Chapter 1

Introduction

1.1 Defining Tribology

1.1.1 What is tribology?

Tribology, derived from the Greek word 'Tribos', is a science that deals with friction, lubrication and wear in all contacting pairs. Tribological knowledge helps in reducing the requirement of maintenance and improves reliability of interacting machine components. Essence of tribology at design stage yields substantial economic benefits.

1.1.2 Need of tribology as a separate subject

Friction, wear and lubrication have been taught in science and engineering classes, but at a rudimentary level. It means empirically derived trends (friction force is proportional to loading force, static friction is greater than kinetic friction, viscous friction in a fluid is proportional to the normal contact force, etc.,) are often the only predictive tools available. These approaches have the drawbacks of being predictive only over a limited range of parameters. Since the underlying physical mechanisms are not well understood, often one does not even know the important parameters or the range over which the observed trends are valid. This poor predictive power has given birth to the field of tribology, being pursued in many scientific quarters as a complete subject.

Most tribological phenomena are inherently complicated and interconnected, making it necessary to understand the concepts of tribology in detail. For example, calculations of contact stresses and surface temperature during sliding require understanding of 'solid mechanics'. Similarly, study of lubricant film formed between different geometric shapes of interacting surfaces demands knowledge of fluid mechanics. Mechanical wear at atomic and micro scales involves thorough understanding of material science. Formation of boundary layer on the solid surface demands information about chemistry. In short, integration of knowledge from multifaceted disciplines (solid mechanics, fluid mechanics, material science, chemistry, etc.,) is essential, and therefore, a separate subject is required.

Solid Mechanics

Solid mechanics governs the response of solid material to applied force, as shown in Fig. 1.1.2.1.

- Based on storage of energy and loss modulus, materials can be categorized as elastic, visco–elastic and plastic materials. Behaviour of these materials with fluid at the interface of contacting solids affects the performance of the system.

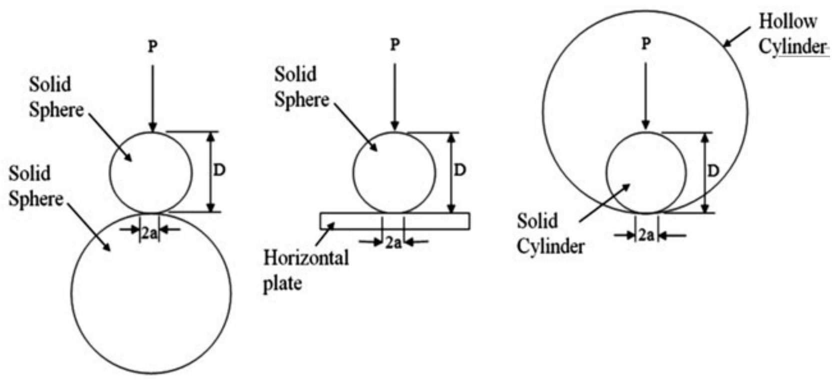

Fig. 1.1.2.1 Solid mechanics

- Surface roughness and real area of contact between surfaces play very important role. Neither zero roughness nor high roughness is desirable. Real area of contact may be 10–50 percent of apparent area. So to achieve high performance from contacting surfaces, account of roughness and real area of contact is essential.
- Behaviour modelling of thin layer coatings having different elastic properties than the substrate is required. The layer may slip or stick at the substrate interface.
- Heat source and heat conduction equations are required to estimate temperature distribution.
- In addition, theories related to crack nucleation, crack propagation and delamination are required.

Fluid Mechanics

A fluid (gas or liquid) is often used to separate two contacting surface as shown in Fig. 1.1.2.2. Following theories/relations are required to estimate tribological behaviour of incorporated fluid:

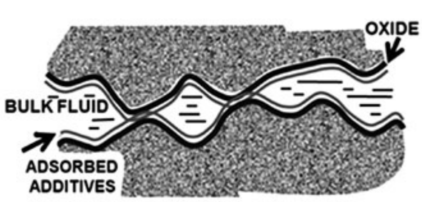

Fig. 1.1.2.2 Fluid mechanics and chemistry

- Hydrodynamic, aerodynamic, hydrostatic, and aerostatic theories of fluid film lubrication.
- Theories related to convective heat transfer.
- Rheological behaviour of liquid to semi-solids.
- Boundary, mixed and elasto–hydrodynamic lubrication mechanisms.
- Study of viscosity thinning and thickening effects.
- Mathematical modelling of thin lubricant film.

Material Science

This science is required to estimate the behaviour of materials in contact as shown in Fig. 1.1.2.3.

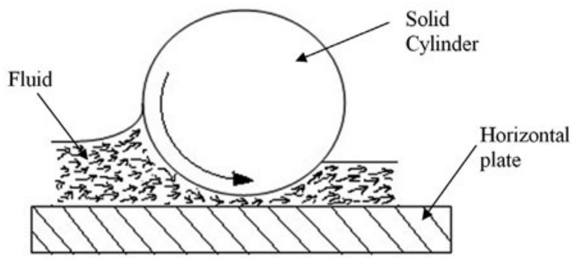

Fig. 1.1.2.3 Material science

Following aspects are important.

- Surface hardening/treatment.
- Development of high/low temperature coatings to provide non-sticking surfaces in bearings, gears, and other tribo–pairs. To deal with formation of adhesive junctions at high temperatures, coatings of ceramics, glass, and metals can be engineered to withstand 75 °C to nearly 200 °C.
- Manufacturing processes to apply nanometre to micrometer thick coating on various materials (material compatibility). Often intermediate coatings are used for better adhesive performance.
- Modelling of thin coatings and linings (thick coatings).

Chemistry

Knowledge of chemistry is required for

- Synthesis of additives: antiwear additives and extreme pressure additives
- Understanding compatibility of lubricants with process fluids and contacting surfaces
- Estimating shelf life of lubricant and its additives
- Predicting performance of lubricant layer as a function of temperature, sliding, etc.
- Optimizing concentration of lubricant additives: covalent, metallic and Van der Waal bonds

As tribology requires understanding of so many subjects, often it is scary to study tribology. In the present book, a step by step approach is followed to learn the subject. The empirical formulae, fundamentals and numerical solutions have been described to study this course.

1.1.3 History of Tribology

In September 1964, a conference on 'Lubrication in Iron and Steel Works in Cardiff (UK)' took place, realizing considerable losses due to lack of knowledge related friction and wear of machine components. After this realization the UK Minister of State for Science formed a committee to investigate into the education, research and the need of industry related to lubrication. The committee after deliberations concluded that only lubrication engineering could not provide complete solution to deal with friction and wear of machine components. An interdisciplinary approach embracing solid and fluid mechanics, chemistry, and material science is essential. Because there was no word for such new concept, a new name 'Tribology' was coined in 1966.

After 1966, the word Tribology has been used for:

1. Basic mechanisms governing interfacial behaviour

2. Basic theories quantifying interfacial mechanisms
3. Solutions to friction and wear problems

Major breakthrough in tribological science came in 1981 with the development of Scanning Tunnelling Microscope (STM) and systematic theory based on Contact Mechanics. Such developments provided tools to predict and estimate the behaviour of a single asperity contact.

Subsequent development of Atomic Force Microscope (AFM) in 1985 allowed measurement (surface topography, friction force) of all engineering surfaces. Atomic Force Microscope can be used to study friction, wear and lubrication at the nanometer level.

The development of tip-based microscopes (STM and AFM) and simulation software to imitate tip–surface interactions and corresponding elastic/plastic deformation, has allowed scientific investigations for interacting surfaces. Tribology is enriched with new findings from such investigations.

1.2 Tribology in Design

Tribology is required for sustainability of the systems having relatively moving machine elements such as gears, cam–follower, bearings, seals, valves, etc. Decrease in the system performance due to aging depends on the rate of wear. Therefore, it is very important to understand the change in system with time and frame suitable maintenance schemes for that system. In addition, if the technology incorporated in the system is expected to be replaced with a newer technology, then the design of the system may be economized by incorporating the tribological aspects alone. Under these conditions the design may be based on the estimated average wear rate for the component to obtain a finite life.

To differentiate mechanical and tribology designs, the following examples of gasket and seal are considered.

1.2.1 Mechanical design of seal and gasket

Seal

The seals are required for closing the unintentional openings or gaps between two or more jointed members for producing a joint that can prevent leakage and withhold fluid under pressure by providing a physical barrier. Basic functions of seal are prevention of fluid leakage between two relatively moving machine components and prevention of entry of foreign particles, like dust or abrasive material into the operating medium. To fulfil these functions, a seal unit requires the following:

- Sealing medium such as an elastomer (i.e., nitrile, silicon and fluoro rubber). The usage of elastomer is advantageous to tolerate, to some extent, misalignment of the shaft and the vibrations.
- Loading mechanism such as spring to deal with seal wear
- Casing that contains seal and restricts its degrees of freedom

The size of seals varies from few millimeters (for micro-bearings) to few thousand millimeters (for canal locks) because of their wide range of applications, e.g. in motor industry, household appliances, power and pump industries, offshore applications, oil refining, automotive, aerospace, construction, agriculture, hydraulics and pneumatics industries. The construction of a typical commercial oil seal unit is shown in Fig.1.2.1. This seal unit consists of an elastomeric seal, circumferential spring, called garter spring, and metallic (carbon steel, aluminium or brass) casing. A press fit between casing and seal is recommended to avoid rotation of the seal material in the casing. To reduce the extent of press-fit casing-bore and seal outer diameters must be machined to the required tolerances and surface roughness. For proper functioning of the oil seals, the shafts should have a highly polished surface free from scratches and tool marks.

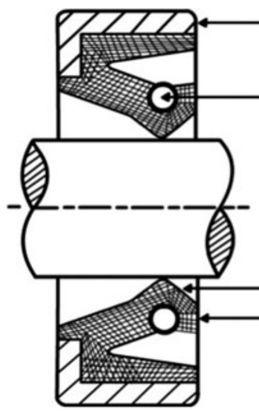

Fig. 1.2.1 Oil seal

Any improper selection or design of the seal may result in failure in the form of leakages, pressure losses, contamination or heat generation that causes reduced efficiencies, energy losses, degraded performance and possibly environmental pollution. The failure costs are generally high as it includes down time costs in addition to seal cost and disassembly & installation costs. As per mechanical design, the diameter of the sealing lip must be slightly less than the shaft diameter so that the seal is elastically deformed while being mounted on the shaft. The elastic deformation and spring action creates contact pressure between the sealing lip (small portion of the elastomer which makes mechanical contact over the surface of the rotating shaft) and the rotating shaft. Too large magnitude of the contact pressure causes excessive friction, resulting in high temperature and rapid wear of the sealing lip. On the other hand, excessive leakage happens due to too little contact pressure. An estimation of the correct contact pressure between the seal lip and the shaft is essential to reduce friction, increase seal life and getting good sealing.

Gasket

If the members of the joint do not have any relative motion between them, then the sealing is achieved by the static seals known as gasket (Fig. 1.2.2). A sealing between jointed members using gasket is achieved by the compression (Fig. 1.2.3) between them. This compression causes gasket material to flow into the imperfections on the gasket seating surfaces for ensuring complete contact between the gasket and its seating surfaces thereby preventing the leakage of the pressurized fluid. Like seal, the basic function of gasket is to prevent leakage from or into the system having relatively stationary components. To be more specific, it can be said that:

- A gasket prevents leakage in a relatively stationary joint (velocities of both the components are the same).
- A seal prevents leakage in a relatively moving joint (velocity of one of the component is greater than velocity of other component).

As there is no relative motion, there is no need of tribology and gaskets can be designed based on mechanical design guidelines. From mechanical design point of view, a gasket is required to fill the space between two mating surfaces and should be able to sustain compressive loading. Due to absence of relative motion between mating parts, 'less-than-perfect' mating surfaces are allowed and surface irregularities are filled by gasket material. In other words gasket material must be able to deform and tightly fill the space including any slight irregularities under compressive loading. The gasket is also many a times subjected to a side load due to internal pressure that tends to extrude it through the flange clearance space. To resist this extrusion the effective compression pressure must be greater than the internal fluid pressure. For reliable sealing the gasket material must be able to withstand high compressive pressure (greater than 15MPa).

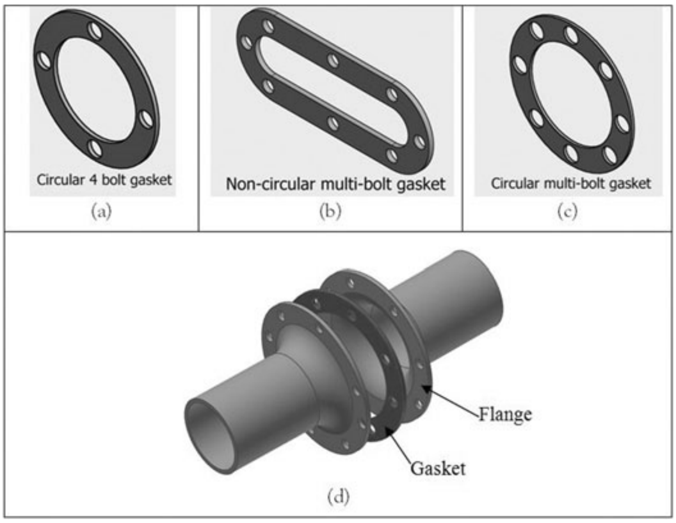

Fig. 1.2.2 (a) Circular 4 bolt gasket, (b) Non-circular multi-bolt gasket, (c) Circular multi-bolt gasket and (d) Gasket assembly

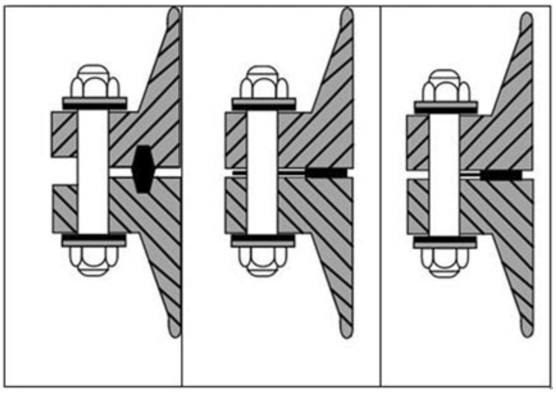

Fig. 1.2.3 Assembled views of different gasket joints

Gaskets are commonly produced by cutting from sheet materials, such as gasket paper, rubber, silicone, metal, cork, felt, neoprene, nitrile rubber, fibreglass, polytetrafluoroethylene (otherwise known as PTFE or teflon) or a plastic polymer (such as polychlorotrifluoroethylene). An example of a gasket used to seal cylinder block and the cylinder head is shown in Fig. 1.2.4.

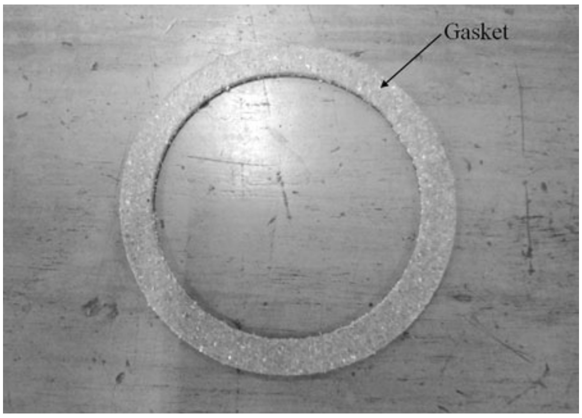

Fig. 1.2.4 Cylinder head gasket

The design of a gasket requires careful consideration of the following aspects:

i. Since the gasket material needs to fill surface irregularities, it must be resilient enough to flow into and fill any irregularities in the surfaces being sealed.
ii. It must remain rigid enough to resist extrusion into the clearance gap between the surfaces under the full system pressure being sealed.
iii. Since a resilient flow is required for gasket closure sufficient closure loading and consequent compressive stresses is required. This requires evaluation of the internal pressure causing leakage problems, the gasket contact pressure produced by the bolt forces, and the gasket materials, which are selected to withstand the operating conditions.
iv. Since the performance of the gasket is degraded due to stress relaxation sufficient contact pressure is required to be maintained to store elastic strain energy to resist relaxation effects. Greater the stored elastic strain energy of the seal greater will be the margin available to resist any relaxation effects during use.
v. Sufficient stresses are required to cause flow of gasket into the imperfections in the seating surfaces. The load required for this deformation is dependent on the gasket material.
vi. The selection of the gasket material must be such that it will withstand the operating pressures and temperature. A material with a low relaxation is preferable as can be used with lower initial compression pressure (otherwise provides higher factor of safety at the same pressure).
vii. Surface finish of gasket seats dictates the thickness and compressibility necessary in the gasket material for providing a physical barrier in the clearance gap.
viii. Flange faces must be parallel and sufficiently rigid to resist distortion on being tightened down and under hydrostatic end loads. Distortion under working loads, often called flange rotation, can appreciably affect the working conditions of the gasket.
ix. The mechanical design of gasket should aim to distribute the load evenly over the whole area of the gasket rather than have a few points of high loading with reduced stress at mid points between the bolts. A more satisfactory arrangement is achieved by employing a large number of smaller diameter bolts rather than fewer bolts of larger diameter. The preload on the bolt must be large enough to achieve minimum seating pressure.

The static sealing of the joint is achieved by proper design of the flanges, selection of gasket type and its material along with its proper installation. The type of flange for a particular application may be selected from ASME standard flanges (which are categorized based on the pressure rating) as given in ASME B16.5. The ASME Unfired Pressure Vessel Code Section VIII, Division I defines the different types of gaskets and their materials to enable selection by the designer as per the operating conditions. The material for the gasket must have good chemical resistance, heat resistance and compressive strength. There are three categories of gasket materials:

1. Non-metallic (elastomers, compressed non-asbestos, PTFE, flexible graphite, mica slabs etc) as given in ASME B16.21
2. Semi-metallic (metallic gaskets containing filler materials like PTFE, flexible graphite, mica, ceramics etc. as given in ASME B16.20 and to be used with ASME standard flanges (ASME B16.5), and
3. Metallic, mostly available in ring type with oval, octagonal and other cross sectional shapes.

The first step in the design of a gasket joint, the selection of the gasket type and its material is made that suits the operating conditions (internal fluid pressure, operating temperature, environment, etc.). The yield value (minimum pressure required to maintain a leak proof gasket joint in the absence of the internal fluid pressure) of the gasket, denoted by 'y', is read from table [3]. Similarly, the ratio between the resultant contact pressure and internal fluid pressure, denoted by 'm', is also read from table [3] corresponding to the given gasket type. This ratio must not be less than a critical value for maintaining leak proof joint because the internal fluid pressure reduces the gasket contact pressure. Since the compression of the gasket is not uniform over its entire compression area due to distortion caused by bolt tightening and fluid pressure, only a narrow band on the outer edge is considered for gasket pressure estimation. It is known as effective gasket yielding width and is denoted by 'b'. The equations for determining this width is available in [1]. The determination of the flange type, bolt size, number of bolts & their arrangement and bolt preload are then determined.

The force in the bolt due to initial tightening is given by:

$$F_g = A_g \,.\, q \qquad \dots (1.1)$$

where F_g is force in the bolt, A_g is the effective gasket area and q is the pressure on the gasket due to bolt tightening.

With the consideration of internal fluid pressure, a force is produced where A_i is the area subjected to internal fluid pressure and p is the internal fluid pressure. The total force on the gasket is thus equal to $A_g\, m\, p$. The total bolt force is given by

$$F_b = p(A_i + A_g \cdot m) \qquad \dots (1.2)$$

The force on gasket must then be equal or greater than F_b.

On equating equation 1.1 and 1.2, we obtain

$$A_g \cdot q = p(A_i + A_g \cdot m) \qquad \dots (1.3)$$

Example: Figure 1.2.5 depicts a joint proposed to be fitted with a gasket for preventing leakage of pressurized fluid. The internal fluid is non-corrosive and is at a pressure 'p' of 12×10^6 N/m^2 (gauge). 24 bolts of 1/2"-12 UNF series are required to keep the cover in place. Select a suitable gasket and bolt loading for ensuring a leak proof joint.

Solution: The figure 1.2.5 depicts a possible gasket arrangement.

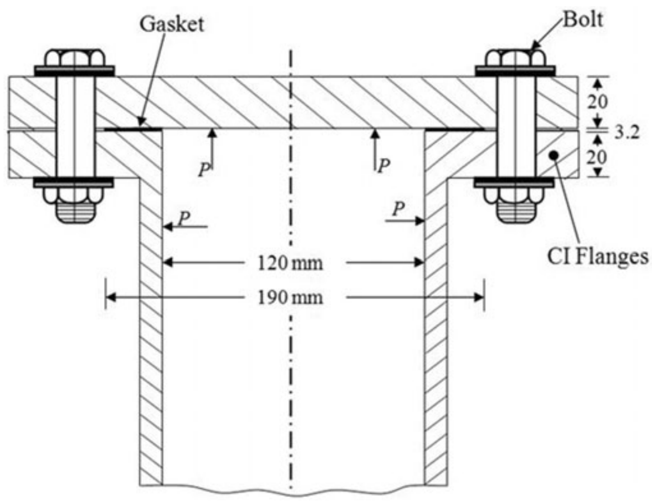

Fig. 1.2.5 Gasket assembly

The fluid is non-corrosive therefore aluminum may be selected as the gasket material. For the aluminum corrugated gasket, the geometry given in figure 1.2.5 may be used. The effective gasket width '*b*' is taken to be 3.2mm. The gasket factor 'm' is 2.5 and yield value for the gasket material 'y' is 20x10^6 N/m^2 (Table 2-5.1, pp 356, Reference [3]).

The area subjected to internal pressure, $A_i = \pi \cdot 0.120 \cdot 0.0032 = 0.0012 \text{ m}^2$

The effective gasket area, $A_g = \frac{\pi}{4}\left(0.190^2 - 0.120^2\right) = 0.017 \text{ m}^2$

Using equation 1.3 with m=2.5, $q = \dfrac{12 \times 10^6 \left(0.0012 + 0.0170 \times 2.5\right)}{0.0170} = 30.85 \times 10^6 \, N/m^2$

Since the gasket pressure is more than the yield value for the gasket material (20x10^6 N/m^2), hence this pressure causes the yielding of the gasket that fills the surface asperities and produces a leak proof joint.

The bolt force required for the joint is $F_b = p\left(A_i + A_g \cdot m\right) = 0.5$ MN.

1.2.2 Tribological design of seals and gasket

Since a gasket requires sealing of stationary members, its design procedure does not involve any tribological considerations. However a seal requires consideration of tribological aspects for its proper design. The seals may be classified into contact seals and clearance (non-contacting) seals. Contact seals bears against its mating surface under positive pressure whereas clearance seals operates with positive clearance (i.e. no rubbing contact). For both kinds of seals, knowledge of tribology is essential.

The majority of the seal types are contact seals, operating with rubbing contact. Friction and wear can be held to a minimum by ensuring that the contact surface is adequately lubricated. In designing a seal, the thickness of oil film between seal faces play an important part in determining seal performance and seal life. If thickness of oil film too small it can be bridged by surface irregularities, producing high friction and rapid seal wear. If too thick, the meniscus will break down, producing a high leakage rate. In practice, the seal will not be perfect under dynamic conditions (i.e. it will not be zero–leakage without excessive preload pressure), and its performance will also depend on load, speed and fluid viscosity.

The prevailing failure mode of contacting seals is wear of seal material. Seal wear is dependent on the contact pressure and surface finish of the surface against which the seal rubs, this in turn being determined to a large extent by the production method. Wear can then be aggravated by lack of lubrication, shaft irregularities, excessive frictional heat, a seal compound which is too soft, etc. To reduce wear and avoid leakage, the sealing interface can be designed targeting almost nil to full hydrodynamic lubrication mechanism. On one hand, wear rate depends on the designed lubrication regime and is the lowest for hydrodynamic lubrication. However, on the other hand the rate of leakage strongly depends on the operating clearance (leakage gap) between the sealing surfaces. Therefore, tribological design of seal is a trade-off between wear rate and leakage rate. It is important to understand that even a modest degree of waviness or slight distortion may provide favourable lubrication or excessive leakage. Therefore a detailed study of these aspects is essential for sustainable seal operation.

For tribological design of seals, consideration of fluid surface tension must be accounted. The usual formula to calculate the pressure due to capillarity is

$$\Delta p = \gamma\left[\frac{1}{R_1} + \frac{1}{R_2}\right] \tag{1.4}$$

where γ = surface tension, R_1 and R_2 = radii of the meniscus in mutually perpendicular planes.

In case of parallel plane surfaces, R_2 can be taken as infinity and R_1 as approximately h/2 where h is the separation of the surfaces. From equation (1.4) it can be said that higher surface tension is desirable to reduce leakage rate. For example, the contact angles of oil against synthetic rubber and steel under industrial conditions are found to be high, so that the sealed oil does not spread along the steel shaft and leakage rate is negligible if pressure difference across the sealing surfaces is lesser than Δp predicted from Eq. 1.4.

The surface tension is a function of temperature; therefore, increase in temperature will increase the leakage rate. In addition, wear rate also increases with increase in temperature; this means tribological design of seal must incorporate thermal analysis.

If the fluid pressure across the seal is over and above the required pressure to overcome the surface tension (Eq. 1.4), an estimate of the volume of the leakage may be made using the following formula:

$$\text{flow/ unit width} = -\frac{1}{12} h^3 \frac{dp}{dx} \tag{1.5}$$

where, η = viscosity,

h = separation distance between the surfaces, and

$\frac{dp}{dx}$ = pressure gradient

1.3 Tribology in Industry (Maintenance)

Let us consider few failed machine components, failure of which could have been avoided using tribological knowledge.

1.3.1 Example: seal

As shown in Fig. 1.3.1, carbon graphite seal is employed to avoid leakage of steam from rotary joints of paper industry. Failure of this component occurs due to adhesive wear. Adhesive wear causes uneven