creates a flexibility in the tubes and allows for a limited expansion of the tubes. This method is adopted only in a few cases.

9.3.15 Nozzles

The following nozzles are generally provided on the heat exchanger:

(*a*) inlet and outlet for shell side fluid
(*b*) inlet and outlet for tube side fluid
(*c*) instrument connection

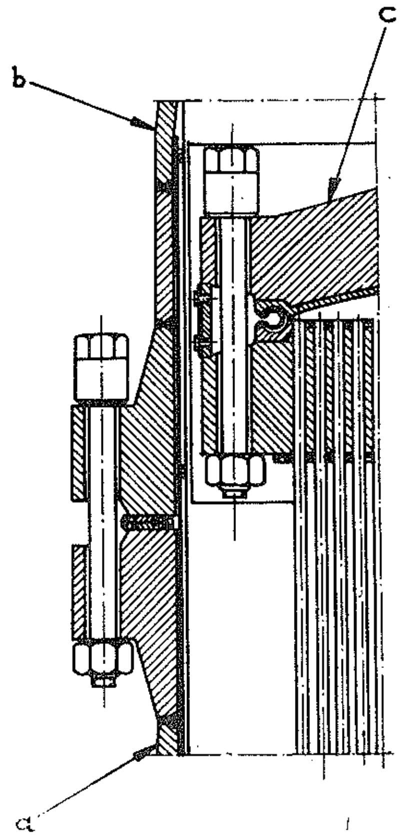


Fig. 9.17 Lined floating head details
(a) shell (b) head (c) lined floating head

(*d*) drain
(*e*) vent.

Nozzles on shells, shell covers and channels are either integral or welded. Shell nozzles have to be flush with the inside contour of the shell. Channel nozzles may protrude inside the channel except for drain and vent. Venting and draining

nozzles may be provided in the diaphragm of a fixed tube sheet exchanger (Fig. 9.13). For a vertical heat exchange, draining can be provided through the tube sheet (Fig. 9.18).

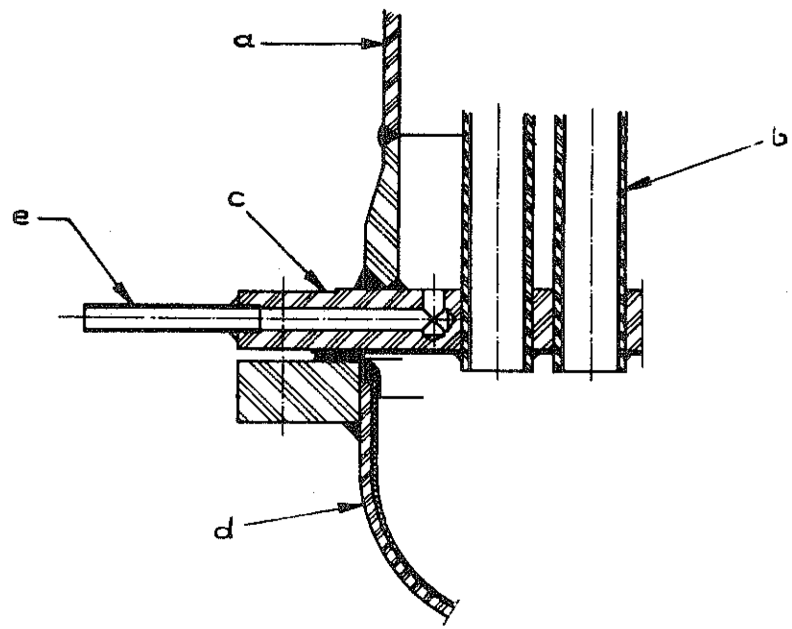


Fig. 9.18 Vertical heat exchangers with drain through flange
(a) shell (b) tube (c) tube sheet (d) channel (e) drain

9.3.16 SUPPORTS

Horizontal shells are supported on two saddle supports, while vertical shells are supported on brackets. In the case of saddle supports for fixed tube sheet exchanger one of the supports is fixed, while the other is placed on rollers, to facilitate expansion of shell (Fig. 9.19). Details regarding design of supports are given in Chapter 13 (see table 13.2).

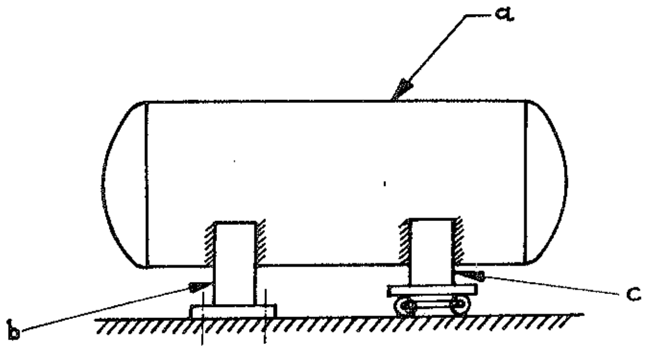


Fig. 9.19 Roller support
(a) shell (b) fixed support (c) roller support

9.4 Numerical Problem

Shell and tube heat exchanger data (Plate II)

Horizontal cooler 'U' bend type

(a) Shell side

Material carbon steel (Corrosion allowance—3 mm)

Number of shell	—	1
Number of passes	—	1
Fluid	—	Water
Working pressure	—	3.3 kg/cm²
Design pressure	—	5 kg/cm²
Temperature inlet	—	30°C
,, outlet	—	50°C

Segmental baffles (25% cut) with tie rods and spacers.

Head

Crown radius	—	400 mm
knuckle radius	—	40 mm

Shell flange—female facing

Gasket — flat metal gasketed asbestos filled

Bolts — 5% C_rM_o steel.

Nozzles — Inlet and outlet—75 mm

Vent	25 mm
Drain	25 mm
Opening for relief valve	50 mm

Permissible stress for carbon steel—9.50 kg/mm²

Permissible stress for bolt material—14.06 kg/mm²

(b) Tube side

Tube and tube sheet material—stainless steel (IS-grade 10)

Number of tubes	—	54
Outside diameter	—	18 mm
Length	—	12 mm
Pitch (square)	—	25 mm
Fluid	—	carbon dioxide
Working pressure	—	190 kg/cm²
Design pressure	—	215 kg/cm²
Inlet temperature	—	150°C
Outlet temperature	—	55°C
Permissible stress	—	10.06 kg/mm²

(c) Channel and channel cover

Material—carbon steel

Joint with tube sheet—ring facing

Gasket—steel jacketed asbestos

Nozzle—inlet and outlet—75 mm

Permissible stress—9.50 kg/mm²

9.4.1 √Shell Side

9.4.1.1 SHELL DIAMETER

From equations (9.4 and 9.5)

(a) Triangular pitch of tubes $a = 0.866\, S_T^2$

$$= 0.866 \times (2.5)^2$$
$$= 5.4 \text{ cm}^2$$

$a_n = 2\times 54\times 5.4 = 584$ cm^2 (2 pass U-bundle)

$$A_s = \frac{\pi D^2}{4} = \frac{584}{0.8}$$

$$D = \sqrt{\frac{584}{0.8}\times\frac{4}{\pi}}$$

$D = 30.5$ cm

(b) Square pitch of tubes

$a = S_T^2 = (2.5)^2 = 6.25$ cm^2

$a_n = 2\times 54\times 6.25 = 703$ cm^2 (2 pass U-bundle)

$$A_s = \frac{\pi D^2}{4} = \frac{703}{0.7}$$

$$D = \sqrt{\frac{703}{0.7}\times\frac{4}{\pi}}$$

$D = 35.8$ cm.

For square pitch of tubes the shell is selected with 400 mm nominal diameter (IS-2844).

9.4.1.2 SHELL THICKNESS

From equation 6.3

$$t_s = \frac{pD}{2fJ+p} \qquad J = 85\ \%$$

$$= \frac{5\times 400}{(2\times 950\times 0.85)+5} = 1.23 \text{ mm}$$

IS-4503—Table (4) gives a minimum thickness of 6.3 mm including corrosion allowance. Use 8 mm thickness.

9.4.1.3 NOZZLES-THICKNESS (DIAMETER 75 mm)

From equation 6.3

$$t_n = \frac{pD}{2fJ-p} = \frac{5\times 75}{2\times 950-5} = 0.1975 \text{ mm}$$

$J = 1$ (seamless pipe)

Corrosion allowance 3 mm

$t_n = 4$ mm

9.4.1.4 HEAD-THICKNESS

From equation 6.23

$$t_h = \frac{5\times 400\times 1.34}{2\times 950\times 1} \qquad W = \tfrac{1}{4}\left(3+\sqrt{R_c/R_i}\right)$$

$$= 1.41 \text{ mm} \qquad = 1.34$$

and $J = 1$

Use thickness same as for shell i.e., 8 mm including corrosion allowance.

9.4.1.5 TRANSVERSE BAFFLES

Spacing between baffles $= \frac{4000}{5} = 80$ mm

Thickness of baffles $= 6$ mm

9.4.1.6 TIE RODS AND SPACERS

Number of tie rods $= 6$

Diameter of tie rod $= 10$ mm.

9.4.1.7 FLANGE JOINT (BETWEEN SHELL AND TUBE SHEET)

This will be made to satisfy the requirements of the flange joint between tube sheet and channel. The bolt size and the pitch circle diameter will be identical. These are determined under 9.4.2.5.

Number of bolts—16

Size of bolts—M 48

Pitch circle diameter (B)—525 mm

9.4.1.8 FLANGE THICKNESS (MALE AND FEMALE FACING)

Gasket size $= 440$ mm outside diameter,

416 mm inside diameter.

Gasket diameter (G) $= 428$ mm

Gasket width (N) $= 26$ mm

$$b_o = \frac{N}{2} = \frac{24}{2} = 12 \qquad b = 2.5\sqrt{12} = 8.64 \text{ mm}$$

Seating stress=5.34

Gasket factor $m=3.75$

From equation 6.36

$$W_{m_1} = \pi \times 428 \times 8.64 \times 5.34 = 62000 \text{ kg.}$$

From equation 6.37

$$W_{m_2} = 2 \times \frac{8.64}{10} \times \pi \times \frac{428}{10} \times 3.75 \times 5 + \frac{\pi}{4} \times \left(\frac{428}{10}\right)^2 \times 5$$

$$= 4350 + 7200 = 11550 \text{ kg}$$

From equation 6.41

$$k = 1\left[0.3 + \frac{1.5 \times 62000}{7200 \times 428} \times \frac{(525-428)}{2}\right] \quad B=525 \text{ mm}$$

$$= \frac{1}{0.3+1.47}$$

$$= 0.621$$

$$t_f = 428\sqrt{\frac{5}{0.621 \times 950}}$$

$$= 39.4 \text{ mm}$$

Use a 45 mm thick flange including corrosion allowance.

9.4.2 Tube Side

9.4.2.1 THICKNESS OF TUBE

From equation 6.3

$$t_1 = \frac{pD_o}{2fJ+p} \qquad J=1 \text{ (seamless tube)}$$

$$= \frac{215 \times 18}{2 \times 1006 \times 1 + 215} = 1.73 \text{ mm}$$

No corrosion allowance, since the tubes are of stainless steel. Use a thickness of 2 mm.

9.4.2.2 TUBE SHEET

The tube sheet is held between shell flange and the channel. The joint on the shell flange side is of male and female facing and on the channel side, of ring facing, since the pressure on the channel side is high.

Based on tube side design pressure of 215 kg/cm² and a mean gasket diameter (G) for the ring type of gasket of 375 mm, tube sheet thickness is calculated from equation 9.1

$$t_{ts} = 1.25 \times 375 \sqrt{\frac{0.25 \times 215}{1006}}$$

$$= 105 \text{ mm}$$

9.4.2.3 CHANNEL AND CHANNEL COVER

Since the pressure on the tube side is high and the velocity of carbon dioxide is low, it is proposed to make the channel and cover out of a single plate as shown in Plate II.

Thickness of channel portion from equation 9.7

$$t_h = G_c \sqrt{\frac{kp}{f}} \qquad k=0.3 \text{ for ring type gasket}$$

$$= 375 \sqrt{\frac{0.3 \times 215}{950}}$$

$$= 94.7 \text{ mm}$$

Use 100 mm thickness including corrosion allowance.

9.4.2.4 FLANGE JOINT (BETWEEN TUBE SHEET AND CHANNEL)

Gasket and bolting calculations—(steel jacketed asbestos)

$G=375$ mm

Ring gasket width (W)=22 mm

$$b_o = \frac{W}{8} = 2.75 \text{ mm}$$

$Y_a = 12.66$ kg/mm²

$m=5.5$

$b=b_o$

From equation 6.36

$$W_{m_1} = \pi G b Y = \pi \times 375 \times 2.75 \times 12.66$$
$$= 41000 \text{ kg}$$

From equation 6.37

$$W_{m_2} = 2b\,\pi\, G_m p + \frac{\pi G^2 p}{4}$$
$$= 2 \times \frac{2.75}{10} \times \pi \frac{375}{10} \times 5.5 \times 215 + \frac{\pi}{4}\,\frac{(375)^2}{(10)} \times 215$$
$$= 311600 \text{ kg}$$

$$A_{m_1} = \frac{41000}{1406} = 292 \text{ cm}^2 \quad \text{and} \quad A_{m_2} = \frac{31160}{1406} = 221 \text{ cm}^2$$

$$\text{Number of bolts} = \frac{375\pi}{10 \times 2.5} = 15$$

Use 16 bolts.

From equation 6.39

$$\text{Diameter of bolt} = \sqrt{\frac{221}{16} \times \frac{4}{\pi}} = 4.2 \text{ cm}$$

Use M 48 bolt   Pitch diameter $= 44.681$ mm

Miner diameter $= 41.795$ mm (IS-3139)

$$\text{Actual bolt area} = \frac{\pi}{4}\left(\frac{44.68 + 41.795}{2 \times 100}\right)^2 = 294 \text{ cm}^2$$

Min. pitch circle diameter $(B) = 375 + 22 + 2 \times 48 = 493$ mm

Take $B = 525$ mm

9.4.2.5 FLANGE THICKNESS

From equation 6.41

$$K = \frac{1}{0.3 + \dfrac{1.5 \times 311600}{238000 \times 375}\left[\dfrac{525 - 375}{2}\right]}$$
$$= \frac{1}{0.3 + 0.393}$$
$$= 1.44$$

$$t_f = 375 \sqrt{\frac{215}{1.44 \times 950}}$$
$$= 138.5 \text{ mm}$$

Use a thickness of 145 mm.

9.4.2.6 NOZZLES

Inlet and outlet (75 mm).

Thickness of nozzle from equation 6.3

$$t_n = \frac{pD}{2fJ - p} \qquad J = 1 \text{ (seamless pipe)}$$
$$= \frac{215 \times 75}{2 \times 950 - 215} = 6.6 \text{ mm}$$

Add corrosion allowance of 3 mm.

Thickness $t_n = 14$ mm

Considering the size of the nozzle and the pressure rating, it is necessary to provide for a reinforcing pad on the channel cover. (see 86.8.3.2)

Area required to be compensated for each nozzle is

$$A = d\, t_h = 75 \times 94.7 = 7930 \text{ mm}^2$$

Compensation will be available from the additional thickness of channel cover and nozzle. It is proposed to use a 50 mm thick pad as shown in Plate II. It can be shown by calculations on the basis of area for area method [6.8.32(a)] that this reinforcement is satisfactory.

Reading References

Kern, D.Q., *Process Heat Transfer*, McGraw-Hill, New York, (1950).

Stoever, H.J., *Applied Heat Transmission*, McGraw-Hill, New York, (1941).

Rubin, F.L., *Practical Heat Exchanger Design*, Chem. Eng. Prog., 64(12), 44 (1968).

Narayan, S., *Heat Exchangers*, Chem. Age of India, 20(8), 673 (1969).

Lord, R.D. et al., *Design of Heat Exchangers*, Chem. Eng., 77(2), 96 (1970).

Vijay Kumar, *A Systematic Approach to the Design of Exchanger Tube Sheet*, Chem. Age of India, **25**(4), 244(1974).

Ludwig, E.E., *Applied Process Design for Chemical and Petrochemical Plants*, Vol. 3, Gulf Publishing Co., Houston, (1964).

Bevevino, J.W., *Tube to Tubesheet welding Technique*, Chem. Eng. Prog, **70** (7), 71 (1974).

Ackley, E.J., *Heat Transfer through Glassed Steel*, Chem. Eng., **66**(8), 181 (1959).

A.I. Ch. E., *Standard Testing Procedure, Heat Exchangers*—Section I, Section II, New York.

Brown, G.G., *Unit Operations*, John Wiley, New York, (1956).

Frass, A.P. and Ozisik, M.N., *Heat Exchanger Design*, John Wiley New York, 1965).

Hills, D.E.G., *Graphite Heat, Exchangers* I-Chem. Engg., **81**(27), 80 (1974) ; II-Chem. Engg., **82**(2), 116 (1975).

Westerman, V., *Bellows for Heat Exchangers*, Processing, **21**(5), 47(1975).

Chugh, R.D., *Methods of Tube to Tubesheet Attachment in Heat Exchangers*, Chem. Age of India, **26**(6), 441(1975).

CHAPTER 10

Evaporators and Crystallisers

10.1 Evaporators

10.1.1 Introduction

Evaporators are used in the chemical industry for concentrating liquids. The operation is performed normally by use of low pressure, dry, saturated steam. For special purposes organic vapours or fuel gases may be adopted. In most cases water, which is the solvent, is evaporated. With the vaporisation of a part of the water, the useful product is a concentrated solution. If the solution contains dissolved solids the resulting strong liquor may become saturated so that crystals are formed.

In majority of the evaporators the heat transfer is through tubular surfaces. The body of the evaporator is in the form of a cylindrical shell, in which the tubes are placed. In order to create effective heat transfer, the liquid must flow across the heat transfer surface. As a result of the generation of the vapour, a natural circulation is created. In evaporating viscous liquids or in cases where natural circulation is inadequate, forced circulation is adopted by use of a pump or by a propeller. With forced circulation it is possible to control the velocity of flow across the heating surface. As the solution boils, a more or less stable foam is formed, on the surface. The vapours generated have, therefore, liquid droplets in suspension. Sufficient vapour space (known as vapour release chamber) must be available for releasing the water vapour from the liquor. For separation of liquid droplets various devices, such as mesh pads, baffles and impingement or centrifugal separators, are adopted.

10.2 Types of Evaporators

Evaporators can be classified according to the heating medium, heating surface and its disposition and the method of circulation. Solar heat, gases, organic vapours and steam are used as heating media. The most common heating medium, namely, steam, can be used in jackets attached to vessels as in the case of kettles or pans and in horizontal shell and tube type of heat exchangers. Standard evaporators usually consist of vertical tubes placed in a cylindrical vessel, known as calendria, which have a provision for vapour space either as an extension of the calendria or as a separate vessel. Evaporators can also be made with horizontal or inclined tubes. Pipe coils, oblong or helical, submerged in the solution contained in a vessel are also used as evaporators.

10.2.1 Standard Vertical Short Tube Evaporators (Calendria Type)

These evaporators (Fig. 10.1) consist of a large vertical cylindrical vessel or drum, ranging from 1 metre to 6 metres

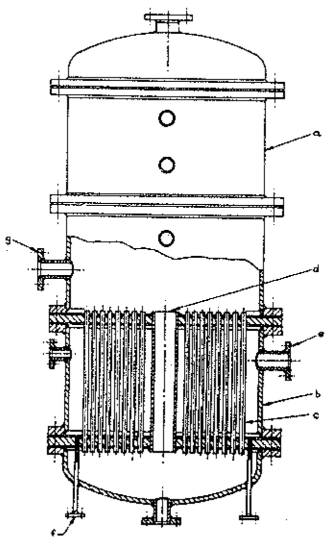
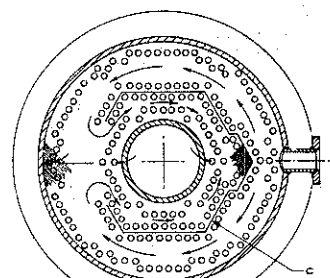


Fig. 10.1 Standard short tube vertical evaporatoı
(a) drum (b) calendria (c) tubes (d) downtake (e) steam inlet (f) condensate outlet (g) feed

diameter with a short vertical tube bundle, consisting of 2.5 cm to 7.5 cm outside diameter tubes, 75 cm to 200 cm long, located in the bottom portion of the vessel. The liquor to be evaporated is inside the tubes and the steam is outside. The tube bundle is held between two tube sheets, placed horizontally and bolted to shell flanges. A central downtake promotes circulation. The area of the central downtake varies from about 40% to 100% of the total cross-sectional area of the surrounding tubes. Multiple downtakes promote even circulation, which reduces the possibility of localized scale formation. The layout of the tubes is shown in Fig. 10.2. A



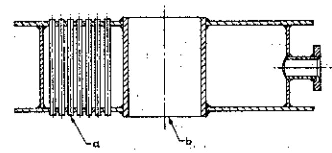


Fig. 10.2 Tube layout of short tube vertical evaporator
(a) tube (b) downtake (c) baffle

baffling system is used within the steam chest, so that there is a uniform distribution of steam. A propeller can be installed in the dished or conical bottom to increase the rate of circulation. Since scaling occurs inside the tubes, it is possible to use the standard evaporator for more vigorous services than the horizontal tube evaporator. The bottom of the cylindrical vessel may be a dished head or a conical head. Considerable vapour space is provided above the tube bundle.

The main components of this evaporator are, therefore, (a) tube bundle with surrounding shell known as calendria (b) evaporator body or vapour release chamber and (c) two heads, one at the bottom and one at the top. The calendria is designed as a shell with internal steam pressure, with tube sheets of a thickness based on equation 9.1 and Table 9.3. It is held between the vapour chamber and the bottom head by flanged joints or is welded to the vapour chamber. For large

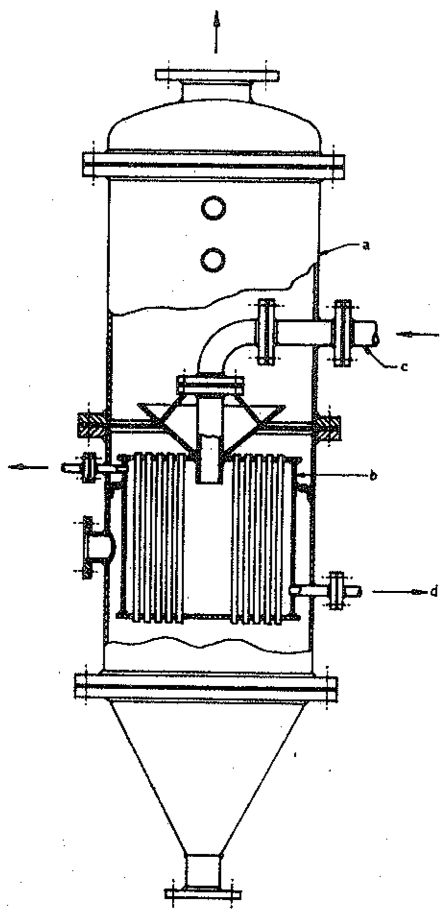


Fig. 10.3 Basket type short tube vertical evaporator
(a) drum (b) tube bundle (c) steam inlet (d) condensate outlet

diameters, a spherical vapour release chamber is more economical. The vapour chamber may be made up of cylindrical sections welded to form the necessary height or it may be made up of flanged corrugated sections, bolted together. A head is also flanged and is bolted at the top. Nozzles are provided for inlet of the solution, vapour outlet, liquor outlet, steam inlet, condensate outlet, drain, vent, manhole, sight glass, level gauge, etc. Supporting legs are of conventional structural sections. Nozzles and supports are welded to the shell.

10.2.2 Basket Type Evaporators

These (Fig. 10.3) are somewhat similar to the standard vertical evaporator. However, the tube bundle in this case is suspended on brackets, and the downtake is formed by the annular space between the bundle and the shell, instead of a central downtake. Since the tube sheets are freely suspended, the problem of differential expansion between the tubes and steam chest shell is not important. These evaporators are not used for liquids with high viscosities or large rates of scaling.

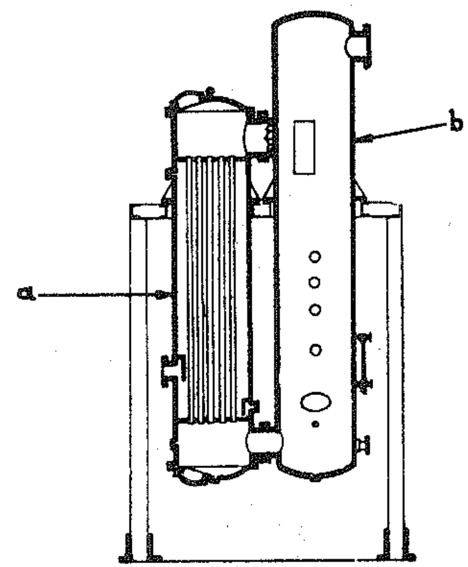


Fig. 10.4 External calendria vertical short tube evaporator
(a) external calendria (b) vapour drum

10.2.3 External Calendria Vertical Tube Evaporators

These consist of a short tube, single effect vertical calendria, but with a separate vapour drum placed by the side of the calendria (Fig. 10.4). The liquid level is kept low, and the foaming is, therefore, confined largely to the inside of the tubes.

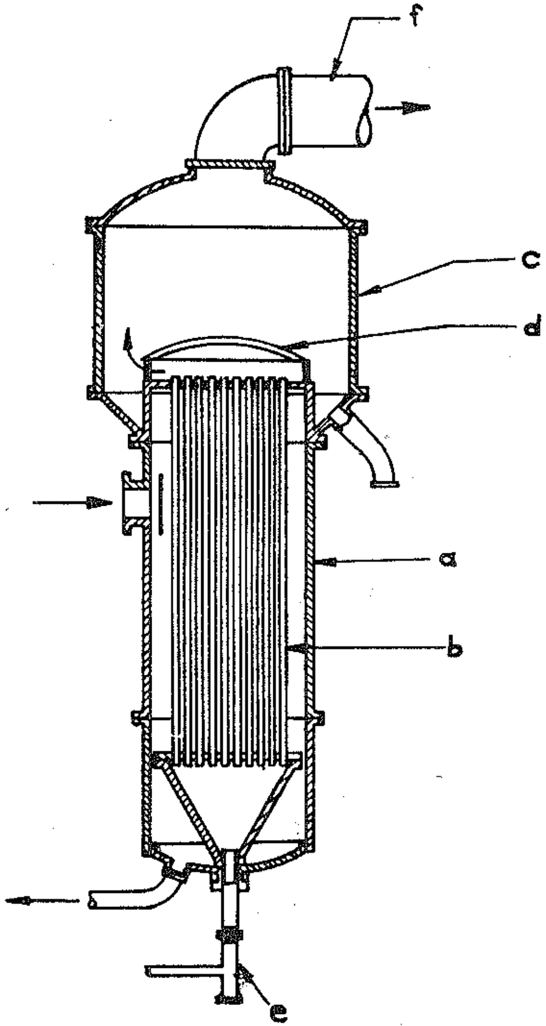


Fig. 10.5 Long tube vertical evaporator
(a) shell (b) tubes (c) drum (d) baffle (e) liquid outlet (f) vapour outlet

These tube contains a mixture of vapour and liquid travelling vertically with high speed. The external calendria can be easily opened and cleaned.

10.2.4 Long Tube Vertical Evaporators

These evaporators (Fig. 10.5) are characterised by relatively long vertical tubes of 3 cm to 5 cm diameter and ranging from 3 metres to 6 metres length. The tubes are enclosed in a shell. The liquor is filled inside the tubes and the steam is fed to the shell. The solution is introduced at a fixed rate. In controlled level type evaporators the liquid level is maintained at one-third to one-half of the height of the tubes. The vapour discharges either into a vapour head directly above the tube bundle or into a separate vapour drum. The liquid is returned to the bottom of the shell through an external downtake.

In the film-type evaporator the liquid level is not controlled. The liquor makes only a single pass through the tubes. Inclined tube evaporators are similar except that the tubes are inclined at an angle of 45°. The liquid is circulated through an external downtake.

10.2.5 Horizontal tube Evaporators

These evaporators (Fig. 10.6) have a vertical cylindrical or a square shell to which are attached two square steam chests. The steam is admitted inside the tubes and the liquor is outside, submerging the tube. The horizontal tubes extend between two tube plates to which they are fastened by packing plates or by expansion. In one method, the holes in the tube sheets are countersunk to receive conical gaskets which are fitted over the ends of the tubes and are forced in by packing plates mounted on studs from the tube plate. Tubes are of 2 to 3 cm diameter. The body of the evaporator is from 1 metre to 4 metres diameter and $2\frac{1}{2}$ metres to 4 metres height. These evaporators are unsatisfactory for fluids which form a scale or deposit salt. They are suitable for processes in which the final product is a liquor instead of a solid.

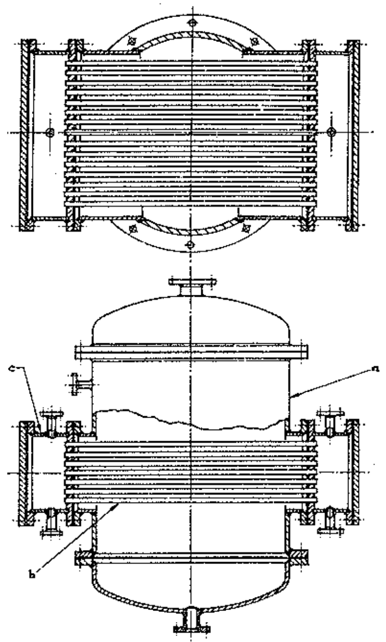


Fig. 10.6 Horizontal tube evaporator
(a) drum (b) tube (c) steam chest

10.2.6 Forced Circulation Evaporators

These evaporators are similar to long tube vertical evaporators, except that a centrifugal pump is placed between the external downtake from the vapour drum and the inlet to the tube bundle. Due to high velocities and consequently high rates of heat transfer, less heating surface is required. This is particularly applicable for heating surface formed out of expensive materials.

The liquor which emerges from the heating elements at high velocities is directed into the vapour drum, where the vapour is separated from the liquid. The vapour drum may be either a separate drum or superimposed on top of the tube bundle. The liquid from the drum is fed to the pump with the

help of the downtake. Forced circulation evaporators may not be as economical in operation as natural circulation evaporators, but they are necessary where the concentration problem involves a solution with poor flow.

10.2.7 Multiple Effect Evaporator

A number of evaporators are used in series, in order to economise the use of steam (Fig. 10.7). The vapours from each evaporator or effect, are led to the steam chest of the next, while the solution flows in the same direction, losing water as it flows. There is no theoretical limit to the number of evaporators which can be used in series. A practical limit is reached when heat transfer rates become low due to inadequate temperature difference and an economic limit is reached when the plant cost exceeds the saving of steam.

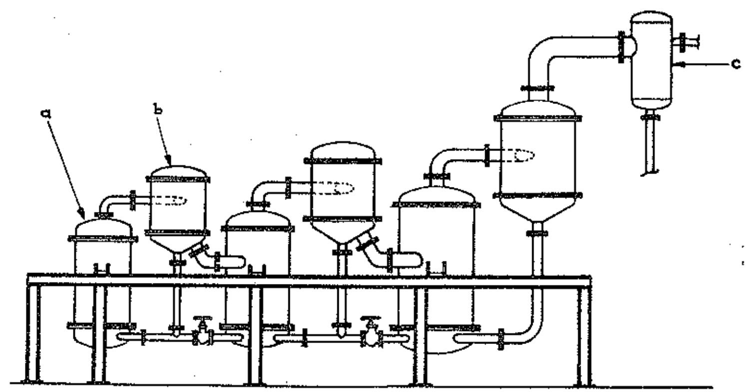


Fig. 10.7 Multiple effect evaporator
(a) evaporator (b) separator (c) barometric condenser

In order to utilise the vapours from each stage as steam in a subsequent stage, it is necessary to use lesser steam pressures in each stage. This involves passing the vapours to a surface condenser or a barometric condenser, where the vapours are finally condensed, and produce the necessary vacuum. In addition a suitable pump is provided to remove air and non-condensables entrained by the liquor or entering through leakage.

10.3 Entrainment Separators

These are used in most evaporators to obtain dry vapours. Entrainment results wherever a vapour is generated from a liquid. When this occurs the vapour carries with it varying quantities and sizes of liquid droplets. Separators provide a means for separating a vapour from the liquid with minimum liquid carry-over or dry vapours. Various methods are adopted for such separation.

10.3.1 Vapour Release Chamber

A large chamber is used to reduce the velocity of vapour stream. This enables the droplets to settle out by gravity. The vapour release drum may either be placed just above the tube bundle shell or it is a separate unit placed adjoining to tube bundle shell being connected to it by a large pipe. It may not be economically practical to make the vapour head large enough to accomplish the entire decontamination of the vapour. Further, increasing vapour space decreases entrainment of larger drops, but has no effect on small drops.

The vapour disengagement rate from a boiling liquid surface should not normally be more than 30 cm per second for normal solutions at atmospheric pressure and may be about 3 cm per second with crude solutions. Even allowing for sufficient vapour disengagement space it is common practice to provide spray traps. These traps are merely a series of baffles giving rapid changes of direction to the vapour stream.

10.3.2 Wire Pad

Pads of finely woven wire, (Fig. 10.8) set in the vapour release chamber at right angles to the vapour flow are used for

entrainment. As the vapour and its entrained liquid pass through the pad, the liquid particles agglomerate, eventually falling back into the vapour release chamber. A highly purified dry vapour leaves the top of the pad.

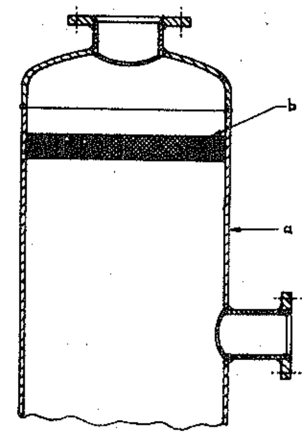


Fig. 10.8 Entrainment separator
(a) drum (b) pad

Application of such pads may be difficult for vapours with suspended solids, fibres or scale forming materials, which will block the wire mesh. In such cases washing facilities at proper intervals may be provided.

10.3.3 Vapour Release Drum Size

The size of drum provided above the tube bundle in most of the evaporators, is decided by three important considerations. They are (a) the foaming of the liquid in the evaporator, (b) the vapour velocity' and (c) entrainment separation.

Foaming takes place above the liquid level and occupies a certain space of the vapour drum. The vapour velocity sets the minimum drum diameter. The following equation helps to determine the drum diameter.

$$R_d = \frac{\frac{V}{A}}{0.0172 \times \sqrt{\frac{\rho_L - \rho_V}{\rho_v}}} \qquad (10.1)$$

where V—volumetric flow rate of vapour in cubic metres/sec

A—area of cross-section of the drum in m^2

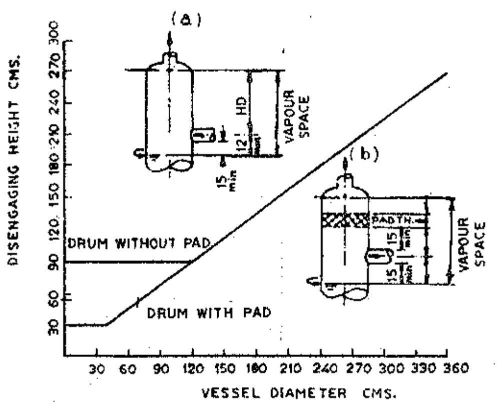
ρ_L—liquid density in kg/m^3

ρ_V—vapour density in kg/m^3

The values of R_d may be taken as, 0.5 for drums without entrainment separators. For drums having a wire mesh as entrainment separator device, R_d may be taken as 1.3.

In the above equation, if the other values are known, it is possible to get the value of the area of cross-section of the drum and hence its diameter.

The height of the drum, considered as disengaging height which helps the process of liquid separation is based on the drum diameter. A graph showing the relationship between the drum diameters (D) and the disengaging height (H_D) is recommended for many applications. Fig. 10.9 shows the requirements of liquid level, feed vapour space, etc.



(c)

Fig. 10.9 Drum diameter disengaging height requirements
(a) drum without pad (b) drum with pad
(c) relationship between drum diameter (D) and disengaging height (H_D)

In cases where the entrainment separator forms an independent unit, the main drum can have a shorter disengaging height.

10.3.4 Centrifugal Separator

This is a separate drum in which the vapours are admitted tangentially and are made to flow in a helical path by use of baffles. The vapours leave either from the top of the drum or through a central pipe [Fig. 10.10 (A)]. A centrifugal type baffling system is shown in [Fig. 10.10 (B)] fitted at the top of the drum. The vapours enter from the central passage and are diverted by the baffles separating the liquid in the process.

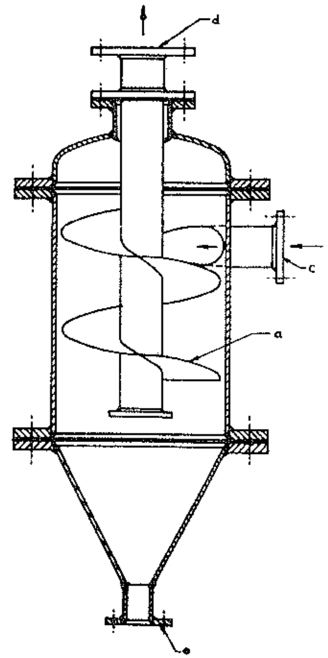


Fig. 10.10 Entrainment separators
(A) Helical baffle type
(a) baffle (b) vapour inlet (c) liquid outlet (d) vapour outlet

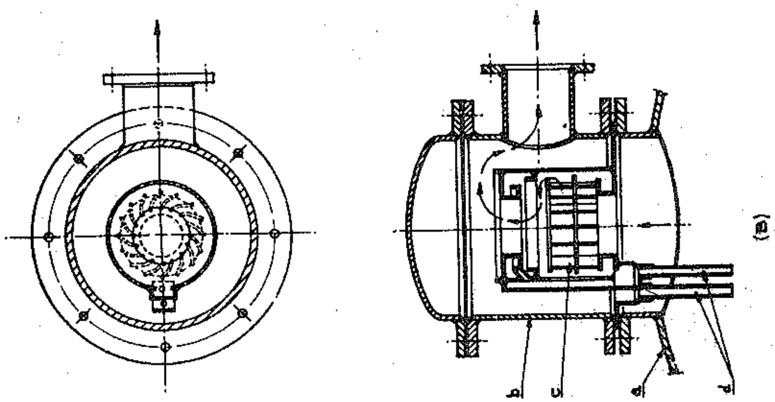


Fig. 10.10 Entrainment separators
(B) Centrifugal type
(a) drum (b) baffles (c) vapour outlet

10.4 Materials of Construction

Evaporator bodies are fabricated from mild steel which is the least expensive and is also easy to fabricate. Materials like cast iron, monel, inconel and stainless steel are used when corrosive action is to be prevented. Tubes are made of copper, aluminium or stainless steel, and tube sheet may be of cast bronze, nickel clad steel or stainless steel.

10.5 Design Considerations

Evaporator drums invariably operate under vacuum. These are designed for an external pressure of 1 kg/cm^2 (see 6.8.1.3). The bottom head may be conical in many cases and may be designed for similar pressure rating (equation 6.22 and section 6.8.2.2 (b)). The top head may be flanged or flared and dished shape or conical. The calendria which has the tubular heating surface is designed as a shell and tube heat exchanger (see Chapter 9). Since steam under pressure is usually accepted as the heating medium, the design is based on

the pressure of steam. The entire evaporator body must be rigid. The conical head, the calendria and the vapour drum are connected by flanged joints or directly welded. The vapour drum may be made up of separate cylindrical pieces and joined by flanges. Large nozzles like manholes, sight glasses must be reinforced with compensating rings. Supports may be placed below the brackets welded to the vapour drum or to the calendria. External calendria is also designed as a shell and tube heat exchanger.

10.6 Crystallisers

10.6.1 Introduction

Crystallisation is a process in which solid crystalline phases separate from liquids. The object of the process is usually the recovery of the solute from the solvent. The performance of the process is evaluated by the size, shape, structure and purity of the solute. The following equipment is used for the operation.

10.6.2 The Tank Crystalliser

This is the simplest and perhaps the most economical unit at present. It is essentially a cylindrical vessel or pan which is cooled by water. Hot saturated liquor is supplied to the unit and slowly cooled. High supersaturation at the cooling surface is usually unavoidable and control is difficult. This normally results in the cooling surfaces becoming rapidly fouled with adhering crystals. The surface, therefore, might need frequent washing or scraping. Cooling can be effected either by a jacket or a coil or by the use of a separate single pass heat exchanger. An agitator is used to improve heat transfer. Jackets are preferred to coils because the latter tend to become coated with a hard crystalline deposit and cease to function efficiently. If a cooling jacket is employed, the inner cold surfaces of the crystalliser should be as smooth and as flat as possible. This minimises crystal build-up on the cold surfaces.

10.6.3 Vacuum Crystalliser

In this unit a high vacuum is drawn in a vessel by means of a steam jet ejector with a barometric condenser. Circulation of the liquor is effected by an agitator or a separate circulating pump.

(a) *Draft tube type (Fig. 10.11)*

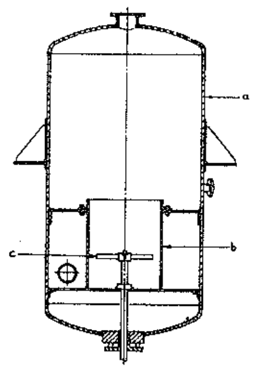


Fig. 10.11 Draft tube type crystalliser
(a) shell (b) draft tube (c) agitator

It is used frequently for vacuum cooling or chemical reaction crystallisation. The feed is supplied near the agitator and is thoroughly mixed. The agitator shaft can be made to enter from either the bottom or from the top. A draft tube surrounding the agitator helps to create circulation and prevents short circuiting of feed. Such a unit is simple and easy to operate and, if well designed can work for very long periods between washouts.

(b) *Evaporative type*

The calendria type evaporator (Fig. 10.12) preferably with short wide tubes and a large central downtake, with the bottom in the form of a conical head, can be used as a crystalliser. A barometric leg discharge can be used for vacuum production.

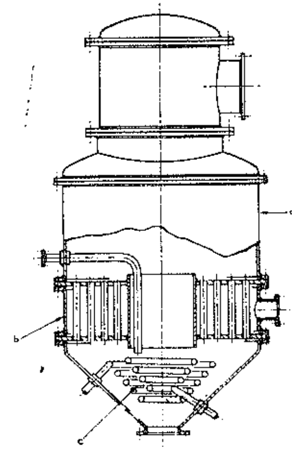


Fig. 10.12 Evaporative type crystalliser
(a) drum (b) calendria (c) steam coil

A unit may also consist of separate circulation pump and shell and tube heat exchanger. This gives sufficient circulation and ample agitation in the main body of the crystalliser. Long periods between washouts are possible.

10.6.4 Growth Type Crystalliser

This is used for the production of larger crystals with a narrow size range and truly close control in the crystallisation process. In such a unit additional operations such as settling of the crystals in order to increase the sludge density in the crystalliser and removal of excess fines from the system are usually needed. Equipment for these functions can be added as external units to any of the crystallisers already discussed, but it is generally much more effective to combine these as an integral part of the crystalliser itself.

In the simplest unit, an agitator is provided at the bottom of a vessel with an open settling chamber at the top. Another type works under vacuum and uses draft tubes for circulation (Fig. 10.13). Agitation is either by an agitator in the vessel or by an external circulation unit like a pump. The feed liquor

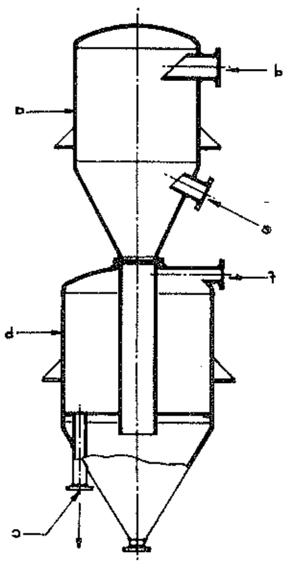


Fig. 10.13 Growth type crystalliser
(a) flash vessel (b) crystallising vessel (c) crystal discharge (d) vapour outlet (e) liquid inlet (f) liquid outlet to heater

is introduced into the unit below the agitator in a draft tube unit, or before the pump in an external circulation unit.

10.6.5 JACKETED TROUGH CRYSTALLISER

This (Plate IV) is probably the most widely used crystalliser. It consists of a U-shaped trough of about 3 to 6 metres long, with a jacket divided into sections, so that differential cooling may be used in the various zones. A close fitting long pitch spiral agitator keeps the crystal suspension in motion. The liquor is fed at one end, and the crystals are discharged at the other end. The solution moves countercurrent to the coolant in the jacket. Four troughs of 3 metre length each may be arranged is series. If the flow of the liquor is to be more rapid than the cooling area permits, troughs are placed one over the other forming two or three decks. The incompletely crystallised liquor from the top deck is cascaded to the second deck and then to the lower deck.

The trough is formed out of metal sheets welded to an angle-iron frame at the top. A jacket covering the entire trough also made of mild steel sheets is attached. The spiral agitator is supported at either end outside the trough. A stuffing-box and gland attached to the cover, prevents the leakage of liquor from the opening provided for shaft. The shaft is driven by chain or belt.

10.6.6 DOUBLE PIPE CRYSTALLISER

This (Fig. 10.14) is somewhat similar to the trough crystalliser. It consists of a (concentric) double pipe, the outer pipe acting as a jacket. A long pitch agitator is rotated in the inner pipe.

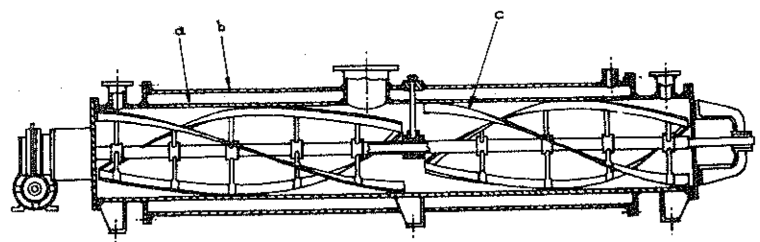


Fig. 10.14 Double pipe crystalliser
(a) inner pipe (b) outer pipe (c) agitator

A set of three lengths of pipes of about 3 metres each is connected in series. The crystalliser can be formed in tiers of three or more, with a set of three lengths in each tier. The hot liquor is fed to the upper tier. The outer pipe is usually of cast iron, while the inner pipe may be of cast iron, steel or stainless steel. The outer jacket pipe is provided with baffles, to create a spiral path for the flow of the cooling water.

The agitators are driven by sprockets and chains at 5 to 30 rpm. The blades of the agitator scrape the bottom of the pipe. A stuffing-box and gland arrangement is used to prevent leakage. The operation of the crystalliser is continuous. Liquor is fed at the top inlet and slurry is collected from the outlet continuously.

10.6.7 Coil Crystalliser

This (Fig. 10.15) consists of a horizontal cylindrical vessel, with a central shaft. An eccentric coil is mounted on the shaft, which also carries an agitator scroll, supported by diamond shaped self-cleaning arms. The cooling water enters and leaves the coil through the hollow shaft. The liquor is introduced through the charging nozzle at the top of the cylindrical vessel and discharged through the nozzle in the end plate near the bottom. The rotation of the coil is extremely slow, and varies from 1/4 to 1/7 rpm.

10.7 Design Considerations

The tank crystalliser is essentially a cylindrical vessel or trough. Tank crystallisers are non-pressure units and can be made

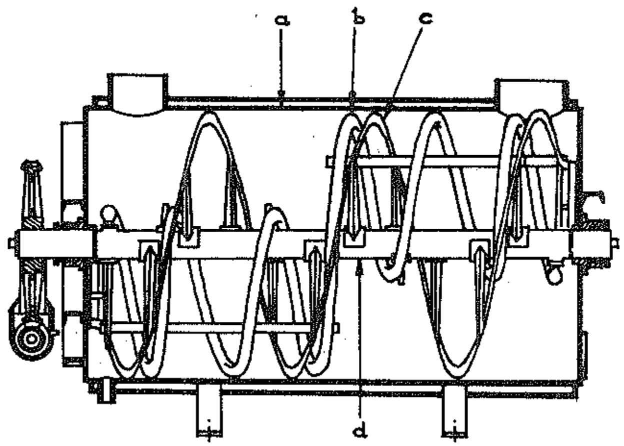


Fig 10.15 Coil crystalliser
(a) shell (b) coil (c) agitator (d) shaft

for minimum sheet thickness. The end closures are attached by simple flanged joints. The materials used must be such that they allow the transfer of heat and the unit must be structurally strong to resist erosion by the crystals and the action of the agitators. Fouling in the jacket is a common difficulty.

Vacuum crystalliser requires a cylindrical vessel with either a dished head at the top and a dished or conical head at the bottom. The construction is similar to evaporators. The vessel is to be designed for vacuum or external pressure and must have structural stability. In constructing this unit, all internal welds must be ground smooth and no projections or other points of possible turbulance or salt build-up must be left. Sight glasses above the boiling surface and in the vapour space ducting are essential. Manholes on the crystalliser should be well above the liquid level. Those placed within the spray or supersaturation zone, or in the settling chamber are, focal points for salt build-up and are very difficult to open for inspection. The high points in the settling chamber must be vented to the vapour section to prevent stagnant air build-up. Stagnant air build up results in corrosion and occasional surging of the liquor. Polished stainless steel or glass-lined mild steel are excellent materials of construction for the inner surfaces of crystallisers. Vessel design is given in Chapter 6.

The agitator must be run at a slow speed and should create only a gentle agitation. A stuffing-box and gland is used to prevent leakage between the shaft and the vessel wall. Details of agitator design are given in Chapter 14.

10.8 Numerical Problem

Standard vertical short tube evaporator (Plate III) (Calendria type).

Data

Evaporator drum under vacuum—external pressure 1 kg/cm^2
Amount of water to be evaporated—2500 kg/hr

Heating surface required — 220 m^2
Steam pressure — 1.5 kg/cm^2
Density of liquid — 990 kg/m^3
Density of vapour — 0.083 kg/m^3

Material

Evaporator — low carbon steel
Tubes — brass

Permissible stress for low carbon steel—980 kg/cm^2
Modulus of elasticity for low carbon steel—19.0×10^5 kg/cm^2
Modulus of elasticity for brass—9.5×10^5 kg/cm^2
Conical head at bottom—
Cone angle — 120°
Conical head at top—
Cone angle — 120°

10.8.1 CALENDRIA WITH VERTICAL TUBES

Use 100 mm outside diameter 1.5 mm thick tubes. The length of each tube is taken as 1220 mm, the effective length being 1165 mm.

$$\text{Number of tubes} = \frac{\text{Heat transfer area}}{\pi\times O.D.\text{ of tube}\times\text{length}} = \frac{220}{\pi\times0.1\times1.165}=605$$

Pitch of tube (Triangular)—125 mm

Area occupied by tubes from equation 9.5

$$a=605\times0.866\times(0.125)^2$$
$$=8.16\ m^2$$

Let the proportionality factor β be 0.9.

From equation 9.6

$$\text{Area required} = \frac{8.16}{0.9}=9.86\ m^2$$

Required area of central downtake

$$= 40\%\times\text{cross-sectional area of tubes}$$
$$= 0.4\times605\times\frac{\pi\times0.01}{4}$$
$$= 1.88\ m^2$$

Use a 1500 mm inside diameter and 1520 mm outside diameter pipe as a central downtake.

Actual area of downtake

$$= \frac{\pi\times1.52^2}{4}=1.81\ m^2$$

Total area of tube sheet

$$= 9.06+1.81=10.87\ m^2$$

Diameter of tube sheet

$$=\sqrt{\frac{10.87\times4}{\pi}}$$
$$= 3.71\ m$$
$$= 3710\ mm$$

10.8.1.1 CALENDRIA SHEET THICKNESS

From equation 6.3

$$t_s = \frac{1.65\times3710}{2\times980\times0.35-1.65}=3.67\ mm$$

The actual thickness must be much more so as to allow for corrosion and give rigidity to the shell. It may be taken as 10 to 12 mm.

10.8.1.2 TUBE SHEET THICKNESS

The tube sheet is similar to a fixed tube sheet heat exchanger.

From equation 9.3

$$K=\frac{E_s t_s(D_o-t_s)}{E_t N_2 t(d_o-t_t)}$$
$$=\frac{19.0\times10^5\times10(3710)}{9.5\times10^5\times605\times1.5\times(100-1.5)}$$
$$= 0.832$$

From equation 9.2

$$F=\left(\frac{0.832}{2+3\times0.832}\right)^{\frac{1}{4}}=0.43$$

From equation 9.1

$$t_{ts}= 0.43\times3710\sqrt{\frac{0.25\times1.65}{9.80}}$$
$$= 32.8\ mm$$

With corrosion allowance the thickness may be taken as 36 mm.

10.8.1.3 BOTTOM FLANGE OF CALENDRIA

A flange is provided at the bottom of calendria for fixing the conical head.

The size of flange selected is as follows :

Thickness of flange	40 mm
Number of bolts	112
Pitch circle diameter	3825 mm
Size of bolts	20 M
Outside diameter	3894 mm

10.8.2 EVAPORATOR DRUM

The entrainment separator to be used in the form of baffles, placed at the top of the drum. The diameter of the drum may be same as that for the calendria. However, it is necessary to check the size from the point of satisfactory entrainment separation.

From equation 10.1

$$R_a = \frac{\dfrac{V}{A}}{0.0172 \times \dfrac{\rho_L - \rho_V}{\rho_V}}$$

Assuming R_a $= 0.8$ for baffle system

$$A = \frac{\dfrac{2500}{3600} \times \dfrac{1}{0.013}}{0.8 \times 0.0172 \left\{\dfrac{990-0.083}{0.083}\right\}^{\frac{1}{2}}}$$

$$= 5.7 \text{ m}^2$$

Drum diameter $D = \sqrt{\dfrac{5.7 \times 4}{\pi}}$

$= 2.69$ m

Drum height (appr.) $= 3$ m

10.8.2.1 DRUM THICKNESS

Drum is under vacuum. Design is therefore based on an external pressure of 1 kg/cm². Assume a thickness of 10 mm.

From equation 6.14 (b), the critical external pressure is given by

$$p_c = \frac{2.42E}{(1-\mu^2)^{\frac{3}{4}}} \frac{\left(\dfrac{t}{D_0}\right)^{\frac{5}{2}}}{\left[\dfrac{L}{D_0} - 0.45\left(\dfrac{t}{D_0}\right)^{\frac{1}{2}}\right]}$$

$$= \frac{2.42 \times 19 \times 10^5}{(1-0.3^2)^{\frac{3}{4}}} \times \frac{\dfrac{10}{3630}}{\left[\left(\dfrac{3000}{3730}\right) - 0.45\left(\dfrac{10}{3730}\right)^{\frac{1}{2}}\right]}$$

$$= 2.35 \text{ kg/cm}^2$$

$$p_{all} = \frac{p_c}{4} = 0.588 \text{ kg/cm}^2$$

According to IS-2825 Appendix *F*

$$\frac{L}{D_0} = \frac{3000}{3710+20} = 0.806$$

$$\frac{D_0}{t} = \frac{3710+20}{10} = 373$$

Factor $B = 3500$

$$p_{all} = \frac{B}{14.22 \times 373} = \frac{3500}{14.22 \times 373}$$

$$= 0.662 \text{ kg/cm}^2$$

The thickness of 10 mm is not satisfactory. Assume a thickness of 12 mm

$$p_c = \frac{2.42 \times 19.10^5}{(1-0.3^2)^{\frac{3}{4}}} \times \frac{\left(\dfrac{12}{3734}\right)}{\left[\dfrac{3000}{3734} - 0.45\left(\dfrac{12}{3734}\right)^{\frac{1}{2}}\right]}$$

$$= 3.71 \text{ kg/cm}^2$$

$$p_{all} = \frac{3.71}{4} = 0.93 \text{ kg/cm}^2$$

According to IS-2825 Appendix *F*

$$\frac{L}{D_o} = \frac{3000}{3710+24} = 0.806$$

$$\frac{D_o}{t} = \frac{3710+24}{12} = 310$$

Factor $B = 4500$

$$p_{all} = \frac{4500}{14.22\times310} = 1.02 \text{ kg/cm}^2$$

The thickness may be taken as 13 mm inclusive of corrosion allowance.

The compressive stress from equation 6.16

$$f_c = \frac{pD}{2t} = \frac{1.02\times3734}{2\times12} = 159 \text{ kg/cm}^2$$

which is well within the permissible stress value.

The drum thickness may be reduced by use of stiffening rings.

10.8.2.2 CONICAL HEADS AT TOP AND BOTTOM

The head thickness is based on an external pressure of 1 kg/cm^2. Assume a thickness of 14 mm. As per (6.8.2.2) (b) (i) the conical head with an apex angle of 120° is taken as equivalent to the shell, with the length of shell as equal to the diameter of shell.

According to IS-2825 Appendix *F*

$$\frac{L}{D_o} = \frac{3710}{3710+28} = 0.996 \simeq 1$$

$$\frac{D_o}{t} = \frac{3710+28}{14} = 267$$

Factor $B = 4500$

$$p_{all} = \frac{4500}{14.22\times267} = 1.18 \text{ kg/cm}^2$$

The thickness may be taken same as the drum thickness.

Reading References

Coulson, J.M. and Richardson, J.F., *Chemical Engineering*, Vol. II, Pergamon, Oxford, (1968).

Cramer, H.W., *Chemical Engineering Practice*, Vol. VI, Butterworths, London, (1958).

Badger, W.L. and Banchero J.H., *Introduction to Chemical Engineering*, McGraw-Hill, New York, (1955).

Rumford, E., *Chemical Engineering Operations*, Constable, London, (1957).

Facoby, H.E., *Recent developments in Evaporation Equipment*, Chem. Eng. Prog., **59** (1), 72 (1963).

Scheimon, A.D., *Size Vapour-liquid Separators*, Petroleum Refiner, **42** (10), 165 (1963).

Crystallization Equipment Review, Chem. Proc. Eng., **51** (11), 67 (1970).

Perry, J.H., *Chemical Engineers' Handbook*, 5th ed., McGraw-Hill, New York, (1973).

Armstrong, A.J., *Scrapped Surface Crystallizers*, Chem. Process Eng., **51** (11), 59 (1970).

Brown, G.G., *Unit Operations*, John Wiley, New York, (1956).

A.I. Chem, E., *Equipment Testing Procedure Evaporators*, (1961).

Jorden, D.G., *Chemical Process Development*, Parts I and II, Interscience, New York, (1968).

CHAPTER 11

Distillation and Fractionation Equipment

11.1 Introduction

Separation of two or more components of a mixture by distillation, when the materials concerned have a definite vapour pressure, has been one of the most widely used processes. Separations, which are easy, can be accomplished by simple distillation, while more difficult separations are made by fractionation which is a process of multiple distillation for effecting the separation of two or more volatile components. Distillation is usually a batch process and may be carried out in a pipe still or a cylindrical vessel. Multiple distillation is carried out in a fractionating column, where continuous interchange between liquid from a condenser and vapour from a still or reboiler leads to the concentration of the lower boiling constituents of a mixture at the top of the column. The aim of the column design is to bring about intimate contact between vapour and liquid, while at the same time keeping both streams flowing evenly in opposite directions.

11.2 Basic Features of Fractionation Equipment

The entire equipment consists of a fractionating column, a still and a condenser or a receiver. Except for the column the design of the other items has been given elsewhere. Details of the fractionating column, are given in this chapter. The column, also known as tower, is essentially a tall vertical cylindrical vessel, with a number of nozzles. The internals of the column consist of a series of plates or trays or a variety of packings. Due to considerable height of the column, a ladder and a platform are provided for inspection and maintenance work. Apart from the normal features of a pressure

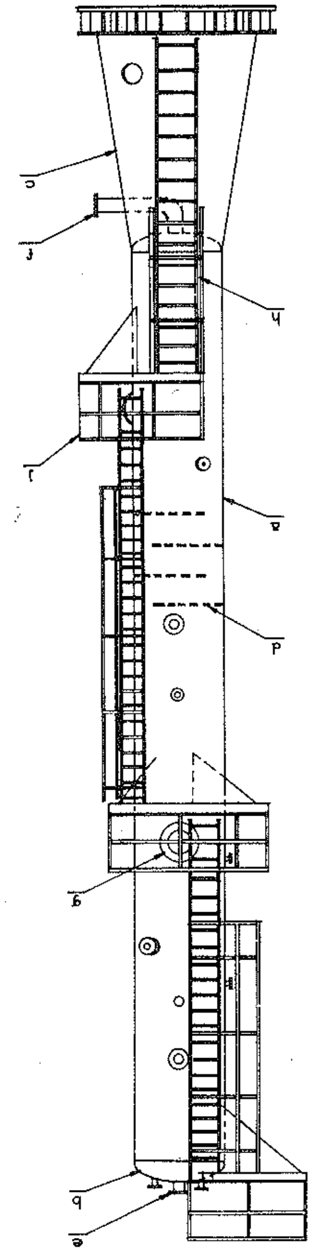


Fig. 11.1 Layout of a fractionating column
(a) shell (b) head (c) skirt support (d) tray (e) vapour outlet
(f) bottom outlet (g) manhole (h) ladder (i) platform

vessel, the column requires adequate supporting structure. Fig. 11.1 shows the main components of a column.

11.3 Stresses in Column Shell

The column is designed as a tall vertical vessel, either as a self-supporting structure or as a stabilized structure with guy ropes. Considering the interference caused by guy ropes, it is appropriate to design it as a self-supporting structure. In this case the stresses in the shell of the column may be assessed as follows:

(a) Circumferential and axial stresses due to internal pressure or vacuum inside the vessel.

(b) Compressive stresses resulting from dead loads such as the weight of the vessel and its contents, the weight of insulation, and attached equipment.

(c) Stresses resulting from bending moment caused by wind loads acting on the vessel and its attachments.

(d) Stresses due to eccentricity as a result of irregular load distribution.

(e) Stresses due to seismic (earthquake) forces

The above stresses may be combined to establish the final controlling stress. The stresses due to pressure act both in the circumferential and axial direction of the shell. The stresses due to other loads are all in the axial direction. It is possible to combine all the above stresses on the basis of shear energy criterion (distortion energy criterion, equation 6.11). However, the stresses in the axial direction are not the same over the entire height of the shell, and the result and stresses in the shell are therefore, likely to increase towards the bottom of the shell.

In analysing the stresses at various heights of the shell, calculations are made beginning at the top of the column. The selection and thickness of the top head is based on the internal pressure or vacuum. The minimum shell thickness in the upper portion of the column is usually determined by the circumferential stress resulting from internal pressure or vacuum.

The shell plate thickness at the top of the column can be specified by taking into consideration the corrosion allowance. The thickness for lower portions of the column are determined by taking into consideration all the other stresses. It is assumed that the stresses due to wind load and earthquake load will not occur simultaneously. The maximum value of either is therefore accepted for evaluation of combined stresses. At a height X from the top of the shell, the combined stresses in the axial direction are (excluding stresses due to eccentric loading) :

(a) Internal pressure and upwind side

$$f_{t_{max}} = (f_{wx} \text{ or } f_{sx}) + f_{ap} - f_{dx} \tag{11.1}$$

(b) External pressure and upwind side

$$f_{t_{max}} = (f_{wx} \text{ or } f_{sx}) - f_{ap} - f_{dx} \tag{11.2}$$

(c) Internal pressure and downwind side

$$f_{c_{max}} = (f_{wx} \text{ or } f_{sx}) + f_{dx} - f_{ap} \tag{11.3}$$

(d) External pressure and downwind side

$$f_{t_{max}} = (f_{wx} \text{ or } f_{sx}) + f_{dx} + f_{ap} \tag{11.4}$$

where

$f_{t_{max}}$ and $f_{c_{max}}$ — maximum tensile and compressive stresses

f_{wx} — stress due to wind load

f_{sx} — stress due to seismic load

f_{ap} — axial stress from internal or external pressure (uniform over the entire height)

f_{dx} — stress due to dead loads.

11.4 Determination of Shell Thickness (at Different Heights)

The procedure for determination of combined axial stresses at any distance 'X' is outlined above. The usual criterion of shell thickness is the maximum tensile stress, which should not exceed the allowable permissible tensile stress in the material of the shell. The minimum thickness is at the top of the shell end and is determined only on the basis of circumferential stress.

$$t_s = \frac{p\,D_i}{2fJ - p} + c \tag{11.5}$$

where t_s — shell thickness
p — design pressure
D_i — internal diameter of shell
f — allowable stress
J — joint efficiency
c — corrosion allowance.

This thickness may be satisfactory upto a certain distance from the top of the shell. In case of vacuum or low pressure the thickness t_s will be much less. A minimum value for thickness may be assumed in such cases according to the material of construction. In order to evaluate the distance 'X' it is necessary to determine the combined maximum stress at this distance. This stress should not exceed the allowable stress. The individual stresses at the distance 'X' in the axial direction are given below.

11.4.1 Axial Stress due to Pressure

$$f_{ap} = \frac{p\,D_i}{4(t_s - c)} \tag{11.6}$$

This is the same throughout the column height.

11.4.2 Stresses due to Dead Loads

(a) Compressive stress due to weight of shell upto a distance 'X'

$$f_{ds} = \frac{\text{weight of shell}}{\text{cross-section of shell}}$$

$$= \frac{\pi/4(D_o^2 - D_i^2)\,\rho_s(X)}{\pi/4(D_o^2 - D_i^2)}$$

$$= \frac{\text{weight of shell per unit height }(X)}{\pi\,D_m\,(t_s - c)} \tag{11.7}$$

where D_o and D_i are internal and external diameters of shell
ρ_s — density of shell material
D_m — mean diameter of shell

(b) Compressive stress due to weight of insulation at height 'X'

$$f_{d(ins)} = \frac{\pi\,D_{ins}\,t_{ins}\,\rho_{ins}}{\pi\,D_m\,(t_s - c)}$$

$$= \frac{\text{weight of insulation per unit height }(X)}{\pi\,D_m\,(t_s - c)} \tag{11.8}$$

D_{ins}, t_{ins} and ρ_{ins} are the diameter, thickness and density of insulation respectively.
D_m — mean diameter of shell
$D_{ins} \simeq D_m$ for large diameter columns.

(c) Compressive stress due to liquid in the column upto a height 'X'

$$f_{d(liq)} = \frac{\Sigma\text{ liquid weight per unit height }(X)}{\pi\,D_m\,(t_s - c)} \tag{11.9}$$

(d) Compressive stress due to attachments, such as internals, top head, platforms and ladders upto a height of X.

$$f_{d(att)} = \frac{\Sigma\text{ weight of attachments per unit height }(X)}{\pi\,D_m(t_s - c)} \tag{11.10}$$

Total compressive dead weight stress

$$f_{dx} = f_{ds} + f_{d(ins)} + f_{d(liq)} + f_{d(att)} \qquad 11.11$$

11.4.3 Stress due to Wind Load at a Distance 'X'

For details see 13.2.3.1 (b)

$$f_{wx} = \frac{M_w}{Z} \tag{11.12}$$

where Z—modulus of section for the area of shell

$$\simeq \frac{\pi}{4} D_o^2(t_s - c)$$

M_w is the bending moment due to wind load at a distance X.

The column is considered as a uniformly loaded cantilever beam. The stress will be compressive on the downwind side and tensile on the upwind side. The wind pressure is related to wind velocity. For calculations a maximum value of 125 kg per square metre on a flat surface may be taken as satisfactory. For round vessels a shape factor of 0.7 may be used. The projected area of either uninsulated or insulated tower may be taken for operation of wind load.

$$M_w = \frac{\text{wind load} \times \text{distance}}{2}$$

$$= \frac{0.7\,p_w\,D_o \times X}{2} \tag{11.13}$$

where p_w—wind pressure

$$f_{wx}=\frac{1.4\,p_w\,X^2}{\pi\,D_o\,(t_s-c)} \tag{11.14}$$

(compare equation 13.18)

11.4.4 Stress due to Eccentricity of Loads Tensile or Compressive according to the Position of Load

$$f_e=\frac{W_e\,(e)}{\pi/4\,D_o^2\,(t_s-c)} \tag{11.15}$$

where W_e—summation of eccentric loads

e—eccentricity.

11.4.5 Stresses due to Seismic Load

For details see 13.2.3.1 (c)

$$f_{sx}=\frac{M_{sx}}{\frac{\pi}{4}D_o^2\,(t_s-c)} \tag{11.16}$$

(Ref. equation 13.22)

where bending moment M_{sx} at a distance X is given by

(Ref. equation 13.21)

$$M_{sx}=\frac{CW(X^2)}{3}\left(\frac{3H-X}{H^2}\right) \tag{11.17}$$

where C—seismic coefficient

W—total weight of column

H—total height of column

11.4.6 Determination of Height 'X'

To determine the value of X, all the stresses (except stresses due to eccentricity of load and seismic load) acting in the axial direction may be added and equated to the allowable tensile stress as follows:

$$\frac{p_w\,X^2}{\pi/4\,D_o^2(t_s-c)}+\frac{pD_i}{4(t_s-c)}-\frac{\Sigma\text{ dead weights }(X)}{\pi D_m\,(t_s-c)}=f_{tmax} \tag{11.18}$$

If the joint efficiency J is taken into account

$$\frac{p_w\,X^2}{\pi/4\,D_o^2\,(t_s-c)}+\frac{\text{dead weight }(X)}{\pi\,D_m\,(t_s-c)}+\frac{pD_i}{4\,(f_s-c)}-f_{(all)}\,J=0 \tag{11.19}$$

In the above equation (f_{tmax}) is substituted by allowable tensile stress $f_{(all)}$. The possibility of the wind load and earthquake load operating simultaneously is remote. Therefore the stresses due to wind load and earthquake load are computed separately and the most adverse loading condition is used to calculate the maximum axial stress.

The above equation can be stated in the form of a quadratic equation.

$$aX^2+bX+c=0$$

from which

$$X=\frac{-b\pm\sqrt{b^2-4ac}}{2a} \tag{11.20}$$

After determining the value of X, it is desirable to adjust the plate thickness, t_s, for the top course so that the height of portion X, will be a multiple of the plate width used. Usually the plate thickness originally selected is satisfactory upto a considerable height. Plates below the distance X must have an increased thickness. Calculations of axial stresses with the increase in thickness according to equations (11.7 to 11.20) are repeated, and subsequent height ranges to correspond with increased thicknesses determined. The procedure is repeated till the entire height of the column is covered. Different sections of the column shall be formed out of a single plate with one longitudinal joint only. Longitudinal joints of adjacent sections shall be staggered. Weld seams shall be, to the extent possible, clear of brackets, supports, etc., so that seams are readily visible for examination.

The stressed condition of the shell indicated so far, is on the assumption that the column is in operation, i.e., it is working with all attachments, insulation, etc. and under requisite pressure and temperature conditions. However, during certain periods the column may be shut down, when stresses due to pressure conditions are absent. It is also necessary to consider the stresses during erection of the column, when there is no insulation load or internal and external attachments, liquid load, etc. The pressure conditions are also absent. Various cases of stressed conditions should be investigated individually to determine the controlling condition. If the column is designed for high pressure service, the limiting condition will usually exist when the shell is under pressure and under a wind load. Columns designed for low pressure service may have maximum stress conditions when the erected empty column is exposed to a high wind load.

11.5 Elastic Stability under Compressive Stresses

Determination of shell thickness t_s is based on the permissible tensile stress requirements. The compressive stress produced on downwind side must be checked to ensure elastic stability. Failure due to Euler's buckling involves bending of the shell as a strut, which is generally not significant in vertical vessels. Wrinkling of the shell must, however, be checked and avoided. The safe compressive stress, which can be imposed without failure by wrinkling is given by

$$f_c = \frac{1}{12}\,\frac{E}{\sqrt{3(1-\mu^2)}} \times \frac{(t_s - c)}{D_o/2} \tag{11.21}$$

If $\mu = 0.3$

$$f_c = 0.105\, E\left(\frac{t_s - c}{D_o}\right) \tag{11.22}$$

where μ—Poisson's ratio

E—modulus of elasticity.

11.6 Allowable Deflection

Column deflection becomes an important factor in the design when the ratio of height to diameter is relatively large. Determination of static deflection of column is essential for accurate vibration analysis. It is general practice to allow a maximum

deflection in mm of (5×overall height of column in metres) for vertical columns. A vibration analysis shall be made on columns with 30 metres overall height and above and with height to diameter ratio ranging from 18 to 30. The total deflection at the top of the column is calculated as an algebraic sum by the following equation

$$\delta = \sum_{i=1}^{n}\left\{\left[\frac{Wl^3}{8EI} + \frac{(\Sigma W)l^3}{3EI} + \frac{(\Sigma M)l^2}{2EI}\right] + \sum_{1}^{l}\left[\frac{Wl^2}{6EI} + \frac{(\Sigma W)l^2}{2EI} + \frac{(\Sigma M)l}{EI}\right]_i (l_{i+1})\right\} \tag{11.23}$$

where δ—deflection

n—number of sections from top in which the column is suitably divided

W—wind load on the section

ΣW—concentrated load on the section at the end (due to wind load on sections above)

ΣM—moment at the end of the section (due to moment of the sections above)

I—moment of inertia of the cross-section

E—modulus of elasticity

l—length of the column section under consideration.

To calculate the total deflection it is necessary to calculate the deflection of each section of the column having uniform thickness and constant value of E and I. If the wall thickness changes within the section to be calculated, then that section should be divided into two or more sections in accordance with the change in wall thickness. The topmost section of the column has no concentrated load or moment, therefore ΣW and ΣM are equal to zero, for calculation of deflection of this section. If the deflection estimated by the above equation is greater than the permissible deflection for the column height, the shell thickness will have to be increased. The new deflection will be recalculated with the revised shell thickness until it falls within the specification.

11.7 Column Internal Details

Various components known as internals, located inside a fractionating column, are designed to create an intimate contact between the vapour and the liquid, while at the same time keeping both streams flowing evenly in opposite directions. According to the mode of contact between vapour and liquid, columns can be classified as equilibrium stage columns and differential stage columns. In equilibrium stage columns the principal mass transfer between phases takes place in stages on a series of plates or trays or in the vicinity of either. In differential columns the mass transfer between phases takes place gradually throughout the entire height of the column.

11.7.1 Equilibrium Stage Columns

Depending on the method of contact and the flow of vapour and liquid, the contacting equipment may be classified as follows.

11.7.1.1 PLATES WITHOUT DOWNCOMERS

These are generally flat circular plates of a diameter slightly smaller than the internal diameter of the column, located with suitable spacing from the bottom to the top of the column. They are provided with a large number of holes or slots through which the liquid and the vapour pass. The Turbogrid (Fig. 11.2) is a flat plate with long rectangular slots. The grid can also be formed by closely spaced parallel bars located

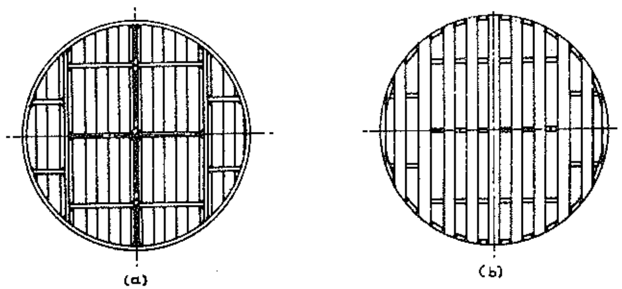


Fig. 11.2 Turbogrid plate or tray
(a) top view (b) bottom view

in one circular plane, instead of a single flat plate. The Dual flow is a flat plate with circular holes. The Ripple tray is corrugated instead of flat. The Kittel tray has slots, which cause the vapour to move the liquid towards the centre of one tray and towards the wall of the next. Invariably plates are assembled with slots of one plate at right angles to those on the adjacent plate, so as to prevent the liquid falling from one slot through the corresponding slot on the plate below. Holes on consecutive plates are also staggered.

11.7.1.2 PLATES WITH DOWNCOMERS

These are circular flat plates or trays with provision of passages known as downcomers for the downward flow of the liquid. The downcomer area for each plate will be generally limited to 10% of the total area of the plate. Various arrangements of downcomers, weirs, and contacting devices are adopted for handling liquids and vapours.

(a) Downcomers and weirs

The function of the downcomer is to provide a passage for the downward flow of the liquid from a top tray to a lower tray. An inlet weir helps to distribute the liquid as it enters the tray from the downcomer, and prevents direct impingement of the liquid on the contacting devices. For small liquid flows circular or pipe type downcomers (Fig. 11.3) are commonly

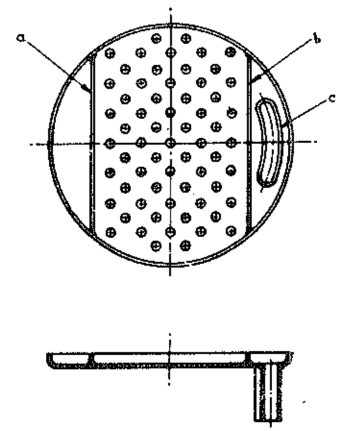


Fig. 11.3 Circular or pipe type downcomer and weir
(a) inlet weir (b) outlet weir (c) down pipe

used. A projection of a corresponding shape, provided on the downcomer acts as an exit weir. The downcomer can be formed as segmental shape (Fig. 11.4) where a raised portion on edge acts as a weir. A downcomer can also be formed by creating

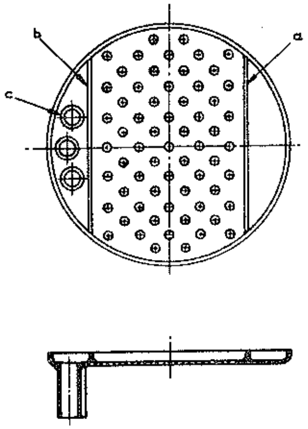


Fig. 11.4 Segmental downcomer and weir
(a) inlet weir (b) outlet weir (c) segmental downcomer

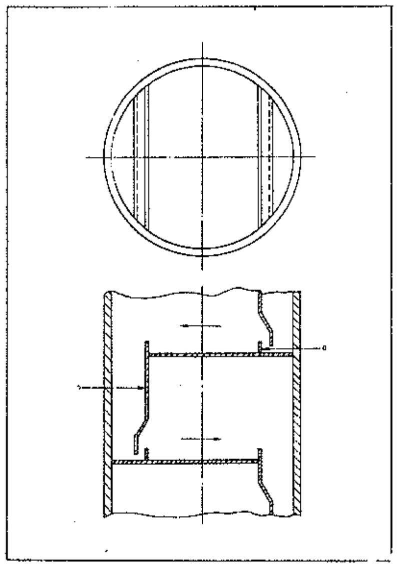


Fig. 11.5 Chord type downcomer and weir (across flow)
(a) inlet weir (b) outlet weir with downcomer

a passage between the column wall and a vertical plate extending across the chord of the tray. The weir in such a case consists of a flat plate extending across the chord, and is known as a chord type weir. (Fig. 11.5). The flow of liquid in these cases is across the tray and is known as single pass cross flow. A reverse flow is created by a weir system shown in Fig. 11.6 (a). It is recommended when the liquid flows become small in comparison to vapour flows. A radial flow is created by downcomers placed at the centre and around the circumference of alternate trays [Fig. 11.6 (b)].

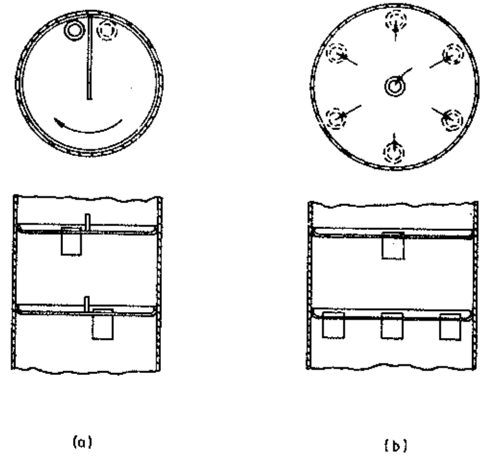


Fig. 11.6 Single pass flow
(a) reverse flow (b) radial flow

For multipass arrangements the system of downcomers and weirs is as shown in Fig. 11.7. The downcomers in these cases are formed of two parallel plates extending across the diameter of the column. Such downcomers are generally tapered at the bottom to facilitate maintenance of liquid seal. Reverse flow, radial flow and multipass arrangements help to obtain satisfactory liquid flow and distribution particularly in large columns. In all the cases, the weirs should be of sufficient height.

Multiple downcomer trays are also designed with a row of narrow rectangular troughs spaced across the tray to act as downcomers. The bottom portion of the trough is not fully

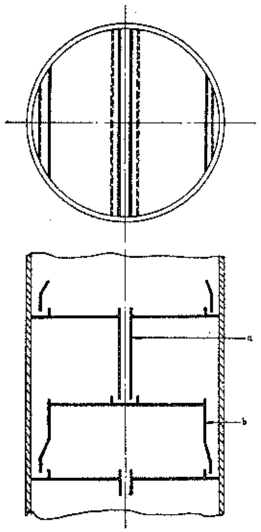


Fig. 11.7 Multipass flow
(a) central downcomer (b) side downcomer

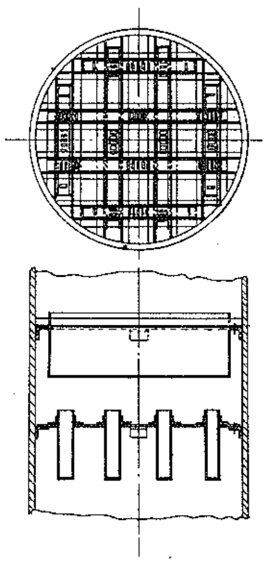


Fig. 11.8 Multiple downcomer tray

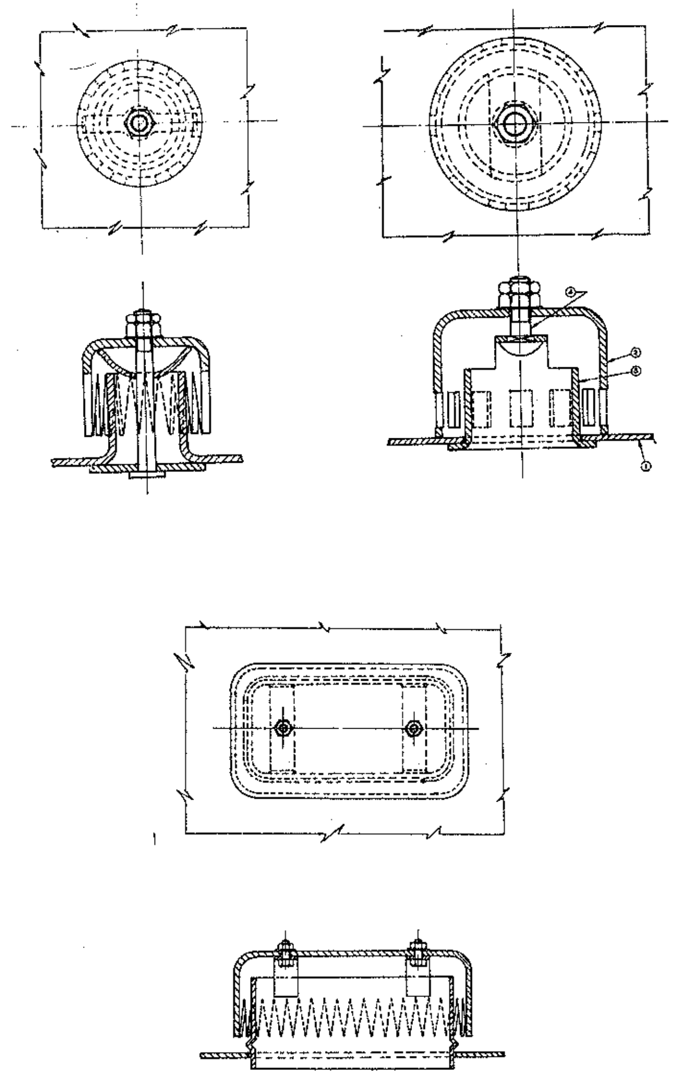


Fig. 11.9 Bubble caps
(a) circular cap with riser (slots on edge) (b) circular cap with riser
(1) tray (2) cap (3) riser (4) bolt
(c) rectangular cap with riser

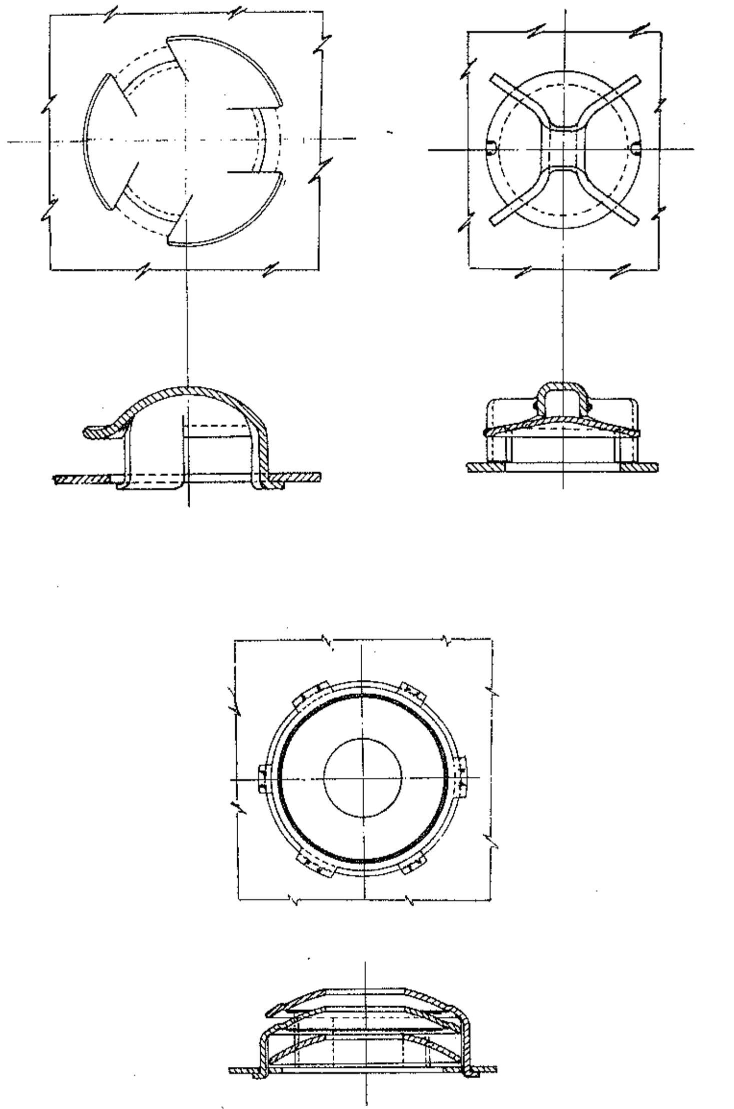


Fig. 11.10 Valves
(a) and (b) flexi-valve tray (c) glitsch, ballast tray

open, but is provided with slots. The position of the downcomers is staggered on consecutive trays (Fig. 11.8). The length of the weir is generally 60 to 75% of the diameter for a cross flow tray and 50 to 60% of the diameter for a multipass.

(b) *Contacting devices*

(i) *Bubble caps*—The area of the plate or tray, between the inlet and exit weirs is provided with a large number of holes over which cylindrical risers or chimneys are placed. Caps in the form of inverted cups are located over the risers. The vapour rises up through the holes and the risers and subsequently through a number of rectangular, triangular or trapezoidal slots provided at the rim of the caps [Fig. 11.9 (a) and (b)]. Table 11.1 gives the size of caps used. In special cases rectangular risers or chimneys with tunnel shaped caps are used [Fig. 11.9 (c)]. Uniflux tray is made up of interlocking S-shaped sections which form a series of transverse tunnel caps with vapour outlets on one side only.

Table 11.1

Diameter of towers (metres)	Cap size (round) (cm)
0.8	5
0.8—1.5	7.5
1.2—4.5	10
3—6	15

(ii) *Valves*—The bubble caps can be replaced by poppet type valves, which are lifted by rising vapours and act as variable orifices. These permit effective mixing of liquid and vapour over a wide range of loading. These valves are circular with domed or flat caps. Rectangular caps and caps with downward facing cones cannot rotate as the circular ones might do under certain process conditions.

Circular liftable caps are restricted in their lift by suitable constructional features such as special legs or spiral webs. Some caps are designed by a double lift system, where at low vapour loads the inner light cap lifts and as the pressure drop through the cap increases and overcomes the weight and friction of the heavier cap, it also lifts and creates an additional area for flow of the vapour. Trays with such caps are known as Flexi trays [Fig. 11.10 (a) and (b)]. Ballast trays have valves with flat caps or discs [Fig. 11.10 (c)].

(iii) *Sieve plates*—There is no separate contacting device in these type of plates. Instead, a large number of holes distributed uniformly over the plate act as passage for vapours, but at the same time the liquid is prevented from draining through the holes. In some cases, holes are punched with protruding lips. This is known as a jet tray. In another design, in addition to small holes a few angular slots are made in various positions. In a modified design of the sieve tray, a layer of expanded metal is attached to the tray. The vapour flowing upwards through the perforated tray is uniformly deflected by the expanded metal thus imparting a horizontal component to the vapour flow.

(c) *Distribution baffles*

With uniform cap spacing it is not possible to locate all caps at the end of the rows close to the column wall. Due to excessive end clearance the liquid will tend to short circuit such rows. If the end space exceeds the spacing by 2.5 cm, the gaps are closed by redistribution baffles. The clearance between these baffles and the caps should be equal to the cap spacing.

(d) *Tray drainage*

Tray drain holes are required to permit drainage of the liquid hold up of the column in case of shut down. These holes must be sufficiently large, so that they may not get plugged by sediments or polymers. The current practice is to make holes of 1 to 1.5 cm diameters along the weir.

11.7.1.3 FEED SYSTEMS

The liquid feed to the column is above the top tray and at an intermediate position. A pipe line located above the top tray,

and just opposite the inlet weir feeds the top tray. An inlet baffle may be provided opposite the pipe [Fig. 11.11(a) and

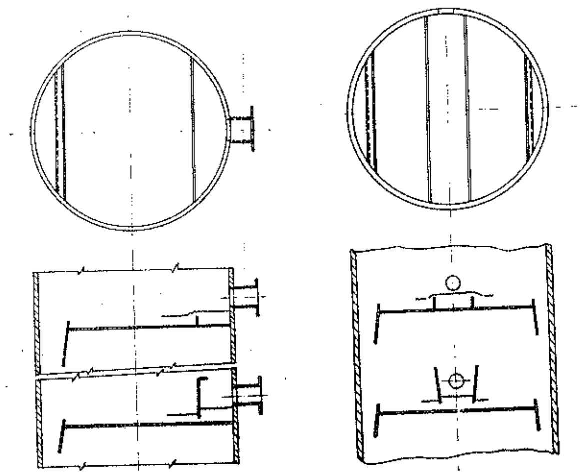


Fig. 11.11 Top tray feed inlet with weir or baffle
(a) single pass (b) multipass

(b)] to create uniform distribution. Various methods of intermediate feed arrangements are shown in Fig. 11.12. Liquid feed can be introduced at A, B, C or F. Location A may be satisfactory. However in the case of a high pressure system where vapour disengagement out of the downcomer is extremely critical or in cases where the temperature of the feed inlet would cause flashing of the downcomer liquid, this arrangement may limit column capacity. Location B is generally not desirable because of the unnecessary turbulance introduced in the downcomer area. Location C is the most convenient location. It is provided with a baffle plate and a guard plate. Location F is preferable for high velocity feeds.

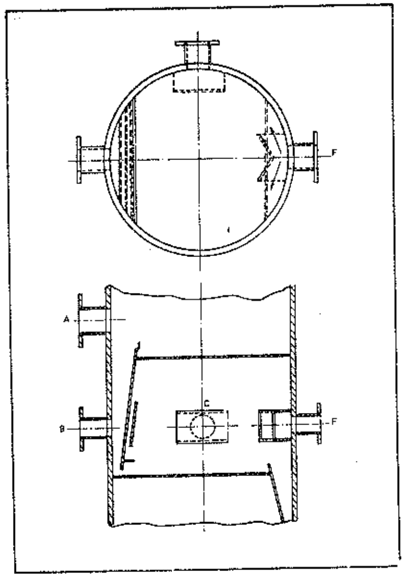


Fig. 11.12 Intermediate feed. A B, C and F—Alternate liquid feeds

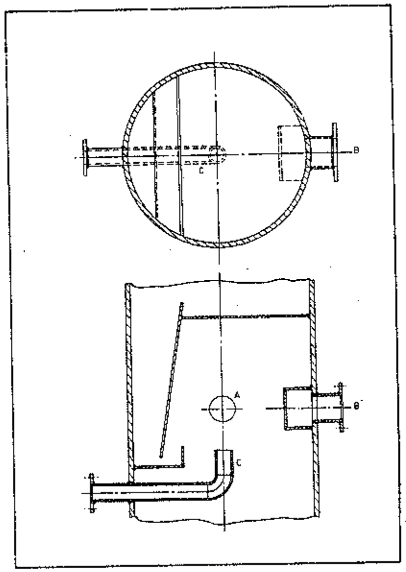


Fig. 11.13 Vapour feeds. A, B, C—Alternate vapour feeds

The vapour feed is generally located below the bottom tray at A as shown in Fig. 11.13. The vapour is introduced parallel to the bottom downcomer at a spacing of about 50 to 60 cm. For multipass trays it is necessary to feed each compartment equally and allow for vapour equalisation between sections. A vapour inlet nozzle causing impingement of the vapour stream should be provided with a baffle B or piping C.

11.7.1.4 DRAW OFFS

The drawing of the liquid from the bottom of the column is either by use of a chimney tray or by a draw pan as indicated in Fig. 11.14 (a) and (b). The chimney tray has no down-

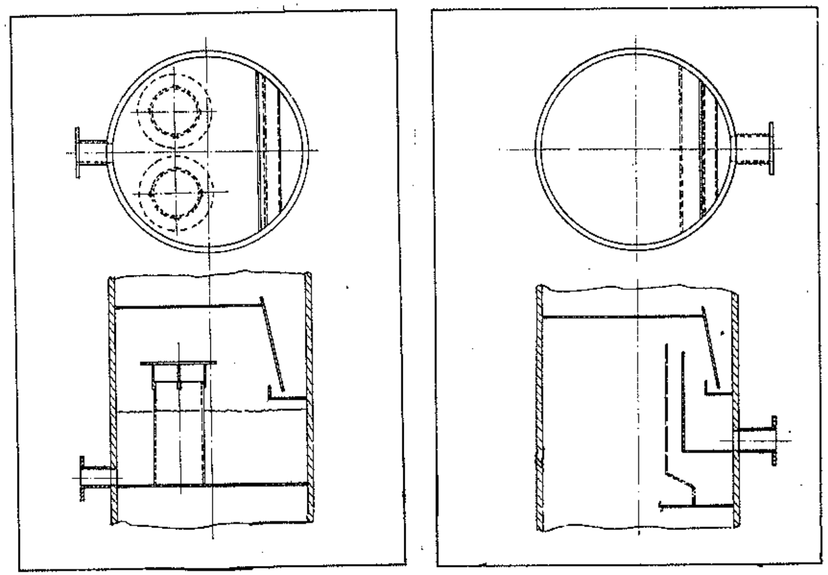


Fig. 11.14 Draw off (a) chimney tray (b) draw pan

comer and is in flush with a draw nozzle. Chimneys are normally of a size approximately 15% of tower area. These should be so located as to prevent directing vapour against the adjoining downcomer.

11.7.1.5 MANHOLES

Large trays are provided with manholes (Fig. 11.15) at intervals with a minimum opening of 30 cm×40 cm. At these

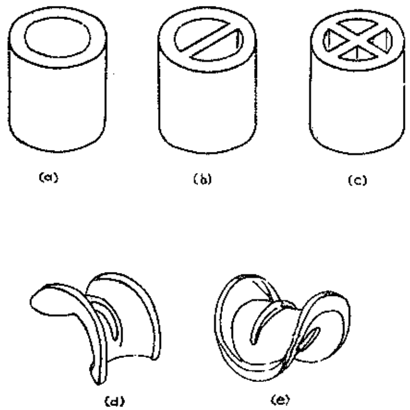


Fig. 11.15 Packings (a) raschig ring (b) lessig ring (c) cross partition ring (d) intolax saddle (e) berl saddle

intervals the plate spacing is increased to facilitate entry of a person into the tower.

11.7.2 DIFFERENTIAL COLUMN

11.7.2.1 PACKED COLUMN

Instead of using plates or trays this column is filled with suitable packing material. The liquid flows downwards over the surface of the packing in the form of thin films, while the vapour rises. The liquid is introduced at the top of the packing by means of a distributing plate and the vapour is introduced beneath a grid which supports the packing. The packings have different shapes. 'Raschig rings' are short cylindrical rings, with length equal to diameter, while Berl saddles are shaped somewhat similar to saddles. Fig. 11.15 shows other types of packings also.

11.7.2.2 LIQUID DISTRIBUTORS

Various methods are adopted for distribution of the liquid, to effect proper wetting of the packing. Piping arrangements

flat perforated plates, notched troughs or weir type distributors (Fig. 11.16) are commonly used. For column diameters less than 60 cm a single pipe nozzle or a shower with multiple pipe nozzles may be satisfactory. In case of single pipe nozzles, a flat plate

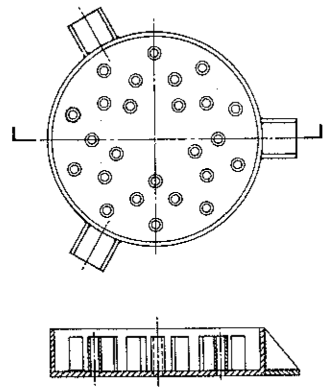


Fig. 11.16 Weir type liquid distributor

should be provided at the centre to allow the liquid to splash on it and thereby distribute it. The distance between the nozzle tip and the splash plate should be kept at 15 to 25 cm. In the case

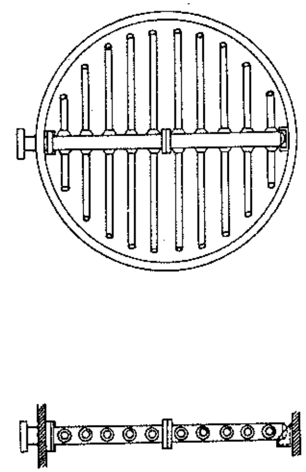


Fig. 11.17 Spider type distributor

of a shower, the spray should cover 50% of the cross-section. For large diameters cross pipes or a spider pipe with multiple arms (Fig. 11.17) or a pipe bent to a circular form should be provided with nozzles, so that the liquid can be distributed evenly on the cross-section.

11.7.2.3 LIQUID REDISTRIBUTION AND WALL WIPERS

Despite its efficient distribution on the top plate, the liquid has a tendency to drift towards the wall as it moves downwards. After a certain vertical height of liquid travel, liquid concentration falls considerably in the inner cross-section. It is therefore necessary to direct the liquid towards the centre after it has travelled a certain distance. This is done by dividing the packing height into more than one section. A general criterion is to limit the heights to approximately 3 column diameters for Raschig rings and 5 to 10 column diameters for saddle packings. Between two packing sections, redistributors and wall wipers are provided to bring the liquid back to the centre from the wall (Fig. 11.18).

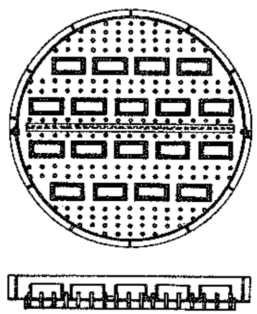


Fig. 11.18 Liquid redistributor

11.7.2.4 SUPPORT PLATES

These plates are located at the bottom of the column and support the packings. Conventional type of support plates are perforated plates or screens and grid bar assembly (Fig. 11.19). In the former the free area is limited to 20 to 30% and

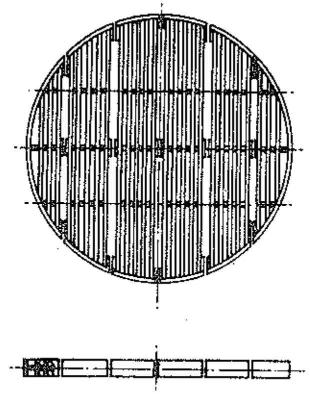


Fig. 11.19 Grid bar support plate

in the latter to 30 to 40%. In order to achieve better performance a minimum free area of 50% is required. In such cases a gas-injection-weir-type (Fig. 11.20) support plate is used. These plates have several advantages over the conventional types.

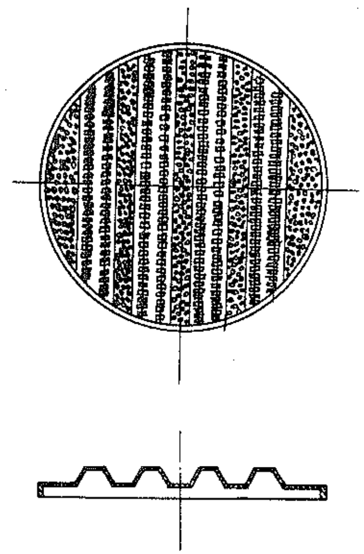


Fig. 11.20 Gas-injection type support plate

11.7.2.5 HOLD-DOWN PLATES

A hold-down plate (Fig. 11.21) on top of a packed section serves an important purpose by resting directly on top of the packing and restraining the bed under conditions of high gas

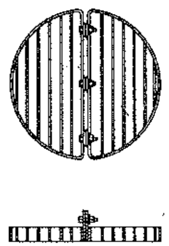


Fig. 11.21 Hold-down plate

rates or fluctuating gas flows, when, without such restraint, ceramic or carbon packings may break and fall back through the bed, thus reducing its capacity. The free area of the plate is decided between as large a value of the area as possible and a value small enough to prevent the packing passing through the plate.

11.8 Design and Construction Features of Column Internals

11.8.1 PLATE AND TRAY CONSTRUCTION

Several factors control the design and construction of plates and trays. These factors are:

(1) load on the tray due to dead weight, liquid weight and impact due to downcoming liquid
(2) expansion due to rise in temperature
(3) ease of installation and fabrication
(4) ease of access and maintenance
(5) method of support
(6) material of construction
(7) safety.

Plates or trays can be constructed either as one piece trays or as sectional trays. A one piece tray may be made of cast iron, cast as one piece including risers and downcomers. It may also be formed conveniently from a thin sheet of ductile material such as copper or steel, with a thickness from 2 mm to 6 mm depending on the diameter and the material. A one piece sieve tray is essentially a flat plate, which is drilled or punched for providing the holes. Small holes are drilled, while large holes can be punched. Holes are usually from 4 to 6 mm diameter located on an equilateral triangular spacing on centres varying from 2.5 to 5 times the hole diameter. In special cases holes can be upto 20 mm diameter. In the case of punched holes, in copper alloy or carbon steel plates the minimum hole diameter which can be punched is equal to the tray thickness, for stainless steel plates, the hole diameter must be $1\frac{1}{2}$ to 2 times the plate thickness. In the case of bubble cap and valve trays, the tray is formed by cutting a circular blank from a thin sheet of ductile material and then rolling and bending the edge of the blank to form the vertical rim of the tray. Holes are drilled in the tray to suit the size of the bubble cap risers or valves. The spacing is usually on equilateral triangular positions. The sectional tray is built from sections in the form of floor plates cut from sheets, which are laid on the supporting beams and a peripheral ring. A clearance is provided between adjacent sections and clamping devices are used for fixing.

The cast iron tray is able to withstand compressive forces created due to thermal expansion within reasonable limits. The diameters of such trays are also limited to small sizes. The one piece wrought tray made of ductile materials is relatively thin and has therefore a limited ability to absorb forces due to thermal expansion. The provision of a packing seal between the edge of the tray and the column wall (Fig. 11.22) will help to relieve these and would therefore prevent distortion of the tray floor. One of the main advantage of the sectional tray is its ability to cope up with thermal expansion. The individual sections of the tray are placed on the supporting structure, with an asbestos jointing material inserted between sections and between the support member. Each

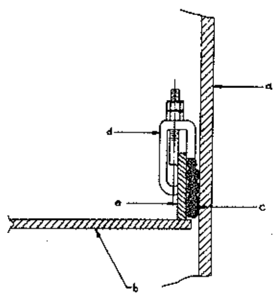


Fig. 11.22 Tray packing arrangements (a) shell (b) tray (c) packing (d) retaining clamp (e) packing collar

section is finally held by frictional clamping devices (Fig. 11.23). The sections therefore move slightly under the influence of the thermal forces and any tendency to distortion is prevented. However such trays might leak if the joints between sections are not satisfactorily made.

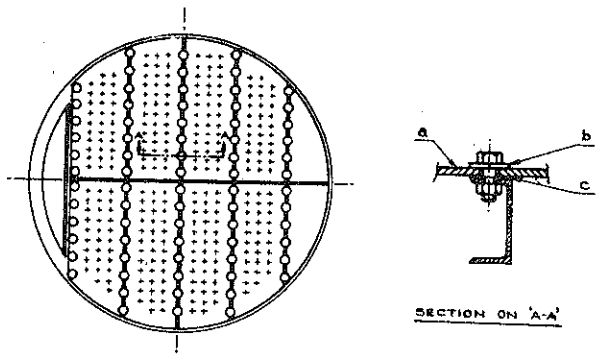


Fig. 11.23 Sectional tray (a) tray (b) frictional washer (c) packing

11.8.2 Loading Conditions for Plates and Trays

Plates and trays have to be maintained flat in order to provide a uniform seal of the liquid on their surfaces. Under the action of the various loads, plates and trays are likely to

deflect excessively, unless they are made sufficiently thick or an adequate supporting system is provided. The loads which cause deflection are :

(1) weight of the tray with contacting devices and downcomers

(2) weight of the liquid

(3) impact load of the downcoming liquid, which is given by,

$$\text{load} = \frac{wv}{g} \tag{11.24}$$

where

w—wt. of the liquid per second

v—velocity per second

g—gravitation constant

(4) Expansion due to rise in temperature (if prevented). It is usual to provide for free expansion. Load due to this may therefore be ignored.

(5) Weight of maintenance personnel and tools.

During working conditions the loads (1) to (3) need to be considered for assessment of deflection. In general a deflection of 3 mm is permissible. In special cases this may be limited to 2 mm.

During cleaning and assembly or inspection operations, loads (1) and (5) should be considered. In this case load due to item (5) is taken as a maximum of 150 kg concentrated load at any point on the tray. The design is based not on the permissible deflection but on the permissible stresses. Certain guide lines for tray loadings are as follows—

(*a*) Fractionating trays shall be designed for a uniform live load of 60 kg/m² or the weight of the liquid at maximum height of weir setting whichever is greater.

(*b*) Maximum deviation from horizontal at normal tray loading shall not exceed $\frac{1}{900}$th of span.

(*c*) Areas under downcomers shall be designed for uniform load of 320 kg/m² or weight of liquid for one-half the height of downcomer whichever is greater.

(*d*) Pans shall be designed for a uniform live load of 80 kg/m^2 or the weight of liquid at the maximum operating level on the pan, whichever is greater.

(*e*) Baffles shall be designed for a live load of 80 kg/m^2 on the projected horizontal area.

(*f*) Trays, baffles and pans, together with their support members shall be designed for a concentrated load of 150 kg at any point on the installed assembly independent of other design live loads and based on allowable stress at normal temperature.

11.8.3 DEFLECTION AND STRESSES

Deflection and stress determination for trays will depend on the methods used for the supporting structure. Usually three methods are adopted.

(*a*) The tray is supported on a peripheral ring. This method is adopted only for small diameter columns (Fig. 11.21).

(*b*) The tray is supported on a truss made up of angles, channel or trapezoidal section members rolled or fabricated from sheets. The ends of the members or purlins of the truss, are supported by the column wall, through clamps, which are welded to the wall. Alternately these may be supported on a support ring, welded to the wall (Fig. 11.24). For large trays,

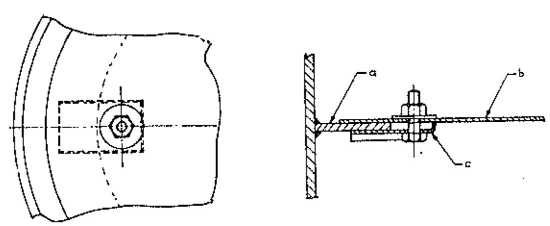


Fig. 11.24 Peripheral ring supports for tray
(a) ring (b) tray (c) clamp

the tray is supported on a number of parallel purlins. All the purlins are made to rest centrally on a major beam specially fabricated in the form of an I-section with variable depth

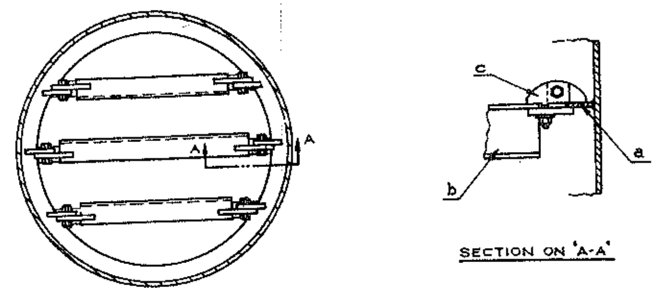


Fig. 11.25 Truss support for tray
(a) ring (b) structural section (c) clamp (d) fixing bolt

(Fig. 11.25). In addition, these purlins are supported and clamped at their ends to a peripheral ring. For sectional trays this arrangement is most suitable.

(*c*) Vertical supports in the form of round or hexagonal bars, extend between trays and are evenly spaced across the area of the tray (Fig. 11.24). This arrangement is suitable for one piece wrought trays. The deflections and stresses in the three methods indicated can be calculated as follows.

(*a*) *Tray supported on a peripheral ring*

This may be considered as a circular flat plate, fixed at the circumference and subjected to a uniform load over its surface. The deflection is given by

$$\delta = \frac{3}{16} \, \frac{m^2-1}{Em^2\, t^3} \cdot p_L R^4 \qquad (11.25)$$

If Poisson's ratio for metal is taken as $\frac{1}{3}$, i.e., $m=3$

then
$$\delta = \frac{1}{6} \, \frac{p_L R^4}{E\, t^3} \qquad (11.26)$$

where

E — modulus of elasticity

t — thickness of the plate

$$p_L = \frac{\text{loads (1) to (3) causing deflection as per (11.8.2)}}{\text{area of the plate}}$$

R — radius of the plate

$\frac{1}{m}$ — Poisson's ratio

The above equations may have to be modified in view of the actual construction of each type of tray.

(1) Fixing of the tray at the edge may not be complete, it may only be partial.

(2) The load may not be as uniformly distributed, as is assumed.

(3) The tray is perforated and not solid, as in the case of Dual flow, Turbogrid or sieve tray. In the case of Bubble cap and valve tray, the holes are reinforced by risers.

The value of the constant in the equation 11.26 may be taken as $\frac{1}{2}$ instead of $\frac{1}{8}$.

During cleaning and assembly operations as per 11.8.2, (1) a uniformly distributed load of tray and downcomers will produce a stress in the tray which is given by

$$f_1 = \frac{3}{4}\left(\frac{pR^2}{t^2}\right) \tag{11.27}$$

where p — uniformly distributed load per unit area

R — radius of tray

t — thickness of tray

While due to the concentrated load of maintenance personnel and tools [(11.8.2 (5)] the stress produced in the tray will be given by

$$f_2 = \frac{3w}{2\pi t^2}\left(1.33 \log \frac{R}{x} - 1\right) \tag{11.28}$$

where w — concentrated load at the centre of tray

t — thickness of tray

R — radius of tray

x — any intermediate radius

(In equation 11.27 and 11.28, the value of the Poisson's ratio is taken as 0.33).

The load in equation 11.28 is taken at the centre of tray for maximum stress condition. At the centre of the tray ($x=0$), the stress f_2 will be infinitely large. However in actual practice the load will be operating on a small area. The stress f_2 will be then finite. The total stress will be sum f_1+f_2.

It may be seen from the above calculations of deflection and stress, that a tray supported merely on a peripheral ring has to be carefully handled with minimum loadings during actual operation and maintenance. Its application is strictly limited to small diameter columns.

(*b*) *Tray supported on a truss*

The size of each purlin or beam of the truss will be determined by the span and the load shared by the purlin. The central purlin will have the maximum span which is almost equal to the internal diameter of the column. Intermediate purlins will have lesser spans. The load on the tray will be shared by the purlins in proportion to the area of the tray supported by it. The deflection of each purlin has to be limited. The deflection is given by (Table 3.1).

$$\delta = \frac{5Wl^3}{384\,EI} \tag{11.29}$$

where W — total load carried by the purlin, including its own weight, during working

l — span of purlin

E — modulus of elasticity

I — moment of inertia of the cross-section of purlin.

The stress in the purlin is given by

$$f_1 = \frac{Wl}{8} \times \frac{1}{Z} \tag{11.30}$$

for uniformly distributed load of (1) indicated in (11.8.2)

and $$f_2 = \frac{Wl}{4} \times \frac{1}{Z}$$

for concentrated load of (5) indicated in (11.8.2)

where Z — modulus of section of the purlin.

(*i*) *Support rings*—The recommended sizes of support rings for different column diameters are given in Table 11.2.

Table 11.2

Column Diameter (metres)	Approximate Ring Size	
	Carbon steel (mm)	Alloy steel (mm)
1.5	35×6	35×6
1.5—2.5	50×6	50×6
2.5—3.0	65×6	65×6
3.0—3.5	65×8	65×6
3.5—4.5	75×8	75×6
4.5—6.0	90×8	90×6

(*ii*) *Purlins or beams*—These may be fabricated in the form of angles, channels or trapezoidal sections from sheets, of approximately 2 to 6 mm thickness depending on the material of construction and the load. The following conditions may be observed as far as possible.

(1) Supports beam axis shall preferably be normal to the direction of liquid flow on the tray.

(2) Beam depth shall not exceed 20% of vertical distance between trays.

(3) Tray areas blanked by supporting members shall not be more than 75 mm for minor beams and 200 mm for major beams.

(4) A minimum overlap of 25 mm shall be provided between outside diameter of trays, pans, etc., and support rings.

(5) Holding down bolts shall be of 10 mm diameter.

(6) Bolt spacing around trays shall not exceed 100 mm and that on the downcomers shall not exceed 75 mm (Fig. 11.26).

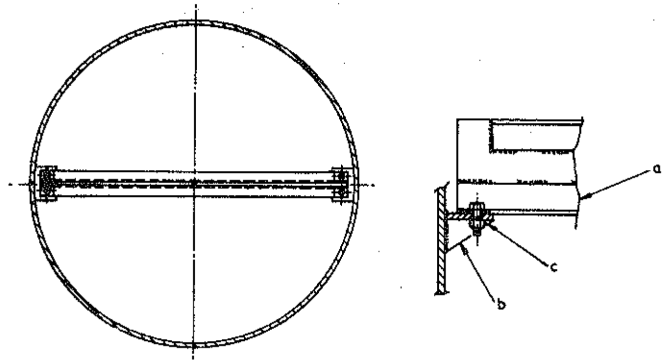


Fig. 11.26 Major beam support
(a) beam (b) foot rest (c) bolt

(*c*) *Vertical supports*

The load on the vertical supports is the total weight of all the trays including the liquid and the total impact load. The load is shared by a large number of vertical bars. These bars are supported by the bottom head. A suitable arrangement of brackets, welded to the bottom head acts as a supporting structure (Fig. 11.27). By providing sufficient number of

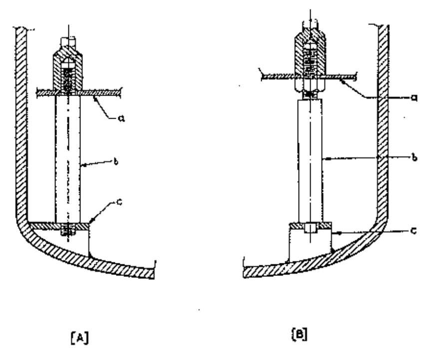


Fig. 11.27 Vertical supports

(A) fixed support (a) tray (b) support rod (c) bracket
(B) adjustable Support

vertical supports, it is possible to limit deflection of the tray to the desired range.

Support bars of carbon steel are either round or hexagonal. The recommended sizes are given in Table 11.3. Sizes for hexagonal bars are to be taken across the flats.

Table 11.3

Required Minimum Diameter (mm)

Length	Load in kg per support					
	400	600	800	1200	1600	2000
30						
37.5						
45	22	22	22	22	22	22
60						
75						
90						
105	25	25	25	25	28	30
120	28	28	28	28	32	35

11.8.4 Support Plates in Packed Columns

The loads to be carried by the support plates at the bottom of the packed column are

(1) weight of the packing

(2) weight of the liquid

(3) force due to pressure surges

(4) weight of intermediate supports or redistributors

If there are no intermediate supports, the entire load will be taken by the bottom support. This is not desirable. In general the packing height per support plate should not exceed 8 to 10 metres depending on the type of packing. The support plate may be designed as a tray resting on a peripheral ring (equations 11.27, 11.28), since the diameters of packed columns are usually small.

Support system for the whole column is indicated in Chapter 13. (see 13.4).

11.9 Numerical Problem

Fractionating column data

(a) *Shell*

Diameter	3000 mm
Working pressure	16 kg/cm^2
Design pressure	17.6 kg/cm^2
Working temperature	180°C
Design temperature	200°C
Base chamber height	2.74 m
Top chamber height	1.05 m
Feed chamber height	0.688 m
Material—carbon steel (sp. gr. 7.7)	
Permissible tensile stress	950 kg/cm^2
Insulation thickness	100 mm
Density of insulation	770 kg/m^3

(b) *Heads*—elliptical, welded to shell (ratio of major to minor axis—2)

Material—carbon steel

Permissible tensile stress 950 kg/cm^2

(c) *Support*—skirt

Height 4.9 m

Material—carbon steel

(d) *Nozzles*

	Number	Size in mm	Location
Feed	3	100	Plates 34, 38 and 42
Vapour outlet	1	450	top head
Reflux	1	200	plate 50
Liquid side-draw	1	75	base chamber
Vapour inlet	1	500	base chamber
Level control	2	50	base chamber
Gauge glass	2	75	base chamber
Thermometer	22	25	

Manhole	5	450	plates 10, 20, 30, 42, and 50
Liquid sample	4	25	plates 7, 13, 23 and 41
Reboiler feed	1	450	bottom head
Pressure gauge	1	25	base chamber

(e) *Trays*—sieve type

Number	50
Spacing	0.686 m
Holes (diameter)	5 mm
Number of holes	21100 (tray nos 1 to 7) 24850 (tray nos 8 to 34) 29,400 (tray nos 34 to 50)
Thickness	2 mm

Downcomer

Centre—rectangular	size 30×262 cm
Side—chord type	size 30×170 cm
Clearance from tray surface	50 mm

Weir

Height above tray 25 mm

Effective length

(i) Centre to side	—distributing	262 cm
	overflow	170 cm
(ii) Side to centre	—distributing	170 cm
	overflow	262 cm

Material for trays, downcomers and weirs—stainless steel.

(f) *Supports for tray*

Purlins—channels and angles
Live load—(liquid+liquid downcomer impact)—210 kg/m²
Material—carbon steel
Permissible tensile stress—1275 kg/cm²

(g) *Weight of attachments* —(pipes, ladder, platform, etc.) —140 kg/m

Weight of liquid and tray, etc.	—92 kg/m²
Weight of column (approx.)	—200,000 kg
Wind pressure	—130 kg/m²

11.9.1 SHELL—(MINIMUM THICKNESS)

From equation 11.5, minimum thickness at the top of shell is given by (corrosion allowance—2 mm)

$$t_{sh}=\frac{17.6\times 3000}{2\times 950\times 0.85-17.6}+C \qquad J=85\%$$

$$=33.0+2$$

$$=35 \text{ mm}$$

11.9.2 HEAD

Elliptical head with ratio of major to minor axis of 2, will have the same thickness as the shell, i.e., 35 mm.

Weight of the head—2670 kg (approx.)

11.9.3 SHELL THICKNESS AT DIFFERENT HEIGHTS

At a distance (X) metres from the top of the shell the stresses are, from equation 11.6

(i) Axial stress $f_{ap}=\dfrac{17.6\times 300}{4(35-2)}=400 \text{ kg/cm}^2$ (tensile)

From equation 11.7

(ii) $f_{as}=\dfrac{\pi/4(3070^2-3000^2)7.7(X)}{\pi/4(3070^2-3000)^2\times 10}=0.77(X) \text{ kg/cm}^2$

From equation 11.8

(iii) $f_d=\dfrac{0.77(X)(100)}{(35-2)\times 10}=0.233(X) \text{ kg/cm}^2$

From equation 11.9

$$f_{d(liq)}=\frac{\Sigma \text{ liquid and tray weight per unit height } (X)}{\pi\, D_m(t_{sh}-c)}$$

Calculation of liquid and tray weight for 'X' metre height is as follows :

The top chamber of height 1.05 metres does not contain any liquid or tray. The tray spacing is 67.5 cm. Liquid and tray weight for 'X' metre height.

Liquid and tray weight for 'X' metre height

$$F_{liq}=\left[\left(\frac{X-1.05}{0.675}\right)+1\right]\frac{\pi D_i^2}{4}\times 92 \text{ kg}$$

(iv) $f_{d(liq)}=\dfrac{F_{liq}\times 10}{\pi (300)(35-2)}$

$$=[0.309X-0.115] \text{ kg/cm}^2$$

(v) From equation 11.10

$$f_{d(att)}=\frac{\Sigma \text{ Weight of attachments per unit height } (X)}{\pi D_m (t_5-c)}$$

Total weight upto 'X' metre height

=weight of top head+pipes+ladder etc.

=[2670+140(X)] kg

$$\therefore \quad f_{d(att)}=\frac{[2670+140(X)]10}{\pi\times 300\times(35-2)}=[0.86+0.045\,(X)\text{ kg/cm}^2]$$

Total compressive dead weight stress at a height (X), from equation 11.11 [addition of (ii) to (v)]

(vi) $f_{dx}=0.78\,(X)+0.233\,(X)+0.309\,(X)-0.115+0.86+0.045\,(X)$

$$=1.357\,(X)+0.745$$

From equation 11.3, bending moment due to wind load

$$M_{wx}=0.7\times 130\times 3.2\frac{(X)^2}{2} \text{ kgm}$$

(vii) Stress due to wind load

$$f_{wx}=\frac{1.4\times 130\,(X)^2\times 10}{\pi\times 30.7\times(35-2)}$$

$$=0.0188\,(X)^2 \text{ kg/cm}^2$$

Tensile or compressive stress due to seismic load

From equation 11.17, bending moment due to seismic load ($C-0.08$)

$$M_{sx}=\frac{0.08\times 200{,}000}{3}(X^2)\left[\frac{3\times 44-(X)}{44^2}\right]\text{kgm}$$

(viii) $f_{sx}=\dfrac{M_{sx}\times 10^3}{\pi/4\times(307)^2\,(35-2)}$

$$=[1.4(X^2)-0\,009(X)] \text{ kg/cm}^2$$

To determine the value of (X), combine the stresses on the upward side as per equation 11.1

$$0.0188(X)^2+400-[1.357(X)+0.745]=f_{t\,(max)}$$

From equation 11.19

$$0.0188(X^2)-[1.357(X)+0.745]+[400-(950\times 0.8)]=0$$

$$X^2-72.3\,X-21650=0$$

$$X=186 \text{ metres}$$

The thickness will be the same for the entire height of the column.

On the downward side of the column, the compressive stress as per equation 11.3

$$f_{c(max)}=0.0188(X^2)-400+(1.357(X)+0.745)$$

The column height is 39 m, for which the maximum value of

$$f_c=0.0188(39)^2-400+(1.357(39)+0.745)$$

$$=-317.745 \text{ kg/cm}^2$$

which shows that the stress on the downward side is also tensile and not compressive. This means that the shell is under tensile stress, under all conditions. If the column is considered without internal pressure operating, then the compressive stress shall be

$$f_c=0.0188\,(39^2)+[1.357(39)+0.745]$$

$$=82.345 \text{ kg/cm}^2$$

which is well within the permissible stress for elastic stability (equation 11.22).

11.9.4 Nozzles

The column is provided with several nozzles. All nozzles above 5 cm diameter have to be checked for reinforcement.

Calculations of reinforcements are made as per area for area method [see 6.8.3.2 (a)]. Similarly all nozzles above 5 cm diameter are to be provided with standard pipe flanges.

11.9.5 TRAYS

The column of this type, which is used in the petroleum industry is made up of a shell, fabricated from sheets by welding. It has no flanged joints. This naturally requires a special design of trays which can be easily dismantled and removed. Such trays are the sectional trays. These trays are made of stainless steel, with a thickness of 2 mm and with holes of 5 mm diameter. Downtakes and weirs are bolted in the appropriate positions. The downcomers on the even number of trays are located at the ends. The shape of the downcomer is segmental with size 30 cm $\times$ 170 cm at the top and 22.5 cm $\times$ 170 cm at the lower end. The downcomers on the odd number trays are at the centre. The shape of the downcomer is rectangular with size 30 cm $\times$ 262 cm at the top and 20 cm $\times$ 262 cm at the lower end.

11.9.6 SUPPORT FOR TRAYS

The supporting system consists of six purlins symmetrically placed underneath each tray. The ends of purlins are attached to two sectors (support rings) which are fixed to the column wall. The length of each purlin and the distance between purlins is as shown in Fig. 11.28 (AA, BB, CC). The load carried by each purlin is in proportion to the area of the tray supported by the purlin.

Purlin AA

Loads

Live load (liquid+impact)	$210 \times \frac{298}{100} \times \frac{38.5}{100} = 241$ kg
Proportional tray weight	16 kg (Approx.)
Downcomer weight	25 kg (Approx.)
Purlin weight	50 kg (estimated)
Total load	332 kg

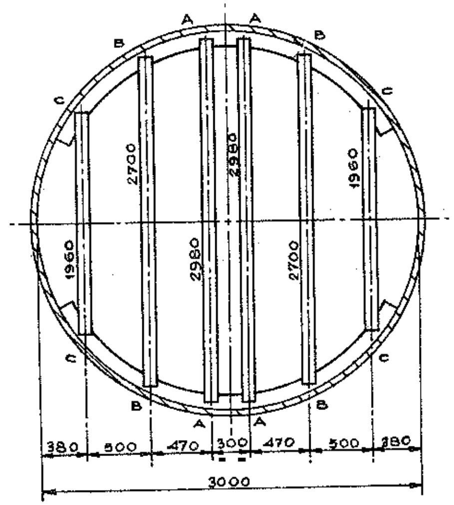


Fig. 11.28 Truss support purlins

If the deflection is to be limited to 3 mm under working conditions then as per equation 11.29

$$\frac{3}{10} = \frac{5}{384} \times \frac{332 \times 298}{2 \times 10^6 \times I}$$

$$I = \frac{5}{384} \times \frac{332 \times 298^3}{2 \times 10^6 \times 0.3} = 191.0 \text{ cm}^4$$

During assembly and cleaning operations the loads are (i) concentrated load of 150 kg (ii) uniformly distributed load of (16+25+50)=91 kg.

From equation 11.31 for concentrated load

$$1275 = \frac{150 \times 298}{4 \times Z}$$

$$Z_1 = \frac{150 \times 298}{4 \times 1275} = 8.75 \text{ cm}^3$$

From equation 11.30 for uniformly distributed load.

$$1275 = \frac{91 \times 298}{8 \times Z_2}$$

$$Z_2=\frac{91\times 298}{8\times 1275}=2.78 \text{ cm}^3$$

$$\text{Total } Z=8.75+2.78=11.53 \text{ cm}^3$$

A suitable channel section has to be chosen to satisfy the value of moment of inertia I and the modulus of section, Z.

Choose channel $120\times 60\times 4$ if the material is stainless steel, and $120\times 60\times 5$ if the material is carbon steel.

Purlin BB

Loads

Live load (liquid only—no impact load) $=60\times\frac{270}{100}\times\frac{58}{100}$

	$=94$ kg
Proportional tray weight	$=22$ kg
Purlin weight	$=38$ kg
Total	154 kg

If the deflection is to be limited to 3 mm under operating conditions, then as per equation 11.29

$$\frac{3}{10}=\frac{5}{384}\times\frac{154\times 270}{2\times 10^6\times I}$$

$$I=\frac{5}{384}\times\frac{154\times(270)^3}{2\times 10^6\times\frac{3}{10}}$$

$$=65.6 \text{ cm}^4$$

During assembly and cleaning operations the loads are (i) concentrated load of 150 kg (ii) uniformly distributed load of $(22+38=60$ kg).

From equation 11.31 for concentrated load

$$1275=\frac{150\times 270}{4\times Z_1}$$

$$Z_1=\frac{150\times 270}{4\times 1275}=7.95 \text{ cm}^3$$

From equation 11.30 for uniformly distributed load

$$1275=\frac{60\times 270}{8\times Z_2}$$

$$Z_2=\frac{60\times 270}{8\times 1275}=1.585 \text{ cm}^3$$

$$\text{Total } Z=7.95+1.585=9.535 \text{ cm}^3$$

Choose a channel section $100\times 50\times 3$ if the material is stainless steel, otherwise $100\times 50\times 4$ if the material used is carbon steel.

Purlin CC

Loads

Live load—(liquid+impact load) $=210\times\frac{196}{100}\times\frac{44}{100}$

	$=181$ kg
Proportional tray weight	$=12$ kg (Approx.)
Downcomer weight	$=15$ kg (Approx.)
Purlin weight	$=20$ kg (estimated)
Total weight	$=228$ kg

If the deflection is to be limited to 3 mm under operating conditions, then as per equation 11.29

$$\frac{3}{10}=\frac{5}{384}\times\frac{228\times 196^3}{2\times 10^6\times I}$$

$$I=\frac{5}{384}\times\frac{228\times 196^3}{2\times 10^6\times\frac{3}{10}}$$

During cleaning and assembly operations the loads are (i) concentrated load of 150 kg (ii) uniformly distributed load of $(12+15+20=47$ kg).

From equation 11.31 for concentrated load

$$1275=\frac{150\times 196}{4\times Z_1}$$

$$Z_1=5.77 \text{ cm}^3$$

From equation 11.30 for uniformly distributed load

$$1275=\frac{47\times 196}{8\times Z_2}$$

$$Z_2=\frac{47\times 196}{8\times 1275}$$

$$=0.904 \text{ cm}^2$$

$$Z=5.77\times 0.904=6.674 \text{ cm}^3$$

Choose a channel of the same size as for purlin BB.

Reading References

Van Winkle, M., *Distillation*, McGraw-Hill, New York, (1967).

Ludwig, E.E., *Applied Process Design for Chemical and Petrochemical Plants*, Vol. II, Gulf, Houston, (1964).

Taganov, N.I., *Process Equipment Design*, Academic Books, Bombay, (1968).

Oliver, E.D., *Diffusional Separation Processes*, John Wiley, New York, (1966).

Robin, B.J., *Mechanical Problems in Bubble and Grid Tray Design*, Brit. Chem. Eng., **3**, 480 (1958).

Robin, B.J., *The Ballast Tray*, Brit. Chem. Eng., **4**, 351(1959)

Proctor, J.F., *Sieve and Bubble Cap Plates*, Chem. Eng. Prog., **59**(3), 47(1963).

Atkins, et al., *How to Specify Trays*, Petrol. Refiner, **59**(8), 86 (1960).

Tang, S.S., *Shortcut Methods of Calculating Tower Deflections*, Hydro. Process, **47** (11), 230, (1968).

Fair, J.R., *Selecting Fractionation Column Internals*, Chem. Eng., **72** (14), 107 (1965).

Jamison, R.H., *Distillation Internal Design Techniques*, Chem. Eng. Prog., **65**(3), 46 (1969).

Arnis, V.R.S., *New Breakthroughs in Distillation Tray Design*, Chem. Age of India, **21**(10), 877 (1970).

Interess, E., *Practical Limitations on Tray Design*, Chem. Eng., **75**(26), 167 (1971).

Perry, J.H., *Chemical, Engineers' Handbook*, 5th Ed., McGraw-Hill, New York, (1973).

Clay, H.A. et al., *Which Packing for Which Job*, Chem. Eng. Prog., **92**(1), 51 (1966).

Eckert, J.S., *A New Look at Distillation, Tower Packings—Comparative Performance*, Chem. Eng. Prog., **59**(7), 76 (1963).

McClain, R.W., *How to Specify Bubble Cap Trays*, Petrol. Refiner, **39**(8), 92 (1960).

Patton, B.A. and Pritchard B.L. (Jr.), *How to Specify Sieve Trays*, Petrol. Refiner, **39**(8), 95 (1960).

Thrizt, G.C., *How to Specify Valve Trays*, Petrol. Refiner, **39**(8), 93(1960).

Leva, M., *Tower Packings and Packed Tower Design*, 2nd Ed., U.S. Stoneware Co., Akron, Ohio, (1953).

Smith, B.D., *Design of Equilibrium Stage Process*, McGraw-Hill, New York, (1963).

A.I.Chem.E., *Equipment Testing Procedure Plate Distillation Column*, (1962).

Jordon, D.G., *Chemical Process Development*, Parts I and II. Interscience New York, (1968).

CHAPTER 12

High Pressure Vessels

12.1 Introduction

These vessels are generally used as reactors, separators or receivers and heat exchangers. Most batch reactors used for high pressure work are termed as autoclaves. They are vessels with an integral bottom and a removable top head, and are generally provided with an inlet, heating and cooling system and very often an agitator system. Separators for high pressure work are also vessels with an inlet and in some cases a centrifugal device for separating the gas from the liquid. High pressure heat exchangers are of concentric tube type.

12.2 Constructional Features

High pressure vessels are used for a pressure range of 200 atmospheres to a maximum of 3000 atmospheres. These are essentially thick walled cylindrical vessels, ranging in size from small tubes to several metres diameters. Both the size of the vessel and the pressure involved will dictate the type of construction used. Following are the types of constructions.

(A) A solid walled vessel produced by forging or boring a solid rod of metal (Fig. 12.1).

(B) A cylinder formed by bending a sheet of metal with a longitudinal weld.

(C) A vessel built up by wrapping a series of sheets of relatively thin metal tightly round one another over a core tube, and holding each sheet with a longitudinal weld. Rings are inserted in the ends to hold the inner shell round while subsequent layers are added. The inner cylinder is 6 to 12 mm

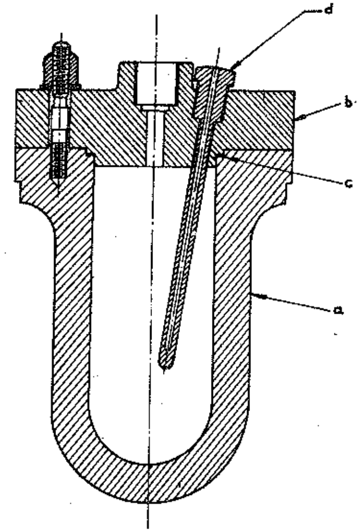


Fig. 12.1 Solid walled vessel
(a) shell (b) cover (c) gasket (d) thermometer pocket

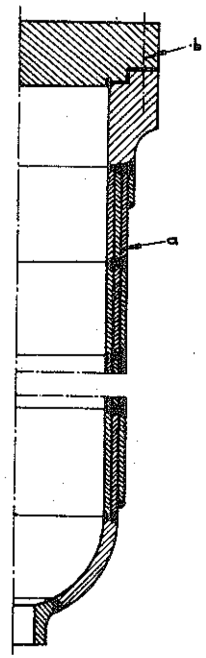


Fig. 12.2 Vessel formed by wrapping of sheets
(a) shell (b) cover

thick, while the subsequent layers are generally less than 6 mm thick (Fig. 12.2).

(D) Shrink-fit construction (Fig. 12.3): Consists of concentric cylindrical shells, fitted together by shrinking one cylinder over the other. The method involves heating the outer shell in order to expand it beyond the interference on the diameter, and then slip it on the inner shell while expanded. In practice it is often desirable to make the innermost component of some special material, for instance, a corrosion-resistant alloy.

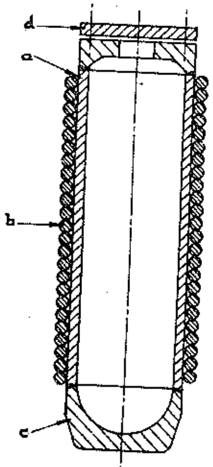


Fig. 12.3 Vessel with shrink-fit construction
(a) shell (b) outer shell (c) head (d) cover

(E) A vessel built up by wire winding around a central cylinder. The wire is wound under tension around a cylinder of about 6 to 10 mm thick (Fig. 12.4).

(F) A vessel built up by wrapping successive layers of interlocking tape around a central cylinder. Spiral grooves are cut on the surface of a central cylinder, which is then wrapped with a tape. The tape is also grooved, so that it will interlock with succeeding layers of tape. The succeeding layers are offset from the preceding ones to get a proper interlock and produce axial strength (Fig. 12.5).

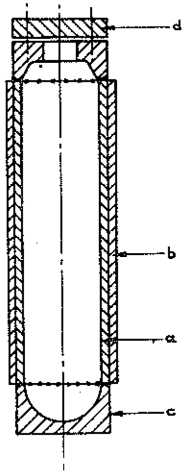


Fig. 12.4 Wire wound vessel
(a) shell (b) wire (c) head (d) cover

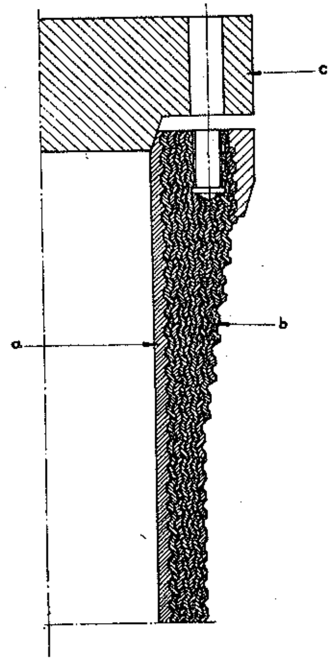


Fig. 12.5 Tape wound vessel
(a) shell (b) tape (c) cover

Table 12.1

High Tensile Steels for Heavy-Wall Pressure Vessels

In Quenched and Tempered Condition

Standard	C	N	Cr	Ni	Mo	V	YP	UTS kgf/mm²	Elongation %
A 10	0.10...0.15	0.50	11.5...13.5	0.50			55	60...75	18
43 OH	0.37...0.45	0.6...0.95	0.65...0.95	1.5...2.0	0.2...0.3		75	80...90	14
En 15	0.27...0.35	0.5...0.7	0.5...0.8	2.3...2.8	0.4...0.7		75	80...100	14
En 26	0.35...0.44	0.5...0.7	0.5...0.8	2.3...2.8	0.4...0.7		85	90...100	11
En 29	0.25...0.35	0.65	2.5...3.5	0.4	0.3...0.7		90	100...115	11
Cr MoV 135	0.17...0.23	0.3...0.5	3.0...3.3		0.5...0.6	0.45...0.55	75	80...95	14
30 Cr MoV 9	0.26...0.34	0.4...0.7	2.3...2.7		0.15...2.25	0.10...0.20	105	125...145	9

(G) Autofrettage : Involves the use of a solid forged vessel, to which is applied a large internal pressure, so as to expand the internal diameter, causing overstrain. When the pressure is released, the outer material will try to contract the inner material. As a result residual compressive stresses are developed in the inner material and tensile stresses in the outer material.

12.3 Materials for High Pressure Vessels

Ductility is an essential requirement in steel for use as a pressure vessel and should guide the choice of steel and its heat treatment.

(a) *Ni-Cr-Mo steels* : These are the mostcommon steels used for high pressure vessels. Heat treatment is generally oil quenching from 850°C with subsequent tempering between 600°C to 650°C.

(b) *Maraging steels* : These steels have been developed recently and their main asset is that they develop their strength as a result of age hardening which is initiated by relatively low temperature heat treatment. A typical steel of this type has a composition Ni 18.5%, Mo 4.9%, Co 8.7%, Ti 0.7%, Al 0.1%. These steels can be machine finished to final dimensions in the solution—annealed condition before heat treatment due to negligible changes in length and the low temperatures involved.

(c) *Creep resistance steels* : Special creep resistance steels which have the property of retaining their strength at these temperatures are used above 300°C.

(d) *Non-ferrous alloys* : For temperatures above 500°C non-ferrous alloys such as Nimonic or Hastelloy are required, the latter giving superior creep resistance. These are essentially Ni–Cr–Co alloys. Creep however cannot be wholly avoided.

12.4 Solid Walled Vessel

This type of vessel consists of a single cylindrical shell, with closed ends. Due to high internal pressure and large thickness the shell is considered as a 'thick' cylinder. In general, the physical criterion is governed by the ratio of diameter to wall thickness and the shell is designed as thick cylinder, if its wall thickness exceeds one-tenth of the inside diameter.

12.4.1 Stresses in Shell

The main loading in the shell is due to internal or external pressure. The stresses across the shell thickness due to pressure are not uniform. Stresses due to other loads, such as dead loads are comparatively smaller in magnitude and may therefore be ignored.

Three principal stresses are produced in the wall of the shell due to pressure. These stresses are (a) tangential (circumferential or hoop), (b) radial and (c) longitudinal (axial). The tangential stresses are of high magnitude. According to Lame's analysis (assuming a fully elastic cylinder) the variation of tangential and radial stresses along the radius of the shell is given by (Fig. 3.9 and equations 3.27, 3.28, 3.29).

Tangential stress $$f_p = A + \frac{B}{R^2} \tag{12.1}$$

Radial stress $$f_R = A - \frac{B}{R^2} \tag{12.2}$$

Longitudinal stress $$f_a = A \tag{12.3}$$

Where R—any radius ; A, B—constants.

The tangential stress may be either tensile or compressive depending on the relative magnitude of internal and external pressure. The radial stress is always compressive and the axial stress is tensile. It is assumed that the axial force due to pressure is wholly carried by the wall.

If at the inner radius R_1, the pressure is p_i and the outer radius R_2, the pressure is p_o, the constants A and B can be determined as

$$A = \frac{p_i R_1^2 - p_o R_2^2}{R_2 - R_1} = \frac{p_i - p_o K^2}{K^2 - 1} \tag{12.4}$$

Where $$K = \frac{R_2}{R_1}$$

$$B = \frac{(p_o - p_i)\, R_1^2\, R_2^2}{(R_2^2 - R_1^2)} = \frac{(p_o - p_i)\, K^2 R_1^2}{K^2 - 1} \qquad (12.5)$$

These relations show some important features. (1) The tangential and radial stresses consist of a pure shear stress $\left(q = -\frac{B}{R^2}\right)$ with a superimposed hydrostatic stress (A). (2) The maximum values of the stresses occur at the inner radius. (3) These maximum values depend only on the ratio of the radii and not on their absolute magnitude. In many cases, only the internal pressure p_i is present and the above equations therefore are simplified. The stresses at the inner surface are (at $R=R_1$) (Fig. 12.6).

$$f_p = A + \frac{B}{R^2} = \frac{p_i\, R_1^2}{R_2^2 - R_1^2}\left(1 + \frac{R_2^2}{R_1^2}\right)$$

$$= p_i \left(\frac{K^2+1}{K^2-1}\right) - 1 \qquad (12.6)$$

$$f_R = A - \frac{B}{R^2} = \frac{p_i\, R_1^2}{R_2^2 - R_1^2}\left(\frac{R_2^2}{R_1^2} - 1\right) = p_i \qquad (12.7)$$

$$f_a = A = p_i \frac{1}{K^2 - 1} \qquad (12.8)$$

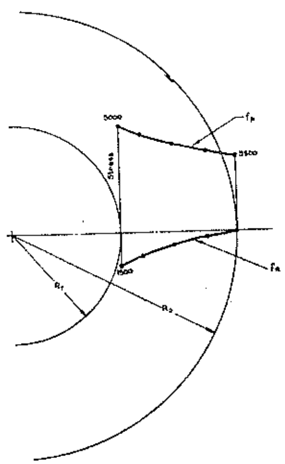


Fig. 12.6 Stresses in a solid walled shell

12.4.2 Application to Shell Design

12.4.2.1 Design based on tangential stress

The thickness of the cylinder can be determined on the basis of the maximum tangential stress at inner surface of the cylinder. Equation 12.6 can be written in terms of thickness, $[t = (R_2 - R_1)]$.

$$f_p\,(\text{Max}) = \frac{p_i}{t}\left(\frac{R_1^2 + R_2^2}{R_2^2 - R_1^2}\right) \qquad (12.9)$$

If f_p is taken as the allowable tensile stress f

$$\text{then } t = R_1\left(\sqrt{\frac{fJ + p_i}{fJ - p_i}} - 1\right) \qquad (12.10)$$

Where J—Joint efficiency

Corrosion allowance is neglected if the thickness is more than 30 mm.

12.4.2.2. Design based on shear stress

Another approach to design is based on the maximum shear stress produced at any point on the wall of the cylinder. This is given by

$$f_{s(max)} = \frac{1}{2}\,(f_p - f_R) = p_i\left[\frac{k^2}{k^2 - 1}\right] \qquad (12.11)$$

The limit of elastic action occurs when

f_s (at yield point) $= \frac{1}{3} f$ (direct tensile stress at yield point).

If the cylindrical shell is to be designed within the limits of elastic action, the maximum internal pressure at the inner radius is given by

$$p_{max} = \left[\frac{1}{\sqrt{3}}\, fr.\, p.\, \frac{(k^2-1)}{k^2}\right] \qquad (12.12)$$

For purpose of design a factor of safety should be used to get the allowable stress. The value of k can then be determined as

$$k = \sqrt{\frac{f}{f - p_i\sqrt{3}}} \qquad (12.13)$$

(where f is the allowable stress) from which the thickness t can be obtained.

12.4.2.3 DESIGN BASED ON MAXIMUM ENERGY OF DISTORSION

A third criterion of design is based on the maximum energy distortion theory, also known as Von Mises or Hencky criterion. The limit of elastic action is reached according to this criterion when the relation between f_p, f_r and f_a is (see equation 4.7)

$$(f_p-f_r)^2+(f_R-f_a)^2+(f_a-f_p)^2=2\,(fr.\,p.)^2 \qquad (12.14)$$

where $fr.\ p.$ is the yield point of the material in simple tension. The three stresses can be written for the internal pressure alone. The value of f_p is given by equation (12.9). Similar values for f_R and f_a are

$$f_R=-\frac{p_i}{t}\,(R_2-R_1) \qquad (12.15)$$

$$f_a=\frac{p_i}{t}\left(\frac{R_1^2}{R_1+R_2}\right) \qquad (12.16)$$

Combining the stresses as per equation (12.14) and substituting $f_r\ p$ by the allowable stress f, the permissible internal pressure is given by

$$p_i=2.31\,f\left(\frac{t}{2R_2}\right)\left(1-\frac{t}{2R_2}\right) \qquad (12.17)$$

Alternately, if the internal pressure, is known, the thickness t of the cylinder can be determined.

The maximum shear stress criterion of design and the maximum energy of distortion theory mentioned above are based on the allowable yield stress of the material. The thickness of the vessel can be reduced if the yield stress of the material can be increased. This is made possible by use of alloy steels and by heat treatment. The alloying elements are chromium, nickel, molybdenum and vanadium. The three criteria for design of the thickness generally give widely differing values of the required thickness (see problem 12.6). Experimental results satisfy one criterion for one material and another criterion for another material. When working with a new material, it is important to make a preliminary test with a vessel constructed of the material. The experimental vessel may be hydrostatically tested until a permanent set is obtained. The pressure measured at the beginning of the permanent set can be compared with that predicted by the various criteria and the most appropriate equation. Steels with high tensile strength would be expected to fall by shear. A few typical compositions and their strength properties are indicated in Table 12.1.

12.4.2.4 STRAINS IN SHELL

In many designs of 'thick walled' vessels, a knowledge of the magnitude of the change in dimensions of the vessel is pertinent, where a more critical analysis is desired. With internal pressure p_i, the change in inside radius ΔR_1 and outside radius ΔR_2 is as follows.

$$\Delta R_1=p_i\,\frac{R_1}{E}\left[\frac{R_2^2+R_1^2}{R_2^2-R_1^2}-\mu\left(\frac{R_1^2}{R_2^2-R_1^2}-1\right)\right] \qquad (12.18)$$

$$\Delta R_2=p_i\,\frac{R_2}{E}\left[\frac{R_2^2}{R_2^2-R_1^2}\,(2-\mu)\right] \qquad (12.19)$$

where μ is Poisson's ratio, and E, the modulus of elasticity.

12.4.2.5 VESSELS WITH THERMAL STRESSES

If the temperature of the vessel wall is not uniform, thermal stresses must be taken into account in determining the safe internal pressure or thickness as desired. The nature of the thermal stresses created in the wall has been stated previously (equations 6.49 and 6.50). Two cases may be considered, (a) in which the vessel is subjected to internal pressure and heated internally and (b) the vessel is subjected to internal pressure, but heated externally. In the former case the design is based on the larger of the two stresses namely, stress due to internal pressure or thermal stress. In the latter case the design is based on the combined stress due to pressure and thermal gradients.

12.4.3 Shells Formed by Bending a Sheet of Metal with a Single Longitudinal Weld

This type of vessel can be designed on the same basis as the solid walled vessel except that the joint efficiency must be taken into consideration in all calculations of thickness determination. The permissible stress may be multiplied by the joint efficiency.

12.4.4 Multi-shell and Prestressing

So far the design of a single shell formed out of a solid block of material or shell formed from a sheet by welding was considered. As the internal pressure in the shell increases, the required shell thickness also increases. With higher pressures the problems of economic use of material as well as those of fabrication become critical. The remaining methods of construction involve use of additional shells or layers of materials or autofrettage. In all these methods of construction, a better utilisation of the material is attempted by setting up an initial stress in the shell, known as prestressing.

As may be seen from Fig. 12.6 in a solid walled vessel, the internal pressure causes a tangential stress, which is maximum on the inside of the cylinder. The thickness of the wall is determined by the allowable maximum stress. To utilise the material more economically it would be desirable to create a uniform stress in the entire thickness of the vessel wall. It is possible to satisfy this condition, by prestressing. The extent of prestressing required to create uniform stress is assessed as

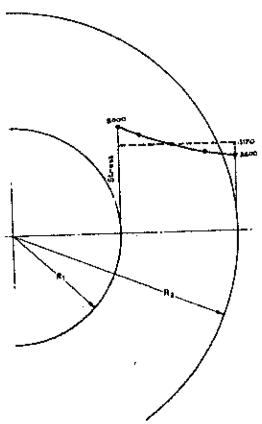


Fig. 12.7 Prestressed shell

follows. The average stress in the vessel wall can be calculated as

$$f_{av}=\frac{p_i R_1}{t}=p_i\left(\frac{1}{k-1}\right) \tag{12.20}$$

If the tangential stresses due to internal pressure are f_i and f_o at the inner and outer radius, then, the extent of prestressing would be (f_i-f_{av}) compressive stress in the part of the wall thickness and $(f_{av}-f_o)$ tensile stress in the remaining wall thickness (Fig. 12.7). The various methods of construction satisfy these conditions only partially. Ideal prestressing to create uniform stress is difficult to achieve.

12.5 Multi-Shell Construction

These vessels are built up by wrapping a series of sheets over a core tube. The construction involves the use of several layers of material, usually for the purpose of quality control and optimum properties. Each layer may be sufficiently thick, and is considered as a thick walled cylinder. Between the two cylinders in contact there is an interface pressure (p_f) which acts on the outer surface of the inner cylinder and on the inner surface of the outer cylinder. Thus with only two cylinders the inner cylinder is subjected to an internal pressure p_i and an external pressure p_f; the outer cylinder is subjected to an internal pressure p_f only. The value of interface pressure p_f is determined from the deformation at the interface of both cylinders.

12.5.1 Shrink Fit Construction

In this construction the vessel is built up of two or more concentric shells, each shell progressively shrunk on from the inside outward. As a result, the inner shells are subjected to an initial compressive circumferential stress so that when the internal pressure is applied, the resultant tensile circumferential stress on the shell layer does not exceed the allowable safe working stress. The magnitude of the compressive stress due to shrinkage is determined by the initial difference between the bore of the outer and the external diameter of the inner cylinder. The permissible extent of shrinkage involves a consideration of the maximum temperature to which the outer cylinder can conveniently be heated without impairing the properties of the material. From economic and fabrication considerations, the number of shells should be limited to two.

12.5.2 Stresses in Multi-shell or Shrink Fit Construction

In the general case of two shell construction the two components have different properties and are held together by a interface pressure p_f. The radial deformations of the two shells can be given as follows.

For the inner shell

$$\delta_i = \frac{p_f R_2}{E_i}\left(\frac{R_1^2+R_2}{R_2^2-R_1} - \mu_i\right) \tag{12.21}$$

For the outer shell

$$\delta_o = \frac{p_f R_2}{E_o}\left(\frac{R_2^2+R_3^2}{R_3^2-R_2} + \mu_o\right) \tag{12.22}$$

where R_1, R_2 and R_3 are the radii as shown in Fig. 12.8. E_i and E_o—modulli of elasticity of respective shells and μ_i and μ_o—Poisson's ratio for respective shells.

The total shrinkage is the absolute sum of the individual deformations.

$$\Delta = \delta_o + \delta_i$$

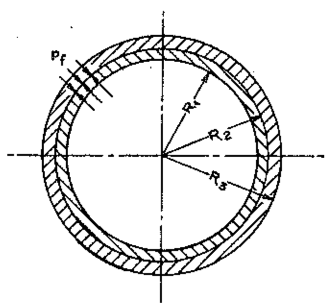


Fig. 12.8 Two shell construction

If the two shells are of the same material then $E_i = E_o$ and $\mu_i = \mu_o$, and the interface pressure p_f is given by

$$p_f = \frac{E\Delta}{R_2}\left[\frac{(R_2^2-R_1^2)(R_3^2-R_2^2)}{2R_2^2(R_3^2-R_1^2)}\right] \tag{12.24}$$

In shrink fit construction the radial deformation of the cylinder is obtained by heating the outer cylinder. The temperature required for shrinking operation is given as

$$t_f - t_i = \frac{R_f - R_i}{\alpha R_i} = \frac{\delta}{\alpha R_1} \tag{12.25}$$

where, R_f, R_i—final and initial radii, α=coefficient of thermal expansion and t_f, t_i=final and initial temperatures. In either method, it is necessary to calculate the interface pressure. If such a built up cylinder is now subjected to internal pressure, the stresses produced by this pressure are the same as those in a solid wall cylinder of thickness equal to the sum of those of the individual cylinders, with radii R_1 and R_3. These are superimposed on the stresses produced due to p_f. Thus the maximum tangential tensile stress is reduced, creating a more favourable stress distribution (Fig. 12.9). (See numerical problem).

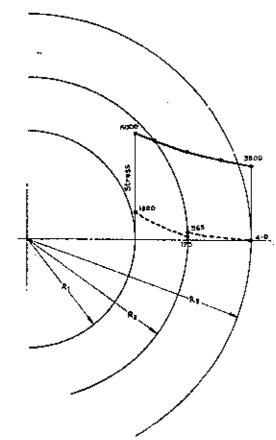


Fig. 12.9 Stresses in single and two shell construction

12.5.3 Optimum Design of Multi-shell Construction

For certain special applications where weight is critical and maximum utilisation of the material is desired it is necessary to design for optimum conditions. Thus if all the shells are of similar strength, the completed vessel is designed, so that when there is internal pressure and no external pressure, each component carries the same maximum shear stress, which is given by

$$f_{s\,max} = \frac{p_i\, k^{2/m}}{m(k^{2/m}-1)} \tag{12.26}$$

where

m — number of shells in the compound cylinder

k — ratio of radii

If the maximum shear stress is taken as equal to the shear yield strength, then the maximum internal pressure is given by

$$p_{max} = \frac{m f_{yp}\,(k^{2/m}-1)}{k^{2/m}} = \frac{m f_{yp}\,(k^{2/m}-1)}{\sqrt{3}\,k^{2/m}} \tag{12.27}$$

(compare equation 12.12)

The allowable stress value for f_{yp} will give the design value of k, if internal pressure is known. There is a limit to the maximum allowable pressure in a compound cylinder regardless of dimensions, which is given by

$$p = \frac{2 f_{yp}\,(k^2-1)}{\sqrt{3}\,k^2} \tag{12.28}$$

There is also a limit on the temperature difference which can be used for shrinkage operation. The optimum shrinkage per unit radial length is $\frac{2p_i\mu}{E}$.

In an actual shrinkage operation the outer cylinder must be heated so that the thermal expansion difference is equal to the shrinkage required plus the radial clearance needed for construction. The optimum value of the interface radius is given by

$$R_2 = \sqrt{R_1 R_3} \tag{12.29}$$

12.6 Vessel Closures

12.6.1 Permanent Closures

It is the usual practice to close one end of the cylindrical vessel permanently by a head which is forged or machined from a solid bar to form the shell and the head as the single piece. Alternately a closure in the form of a cup or dish may be welded to the shell. For vessels of small diameter, a simple flat disk may be welded to the shell. Closures welded to shell can withstand only moderate pressures.

12.6.2 Removable Closures

The type of the closure and the method of sealing and fixing the closure to the shell depend on a number of factors such as the size of vessel, the nature of its contents, the working temperature and pressure and the frequency with which the closure is removed. The joints made between the body of the vessel or shell and the closure are of different types depending on whether the joint can be made without a gasket or with the help of a gasket. Of the latter a gasket joint can be either of the compression type or of the unsupported type. Compression type gasket joints are discussed under 6.8.4.1 and are either of the unconfined type, such as a full face gasket and a ring gasket on raised face, or confined gaskets located on male and female or tongue and groove facings. Unsupported gasket joints, also known as self-sealing types, are especially applicable to high pressure vessel closures.

The plain joint which is made without a gasket, does not require excessively large clamping forces since the two components namely, the vessel and its cover make contact over a comparatively small area. Fig. 12.10 illustrates the single cone joint which is one of the most common. The maximum pressure the joint can withstand is determined by the initial compressive stress that can be developed across the sealing surfaces. With these joints the effect of over-tightening is to increase the area of contact between the two components and thus increase the difficulty of obtaining the required initial stress across the seating. Such joints are used for small

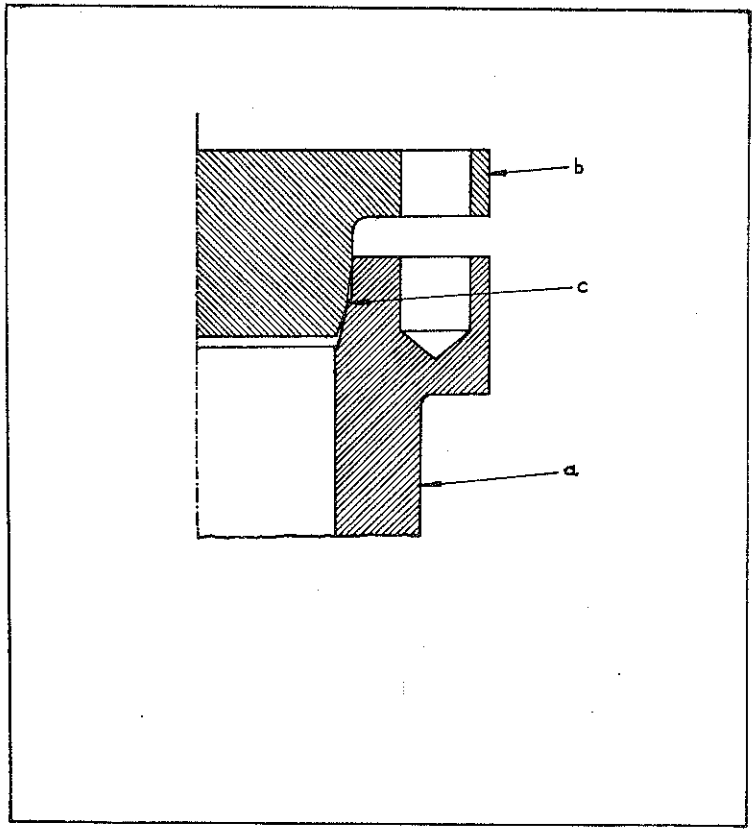


Fig. 12.10 Single cone joint
(a) shell (b) cover (c) cone

vessels operating at moderate pressure. Compressive type of gasket joints are made with gaskets which are comparatively soft and ductile. They are able to sustain an initial high compressive stress. The gasket diameter and width are kept as small as possible. For high pressure work the gasket is confined and supported and is made of copper, silver, iron or aluminium. Unsupported or self-sealing gaskets are so constructed that the stress between the adjoining faces is automatically maintained at a higher value than the pressure in the

vessel. Several designs of self-sealing type joints have been developed, the most widely adopted forms are indicated below.

12.6.2.1 DELTA-RING CLOSURES

These are used for vessels operating under wide range of pressures upto 1000 kg/cm^2 (Fig. 12.11). The ring has a triangular

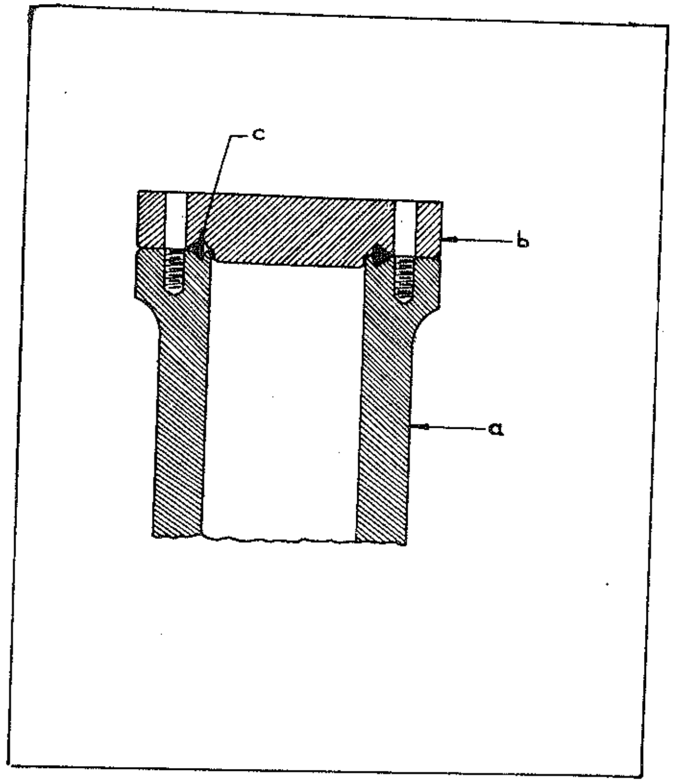


Fig. 12.11 Delta-ring closure
(a) shell (b) cover (c) delta-ring

cross-section with the longest side of the triangle facing the centre of the ring. V-shaped grooves are cut in the flange of the vessel and in the underside of the cover. These grooves match the delta-ring, so that when the cover is tightened down on the flange, it will form a seal. The delta-ring is made of steel, machined, polished and copper plated. The copper plating serves as a gasket. The ring is designed to function below its elastic limit under bolt operating and hydrostatic testing conditions.

Delta-ring closures are simple to fabricate and install. The closure may be opened and closed with minimum time. The disadvantage is that when the closure is very large, the delta-ring lacks rigidity during machining and dimensional control becomes difficult.

12.6.2.2 DOUBLE-CONE SEAL RING (LENS RING)

In this type of closure a ring of soft steel with a plain vertical face towards the inside, and two bevelled edges towards the outside are used (Fig. 12.12). The bevels extend nearly to the outside middle line of the ring and each bevel makes an angle of 30° with the axis of the vessel. The ring is made from an endless ribbon formed from an aluminium sheet about 1 mm. thick. The length of the ribbon is made equal to the inside circum-

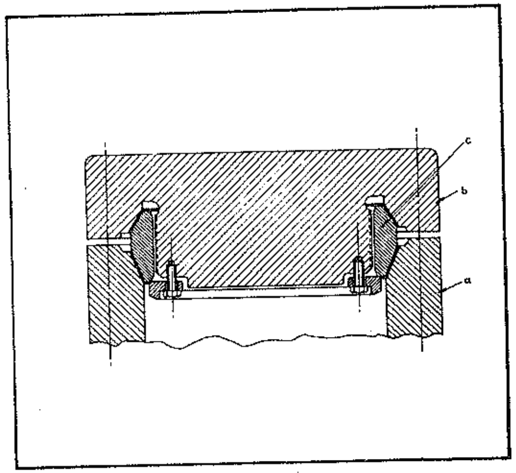


Fig. 12.12 Double-cone seal ring closure
(a) shell (b) cover (c) double-cone ring

ference of the double-cone ring and a width equal to 2½ times the height of the ring. This ribbon fits the inside circumference

of the ring, wraps around and covers the conical surfaces of the ring, thus forming a gasket. No groove is required in the flange of the shell. The studs can be placed on a smaller pitch circle, thus reducing the size of the closure. These seal rings are used upto 500 kg/cm² pressure. The above gaskets are unsupported in the radial direction.

12.6.2.3 THE BRIDGEMAN CLOSURE

This consists of a partly triangular gasket of a relatively soft metal like copper or aluminium backed by a steel seal-ring. These two rings are accurately machined to fill the space between the mouth of the vessel and a floating cover (Fig. 12.13).

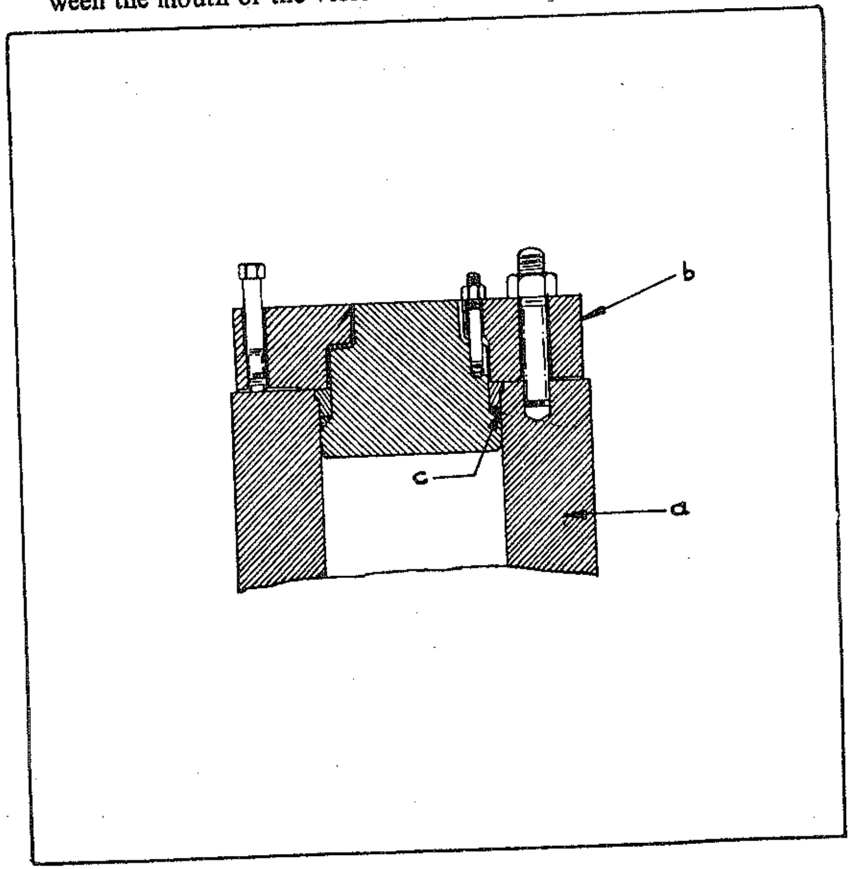


Fig. 12.13 Bridgeman closure
(a) shell (b) cap (c) seal-ring

A narrow ledge in the mouth of the vessel prevents the floating cover from going down too far. The floating cover is suspended by screws from a ring-shaped cap, which is fastened to the flanges of the vessel by studs. In making the closure, these suspension screws are tightened just enough to make the vessel gas-tight without internal pressure.

When the internal pressure on the vessel is applied, the axial pressure on the underside of floating cover pushes it against the gasket. A component of the pressure then pushes the gasket against the mouth of the vessel making a seal, the tightness of which is proportional to the internal pressure.

The chief advantages of this closure are (a) the gasket is totally confined, and is in no danger of being blown out even at very high pressure (b) the gasket is unsupported in the axial direction and hence the function of taking the end load and of prestressing the joint between the vessel and its cover can be separated and (c) the closure is simple to fabricate, easy to install and can be used for any diameter.

12.6.2.4 THE UNDE-BREDTSCHNEIDER CLOSURE

It consists of a floating cap with spherical shoulder and a seal-ring with a wedge-shaped cross-section (Fig. 12.14), which fits into the slightly tapered throat of the vessel. This feature facilitates removal of the seal-ring at any time after the vessel has been under testing or under normal operation. The seal-

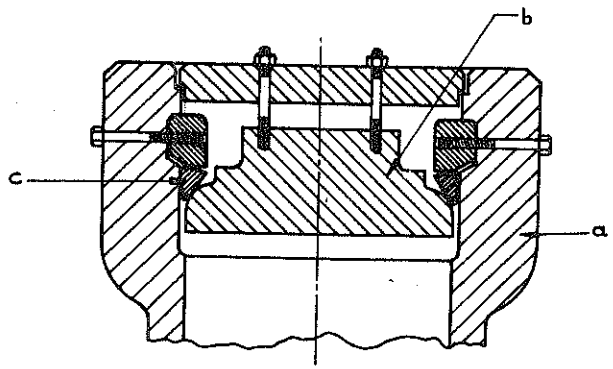


Fig. 12.14 Unde-Bredtschneider closure
(a) shell (b) floating cap (c) seal-ring

ring is designed to function within the elastic limit under operating and testing conditions, and is therefore made of steel.

12.7 Jackets for Vessels

Some high pressure vessels need heating and cooling devices. Fig. 12.15 shows two methods of providing jackets, one for a solid-wall vessel and the other for a strip-wound vessel. In both the cases a half coil is used, which (see 8.6.2) is welded to the wall or to the outer strip in strip-wound vessel. Water or steam with a pressure of upto 100 kg/cm^2 can be used in these coils

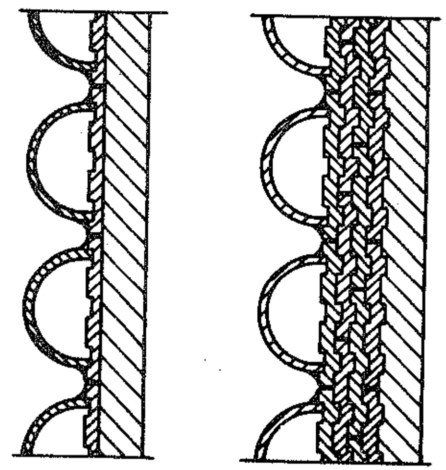


Fig. 12.15 Jackets

12.8. Numerical Problem

High Pressure vessel shell

Data

Internal diameter of shell	— 30 cm
Internal pressure	— 1500 kg/cm^2

External pressure	— atmospheric
Material	— High tensile steel (Ni-Cr-Mo)
Permissible tensile stress	— 5000 kg/cm^2
Modulus of elasticity	— 2×10^6 kg/cm^2
Coefficient of linear expansion	— 12.5×10^{-6} per °C

12.8.1 VESSEL SHELL THICKNESS

From equation 12.10

$$t=R_1\left[\left\{\frac{fJ+p_i}{fJ-p_i}\right\}^{\frac{1}{2}}-1\right] \quad J=1 \text{ (forged shell)}$$

$$=\frac{30}{2}\left[\left\{\frac{5000\times1+1500}{5000\times1-1500}\right\}^{\frac{1}{2}}-1\right]=5.4 \text{ cm}$$

$$R_2=R_1+5.4=15+5.4=20.4 \text{ cm}$$

$$k=\frac{R_2}{R_1}=\frac{20.4}{15}=1.36$$

12.8.2 VARIATION OF STRESS ALONG THE THICKNESS

From equation 12.4

$$A=\frac{1500\times(15)^2}{(20.4)^2-(15)^2}=1750$$

From equation 12.5

$$B=\frac{1500\times15^2\times20.4^2}{(20.4)^2-(15)^2}=7320\times10^2$$

From constants A and B, the stresses f_p and f_R can be calculated. Equations 12.1 and 12.2 give

at $R=15$
$$f_p=A+\frac{B}{R^2}=1750+\frac{7320\times10^2}{(15)^2}$$
$$=5000 \text{ kg/cm}^2 \text{ (tensile)}$$
$$f_R=A-\frac{B}{R^2}=1750-\frac{7320\times10^2}{(15)^2}$$
$$=-1500 \text{ kg/cm}^2 \text{ (compressive)}$$

Similarly for

$R=16$	$f_p=4610$	$f_R=1110$
$R=17.5$	$f_p=4140$	$f_R=-640$
$R=19$	$f_p=3780$	$f_R=-280$
$R=20.4$	$f_p=3500$	$f_R=0$

Fig. 12.6 shows the variation of stresses

From equation 12.3, axial stress

$$f_a=A=1750 \text{ kg/cm}^2 \text{ (Tensile)}$$

From equation 12.10

$$f_{av}=\frac{p_i\,R_i}{t}=\frac{1500\times15}{5.4}=4170 \text{ kg/cm}^2 \text{ (Tensile)}$$

From Fig. 12.7 it may be seen that the difference in stresses shows the extent of prestressing

$$f_i-f_{av}=5000-4170=830 \text{ kg/cm}^2 \text{ (Compressive)}$$
$$f_{av}-f_o=4170-3500=670 \text{ kg/cm}^2 \text{ (Tensile)}$$

12.8.3 MAXIMUM SHEAR STRESS THEORY

According to the maximum shear stress theory the value of k can be determined from equation 12.13.

$$k=\left(\frac{f}{f-p_i\sqrt{3}}\right)^{\frac{1}{2}}=\left(\frac{5000}{5000-1500\sqrt{3}}\right)^{\frac{1}{2}}=1.44$$

$$R_2=15\times1.44=21.6 \text{ cm}$$
$$t=R_2-R_1=21.6-15=6.6 \text{ cm}$$

12.8.4 MAXIMUM ENERGY OF DISTORTION THEORY

Equation 12.17 gives

$$p_i=2.31\ f\left(\frac{t}{2R_2}\right)\left(1-\frac{t}{2R_2}\right)$$

If the thickness 't' is 5.4 cm, then

$$p_i=2.31\times500\left(\frac{5.4}{2\times20.4}\right)\left(1-\frac{5.4}{2\times20.4}\right)$$
$$=1325 \text{ kg/cm}^2$$

The thickness will have to be increased for an internal pressure 1500 kg/cm^2.

12.8.5 MULTI-SHELL SHRINK FIT CONSTRUCTION

If the shell is made up of two concentric shells, then for optimum design, from equation 12.29

$$R_2=\sqrt{R_1 R_3}=\sqrt{15\times 20.4}=17.5 \text{ cm}$$

Assuming initial inner radius of the outer shell to be 17.46 cm, so that after heating it will be 17.5 cm, i.e., the difference in radius will be 0.04 cm, which is the expansion $\triangle$.

The temperature increase required for shrinking operation is given by equation 12.25.

$$t_f-t_i=\frac{(R_f-R_i)}{\alpha R_i}=\frac{17.5-17.46}{12.2\times 10^{-6}\times 17.46}$$

$$=183.5°\text{C}$$

From equation 12.24

$$p_f=\frac{E\triangle}{R_2}\,\frac{(R_2^2-R_1^2)(R_3^2-R_2^2)}{2R_2^2(R_3^2-R_1^2)}$$

$$=\frac{2\times 10^6\times 0.04}{17.5}\times\frac{(17.5^2-15^2)(20.4^2-17.5^2)}{2\times(17.5)^2\,(20.4^2-15)}$$

$$=348 \text{ kg/cm}^2$$

12.8.6 Stresses in Multi-shell Construction

Inner shell

internal pressure (p_i)—1500 kg/cm²

external pressure (p_f)—348 kg/cm²

For the determination of stresses the constants A and B are from equations 12.4 and 12.5

$$A=\frac{1500-348\times 1.167}{1.167^2-1} \qquad \left[k=\frac{17.5}{15}=1.167\right]$$

$$=3040$$

$$B=\frac{(348-1500)\times 1.167^2\times 15^2}{1.167^2-1}$$

$$=-980\times 10^3$$

$$f_p \text{ (stress at inner radius)}=3040-\frac{980\times 10^3}{(15)^2}$$

$$=-1320 \text{ kg/cm}^2 \text{ (Tensile)}$$

$$f_p \text{ (stress at outer radius)}=3040-\frac{980\times 10^3}{(17.5)^2}$$

$$=-170 \text{ kg/cm}^2 \text{ (Tensile)}$$

Outer shell

internal pressure—348 kg/cm²

external pressure—0

Constants A and B from equations 12.4 and 12.5 are

$$A=\frac{348}{1.167^2-1}=965 \qquad \left[k=\frac{20.4}{17.5}=1.167\right]$$

$$B=\frac{-348\times 1.167^2\times 17.5^2}{1.167^2-1}$$

$$=-402.5\times 10^3$$

$$f_p \text{ (stress at inner radius)}=965-\frac{402.5\times 10^3}{(17.5)^2}$$

$$=-365 \text{ kg/cm}^2 \text{ (Tensile)}$$

$$f_p \text{ (stress at outer radius)}=965-\frac{402.5\times 10^3}{(20.4)^2}$$

$$=965-969$$

$$=-4 \text{ kg/cm}^2 \text{ (Tensile)}$$

If the shrink fit multi-shell construction is compared with a single solid wall shell it may be seen that the maximum stress which occurs at the inner radius is reduced from 5000 to 1320 kg/cm², as shown in Fig. 12.9.

Reading References

Coming, E., *High Pressure Technology*, McGraw-Hill, New York, (1956).

Manning, R., *High Pressure Engineering*, Leonard-Hill, London, (1971).

Strauss, W., *Vessel for Very High Pressures* I, Brit. Chem. Eng., **15**(7), 889 (1970), **15**(8), 1057 (1970).

Strauss, W. and Pollard, L.J., *Pressure Vessels for very High Pressures* II, Brit. Chem. Eng., **15**(9), 1169 (1970), III, **16**(1), 49(1970).

Strezoff, S. et al., *Trends in Pressure Vessel and Closures*, Chem. Eng., **75**(22), 143 (1968), **75**(23), 141(1968).

McCarry, R.R. and Fey R.F., *Design of Thick-walled Pressure Vessel Shells*, Chem. Eng., **56** (August), 124 (1949), **56** (September), 105 (1949) **56** (October), 120 (1949).

Szantay, B.G., *Vessels with High Pressures*, Chem. Age of India, **22**(6), 361 (1971).

Ghosh, G.C., *High Pressure Vessels with Multilayer Construction*, Chem. Age of India, **23**(8), 669 (1972).

Bhattacharya, B.C., *High Pressure Vessel Design*, (*A jacketed construction*), Chem. Age of India, **25**(7), 455 (1974).

Mukherjee, S., *Self Energising Delta Gaskets for High Pressure Closures*, Chem. Age of India, **25**(6), 361 (1974).

Noel, M.R., *Multiwall Pressure Vessels*, Brit. Chem. Eng., **15** (7), 879 (1970).

Tangue, H., *The Design and Construction of High Pressure Chemical Plant*, Champan Hall, London, (1959).

Karl, E., *Strip wound Pressure Vessels, Safety, in High Pressure Polyethylene Plant*, CEP Technical Manual (26) (1973).

Bickel, M.B. and Ruiz C., *Pressure Vessel Design and Analysis*, Macmillan, London, (1967).

CHAPTER 13

Supports for Vessels

13.1 Introduction

Cylindrical and other types of vessels have to be supported by different methods. Vertical vessels are supported by bracket, column, skirt or stool supports, while horizontal vessels are supported by saddles. The choice of the type of support depends on the height and diameter of the vessel, available floor space, convenience of location, operating temperature and the material of construction.

The attachment of supports to the vessel walls cause additional stresses in the wall which the vessel should withstand in addition to the stresses created due to the operating pressure. Calculations of bending and other stresses in the shell due to supports are very complex. It is advisable to assess such stresses only when the effect of supports is significant in comparison to internal pressure. It is also necessary to ensure that the attachment of the support to the vessel, which is usually by fillet welds should be able to transfer the load safely from vessel to support and that the support should be strong enough to withstand the load of the vessel.

13.2 Bracket or Lug Supports

These (Fig. 13.1) can be easily fabricated from plates and attached to the vessel wall with minimum welding length. They are made to rest on short columns or on beams of a structure depending on the elevation required. They can be easily levelled. Due to the eccentricity of these supports and the resulting bending moment, compressive, tensile and shear stresses are induced in the vessel wall. These stresses must

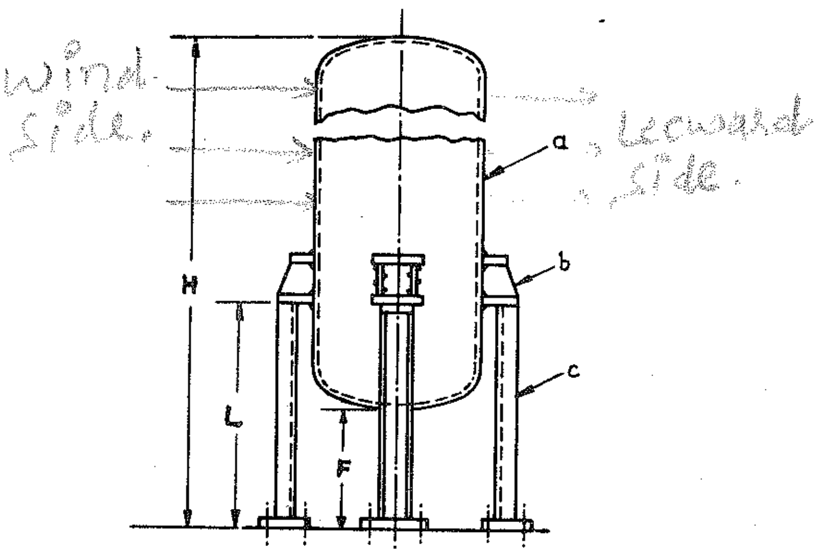


Fig. 13.1 Vessel supported on brackets
(a) shell (b) bracket (c) leg

be combined with circumferential and longitudinal stresses produced in the vessel wall due to operating pressure. The shear stresses being of a smaller magnitude can be ignored.

Bracket supports are most suitable for vessels with thick walls, as these are capable of absorbing the bending stresses due to eccentricity of loads. In vessels with thin walls, it is necessary either to reinforce certain area of the vessel wall, where the bracket is attached, or to use many brackets. In case a pad is used it is welded to the shell. The bracket is then welded to the pad. The thickness of the pad is usually made equal to or greater than the thickness of the shell with a minimum thickness of 6 mm. It is usual to provide 2 brackets for vessels upto 0.6 metre diameter, 4 brackets for vessels upto 3 metre diameter, 6 brackets for vessels upto 5 metre diameter and 8 brackets for vessels above 5 metre diameter.

The main loads on the bracket supports are the dead weight of the vessel with its contents and the wind load. The wind

load tends to overturn the vessel, particularly when it is empty. The maximum compressive stresses in the supports occur on the leeward side when the vessel is full, since dead load and wind load have a similar effect. The maximum tensile stresses are set up on the wind side when the vessel is empty, since dead load and wind load have opposing effects. Therefore the stresses on the leeward side are the determining factor for design of supports. The maximum total compressive load in the most remote support is (Fig. 13.1),

$$P = \frac{4P_w(H-L)}{n\,D_b} + \frac{\Sigma W}{n} \tag{13.1}$$

where P_w – total force due to wind load acting on the vessel [Equation 13.16 (a)]

H – height of vessel above the foundation

L – vessel clearance from foundation to vessel bottom

D_b – diameter of the bolt circle

ΣW – maximum weight of vessel with attachments and contents

n – number of brackets.

The windload is neglected if the vessel is indoors or if the height of the vessel is limited. The load P would then be only due to the weight of the vessel and its contents,

$$P = \left(\frac{\Sigma W}{n}\right)$$

Two types of brackets are generally used (Fig. 13.2).

(*a*) One type consists of two vertical gusset plates, with two additional horizontal plate stiffeners. All the plates are welded to the pressure vessel shell. Both the top and bottom stiffener plates are continuously welded to the shell, as the maximum compressive and tensile stress occurs in these two plates, respectively. The gusset plates are welded intermittently. All the welds are under a shearing load.

The compression load P induces a reaction in the pressure vessel wall around the bracket. The bending moment created is given by

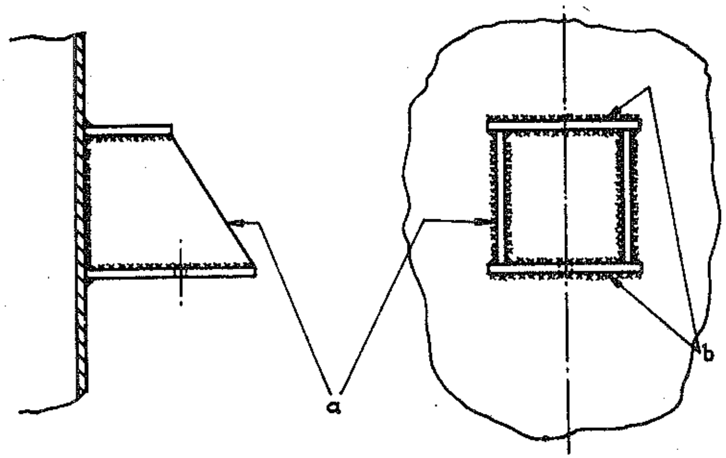


Fig. 13.2 Welding of bracket to shell
(a) gusset plates (b) stiffeners.

$$M_o = \frac{\beta^3 t^2 \, Pb \, R^2}{6\,(1-\mu^2)Ah} \tag{13.2}$$

where $\beta - \sqrt[4]{\frac{3(1-\mu^2)}{R^2\,t^2}}$

t—shell thickness
R—radius of shell
P—compressive load—(equation 13.1)
b—distance from the centre line of the shell plate to the centre line of the column
A—width of the compression plate
h—gusset height
μ—poison's ratio.

The axial stress in the vessel wall is, therefore, given by

$$f_a = \frac{6M_o}{t^2} = \frac{\beta^3 Pb\, R^2}{(1-\mu^2)Ah} \tag{13.3}$$

This is in addition to the axial stress produced by the internal pressure. A reinforcing pad is welded to the shell to take care of these additional stresses if they are of large magnitude.

(*b*) In the second case only the vertical gusset plates are welded to the shell. The bottom plate is not welded. The method of calculating moments and stresses for this type of construction, is complex and is, therefore, not considered here.

13.2.1 Thickness of the Base Plate

The plate is fixed on the edges with the load P considered as distributed over about half the area of the plate.

Average pressure on the plate $p_{av} = \frac{P}{aB}$ (13.4)

Maximum stress in a rectangular plate subjected to a pressure p_{av} and fixed at the edges is given by

$$f = 0.5\, p_{av} \frac{B^2}{T_1^2}\left(\frac{a^4}{B^4+a^4}\right) \tag{13.5}$$

where a, B, and T_1 are as shown in Fig. 13.3.

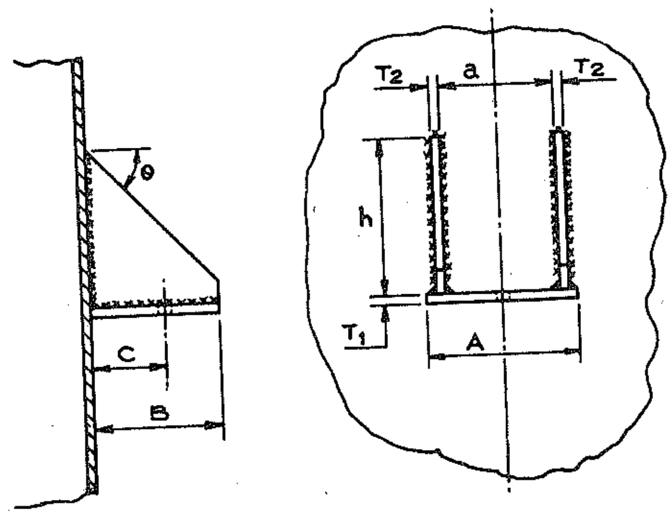


Fig. 13.3 Bracket details

Since, in this case the load is only distributed on the surface of contact between the base plate and the supporting beam, the actual stress may be taken as 40% more.

Therefore $$f=0.7p_{av}\frac{B^2}{T_1^2}\left(\frac{a^4}{B^4+a^4}\right) \tag{13.6}$$

The thickness T_1 should be so chosen that the stress (f) shall be within permissible limits.

13.2.2 Thickness of the Web (Gusset) Plates

There are two web plates for each bracket. The bending moment for each plate $= \frac{P}{2}\times C$

Section modulus of the plate $=\frac{1}{6}\,T_2h^2$

Stress at the edge $$f=\frac{3P\,C}{T_2h^2} \tag{13.7}$$

The edge is at an angle θ from the horizontal.

Therefore, the maximum compressive stress parallel to the edge of the web plate $$=\frac{3PC}{T_2h^2}\times\frac{1}{\cos\theta} \tag{13.8}$$

where C, T_2, h and θ are as shown in Fig. 13.3. T_2 should be so chosen that the stress shall not exceed the permissible value.

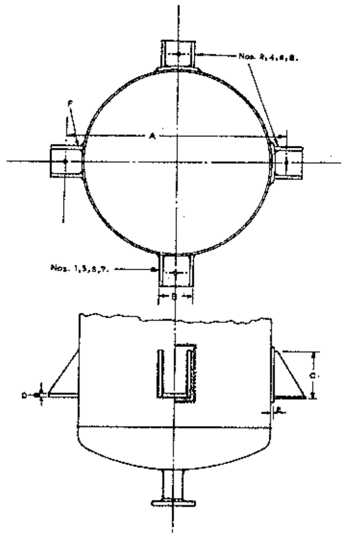


Fig. 13.4 Sizes of brackets

Fig. 13.4 and Table 13.1 give some of the major dimensions of the brackets used in practice.

13.2.3 Column Supports for Brackets

Columns are attached under the brackets. Hence the load on the column is not concentric with the weight of the vessel and its contents. These loading conditions act to produce an axial compression as well as bending due to eccentricity.

The nature of stresses induced in struts, acting as columns is indicated under Section 3.9. Column supports in steel are designed for slenderness ratios upto 120. Compressive stresses can be determined either on the basis of struts under direct compression or on the basis of the Rankine-Gordon formula, depending on the slenderness ratio.

If the load P is acting eccentric on a short column, the maximum combined bending and direct stress is given by

$$f=\frac{\Sigma W}{A\times n}+\frac{\Sigma W\times e}{n\times Z} \tag{13.9}$$

where ΣW—load on column

A—area of cross-section

e—eccentricity

Z—modulus of section of the cross-section

n—number of columns.

If the column is long, the stress

$$f=\frac{\Sigma W}{nA}\left[1+a\left(\frac{l_e}{r}\right)^2\right]+\frac{\Sigma W\times e}{n\,Z} \tag{13.10}$$

where l_e—effective length of the column

a—constant

r—radius of gyration

$l_e=\frac{l}{2}$ (for fixed ends of the column)

In the case of vessels which are outside the building, column supports will also be subjected to wind load. The stress induced due to wind load will be given by

$$f_w=\left(\frac{p_w}{n}\times\frac{l}{2}\right)\frac{1}{Z} \tag{13.11}$$

Table 13.1
Brackets for Vertical Vessels
Centre to Centre of Bolt-holes in Lugs (Dimension A)

Tank dia (m)	0.75	0.90	1.05	1.20	1.35	1.50	1.80	2.10	2.40	2.70
Dim A (m)	0.90	1.05	1.20	1.35	1.50	1.65	1.95	2.35	2.65	2.95

Support No.	DIMENSIONS B (cm)	C (cm)	D (mm)	E (mm)	F (mm)	Total Maximum weight Four Lugs can support (kg)
1	15	15	10	—	8	11300
2	15	15	10	10	8	
3	15	23	10	—	8	22600
4	15	23	10	10	8	
5	20	25	13	—	10	45200
6	20	25	13	13	10	
7	20	25	13	—	10	68000
8	20	25	13	13	10	

where p_w—total force acting on the vessel due to wind as per equation 13.16 (a)

l—column height

n—number of columns.

Stress due to wind load should be combined with the stress due to dead loads.

In designing the size of an appropriate column, it is usual to choose a structural section such as a joist, a channel or a T-section of a suitable size and determine the stresses as per the above equations. These stresses should be well within the range of permissible stresses. It is desirable to be conservative in the choice of the cross-section of the column support and work with lower stresses than permissible.

13.3 Leg Supports

Structural sections such as angles, channels or joists can be directly welded to the pressure vessel shell to form vertical legs as shown in Fig. 13.5. Sometimes these legs can be made detachable by bolting leg plates welded to shell. The design of such supports is similar to those of column supports for brackets. The number of legs may be same as those specified for bracket supports. The legs are attached to the vessel by fillet welds. The shear stress produced in the weld by the load is given by

$$f_{ws} = \frac{\Sigma W}{0.707 \times t_w \times l_w \times n} \qquad (13.12)$$

where ΣW—load

t_w—side of weld (weld height)

l_w—length of weld.

n—number of legs

A tensile stress will also be produced in the weld due to the eccentricity of the load. This stress may be neglected.

This type of support is permitted only for small vessels. Severe local stresses are produced at the connection of the support to the vessel wall, which should be spread over a sufficient area by providing a reinforcing pad, as in the case

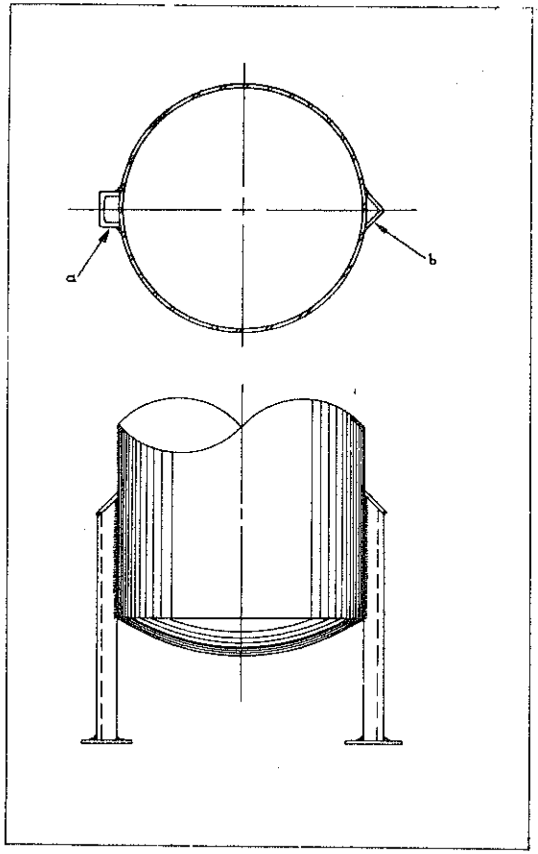


Fig. 13.5 Leg supports
(a) angle leg (b) channel leg

of brackets. While determining the stresses in the supports the length may be taken as height of the leg between the lower edge of the welded portion and the base plate.

13.3.1 Base Plate for Column or Leg Supports

The column support is usually fabricated from a structural section such as joist or a channel. It is assumed that the load from the column is transmitted to the base plate uniformly on

a rectangular area having sides equal to 0.8 times the width of the section and 0.95 times the depth of section (Fig. 13.6). It

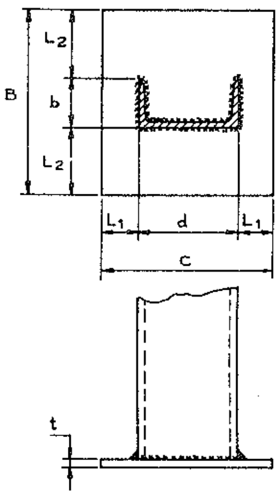


Fig. 13.6 Base plate for column or leg support

is also assumed that the base plate has a uniform bearing pressure (p_b) on its surface, which is resting on the foundation.

$$p_b = P \times \frac{1}{B \times C} \qquad (13.13)$$

B and C are the two sides of the base plate.

If p_b is the allowable bearing pressure of steel plates on the foundation, it is possible to determine the dimensions B and C. The allowable pressure is based on the material used for foundations.

As shown in Fig. 13.6 let the projections of the base plate be L_1 and L_2 and the thickness of the plate be t. Taking a unit width of the strip of base plate, the maximum bending moment in the plate is

$$M = \frac{p_b L_1^2}{2} \text{ or } \frac{p_b L_2^2}{2} \text{ whichever is greater.}$$

Maximum stress in the plate, $f = \frac{6M}{t^2}$ (13.14)

13.4 Skirt Supports

Tall vertical vessels are usually supported by cylindrical shells or skirts (Fig. 13.7). The cross-section of the skirt is uniformly

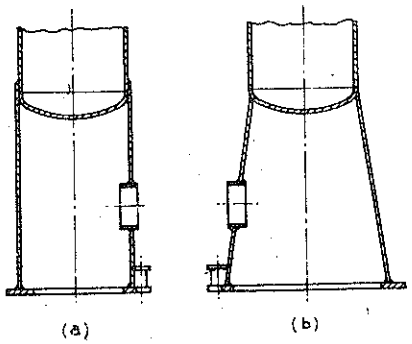


Fig. 13.7 Skirt supports (*a*) straight skirt (*b*) angular skirt

distributed at a sufficient distance from the axis. This gives a large value of the section modulus and helps to increase the resistance to bending action. The skirt is, therefore, a suitable supporting structure for tall vessels which are subjected to wind, seismic and other loads which cause a bending moment at the base of the vessel.

The skirt is welded to the bottom dished head, either flush with the shell or to the outside of the shell. In the first case, in the absence of wind and seismic loads, the weld is in compression. In the latter method of welding the weld is in shear. A bearing plate is attached to the bottom of the skirt. This plate is made to rest on a concrete foundation and is securely anchored to the foundation by means of anchor bolts embedded in concrete to prevent overturning from the moments induced by wind or seismic loads. The bearing plate is in the form of a rolled angle or a single flat ring with or without gussets. Alternately a bolting chair formed by two flat rings namely a bearing plate and compression plate, with gusset plates in between, is used for securing the skirt, to the foundation.

13.4.1 Skirt Design

The cylindrical shell of the skirt is designed for the combination of the stresses due to vessel dead weight, wind load and seismic load. The skirt thickness is uniform and is designed to withstand the maximum values of tensile or compressive stresses. These stresses are :

(*a*) Due to dead weight

$$f_d = \frac{\Sigma W}{\pi\, D_{ok}\, t_{sk}} \tag{13.15}$$

where f_d—stress

w—dead weight of vessel contents and attachments

D_{ok}—outside diameter of skirt

t_{sk}—thickness of skirt.

(*b*) *Due to wind load* : The forces due to wind load acting on the lower and upper parts of the vessel are shown in Fig. 13.8

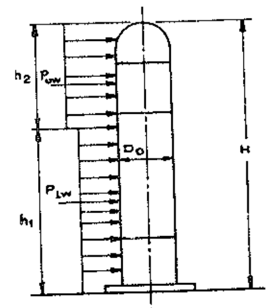


Fig. 13.8 Incidence of wind load.

and are determined as follows :

$$p_{lw} = k\, p_1\, h_1\, D_o \qquad [13.16(a)]$$

upto 20 m height

$$p_{uw} = k\, p_2\, h_2\, D_o \qquad [13.16(b)]$$

above 20 m height

where p_1—wind pressure for the lower part of the vessel (40 to 100 kg/cm^2)

p_2—wind pressure for the upper part of the vessel (upto 200 kg/m^2)

k—coefficient depending on the shape factor (0.7 for cylindrical surface)

D_o—outside diameter of the vessel.

The bending moment due to wind at the base of the vessel is determined by

$$M_w = p_{bw}\frac{H}{2} \qquad [13.17\,(a)]$$

(Upto $H \leqslant 20$ m)

$$= p_{lw}\,\frac{h_1}{2} + p_{uw}\left(h_1 + \frac{h_2}{2}\right) \qquad [13.17\,(b)]$$

($H > 20$ m)

$$\text{The stress } f_{wb} = \frac{M_w}{Z} = \frac{4\,M_w}{\pi\,D_o^2\,t_{sk}} \qquad (13.18)$$

where Z—modulus of section of skirt cross-section

t_{sk}—skirt thickness

D_{ok}—outside diameter of skirt.

(c) *Due to seismic load* : This is a vibrational load resulting from earthquakes. The load is normal to the vessel shell and decreases linearly as shown in Fig. 13.9. The load may therefore be considered as acting at a distance $\frac{2}{3}$ the height of the vessel from bottom.

$$\text{Load } F = CW \qquad (13.19)$$

where W—total weight of vessel

C—Seismic coefficient

This load is actually distributed as a triangular shaped load.

$$(\text{load upto the height } X) = V_x = (L + L_x) \times \frac{X}{2}$$

$$\text{B.M. } M_{sx} = \frac{CW\,X^2}{3} \times \frac{(3H - X)}{H^2} \qquad (13.21)$$

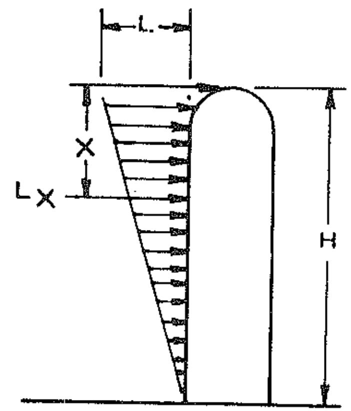


Fig. 13.9 Distribution of seismic load

$$\text{Stress } f_{sx} = \frac{M_{sx}}{\pi\,R_{ok}^2\,t_{sk}} \qquad (13.22)$$

The maximum bending moment is at the base of the skirt and may be found by substituting $X = H$

$$V_{sb} = CW$$

$$M_{sb} = CW \times \tfrac{2}{3}H = \tfrac{2}{3}CWH$$

$$\text{Stress at base, } f_{sb} = \frac{M_{sb}}{\pi\,R_{ok}^2\,t_{sk}} = \frac{2}{3} \times \frac{CWH}{\pi R_{ok}^2\,t_{sk}} \qquad (13.23)$$

The possibility of the wind load and earthquake load operating simultaneously is remote. Therefore the stresses due to wind load and earthquake loads are computed separately and the most adverse loading condition is used to calculate the maximum resultant stress.

Max. tensile stress at the bottom of skirt

$$= (f_{wb} \text{ or } f_{sb}) - f_{db} \qquad (13.24)$$

Max. compressive stress on skirt

$$= (f_{wb} \text{ or } f_{sb}) + f_{db} \qquad (13.25)$$

In view of the complexity of the final equations for maximum stresses, it is usual to assume a suitable thickness (t_{sk}) of the skirt and check for the maximum stresses, which should not exceed the permissible values.

13.4.2 Skirt Bearing Plate

The maximum compressive stress between the bearing plate and the concrete foundation is given by the following equation :

$$f_c = \frac{\Sigma W}{A} + \frac{M_w}{Z} \qquad (13.26)$$

where ΣW—weight of vessel, contents and attachments

A—area of contact between the bearing plate and foundation

M_w—bending moment due to wind

Z—section modulus of the area.

The maximum compressive stress (f_c) must be less than the permissible compressive stress in concrete. The thickness of the bearing plate is determined by considering it as a uniformly loaded cantilever with f_c as the uniform load. The maximum bending moment for the contilever occurs at the junction of the skirt and bearing plate. This is given by (Fig. 13.7).

$$M_{max} = f_c\, bl \times \frac{l}{2} = f_c \frac{bl^2}{2} \qquad (13.27)$$

where l—difference between outer radius of the bearing plate and outer radius of skirt.

b—circumferential length.

The stress is

$$f = \frac{6\, M_{(max)}}{b\, t_B^2}$$

$$= \frac{3\, f_c\, bl^2}{b\, t_B^2} = \frac{3\, f_c\, l^2}{t_B^2} \qquad (13.28)$$

where t_B—thickness of bearing plate.

from which the thickness t_B can be determined.

If the calculated thickness is less than 12 mm a steel rolled angle may be used as a bearing plate (Fig. 13.10). If however the thickness is between 12 to 18 mm a single ring bearing plate with or without gusset plates may be used (Fig. 13.11). The strengthening effect of the gusset plates will help to reduce the thickness of the bearing plate.

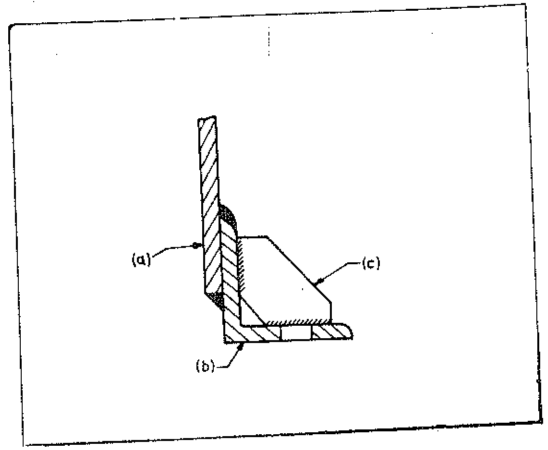


Fig. 13.10 Angle bearing plate
(*a*) skirt (*b*) angle (*c*) gusset plate

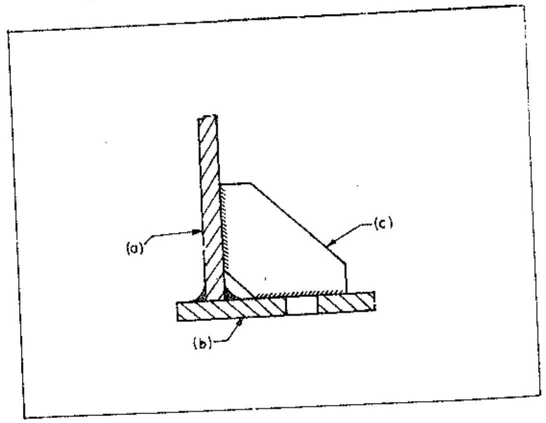


Fig. 13.11 Ring Bearing Plate
(*a*) skirt (*b*) ring plate (*c*) gusset plate

13.4.3 Anchor Bolt Design

The minimum stress between the bearing plate and the concrete foundation will be (compare equation 13.1)

$$f_{c(min)} = \frac{W_{min}}{A} - \frac{M_w}{Z} \qquad (13.29)$$

where W_{min}—minimum weight of the empty vessel.

If the value of f_c (min) is positive or equals zero it is necessary to assess the so-called coefficient of stability (Y) from the following equation :

$$Y = \frac{M_{weight}}{M_{wind}} = \frac{W_{min} \times R}{M_{wind}} \qquad (13.30)$$

where M—moment

R—arm of force of weight of vessel

—0.42 D_o

If $Y > 1.5$, the vessel need not be anchored. In this case the bolts are used only for fixing the vessel to the foundation. If f_c(min) is negative, the vessel must be anchored to concrete foundation by means of anchor bolts to prevent overturning owing to the moment produced by the wind or seismic load.

The approximate value of the load on each bolt may be determined by the equation.

$$p_{bolt} = f_c(\text{min}) \frac{A}{n} \qquad (13.31)$$

where p_{bolt}—load on one anchor bolt

n—number of bolts

f_c—stress determined by equation 13.29

A—Area of contact between bearing plate and foundation (Fig. 13.13).

The stress in the bolt

$$f_{bolt} = \frac{p_{bolt}}{A_b} \qquad (13.32)$$

where A_b—Area cf cross-section of bolt.

The stress f_{bolt} should not exceed the permissible tensile stress in the bolt material.

13.4.4 Design of Bolting Chair

If the thickness of the bearing plate is more than 18 mm, it is advisable to use a bolting chair. This could be of the centered type (Fig. 13.12) or external type (Fig. 13.13). Centered chairs are used when the skirt diameter is more than 3 metres, and the number of bolts does not exceed 24. For larger skirt diameters, an external bolting chair is suitable.

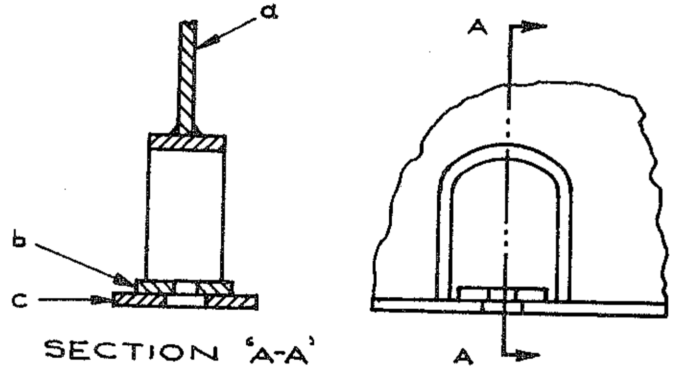


Fig. 13.12 Bolting chair—Centered type
(a) skirt (b) washer (c) bearing plate

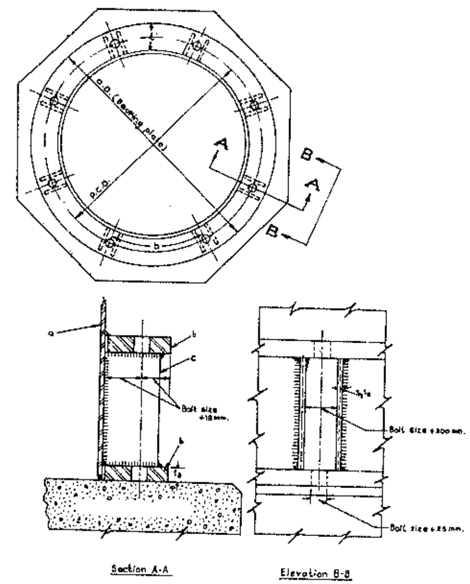


Fig. 13.13 Bolting chair—External type
(a) skirt (b) compression ring (c) gusset plate (d) bearing plate

(*a*) *Centered chair*

While calculating the bearing plate thickness for a centered chair, the plate inside the stiffeners is considered as a beam with fixed ends and with a concentrated load. The maximum bending moment in the bearing plate inside the chair will be near the bolt, and can be determined by the relation

$$M_{max} = \frac{p_{bolt} \times b}{8} \tag{13.33}$$

where b—spacing inside chair

p—maximum bolt load

The thickness of the bearing plate considering the reduction of section due to the bolt hole will be

$$t_{B_p} = \sqrt{\frac{6M_{max}}{(w_{B_p} - d_{bhd}) f_{all}}} \tag{13.34}$$

where t_{B_p}—bearing plate thickness

w_{B_p}—width of bearing plate

d_{bhd}—bolt hole diameter in bearing plate

f_{all}—allowable tensile stress in bolt.

The thickness of bearing plate between the chairs can be checked by use of equation (13.28).

(*b*) *External chair*

The size and pitch of bolts may be calculated as indicated earlier. The dimensions of the chair may be determined approximately by the relationship given in Fig. 13.13.

13.5 Saddle Supports

Horizontal cylindrical vessels are supported on saddles. These are placed at two positions. If necessary the shell of the vessel is strengthened by stiffeners, located on the shell area surrounding the saddle [Fig. 13.14 (a)]. Supports in the form of rings are preferable for vessels in which supports at more than two positions, are unavoidable. For large thin-walled vessels or vessels under vacuum, which are liable to be weakened by heavy loads of fittings or other supported structures, it is

necessary to provide ring supports. For small vessels simple leg supports may be used.

The location of the saddle support should be such that the distance from the tangent line to the centre line of the saddle should be less than the radius of the vessel, in order to take advantage of the stiffening effect of the head. This distance '*A*' is usually 0.4 to 0.5 *R* and less than 0.2 '*L*' (Fig. 13.14). The included angle θ should not normally be less than 120°.

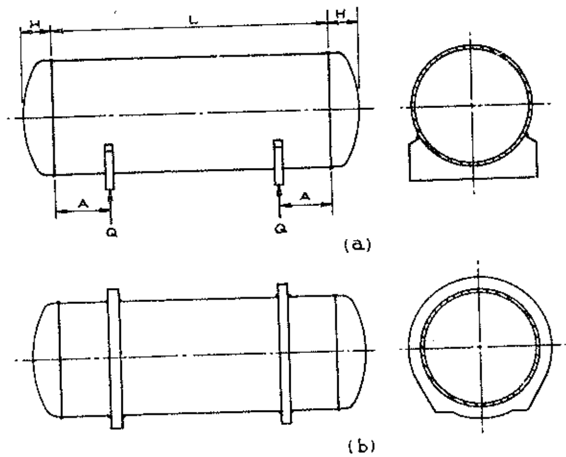


Fig. 13.14 Saddle supports
(a) plate type (b) ring type

Horizontal vessels resting on saddle supports behave as beams. They are subjected to longitudinal bending moments and local shear stresses due to the weight of the vessel, its contents and any attachment. The resulting stresses in the shell are longitudinal stresses, tangential shear stresses and circumferential stresses.

13.5.1 Longitudinal Bending Moments

A cylindrical vessel with dished closures at the ends may be treated as an equivalent cylinder having a length equal to $(L + \frac{4}{3} H)$, where L is the length between the tangent lines and H is the depth of dished closure.

The weight of fluid and the vessel may be considered as a uniform load on the equivalent length. If two symmetrical supports are considered each will carry a load

$$Q = \frac{w}{2}\left(L + \frac{4}{3} H\right) \tag{13.35}$$

where w — uniformly distributed load

In the loaded condition the shell and load of heads can be taken as an overhanging beam with two supports, so that maximum bending moments are created. One maximum is at the saddle supports, the other acts in the centre of the span.

The bending moment at the supports is

$$M_1 = QA\left[1 - \frac{1 - \frac{A}{L} + \frac{(R^2 - H^2)}{2AL}}{1 + \frac{4}{3}\frac{H}{L}}\right] \tag{13.36}$$

(where A, L, R are as shown in Fig. 13.14).

The bending moment at the centre of the span is given by

$$M_2 = \frac{QL}{4}\left[\frac{1 + 2\left(\frac{R^2 - H^2}{L^2}\right)}{1 + \frac{4}{3}\frac{H}{L}} - \frac{4A}{L}\right] \tag{13.37}$$

13.5.2 Stress in Shell at the Saddle

The longitudinal bending stress in the shell will depend on the stiffness of the shell at the cross-section where the bending is maximum. If the shell does not retain its round shape under the load, a portion of the upper part of its cross-section is ineffective against bending.

(*a*) In case the stiffness is enough to maintain a circular cross-section (i.e. $A < 0.5\ R$) the whole cross-section is effective and, therefore, the stress due to bending is given by

(i) At the topmost fibre of the cross-section

$$f_1 = \frac{M_1}{k_1 \pi R^2 t} \tag{13.38}$$

(ii) At the bottommost fibre of the cross-section

$$f_2 = \frac{M_2}{k_2 \pi R^2 t} \tag{13.39}$$

where t—thickness of shell and $k_1 = k_2 = 1.0$

(*b*) For $A > 0.5R$, the shell is not sufficiently stiffened by the end.

The value of the factor $k_1 = 0.107$ for $\theta = 120°$
$= 0.161$ for $\theta = 150°$

and $k_2 = 0.192$ for $\theta = 120°$
$= 0.279$ for $\theta = 150°$

where θ is as shown in Fig. 13.14.

If there is any internal pressure on the vessel, the resultant stress will be either tensile or compressive.

The tagential shear stress is generally of a small magnitude and may be ignored. The circumferential stress at the saddle due to bending may be of limited value if the thickness of shell is greater than $0.1\ R$ and the unsupported length 'A' is less than $0.5\ R$.

13.5.3 Stresses in the Shell at the Mid-Span

The stress at the mid-span is

$$f_3 = \frac{M_2}{\pi R^2 t} \tag{13.40}$$

which is either tensile or compressive depending on the position of the fibre.

The resultant tensile stresses (including the axial stress due to internal pressure) should not exceed the permissible stress, and the resultant compressive stress should not exceed the permissible compressive stress or $\left(\frac{Et}{16\ R}\right)$, whichever is less, where E is modulus of elasticity.

13.5.4 Wear Plates and Stiffeners

The stresses in the shell band adjoining the saddle may be reduced by attaching a wear plate to the shell directly over the saddle, somewhat larger than the surface of the saddle (Fig. 13.15). In the case of thin-walled vessels or in the case of a saddle located away from the head ($A > 0.5\ R$) the shell alone may not resist the circumferential bending moment. Ring stiffeners are then attached to the shell to alleviate the load

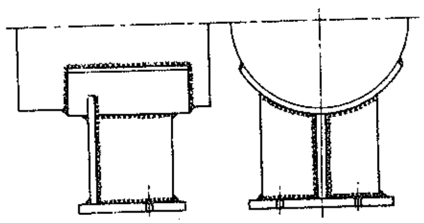


Fig. 13.15 Wear plate for saddle

on the shell. These are in the form of a ring welded either inside or outside the shell. An inside ring stiffener is more desirable, from the point of view of strength (Figs. 13.16 and 13.17).

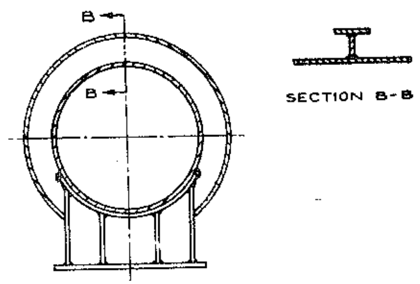


Fig. 13.16 Internal stiffening ring

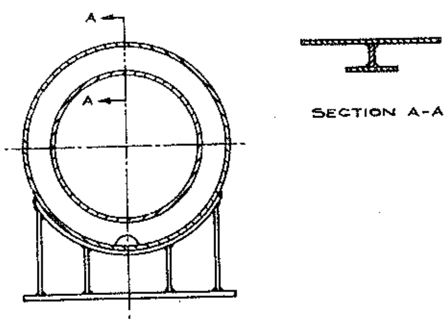


Fig. 13.17 External stiffening ring

13.5.5 Design of Saddles

Saddles should be strong enough to withstand the loads imposed on the vessel. Suitable sizes of saddles suggested for different diameters of vessels are indicated in Table 13.2 and Fig. 13.18.

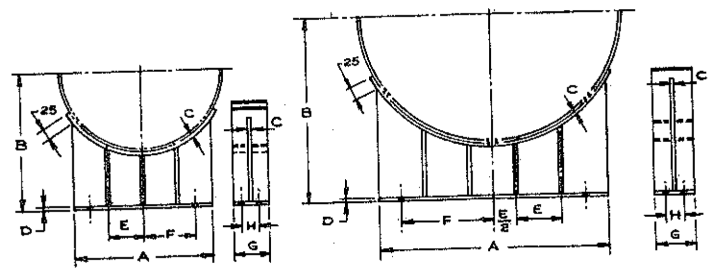


Fig. 13.18 Saddle support sizes
(a) upto 120 cm drum diameter (b) over 120 cm drum diameter

13.6 Numerical Problems

13.6.1 Bracket or Lug Support for a Vertical Cylindrical Vessel

Data—(Fig. 13.1)

Diameter of vessel		1.5 m
Height of vessel		2.0 m
Clearance from vessel bottom to foundation		1.0 m
Weight of vessel with contents		4000 kg
Wind pressure		128.5 kg/m^2
Number of brackets		4
Diameter of anchor bolt circle (From table 13.1)		1.65 m
Height of bracket from foundation		2.25 m
Permissible stresses for structural steel		
(IS-800)	Tension	1400 kg/cm^2
	Compression	1233 kg/cm^2
	Bending	1575 kg/cm^2
Permissible bearing pressure for concrete		35kg/cm^2

Table 13.2
Welded saddles for Horizontal Vessels

Drum dia (m)	A (cm)	B (cm)	C (cm)	D (cm)	E (cm)	F(cm)	G (cm)	H (cm)	Bolt dia (mm)	Weld size (mm)	Rib size (mm)	Max allow load (kg)
0.60	55	50	10	13	19	24	15	10	19	6	70×10	2300
0.90	80	65	10	13	28	34	15	10	19	6	70×10	3900
1.20	110	80	13	16	37	45	20	14	22	6	90×13	6300
1.50	135	95	13	16	33	58	20	14	22	6	90×13	7100
1.80	160	110	13	16	40	71	20	14	22	6	90×13	12300
2.10	190	125	16	19	48	81	23	15	22	8	100×16	44600
2.40	215	140	16	19	53	94	23	15	25	8	100×16	44100
2.70	240	155	16	19	61	106	25	18	25	8	115×16	50800
3.00	265	170	16	19	66	117	25	18	25	8	115×16	49500
3.30	295	185	16	19	73	129	25	18	25	8	115×16	49200
3.60	320	200	16	19	80	140	25	18	25	8	115×16	47900

13.6.1.1 MAXIMUM COMPRESSIVE LOAD

Wind pressure from equation 13.16

$$P_w = K\,p.h.D_o$$
$$= 0.7 \times 128.5 \times 2 \times 1.5$$
$$= 270 \text{ kg}$$

From equation 13.1

$$P = \frac{4 \times 270 \times 2}{4 \times 1.65} + \frac{4000}{4}$$
$$= 1327 \text{ kg}$$

13.6.1.2 BRACKET

(a) *Base plate* : From table 13.1 suitable base plate size

$a = 140$ mm $\qquad B = 150$ mm

From equation 13.4

$$p_{aw} = \frac{1327}{14 \times 15} = 6.32 \text{ kg/cm}^2$$

From equations 13.5 and 13.6

$$f = 0.7 \times 6.32\ \frac{15^2}{T^2}\left(\frac{14^2}{15^2 + 14^2}\right)$$
$$= \frac{465}{T^2}$$
$$f = 1575 \text{ kg/cm}^2$$
$$T^2 = \frac{465}{1575} \times 100 \text{ mm}^2$$
$$T_1 = 5.4 \text{ mm}$$

Use a 6 mm thick plate.

(b) *Web plate* : From equations 13.7 and 13.8

$$\text{Bending moment of each plate} = \frac{1327}{2} \times \frac{(1.65 - 1.5)}{2} \times 100$$
$$= 5150 \text{ kg cm}$$

$$\text{Stress at the edge } f = \frac{5150}{T_2 \times 14 \times 14} \times \frac{1}{0.707}$$

$$f = 1575$$
$$T_2 = 0.1425$$
$$= 1.1425 \text{ mm}$$

T_2 may be taken as 4-6 mm

13.6.1.3 COLUMN SUPPORT FOR BRACKET

It is proposed to use a channel section as column. The size chosen is ISMC 150.

Size—150×75

Area of cross-section (A)—20.88 cm^2

Modulus of section (Z_{yy})—19.4 cm^3

Radius of gyration (r_{yy})—2.21 cm

Weight —16.4 kg/m

Height from foundation —2.25 m

Equivalent length for fixed ends $l_e=\frac{l}{2}=\frac{2.25}{2}=1.125$ m

Slenderness ratio $\frac{l_e}{r}=\frac{1.25\times100}{2.21}=51.0$

From equation 13.9

$$f=\frac{1327}{20.88}+\frac{1327\times7.5}{19.4}=576.5 \text{ kg/cm}^2$$

From equation 13.10

$$f_c=\frac{1327}{20.88}\left[1+\frac{1}{7500}\times(51.0)^2\right]+\frac{1327\times7.5}{19.4}$$

$$=604 \text{ kg/cm}^2$$

The calculated values are less than the permissible compressive stress and hence the channel selected is satisfactory.

13.6.1.4 BASE PLATE FOR COLUMN

The size of the column is 150×75. It is assumed that the base plate extends 20 mm on either side of the channel.

Side B—0.8×75+2×20 =100 mm

Side C—0.95×150+2×20=182.5 mm

From equation 13.13

Bearing pressure $p_b=\frac{1327}{4}\times\frac{1}{10\times18.25}=1.82$ kg/cm²

This is less than the permissible bearing pressure for concrete.

$$\text{Stress in the plate } f=\left[\frac{\frac{1.82}{2}\times\frac{20^2}{10}}{t^2/6}\right]=\frac{21.8}{t^2}$$

$$f=1575 \text{ kg/cm}^2$$

$$t^2=\frac{21.8}{1575}\times100 \text{ mm}^2$$

$$t=1.38 \text{ mm}$$

It is usual to select a plate 4-6 mm thick.

13.6.2 SKIRT SUPPORT FOR A VERTICAL CYLINDRICAL VESSEL

Refer to Figs. 13.7 and 13.12

Data

Diameter of vessel	3000 mm
Height of vessel	37.5 m
Weight of vessel, attachments, etc.	200,000 kg
Diameter of skirt (straight)	3000 mm
Height of skirt	4.8 m
Wind pressure	128.5 kg/m²

13.6.2.1 SKIRT

Stress due to dead weight from equation 13.15

$$f_d=\frac{200{,}000}{\pi\times300\times t_{sk}}=\frac{212.5}{t_{sk}} \text{ kg/cm}^2$$

Stress due to wind load from equation 13.18 and Fig. 13.7

$$f_{wb}=\frac{(0.7\times128.5\times20\times3\times\frac{20}{2}\times100)}{\pi\left(\frac{3\times100}{2}\right)^2 t_{sk}}+\frac{0.7\times128.5\times(37.5+4.8-20)3\times(20+11.15\times100)}{\pi\left(\frac{3\times100}{2}\right)^2 t_{sk}}$$

$$=\frac{342.5}{t_{sk}} \text{ kg/cm}^2$$

Stress due to seismic load from equation 13.23

$$f_{sb}=\frac{2}{3}\times\frac{0.08\times200{,}000}{\pi(3\times100)^2\frac{t_{sk}}{2}}$$

$$f_{sb} = \frac{2}{3}\cdot\frac{CWH}{\pi R_o K \frac{2}{}t_{sk}}$$

where $C=0.08$

$$f_{sb}=\frac{0.151}{t_{sk}}\ \text{kg/cm}^2$$

Max tensile stress from equation 13.24

$$f_{t(max)}=\frac{342.5}{t_{sk}}-\frac{212.5}{t_{sk}}=\frac{130}{t_{sk}}\ \text{kg/cm}^2$$

Permissible tensile stress—1400 kg/cm²

$$t_{sk}=\frac{130}{1400}\ \text{cm}$$

$$=0.929\ \text{mm}$$

Max compressive stress from equation 13.25

$$f_{c(max)}=\frac{342.5}{t_{sk}}+\frac{212.5}{t_{sk}}=\frac{555}{t_{sk}}\ \text{kg/cm}^2$$

$$f_c\ \text{permissible} \leqslant \frac{1}{3}\ \text{yield point}$$

$$\leqslant\frac{2000}{3}=666\ \text{kg/cm}^2$$

$$t_{sk}=\frac{555}{666}\ \text{cm}$$

$$=8.32\ \text{mm}$$

Use a thickness of 10 mm

13.6.2.2 SKIRT BEARING PLATE (Fig. 13.7)

Assume bolt circle diameter = skirt diameter + 32.5 cm
= 332.5 cm.

Compressive stress between bearing plate and concrete foundation

$$f_c = \frac{200{,}000}{A}+\frac{0.7\times128.5\times3\times42.3\times42.3}{2\times Z}$$

$$=\frac{200{,}000\times4}{\pi(332.5^2-300^2)}+\frac{90\times3\times(42.3)^2}{\frac{2\pi(332.5^4-300^4)}{32\times332.5}}$$

$$=12.2\ \text{kg/cm}^2$$

which is less than the permissible value for concrete. Maximum bending moment in bearing plate from equation 13.37.

$$M_{max} = \frac{12.2\times(16-25)^2}{2}$$

$$\text{Stress } f = \frac{6\times12.2}{t^2_B}\times\frac{(16.25)^2}{2}=\frac{9670}{t_B}$$

Permissible stress in bending is 1575 kg/cm²

$$t_B^2 = \frac{967}{1575}\ \text{cm}^2$$

$$t_B = 61.5\ \text{mm}$$

A bolting chair has to be used.

13.6.2.3 ANCHOR BOLTS

Minimum weight of the vessel—173.75×10^3 kg (assumed)
From equation (13.29)

$$f_c = \frac{173.75\times10^3\times4}{\pi(332.5^2-300^2)}-\frac{0.7\times127.5\times3\times(42.3)^2}{\frac{2\pi\,(332.5^4-300^4)}{32\times332.5}}$$

$$= 11.0-20.4$$

$$= -9.4\ \text{kg/cm}^2$$

Since f_c is negative, the vessel skirt must be anchored to the concrete foundation by anchor bolts

From equation 13.31 assuming 24 bolts

$$P_{bolt} = \frac{9.4}{24}\times\frac{332.5^2-300}{4} = 6650\ \text{kg}$$

13.6.3 SADDLE SUPPORT FOR HORIZONTAL VESSEL

Data

Material—low carbon steel
Vessel diameter—1230 mm
Length of shell—8000 mm
Torispherical head—crown radius—1250 mm
knuckle radius—6% of diameter; total depth of head—257 mm

$\pi d_i l$

Working pressure—5 kg/cm^2
Shell thickness—10 mm
Head thickness—12 mm
Corrosion allowance—1.5 mm
Permissible stress—950 kg/cm^2
Weight of vessel and contents—11943 kg
Distance of saddle centre line from shell end—320 mm

13.6.3.1 LONGITUDINAL BENDING MOMENTS

From equation 13.36

$$M_1 = 5972.5\times 32 \left[1-\frac{1-\dfrac{32}{800}+\dfrac{61.5^2-25.7^2}{2\times 32\times 800}}{1+\dfrac{4}{3}\times\dfrac{25.7}{800}}\right]$$

$$= 5972.5\times 32\left(1-\frac{1.0213}{1.0428}\right)$$

$$= 3810 \text{ kg cm}$$

From equation 13.37

$$M_2=\frac{5972.5\times 800}{4}\left[\frac{1+2\,\dfrac{(61.5^2-25.7^2)}{800^2}}{1+\dfrac{4}{3}\times\dfrac{25.7}{800}}-\frac{4\times 32}{800}\right]$$

$$= 1062{,}000 \text{ kg cm}$$

13.6.3.2 STRESSES IN SHELL AT THE SADDLE

From equation 13.38

$$f_1 = \frac{3810}{\pi(61.5)^2\dfrac{(10-1.5)}{10}}$$

$$= 0.38 \text{ kg/cm}^2$$

From equation 13.39

$$f_2 = \frac{1062{,}000}{\pi(61.5)^2\dfrac{(10-1.5)}{10}}$$

$$= 105.2 \text{ kg/cm}^2$$

The stresses are well within the permissible values.

13.6.3.3 STRESS IN THE SHELL AT MID-SPAN

From equation 13.40

$$f_3=\frac{1062{,}000}{\pi\,(61.5)^2\,\dfrac{(10-1.5)}{10}}$$

$$= 105.2 \text{ kg/cm}^2$$

From equation 6.2, axial stress in the shell due to internal pressure—

$$f_p = \frac{5.5\times 123\times 10}{4(10-1.5)}$$

$$= 200 \text{ kg/cm}^2$$

The sum of f_3 and f_p is well within the permissible values.

Reading References

Brownell, L.E. and Young, E.H., *Process Equipment Designs*, John Wiley, New York, (1968).

Taganov, N.I., *Process Equipment Design*, Academic Books, Bombay, (1958).

CHAPTER 14

Agitators

14.1 Introduction

Certain processing operations, such as blending, dispersion, dissolution, gas absorption, crystallization etc., need agitation of the liquids. In such operations an agitator system has to be provided, along with the basic equipment. The basic equipment may be a tank, a reaction vessel, a kettle or a crystalliser. Selection of an efficient agitation system will depend on the nature of liquid, operating conditions and the intensity of circulation and shear. A variety of agitation systems are available, each one having a useful operating range. The limits of usefulness of each system are not clearly defined. The factors to be considered are, (1) type of agitator (2) circulation pattern (3) location of the agitator in the basic equipment (4) shape and size of tank (5) diameter and width of agitator (6) method of baffling (7) power required for agitation (8) shaft overhang and (9) type of stuffing box or seal, bearings, drive system etc. Some of the factors are considered below in detail.

14.2 Types of Agitators

Mechanical agitators can be divided into seven basic groups namely paddles, turbines, propellers, helical screws, cones, radial propellers and high speed discs. Mixing by agitators takes place due to momentum transfer. High velocity streams produced by the impeller entrain the slower mixing or stagnant liquid areas in all parts of the vessel and uniform mixing occurs. As the viscosity of the liquid increases, frictional drag forces retard the high velocity streams and confine them to the

immediate vicinity of the impeller. Thus stagnant areas develop and uniform mixing is not achieved.

Agitators having a small blade area which rotate at high speeds such as propellers, flat and curved blade turbines, are used to mix liquids having low and medium viscosities, upto 1000 and 50,000 cp respectively. Agitators having a large blade area which rotate at slow speeds, such as anchors, gates and helical screws are more effective for mixing high viscosity liquids.

14.2.1 Paddle Agitator

Fig. 14.1 represents some paddle agitators. The blades of these agitators normally extend close to the tank wall. They

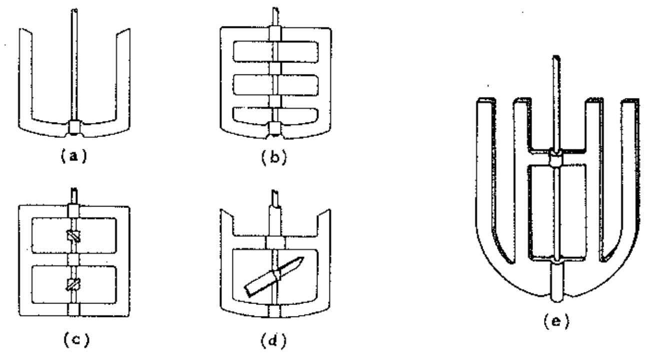


Fig. 14.1 Paddle agitators
(a) anchor (b) gate (c) gate with pitched cross arms (d) anchor with pitched cross arms (e) Combined anchor and gate

are simply pushers and cause the mass to rotate in a laminar swirling motion with practically no radial flow along the paddle blades or any axial flow. The circulation is poor and the mixing action is inefficient. The speed of rotation is very slow, and is generally between 80 to 150 metres per minute. Highly viscous liquids and pastes are agitated by multiple blade paddles. Such paddles are made to rotate between intermeshing stationary fingers. A counter rotating pair of multiple paddles creates a high localised shear. The width of the blades is 1/4th to 1/10th of the paddle diameter. The most common paddle diameter is 0.8 times the tank diameter.

14.2.2. Turbine Agitators

These have a variety of shapes such as radial, pitched and back-sloped (Fig. 14.2). They are capable of creating a

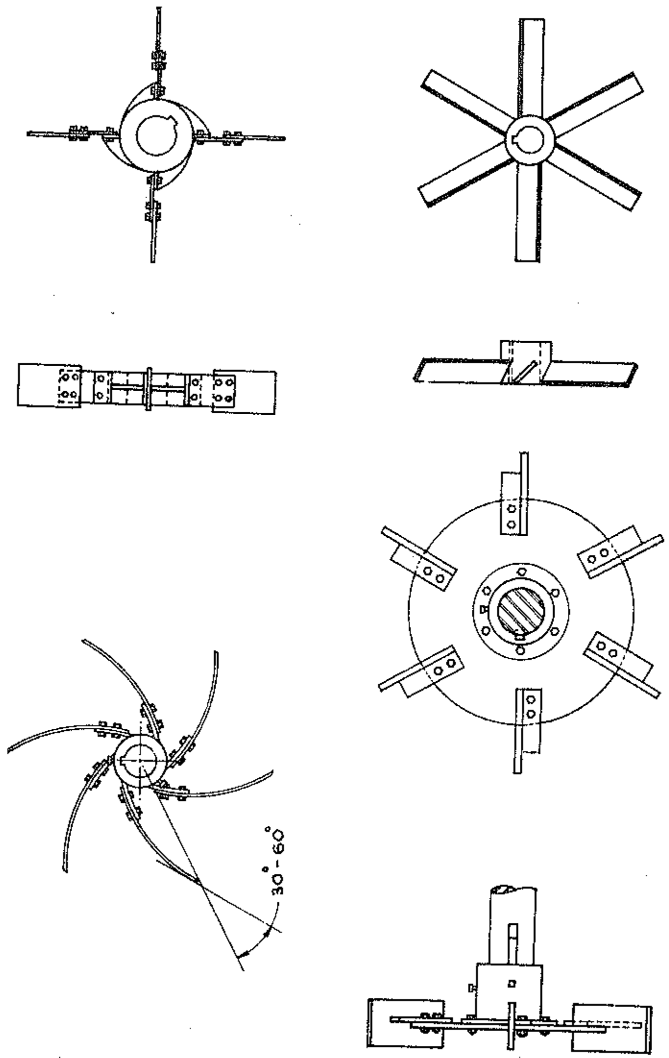


Fig. 14.2 Turbine agitators
(a) straight flat blade (b) pitched blade (c) curved back-sloped
(d) straight blades attached to disc

vigorous mixing action, due to centrifugal and rotational motion, generated by them. The back-sloped turbine agitator offers maximum power economy. A stator ring surrounding this agitator gives an efficient mixing action. A radial bladed turbine gives a higher discharge velocity, but requires more power. The more commonly used type is the pitched blade turbine Fig. 14.2 (b) which gives both radial and axial flow. The axial flow in this type of agitator is the lesser of the two flows and varies with the number of blades, the pitch angle and the blade height. The approximate peripheral speed for the turbine agitator is 200 to 250 metres per minute. All the three types of turbine agitators can be constructed either with blades directly attached to a hub or blades attached to a centre disc [Fig. 14.2 (d)]. Generally the diameter of the agitator is kept between a third and a sixth of the tank diameter depending on the circulation required. The blade length is a fourth of the agitator diameter. With central disc it is 1/8th the agitator diameter. The blade angle of a curved blade turbine agitator may vary between 30° and 60°. In general the higher the viscosity, the greater must be the blade angle to keep the power requirements to a minimum. The number of turbine agitators to be used is determined by the following equation :

$$\text{No. of agitators} = \frac{\text{maximum liquid height} \times \text{average sp.gr}}{\text{tank diameter}}$$

The distance between the agitators should be 1 to 1.5 agitator diameters.

It is common to locate the agitator, at a height not less than one agitator diameter length from the bottom and the agitator should be submerged with liquid by a depth equal to twice the agitator diameter at low speeds and four times at high speeds. If the depth of liquid is more than twice the agitator diameter, it may be advisable to use two agitators.

14.2.3. Prupeller Agitator

A propeller agitator is shaped with a tapering blade to minimize the effect of centrifugal force and produce a maximum axial flow (Fig. 14.3). It may be mounted centrally, off-centre or at an angle to the tank. It is simple and portable. The diameter of this agitator is usually between 15 and 30% of the tank diameter and its peripheral speed is generally between 300 and 500 metres per minute. It can therefore be directly

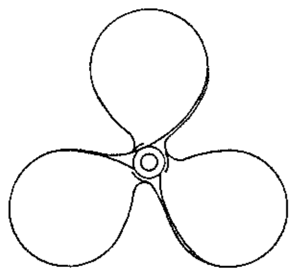


Fig. 14.3 Propeller agitator

coupled to a standard electric motor. It is considered as the most economic unit for simpler mixing jobs, particularly in small tanks.

14.2.4 Helical Screw Agitator

It is in the shape of a screw as shown in Fig. 14.4. It is an effective device for mixing high viscosity liquids. The screw drives the liquid from the vessel bottom to the top surface of the liquid. Alternatively, screws may be operated in the reverse direction to drive the liquid to the bottom of the vessel.

Baffles in a screw agitated vessel help to create turbulance. These should be set away from the vessel wall, thus allowing the turbulance created to travel around the baffle and entrain the otherwise slowly moving liquid in contact with the vessel wall. Sometimes the screw is placed off centre in which case baffles may be avoided. However this arrangement requires more power.

14.2.5 Cone Type Agitator

This agitator is built with shallow vertical vanes on the inside periphery (Fig. 14.5). Flow is generated between the peripheries of the narrow and wide throats of the cone. It is specially used for handling fibrous and dense slurries. Its speed is similar to that of the turbine.

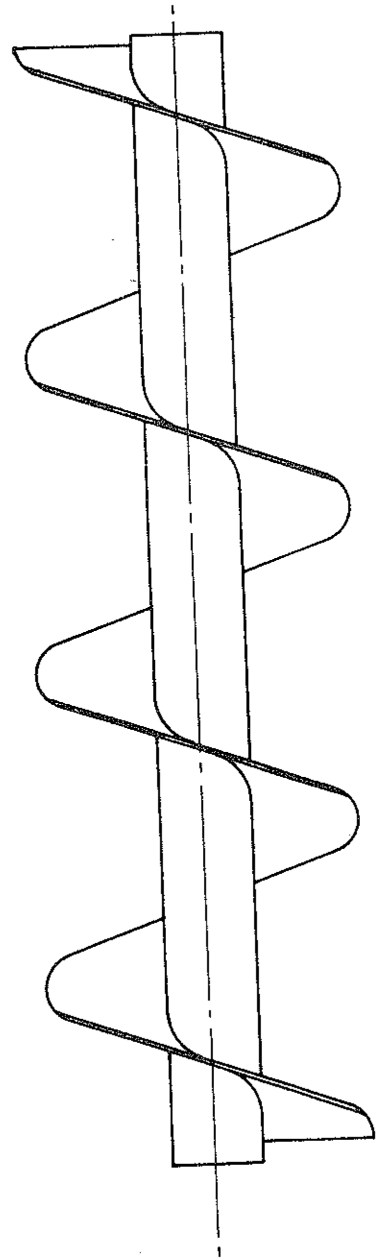


Fig. 14.4 Helical screw agitator

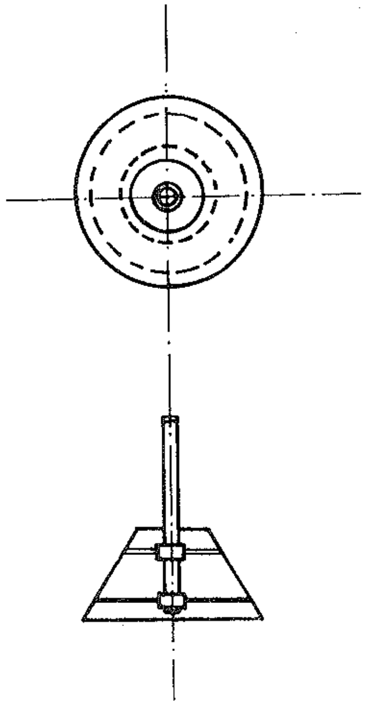


Fig. 14.5 Cone type agitator

14.2.6 Radial Propeller Agitator

It has blades fixed at the end of a rotating arm (Fig. 14.6). The blades are pitched to the direction of rotation. By varying the height, width, and particularly the angle of pitch, turbulance or shear can be controlled.

14.2.7 High Speed Disc Agitator

This has a disc with corrugations (Fig. 14.7). The speed of the agitator provides sufficient centrifugal force through surface friction to generate a flow. It produces high shear and helps disintegration of low density fibrous solids.

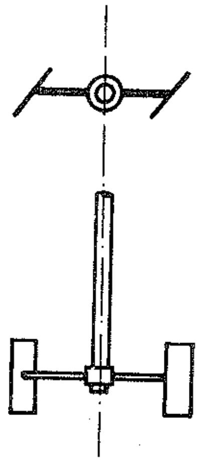


Fig. 14.6 Radial propeller agitator

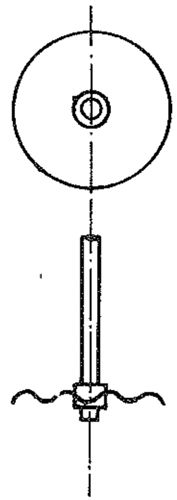


Fig. 14.7 High speed disc agitator

14.3 Baffling

Baffling is essential for efficient mixing action. If a simple swirling motion is required no baffling is necessary. The most

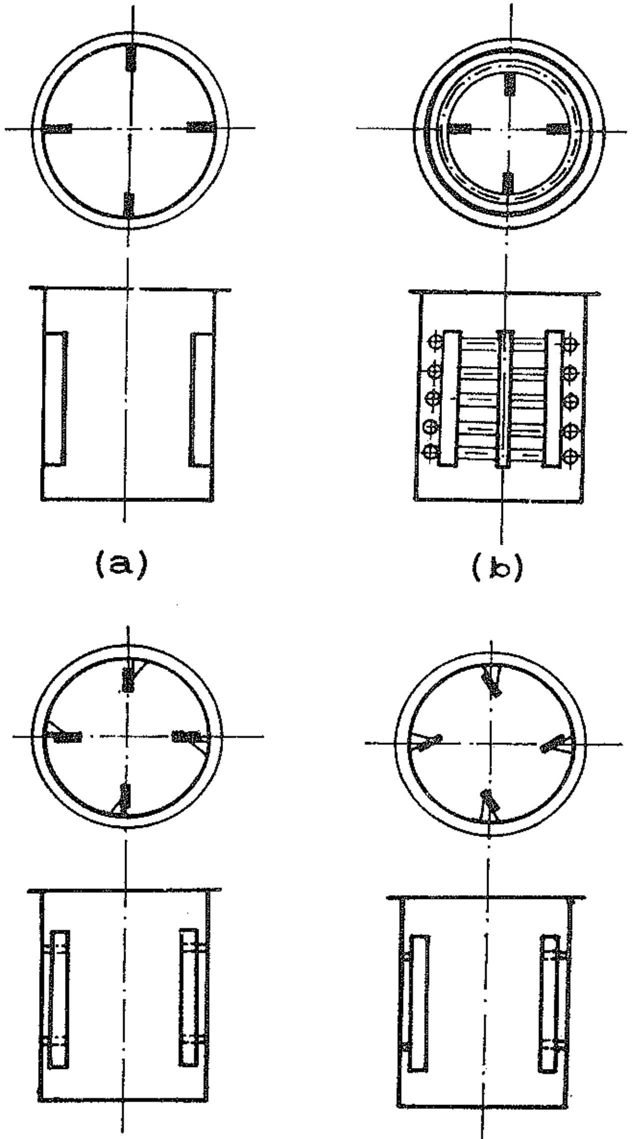


Fig. 14.8 Baffle arrangements
(a) normal baffle (b) baffle with coil (c) outset baffle
(d) angled baffle

perfect baffling is the curved deflecting ring. However, it is limited to turbine type impellers, and acts as an impedance to free discharge. The common practice is to use baffles attached to tank walls (Fig. 14.8). Their size and disposition are as follows :

(1) Four baffles should be mounted vertically on the tank wall, projecting radially from the wall and located 90° apart.
(2) Baffle width should be a tenth to a twelfth of the tank diameter.
(3) Baffle height should be at least two impeller diameters and approximately centered on the agitator.
(4) For baffles set out from the wall, the width may be 1/12th of the tank diameter with an offset of approximately 1/5th the baffle width.
(5) With coils in the tank, the baffles should be placed inside the coil.

For large diameter tanks (6 metres and above) the number of baffles may be increased to six and they may be set out from the wall to reduce the distance from the agitator to the baffle. For high viscosity liquids viscosity in the region of 200 poise angled baffles are preferable. For viscosities above 600 poise, the baffles may be eliminated completely as the viscous drag becomes sufficient to control swirl and assure vertical circulation.

A chart indicating the selection of agitators for different services is given in Table 14.1.

14.4 Power Requirements for Agitation

14.4.1 Power Absorbed by the Agitator

Power required to operate an agitator depends on several factors such as the properties of the liquid, height of the liquid, agitator type and size, the tank or vessel size and speed of agitation. No single equation is, therefore, capable of determining the power requirement of different agitators operating under various conditions. A convenient method accepted in many cases is the use of what are known as power curves. An individual power curve is only true for a particular geometrical configuration, but is independent of the vessel size. Numerous power curves are available, representing a wide variety of geometries.

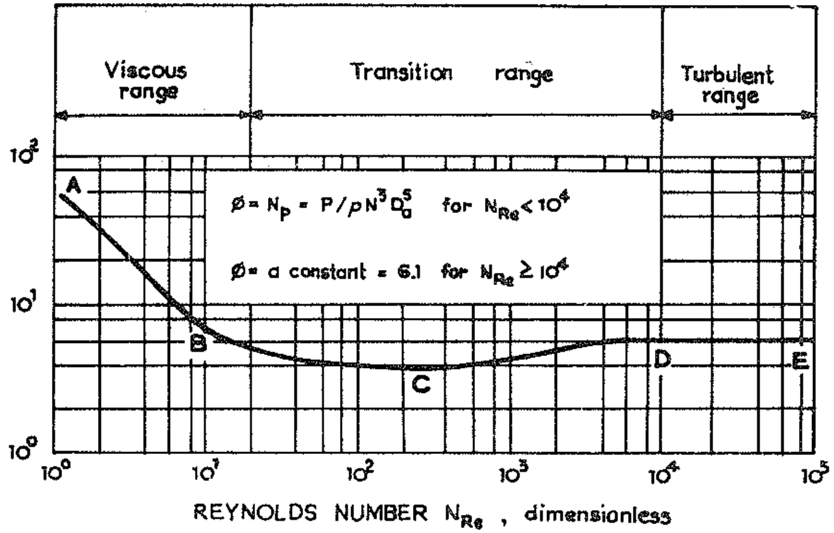


Fig. 14.9 Power curve for standard tank configuration.

Fig. 14.9 shows the power curve for a standard tank configuration, which consists of a flat bottom cylindrical tank, with 6 flat blade turbine agitator and filled with liquid upto a height equal to the tank diameter. The tank is also provided with four baffles. As may be seen, the curve is a log-log graph of the dimensionless Power function ϕ and dimensionless Reynolds number N_{Re}, where

$$\phi = N_p = \frac{Pg_c}{\rho N^3 D_a^5} \tag{14.1}$$

and

$$N_{Re} = \frac{\rho N D_a^2}{\mu} \tag{14.2}$$

where N_p —Power number

P —Power in kg metres

g_c —Gravitational acceleration in metre/sec²

ρ —Density of liquid in kg/m²

N —Speed of agitator in revolutions per second

D_a—Agitator diameter in metres

μ —Viscosity in kg/m sec

In an unbaffled vessel with similar tank configuration the mixing action is different and the relation between power function ϕ and Reynolds number N_{Re} is as shown in Fig. 14.10, where

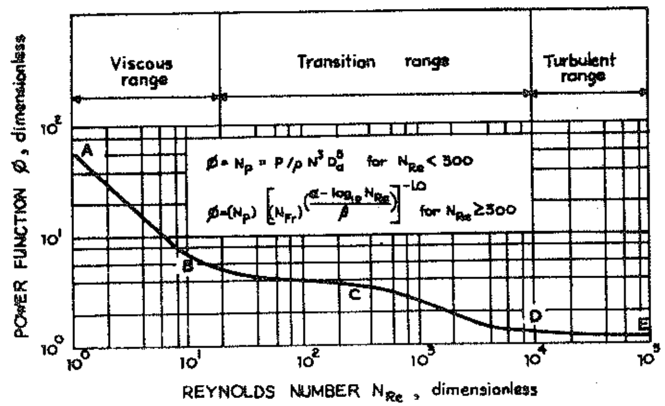


Fig. 14.10 Power curve for unbaffled vessel

$$\phi = N_p = \frac{pg_c}{\rho N^3 D_a^5} \text{ for } N_{Re} < 300 \tag{14.3}$$

$$\text{and } \phi = N_r \frac{(\alpha - \log_{10} N_{Re})}{N_{Fr}\,\beta} \text{ for } N_{Re} > 300 \tag{14.4}$$

where N_{Fr} is known as Froude number and is given by

$$N_{Fr} = \frac{N^2 D_a}{g} \tag{14.5}$$

Values of α and β are given in Table 14.2.

The above curves are a particular tank configuration, it may be used to calculate power for various agitator speeds, liquid viscosities and densities. Similar curves are available for other configurations.

Table 14.2

6 Blade, Flat Blade Turbine Agitator

Diameter (D_a) cm	D_a/D	α	β
10	0.3	1.0	40.0
15	0.33	1.0	40.0

14.4.2 Power Absorbed by Vessel Fittings

Fittings such as dip pipes, thermowells as also coils will cause a power loss. This loss depends on the position of the fittings. Fittings close to the shaft make no appreciable difference to the power consumption. Fittings close to the tank wall absorb more power.

14.4.3 Transmission and Gland Losses

The power loss in glands varies from 0.5 hp for smaller agitator shafts, upto 5 hp for larger shafts. No reliable method for calculating this loss can be given. As a very rough approximation, however the gland loss may be taken as 10% of the agitator power consumption or 0.5 hp whichever is greater. It is usual to allow 20% of the maximum input rating as the gear box and V-belt drive loss.

14.4.4 Horse Power of Driving Motor

The required hp output of electric motor is obtained by the addition of the above power consumption factors. Due allowance must be made for the possible occurrence of extra heavy loads during start-up periods or during some stage of the process. Some empirical allowances may be made. Losses in the motor at various loads can be determined from output efficiency curves.

14.5 Design of Agitation System Components

In order to rotate the agitator at the required speed, it is attached to a shaft. The driving system usually consists of the

shaft, coupling, bearings, gearing, pulleys and belts. The power is supplied by an electric motor, a hydraulic motor or a steam turbine. In the case of toxic, flammable or volatile materials or liquids under pressure, special provision has to be made to prevent leakage, between the shaft and the portion of the vessel surrounding the shaft. To allow free rotation of the shaft a clearance must be maintained between these adjacent parts. A stuffing box or a seal is used for preventing leakage through this clearance. Figs. 14.11 and 14.16 indicate the arrangement

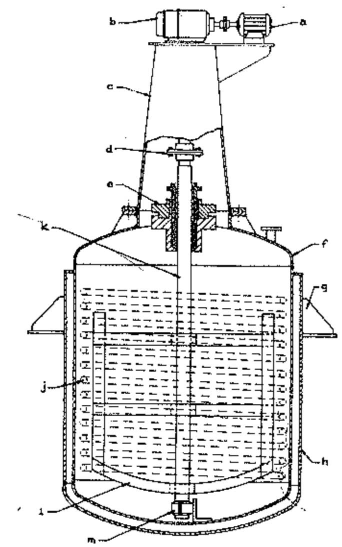


Fig. 14.11 Typical drive system
(a) motor (b) gear box (c) support bracket (d) coupling
(e) stuffing box (f) vessel (g) bracket (h) jacket (j) shaft
(l) anchor agitator (m) steady bearing

and components of drive system, along with an agitator as provided for a pressure vessel. Details of some of the components and the design features are considered below.

14.5.1 Shaft Design

The shaft is attached to the agitator and may be located in horizontal, vertical or angular position. It is preferable to support the shaft by use of bearings outside the vessel or equipment. In special cases steady bearings may be used, which may be located inside the equipment, such as bearings placed at the bottom of a vessel. A stabilizer ring just below the agitator may be used to reduce vibrations and deflection of the shaft.

The shaft design is based on criteria indicated in detail in Section 5.2.

14.5.1.1 DESIGN BASED ON TORQUE AND BENDING MOMENT

The normal power required for the agitator and frictional losses are indicated under 'Power Requirement for Agitation'. (Section 14.4). From this it is possible to determine the continuous average rated torque on the agitator shaft.

$$T_c = \frac{\text{hp} \times 75 \times 60}{2\pi \times N} \tag{14.8}$$

where N — rpm

During start up the shaft has to withstand higher torque. During running, apart from the torque, various forces acting on the shaft have to be taken into account. These are transient unbalanced hydraulic forces due to turbulance in the liquid or asymmetrical construction of the agitator and baffles, acting laterally on the shaft in a cyclic manner. Centrifugal forces are also present while the shaft is rotating and the agitator is out of balance. Further there is the possibility of the agitator being checked when the material is added to the vessels by tipping bags or other containers. The worst cumulative conditions are assumed to be equivalent to those in which the agitator blade is jammed at a point 75% of its length from the shaft.

During starting the shaft must be capable of resisting $1\frac{1}{2}$ times the continuous average torque (T_c) at low speeds and $2\frac{1}{2}$ times at high speeds. The shaft is, therefore, designed for either of these maximum values. The maximum shear stress developed is given by

$$f_s = \frac{(1\frac{1}{2} \text{ or } 2\frac{1}{2}) \times T_c}{Z_p} = \frac{T_m}{Z_p} \tag{14.9}$$

where T_m = max. torque
f_s = shear stress
Z_p = polar modulus of section of the shaft cross-section

If the permissible stress value of f_s is known, the shaft diameter can be determined. Another criterion of design is based on fluctuating loads during running. Consider the agitator blade jammed at 75% of its length for a short period. In this case it is necessary to work out the equivalent bending moment which is the resultant of the bending moment and maximum torque (Ref. equation 5.2).

$$M_e = \tfrac{1}{2}[M + \sqrt{M^2 + (T_m)^2}] \tag{14.10}$$

where T_m — (1.5 or 2.5) T_c
M — bending moment

The bending moment M is determined as follows :

The torque T_m is resisted by a force F_m acting at a radius of 0.75 R_b from the axis of the agitator shaft.

$$F_m = \frac{T_m}{0.75\ R_b} \tag{14.11}$$

where R_b — radius of blade.

The maximum bending moment M occurs at a point near the bearing, from which the shaft overhangs

$$M = F_m \times l \tag{14.12}$$

where l—shaft length between agitator and bearing.

The stress due to equivalent bending moment is given by

$$f = \frac{M_e}{Z} \tag{14.13}$$

where Z—modulus of section of the shaft cross-section.

The stress f should not exceed the yield stress of the material or 0.2% proof stress. It is assumed that the force F_m acts as a transient force and hence the design is based on yield stress. From equations 14.9 and 14.13 the shaft diameter may be selected. The higher value is usually chosen.

14.5.1.2 DESIGN BASED ON CRITICAL SPEED

It is difficult to assess the unbalanced hydraulic forces or the unbalanced forces due to asymmetric construction of the agitator. The latter forces can be eliminated to a considerable extent if the agitator is statically and dynamically balanced. This is usually effected by fixing certain counter balance weights in the opposite direction to the unbalanced forces. As the shaft rotates, centrifugal forces are created. If the speed of the shaft varies, it is not possible to balance these dynamic forces completely. Only a partial balance may be possible.

It is necessary to control the deflection of the shaft by adequate supports. If the shaft is overhung it may be necessary to increase the shaft diameter for the cantilever portion and reduce deflection. Higher deflection increases eccentricity and therefore centrifugal forces. These forces change direction as the shaft rotates, and cause vibrations. The speed at which the shaft vibrates violently is known as critical speed. It is recommended that the range of speed between 70% and 130% of the critical speed should be avoided. The shaft diameter should be so chosen that its normal working speed range does not fall within the critical speed range. Low speed agitators such as paddles and turbines normally operate between 50% and 65% of their critical speed. High speed impellers, such as propellers and disks, normally operate above the critical speed. Determination of critical speeds for shafts is done as follows :

Fig. 14.12 shows a horizontal shaft, supported at two bearings A and B. The loads or forces w_1, w_2, w_3, are acting perpendicular to the shaft. The deflection due to each load acting independently is given by

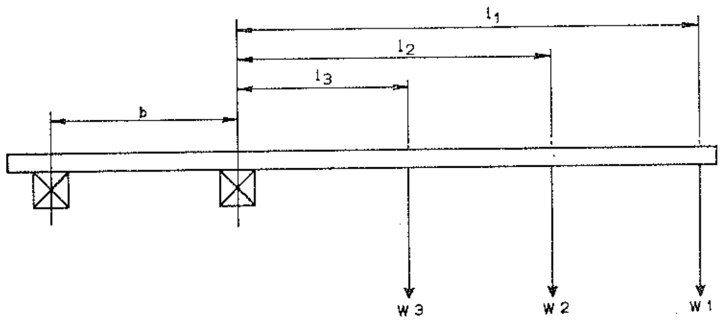


Fig. 14.12 Shaft supported in bearings

$$\delta = \frac{Wl^3}{3EI} \tag{14.14}$$

Similarly the maximum deflection due to shaft weight is given by

$$\delta_s = \frac{wl^4}{8EI} \tag{14.15}$$

where l—appropriate length

W—concentrated load

w—uniformly distributed load per unit length

E—modulus of elasticity

I—moment of inertia of the cross-section of shaft.

The critical speed can be determined as follows :

$$N_c = \frac{60 \times 4.987}{\left(\delta_1 + \delta_2 + \delta_3 + \frac{\delta_s}{1.27}\right)^2} \tag{14.16}$$

N_c = rpm

where δ_1, δ_2, δ_3 and δ_s are the deflections in cm due to each load.

In the case of vertical shafts only unbalanced forces on the impeller may be considered for evaluating critical speed.

14.5.2 Agitator

The main parts of an agitator are the hub and the blades. The hub is attached to the shaft by the use of keys or bolts. Agitators may either be cast as one piece or in two pieces split and bolted at the hub. Cast agitators have two basic advantages, namely uniformity of material and a hard surface. Agitators are also fabricated from metal sheets by welding. Such agitators are liable to corrode, particularly at the welded areas, if precautions are not taken. It is desirable to restrict welding to areas near the hub, where the flow velocity is minimum. Agitators of bolted construction, with blades bolted at the hub (Fig. 14.2) are also fabricated. These are liable to corrosion and failure of exposed threads.

Wooden agitators are useful only for relatively low speeds. They are difficult to balance and tend, not to remain in balance due to working and absorption of moisture. Simple paddles or gates can be made satisfactorily. Wood is used successfully in many aqueous processes where no suitable metals are available or where the size of the equipment is large and the cost of other materials would be excessive. Agitators fabricated from solid lead are suitable at low speeds. Homogeneous lead covering is of wider application. Erosion or creep can cause rapid failure. Enamelled cast iron or enamelled mild steel agitators can be made in a variety of forms, but the outer contours must be smoothly rounded and the sections must be uniform as far as possible. Agitators can also be covered with rubber which offers high resistance to erosion.

Enamel, rubber or lead lined agitators are likely to be troublesome where high intensity agitation is required. In such cases it is economical to use special alloys for impellers and shafts.

As indicated in 14.5.1.1 the load on the blade is assumed to act at 75% of the agitator radius. This will create a bending moment, which will be maximum at the point where the blade is attached to the hub.

$$\text{Max. BM} = F\,(0.75\,R_b - R_h) \tag{14.6}$$

where F—force on each blade

R_b—radius of blade

R_h—radius of hub

If the blade is flat, the stress in the blade will be given by

$$f = \frac{\text{Max BM}}{Z} = \frac{F\,(0.75\,R_b - R_h)}{\dfrac{b_t \times b_w^2}{6}} \tag{14.7}$$

where, b_w—blade width

b_t—blade thickness

Since the blades are subject to high frequency oscillation, due to turbulance in the liquid, they are liable to fatigue. The permissible values of the stress should never exceed the endurance limit for the blade material.

The hub is fixed to the shaft by a key, or setscrew, which transmits the shaft torque to the impeller. The hub is subjected to bending moments due to the forces on the blades and to shear forces due to torque. It is usual to assume the outside hub diameter as twice the shaft diameter and check the shear stress due to torque. The detailed design of key and hub is indicated under sections 5.3 and 5.4.

14.5.3 Couplings

For connecting the agitator shaft to the drive shaft three types of couplings are generally used. These are flange coupling, both rigid and flexible, split muff coupling and clamp coupling. They must be capable of transmitting torque, sustain the end thrust due to the weight of the agitator and should withstand the bending moments impressed upon by the unbalanced hydraulic and centrifugal forces. For small agitators with shafts upto 4 cm diameter and 1.25 m length, a sleeve coupling may be used. The details of construction and design are indicated in section 5.4.

14.5.4 Bearings

It is common to use bearings as far as possible outside the equipment [Fig. 14.16 (h)] in which agitation is carried out. However, intermediate bearings and bearings at the free end of the shaft may also be used. In deep tanks with long agitator shafts, excessive vibrations can be reduced by using an auxiliary or steady bearing at the bottom of the shaft. [Fig. 14.11 (m)]. This should be mounted on a support fastened to the bottom or side of the tank. The mounting should allow the shaft to be adjusted to align the agitator shaft. Serving a steady bearing involves the inconvenience of shutting down the equipment. These bearings also create problems of alignment, lubrication, corrosion and maintenance. Whenever possible the use of such bearings should be avoided. If it is absolutely necessary to use these types of bearings, replacement sleeves for shaft and bearing surfaces are advantageous. Bearings outside the vessel are nearly always of the ball or roller type. However plain bearings with bush may be satisfactory for lower shaft speeds, although difficult to instal. Bushes may be made of cast iron, phenolic material, phosphor bronze, rubber, etc. Grease lubrication is normal and protection against entry of the condensate, dust or fumes from the vessel should be provided. Details of bearings design are given under section 5.5.

14.5.5 STABILIZERS

As mentioned earlier the critical speed range for the rotation of shaft should be avoided. However, with variable speed agitators, it may be necessary to rotate the shaft within this range. The unbalanced forces may therefore cause the shaft to vibrate vigorously. To damp such vibrations, a stabilizer is sometimes used. This is essentially in the form of a ring (Fig. 14.13) installed at the bottom end of the cantilever shaft, below the agitator. When the transient unbalanced hydraulic

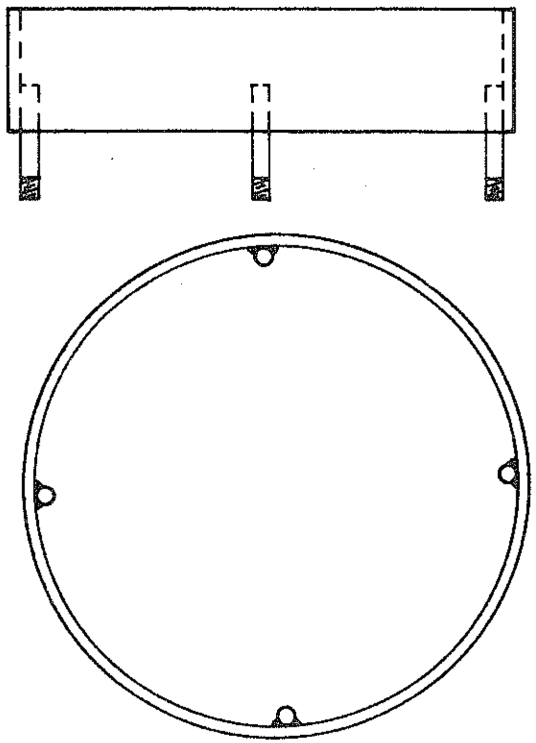


Fig. 14.13 Stabilizer ring

force pushes the agitator away from the axis of rotation the movement creates a counter balancing drag force on the stabilizer. This drag force provides damping and reduces vibrations and stresses.

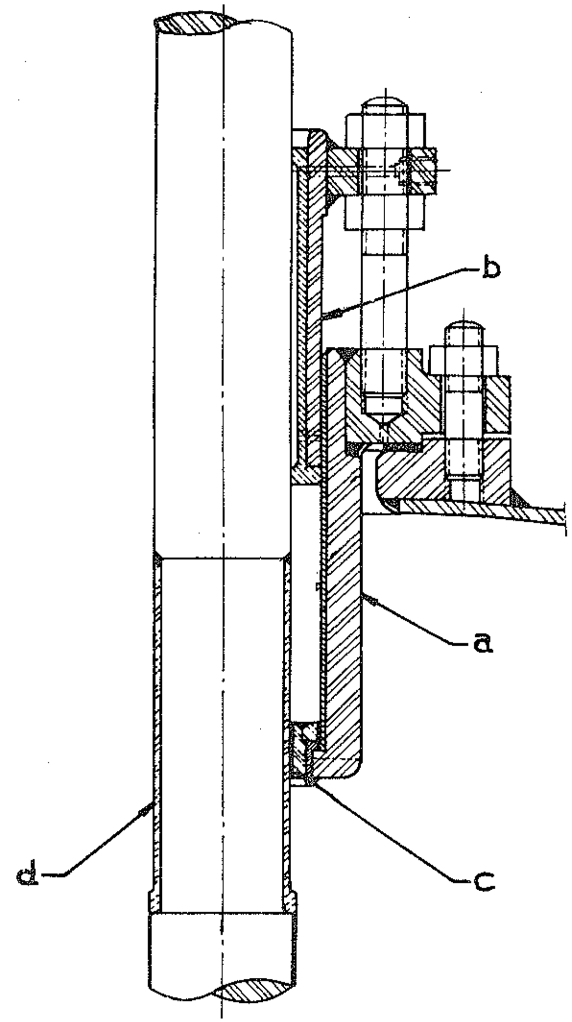


Fig. 14.14 Stuffing box and gland.
(a) stuffing box (b) gland with sleeve (c) bush (d) shaft lining

14.5.6 Shaft Seals

In the case of toxic or flammable materials, and during the evaporation of volatile materials, or when the process pressure is different from the atmospheric pressure, the agitator shaft must be sealed to isolate the contents of the equipment, from the atmospheric surroundings and prevent outflow of the fluid from the equipment on inflow of foreign material into the equipment. The most common methods for sealing are stuffing boxes and mechanical seals. Details regarding both the methods are indicated in sections 5.12, 5.13, and 5.14.

A stuffing box may be located partially within the vessel (Figs. 14.14, 14.15 and 14.16) or outside the vessel. In either case it is fixed to a pad welded to the top head of the reaction

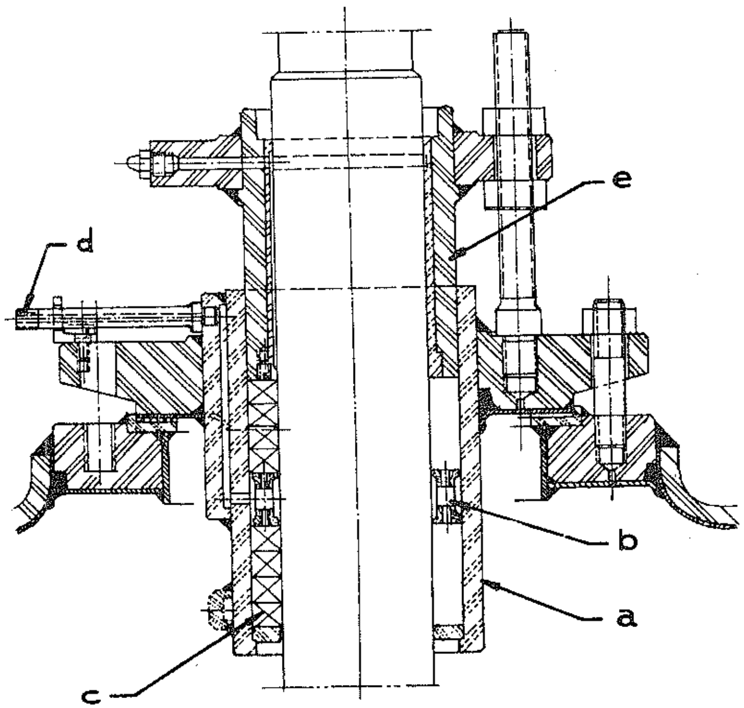


Fig. 14.15 Lubricated stuffing box and gland.
(a) stuffing box (b) lantern ring (c) packing (d) inlet of lubricating fluid (e) gland with sleeve

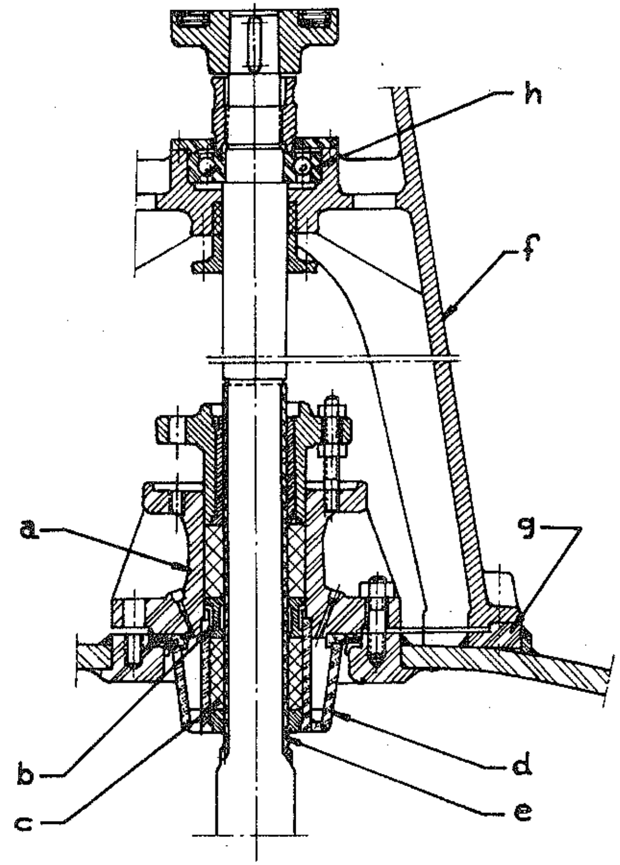


Fig. 14.16 Lubricated and jacketed stuffing box.
(a) stuffing box (b) lantern ring (c) packing (d) cooling jacket (e) shaft lining (f) cast supporting from bearing (g) pad for fixing the frame to head (h) bearing with housing

vessel. Fig. 14.14 shows a bush in the base of the stuffing box. This may be necessary when shafts are likely to deflect or where a high standard of performance is expected of the gland. Cast iron, plastic impregnated asbestos or cotton bushes are very often satisfactory where the vessel and the agitator are lined with lead, enamel or rubber. For severe duties, as may be seen in the figure, it is possible to provide a shoulder on the shaft with a wearing sleeve of resistant material upto the shoulder and through the gland. These sleeves may be of hastelloy, high silicon iron, phosphor bronze, etc. The gland may be constructed out of steel with a sleeve of soft material. Figs. 14.15 and 14.16 show a method of lubricating the packing, through a lantern ring located in the centre of the stuffing box. It is supplied with a lubricating fluid. A heavy duty stuffing box is shown in Fig. 14.17. It is

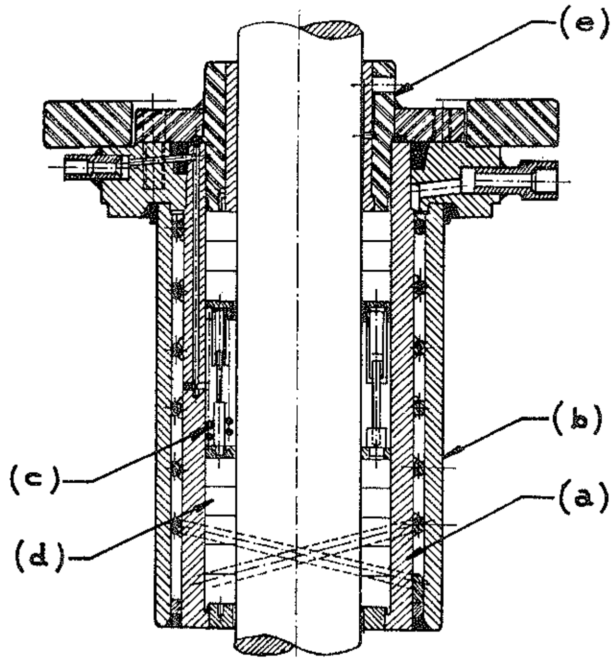


Fig. 14.17 Jacketed stuffing box
(a) stuffing box (b) jacket (c) spring (d) packing (e) gland with sleeve

provided with a cooling jacket, supplied with a circulating liquid. A few springs are also placed in the centre of the stuffing box. These springs exert a pressure on two rings placed on either side. The pressure is transferred to the

packings, which help to create a uniform pressure within the stuffing box thus reducing the leakage from the gland.

14.6 Drive for Agitators

Agitator drive system may consist of a motor, a belt drive and/or a gear drive. Usually a V-belt is preferred due to its simplicity and the short centre distance between motor pulley and shaft pulley. It is satisfactory upto a speed reduction ratio of 5 to 1. Spur, helical, bevel and worm gearing may be used according to convenience and shaft arrangement. Helical gear teeth have greater strength and smoother engagement. The transfer of power is also gradual. The teeth are also subjected to less shock. Helical gear trains are efficient in power transmission and are preferred to other types of gears. Worm gearing is suitable for transmission of power between perpendicular shafts. With this gearing it is possible to obtain high velocity ratios in minimum space but usually at the sacrifice of efficiency. Gear boxes are expensive.

14.6.1 Vertical Drives

In vertical agitator applications the motor may be placed in a vertical position or in a horizontal position. In most applications a reduction in speed is desired, since the agitators rotate at speeds lesser than the speed of the motor. This involves a single, double or triple reduction, depending on the required agitator speed. High speed agitators such as propellers require only one set of reduction gears. Low speed agitators such as paddles require two or three sets of reduction gears. The vertically mounted motor can be directly coupled to the input shaft of an in-line vertical reducer and the output shaft of the reducer can be directly coupled to the agitator shaft. In the case of a horizontal motor it is necessary to provide for the transition from the horizontal to vertical position of the agitator shaft. This can be achieved with either a worm or a bevel gearing, along with a belt drive or a gear train.

14.6.2 Side Entering Agitators

In this arrangement the agitator shaft is horizontal. The motor may be either directly coupled to the agitator shaft (Fig. 14.18.) or through a single reduction gear unit or a V-belt arrangement. The reduction is between the parallel shaft of the agitator and that of the motor. In special cases a bevel or spiral bevel gear may be used on the agitator shaft with the input pinion mounted vertically along with the motor.

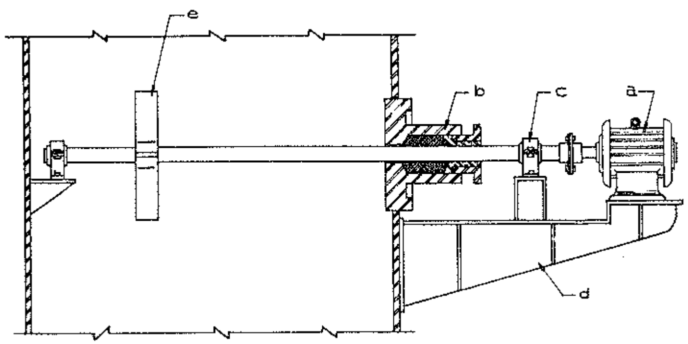


Fig. 14.18 Side entering agitator shaft.
(a) motor (b) stuffing box and gland (c) bearing
(d) support bracket (e) agitator

14.6.3 Special Drives

Instead of using an electric motor, a hydraulic drive system may be used, particularly where high torque at low speeds is required. The hydraulic drive motor can be directly mounted to a vertical shaft, eliminating gear boxes. A wide range of speeds is feasible. However this needs additional equipment such as pumps, valves, etc.

14.6.4 Mountings

The drive system assembly must be located at the top of the vessel with preferably an independent supporting structure. In the case of an open tank, the assembly can be supported

between a support bracket and a horizontal girder placed on top of the tank. Alternately it can be supported between the brackets, over which girders can be placed.

In vessels with heads, it is necessary to consider the rigidity of the support to eliminate vibration or flexure detrimental to stuffing box operating and packing life, and also the effect of load on the head thickness.

One method of avoiding any load on the head is to provide a beam support, the ends of the beams can be extended down the vessel shell to distribute the load over a wider area. In the case of low drive assembly load, it may be mounted on pads fixed to the head with a stool support [Fig. 14.11 (g), 14.16 (f) and (g) and 14.19]. In such cases and with full vacuum in the vessel additional agitator drive load must be given special consideration.

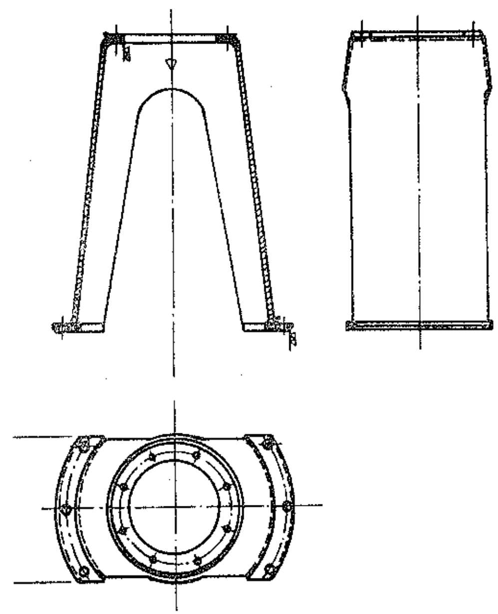


Fig. 14.19 Stool for supporting drive system (Welded construction)

The effect of eccentric load on the mounting arrangement may cause stresses in the joint. This is particularly relevant with horizontal motors. Special supports or brackets must be provided beneath the centre of the drive assembly in such cases.

Side entering agitators must be mounted so that the drive assembly has the least possible overhang. The unit has to be supported by tie-rods or brackets of sufficient strength (Fig. 14.18). It is advisable to support outboard machinery with any structure anchored to the ground, since slight shifts in the position of process vessel lead to incorrect alignment between the shaft and the seal. This results in excessive shaft vibration, wear and leakage of stuffing box or seal.

14.7 Numerical Problem

Data: Turbine agitator operating in a vessel of 1500 mm diameter.

Internal pressure in vessel	5 kg/cm^2
Diameter of agitator	500 mm
Speed (maximum)	200 rpm
Liquid in vessel (i) sp. gravity	1.2
(ii) viscosity	600 cp
Overhang of agitator shaft between bearing and agitator	1300 mm
Agitator blades (flat) Nos.	6
Width of blade	75 mm
Thickness of blade	8 mm
Baffles at tank wall Nos.	4
Shaft material—commercial cold rolled steel	
Permissible shear stress in shaft	550 kg/cm^2
Elastic limit in tension	1460 kg/cm^2
Modulus of elasticity	19.5×10^5 kg/cm^2
Permissible stresses for key (carbon steel)	
Shear	650 kg/cm^2
Crushing	1300 kg/cm^2
Stuffing box—(carbon steel)	
Permissible stress	950 kg/cm^2

Studs and bolts (Hot rolled carbon steel)	
Permissible stress	587 kg/cm^2

It is assumed that the vessel geometry conforms to the standard tank configuration. The power curve referred to is therefore as in Fig. 14.9.

From equation 14.2

$$\frac{\rho N D_a^2}{\mu}=\frac{1.2\times10^3\times\frac{200}{60}\times\frac{(500)^2}{(1000)}}{600\times10^{-3}}$$

$$=1670$$

From power curve 14.19

$$N_p=4.5$$

From equation 14.1

$$P=\frac{N_p\,\rho\,N^3D_a^5}{g_c\times75}=\frac{4.5\times1.2\times10^3\times\frac{(200)^3}{(60)}\frac{(500)^5}{(1000)}}{9.81\times75}$$

$$=8\ 5\text{ hp}$$

Gland losses 10%—0.85 hp

Power input$=8.5+0.85=9.35$ hp

Transmission system losses 20%$=9.35\times0.2=1.87$ hp

Total hp$=9.35+1.87=11.22$

This will be taken as 12.5 to allow for fitting losses. It is advisable to use 15 hp motor.

14.7.1 Shaft Design

For shaft design assume power required as 12.5 hp. Use solid shaft diameter (d)

From equation 14.8

$$T_c=\frac{12.5\times75\times60}{2\pi\times200}=44.8\text{ kg m}$$

$$T_m=1.5\times44.8=67.3\text{ kg m}$$

From equation 14.9

$$Z_p=\frac{1.5\times T_c}{f_s}=\frac{67.3\times100}{550}$$

$$=12.2\text{ cm}^3$$

$$\frac{\pi d^3}{16}=12.2$$

$d^3=62.2$, $d=3.95$ cm $\simeq 4$ cm

From 14.11

$$F_m=\frac{1.5\times 44.8\times 100}{0.75\times 25}=359 \text{ kg}$$

$$M=F_m\times l$$

$$=359\times 1.3=467 \text{ kg m}$$

From equation 14.10

$$M_e=\frac{1}{2}\left[467+\sqrt{467^2+67.3^2}\right]$$

$$=469 \text{ kg m}$$

From equation 14.13

$$f=\frac{469\times 100}{\frac{\pi\times(4)^3}{32}}=\frac{540\times 100\times 32}{\pi\times 64}$$

$$=7470 \text{ kg/cm}^2$$

Stress f is higher than the permissible elastic limit (2460 kg/cm^2) Therefore, use a 6 cm diameter shaft for which the stress will be

$$f=2210 \text{ kg/cm}^2$$

Deflection of shaft — From equation 14.14

$$\delta=\frac{wl^3}{3\,EI}=\frac{F_m\times(130)^3}{3\times 19.5\times 10^5\times\frac{w\times(6)^4}{64}}$$

$$=\frac{359\times(150)^3\times 64}{3\times 19.5\times\pi(6.0)^4}$$

$$=1.38 \text{ cm}$$

From equation 14.16

$$\text{Critical speed}\quad N_c=\frac{60\times 4.987}{\sqrt{\delta}}$$

$$=\frac{60\times 4.987}{\sqrt{1.38}}=254 \text{ rpm}$$

Since the actual shaft speed is 200 rpm which is 79% of the critical speed, it is necessary to increase the value of critical

speed, by decreasing the deflection. Choose therefore a 6.5 cm diameter shaft

$$\therefore\ \delta=1.00 \text{ cm}$$

$$N_c=\frac{60\times 4.987}{1.00}=300 \text{ rpm}$$

Actual speed of 200 rpm is satisfactory, which is 66.6% of the critical speed.

14.7.2 Blade Design

Equations 14.6 and 14.7 give

$$f=\frac{67.3\times 10}{\frac{0.8\times(7.5)^2}{6}}=900 \text{ kg/cm}^2$$

The value of the stress is well within the endurance limit for carbon steel.

14.7.3 Hub and Key Design

Hub diameter of agitator $=2\times$ shaft diameter

$=2\times 6.5=13$ cm

Length of Hub—$2.5\times 6.5=16.25$ cm

Length of Key—$1.5\times$ shaft diameter $=1.5\times 6.5=9.75$ cm

From equation 5.6

$$\frac{T_{max}}{d/2}=lbf_s=\frac{lt}{2}f_c=\frac{67.3\times 100}{6.5/2}$$

$$=9.75\times b\times 650=9.75\times\frac{t}{2}\times 1300$$

$\therefore\ b=3.27$ mm

$t=3.27$ mm

Use 4 mm $\times$ 4 mm $\times$ 10 cm key.

14.7.4 Stuffing Box and Gland

Internal design pressure—5.5 kg/cm^2

From equations 5.36 to 5.40

$$b=d+\sqrt{d}=6.5+\sqrt{6.5}=9.54 \text{ cm}$$

$$t=\frac{P_b}{2f}+c$$

$$=\frac{5.5\times 9.54\times 10}{2\times 950}=0.275\text{ cm}+c=4\text{ mm}$$

$$a=b+2t=9.54+0.8=10.34\text{ mm}\cong 10.5\text{ cm}$$

Load on gland

$$F=\pi/4\times 5.5\,(9.54^2-6.5^2)$$

$$=210\text{ kg}$$

Size of stud

$$210=\frac{\pi d_o^2}{4}\times n\times f_t$$

$$=\frac{\pi\times d_o^2}{4}\times 4\times 587$$

$$d_o^2=0.114\text{ cm}=1.14\text{ mm}$$

$$d_o=1.07\text{ mm}$$

Minimum stud diameter = 15 mm

Flange thickness $=1.75\times 15=27.25$ mm $\cong$ 30 mm

14.7.5 COUPLING

A clamp coupling is suggested (Fig. 5.4). It is made of cast iron

Equation 5.15 gives

$$\text{Force per bolt}=\frac{2\,T_{max}}{\pi\,\mu\,d\times\frac{n}{2}}=\frac{2\times 67.3\times 100}{\pi\times 0.25\times 6.5\times\frac{(8)}{(2)}}$$

$$=662\text{ kg}$$

$$\text{Area of bolt}=\frac{662}{587}=1.125\text{ cm}^2.$$

$$\text{Diameter of bolt}=\frac{1.125\times 4}{\pi}=1.15\text{ cm}$$

$$=11.5\text{ mm}$$

Use 15 M size bolts

Overall diameter of coupling $=6.5\times 2=13$ cm

Reading References

Uhl, V.W. and Gray J.B., *Mixing Theory and Practice*, Vol. I, Academic Press, London, (1966).

Uhl, V.W. and Gray J.B., *Mixing Theory and Practice*, Vol. II, Academic Press, New York, (1967).

Holland, F.A., and Chapman, F.S., *Liquid Mixing in Stirred Tanks*, Reinhold. New York, (1966).

Agitator Selection and Design : Handbook 9, Engineering Equipment Users Association, Constable, London (1963).

Parker, N.H., *Mixing*, Chem. Eng., **71**(12), 165 (1964).

Weber, A.P., *Selecting Turbine Agitators*, Chem. Eng., **71**(25), 169 (1964).

McCabe, W.L., and Smith, J.C., *Unit Operations in Chemical Engineering*. McGraw-Hill, New York, (1967).

Penny, W.R., *Recent Trends in Mixing Equipment Report*, Chem. Eng, **78**(7), 86 (1971).

Weber, A.P., *Selecting Propeller Mixers*, Chem. Eng., **70**(18), 91 (1963).

Bates, R.L., *How to Specify Agitators*, Petrol. Refiner, 39(11), 243 (1960).

Parker, N.H., *How to Select Double Arm Mixers*, Chem. Eng., **72**(18), 121 (1965).

Ludwig. E.E., *Applied Process Design for Chemical and Petrochemical Plants*, Vol. I, Gulf, Houston, (1964).

Jensen W.P. and Talton R.T., *Design of Mixers for Viscous Materials*, A.I. Chem. E., Symposium series **10**(82), (1965).

Bourne J.R., and Butler H., *Some Characteristics of Helical Impellers in Viscous Liquids* A.I. Chem. E. Symposium Series No. **10** (89) (1965).

A.I. Chem. E., *Standard Testing Procedure Mixing Equipment* (Impeller Type) (1965).

Penny, R.W., *Guide to Trouble Free Mixers*, Chem. Eng., **77**(12), 171 (1970).

CHAPTER 15

Filters

15.1 Classification

The function of filtration equipment is to filter a slurry, so as to separate the solid more or less completely from the liquid. Depending on the nature of the slurry, the driving force required for separation may be divided into four categories, namely, gravity, vacuum, pressure and centrifugal.

(a) *Gravity*: This is the simplest method of filtration. But the equipment is generally very bulky, and the filtration is not complete.

(b) *Vacuum*: The equipment used for generating the vacuum is simple. But the filtration equipment may be bulky and relatively expensive. This type of filter is normally intended to collect only small amounts of solids from relatively large volumes of liquid.

(c) *Pressure*: This method gives higher outputs. The equipment may, therefore, be smaller in size. It can handle volatile liquids without much difficulty. These filters are widely used.

(d) *Centrifugal*: This method utilizes maximum separating forces due to centrifugal action. High outputs can be obtained.

It is proposed to consider the design and construction features of some of the more important types of filtration equipment. Such equipment may be either for batch production or for continuous production.

15.2 Vacuum Filters

15.2.1 Batch Type

(1) Tank filter (Nutsch filter) and (2) Leaf filter (Moore filter).

15.2.1.1 TANK FILTER

This filter is simple in construction and is used widely for small scale filtration work. It consists of a tank or cylindrical vessel, divided into two chambers by a perforated plate. The plate can be covered with a filter cloth or wire screen, placed loosely in position or sealed around the edge by means of a caulking ring or a joint ring. A stoneware porous plate can also act as a filter medium. The lower chamber is connected to a vacuum line, the vacuum being created by either a dry vacuum pump or a steam ejector. The size of the vessel is determined by the quantity supplied in each batch. Where large capacities are desired the lower chamber may be evacuated of the filtrate intermittently.

The vacuum produced in the lower chamber draws the liquor down through the filter medium leaving the solids in the upper chamber. The accumulated solid or cake may be washed. The tank or the cylindrical vessel must be designed for vacuum. If a rectangular tank is used it must be sufficiently reinforced, so that buckling is prevented. A cylindrical vessel has, obviously, a greater strength.

15.2.1.2 LEAF FILTER

There are a number of ways in which a leaf can be formed. In a simpler method of fabrication, a screen or perforated plate is welded to a channel frame with a drainage connection at the top. The filter medium is welded to the screen. If a fabric material is to be used as a medium, it is made to cover the screen to form a bag which is fastened at the top. A basket is formed by assembling the leaves in parallel and held in position by two parallel beams, which are placed at right angles to the axis of the leaves (Fig. 15.1).

The basket is submerged in a feed liquor tank. Vacuum is applied at the top of the leaves, through a hose connection, which is attached to each of the submerged filter leaves. After the cake is deposited the basket of leaves is lifted out of the filter tank with a crane and is placed in a wash tank. Vacuum is maintained during the transfer by means of flexible hoses. Wash liquor draws through the cake in the same manner. The leaves are then transferred to a tank, where air blow discharges the cakes.

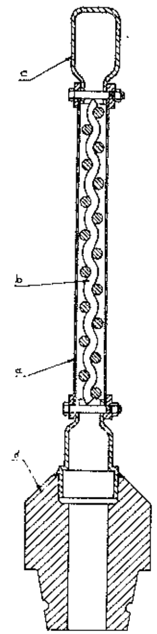


Fig. 15.1 Filter leaf
(a) filter medium (b) screen (c) hoop (d) coupling

The leaf should be sufficiently rigid. The channel frame and the screen or plate must withstand the pressure, corresponding to the vacuum applied. The cross-section of the channel and the thickness of the screen will depend on the size of the frame. The channel can be formed from a sheet of metal.

15.2.2 Continuous Type

(1) Rotary drum filter
(2) Rotary disc filter
(3) Horizontal rotary filter

15.2.2.1 ROTARY DRUM FILTER

This filter (Fig. 15.2) is one of the most widely used types

in industry, finding application both where cake washing is required and where it is not required. Constructionally, vacuum drum filters fall into two groups, namely, those in which the whole drum is under vacuum and those in which vacuum is applied only to an outer annular series of compartments. The second category is by far the most common. The filter consists of the following parts:

(a) Drum (b) Filter medium (c) Automatic filter valve (d) Filter tank (e) Agitator (f) Cake discharge system (g) Shaft (h) Bearings (j) Drive system (k) Supporting framework.

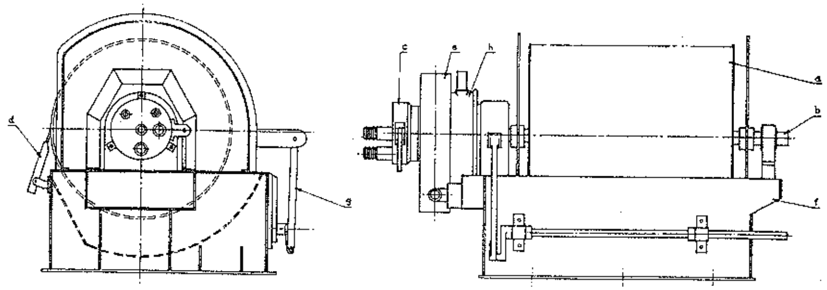


Fig. 15.2 Layout of rotary drum filter.
(a) drum (b) shaft (stub) (c) filter valve (d) scraper (e) driving gear (f) slurry tank (g) oscillating agitator (h) bearing

(a) *Drum*

The drum (Fig. 15.3) diameters vary from about half a metre to about four metres, and the face width from about half a metre to eight metres. Drums are made of wood, cast iron or steel. The surface of the drum is corrugated or covered with corrugated sheets made of copper, stainless steel or other materials to avoid corrosion. This surface is divided into a number of shallow sections, usually 12 to 24, depending on the drum diameter. The sections are sealed from each other by means of longitudinal strips, and are covered with screens or grids for supporting the filter medium. The strips are made from metallic sheets, by bending a flat strip into an approximate U shape. Each section forms an independent compartment. The

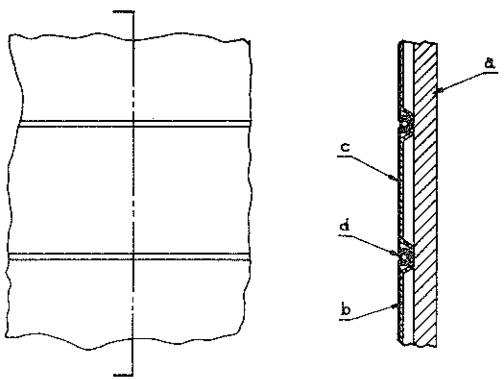


Fig. 15.3 Rotary drum section
(a) drum (b) screen (c) filter cloth (d) longitudinal strip

screen or grid must be so shaped that adequate drainage space (about 2.5 cm depth) is provided between the filter cloth and surface forming the bottom floor of the section. Each section has one or several outlets depending upon the length of arm and the amount of filtrate to be handled. The outlet pipes from each section are connected to a larger common pipe leading to the automatic filter valve. Where maximum separation of filtrate and wash liquor is necessary the outlets are normally arranged along the trailing edge of each section. If both cake dryness and good separation are important a compromise can be made by using both leading and trailing edge piping. In very special cases, where it is essential to have both a sharp separation of liquor and maximum cake dryness, two automatic filter valves can be provided, one mounted on each end of the drum. All leading edge piping is connected to one valve and all trailing edge piping to the other. The ends of the drum may be open or closed, the open design being slightly cheaper. It also facilitates access for maintenance.

(b) *Filter medium*

The filter medium is usually cotton duck, metal woven, cloth or plastic materials in the form of cloth. The common method of screening the filter medium to the screen surface is by

caulking the cloth into the longitudinal divisional strip. The cloth is held taut in position inside the U-shaped strips by blocks of rubber sheathed with a metallic cover. Ends of cloth are sealed either by caulking or by the use of bands or wire. In addition the cloth may be held taut, sometimes by a wire wound at a pitch of about 4 cm. In certain cases, the filter cloth is not fixed to the screen or grid, but is in the form of an endless belt which remains in contact with the drum for only a part of its travel.

(c) *Automatic filter valve*

An important part of the filter is the filter valve. As the drum rotates the valve connects each section on the drum to suction line or to wash line or to pressure line as required by the sequence of operations. The valve consists of three parts [Fig. 15.4 (*a*)].

(*i*) *Rotating tube plate* (*Moving sector block*). This is made as an integral part of the rotating trunnion or shaft which supports the drum. It is essentially a ring with circular ports, to which the filtrate piping from each section is connected. The tube plate, the drum, the filtrate piping and the shaft are interconnected and therefore rotate at the same speed.

(*ii*) *Valve head* (*Fixed distributor block*). It is in the form of a stationary block with an annular channel facing the circular parts in the tube plate. The annular channel is divided into several segments by means of bridge pieces, which may either be fixed or movable. The segments of annular channel are so shaped that they form a piping manifold on the outside. The valve head has one of its surface facing the tube plate and the other side has the manifold for connecting to suction line, wash line and compressed air line. Fig. 15.4 (*b*) shows an alternative arrangement of valve head.

(*iii*) *Wear plate.* It is in the form of a ring which is placed between the rotating tube plate and the stationary valve head. It helps to prevent leakage of the filtrate as it flows from the tube plate to the valve head. The assembly is held tight by a stud with a spring, which is located at the centre of the valve head. The rotation of the tube plate causes wear of the wear plate surface which may be reduced by lubrication with the

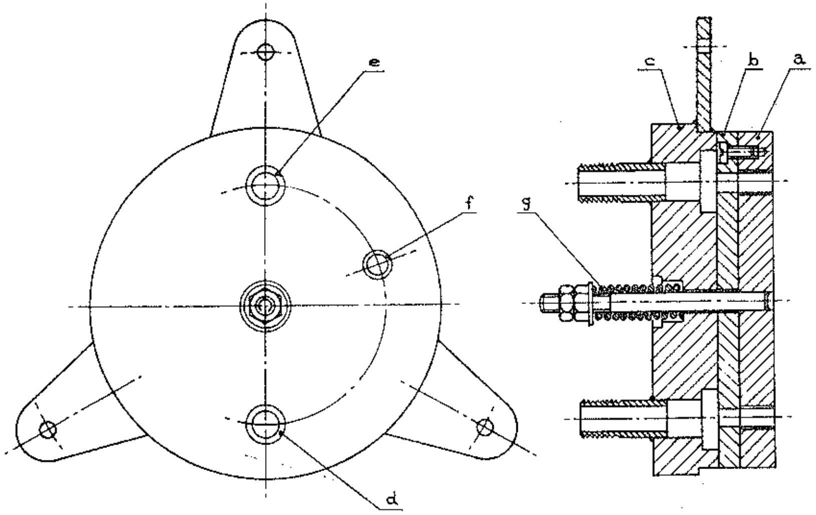
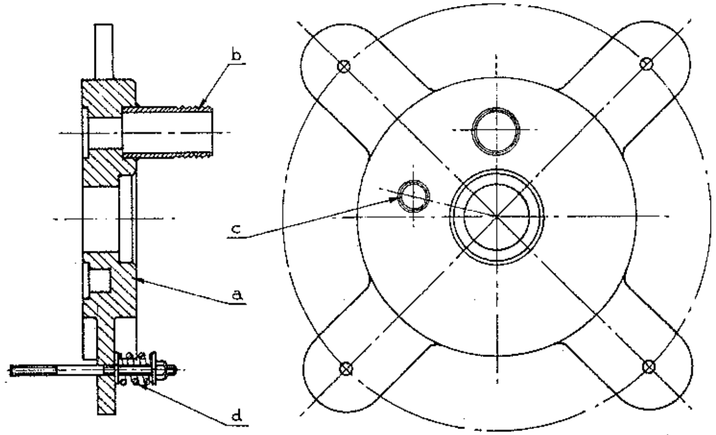


Fig. 15.4 (A) Filter valve components
(a) rotating tube plate (b) valve head (c) wear plate (d) section line
(e) wash line (f) compressed air line (g) spring



(B) (a) valve head (b) section line (c) compressed air line (d) spring

help of a grease cup. The wear plate is replaced when required.

(d) Filter tank (e) Agitator

The tank is placed below the drum and covers a portion of the drum, over its entire length. The tank is fabricated from sheets of metal usually about 4 mm to 6 mm thick. The most common filter tank is designed for about 40% submergence of the drum. The slurry in the tank is stirred by a horizontal agitator, which is placed close to the tank bottom. The agitator may be of the paddle type. A better agitation system consists of oscillating cross bars supported by arcs hung from the tank. The agitator moves back and forth at a slow rate.

(f) Cake discharge system

The cake formed on the filter cloth surface, is discharged as each section of the drum moves towards the cake discharge point, while the filter valve shuts off the vacuum on that particular section. Depending on the type of cake, it is removed by one of the several available methods.

(i) Scraper discharge for easy filtering of solids.
(ii) Roll discharge for materials which are sticky and thin but will peel off.
(iii) Longitudinal wire under considerable tension for discharge of semi-sticky cakes.
(iv) String type discharge for cakes which tend to adhere to the filter cloth.
(v) Belt discharge where clean cloth is essential.
(vi) Precoat scraper where measured amount of cake is removed for feeds low in solids and producing a very thin cake.

A scraper discharge arrangement (Fig. 15.5) consists of a narrow scraper supported by, and attached to the filter tank. It is free to oscillate about a hinged support, so that it can float and follow the contour of the cloth, which is slightly inflated by air pressure, to facilitate cake discharge. The scraper is balanced so that it rests slightly on the filter medium. The scraper may have a rubber edge or a metal edge.

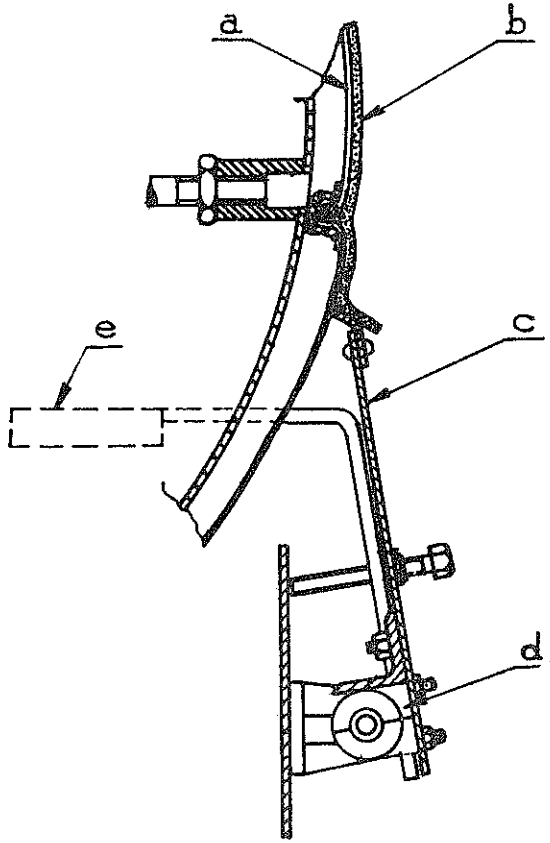


Fig. 15.5 Scraper mechanism
(a) filter medium (b) filter cake (c) scraper (d) scraper support
(e) counterweight

(g) Shaft

The drum is fixed to the shaft through the end covers by providing suitable hubs at the centre. For open end drum the shaft is fixed through the spider frame. The shaft extends on either side of drum, and is supported by bearings located close to the drum. In most cases the shaft is not continuous, but is in the form of a trunnion fixed to the cover at either end. The

drum, therefore, acts as a connecting component between the two trunnions. On one side of the drum the shaft is extended beyond the bearing for connecting it to the drive system, and for fixing the rotating tube plate of the filter valve. The loading of the shaft will be as shown in Fig. 15.6. It will transmit power required for the following loads :

(i) Acceleration of the drum from the stationary conditions. The rotation of the drum is extremely slow, usually less than one revolution per minute.
(ii) Friction at the bearings, wear plate and scraper knife.
(iii) Resistance of the slurry in the tank, in which the drum is partially immersed.

Each of the above loads can be assessed in terms of the torque required, depending on the radius at which the above loads are acting. The shaft can also be considered as a beam with loads due to the weight of the drum, driving gear and the tube plate, and supported at the bearings. From these loads, it is possible to calculate the maximum bending moment. From equations (5.2) and (5.3), the shaft diameter can be determined, with the values of maximum torque and bending moment. Since the shaft is generally not continuous, but in the form of a trunnion at either end, each trunnion will require an independent assessment of stresses. The drum will also transmit power in this case.

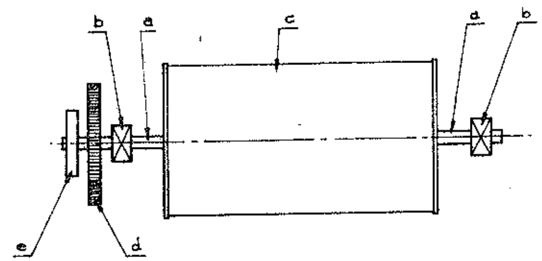


Fig. 15.6 Shaft loading
(a) shaft (b) bearing (c) drum (d) gear (e) valve

(h) Bearings

Usually two cup lubricated journal bearings are provided for supporting the shaft. The load is primarily radial. Assuming a permissible bearing pressure of 120 kg/cm^2 the size of the bearing bush can be determined (equation 5.17). The features of the bearing are as shown in Fig. 15.6.

(j) Drive system

Considering an extremely slow speed at which the drum is to rotate, the appropriate choice of the driving gear is the worm gear (Fig. 5.15). The worm gear is mounted on the drum shaft, while the worm is fixed to a perpendicular shaft. The worm shaft is driven through a chain or V-belt drive from an electric motor. The major speed reduction is obtained from the worm and worm gear. Ball bearings are used to support the other shafts.

(k) Supporting framework

It is essential to build a framework surrounding the drum. This is made up of structural sections and plates welded to form a rigid structure. The framework has to support the load of the drum, through the bearings, automatic valve, tank with agitator, discharge mechanism and washing equipment.

15.2.2.2 ROTARY DISC FILTER

The vacuum disc filter comprises one or more circular discs mounted on a common horizontal central shaft which is driven so that the discs rotate (Fig. 15.7). The lower part of each

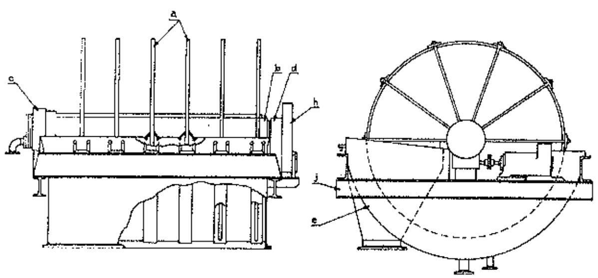


Fig. 15.7 Rotary disc filter
(a) filter disc (b) shaft (c) rotary valve (d) bearing
(e) slurry tank (h) gear drive (j) supporting frame

disc is submerged in a tank of slurry during passage through which a cake is deposited on each side of the disc as a result of suction through piping located inside the central shaft. The cake is then dried by further suction as it rotates through air and is finally discharged from the disc, generally by a reversed blast of air combined with the use of a scraper.

The main components of the filter are (a) filter disc (b) shaft (c) rotary valve (d) bearings (e) slurry tank (f) agitator (g) cake discharge system (h) drive system (j) supporting framework.

(a) *Filter disc*

It (Fig. 15.8) is constructed from a number of segmental elements which can be detached. The segment may either be of solid construction with each face grooved or it may be hollow with each face perforated. The disc is covered with a filter cloth in the form of a bag, which slips over the sectors. The sector narrows to an apex through which protrudes a radial pipe nipple which is plugged into the central shaft. The bag is folded over and clamped at the edges.

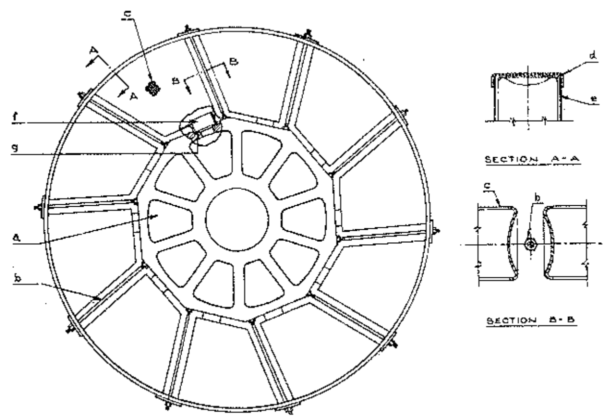


Fig. 15.8 Disc details
(a) channel for filtrate (b) the rod (c) sector (d) sector clamp
(e) perforated plate with wire mesh (f) pipe

The discs are made of wood, carbon steel, stainless steel or bronze. The diameters of discs range from half a metre to 5 metres. Each disc is composed of ten or more sectors. Multiples of upto 12 discs are assembled on one shaft, which rotates at a speed varying between 2 to 5 revolution per minutes.

(b) *Shaft*

The central shaft (Fig. 15.9) is hollow, with a large central hole and a number of channels (manifolds) concentric with the central hole but separated from it. These channels are closed at one end, while at the other, they lead into a rotary valve. In view of the hollow shape of the shaft, it is easier to construct the shaft in cast iron.

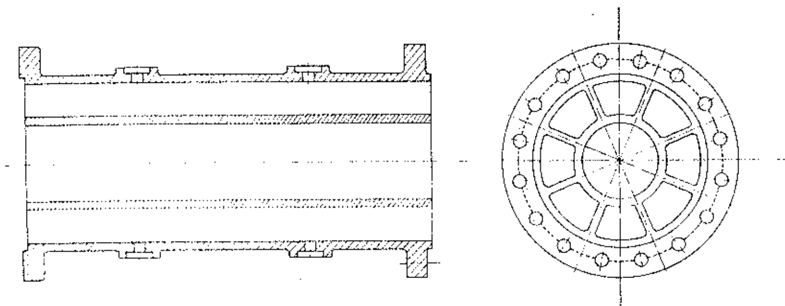


Fig. 15.9 Shaft for disc filter

(c) *Rotary valve*

The valve (Fig. 15.10) is somewhat similar to the type used in rotary drum filters. It has a large part for filtrate outlet, and smaller connection for the blow-back of air.

(d) *Bearings*

The shaft is extended on either side by a flanged connection and is supported in bearings at either end. The bearings are similar to those in the drum filter.

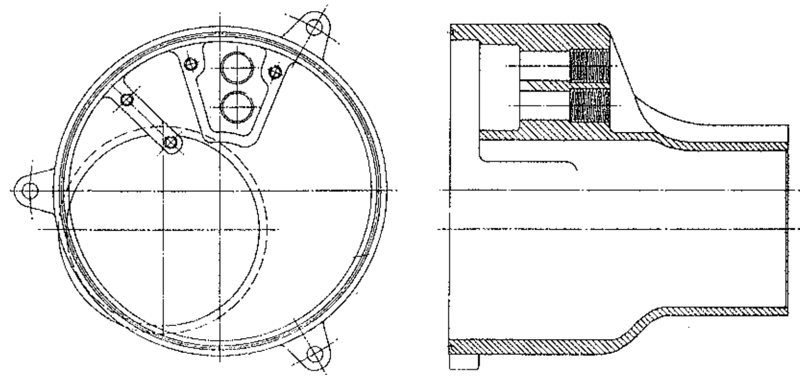


Fig. 15.10 Valve for disc filter

(e) *Slurry tank*, (f) *Agitator*

The tank is semi-circular in cross-section and a part of it is provided with crenelations—one for each disc. The level of the slurry in the tank is critical, since it must always be sufficient so that, a segment after cake discharge, is fully immersed before it is connected to the vacuum source. The exact point of rotation where suction is applied is controlled by the location of one of the bridge pieces on the rotary valve.

If the drive shaft is to be partly or wholly submerged in the slurry, a stuffing box has to be fitted. The slurry must not be so high as to foul the cake discharge knife. The surface available for cake formation is restricted to 20 to 45%.

The agitator is somewhat similar to the one used in the slurry tank of the drum filter.

(g) *Cake discharge system*

The discharge of the cake is facilitated by a pair of radial blades touching the disc on either side. As the shaft revolves the sectors are brought in contact with the radial blades, while an air-blow bulges out the filter cloth. The blade flattens out the bulge so that the cake falls off and drops, guided by a deflector through the spaces provided between the crenelations to a chute or conveyer.

(h) *Drive system*

This is somewhat similar to the drive system suggested for drum filter. The speed of the disc is extremely slow and is also variable. The calculations of power required may be made from acceleration and frictional loads. The motor power is usually between $\frac{1}{2}$ hp and 10 hp depending on the filter size.

(j) *Supporting framework*

The framework is built up from structural sections and plates by welding. The thickness of sections or plates may vary between 6 mm and 12 mm.

15.2.2.3 HORIZONTAL ROTARY FILTER

This filter (Plate V) is essentially a rotating horizontal circular table divided into a number of sectors. Each sector is a separate compartment. Vacuum is applied from the automatic valve concentrically located beneath the filter. The bottom of each compartment slopes towards the centre of the filter. The filter medium is supported on a punched plate or wire mesh drainage deck and caulked into dovetailed grooves at the periphery of the sectors. An elevated centre island and a circumferential vertical rim help to keep the slurry and the wash liquors on the filtering surface.

Feed is applied by a pipe and a weir box distributor above the filter, which extends radially across the width of the surface. Dewatered and washed cake is discharged by a spiral scroll located just ahead of the feed. A radial dam is placed between the scroll and the point of feed to prevent slurry running back to the cake discharging zone.

There is no filter tank. The unit is mainly controlled by rotational speed with a variable speed drive and by the rate of slurry feed.

The filter may be built of steel, lead or rubber lined steel and stainless steel. The rotating table is supported on the lower side, the filter medium being located on the upper side. The drive may be either by a gear ring attached to the rim of the table, with a driving pinion meshing with it or by a sprocket and chain or V-belt arrangement. In either case a variable speed drive must be provided for so as to permit operation in the range of 1-10 minutes per revolution.

15.3 Pressure Filters

15.3.1 BATCH PRESSURE FILTER

(1) Filter press
(2) Mechanised filter press
(3) Pressure leaf filter

15.3.1.1 FILTER PRESS

This filter (Fig. 15.11) consists essentially of a skeleton framework made up of two end supports connected by horizontal bars. On these bars a varying number of filter chambers are assembled. These chambers are formed either by cloth covered recessed filter plates or by alternate frames and cloth covered plates. The chambers are closed and tightened by a screw or hydraulic ram which forces the plate or plates and frames together, making gasketed joint of the filter cloth. The slurry is forced into these chambers, which are designed in such a way that the filtrate cannot leave the filter press except by passing through the filter medium. The filtered cake is held back by the cloth. The flow of the slurry through the filter press is not from chamber to chamber consecutively, but through all the chambers simultaneously, so that each chamber fills at the same rate.

Different parts of the filter press are (a) plates and frames (b) side bars (c) movable head (d) stationary head (e) screw (f) screw standard (g) supporting legs.

Table 15.1 gives the major dimensions of some standard sizes of filter presses.

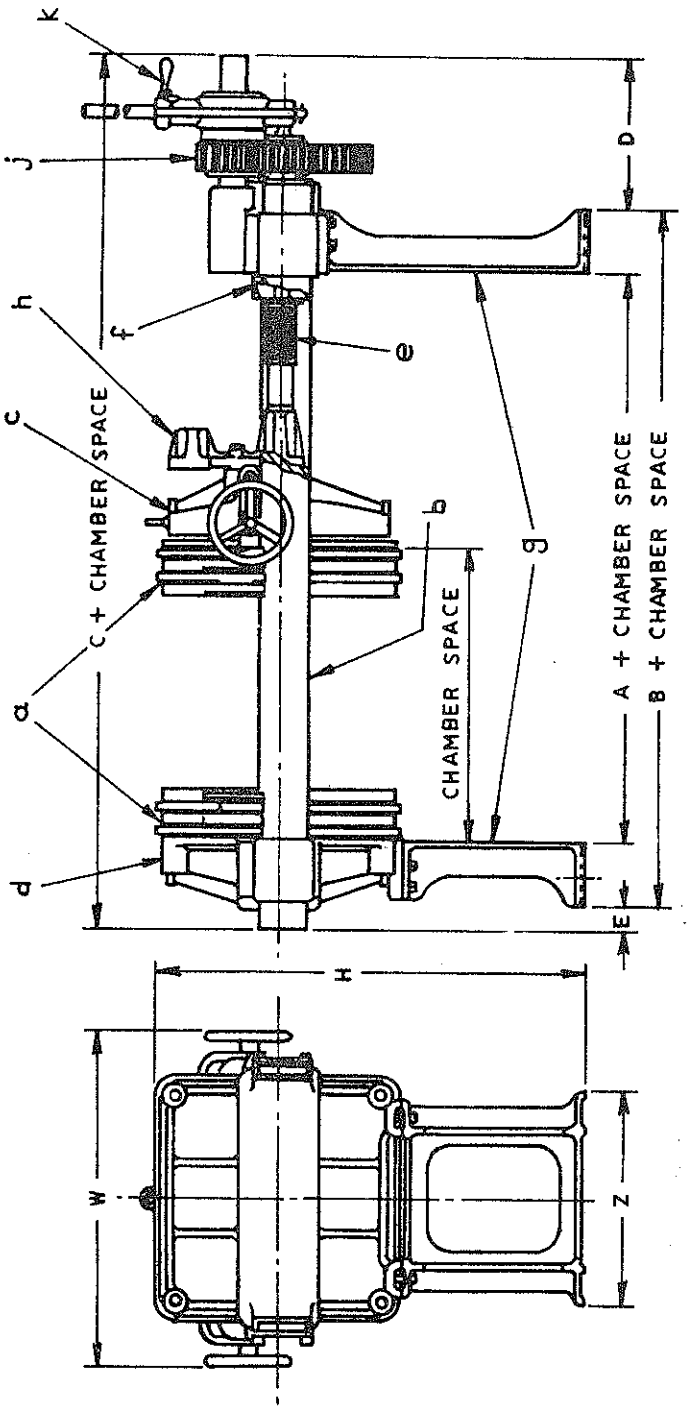


Fig. 15.11 Filter Press
(a) plates and frames (b) side bar (c) movable head (d) stationary head (e) screw (f) screw standard (g) supporting legs (h) thrust block (j) gears (k) wheel

(a) *Plates and Frames*

These are square or rectangular in shape (Fig. 15.12). They are made of cast iron, cast steel, stainless steel, aluminium, lead, bronze, nickel, monel or special alloys. With non-ferrous materials plates are made of wood, hard rubber or plastics. Coated materials are also used.

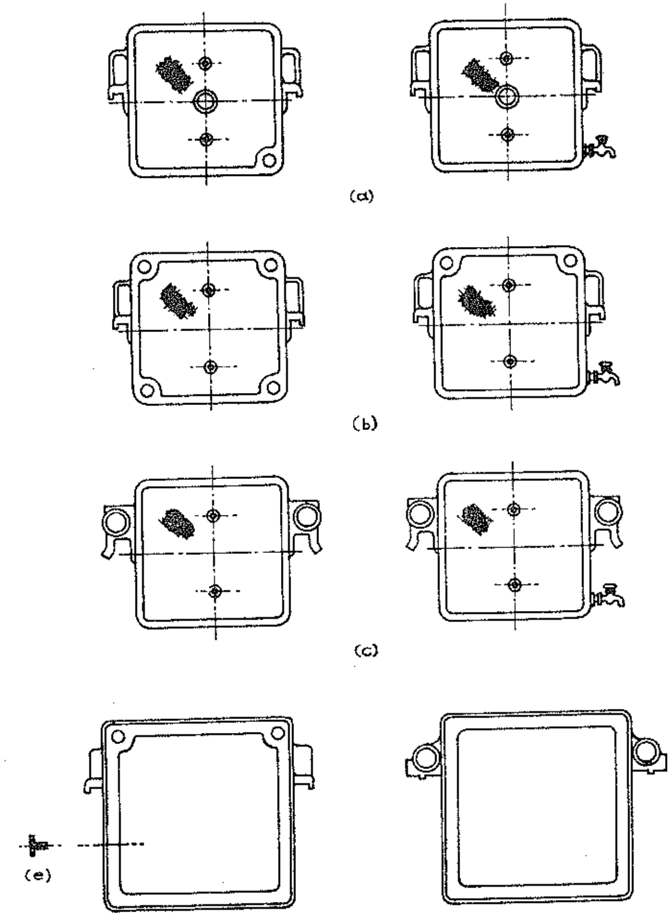


Fig. 15.12 Plates and frames
(a) plates with centre feed corner discharge (left view—closed discharge; right view—open discharge) (b) plates with feed and discharge at corners (c) plates with side feed and corner disc (d) frame (e) cross-section of side of frame

Table 15.1

Dimensions of Filter Presses

Size of Plate	Plate Thickness		All dimensions in mm as per Fig. 15.11							
	Standard cast iron	Standard wood	A	B +	D +	E =	C	H	W	Z
175	12.5	—	178	284	94	16	394	300	238	138
300	12.5	31	294	468	156	49	668	575	400	250
450	19	44	575	836	238	50	1124	875	725	375
600	25	44	725	1060	450	64	1574	1100	850	550
750	25	44	825	1228	412	72	1712	1275	1100	600
900	25	—	1090	1590	405	81	2076	1400	1250	725
900	—	56	915	1415	405	81	1901	1400	1250	725
1080	28	—	1350	2000	496	125	2508.5	1475	1525	1050
1080	—	56	1175	1828	496	125	2336.6	1475	1525	1050
1200	31	62	1495	2140	512	112	2764	1575	1725	1050
1400	—	69	1450	2100	500	112	2712	1775	1950	1100

(a_1) *Plate arrangements*: Depending on the type of filter press, there are two general types of arrangements: (i) flush plate and frame (ii) recessed plate.

The first arrangement is made up of alternate plates and frames forming chambers. The plates generally have the drainage surfaces practically flush with the joint surfaces. The frames are hollow and provide the space for filter cake and can be made for any desired thickness of cake, the usual thickness range being 25 to 50 mm. Filter cloths are placed over each plate to cover the plate surface on both sides. The joints between plates and frames are made tight on closure by the gasket joints formed by filter cloth. The feed and discharge holes are located either in the corners of the plates and the frames or on lugs placed on the sides.

The second arrangement has only plates, the chamber being formed in the recesses in the adjacent plates. The depth of recess seldom exceeds 15 mm making a cake of 30 mm. Where necessary frames may be inserted between recessed plates to permit building up of a thick cake. The feed is generally located in the centre of the plate. Filter cloth on the recessed surface of each side of the plate is sealed around the feed opening by two cloths sewn together at the hole or by clip nuts. The plates are not under any pressure, but a slight pressure might be exerted due to clogging. The plate thickness is such as to provide for enough rigidity. It varies according to the material. The frames are however under pressure, which acts on the inner periphery of the frame. The frames are held in position against the plate and therefore receive considerable support. The bending moment due to pressure acting on each side of the frame is given by

$$M = \frac{Pl^2}{4} \tag{15.1}$$

where P—pumping pressure

l—length of the side of frame.

The stress in the cross-section of the side of frame is given by

$$f = \frac{M}{Z} \tag{15.2}$$

where Z—modulus of section of the cross-section of the side of frame.

If the permissible stress (f) is known, the size can be determined. The cross-section is rectangular as shown in Fig. 15.12 for which modulus of section (Z) can be determined. Alternately, assuming a suitable cross-section, the stress can be assessed. The stress should not exceed the permissible value.

(a_2) *Method of filtrate discharge*: Filtrate discharge may be from outlets in each plate, with or without control cocks. This is known as 'open' discharge. A closed channel in the plates and through the fixed head is connected by ports to each plate. This channel may be located in the joint surface or outside it, and carries the filtrate discharge of each plate. Such plates are known as closed discharge. Test and shut-off cocks can be provided to permit examination of filtrate from any plate of a closed discharge filter press and to shut off the flow from any plate when necessary.

(a_3) *Drainage surface of the plate*: The pyramid type drainage surface is commonly adopted to obtain maximum amount of cloth support without losing effective filtering area. Radial drainage grooves or corrugated surface metal plates are also used. For wood or rubber plates a corrugated surface is provided.

(a_4) *Control of leakage*: Certain materials, particularly very thin fluids, tend to leak through gasket joints made by the filter cloth between plates and frames. To overcome this difficulty, several types of plates and frames are used. Frames are cast with special grooves in the joint surfaces. The grooves are connected to openings forming a channel which discharges through the head of the filter pressure. Alternately, plates are made with cast or milled grooves into which rubber gaskets are fitted. In another arrangement, the filter cloth is held against the plate by means of metal bands screwed against a rim inside the joint surface. A flat rubber gasket is cemented to the joint surface.

(b) *Side bars*

All the plates and frames, and the movable head with the thrust block are supported on two side bars. According to the number of plates the load may be resting only on the part of the length of the bars. However assuming that the entire span of the bars is used to support maximum number of plates, it is possible to determine the size of the bar. The ends of the bar are fixed. The bending moment due to the total load divided between two bars is given by

$$M = \frac{Wl}{12} \tag{15.3}$$

where W—load on each bar=half the total load

l—length of bar

The stress, $$f = \frac{M}{Z} \tag{15.4}$$

where Z—modulus of section for the cross-section of the bar.

The cross-section of the bar is rectangular with the depth 4 to 6 times the thickness. Assuming a suitable cross-section, the stress (f), which should not exceed the permissible value, can be calculated.

(c) *Movable head (Follower head)*

The movable head, Fig. (15.13) is cast, to the same size as that of the plate. On one side it has a surface similar to that of

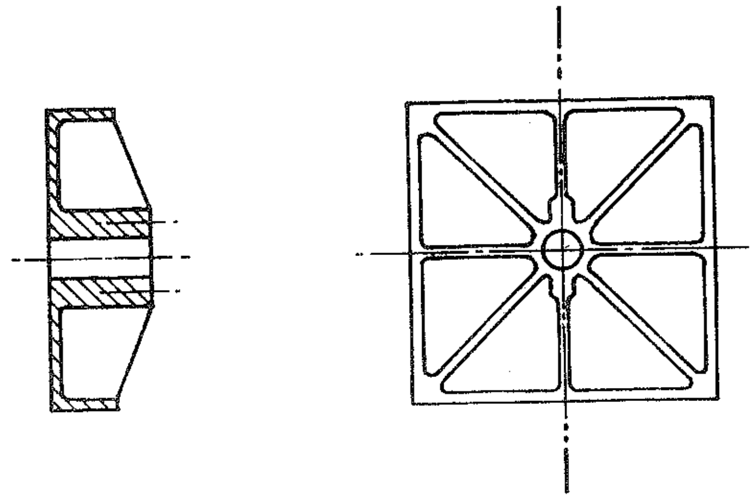


Fig. 15.13 Movable head

the plate, which is in contact with the adjoining plate or frame, and therefore acts as a plate. On the outer side it is provided with radial ribs terminating into a centered boss. All the plates and frames are pressed together by the movable head towards the fixed head. The movable head is pressed by either a screw or a hydraulic ram, for holding the plates and frames tight. A thrust block is mounted on the head and the screw is fitted in the thrust block. The function of the thrust block is to facilitate quick drawback of the movable head. The movable head is supported on the two side bars and is provided with rollers and hand wheels on either side for easy movement.

The thickness of the movable head can be determined on the basis of the loading conditions. On one side of the head the screw exerts a force at the centre of the head. This force is required to keep the plates and frames tight, through the gasket joints. The force required on the joints can be calculated as (equation 6.36).

$$W = l \times b \times Y \times n \tag{15.5}$$

where W = force

l—length of the gasket joint (periphery of the plate forming the joint)

b—effective seating width of the filter cloth acting as gasket

Y—design seating stress for filter cloth material

n—number of joints

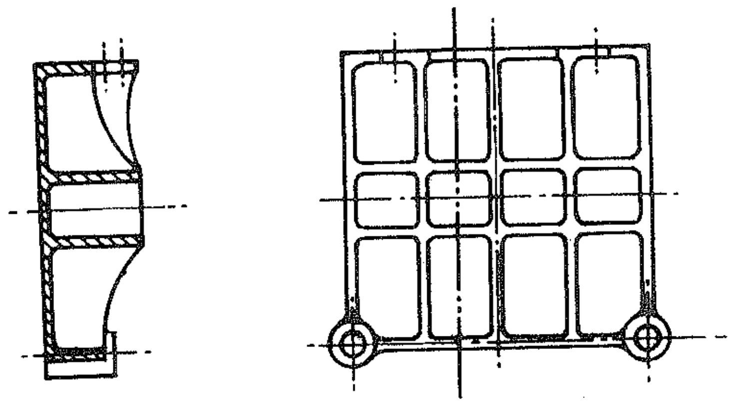


Fig. 15.14 Stationary head

The above load may be considered as a uniformly distributed load, with the support in the centre of the span. The maximum bending moment will be given by

$$M = \frac{W \times L}{8} \tag{15.6}$$

where L—length of longer side of the plate.

The cross-section of the head will be as shown in the Fig. 15.13. Assuming a suitable thickness, and knowing the size of the plate, it should be possible to determine the modulus of section (Z). The stress will be given by

$$f = \frac{M}{Z} \tag{15.7}$$

This stress should not exceed the permissible value. The average minimum thickness for grey iron castings is about 6 mm for parts upto about 45 cm in length. It gradually increases to 18 mm for large and heavy castings. Ribs are added to make the construction more rigid. While they always increase rigidity, they may fail to increase strength. The provision of ribs for the construction of the head helps to make it rigid and limits deflection. This is essential to maintain uniform pressure on the gasket joints.

(d) *Stationary head*

The stationary head (Fig. 15.14) is cast to the same size as the plate. On one side it has a surface similar to that of the plate and acts as a plate. On the other side it has cross ribs. The head is supported on its edge by the legs, and is attached to it by bolts. The load on the stationary head is given by equation 15.5. If the load is considered as a uniformly distributed load the maximum bending moment is given by

$$M = \frac{WL}{Z} \tag{15.8}$$

where L—length of the longer side of the plate.

The cross-section of the stationary head is as shown in Fig. 15.14. The thickness may be assumed and the stress may be checked on the same basis as the design of movable head. Ribs are provided for rigidity which is expected to ensure uniform pressure on the joints.

(e) *Screw*, (f) *Screw standard*

The plates and frames are pressed between the two heads by means of a screw acting on the central boss of the movable head, or on a thrust block which is interposed between the end of the screw and the movable head. The screw standard is a fixed casting block supported on the legs. The side bars are made to terminate within this block. The screw standard has a nut located in the centre which meshes with the screw. The screw is tightened directly by a wheel fixed at its end or by a gear and pinion mechanism attached to the end of the screw. A tommy bar of suitable length may give additional leverage for turning the screw. On large size filter presses, the screw is replaced by a hydraulic ram.

The screw is made of steel and has square threads. The load on the screw is mainly the tightening force W (equation 15.5). It is therefore subjected to the following stresses

(i) Compressive stress resulting from the load W

$$f_c = \frac{4\,W}{\pi\,d_1^2} \tag{15.9}$$

where d_1—root diameter of screw

If the length of the screw exceeds eight times the root diameter, the screw must be treated as a column (See 3.9.3).

(ii) Torsional shear induced in the screw by the external turning moment T (force applied to wheel) to overcome the force W and thread friction.

$$T = \frac{1}{2}\; W\,d_m\,t_{an}\,(\alpha + \phi) \tag{15.10}$$

where

d_m—mean diameter of thread

α—helic angle of screw

ϕ—angle of friction

$$\text{Shear stress } f_s = \frac{T}{\frac{\pi}{16} d_m^3} \tag{15.11}$$

The design of screw is based primarily on the permissible compressive stress f_c. The other stresses are checked for permissible values.

The nut placed in the screw standard is made of brass or bronze. The number of threads to be provided are given by

$$n = \frac{W}{\frac{\pi}{4}(d_2 - d_1)p_b} \tag{15.12}$$

where

d_2—inside diameter of nut threads

d_1—outside diameter of nut threads

p_b—bearing pressure

Safe bearing pressure for steel screw and bronze nut is between 160 kg/cm^2 and 220 kg/cm^2

(g) *Supporting legs*

The legs are usually made of cast iron with the cross-section in the form of an angle or T-section. They are mainly under compression due to the weight of the fixed head on one side and the screw standard on the other side. The rest of the weight consisting of the plates and frames, side bars, movable head, etc., is eccentric and will create a bending stress in addition to the compressive stress. It is usual to choose a suitable cross-section of the leg and check for the stresses.

15.3.1.2 MECHANISED FILTER PRESS

A modification of the standard plate and frame filter is a mechanised unit, which consists of circular plates fixed in the vertical position and supported at the centre line by a horizontal shaft on each side of the filter. Two sets of circular frames mounted 180° apart are attached to one of the side shafts, which can be rotated. When one set is in operating position, the other is in cleaning or discharge position. The plates are made up as two movable parts sealed to each other by O-rings. Between them they form a closed compartment. For closing and sealing the filter, air pressure in excess of operating pressure is applied to the compartment, spreading the two parts against the adjacent gasketed frames.

15.3.1.3 PRESSURE LEAF FILTER

This consists of a series of uniformly spaced leaves mounted in a vertical or horizontal cylindrical or rectangular pressure vessel. In general the leaves consist of a heavy wire drainage screen mounted in a tubular frame, which acts as a support and filtrate conduit. Filter medium may be either fine wire mesh cloth or any natural or synthetic fabric. The wire cloth is sealed, between flanges on the frame by rivets, bolts, etc.

The vertical pressure vessel type can be equipped with a quick opening door. On the horizontal vessel type the leaf and a manifold below it are attached to the movable head. Leaves and movable head are removed from the vessel for discharging and cleaning by internal rollers and a monorail.

15.4 Centrifugals

These machines are used for separating solids from liquids either by filtration or by sedimentation. Advantage is taken of the centrifugal force created due to rotation of a container in which the slurry is fed. There are two main types of centrifugal machines.

(a) Machines in which the solids are retained by a porous medium, while the liquid is forced through.

(b) Machines in which the solids are separated as a cake while the liquid is drained off by overflow.

Both types can be operated either as batch operation or as continuous operation and with or without facilities for washing.

15.4.1 Centrifugal with Perforated Basket

This is the most widely used centrifugal in chemical and process industries. It consists of a perforated metal basket. The axis of the basket is either vertical or horizontal. In the vertical mounting the basket may be either under driven (Plate VI) or over driven (Fig. 15.15).

Surrounding the basket is a stationary monitor case or curb (solid shell), with an annular gutter at the bottom provided with a discharge pipe. Slurry is generally fed through a pipe which directs it against the side of the rotating basket. Occasionally a device such as a cone moving on the vertical axis is used to promote the formation of a more even cake from top to bottom of the basket. The solids deposit on the basket liner, while the filtrate passes through and is collected in the curb and flows through the annular gutter to the discharge. The solids may be washed most efficiently by spraying water up and down the vertical wall. The cake is discharged through the openings in the bottom. An annular valve which forms part of the bottom is opened to allow the cake

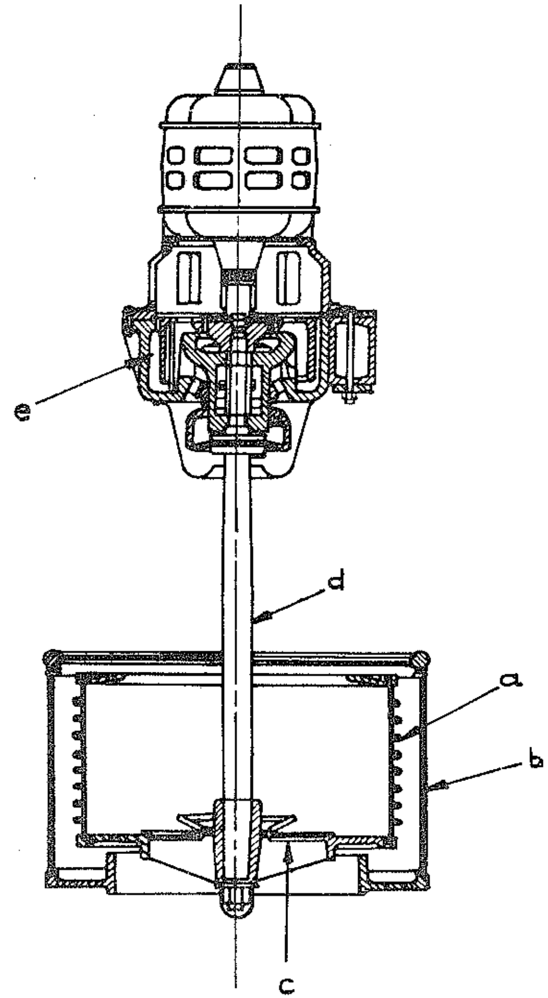


Fig. 15.15 Overdriven basket centrifugal
(a) basket (b) curb (c) valve (d) shaft (e) drive system

discharge. The valve is closed after the discharging action is complete. The solids drop to a chute, to a conveyer or to a rotating screen. The cake is dislodged from the basket walls manually by use of a wooden spade or automatically by a plough blade operated by a rack and pinion to allow its up and down movements and a hinge for sideways motion. In the case of free flowing crystals, baskets have a conical bottom which is self discharging. An alternative cake release device is known as a peeler, which comprises a narrow blade which moves back and forth across the face of the cake while it simultaneously advances towards the screen surface.

In horizontal axis basket centrifuge, the basket is directly mounted on the extension of the electric motor shaft and rotates at the same speed as the motor. The shaft is supported on tapered roller bearings on one side of the motor, while the basket is overhanging. The charge is fed to the basket through a horizontal spout. After washing and drying, a travelling knife slices out the cake. The movement of the knife is well controlled so that its edge stops at one mm from the filter cloth. The movement of the knife is controlled by a mechanism located on the end cover.

The main components of the basket centrifuge are (a) basket (b) mesh or screen with filter cloth (c) plough blade (d) curb (e) cover (f) shaft (g) bearings (h) drive system (i) brake.

(a) *Basket*

The main parts of the centrifuge such as the basket, screen, valve, etc., are constructed of mild steel, copper and various alloys such as bronze, monel and stainless steel. In case of carbon steel, they may be lined or coated with lead, rubber, tin etc. The basket may be either underdriven with open top or overdriven with a suspended basket. The open top machines are preferred for intermittent service and for handling of volatile solvents and fumes. The suspended basket type is used for high running speeds, bottom discharge of solids and the greatest stability with unbalanced loads.

Table 15.2 gives sizes of baskets. The speed of rotation varies between 50 and 300 rpm. Basket shells have perforations and may be strengthened by hoops located evenly over the depth of the baskets. The number of hoops might vary from one to five depending on the depth. These hoops may be made of materials of higher tensile strength than the basket materials. Table 15.3 gives a list of materials used for basket shells and hoops, and the allowable stress values.

Table 15.2
Sizes of Baskets of Centrifuge

Mean Diameter of Basket Shell D (mm)	Recommended Basket Depth at Lip (mm)
400	250
500	275
630	325
750	350
900	350/450
1000	400/500
1200	450/600/750
1250	450/750/830

The load on the basket is due to the centrifugal force of the rotating mass of the basket, hoop and material charged which may be taken as distributed over the entire surface (See 3.14)

(i) *Unhooped basket* : The centrifugal force

$$F=\frac{2\pi^2 N^2 D}{g}\times W \tag{15.13}$$

where

W—weight of basket shell and material charged

N—revolutions per second

D—mean diameter of basket, shell.

Surface area of the basket $(A)=\pi D\ L$ (15.14)

where

L—depth of basket

Internal pressure on the basket wall due to force F (assumed to be uniformly distributed)

$$p=\frac{2\pi^2 N^2 D}{g}\times\frac{W}{\pi\ DL}=\frac{2\pi\ N^2}{g}\times\frac{W}{L} \tag{15.15}$$

The circumferential stress due to internal pressure

$$f=\frac{p\ D}{2\ t}\left(\frac{P}{P-d}\right)$$

$$=\frac{\pi N^2 DW}{g L t}\left(\frac{P}{P-d}\right) \tag{15.16}$$

where

t—thickness of basket

P—axial pitch of perforations

d—diameter of perforation

(*ii*) *Hooped basket* (basket shell and hoops made of same material).

The stress in basket due to centrifugal force (f)

$$=\frac{\pi^2 N^2 D}{g}\times\frac{W_1\ k}{(L+K+A)}\times\frac{P-d}{P} \tag{15.17}$$

The stress in the hoop

$$=\frac{\pi N^2 D}{g}\times\frac{W_1}{(L+K+A)}\times\frac{1}{J} \tag{15.18}$$

where

D—mean diameter of basket shell

N—revolution per second

W_1—weight of basket shell, hoop and material charged

L—depth of basket

K—$\frac{P_1 P-2d^2}{P_1 P}$ (a factor allowing for reduction of stiffness due to perforations).

P_1—circumferential pitch of perforations

d—diameter of perforations

J—joint efficiency.

The values of D and L are determined by filter area requirements. Assuming a shell thickness (t), the stress (f) may be determined, which should not exceed the value given in Table 15.3. A minimum basket thickness recommended is given in Table 15.4. The built-up basket shall be adequately balanced by static or dynamic methods appropriate to the shape and size of basket.

(*b*) *Mesh or Screen*

This is built up from a coarse wire screen placed against the wall of the basket covered by a filter cloth of cotton, wood or metal wire fabric. It is held in position either by caulking in the circumferential grooves at the ends of the basket or by means of an expandable ring.

(*c*) *Plough blade*

The function of the blade is to dislodge the thick hard crystalline cake compacted against the filter cloth or mesh, due to centrifugal pressure. The action of the blade causes considerable wear and tear. The blade holder may be fixed to the curb or casing and can be operated by a rack and pinion mechanism so that the blade can be moved vertically along the depth of the basket. The blade presses against the cake during operation and may be withdrawn if required.

(*d*) *Main casing or curb*

This is a stationary cylindrical vessel placed outside the basket and is supported at the bottom by legs. The curb may be fabricated from mild steel or stainless steel sheets of about 6 to 10 mm thickness. It carries a hinged cover at the top and a valve at the bottom.

(*e*) *Cover*

It is made from metal sheets, and fixed to the curb by a gasketed flange joint. A portion of the cover (lid) is made separate and is attached to the main cover by hinges and bolts and wing nuts to facilitate frequent opening.

(*f*) *Shaft*

In the case of overdriven or underdriven vertical centrifugals, the drive shaft is held vertical by bearings located at the top in the first type and at the bottom in the latter type. In the case of horizontal centrifuge the shaft is horizontal and is supported at one end. In certain cases the shaft extends into the basket and is supported in a bearing located at the centre of gravity of the entire rotating mass.

The horse power required is determined from the considerations of the torque required for acceleration of the basket and the frictional torques. The nature and the position of the forces acting on the shaft will depend on whether the shaft is vertical or horizontal and on the drive system.

Table 15.3

Materials of Construction for Basket Shells and Hoops of Centrifuges

Materials	*IS Specification*	*Tensile Strength in* kgf/mm^2	*Maximum Allowable Stress Value in* kgf/mm^2
Mild steel	IS : 2002-1962		11
	(a) Grade I	37 to 45	
	(b) Grade 2	42 to 50	
Stainless steel	IS : 1570-1961	55 min	14
	(a) 04Cr19Ni9		
	(b) 07Cr19NiNb70		
	(c) 07Cr19Ni9Mo2Ti20		
	(d) 05Cr18Ni11Mo3Ti20		
Copper sheets (fully annealed)	IS : 1972-1961	22.5 min	5.5
Jointless copper shells electrolytically deposited	—	25 to 28	6

Table 15.4

Minimum Thickness of Basket Shells
(All dimensions in millimetres)

Basket Mean Diameter, D	*Minimum Thickness, t*	
	Unhooped	Hooped
Below 650	3.2	2.5
650 upto but not including 1250	5.0	3.2
1250 and over	6.0	5.0

In the case of overdriven or underdriven vertical centrifuge, the axial forces acting on the shaft are mainly due to the weight of basket and its contents, and any coupling or pulley mounted on the shaft. The axial forces will create a tensile stress in the case of overdriven and compressive stress in the case of underdriven. Forces perpendicular to the shaft will be acting due to

tensions of belt or chain drive. A resultant stress has to be determined from the shear stress due to torque, tensile or compressive stress due to axial load and a bending stress. Based on equations 4.9, 5.2, 5.3 the minimum diameter of the shaft can be determined. The actual shaft provided may not be of the same diameter over its length. In the case of horizontal centrifuge there will be no axial force on the shaft. The design will, therefore, be based on torque and bending moment due to normal loads. In all cases it is necessary to check the unbalanced forces, deflection and the critical speed of the shaft. Due to variation of speed of the basket it may not be possible to obtain perfect balancing of the centrifugal forces. Deflection of the shaft should be limited to 0.002 cm at critical points. Vibrations can be limited if the normal speed is below 60% of the critical speed (secs. 5.2.5 and 14.5.1.2).

(*g*) *Bearings*

Axial forces on the shaft can be supported by provision of thrust bearings. In addition ball bearings are also provided for radial loads. For an overdriven shaft a self-lubricating bearing shaped in the form of a sphere helps to maintain the shaft in the vertical position in spite of the vibrations created due to unbalance of rotating masses. The choice of the bearings may be made according to the loading conditions.

(*h*) *Drive system*

The basket may be driven either by connecting the shaft directly to the motor shaft or by use of a flexible coupling. For variation of speed a belt or chain drive may be used. The motor may be supported by brackets fixed to the casing or to a framework.

(*i*) *Brake*

A brake in the form of a shoe with a lever is applied to the basket for reducing speed and ultimately making the basket stationary.

15.4.1.2 PUSHER TYPE CENTRIFUGAL

This (Fig. 15.16) consists of a horizontal basket mounted on a shaft. Feed enters through a feed pipe, which is made to enter the basket. The slurry from the feed pipe is accelerated gradually to the basket speed. A cake is deposited on the screen of the basket. To remove the cake a hydraulically operated pusher moves axially shearing the cake along the perforated basket through a short distance. The pusher stroke is controlled by piston operating in a cylinder under oil pressure. Dry cake flies off the end of the basket into the portion of casing (housing) placed in front of the basket. From the casing the cake passes out through the discharge. The pusher returns to its initial position. The length and frequency of stroke of the pusher may be adjusted and is generally set to handle slightly more than the anticipated maximum production of solids. As the case moves along the basket, the filtrate is thrown out in the casing surrounding the basket. This part of the casing is sealed from the front portion of the casing to prevent leakage of the filtrate. A wash water pipe supplies the water for washing.

The basket diameters vary from 60 cm to 240 cm. The pusher shaft is solid and is located inside the hollow shaft of the basket. The hydraulic cylinder forms a part of the hollow shaft. The cylinder cover is fitted to the cylinder by a flanged joint. The operating oil is supplied from either side through an annular gap in the hollow shaft. The hollow shaft and the cylinder are to be designed for transmission of torque and axial thrust due to oil pressure acting on the cylinder. In addition the cylinder and the hollow shaft have to withstand internal oil pressure. The stresses produced are circumferential and axial and a shear stress due to torsion. The hollow shaft is supported between two tapered roller bearings ; the rest of the shaft with the hydraulic cylinder and other mechanisms is enclosed in a housing. The basket over hangs from one of the bearing supports, while the drive to the shaft is from a pulley placed near the other bearing. From the loading on the shaft, namely the weight of the hydraulic mechanism, basket, pulley, pusher rod shaft weight etc., a bending moment diagram can be drawn to assess the maximum bending moment and stresses due to bending can be determined. The design of the shaft is finally based on the permissible resultant stress.

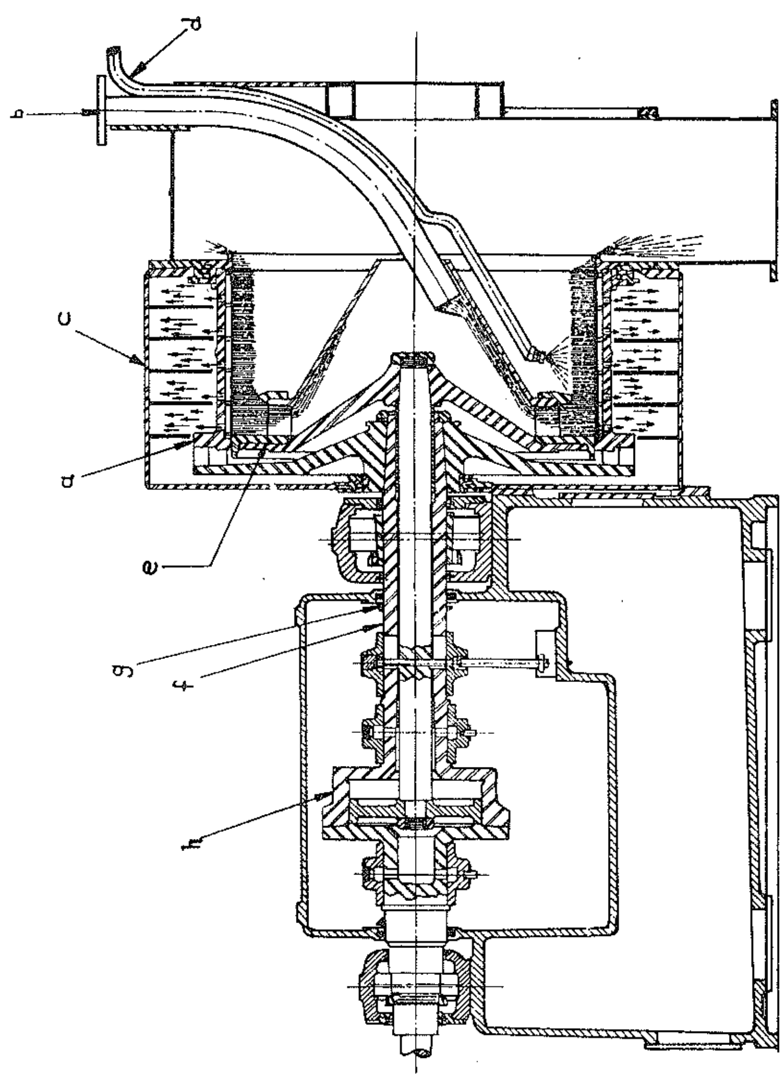


Fig. 15.16. Pusher type centrifugal.
(a) basket (b) feed pipe (c) casing (d) wash water pipe (e) pusher (f) main shaft (g) pusher shaft (h) hydraulic cylinder for operation of pusher shaft

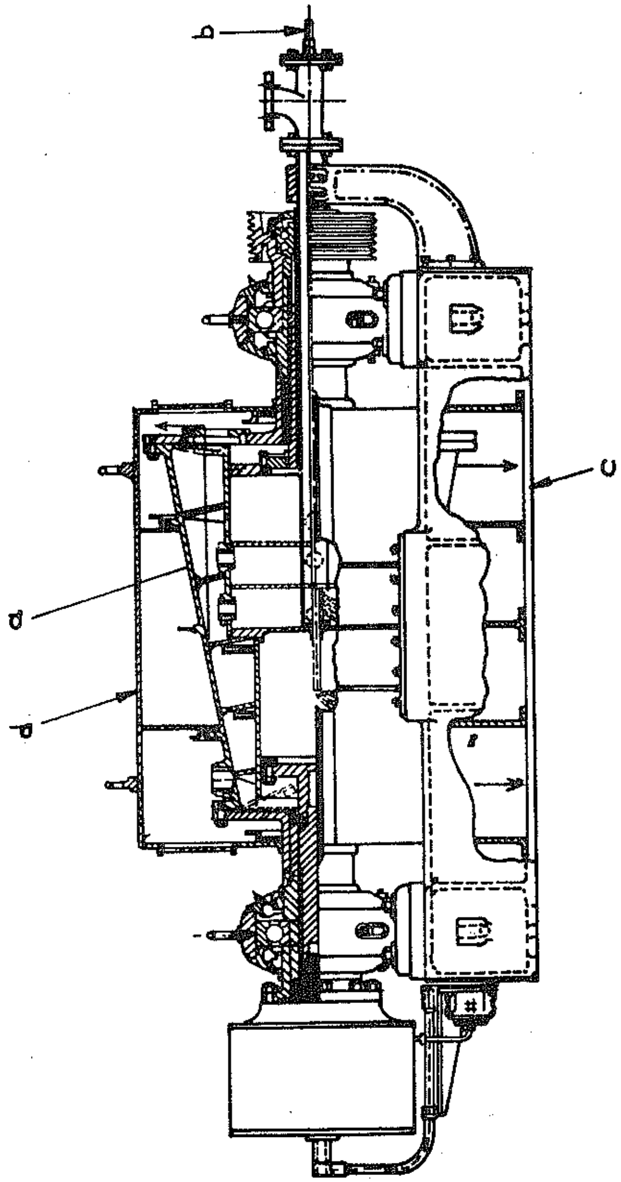


Fig. 15.17 Screw conveyor type centrifugal, (a) solid bowl (b) feed pipe (c) outlet (d) casing

15.4.2 Continuous Centrifugals

There are two types of continuous centrifugals.

(1) Screw conveyor type (Fig. 15.17)

(2) Spiral scraper type.

15.4.2.1 SCREW CONVEYOR TYPE

A solid bowl is mounted horizontally with an internal conveyor scroll. Both the bowl and the conveyor revolve in the same direction. The centrifugal is driven by a V-belt and pulley fixed to the shaft towards the feed end. Rotation of the bowl is transmitted through an epicyclic gear train, placed in a box towards the other end to the conveyor, which is made to rotate at a slower speed than the bowl.

Feed slurry enters into the chamber inside the conveyor assembly through an axially located pipe (hollow shaft). Parts around the periphery of this chamber distribute the material into the bowl. Centrifugal force holds the material against the bowl wall. The conveyor moves the solids along the bowl wall and leads to the solid discharge part. The liquid collects in the lower part of the bowl and flows towards the filtrate part at the other end.

The bowl is cylindrical in shape (partly tapered) preferably with a taper at one end. It may be fabricated from sheets of mild steel, stainless steel, copper, etc., or cast in two pieces jointed together by a flanged joint. The bowl is provided with covers or heads at either end attached by flanged joints. A hollow shaft is fixed to the cover and carries a V-pulley at one end. Pedestal ball bearings are placed at either end to support the two pieces of shaft. The pedestals are in turn supported by a base frame cast or fabricated. The outer casing surrounding the bowl is divided into compartments. It is connected to hoppers for slurry feed and discharge. The casing is fabricated from sheet steel.

Reading References

Purchas, D.B., *Industrial Filtration of Liquids*, Leonard-Hill, London, (1971).

Dickey, G.D., *Filtration*, Reinhold, New York, (1961).

Chamlets, J.M., Elledge, L.R., and Porter, H.F., *Filters*, Chem. Eng. **62** (6), 191 (1955).

Review of Filtration, Chem. Age of India, **21** (3), 1961 (1971).

Purchas, D.B. *The Sizing of Rotary Vacuum Filters*, Part I, Brit. Chem. Eng. **2**, 196 (1957).

Day, R.D., *Technique for Selecting Centrifuges*, Chem. Eng. **81** (10), 98 (1974).

Fleming, D.K., and Dahlstrom, D.A., *Enclosed Continuous Filtration Equipment*, Chem. Eng. Progr., **59** (7), 62 (1963).

Perry, D. Jr., *What You Should Know About Filters*, Hydrocarbon Process. Petrol. Refiner, **45** (4), 145 (1966).

Perry, J.H., *Chemical Engineers' Handbook*, 5th Edition, McGraw-Hill, New York, (1973).

Brown G.G., *Unit Operations*, John Wiley and Sons, New York, (1956).

Jordon D.G., *Chemical Process Development*, Part I and II, Inter Science, New York, (1968).

CHAPTER 16

Dryers

16.1 Introduction

The function of the dryers is to remove water from wet solids or from slurries by evaporation. The process involves heat transfer and diffusion. Accordingly the process of drying can be divided into two stages. During the first stage, in which the material surface is wet, the rate of evaporation is constant. This is known as constant rate period. In the next stage the surface being dry, the water must force itself to the surface by diffusion, which is slower than evaporation. This is therefore known as falling rate period. The entire process therefore involves heat transfer as well as material transfer. Heat transfer can be achieved either by direct contact between the heating medium and the material or by indirect contact in which the material and the medium are separated by a wall. Heat transfer and diffusion can be facilitated if the material is agitated or sprayed or the moisture extracted during the process of drying. Although evaporation is the main operation in drying of the materials, in certain cases it is equally essential that the physical appearance and properties of dried product be preserved. A critical choice of processing conditions and the method of heat transfer is necessary in such cases.

16.2 Types of Dryers

A large number of dryers have been designed for each group of materials, with different arrangements and methods of heat transfer, as also the heating mediums. Dryers can therefore be classified according to material, solids or liquids, according to the method of heat transfer; direct or indirect, or according to the heating medium; air, steam, hot water, etc. On the basis of the method of heat transfer, dryers can be classified as (a) direct dryers (b) indirect dryers. In each category they can further be divided into batch and continuous types. As a general rule production rates of 5000 kg per day are best handled by batch dryers and rates over 50,000 kg in a continuous dryer.

Details of construction and design features of some of the important types of dryers are considered. Flash dryers, fluid bed dryers, spray dryers have complex design features and accessories. These have been excluded.

16.3 Batch Type Driers

16.3.1 Tray Dryer

The simplest type of batch dryer is the tray dryer (Fig. 16.1), which is essentially a cabinet or large compartment with a number of trays. These trays may either be fabricated from sheets or from screens. In these dryers, steam, gas or electrically heated air is used as a drying medium. The air is passed by means of a fan over a radiator or over finned tubes and then over the trays. A portion of the air is let out at the discharge, the remainder is reheated and recirculated. An amount of fresh air equivalent to the volume discharged is admitted at the fan. In modern designs of tray dryers hot air is allowed to travel only a short distance, usually not more than that corresponding to two or three stacks of trays. The air is reheated after each passage over trays, with partial rejection of humid air and its replacement by fresh air. Secondary heating tubes are placed in the path of the air to restore its temperature and heat content.

The cabinet or the chest, in which the shelves and trays are located, is rigidly built from a framework of structural sections, provided with panels and doors of double metallic sheet construction having heat insulation material between. The spacing of trays is such as to maintain low pressure losses and m baye about 7.5 cm. Air velocities of the order of 750 cm per

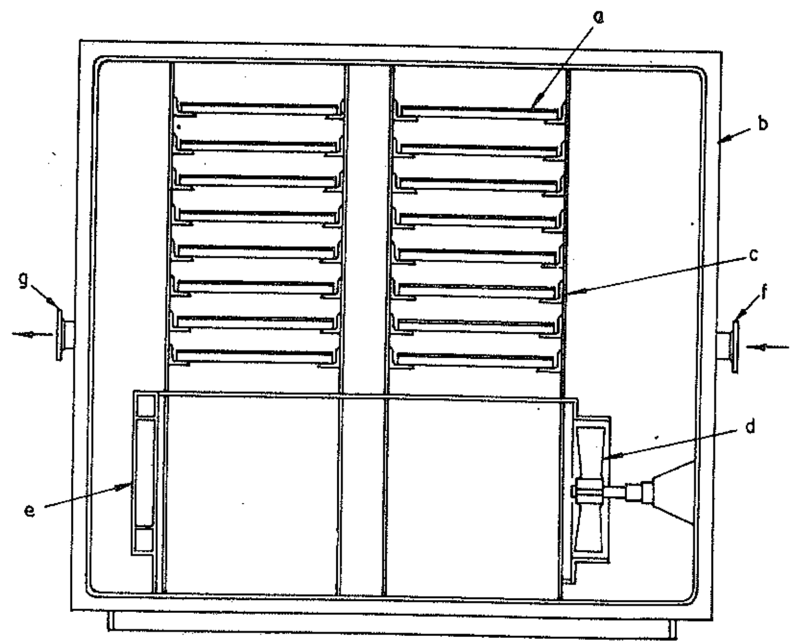


Fig. 16.1 Tray dryer (door removed)
(a) tray (b) cabinet (c) shelf (d) fan (e) radiator (f) air inlet (g) air outlet

minute are used between trays, but where there is a danger of entrainment of the material as drying proceeds, lower velocities should be used. A high volume propeller fan is adequate. It is common to use two air passes, the middle tray acting as a horizontal partition between two passes. Uniform distribution of air is important. Stagnant air pockets should be avoided and for this reason curved deflector baffles are provided at corners. Where extremely uniform drying of material throughout the drying cycle is important, the design of cabinet is such as to permit frequent reversal in the direction of air flow.

Batch vacuum shelf or tray dryers are generally used for materials which are excessively heat sensitive. They consist of a cast iron rectangular or steel cylindrical chamber, fitted with a vacuum-tight charging door. The door is either of the quick-acting type or is provided with several wing-nuts and swivel bolts.

Steam-heated shelves are connected to a steam manifold at one end and a condensate discharge at the other end. A number of trays are arranged on the shelves. The material to be dried is loaded into these trays. The chest is evacuated during each cycle before steam is supplied.

For vacuum tray dryers, the framework with the covering must be strong enough to withstand external pressure of about 1 kg/cm^2. All joints must be air-tight. The walls and the door can be strengthened by welding horizontal and vertical strips on the outside surface of the sheets.

In some cases the trays are mounted on trucks with wheels which may be pulled out of the cabinet for loading and unloading. Several such trucks may be placed in the dryer.

The trays are usually 60 cm wide, 90 to 180 cm long and 3 to 4 cm deep. They are made of mild steel, stainless steel, enamelled iron or other special materials, and are fabricated from sheets of 3 mm to 6 mm thick. A shelf is fabricated from steel structural sections like angles or tees, on which about 10 to 20 trays may be supported. The shelf must be sufficiently rigid to avoid deflection of the framework due to dead load of trays and the material. It is necessary to ensure that the trays remain flat under the load of the material.

The framework consists of horizontal structural sections, acting as beams, while the vertical sections will act as columns. The deflection of each horizontal section may be determined by the following equation,

$$y = \frac{5\,W\,L^3}{384\,EI} \qquad (16.1)$$

where W—load carried by each horizontal section. The total load of trays and material is distributed between all horizontal sections

L—length of section between supports

E—modulus of elasticity of the material of the section

I—moment of inertia of the cross-section.

The cross-section of the structural member should be so chosen that the deflection will be between 1 and 2 mm. The vertical structural members should be designed as columns as per equation 3.16.

16.3.2 Jacketed Pan Dryer

This (Fig. 16.2) consists of a flat bottom shallow cylindrical pan with a steam jacket at the bottom and on the sides. An anchor shaped scraper type of agitator is used to move the wet material over the surface heated by the steam. The material is removed through an outlet at the bottom.

If the pan dryer is to work under vacuum it is fitted with an air-tight conical cover with nozzles. The vapour generated by heating is sucked by a vacuum pump.

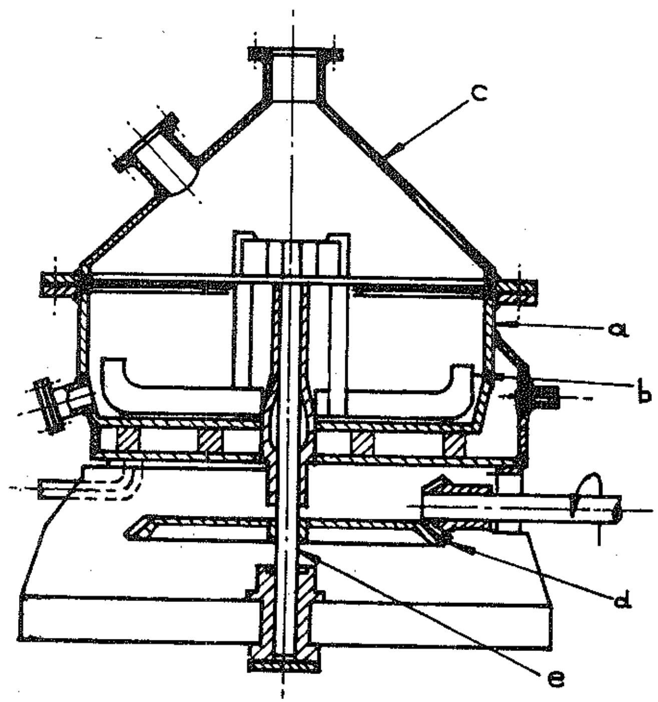


Fig. 16.2 Jacketed pan dryer
(a) pan (b) anchor agitator (c) cover (d) bevel gear (e) shaft

The scraper is rotated at speeds usually between 2 and 20 rpm with the intention of merely preventing the solids from forming a hard cake on the heating surface. When drying slurries and wet pastes, clearance between the scrapers and heated surfaces must be minimal to prevent a skin of dried material building up on these surfaces. The driving system must, therefore, be designed for heavy torque conditions. The usual size of the pan is upto 3 metres in diameter, with capacities up to 5000 litres. The drive to the stirrer is either from the bottom or from the top of the pan with a bevel reduction gear, which in turn is driven by a V-belt and motor.

The pan is designed for an external pressure of about 3 to 6 kg/cm^2, depending on the pressure of steam. In addition if a vacuum pan is to be designed the design pressure will be increased further by 1 kg/cm^2.

The design of the pan dryer is similar to that of a reaction vessel discussed in chapter 7. The stirrer design is also indicated in chapter 14.

16.3.3 Rotary Vacuum Dryer

This is similar in principle to a pan dryer, where a jacket is used for the heating medium and an agitator is used for moving the wet material over the heated surface. The unit consists of a horizontal jacketed cylindrical shell, closed at the ends by suitable heads. The dryer is built in a variety of sizes, ranging from about 75 cm diameter by 4 m long to about 1.6 m diameter by 12 m long. A central shaft is supported in bearings outside the shell. It is sealed by stuffing boxes against leakage through the holes in the heads.

The shaft carries a spiral agitator with two sets of blades. One set of blades moves the material in one direction and the other set in the opposite direction. Doors are provided on the shell, at the top for charging the material and at the bottom for discharging. Some dryers have a rotating shell with stationary agitator. Hot water, steam or suitable fluid is supplied to the jacket for heating. There is also provision for feeding steam into the shaft of the agitator. This involves the use of a rotary joint (Fig. 16.4), which provides a joint between a rotating steam pipe and a stationary condensate piping.

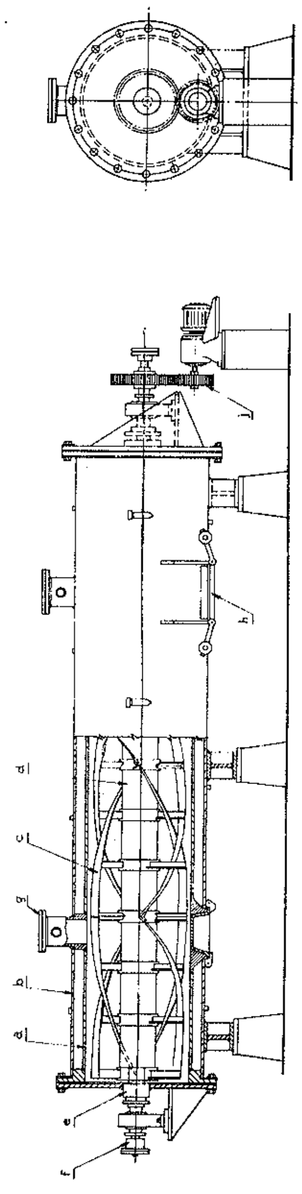


Fig. 16.3 Mechanically agitated dryer
(a) shell (b) jacket (c) agitator (d) shaft (e) stuffing box
(f) rotary joint (g) charging nozzle (h) discharge door (j) drive system

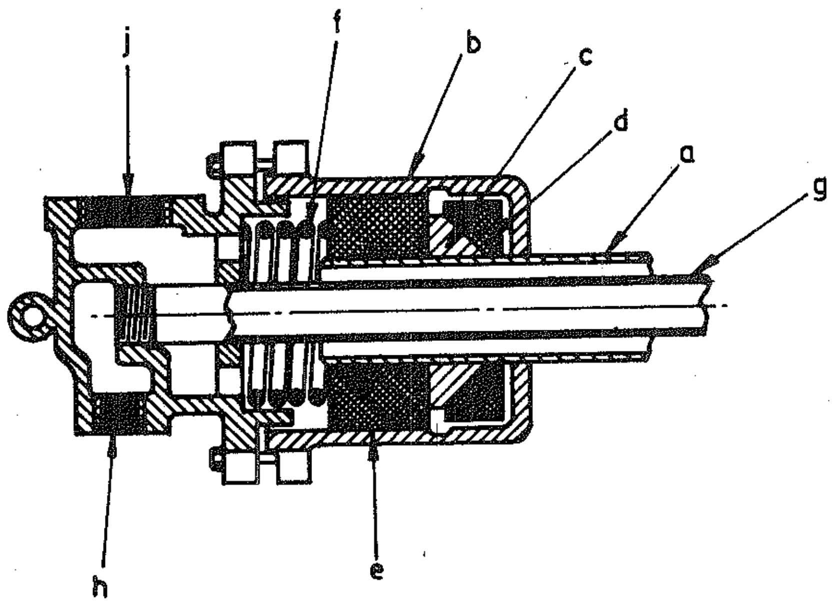


Fig. 16.4 Rotary joint (a) rotating steam pipe (b) joint housing (c) shoulder (d) graphite seat (e) graphite bushing (f) seating spring (g) condensate discharge pipe (h) condensate discharge connection (j) steam connection

16.3.4 Tumbler Dryer

This type of dryer (Fig. 16.5) is, to some extent, replacing the cylindrical rotary vacuum dryer. It has two opposing jacketed cones on a common jacketed short cylindrical base. When the cones are in a vertical position the dryer can be rapidly discharged. The unit is provided with supporting trunnions running in suitable bearings. The trunnions are hollow. A vacuum connection is made through one of the trunnions. The connection pipe is turned upwards in the cone and is fitted with a dust filter at its end. Inlet and outlet pipes pass through the other trunnion for supply of a suitable heating

medium to the jacket. A special design of a rotating valve is used for this purpose. One of the trunnions is driven through a reduction gear and chain drive.

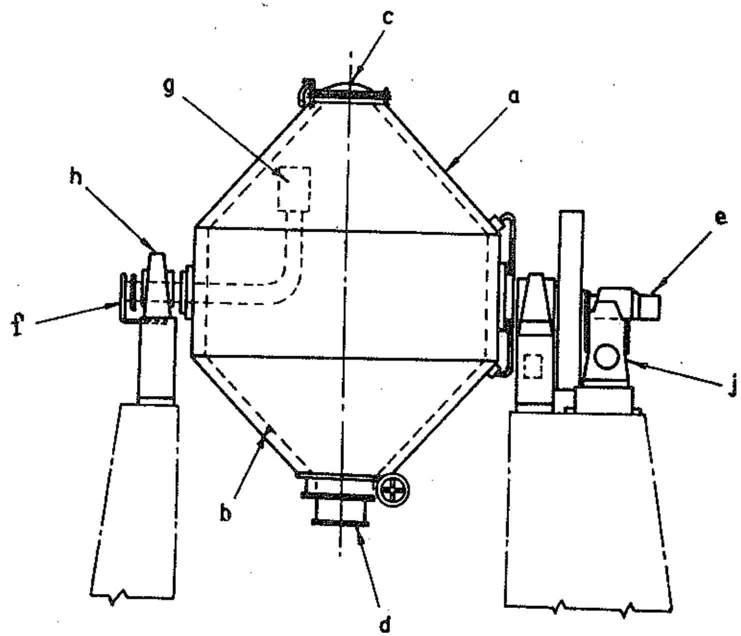


Fig. 16.5 Tumbler dryer (a) conical shell (b) jacket (c) cover (d) discharge (e) steam inlet (f) vacuum connection (g) dust filter (h) bearing (j) drive system

Rotational speeds will range from 12 rpm for small units to 3 to 4 rpm for large commercial installations. The horse powers vary between 1/2 and 15. The base cylinder diameters are about 1 m to 3 m.

16.4 Continuous Dryers

16.4.1 Band Dryer

With the trays on trucks, the dryer can be made continuous by passing the trays with the wet material continuously through a drying chamber. Trays may be placed on a conveyor instead of using trucks. In some cases the material may be placed or attached directly to a conveyor belt. The conveyor belt is made of perforated metal plate or woven wire.

A section of the dryer is used for locating the heater which may be usually of steam heated finned tubes. A fan which is placed above the heater pulls the air through the heater and circulates it through the wet material. Recirculation and reheating may be automatically controlled.

Conveyor widths vary between 40 cm and 250 cm. Lengths range upto 50 m, so that the material can be retained in the chamber for sufficient time. The entire chamber is fabricated out of structural steel sections with steel sheets welded to it. Doors are provided at either end with the necessary ports. The chamber is properly insulatr l.

16.4.2 TURBO DRYER

It consists of a vertical rotating structure placed in a cylindrical housing which carries annular tray assemblies, superimposed on each other at different levels. The material which is loaded in trays drops down from one level to the next lower level by means of a stationary scraper. The structure rotates about a central hollow portion in which a vertical shaft carrying a number of multiblade fans is located. The action of the fans is to draw air radially inwards over the material in the trays, and to recirculate it outwards to the housing which encloses the rotating structure. Heating elements are placed in the space between the rotating structure and the housing. The wet material enters at the top, and the dry discharge is removed from the bottom. The air enters from the bottom and leaves from a stack at the top.

16.4 3 ROTARY DRYER

It (Fig. 16.6) consists of a long cylindrical shell mounted horizontally with a slight slope. The shell is either rotated or may be kept stationary. If it is stationary, an agitator is made to revolve within the shell at a slow speed. The wet material is fed at the upper end, and moves gradually towards the lower end due to the rotation of the cylinder or movement of the agitator. Warm air or a stream of hot gas travels counter-

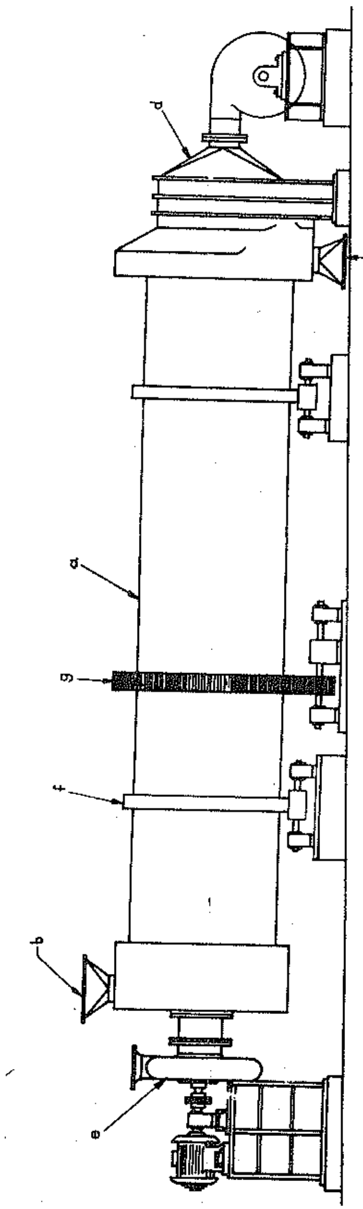


Fig. 16.6 Rotary dryer (a) shell (b) feed hopper (c) discharge hopper (d) air preheater (e) exhauster (f) tyre (g) gear wheel

current to the material. The rate of feed, the speed of rotation or agitation, the volume of the heated air or gases and their temperatures, are so regulated that the material is completely dried before it is discharged at the lower end.

With the rotation of the shell, the material is carried upwards, one-fourth of the circumference and then rolls back to the lower level. The action of lifting the material is carried out more efficiently if lifting devices (Fig. 16.7) such as shelves

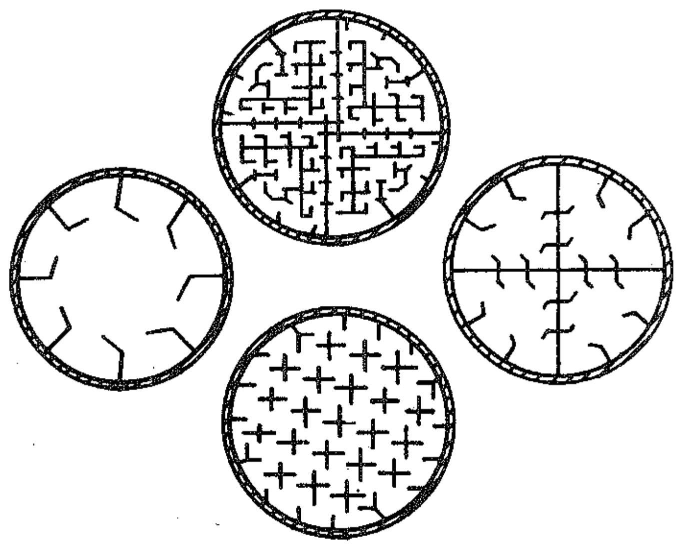


Fig. 16.7 Lifting devices

or baffles are provided on the inner circumference of the shell along the entire length. These are of different designs, and are able to lift the material to half-way round the circumference of the shell and then drop from the topmost position, through the central part of the shell, where the air is hottest. The simplest shelves are longitudinal baffles, plain or serrated, of about 5 cm to 10 cm width on the periphery of the cylinder. Cross shelves covering the entire cross-section of the cylinder give better distribution. Baffles with louvres form a cone section with the narrower end at the feed. In some cases the

cylinder is divided in sections, each section being provided with radial and cross baffles.

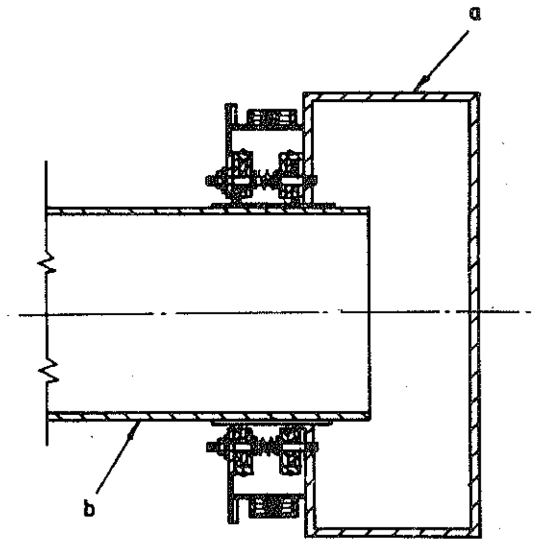
The wet material is fed continuously through a hopper at the upper end and a vibrating feeder or screw regulates the flow of the material. The dry material which drops in a hopper at the lower end may be removed continuously by a bucket elevator. Free flowing feedstock will usually be fed through an inclined chute, rotary valve, vibrating feeder, rotary table or a belt conveyor.

The retention time of the material in the dryer will be determined by several factors such as the slope of the dryer shell, its speed of rotation and the length, the arrangement of flights, etc. The retention time at a given speed of rotation is inversely proportional to the slope of the shell. The rotational speed in rpm × dryer diameter in m lies between 75 to 105. The slope is 18 mm to 54 mm per m.

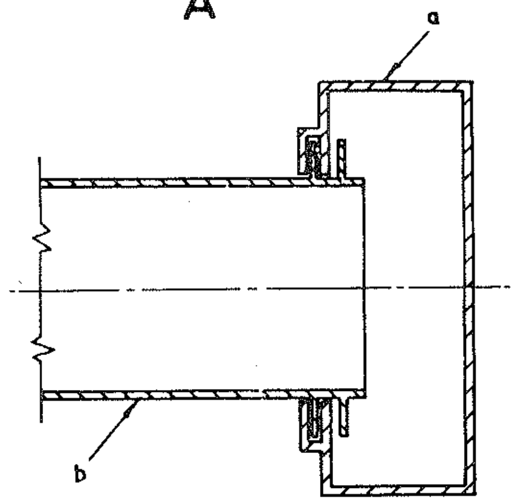
A sealing arrangement at either end has to be provided between the rotating shell and the stationary end feed box forming the material feed and the hot inlet and exhaust gas openings. These seals act to prevent the leakage of air into the cylinder or leakage of hot gases out of the dryer. These seals (Fig. 16.8) can be constructed (A) by use of a friction brake lining attached to the rotating shell and pressed by a spring on the stationary wall of the end box, (B) by use of a labyrinth with intermeshing projections and (C) by use of cloth fastened to the end box and pressed against the rotating shell by a metal band. Whichever type of seal is used, it is important that the dryer shell should have an absolute minimum of eccentricity in its construction.

Heating of the material is done by passing flue gases, or superheated steam or preheated air, which enter at the lower end and leave through an exhauster at the upper end. The flow is thus counter-current to the flow of the material. When the material is sensitive to heat, the flow may be concurrent. In some rotary dryers, steam heated tubes are placed along the entire length of the dryer (Fig. 16.9).

The steam enters through a central inlet pipe and is distributed to all tubes through a header. The condensate from each tube is discharged into a common ring placed at the end

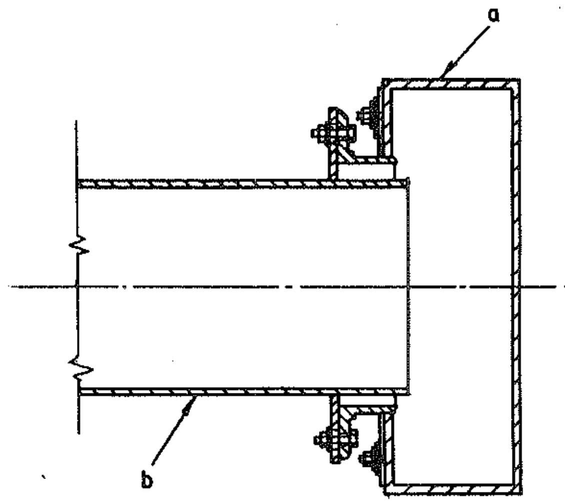


A



B

Fig. 16.8 Sealing arrangements
(A) friction type seal (a) feed box (b) dryer shell
(B) labyrinth type seal



C

(C) Flexible cloth rubbing seal

of the dryer. The ring is concentric with the shell. The tubes are loosely supported in a plate to allow for free expansion.

The rotating shell diameter varies between 0.3 and 3 m, while the length of the shell may range between four and ten times the diameter. The shell is generally 6 to 8 mm thick, and is made as one piece. It may be fabricated from mild steel, stainless steel clad or lined. In order to support the shell, two rings, known as tyres (girth rings), are fixed to the shell at two intermediate positions. The tyres are mounted on tyre bands which are part of the shell wall, and are held in place by retaining blocks. On relatively light dryers, the tyres may be hollow, but for heavy loads, the tyres are solid and are additionally supported by filler blocks, which are metal bands surrounding the tyre band.

Tyres are made to rest on bearing wheels, while the inclined rotating shell is kept from sliding off the bearing wheels by

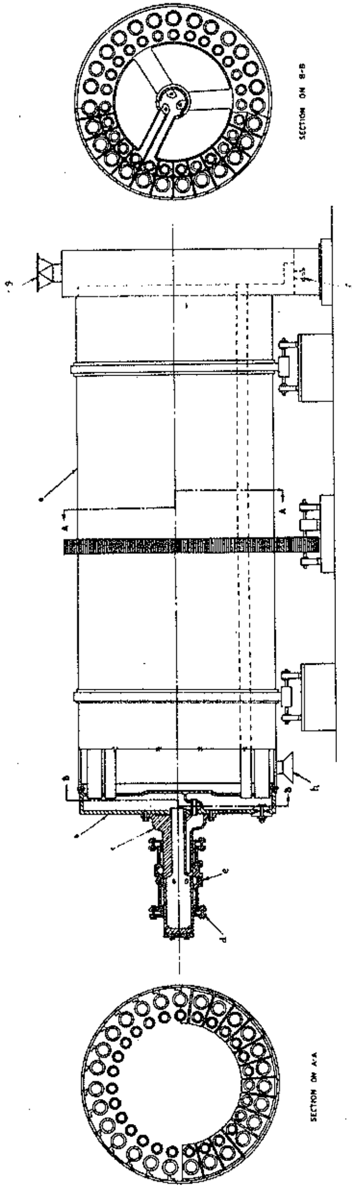


Fig. 16.9 Steam heated rotary dryer (a) shell (b) steam manifold (c) steam neck (d) gland (e) steam inlet (f) condensate (g) feed hopper (h) discharge hopper

'thrust wheels'. The entire assembly of bearing and thrust wheels is mounted on a bearing wheel base (Fig. 16.10). The bearing wheels are supported by means of pressure lubricated roller bearings on stationary shafts. The thrust wheels which roll against edges of the tyres are also supported by pressure-lubricated ball bearings on fixed shafts. Tyres and supporting bearing wheels should be of ample face width and diameter and should be of machined cast steel.

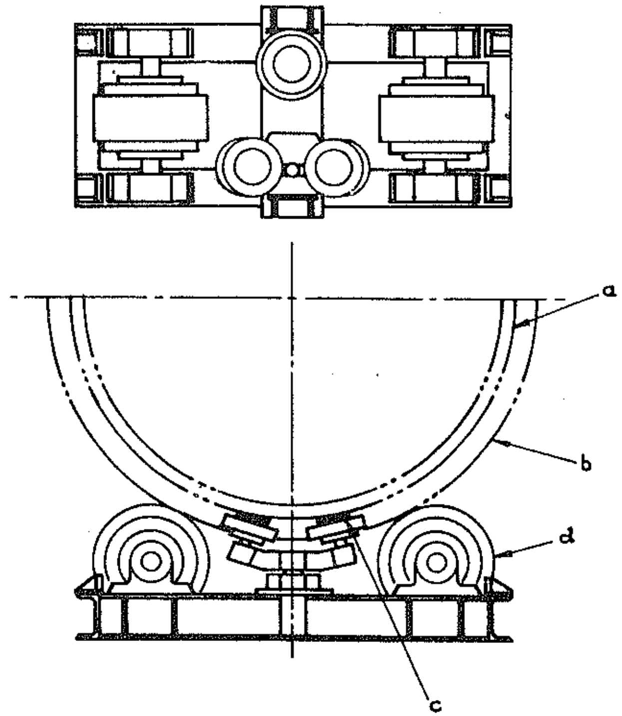


Fig. 16.10 Supporting wheel arrangement
(a) dryer shell (b) tyre (c) thrust wheel (d) bearing wheel

A girth spur gear wheel is fixed to the shell near one of the tyres and is driven by a pinion. The pinion is run by an electric motor with a reduction gear. The pinion and the gear are designed to transmit torque required to overcome friction between the tyres and wheels, and for accelerating the shell. The shafts are designed to transmit frictional torques.

The details of gear and shaft design are indicated in sections 5.8 and 5.2. The total power required can be determined by taking into consideration frictional power, power for acceleration and the loss of power in the drive system.

In double shell dryers, the hot gases first pass down a central tube coaxial with the dryer shell and return through the annular space between the tube and the shell, the material being supplied to the annulus. The material is heated by direct contact with the returning gases in the annulus and also by conduction from the hot central tube.

The rotary louvre dryer is similar to a through circulation rotary dryer. The heating medium is admitted through longitudinal channels formed by a series of radial plates fitted to the inside of the horizontal rotating shell (Fig. 16.11).

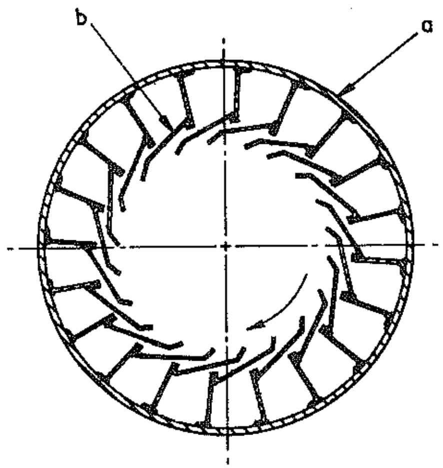


Fig. 16.11 Arrangement of louvres (a) shell (b) louvres

The channels diminish in depth towards the discharge and are covered by tangential louvres which overlap, forming an inner drum designed to provide a passage for the heating medium to the inside of the drum. The louvres also help to prevent the charge of material dropping in the channels. The material travels to the discharge end with a spiral motion forming a compact bed. Heating medium is admitted only

to those channels whose louvres are covered by the bed of the material. The heating medium passes through the bed of material before being exhausted through the vane discharger into an exhaust head situated at the discharge end of the dryer.

16.4.4 Film Drum Dryers

These dryers are operated either under atmospheric conditions or under vacuum. In the latter case, the dryer is enclosed in a vacuum-tight chamber. They have a single-drum, double-drum or twin-drum arrangements. The feedstock is supplied continuously to the effective drying surface of the drums and the dried product is removed by scraper knife. In a single-drum dryer only one drum is used along with a scraper knife assembly. The double-drum dryer has two drying drums with a common drive and common feed arrangement, the latter being formed into a trough above the nip between the two drums by use of spring loaded end plates. The twin-drum dryer consists of two single-drums with a common drive, with generally two separate feed arrangements and two scraper knife assemblies. The direction of rotation of the two drums is almost invariably downwards towards the nip between the cylinders. The clearance at this point is adjustable by changing the position of one drum relative to the other.

The material is fed by various arrangements. Pasty materials are fed by feed rollers from top. Viscous materials are fed at the nip between drums. Slurries are normally fed by submerging the drum partially in a trough. In some arrangements instead of submerging the drum, provision of rollers or sprays is made in the trough. Fig. 16.12 shows a trough with spray feeding for a drum.

The drum, which may be of 1/2 m to 2 m diameter and 1 m to 4 m length can be fabricated from a plate or cast to the required shape. The materials used are cast iron, bronze, chromium plated steel or stainless steel. The surface of the drum should be hard and smooth. The drum is heated internally by steam and has, therefore, to withstand internal pressure. The ends of the drum are closed by flanged type of covers, with trunnions supported on bearings. The steam is

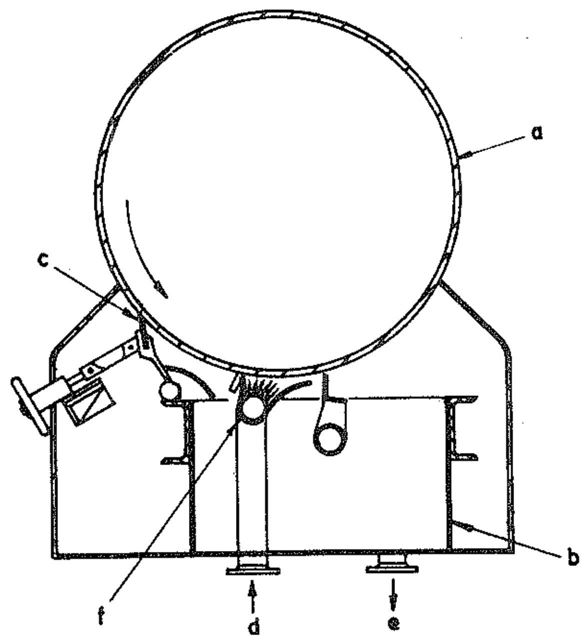


Fig. 16.12 Drum dryer knife
(a) drum (b) feed trough (c) scraper (d) feed inlet
(e) feed outlet (f) feed spray

supplied through a pipe which passes through the centre of the trunnion. The condensate is discharged by means of a scoop or syphon through the second trunnion by means of a rotary joint (Fig. 16.4).

Drums are rotated at speeds in the range of 3 to 20 rpm. As the drum rotates a film of about 1 to 3 mm thickness is formed, which is scraped by doctor knives. Three types are generally used, stationary single-bladed, oscillating single-bladed and multiple adjustable abutting or overlapping bladed. For materials which form a film of dried material, that is easily removed by a knife, a single knife extending the whole length of the drum, with adjusting screws at frequent intervals is satisfactory. The knife is of silicon-carbon steel with fairly small thickness. Sectional knives are usually of higher thicknesses. Each section is adjustable by hand wheels. For

materials which are extremely abrasive the knives are often tipped with abrasion resistant metal. Where corrosion is likely to occur or where metal contamination is to be completely eliminated, the knives may be of suitable plastic material or even toughened glass. Knives are set at an appropriate angle at the point of contact. A sheet metal or stainless steel chute plate is fitted to deflect the dried product away from the knife adjustment gear, where this is located below the knife assembly.

The drum is rotated by a gear and pinion. The pinion is driven through a belt or chain or through a reduction gear from a motor. A variable speed drive may be provided for regulating the speed of the drums. The design of the components is similar to that of a drum filter (see section 15.2.21).

Reading References

Foust, A.S., *Principles of Unit Operations*, John Wiley and Sons, New York (1960).

Salon, C.E., *Drying Systems and Equipment*, Chem. Eng. 74 (13), 167 (1967).

Nonhebel, G. and Moss, A.A.H., *Drying of Solids in the Chemical Industry*, Butterworths, London, (1971).

Perry, J.H., *Chemical Engineers' Handbook*, 5th Edition, McGraw-Hill, New York (1973).

Parker, N.H., *Aids to Dryer Selection*, Chem. Engg., 70 (13), 115 (1963).

Belcher, D.W.D., *et al*, *Design and Uses of Spray Dryers*, Part I, Chem. Eng., 70(20), 83(1963) Part II, Chem. Eng. 70(21), 201 (1963).

A.I. Ch. E., *Equipment Testing Procedure Dryers* (*Rotary Continuous Direct-heat Dryers*).

William, Gardner A., *Industrial Drying*, Leonard Hill Books, London (1971).

Keey, R.B., *Drying Principles and Practice*, Pergamon, London (1972).

CHAPTER 17

Process Hazards and Safety Measures in Equipment Design

17.1 Introduction

The previous chapters dealt with selection of materials, analysis of stresses, determination of strength and the design procedure adopted in equipment design. If the design procedure is strictly adhered to, the selection of the material is appropriate, and the fabrication and final inspection are carefully carried out, it is expected that the equipment should not fail under service conditions. However, failures in service conditions or environments are not entirely eliminated. Singly, or in combination, many specific causes can lead to these failures. Certain safety measures can help to prevent such failures.

17.2 Hazards in Process Industries

Modern technology has been quite successful in developing tailor-made chemicals. However, this effort has also introduced some additional problems, since manufacturing and handling experience is frequently inadequate to deal with the question of hazards. The ever increasing production of flammable organics, the rush to bring new products from the laboratory into full scale production, the problem of familiarisation with a stream of new technology have all extended the probabilities of hazards. Toxic and corrosive chemicals, fires, explosions, falls and faulty mechanised equipment are the major hazards encountered in the operation of plants in chemical industries. The design engineer must be aware of these hazards and must make every attempt to present a design which

provides the maximum protection for the plant personnel and the minimum chance for the occurrence of accidents.

17.3 Analysis of Hazards

An analysis must be made of the probable sources of hazards. It is also necessary to identify and evaluate the hazard potential. Where practical, recognised hazards should be eliminated, but such hazards can be tolerated if proper procedures are used to guard against accidents. The sources of hazards can be divided into two categories, namely, materials hazards and process hazards.

17.3 1 Material Hazards

These are either due to the nature and properties of the materials or the reactions the materials are likely to undergo during processing.

(1) Combustible solids, liquids or gases.
(2) Explosive or blasting agents.
(3) Radio-active materials.
(4) Nuclear materials.
(5) Oxidising materials.
(6) Materials that react with water to produce a combustible gas.
(7) Materials subject to spontaneous heating, spontaneous polymerisation, or explositive decomposition.
(8) Corrosive materials.
(9) Toxic matcrials.

17.3.2 Process Hazards

These hazards might arise due to several factors, some of which are enumerated here.

(1) Storage handling and physical change of materials such as transfer of flammable liquids or mixing in open containers, etc.

(2) Exothermic or endothermic reactions, where there is a strong possibility of reaction getting out of control.

(3) Low pressure processes operating at sub-atmospheric conditions so that air or contaminants leak into the system.

(4) High pressure processes, which may involve pressures of 200 kg/cm^2 or above. A hazard results from the potentially large expansion of fluid released to the atmosphere from elevated pressures.

(5) High temperature processes, which may involve temperatures above flash point, boiling point or auto-ignition.

(6) Dust or mist explosion hazard may arise in processes which involve handling of materials that could create dust or mist.

(7) Sparking of electric motors, switches and alarms.

(8) Electrostatic charges.

(9) Hazards arising out of equipment and/or instrument failure.

In many cases the equipment or instrument fails to function due to instrument air failure, thermocouple burn out, loss of electric power, steam or cooling water failure, possible reversal of flow, plugging of lines or equipment, loss of pressure etc.

17.4 Safety Measures

In every process industry a wide variety of preventive and protective safety measures must be provided. Such measures can be of two types : (1) measures to minimise or avoid mishaps (2) measures which will limit the extent of damage if the situation gets out of control. Some of the measures can be considered as basic preventive and protective features. Others can be recommended as minimum features, while some others are specific preventive features appropriate to each individual situation or equipment.

Prior to use, the properties of each material in a proposed process should be examined to determine if it is unstable at normal temperatures, or if it can explode from mechanical shock, from exposure to high temperature while confined or ignited. The flammability of each chemical should also be determined. Similarly, precautions must be taken while handling toxic chemicals. Radioactive materials should not be used unless appropriate storage, decontamination, testing and disposal facilities are available.

Some of the basic preventive and protective measures are : adequate supply of water for fire protection, over pressure relief devices, segregation of reactive materials in process lines and equipment, grounding of electric equipment, safe location of auxiliary electric gear, drainage from spills, fire fighting water from hose nozzles or sprinkler heads, insulation of hot surfaces that heat to within 80% of auto-ignition temperature, proper building and equipment layout, limitation on glass devices in flammable or hazardous service. Certain protective features which are recommended as minimum, where hazards are limited, are : protection of supports for equipment with fire-resistant materials, water spray to equipment and structure, special 'fail-safe' or remote operated valves and instruments, quick and effective removal of blowdown or spill material, instruments and techniques to ensure that explosive mixtures do not form inside process equipment, monitoring systems which actuate protective systems such as sprinkler systems or ventilation fans, building ventilation, segregating hazardous operations and installation of barrier walls.

Specific preventive measures can be any of the above measures suggested or additional measures to suit the material or process. For instance, in case of high temperature process it is essential to provide (1) instrumentation and/or special devices to minimise flow of flammables (2) combustible gas monitor with alarms (3) combustible gas monitors that turn on deluge system or safety shut down equipment (4) special vent and dump system.

17.5 Safety Measures in Equipment Design

So far a large number of safety measures for preventing or controlling hazards were indicated. Some of these are relevant in equipment design problems. Items of equipment considered in this book are either pressure vessels such as reactors, heat exchangers, etc., or those involving rotary motion such as filters, agitators, etc. Some of the important safety measures which need consideration during the design stages of these items of equipment are discussed in some detail.

17.5.1 Material of Construction

Materials available for construction of equipment, their properties, problems of corrosion and protective coatings have been considered in chapter 2. In some cases, process conditions vary a good deal, which makes the choice of the material difficult. Too often the choice of material is based on similar previous experience, but even small differences in the constituents in fluid streams may make considerable difference to the material to be used. Many vessel failures, not all catastrophic of course, are attributable to a bad choice in the material. A critical analysis of the processing conditions and material handled must be made to ensure safe operation.

17.5.2 Precautions in Design and Construction

A wide variety of precautions is necessary to ensure the safe working of an equipment and prevent failures which may result in hazards, and accidents. These will vary according to the type of equipment. A well designed piece of equipment will have safety and loss prevention features built in The following design considerations are important to prevent failures :

1. Unit reliability.
2. Ease of operation.
3. Unit flexibility.
4. Provisions for future expansion, inspection and maintenance.
5. Adequate emergency shutdown facilities.
6. Equipment standardization for rapid replacement.
7. Design to anticipated pressure range—with overpressure controls.
8. Design to anticipated temperature range—with over-temperature controls..

It is not possible to give a list of precautions for each individual unit of equipment. Some of them are indicated below.

17.5.2.1 PRESSURE VESSELS

(1) The design and construction of pressure vessels and storage tanks should follow Indian Standards codes and vessels should be tested at 1.5 to 2 times the design pressure.

(2) Attempt should be made to keep the vessel as simple as possible, consistent with the fulfilment of its function. It should not be over burdened with manways and inspection openings for which there is only a remote chance of usage.

(3) Careful attention should be given to any welds made on a vessel, particularly if it is thick.

(4) If the vessel is to work under pressure cycling, system changes, vibrations or similar factors which are likely to create fatigue conditions, it is essential to check the fatigue strength.

(5) Careful attention should be given to design criteria indicated under section 6.7.2 of chapter 6 on pressure vessels.

(6) Flange joints must be leak proof. This can be ensured by checking the gasket position, flange faces and bolt holes.

(7) Pressure relief devices should be provided on all pressure vessels.

17.5.2.2 HEAT TRANSFER EQUIPMENT

Equipment such as heat exchangers, vaporisers, reaction kettles, furnaces, require some method of heating, which may be directly fireed with the help of fuel, electric heating, heat transfer media such as steam, or organic fluids. For such units special precautions should be taken, which would prevent overheating, fire and/or explosion. The following precautions in the design and construction of such equipment will help safe operation.

(1) Sufficient heating surface must be provided to avoid excessive rate of heat input per unit surface, so as to minimise excessive liquid film temperatures.

(2) The heat absorbed by the tubes must be removed by the circulating fluids and the liquid film coefficient inside the tubes must be sufficiently high to prevent an excessive temperature rise through the liquid film.

(3) Inspection openings must be sufficient in number and location to permit complete periodic inspection of the vaporiser.

(4) Vent valves should be provided at all high spots in the equipment.

(5) Liquid phase systems must include an expansion or surge tank.

(6) Heater tubes must be tightly secured to headers and vapour drums.

(7) In case of furnaces, combustion space should be designed to avoid flame impingement on tubes.

(8) Pressure relief devices must be designed for high temperatures.

(9) Care should be taken to allow for stresses due to thermal expansion.

(10) Appropriate choice of insulation must be made consistent with the material handled by the equipment.

17.5.2.3 EQUIPMENT INVOLVING ROTARY MOTION

This type of equipment involves a mechanical drive system which should be protected by guards. Bearings should be well lubricated and cooled, if necessary, to reduce temperature.

17.5.2.4 EQUIPMENT INVOLVING ELECTRICAL ENERGY

All electrical installations require safety features which are controlled by Electricity Regulations. IS 5571-(1970) is a guide for selection of electrical equipment for hazardous areas. IS 5572-(1970) gives classification of hazardous areas for electrical installations.

(2) The entry of vapours or gases into unpressured electrical equipment cannot be prevented even by the tightest of gaskets. To avoid an explosion from being extensive the enclosure of the equipment must be strong enough to contain explosion, and its joints must be long enough and clearances small enough so that flame will not propagate to the outside. Such equipment is said to be explosion proof.

(3) Purging or pressuring enclosures with clean air or inert gas will help to prevent the entrance of hazardous dust.

(4) Static electrification can be prevented by earthing, static eliminators and effective surface treatment of equipment.

17.6 Pressure Relief Devices

One of the most important safety devices used on process equipment, is for the prevention of failure from overpressure. The more common causes of overpressure are external fire, closed outlets, liquid expansion, failure of preflux, cooling water, electric power and heat exchanger tubes. IS 2825 gives the conditions under which relief devices are to be used, their types, capacities and installation requirements.

The available relief devices generally fall into six broad categories.

(a) *Safety valve*—Actuated by static pressure upstream of the valve, this device is characterized by rapid full opening or pop action. It is used for steam, gas or vapour service.

(b) *Relief valve*—Unlike the above, this device opens in proportion to the increase in pressure over the opening pressure and is used primarily for liquid service.

(c) *Safety selief valve*—This is suitable for use as either safety or relief valve depending upon application.

(d) *Rupture disc*—This device normally consists of a metal disc and a vacuum support when required, which is held between two special flanges.

(e) *Relief valve-rapture disc combination*—Here the disc is either used in series with a relief valve or is used as a secondary relief device in parallel with relief valve.

(f) *Pressure-vacuum combination relief valve*—This device is generally used on atmospheric-type storage vessels. It is frequently designed to relieve both pressure and vacuum.

17.6.1 RELIEF VALVE

A relief valve should be of sufficient capacity to discharge the maximum quantity of the fluid contained in the vessel without permitting a rise in the pressure vessel of more than 10% above the set pressure, when it is discharging. Equations to determine vapour relief and liquid relief are given in IS 2825-(1969).

No matter how carefully the valve has been sized and selected, poor practice in handling and installation can result in completely unsatisfactory performance. Fig. 17.1 shows a valve mounted on a pressure vessel without a shut-off valve while Fig. 17.2 shows the arrangement with shut-off valve. The pressure drop in the nozzle and valve together should not be more than 3% of set pressure. The long-radius elbow should

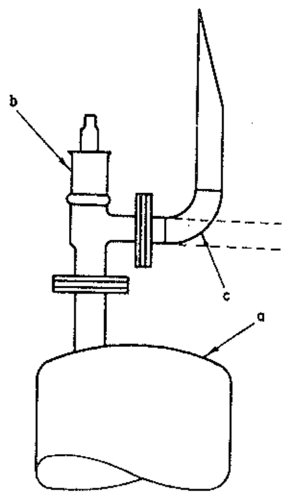


Fig. 17.1 Relief valve mounted on pressure vessel (a) vessel (b) relief valve (c) elbow

be supported. In all the above cases no incoming processing piping connection should be made between the vessel and the valve.

Fig. 17.3 shows alternate relief valve positions on a fractionating column. Location of the valve can influence the size of the relief valve as well as influence the pressure available for the relief system design. Location 'A' is usually preferred when the relief valve is tied up in a closed relief system because of maintenance convenience, shorter length of line to the relief header, and cooler relieving temperature when air fan cooler is used. Pressure drop through condenser must be calculated as

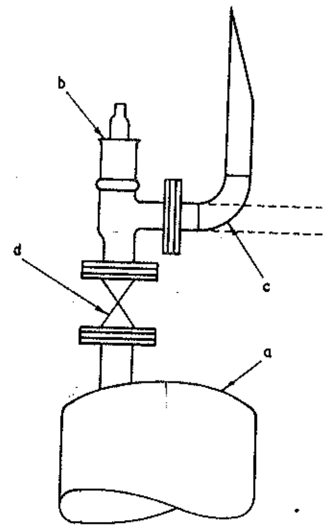


Fig. 17.2 Relief valve with shut-off valve (a) vessel (b) relief valve (c) elbow (d) stuff-off valve

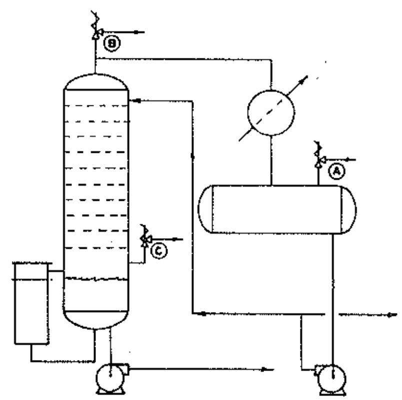


Fig. 17.3 Relief valve positions on a fractionating column

also the relieving condition, to assure proper setting of relief valve. Location 'B' is preferred when the column is to be relieved to the atmosphere. This practice is preferred when relieving low molecular weight gases, or when dispersion is such that the vapour is out of the flammability range before it reaches an elevation low enough to be dangerous. Location 'C' is used on low pressure columns designed for the same top and bottom pressure. This is to account for the pressure drop across the column. All relief valve installations should be built without pockets to avoid trapping liquids.

From a safety standpoint, any relief valve which may discharge flammable liquid should be connected to a closed system. Flammable vapour may be discharged at any point which will not endanger personnel or equipment, however, it should not condense in appreciable quantities. Regardless of whether the relief valve discharges to the atmosphere or to a closed system, safe plant design requires a means of rapidly depressurising the plant and in most instances safe disposal of the contents of the vessels and equipment either to storage or elsewhere under emergency conditions. If there is any probability of liquid discharge into the flare system, a knock-out drum should be provided adjacent to each process unit or a group of units.

17.6.2 Rupture Disc

This device is designed to burst at some predetermined pressure in chemical equipment. This may be used as a primary or the sole relieving device, but once the disc relieves pressure, the equipment is open to atmospheric pressure until the disc is replaced. With highly toxic, poisonous or corrosive materials, it is necessary to release the materials into a vent surge or flare header system. The advantage of the rupture disc is that it will not allow leakage under normal conditions. It is quite common to install a rupture disc upstream of a relief valve. Under normal conditions, the rupture disc is sealed tight and protects the relief valve. If the maximum allowable pressure is exceeded, the disc will break and the relief valve will start to relieve the pressure. As the pressure drops the valve will shut and reclose the process.

The diameter of the rupture disc is calculated as follows :

For Vapour

$$d = 0.21 \sqrt{\frac{W}{p}} \sqrt{\frac{0.56\ T}{M_w}} \tag{17.1}$$

For Steam Dry and Saturated,

$$d = 0.52 \sqrt{\frac{W}{p}} \tag{17.2}$$

Superheated

$$d = 0.52 \sqrt{\frac{W(1+0.000358\ T_s)}{p}} \tag{17.3}$$

Wet

$$d = 0.52 \sqrt{\frac{W(1-0.012\ y)}{p}} \tag{17.4}$$

For Liquid

$$d = 0.5031 \sqrt{Q}\ 1.94 \sqrt{\frac{SG}{p_1}} \tag{17.5}$$

where,

d — Minimum rupture disc dia in cm
M_w — Molecular weight
p — Relieving pressure kg/cm² absolute including allowable accumulations (10% in normal conditions ; 20% in fine conditions)
p_1 — Relieving pressure kg/cm² gauge, including allowable accumulation
Q — Relieving rate, lit/min
SG — Liquid specific gravity, where for water SG = 1.0
T — Relieving temperature K (273+°C)
T_s — Degrees of superheat °C
W — Relieving rate kg/hr
y — % moisture.

17.6.3 Vents and Flame Arresters

Vents are provided on storage tanks, heat exchangers, vessel jackets, etc. The sizes of breather vents used for tanks are given in Table 7.2. Emergency venting is required for cases of excessive internal pressure within the tank from liquid ebullition due to fire exposure. In some cases, the number and size of the venting devices per tank will be governed by pressure requirements, whereas in other cases this will be decided by vacuum requirements. If the tank contains a volatile liquid, a breather valve, which contains both a pressure and vacuum pellet set to open at a specified setting, is fitted to vent opening to minimise evaporation losses to the atmosphere.

Vent valves are not regarded as providing the required protection against fire and explosion to tank contents ; consequently, flame arresters should be employed in conjunction with breather valve when necessary. A flame arrester is placed at or near the outlet end of a vent pipe to protect the tank contents by preventing the propagation of flame. In many cases normal venting requirements may be inadequate to handle the abnormal vapour efflux generated by heat from an exposure to fire. Consequently, some form of safety device that will relieve excessive internal pressure is necessary. These may consist of additional vents, self-closing gauge hatches, lifting type manhole covers, etc. The total emergency venting capacity for any specific liquid can be approximately expressed as

$$V = \frac{31.8Q}{L\sqrt{M}} \tag{17.6}$$

where V — m^3/hr
Q — kcal/hr
L — Latent heat kcal/kg
M — Molecular weight.

Emergency vents are usually set slightly higher than the pressure settings on breather vents, but not higher than the pressure the tank can safely withstand.

Reading References

Mead, W.J., *The Encyclopedia of Chemical Process Equipment*, Reinhold, New York (1964).

Thielsch, H., *Defects and Failures in Pressure Vessels and Piping*, Reinhold, New York (1965).

Jenett, E., *Components of Pressure Relieving Systems*, Chem. Eng., Part I, **70** (14) 125 (1963) and Part II, **70** (17), 151 (1963)

Albrecht, A.R. and Seifert, W.F., *Accident Prevention in High Temperature Heat Transfer Fluid Systems*, Loss Prevention, **4** (67), CEP Technical Manual (1969).

Walker, J.J., *Sizing Relief Areas for Distillation Columns.* Loss Prevention **6** (104), CEP Technical Manual (1969).

Spiegelman A., *Risk Evaluation of Chemical Plants*, Loss Prevention **3**, (1), CEP Technical Manual (1969).

Peterson, P. and Cutler, H.R., *Explosion Protection for Centrifuges*, Loss Prevention **7** (88) CEP Technical Manual (1973).

McFarland I., *Safety in Pressure Vessel Design*, Ammonia Plant Safety, **12** (43), CEP Technical Manual (1970).

Keyser, D.S., *Rupture Disc Selection*, Loss Prevention **6** (82), CEP Technical Manual (1972).

Simon, H., and Thomson S.J., *Relief System Optimization*, Loss Prevention **6** (74), CEP Technical Manual (1972).

Yelland, A.E.J., *Design Considerations When Assessing Safety in Chemical Plant*, The Chemical Engineer **201**, CE 214 (1966).

Burklin, C.R., *Safety Standards, Codes and Practices for Plant Design*, Part I—*Design Considerations*, Chem. Eng. **79** (22), 56 (1972).
Part II—*Safety Standards, Codes and Practices for Plant Design*, Chem. Eng. **79** (23), **113** (1972).

Le Vine, R.Y., *Electrical Safety in Process Plants*, Chem. Eng. **79** (9), 51 (1972) ; **79** (9) 59 (1972).

Russell, W.W., *Safety in Flange Joints*, Loss Prevention **8** (100), CEP Technical Manual (1974).

J.S., Rearick, *How to Design Pressure Relief Systems*, Part I—*Devices Codes and Causes of Overpressure*, Hydrocarbon Processing, **48** (8), 104 (1969).
Part II—*Sizing Relief Systems and Disposal of Vented Material*, Hydrocarbon Processing, **48** (9) 161 (1969).

Proceedings of the Second Symposium on Chemical Process Hazards with Special Reference to Plant Design. The Institution of Chemical Engineers Symposium Series 15 (1963).

Fire and Explosion Safety and Loss Prevention Guide. Hazard Classification and Protection. CEP Technical Manual—(1973) American Institute of Chemical Engineers.

CHAPTER 18

Fundamentals of Computer Aided Design

18.1 Introduction

Problems in engineering design are becoming complex due to several factors which influence the design procedure. Efforts are being made continuously to effect changes in the design procedure, with a view to better utilisation of materia,ls improvement in efficiency and overall economy. In equipment design, theories relating to chemical process principles, mechanics, strength of materials form the basis of a design procedure. These theories usually give rise to a set of equations, which has to be solved to arrive at a satisfactory result. Assumptions have to be made and certain conditions have to be satisfied. According to the variation in operating conditions, solutions have to be obtained for the same set of equations. Likewise if the number of units of each equipment is large, the procedure has to be repeated to suit each individual unit. The entire process in many cases is time consuming, involving checking and rechecking of results to avoid errors. Similarly, to satisfy the different controlling parameters, a trial and error procedure or the method of iterations is adopted to arrive at a satisfactory solution. In this, use of mechanical calculators and electronic computers have helped to yield accurate results with great speed. Design engineers must get fully acquainted with the operation and application of these computing methods. Successful application of the computer in the areas of engineering design requires a careful blend of creativity on the part of the engineering designer, suitably powerful computing hardware and some sophisticated tools to make communication between man and machine simple, direct and practical.

18.2 Electronic Computers

The first general purpose computer was developed in the early 1940's. Present-day computers range from small engineering machines and special purpose computers to very large commercial and scientific computing systems. They are widely used for the solution of problems in many fields. Their use is based on their ability to operate at great speed, to produce accurate results, to store large quantities of information and to carry out long and complex sequences of operations without human intervention. Computers can perform essentially five functions. They can receive information, write out information, store information and perform basic arithmetic and logic operations.

18.2.1 Analog and Digital Computers

The two major types of computers are the analog and the digital computors. Each type has its particular characteristics, which give it an advantage in the solution of some engineering and scientific problems. An analog computer sets up a model of the physical system or problem to be solved by using electrical analogies of the actual magnitudes and dimensions of the variable and parameters. For instance to perform an addition problem, the two voltages may act as an analogy of two quantities, and these voltages would be applied to an adder circuit. The digital computer on the other hand deals exclusively with the manipulation of pure numbers and is, therefore, ideally suited for the solution of numerical problems.

18.2.2 Digital Computer Characteristics

The digital computer has certain distinctive characteristics:

(*a*) *Memory*

This represents the ability of a digital computer to store numbers, alphabetic characters, and other symbols in the internal memory facilities, consisting of thousands of tiny magnetic cores arranged in a cubic matrix. These cores are magnetized in a definite direction depending on the information to be stored. The process requires only a few billionths of a second. Great volumes and data that are used less frequently are stored in an auxiliary memory, which consists of magnetic drums or discs.

(*b*) *Stored Program*

For carrying out any calculation a program of instructions has to be prepared giving details of the sequence of operations. This program can be stored in the computer. The program of instructions may fulfil a variety of functions.

(1) Data to be brought into main memory from an external storage memory.

(2) Arithmetic operation to be performed on selected numerical data.

(3) Logical test to determine the next step of the program.

(4) Results of calculations to be sent from internal storage to an output recording device, such as a printer, typewriter or graph plotter.

Certain digital computers merely act as data processors, which can be worked according to the convenience of the operator, while others are used as control computers, which control a process by receiving information, performing calculations and issuing the necessary control instructions within the stipulated period of the process.

18.3 Computer Application

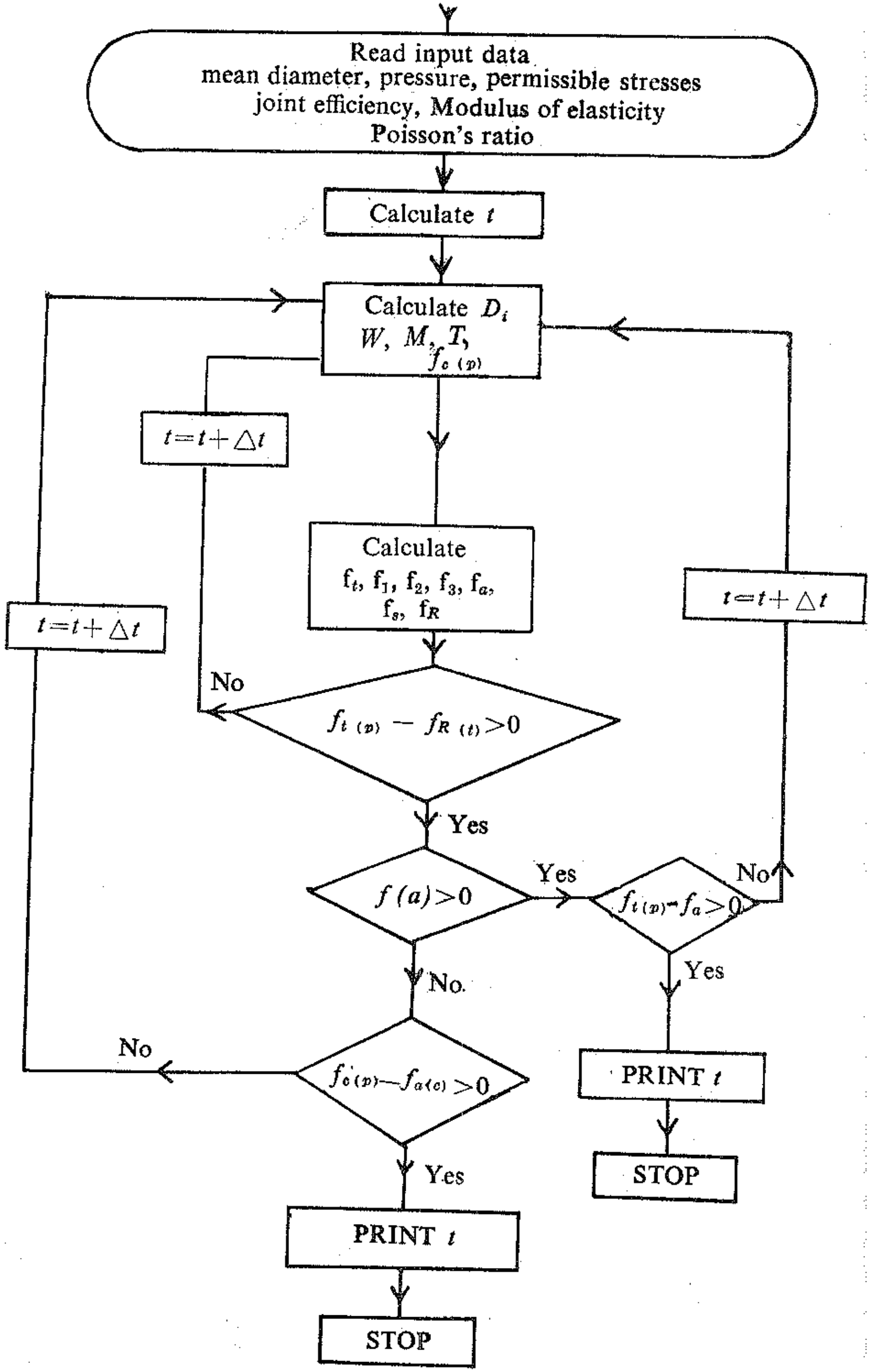
In general computers are capable of solving two broad categories of problems

(a) Straight forward or explicit

(b) Iterative or implicit

In the first category problems the method of solution merely consists of carrying out a sequence of calculations. The computer simply does the lengthy or complex calculations which would be laborious to tackle manually. For instance a problem can be expressed in explicit form as

$$x=f(u, v, w, y, z) \tag{18.1}$$



The value of x can be obtained by substituting known values of u, v, w, y and z and making the calculations either manually or by the use of computer.

In the second category, the relationship between variables cannot be expressed in explicit form leading to a simple solution, but the unknown variable is inextricably interwoven with the known factors on both sides of the equation. Thus, an implicit relationship can be expressed as

$$x = f(u, v, w, x, y, z) \tag{18.2}$$

The unknown x cannot be obtained directly, because it forms a part of the known factors on the right side of the equation. Such problems involve a process of trial and error. A likely estimate of x is made and is tested to satisfy the equation. If the estimate is wrong progressively better estimates or iterations are made until an approximate solution is obtained. A variety of such complex problems can be solved conveniently by the computer very rapidly.

18.4 Problem Solving Procedure

The basic steps adopted for solving a problem by a computer are essentially the same as those which a programmer would follow, if he were to solve the problem manually. However one should take into account the special features, advantages and limitations, which the computer possesses. An outline of the procedure of solving an engineering problem by computer is as follows :

(a) *Defining the Problem*—The first step is to list all variables and parameters and decide what combinations of goals the system must satisfy.

(b) *Mathematical Description*—The process is described mathematically and a feasible approach or method of solution is devised. The mathematical formulation of the problem may not be directly translatable to the language of the computer since the computer can only do arithmetic and make simple quantitative decisions. Trigonometric functions, differential equations, integrals, square roots and logarithms must be expressed in terms of arithmetic operations, by the methods of numerical analysis. For instance a sine function may be written in terms of a Taylor series.

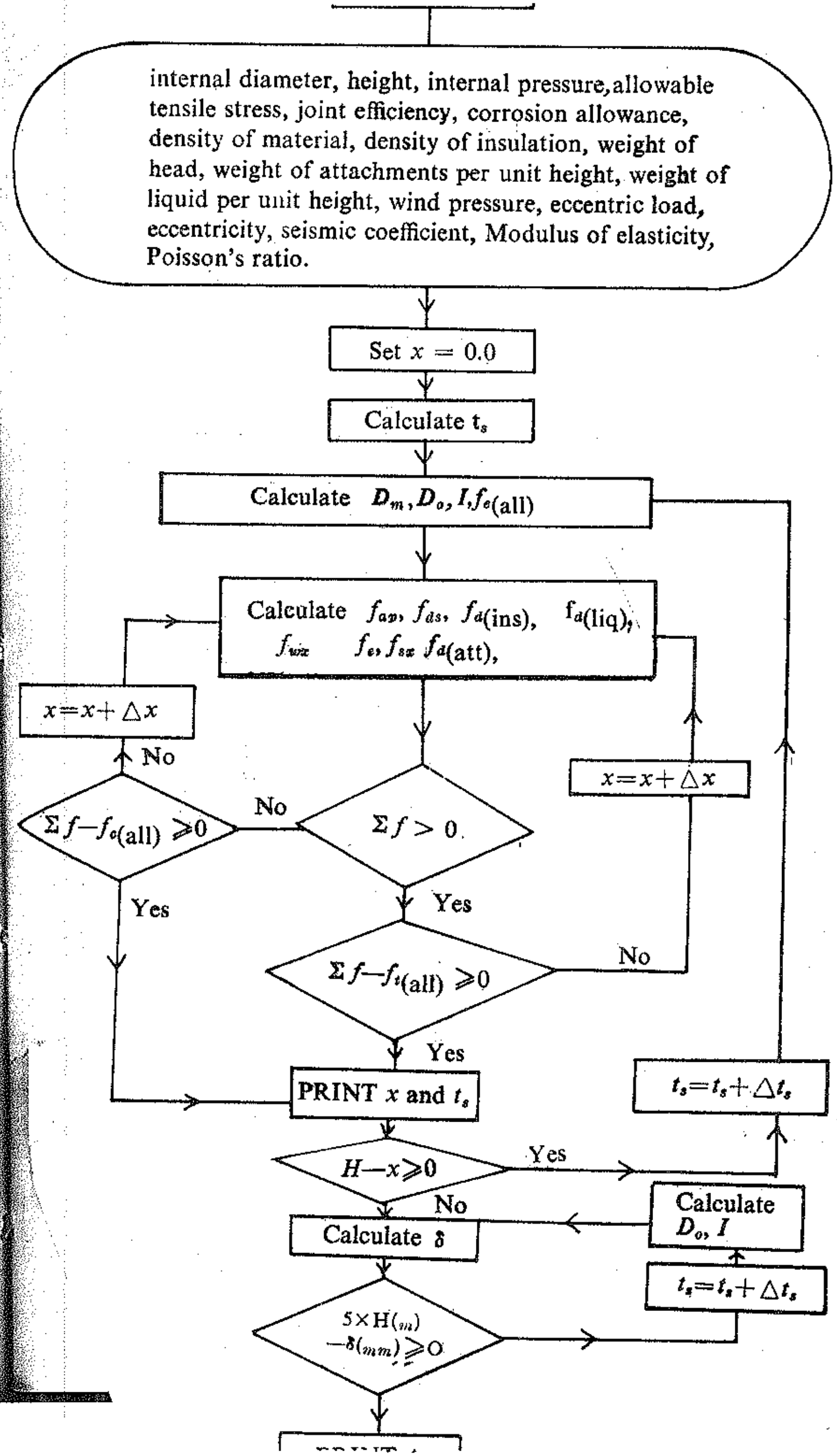
(c) *Programming*—A computer program is a pre-established plan for solving a particular problem or controlling a process. The program defines the problem or process and spells out the method of solution through a sequence of detailed instructions to the computer. The necessary sequence of instructions is written first in the form of block diagrams (flow charts). Each instruction step is then stated in a language that the computer can understand. A variety of computer programming languages are available such as FORTRAN, ALGOL, MAD, etc. For instance FORTRAN provides five basic operations, each represented by a distinct symbol

Addition	+
Subtraction	—
Multiplication	×
Division	/
Exponentiation	**

Once the program is completely encoded, it is punched out on cards, paper tape or recorded on some other medium, and is inserted into the computer storage. When the computer is started, it reads the first instruction and acts on it. The procedure is continued upto the last step.

However, depending on the results of intermediate calculations, the program may instruct the computer, at a certain crucial point to branch to another sequence of instructions, or sub-program, if the predetermined condition occurs. A control unit is provided in the computer to execute the sequence of instructions contained in the program and make a logical decision at certain points of the program, based on the work performed upto the relevant point.

(d) *Operation*—The known data of the problem or process is fed to the computer through a coded prescribed form and recorded. This data, along with the program are stored in a memory unit. Then, as required during the problem, instructions and data are taken out of the memory unit, item by item, and the results are transferred back to the memory unit for later use. The operations within the computer therefore



consist of a continuous flow of information, with the instructions and data shuttling back and forth between storage and processing units. The control unit directs the flow of information and controls the sequence of operations. When the final solution is obtained it is converted back into humanly comprehensible language by an output device, which may be a typewriter, printer, punched card, etc.

18.5 Problems in Equipment Design

Certain problems in equipment design can be solved rapidly with the help of a computer. This may be advantageous, particularly when an equipment is to be designed to satisfy a variety of processing conditions or when a large number of similar units are to be produced. Such problems might arise in the design of pressure vessel components, heat exchangers, distillation columns, agitators, etc. A few problems to illustrate the computer programs are presented below.

18.5.1 Problem

To determine the shell thickness of a cylindrical pressure vessel under internal pressure and at a stipulated temperature.

Solution

The thickness is determined with the application of equations 6.3 and 6.5 to 6.12. For a satisfactory design the conditions to be satisfied are : (Tensile stress is taken as positive)

(1) *Equivalent stress :* f_R (tensile) $\leqslant f_t$ (permissible) at the stipulated temperature.

(2) *Total stress in the axial direction*

f_a (tensile) $\leqslant f_t$ (permissible) at the stipulated temperature.

(3) *Total stress in the axial direction*

f_a (compressive) $\leqslant f_c$ (permissible) at the stipulated temperature.

f_c (permissible) is calculated from equation 6.12.

18.5.2 Problem

Design the shell of a distillation column working under internal pressure and at a stipulated temperature.

Solution

The shell thickness t_s at the top of the column is calculated from equation 11.5. The axial stresses are calculated at the top of the column, where height $x=0$. By giving increments of $\triangle x$, these stresses are determined upto a height where the allowable tensile stress equals the total axial stresses, by application of equations 11.6 to 11.18. If this height is equal to or greater than the total height of the column, the thickness, t_s, calculated is satisfactory. If, however, the height determined is less than the height of the column, the thickness t_s is given an increment of $\triangle t_s$, and the axial stresses are determined again for additional height.

Stresses in the axial direction can be either tensile or compressive. Tensile stresses are assumed to be positive and compressive as negative. In case the resultant stresses are compressive, particularly on the downward side, elastic stability must be checked by application of the safe compressive stress as per equation 11.21. Column deflection must be limited to a maximum value, in mm, of 5 times the overall height of column in m.

18.5.3 Problem—Shaft Design for a Vertical Agitator

The horse power and speed for the shaft are given.

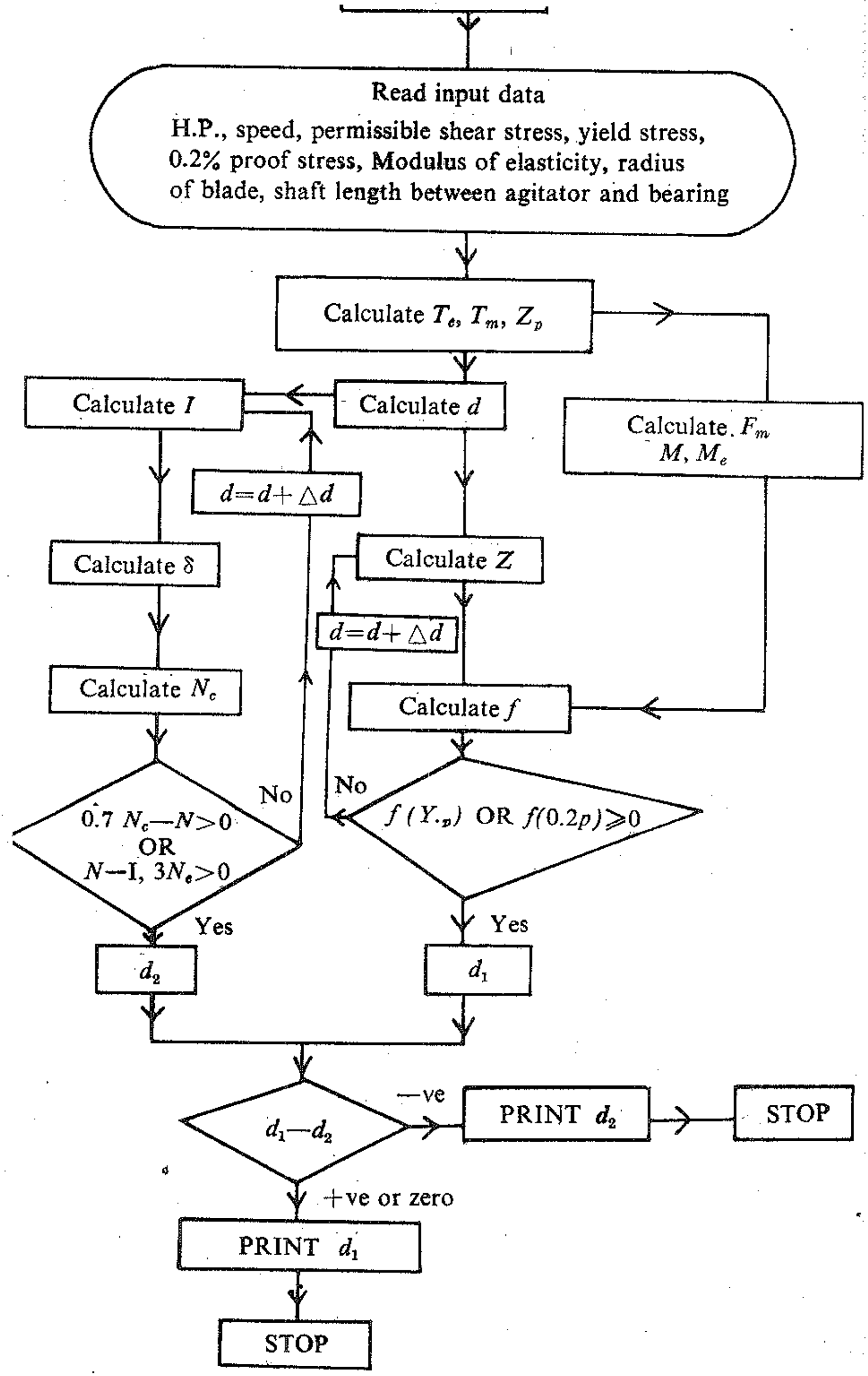
Solution

It is assumed that the horse power required to drive the agitator is already calculated by application of equations 14.1 and 14.2 and by taking into account the losses.

The shaft diameter is determined by the application of equations 14.8 to 14.13. The conditions to be satisfied are:

(1) The shear stress produced due to maximum torque must be within the permissible value.

(2) The stress due to equivalent bending moment should not exceed the elastic yield stress of the material or 0.2% proof stress.



(3) The shaft speed should be below 70% of the critical speed, or above 130% of the critical speed.

The block diagrams illustrated above have to be translated into an appropriate computer language. Details regarding the procedure to be followed for writing the program in a computer language may be obtained from the list of references.

Selected computer programes are now available for pressure vessel design, flange design, nozzle design, vessel stress analysis, heat exchanger tube sheet layout and mechanical design, piping design, etc. A list of such programmes is given in 'Chemical Engineering', 78 (16), 70 (1971).

Reading References

McCracken, D.D. and Dorn, W.S., *Numerical Methods and Fortran Programming*, Wiley International, New York, (1965).

Katz, D.L. et al., *Computers in Engineering*, Design Education; Vol. I to VI, University of Michigan, College of Engineering (1966).

Rosenbrock, H.H. and Storey, C., *Computational Techniques for Chemical Engineers*, Pergamon Press, Oxford, (1966).

Rogers, A.E. and Connoly, T. W., *Analog Computation in Engineering Design* McGraw-Hill Book Co., New York (1960).

Pennington, R.H., *Introductory Computer Methods and Numerical Analysis.* The Macmillan Book Co., New York (1965).

Mochel, M.G., *Fortran, Programming, Prgrams, and Schematic Storage Maps*, McGraw-Hill Book Co., New York (1971).

Korelitz, T.H., *Cut Vessel Flange Costs by Computer*, Petroleum Refiner 43 (7), 125, (1964).

Selected Computer Programs for Chemical Engineers, Chemical Engineering, 78 (16), 70 (1971).

Hodge, B. and Mantey, J.P., *Computer Refresher*, Chemical Engineering, Part 1, 74, (20) 180 (1967).
Part 2, 74, (22) 167 (1967).

	C SW-C25	49	25	18	8.7	8.0	7.2	6.8	6.2	4.4	3.2	2.7	1.6	—	—	—	—
Rivet and Stay Bar																	
IS : 1990-1962		37	0.55 R_{20}	26	8.6	7.9	7.1	6.8	6.5	5.9	4.3	3.6	—	—	—	—	—
		42	0.55 R_{20}	23	9.8	9.0	8.1	7.9	7.4	5.9	4.3	3.6	—	—	—	—	—
Sections, Plates, Bars																	
IS : 226-1962	St 42-S	42	24	23	9.8	9.0	8.1	—	—	—	—	—	—	—	—	—	—
IS : 961-1962	St 55 HTW	50	29	20	11.7	10.7	9.6	—	—	—	—	—	—	—	—	—	—
IS : 2062-1962	St 42-W	42	23	23	9.8	9.0	8.1	—	—	—	—	—	—	—	—	—	—
IS : 3039-1965	Grade A	—	—	—	9.8	[illegible]	8.1	—	—	—	—	—	—	—	—	—	—
	Grade D	—	—	—	11.7	10.7	9.6	—	—	—	—	—	—	—	—	—	—
IS : 3503-1966	Grade 1	37	0.55 R_{20}	26	8.6	7.9	7.1	6.8	6.5	5.9	4.3	3.6	—	—	—	—	—
	Grade 2	42	0.55 R_{20}	25	9.8	9.0	8.1	7.7	7.4	5.9	4.3	3.9	—	—	—	—	—
	Grade 3	44	0.55 R_{20}	23	10.2	9.3	8.5	8.0	7.7	5.9	4.3	3.6	—	—	—	—	—
	Grade 4	47	0.55 R_{20}	22	11.7	10.7	9.6	9.1	8.3	5.9	4.3	3.6	—	—	—	—	—
	Grade 5	50	0.55 R_{20}	21	12.1	11.1	10.0	9.5	8.3	5.9	4.3	3.6	—	—	—	—	—
IS : 3945-1966	Grade A-N	44	24	23	9.8	9.0	8.1	—	—	—	—	—	—	—	—	—	—
	Grade B-N	50	28.5	20	11.7	10.7	9.6	—	—	—	—	—	—	—	—	—	—

Allowable Stress Values for Ferrous and Non-Ferrous Material

Table E.1

Allowable Stress Values for Carbon and Low Alloy Steel in Tension

Material Specification	Grade or Designation	Mechanical Properties: Tensile Strength Min kgf/mm² R_{20}	Yield Stress Min kgf/mm² E_{20}	Percentage Elongation Min on Gauge Length $=5.65\sqrt{S_0}$	Allowable Stress Values in kgf/mm² at Design Temperature °C: Up to 250	Up to 300	Up to 350	Up to 375	Up to 400	Up to 425	Up to 450	Up to 475	Up to 500	Up to 525	Up to 550	Up to 575	Up to 600
				Plates													
IS : 2002-1962	I	37	$0.55R_{20}$	26	9.5	8.7	7.8	7.5	7.2	5.9	4.3	3.6	—	—	—	—	—
IS : 2002-1962	2A	42	$0.50R_{20}$	25	9.8	9.0	8.1	7.7	7.4	5.9	4.3	3.6	—	—	—	—	—
IS : 2002-1962	2B	52	$0.50R_{20}$	20	12.1	11.1	10.0	9.5	8.3	5.9	4.3	3.6	—	—	—	—	—
IS : 2041-1962	20Mo55	48	28	20	14.3	13.2	12.3	11.9	11.5	11.2	10.8	7.7	5.6	3.7	—	—	—
IS : 2041-1962	20Mn2	52	30	20	14.0	12.8	11.6	11.0	8.3	5.9	4.3	3.6	—	—	—	—	—
IS : 1570-1961	15Cr90Mo55	50	30	20	16.0	15.2	14.4	13.8	13.4	13.0	12.6	11.7	8.6	5.8	3.5	—	—
IS : 1570-1961	C15Mn 75	42	23	25	10.7	9.8	8.9	8.4	8.1	5.9	4.3	3.6	—	—	—	—	—
				Forgings													
IS : 2004-1962	Class 1	37	$0.50R_{20}$	—	8.6	7.9	7.1	6.8	6.5	5.9	4.3	3.6	—	—	—	—	—
IS : 2004-1962	Class 2	44	$0.50R_{20}$	15	10.2	9.3	8.5	8.0	7.7	5.9	4.3	3.6	—	—	—	—	—
IS : 2004-1962	Class 3	50	$0.50R_{20}$	21	11.7	10.7	9.6	9.1	8.3	5.9	4.3	3.6	—	—	—	—	—
IS : 2004-1962	Class 4	63	$0.50R_{20}$	15	14.7	13.4	12.2	11.5	8.3	5.9	4.3	3.6	—	—	—	—	—
IS : 1570-1961	20Mo55	48	28	20	14.3	13.2	12.3	11.9	11.5	11.2	10.8	7.7	5.6	3.7	—	—	—
IS : 2611-1964	15Cr90Mo55	50	30	20	16.0	15.2	14.4	13.8	13.4	13.0	12.6	11.7	8.6	5.8	3.5	—	—
IS : 1570-1961	10Cr2Mo1	50	32	20	17.9	17.3	16.4	16.1	15.8	15.3	14.9	12.7	9.6	7.0	4.9	3.2	2.3
				Tubes, Pipes													
IS : 3609-1966	1% Cr ½% Mo Tube normalized and tempered	44	24	$950/R_{20}$	12.8	12.1	11.5	11.1	10.7	10.4	10.0	9.7	8.6	5.8	3.5	—	—
IS : 3609-1966	2¼ Cr 1% Mo Tube normalized and tempered	49	25	$950/R_{20}$	14.0	13.5	12.8	12.6	12.3	12.0	11.6	11.3	9.6	7.0	4.9	—	—
IS : 1570-1961	20Mo 55	46	25	$950/R_{20}$	12.8	11.8	11.0	10.6	10.3	10.0	9.6	7.7	5.6	8.7	--	—	—
IS : 1914-1961	32 kgf/mm², Min Tensile Strength	32	$0.50R_{20}$	$950/R_{20}$	7.4	6.8	6.2	5.8	5.6	5.0	4.3	3.6	—	—	—	—	—
IS : 1914-1961	43 kgf/mm², Min Tensile Strength	43	$0.50R_{20}$	$950/R_{20}$	10.0	9.2	8.3	7.9	7.6	5.9	4.3	3.6	—	—	—	—	—
IS : 2416-1963	32kgf/mm², Min Tensile Strength	32	$0.50R_{20}$	$950/R_{20}$	4.7	6.8	6.2	5.8	5.6	5.0	4.3	3.6	—	—	—	—	—
IS : 1978-1961	St 18	31.6	17.6	—	8.2	7.5	6.7	6.4	6.2	5.9	4.3	3.6	—	—	—	—	—
	St 20	33.7	19.7	—	9.2	8.4	7.6	7.2	6.9	5.9	4.3	3.6	—	—	—	—	—
	St 21	33.7	21.1	—	9.8	9.0	8.1	7.7	7.4	5.9	4.3	3.6	—	—	—	—	—
	St 25	42.2	24.6	—	11.5	10.5	9.5	9.0	8.3	5.9	4.3	3.6	—	—	—	—	—
IS : 1979-1961	St 30	42.2	29.5	—	13.8	12.6	11.5	10.8	8.3	5.9	4.3	3.6	—	—	—	—	—
	St 32	44.3	32.3	—	15.0	13.8	12.5	11.8	8.3	5.9	4.3	3.6	—	—	—	—	—
	St 37	46.4	36.6	—	17.1	15.6	14.1	13.4	8.3	5.9	4.3	3.6	—	—	—	—	—
				Castings													
IS : 3038-1965	Grade 1	55	35	17	12.2	11.2	10.1	9.6	6.2	4.4	3.2	2.7	—	—	—	—	—
	Grade 2	47	25	17	9.6	8.8	8.2	8.0	7.7	7.5	7.2	5.8	4.2	2.8	—	—	—
	Grade 3	52	31	15	11.9	11.0	10.2	9.9	9.6	9.3	8.4	5.8	4.2	2.8	—	—	—
	Grade 4	49	28	17	11.2	10.6	10.1	9.7	9.3	9.1	8.8	8.5	6.5	4.4	2.6	—	—
	Grade 5	52	31	17	13.0	12.5	11.9	11.7	11.4	11.1	10.8	9.5	7.2	5.3	3.7	2.4	—
	Grade 6	63	43	15	17.2	16.3	15.5	14.9	14.4	14.0	13.5	6.7	4.9	3.5	2.6	1.7	0.9
IS : 2865-1964	C SW-C20	42	21	20	7.3	6.7	6.1	5.7	5.5	4.4	3.2	2.7	1.6	—	—	—	—
	C SW-C25	49	25	18	8.7	8.0	7.2	6.8	6.2	4.4	3.2	2.7	1.6	—	—	—	—
				Rivet and Stay Bar													
IS : 1990-1962		37	$0.55\ R_{20}$	26	8.6	7.9	7.1	6.8	6.5	5.9	4.3	3.6	—	—	—	—	—
		42	$0.55\ R_{20}$	23	9.8	9.0	8.1	7.9	7.4	5.9	4.3	3.6	—	—	—	—	—
				Sections, Plates, Bars													
IS : 226-1962	St 42-S	42	24	23	9.8	9.0	8.1	—	—	—	—	—	—	—	—	—	—
IS : 961-1962	St 55 HTW	50	29	20	11.7	10.7	9.6	—	—	—	—	—	—	—	—	—	—
IS : 2062-1962	St 42-W	42	23	23	9.8	9.0	8.1	—	—	—	—	—	—	—	—	—	—
IS : 3039-1965	Grade A	—	—	—	9.8	9.0	8.1	—	—	—	—	—	—	—	—	—	—
	Grade D	—	—	—	11.7	10.7	9.6	—	—	—	—	—	—	—	—	—	—
IS : 3503-1966	Grade 1	37	$0.55\ R_{20}$	26	8.6	7.9	7.1	6.8	6.5	5.9	4.3	3.6	—	—	—	—	—
	Grade 2	42	$0.55\ R_{20}$	25	9.8	9.0	8.1	7.7	7.4	5.9	4.3	3.9	—	—	—	—	—
	Grade 3	44	$0.55\ R_{20}$	23	10.2	9.3	8.5	8.0	7.7	5.9	4.3	3.6	—	—	—	—	—
	Grade 4	47	$0.55\ R_{20}$	22	11.7	10.7	9.6	9.1	8.3	5.9	4.3	3.6	—	—	—	—	—
	Grade 5	50	$0.55\ R_{20}$	21	12.1	11.1	10.0	9.5	8.3	5.9	4.3	3.6	—	—	—	—	—
IS : 3945-1966	Grade A-N	44	24	23	9.8	9.0	8.1	—	—	—	—	—	—	—	—	—	—
	Grade B-N	50	28.5	20	11.7	10.7	9.6	—	—	—	—	—	—	—	—	—	—

Table E.4
Allowable Stress Values for Copper and Copper Alloys

Material Specification	Grade Product	Mechanical Properties: Tensile Strength Min kfg/mm²	Yield Stress Min kgf/mm²	Elongation Per cent Min	Allowable Stress Values in kgf/mm² at Design Temperature °C: Upto 50	Upto 75	Upto 100	Upto 125	Upto 150	Upto 175	Upto 200	Upto 225	Upto 250	Upto 275	Upto 300	Upto 325	Upto 350
Plate, Sheet and Strip																	
IS : 410 -1967	Cu Zn 30	28	—	45	7.03	7.03	7.03	7.03	6.96	5.70	3.83	2.46	—	—	—	—	—
	Cu Zn 37	28	—	45	8.79	8.67	8.30	7.81	7.28	5.38	2.00	—	—	—	—	—	—
	Cu Zn 40	28	—	30	8.79	8.67	8.30	7.81	7.28	5.38	2.00	—	—	—	—	—	—
IS : 1972-1961	All Grades	22.5	—	35	4.71	4.66	4.54	4.30	3.47	2.71	1.90	—	—	—	—	—	—
Bars and Rods																	
IS : 288 -1960	—	40	—	22	7.03	7.03	7.03	7.03	6.96	5.70	3.83	2.46	—	—	—	—	—
IS : 4171-1967	—	40	—	22	7.03	7.03	7.03	7.03	6.96	5.70	3.83	2.46	—	—	—	—	—
Bolting Material																	
IS : 288 -1960	—	40	—	22	1.76	1.76	1.76	1.67	1.54	1.48	1.41	—	—	—	—	—	—
IS : 4171-1967	—	40	—	22	1.76	1.76	1.76	1.67	1.54	1.48	1.41	—	—	—	—	—	—
Sections																	
IS : 291 -1966	Grade 1	35	—	20	8.79	8.67	8.30	7.81	7.28	5.38	2.00	—	—	—	—	—	—
	Grade 2	35	—	20	8.79	8.67	8.30	7.81	7.28	5.38	2.00	—	—	—	—	—	—
Tubes																	
IS : 407 -1966	Alloy 1	29	—	—	7.03	7.03	7.03	7.03	6.96	5.70	3.83	—	—	—	—	—	—
	Alloy 2	29	—	—	8.79	8.67	8.30	7.81	7.28	5.38	2.00	—	—	—	—	—	—
IS : 1545-1960	ISBT 1 ISBT 2	—	—	—	7.03	7.03	7.03	7.03	6.96	5.70	3.83	—	—	—	—	—	—
	ISABT	—	—	—	8.44	8.44	8.44	8.44	8.28	5.43	2.58	1.58	—	—	—	—	—
	ISABZT	—	—	—	8.76	8.67	8.53	8.34	8.09	7.09	4.64	3.16	1.88	—	—	—	—
IS : 2371-1963	Cu Ni 21 Al 2 As	32	—	—	8.44	8.44	8.44	8.44	8.28	5.43	2.58	1.58	—	—	—	—	—
	Cu Ni 31 Mn 1 Fe	42	—	—	8.31	8.08	7.89	7.71	7.59	7.58	7.27	7.14	7.11	6.92	6.83	6.73	6.66
IS : 2501-1963		—	—	—	4.22	4.19	4.13	4.00	3.47	2.71	1.90	—	—	—	—	—	—
Castings																	
IS : 318 -1962	Grade 1	22	11.5	12	5.94	5.88	5.82	5.76	5.69	5.56	5.38	5.13	4.78	4.00	—	—	—
	Grade 2	19	11	7.5	5.21	5.09	4.96	4.84	4.77	4.65	4.58	—	—	—	—	—	—
	Grade 3	17.5	7.5	7.0	4.30	4.12	3.90	3.78	3.65	3.52	3.50	—	—	—	—	—	—

UF (Urea-form alde-hyde)	Up (Ure-thances)	PVAC, PVAI, PVB, PVC PVCAC, PVFM
387-914	12-703	35-633
0.5-1.0	10-1000	2-450
70-105.0	0.7-70.0	35-48.0
58-3164	1406-	70-1547
—	0.28-7.0	- to 48
03-1265	- to 633	- to 1195
1.0-112.0	0.7-25.0	-28.0
M 100- M 120	20 A (shore) M 28	10 A (shore) M85
0.0136 -0.217	- to 0.272	0.0217 - 1.083

Table E.2

Allowable Stress Values for High Alloy Steels in Tension

Material Specification	Designation	Product	Remarks	Mechanical Properties: Tensile Strength kgf/mm² Min R_{20}	Yield Stress kgf/mm² Min E_{20}	Elongation Per cent Min On Gauge Length 5.65 S_0	Allowable Stress Values in kgf/mm² at Design Temperature °C: 50	100	150	200	250	300	350	40
IS : 1570-1961	04 Cr 19 Ni 9	Plates, Sections Bars, forgings and seamless tubes	Austenitic stainless steel	55	24	28	16.00	14.20	12.40	10.60	9.97	9.35	8.70	8.
	04 Cr 19 Ni 9 Ti 20			55	24	28	16.00	14.28	12.56	10.83	10.64	10.60	10.60	10.
	04 Cr 19 Ni 9 Nb 40			55	24	28	16.00	14.28	12.56	10.83	10.64	10.60	10.60	10
	05 Cr 18 Ni 11 Mo 3			55	24	28	16.00	14.50	13.00	11.50	11.24	11.23	11.23	11.
	05 Cr 19 Ni 9 Mo 3 Ti 20			55	24	28	16.00	14.50	13.00	11.50	11.24	11.23	11.23	11.
IS : 3444-1966	Grade 7,8	Castings		47	21	21	14.00	12.94	11.88	10.83	10.64	10.60	10.60	10
	Grade 9,11			47	21	13	14.00	13.47	12.34	11.50	11.24	11.23	11.23	11

Table E.3

Allowable Stress Values for Aluminium and Aluminium Alloys in Tension

Material	Grade and Products	Condition	Mechanical Properties: Tensile Strength Min kgf/mm²	0.2 Proof Stress Min kgf/mm²	Elongation on $\sqrt[4]{S_0}$ Min Per cent	Allowable Stress Values in kgf/mm² at Design Temperature °C: Upto 50	Upto 75	Upto 100	Upto 125	Upto 150	Upto 175	Upto 200
IS : 736 —1966	PIB Plate	M	6.5	—	30							
IS : 737 —1965	SIB Sheet, Strip	O	6.5	—	30	1.36	1.27	1.16	1.05	0.95	0.84	0.74
IS : 733 —1967	EIB Bars, Rods	M	6.5	—	25							
IS : 1285—1958	VIB Extruded	M	6.3	—	25							
IS : 736 —1966	PIB Plate	½H	10	—	8	2.11	2.07	2.00	1.90	1.65	1.41	1.12
IS : 737 —1965	SIB Sheet, Strip	½H	10	—	8							
IS : 737 —1965	NS_3 Sheet, Strip	O	10	—	30	2.30	2.22	2.00	1.89	1.69	1.48	1.26
IS : 373 —1965	NS_3 Sheet, Strip	½H	14	—	7	3.46	3.37	3.25	3.09	2.82	2 46	2.18
IS : 736 —1965	NP_4 Plate	M	19	—	12							
IS : 737 —1965	NS_4 Sheet, Strip	O	17.5	—	18							
IS : 733 —1967	NE_4 Bars, Rods	M	17.5	—	18	4.35	4.35	4.32	4.20	3.80	3.27	2.46
IS : 738 —1966	NT_4 Drawn Tube	O	17.5	—	18							
IS : 1285—1958	NV_4 Extruded Tubes	M	17.3	—	18							
IS : 737 —1965	NS_4 Sheet, Strip	½H	20.5	—	8	5.45	5.42	5.27	4.98	4.50	3.94	2.46
IS : 738 —1966	NT_4 Drawn Tube	½H	23.5	—	5	5.97	5.95	5.78	5.40	4.85	4.28	2.46
IS : 737 —1965	NS_4 Sheet, Strip	½H	23.5	—	5							
IS : 737 —1965	NS_5 Sheet, Strip	O	22	—	18							
IS : 733 —1967	NE_5 Bars, Rods and Sections	M	22	—	18	5.69	—	—	—	—	—	—
IS : 738 —1966	NT_5 Drawn Tubes	O	22	—	18							
IS : 1285—1958	NV_5 Hollow Section	N	22	—	18							
IS : 737 —1965	NS_6 Sheet, Strip	O	27	—	18							
IS : 738 —1966	NT_6 Drawn Tubes	O	27	—	18							
IS : 736 —1966	NP_6 Plate	O	27	—	20	6.68	—	—	—	—	—	—
IS : 733 —1967	NE_6 Bars, Rods	M	27	—	18							
IS : 1285—1958	NV_8 Extruded Tubes	M	26.8	—	18							
IS : 733 —1967	NE_8 Bars, Rods, and Sections	O	27	—	16	7.02	—	—	—	—	—	—
IS : 1285—1958	HV_9 Extruded Round Tubes Hollow Section	M	11	—	15	2.97	2.90	2.78	2.67	2.53	1.93	1.33
IS : 1285—1958	HV_9 do	P	15.7	—	10	3.74	3.61	3.38	3.24	2.95	2.18	1.41
IS : 1285—1958	HV_9 do	WP	18.9	—	12	5.15	4.91	4.65	4.29	3.16	2 18	1.41
IS : 736 —1966	HP_{30} Plate	W	20.5	11.0	15							
IS : 734 —1967	HF_{30} Forging	W	19 0	11.0	18							
IS : 737 —1965	HS_{30} Sheet, Strip	W	20.5	11.0	15	5.18	5.01	4.86	4.71	4.50	3.94	2.81
IS : 733 —1967	HE_{30} Bars, Rods and Sections	W	19.0	11.0	18							
IS : 738 —1966	HT_{30} Drawn Tube	W	22.0	11.0	16							
IS : 738 —1966	HT_{30} Drawn Tube	WP	31.5	25.0	7							
IS : 736 —1966	HP_{30} Plate	WP	30	23.5	8							
IS : 734 —1967	HF_{30} Forging	WP	30	25	10	7.29	7.10	6.85	6.60	5.56	4.36	3.10
IS : 737 —1965	HS_{30} Sheet, Strip	WP	30	25.5	8							
IS : 733 —1967	HE_{30} Bars, Rods and Sections	WP	30	25.0	10							
IS : 1285—1958	HB_{15} Bolting alloy	WP	44	38.0	8	8.82	8.45	7.95	7.30	5.06	3.09	2.11
IS : 1285—1958	NB_6 Bolting alloy	½H	31.5	23.5	—	7.65	—	—	—	—	—	—
IS : 1284—1966	HB_{30} Bolting alloy	WP	30.0	25.0	—	5.85	5.70	5.48	5.27	4.43	3.44	2.32
IS : 617 —1959	A—3 Casting alloy	M(sand cast)	16.5	—	8	2.59	2.36	2.25	2.14	1.90	1.72	1.55
		M(chill cast)	18.9	—	12	2.96	2.68	2.56	2.46	2.18	1.97	1.76

esign Temperature °C

Jpto 225	Upto 250	Upto 275	Upto 300	Upto 325	Upto 350
2.46	—	—	—	—	—
—	—	—	—	—	—
—	—	—	—	—	—
2.46	—	—	—	—	—
2.46	—	—	—	—	—
—	—	—	—	—	—
—	—	—	—	—	—
—	—	—	—	—	—
—	—	—	—	—	—
1.58	—	—	—	—	—
3.16	1.88	—	—	—	—
1.58	—	—	—	—	—
7.14	7.11	6.92	6.83	6.73	6.66
—	—	—	—	—	—
5.13	4.78	4.00	—	—	—
—	—	—	—	—	—

APPENDIX C

Properties of Plastic Materials

Property	ABS Acryloni-trile Butadiene Styrene	EP Epoxies	PC Poly-carbonate	Polyesters	PE Polyethy-lene	PS, SAN, SBP, SRP, (Polysty-rene)	SI Silico-nes)	UF (Urea-form alde-hyde)	Up (Ure-thances)	PVAC, PVAI, PVB, PVC PVCAC, PVFM
Tensile strength, kg/cm²	281-563	281-2109	563-1406	56-3515	70-387	106-1406	56-2461	387-914	12-703	35-633
Elongation, (%)	2-300	0.5-70	0.9-1.30	0.5-310	15-1000	0.75-80	- to 100	0.5-1.0	10-1000	2-450
Tensile modulus 10^3 kg/cm²	16-72	0.07-214	25-130	21-140	0.98-13.0	10.5-98.0	0.06-211.0	70-105.0	0.7-70.0	35-48.0
Compressive strength kg/cm²	492-1547	70-2812	879-1336	844-3515	- to 387	281-1546	7-1265	1758-3164	1406-	70-1547
Compressive modulus 10^3 kg/cm²	12-27	—	21-32.0	—	- to 11.0	- to 37.0	—	—	0.28-7.0	- to 48
Flexural yield strength, kg/cm²	352-1898	70-4218	949-2109	563-5624	- to 492	352-1828	- to 2461	703-1265	- to 633	- to 1195
Flexural modulus 10^3 kg/cm²	1.4-91.0	—	24.0-84	- to 141.0	- to 25.0	- to 126.0	—	91.0-112.0	0.7-25.0	-28.0
Hardness rock well	275-M 100	M 80-M 120	M 70-R 118	60 (bared) E 89	D 30 (shore) R 15	R 50-E 60	-40 (shore) M 95	M 100-M 120	20 A (shore) M 28	10 A (shore) M85
Impact strength kg—m/cm	0.0108-0.544	0-0.544-0.544	0.0652-.952	0.0108-.87	0.0272-.1088	0.0136-.598	- to 0.816	0.0136-0.217	- to 0.272	0.0217-1.088

APPENDIX D

Modulus of Elasticity

Values of E for Ferrous Materials in 10^3 kgf/mm²

Material		Design Temperature °C						
		0	20	100	200	300	400	500
Carbon	C<0.30%	19.6	19.6	19.5	19.0	18.2	17.0	—
Steels	C<0.30%	21.1	21.0	20.7	19.9	19.0	17.3	—
Carbon-molybdenum steels and chrome-molybdenum steels (up to 3% Cr)		21.1	21.0	20.7	20.1	19.4	18.4	17.2
Intermediate chrome molybdenum steels and austenitic stainless steels		19.3	19.3	19.0	18.6	18.0	17.3	16

Note—Intermediate values may be obtained by interpolation.

Values of E for Aluminium and its Alloys in 10^3 kgf/mm²

Material Grade	Design Temperature °C								
	—200	—100	0	50	75	100	125	150	200
IB, N_3, N_4	7.8	7.4	7.1	7.0	7.0	6.9	6.8	5.7	6.4
H9	7.4	7.1	6.8	6.6	6.6	6.5	6.5	6.3	6.0
H15	8.3	7.9	7.5	7.4	7.4	7.3	7.2	7.1	6.8
A6	8.9	8.5	8.2	8.1	8.0	7.9	7.8	7.7	7.4

Note 1. Intermediate values may be obtained by interpolation.

Note 2. Since aluminium and its alloys do not have a well-defined yield point, the above values of E are to be used with caution.

Values of E for Nickel and Nickel Alloys in 10^3 kgf/mm²

Material	Design Temperature °C						
	20	300	400	500	600	700	750
Nickel	21.1	20.4	18.8	16.5	14.0	11.7	10.9
70% Nickel and 30% copper alloy	18.8	18.0	17.6	16.9	16.2	15.5	15.0
75% Nickel, 15% chromium and 10% ferrous alloy	21.8	20.7	20.0	17.6	16.0	13.0	11.9

Values of E for Copper and its Alloys in 10^3 kgf/mm²

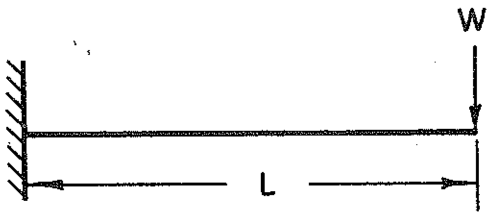
Material	Composition	Design Temperature °C								
		20	50	100	150	200	250	300	350	400
Copper	99.98% Cu	11.2	11.1	11.0	10.0	10.6	10.4	10.1	9.7	—
Commercial brass	66% Cu, 34% Zn	9.8	9.7	9.6	9.5	9.1	8.9	8.6	8.5	—
Leaded in bronze	88% Cu, 6% Sn, 1.5% Pb, 4.5% Zn	9.1	9.0	8.9	8.7	8.4	8.2	8.0	7.7	—
Phosphor Bronze	85.5% Cu, 12.5% Sn, 10% Zn	10.5	10.3	10.2	9.8	9.5	9.1	8.5	6.7	—
Muntz	59% Cu, 39% Zn	10.7	10.2	9.8	9.1	8.3	7.7	—	—	—
Cupro Nickel	80% Cu, 20% Ni or 70% Cu and 30% Ni	13.3	13.1	12.9	12.6	12.4	12.1	11.8	11.5	14.2

APPENDIX B

Bending Moment and Deflection

B ize	I' % ' 10' Zn									
Muntz	59% Cu, 39% Zn	10.7	10.2	9.8	9.1	8.3	7.?	—	—	—
Cupro Nickel	80% Cu, 20% Ni or 70% Cu and 30% Ni	13.3	13.1	12.9	12.6	12.4	12.1	11.8	11.5	14.2

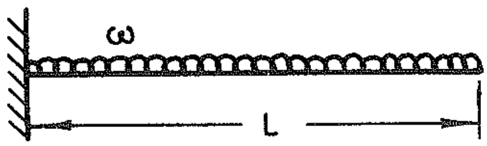
APPENDIX B
Bending Moment and Deflection of Beams



Cantilever—concentrated load at free end

M_{max} (at fixed end) $= WL$

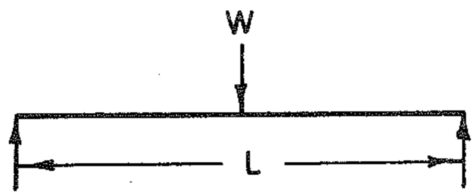
δ_{max} (at free end) $= \frac{WL^3}{3EI}$



Cantilever—uniformly distributed load

M_{max} (at fixed end) $= \frac{\omega L^2}{2}$

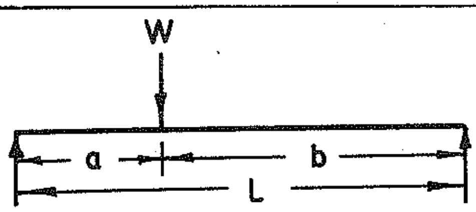
δ_{max} (at free end) $= \frac{\omega L^4}{8EI}$



Simply supported—concentrated load at centre

M_{max} (at point of load) $= \frac{WL}{4}$

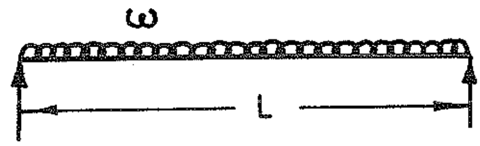
δ_{max} (at point of load) $= \frac{WL^3}{48\,EI}$



Simply Supported—concentrated load at any point

M_{max} (at point of load) $= \frac{Wab}{L}$

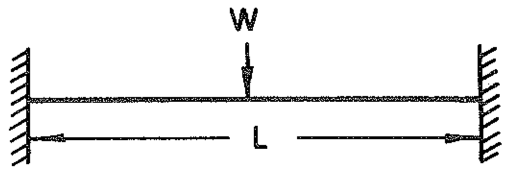
δ (at point of load) $= \frac{Wa^2b^2}{3EIL}$



Simply supported—uniformly distributed load

M_{max} (at the centre) $= \frac{\omega L^2}{8}$

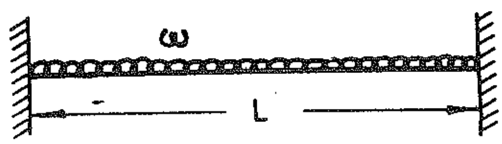
δ_{max} (at the centre) $= \frac{5\omega L^4}{384\,EI}$



Fixed at both ends—concentrated load at centre

M_{max} (at the centre) $= \frac{WL}{8}$

δ_{max} (at the centre) $= \frac{WL^2}{192\,EI}$



Fixed at both ends—uniformly distributed load

M_{max} (at the centre) $= \frac{\omega L^2}{16}$

δ_{max} (at centre) $= \frac{\omega L^4}{384\,EI}$

Notation

W=concentrated load ω=unit load L=span M=bending moment δ=deflection

70% Nickel and
30% copper alloy 18.8 18.0 17.6

APPENDIX F

Allowable Stresses for Flange Bolting Material kgf/mm²

Material	Diameter (mm)	Specified Tensile Strength (kgf/mm²)	Allowable Stress kgf/mm² for Design Metal Temperature Not Exceeding (°C)						
			50	100	200	250	300	350	400
Hot rolled carbon steel	Upto 150	44-52	5.87	5.62	5.45	4.85	—	—	—
1% Cr Mo steel	Upto 63.5	86 Min	19.68	18.5	17.1	16.2	15.75	15.12	14.27
	Over 63.5 to 102	79 Min	17.79	16.66	15.47	14.76	14.41	13.71	12.94
5% Cr Mo steel	Upto 6.35 over 63.5	71 Min 66 Min	14.06	14.06	14.06	14.06	14.06	14.06	14.06
1% Cr Mo V steel	Upto 63.5	86 Min	19.68	19.05	18.49	17.93	17.29	16.80	16.03
	Over 63.5 upto 102	82 Min	17.79	17.23	16.66	16.24	15.54	15.26	14.55
13% Cr Ni steel	Upto 102	71 Min	17.93	16.45	14.34	13.64	12.86	12.16	10.65
18/8 Cr Ni steel	All (1) (2)		13.18	11.07	8.65	8.01	7.73	7.45	7.31
18/8 Cr Ni Ti stabilized steel	All (1) (2)		13.18	11.53	10.19	9.49	9.14	8.79	8.58
18/9 Cr Ni Nb stabilized steel	All (1) (2)		13.18	11.53	10.19	9.49	9.14	8.79	8.58
17/10 2½ Cr Ni Mo steel	All (1) (2)		13.18	11.18	9.56	8.86	8.51	8.08	7.94
18 Cr 2 Ni steel	Upto 102	86 Min	21.58	19.90	17.30	16.40	15.54	14.69	12.94

APPENDIX G

Permissible Internal Pressure for Pipes

(a) Steel and iron pipes—diameters between 0.6 to 12.5 cm

$$p = \frac{2f}{d_0}(t—0.165) —8.79$$

(b) Steel and iron pipes—diameters over 12.5 cm

$$p = \frac{2f}{d_0}(t—(0.25)$$

(c) Non-ferrous seamless tubes and pipes

$$p = \frac{2ft}{d_0}$$

where

p —Internal working pressure in kg/cm^2
f —Permissible working stress in kg/cm^2 (Table G.1 and G.2)
t —Pipe wall thickness cm
d_0 —Outside diameter of pipe cm

Table G.1
Values of 'f' for Non Ferrous Materials

Material	For temperature in °C not to exceed						
	—20 to 65	120	175	200	230	260	290
Muntz metal tubing and high brass tubing	350	280	175	—	—	—	—
Muntz metal condenser tubes	350	280	175	—	—	—	—
Red brass tubes	420	385	350	315	—	—	—
Copper tubes	420	350	315	280	—	—	—
Copper pipes	420	350	315	280	—	—	—
Admiralty tubing	490	460	420	285	315	—	—
Admiralty condenser tubes	490	460	420	385	315	—	—
Steam bronze	480	440	410	380	350	295	230

Table G.2
Value of 'f' for Steel and Iron

Material	For temperature in °C not to exceed							
	30 to 340	370	400	425	455	480	510	540
Seamless medium carbon steel	840	840	730	585	445	310	185	—
Seamless low carbon steel	660	630	605	500	410	310	185	—
Fusion welded steel	770	730	670	560	445	310	185	—
Fusion welded steel :								
Grade B	700	675	630	525	420	310	185	—
Grade A	630	620	590	485	400	310	185	—
Lap-welded steel	630	620	590	485	400	310	185	—
Butt-welded steel	630	620	590	—	—	—	—	—
Lap-welded wrought iron	560	540	485	—	—	—	—	—
Butt-welded wrought iron	560	540	485	—	—	—	—	—
Seamless alloy steel	770	770	770	755	740	700	560	350

APPENDIX-H

Relevant Indian Standards

Material and Sections

IS-2712-1971	Specification for compressed asbestos jointing.
IS-800-1956	Code of practice for use of structural steel in general building construction.
IS-1972-1961	Specification for plate, sheet, strip for industrial purpose copper.
IS-3965-1969	Dimensions for wrought aluminium and aluminium alloys, bar, rod and section.
IS-2677-1964	Dimensions for plate, wrought aluminium and aluminium alloys.
IS-2676-1964	Dimensions for sheet and strip, wrought aluminium and aluminium alloys.
IS-1870-1965	Comparison of Indian and overseas standard for wrought steels for general engineering purpose.
IS-808-1964	Specification for rolled steel beam, channel and angle sections (revised).
IS-1731-1971	Dimensions for steel flats structural and general engineering purpose.
IS-1730-1961	Dimensions for steel plate, strip and sheet for structural and general engineering purpose.
IS-4687-1968	Specification for gland packing asbestos.
IS-4688-1968	Dimensions for proofed cotton duck gland packing.

Machine Elements

IS-2535-1969	Specifications for basic rack and modules of cylindrical gears for general engineering and heavy engineering.

IS-4460-1967	Method for rating of machine cut spur gears and helical gears.
IS-2389-1968	Specification for precision hexagonal bolts, screws, nuts and lock nuts, (diameter range 1.6 to 5 mm).
IS-2636-1964	Specification for wing nuts.
IS-3824-1966 (Part I)	Rolling bearings: methods of evaluating dynamic load ratings of: Part-I radial ball bearings.
IS-3824-1966 (Part II)	Rolling bearings: methods of evaluating dynamic load ratings of: Part-II radial roller bearings.
IS-3824-1966 (Part III)	Rolling bearings: methods of fevaluating dynamic load ratings of: Part-III thrust ball bearings.
IS-3824-1966 (Part IV)	Rolling bearings: methods of evaluating dynamic load ratings of: Part-IV thrust roller bearings.
IS-3823-1966 (Part I)	Rolling bearings: methods of evaluating static load rating of: Part-I radial ball bearings.
IS-3823-1966 (Part II)	Rolling bearings: methods of evaluating static load ratings of: Part-II radial roller bearings.
IS-3823-1966 (Part III)	Rolling bearings: methods of evaluating static load ratings of: Part-III thrust ball bearings.
IS-3823-1966 (Part IV)	Rolling bearings: methods of evaluating static load ratings of: Part-IV thrust roller bearings.
IS-3132-1965	Recommendations for shafts diameters for chemical equipments.
IS-1363-1967	Specification for black hexagonal bolts, nuts and locknuts (diam. 6 to 39 mm) and black hexagonal screws (diam. 6 to 24 mm).
IS-2585-1968	Specification for black square bolts and nuts, (diam. range 6 to 39 mm) and black square screws (diam. range 6 to 24 mm).
IS-3138-1966	Specification for hexagonal bolts and nuts (M 42 to M 150).
IS-1691-1968	Specification for cast-iron and mild steel flat pulleys.
IS-2693-94	Dimensions for cast iron flexible couplings.
IS-3653-1966	Dimensions for forged end type rigid couplings.
IS-2293-1963	Dimensions of Gib-head keys and keyways.
IS-2403-1964	Specification for transmission steel roller chains and wheels.
IS-2494-1964	Specifications for V-belts for industrial purposes.
IS-3142-1965	Specification for V-grooved pulleys for V-belts groove sections A, B, C, D and E.
IS-1367-1967	Technical supply conditions for threaded fasteners.

Tubes, Pipes and Valves

IS-2371-1963	Specification for solid drawn copper alloy tubes for condensers, evaporators, heaters and coolers using saline and hard water.
IS-3333-1967	Dimensions for petroleum industry pipe threads. Part I —Line pipe threads. Part II —Casing round threads. Part III —Tubing round threads. Part IV —Buttress casing threads.
IS-3382-1965	Specification for stainless steel milk pipes and fittings.
IS-3589-1966	Specification for electrically welded steel pipes (200 mm to 2000 mm nominal diam.).
IS-3516-1966	Specification for cast iron pipe flanges and flanged fittings, class 9, for petroleum industry.
IS-6011-1970	Specification for carbon steel tubes for use on boardships for pressure services.

IS-2501-1963	Specification for copper tubes for general engineering purposes.
IS-2416	Specification for boiler and superheater tubes for marine and naval purpose.
IS-3233-1965	Glossary of terms for safety and relief valves and their parts.
IS-778-1971	Specification for gunmetal gate, globe and check valves for general engineering purposes.
IS-4854-1969	Glossary of terms for valves and their parts. Part I—Screw down stop, check and globe valves, and their parts. Part II—Valves and cocks and their parts.
IS-407-1966	Specification for brass tubes for general purposes (second revision).
IS-1545-1969	Specification for tubes, solid drawn copper alloy.
IS-2678-1963	Dimensions for wrought aluminium and aluminium alloys, drawn tubes.
IS-2673-1964	Dimensions for wrought aluminium and aluminium alloys, extruded tubes (round).
IS-1239-1958	Specification for mild steel tubes and tubulors.
IS-6418-1971	Specification for cast iron and malleable cast iron flanges for general engineering purposes.
IS-6392-1971	Specification for steel pipe flanges.

Pressure Vessels and Components

IS-4682-1968 (Part I)	Code of practice for lining of vessels and equipments for chemical processes. Part I: rubber lining.
IS-4682-1969 (Part II)	Code of practice for lining of vessels and equipments of chemical processes. Part II: glass enamel lining.
IS-4682-1969 (Part III)	Code of practice for lining of vessels and equipment for chemical processes. Part III: lead lining.
IS-4682 (Part IV)	Code of practice for lining of vessels and equipments for chemical process. Part IV : Plasticized P.V.C. lining.
IS-4682-1970 (Part V)	Code of practice for lining of the vessels and equipments for chemical processes. Part V : Epoxide resin lining.
IS-4864-1968	Specification for shell flanges for vessels and equipments.
IS-4865-1968	Specification for shell flanges for vessels and equipments, welded shell flanges for no-pressure service.
IS-4866-1968	Specification for shell flanges for vessels and equipments: Welded shell flanges for carbon steel pressure vessels and equipments.
IS-4867-1968	Specification for shell flanges for vessels and equipments: Welded neck shell flanges for carbon steel for pressure vessels and equipments.
IS-4868-1968	Specification for shell flanges for vessels and equipments: Welded shell flanges for stainless steel pressure vessels and equipments.
IS-4869-1968	Specification for shell flanges for vessels and equipments: Welded shell flanges with hub for stainless steel pressure vessels and equipments.
IS-4870-1968	Specification for shell flanges for vessels and equipments: Flat gaskets for shell flanges.
IS-6202-1971	Specification for flat glass oil level gauges for oil storage tanks.
IS-804-1967	Specification for rectangular pressed steel tanks.
IS-2825-1969	Code for unfixed pressure vessels.

Subject Index