way it must be effective with the sealed fluid as the 'lubricant'. Satisfactory seal performance depends on effective lubrication of the sliding interface.

1.1.2 Design variants

Many variants of the simple design shown in Fig. 1.1 have evolved to cope with the wide variations in shaft size, speed of rotation, sealed pressure, temperature, and fluid properties that occur in practice, not only in centrifugal pumps, the main field of application, but also to cover other applications such as compressors, agitator shafts, etc. These include the following.

Seal configurations
For example, internally and externally mounted; internally and externally pressurized.

Different types of secondary sealing elements
PTFE wedges, 'U' cups, trapezoidal packings, etc.

Different spring arrangements
Single or multi springs in rotating or stationary configurations.

Bellows seals in metals, rubber, or plastics
Bellows seals eliminate the secondary sliding seal and metal bellows also eliminate the spring.

Alternative materials of construction
Faces can be in monobloc, hard faced, or inserted constructions in a wide variety of materials. A wide variety of materials is used for other components.

Different load/balance configurations
Mechanical seals are available in unbalanced and balanced form. The degree of balance in both types can vary, as will be illustrated later.

Whatever the particular construction, the basic objective in radial face mechanical seal design is to achieve a stable fluid film condition at the seal face such that leakage is limited to acceptable levels while achieving the desired life.

The aim of this chapter is to give an insight into the important parameters in design and highlight the ways in which the performance of mechanical seals can be improved.

1.2 FACTORS AFFECTING DESIGN

The equations developed in this chapter are generally for seals pressurized on their outer periphery. The equations for seals pressurized at their inner periphery follow the same principles.

1.2.1 Closing Force: F_t

The total closing force on the floating member of a seal, F_t is the sum of

The spring load, F_s
The hydraulic load, F_h

$$F_t = F_s + F_h \tag{1.1}$$

The specific closing force p_n is based on the area of the sealing interface, A_f

$$p_n = F_t/A_f \tag{1.2}$$

where

$$A_f = \tfrac{1}{4}\pi(D_o^2 - D_i^2)$$

D_o and D_i are the outside and inside diameters of the sealing interface.

The closing force is reacted in the sliding interface by a supporting force which is provided by some combination of

Hydrostatic fluid pressure
Hydrodynamic fluid pressure
Solid or asperity contact

These forces are discussed further in section 1.2.2 under Lubrication.

Spring load: F_s

All mechanical seals have an arrangement (spring or metallic bellows) to apply an initial closing force to the faces and to hold them together in the absence of fluid pressure (Fig. 1.2). The magnitude of the spring load must be sufficient to overcome any axial friction from the dynamic secondary seal, the dynamic effects of any face misalignments and, for general purpose seals, must allow for the possibility of a vacuum in the seal chamber which, in effect, subjects the seal to a reverse pressure. (*Note.* It is also possible to over-pressure a single mechanical seal in the wrong (reverse) direction if it is fitted with an atmospheric quench and the quench pressure is too high. Similarly with tandem and double mechanical seals it is possible to generate reverse pressures which can displace secondary seals such as 'O' rings. This is at least inconvenient, since the seal must be dismantled to replace them, and at worst potentially dangerous, since a major leakage could occur. Major leakages are possible when the primary seal pressure is re-established in the case of tandem seals or the barrier fluid pressure is low in the case of double seals. Designs which are tolerant to transient reverse pressure effects are available.)

The face pressure due to the spring, p_s, also depends on the area of the sliding interface

$$p_s = F_s/A_f \tag{1.3}$$

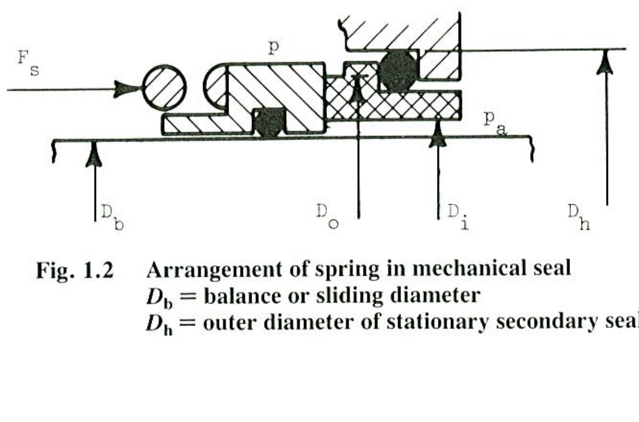

Fig. 1.2 Arrangement of spring in mechanical seal
D_b = balance or sliding diameter
D_h = outer diameter of stationary secondary seal

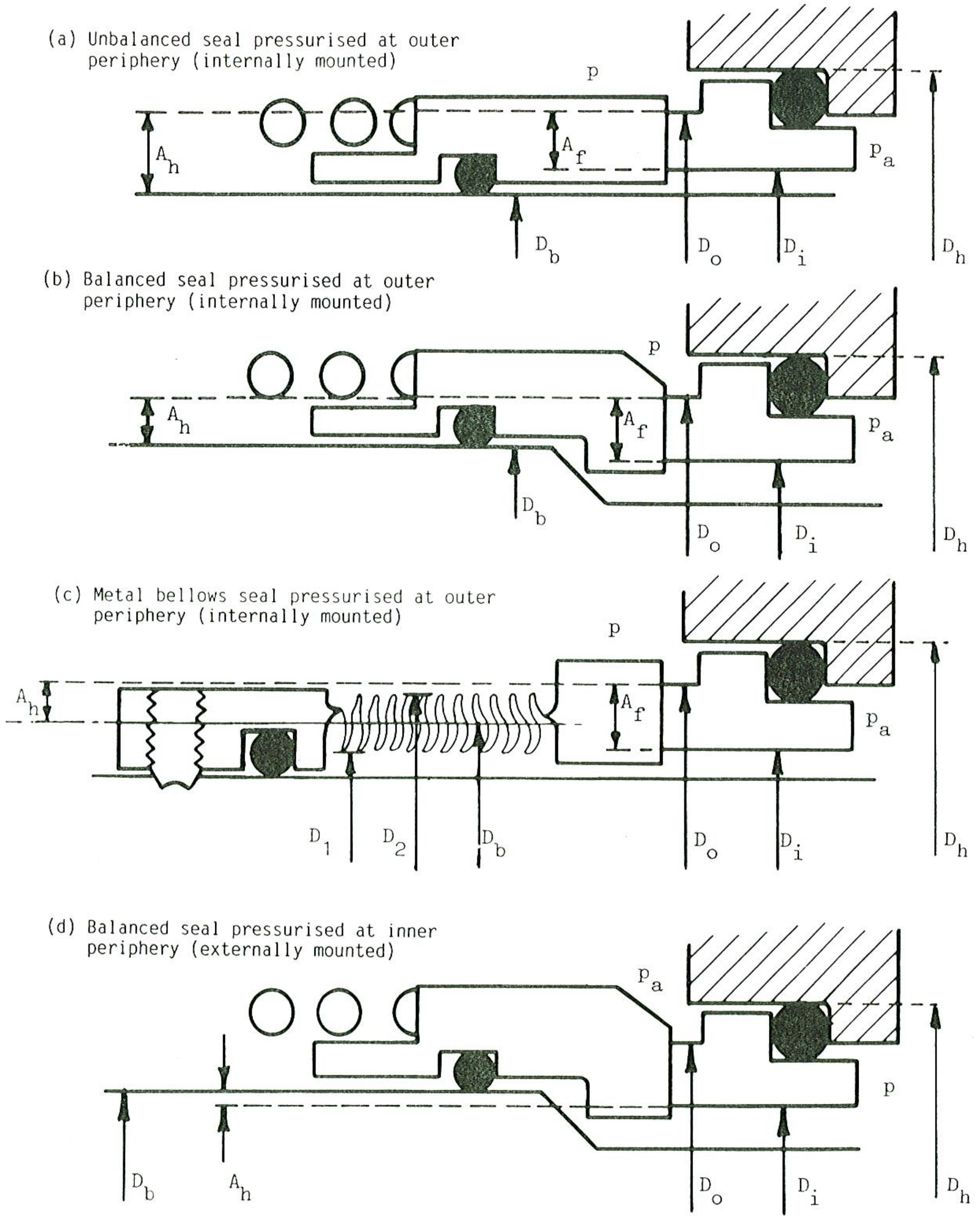

Fig. 1.3 Balance in mechanical seals
A_f = area of sealing interface
A_h = hydraulic loading area
D_2 = outside diameter of bellows
D_1 = inside diameter of bellows
D_o = outside diameter of sealing interface
D_i = inside diameter of sealing interface
D_b = balance diameter
D_h = outer diameter of stationary secondary seal

Typical values of spring load per unit area of sealing interface, (p_s), are 0.1 to 0.3 MPa, though in some designs values as low as 0.02 MPa or as high as 0.5 MPa may be used.

To resist vacuum loads

$$F_s > \tfrac{1}{4}\pi(D_h^2 - D_b^2)(p_a - p) \tag{1.4}$$

or if the stator is clamped

$$F_s > \tfrac{1}{4}\pi(D_o^2 - D_b^2)(p_a - p) \tag{1.5}$$

where

p = absolute pressure on outer periphery of seal
p_a = absolute pressure on inner periphery of seal

Internal/external mounting

Internal mounting is the most common arrangement with the seal mounted inside the pump or sealed vessel. In this arrangement the higher pressure liquid is in contact with the outer diameter of the seal faces. Such an arrangement is shown in Fig. 1.3 (a), (b), (c).

An externally mounted seal is mounted outside the pump or vessel. The higher pressure liquid is in contact with the inner diameter of the seal faces. Such an arrangement is shown in Fig. 1.3 (d).

Balance ratio: B

A fundamental characteristic of a mechanical seal is its 'balance ratio'.

Seals in which the balance ratio is equal to (or greater than 1) are referred to as unbalanced seals; those in which it is less than 1 as balanced seals. Unbalanced seals are used for lower pressures, normally 1 MPa g (10 bar g) max. in simple applications; balanced seals are used for higher pressures, typically up to 7 MPa g (70 bar g) with general purpose balanced seals, and higher with high duty designs.

All mechanical seals contain an area within their design which is responsible for generating the hydraulic closing forces in the seal. This area is annular, enclosed by the outside diameter of the sealing face (D_o) and the balance diameter (D_b). It is the area of the seal face compared to this hydraulic area which determines whether a seal is balanced or unbalanced and this, in turn, determines the prevailing bearing pressure at the faces and the type of fluid film formed and, hence, the seal's ultimate pressure capability.

Pusher seals

The balance ratio, B, can be derived from the following formulae

for an externally pressurized seal (internally mounted)

$$B = \frac{\text{hydraulic loading area}}{\text{sealing interface area}} = \frac{\tfrac{1}{4}\pi(D_o^2 - D_b^2)}{\tfrac{1}{4}\pi(D_o^2 - D_i^2)} = \frac{(D_o^2 - D_b^2)}{(D_o^2 - D_i^2)} \tag{1.6}$$

for an internally pressurized seal (externally mounted) the hydraulic load area is $\tfrac{1}{4}\pi(D_b^2 - D_i^2)$

$$B = (D_b^2 - D_i^2)/(D_o^2 - D_i^2) \tag{1.7}$$

Figure 1.3 shows that balanced seals have face areas which are larger than the hydraulic loading areas by disposing the face area above and below the balance line as indicated. This requires a step in the shaft or sleeve; whereas, in the case of unbalanced seals, the face area is smaller than the hydraulic area; unbalanced seals may be applied directly to a parallel shaft.

The balance ratios of commercially available balanced seals vary typically from 0.65 to 0.85. The choice of balance results from the compromise effected by the particular seal manufacturer. The choice may be influenced by the seal type and its characteristics of design and a compromise between the opposing requirements of high sealing integrity and controlled face loading.

Seals with a high balance ratio, e.g., $B = 0.85$, are more stable and less likely to blow open under the action of hydrostatic pressure between the faces, but at the expense of higher face loadings. Equally, seals with a lower balance ratio, e.g., $B = 0.65$, exhibit lower face loadings and lower amounts of heat generated, but can be hydraulically unstable at higher pressures if not designed carefully.

Unbalanced seals typically have balance ratios of around 1.2; the degree of unbalance depends on the clearance between the shaft and the seal face inner circumference.

Bellows seals

Bellows seals achieve balance without a step in the shaft since the 'O' ring at the sliding diameter is eliminated and the balance or sliding diameter (D_b) is 'built-in' to the bellows. The sliding or balance diameter lies between the inside and outside diameters of the bellows (D_1 and D_2) and, for metal bellows, is normally calculated as the root mean square of those values

$$D_b = \sqrt{(0.5(D_2^2 + D_1^2))} \tag{1.8}$$

Equation (1.8) applies with zero differential pressure across the bellows. With edge welded bellows differential pressure tends to deform the bellows in such a way as to move the balance diameter towards the low pressure side, thereby increasing the balance ratio. Fig. 1.4 shows a typical example of such change.

Nested bellows in which the neighbouring leaves conform closely to each other are the most susceptible. Edge welded bellows of a more open configuration are less susceptible and formed bellows are least susceptible to this effect.

In seals with metal bellows, the spring function is normally performed by the bellows itself, whilst rubber and plastics bellows usually incorporate a metallic spring.

Balance ratios are calculated as previously described, using the balance diameter in conjunction with the seal face diameters (equations (1.6) (1.7)).

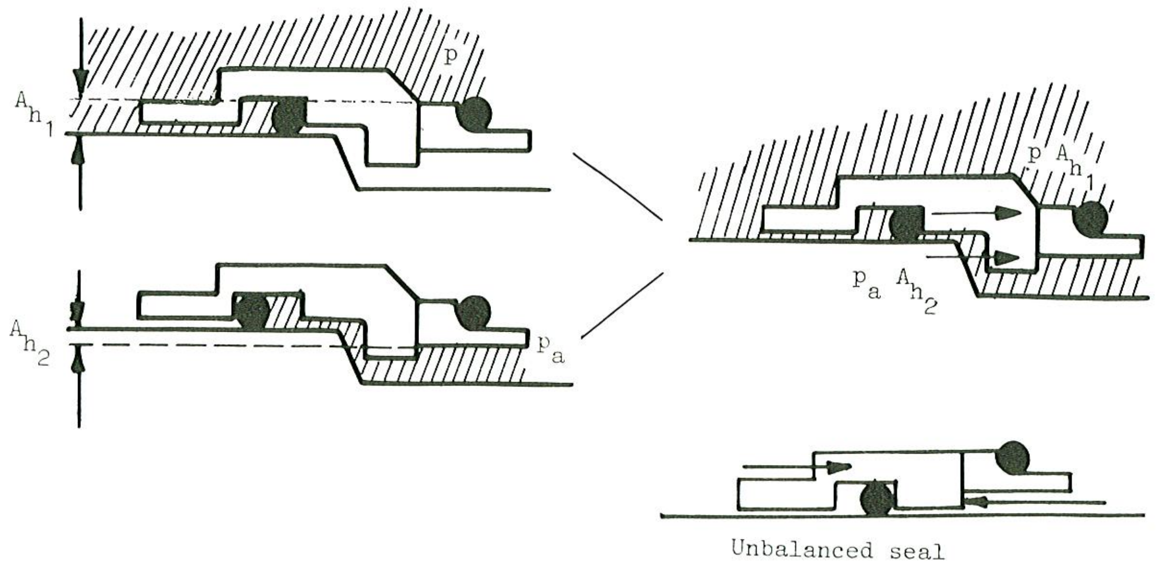

Fig. 1.5 The elements of hydraulic load in a seal

(*Note.* For an unbalanced seal, $B > 1$, the second component of force, derived from p_a acts to reduce the closing force rather than increase it.)

Total closing force: F_t

The total closing force in a mechanical seal is thus very predictable and depends on the initial spring load and the hydraulic closing force.

Total closing force (F_t)

= Spring force (F_s) + Hydraulic closing force (F_h)

$$F_t = F_s + A_f(\Delta p \times B + p_a) \quad (1.11)$$

It is the way in which this load is supported that is not so predictable, as will be described in later sections.

For very high pressure ($p > 30$ bar) seals, sealing to atmosphere ($p_a = 1$ bar a), p_a may be neglected, and equation (1.11) simplifies to

$$F_t = F_s + p \times B \times A_f \quad (1.12)$$

Examples are given wherever possible in this chapter and the 70 mm size seal used here appears regularly throughout.*

Example 1.

A balanced seal (sealing to atmosphere), 70 mm size, sealed pressure: 1.1 MPa (1 MPa g)

$p_a = 0.1$ MPa	$D_i = 80.17$ mm
$p = 1.1$ MPa	$D_b = 82.45$ mm
$\Delta p = 1.0$ MPa	$F_s = 285$ N
$D_o = 87.17$ mm	

$$B = (D_o^2 - D_b^2)/(D_o^2 - D_i^2)$$
$$= (87.17^2 - 82.45^2) \div (87.17^2 - 80.17^2)$$
$$= 0.683$$
$$A_f = \tfrac{1}{4}\pi(87.17^2 - 80.17^2)$$
$$= 920\ \text{mm}^2\ (1.43\ \text{in}^2)$$
$$F_t = 285 + 920(1 \times 0.683 + 0.1)$$
$$= 1005.4\ \text{N}\ (226.2/\text{lbf})$$

Example 2.

An unbalanced seal, 70 mm dia, with the same face area, spring load, and duty conditions. Typical value of $B = 1.2$.

Then

$$F_t = 285 + 920(1 \times 1.2 + 0.1)$$
$$= 1481\ \text{N}\ (333.2\ \text{lbf})$$

The above examples show the fundamental difference in face load between balanced and unbalanced seals. This is not, however, entirely responsible for their different pressure capability, as will be explained in section 1.2.2 under Film formation.

1.2.2 Lubrication

Some form of lubrication between the seal faces is necessary to control frictional heat generation and wear of the faces. Three modes are possible:

Hydrostatic film
Hydrodynamic film
Boundary lubrication

In hydrostratic lubrication the faces are separated by the pressure of the sealed fluid. The pressure in the

* In the examples in this chapter, seal size is based on the maximum shaft or sleeve which will pass freely through the seal. Different conventions are used by different manufacturers and in many cases, the balance diameter itself is the basis for sizing the seal.