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Guidelines for Designing Critical Parameters of Split Mechanical Seal

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***Abstract*—Split mechanical seal has an advantage over non-split type mechanical seal in the area of sealing large rotary equipment, as it takes very less installation time and it can be installed without removing any component of rotary equipment. Many theories regarding the different parameters affecting design of split mechanical seal have been published, however, literatures which discusses the complete design procedure of split mechanical seal is not readily available. An attempt is made in this work to prepare the design guidelines for critical features of split mechanical seal such as balance ratio selection, design against centrifugal leakage and selection of reference plane. Manufacturing method of ring halves is also described which is critical for sealing of gap between ring halves. In split mechanical seal, since leakage across mating faces of split halves is major concern, a testing procedure is presented for checking any leakage before installation.**

***Index Terms*— Mechanical seal; Split design; Balance ratio.**

I. INTRODUCTION

Mechanical seal is used in wide variety of rotary equipment to prevent leakage of pressurised fluid from housing and rotating shaft. Sealing is done by mainly two rings, a stationary ring and a rotating ring. Stationary ring is fixed on stuffing box while rotary is fixed to shaft by means of some flexible element. Mating faces of two rings are lapped, this ensures no leakage across mating faces. This type of seals are called non split seal in which two rings are installed axially on shaft. In some application, especially in large rotary equipment installation and removal of non- split seal is time consuming as it requires removal of bearings and sometimes casing too. This problem has been solved by splitting sealing rings and all other parts radially into two parts so that all the parts of this spit design can be installed radially around the shaft. Design of split seal and non-split seal are similar but in split design splitting of all the components creates more gaps through which leakage must be prevented. Also when ring is split into two halves which can move relative to each other while shaft is rotating hence their alignment and selection of proper reference plane is also necessary. Selection of proper balance ratio is necessary in non-split as well as split design. Lebeck [1] presented theoretical model for selection of stable balance ratio range for two phases, single (Water) and multicomponent mixtures (Propane-methane). Buck [2] described effect of seal face profile on pressure gradient factor. Fliteny [3] presented guidelines regarding selection of flexible element such as single spring, multi-spring or metal bellows for different applications. He also explained guidelines for selection of mechanical seal configuration, considering environment control and application. Lebeck [4] proposes model to predict seal performance (leakage, co-efficient of friction and friction power) considering convergent and divergent seal face contact profile. He developed equations to calculate leakage and

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friction power loss, considering hydrostatic equilibrium of a seal ring. Lebeck & Chiou [5] explained parting of seal faces due to phase transition of sealed liquid from liquid to vapour cross the seal face. Goilkar & Hirani [6] carried out experimental study on friction and wear of carbon-graphite seal ring material for four different balance ratio values to explore effect of operating conditions on the wear rate and frictional loss. Karassik & McGuire [7] presented analytical model for calculation of wear and concluded that wear is directly proportional to pressure multiplied by mean face velocity, and coefficient of wear, but inversely proportional to seal face material hardness. They provided value of coefficient of wear for different face material combination.

II. Selection of Balance or Unbalanced Design

Balance ratio is the ratio of hydraulically loaded area (A_H) to sliding face area (A_F) (Fig. 1). A fluid film is present between to mating faces of seal rings. This fluid film provides means of lubrication between two rubbing faces. Balance ratio should be selected carefully to maintain fluid film between mating faces. Dimensions of seal rings are obtained from the value of balance ratio.

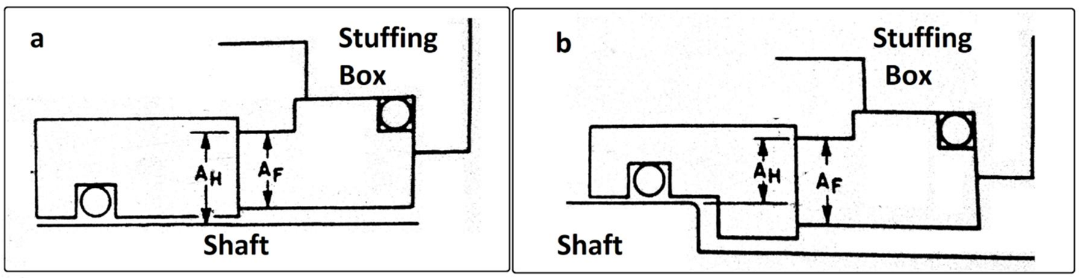

Fig.1 [6](a) unbalanced design; (b) balanced design

Before selecting value of balance ratio, it is necessary to decide whether balance seal or unbalanced seal is to be design (Fig.1). Selection of balanced or unbalanced design depends on viscosity of sealed liquid and severity of operating condition. For balanced design valued of balance ratio is greater than one and for unbalanced design balance ratio has valued less than one. Balanced seal design provides higher life compared to unbalanced design. But balanced design is more prone to leakage compared to unbalanced design. So care should be taken while selecting minimum balance ratio for balanced design. For sealed fluid having viscosity less than water balanced seal is preferred. It is preferable to select balanced design for high and maximum severity condition [8]. Severity of operating conditions has been classified using PV factor (Table 1) [8]. When leakage of fluid is major concern compared to life of seal than unbalanced design is recommended. Unbalance design has balance ratio garter than one which ensures leak proof design.

TABLE I. [10]. Classification of seal working condition

Severity of Operating Conditions	Sealed Pressure(P), MPa	Sliding speed (V), m/s	PV-factor MPa m/s
Low	$P \leq 0.1$	$V \leq 10$	$PV \leq 1$
Medium	$P \leq 1$	$V \leq 10$	$PV \leq 5$
High	$P \leq 5$	$V \leq 20$	$PV \leq 50$
Maximum	$P > 5$	$V > 20$	$PV > 50$

III. Selection of Balance Ratio for Balanced Design

According to Mayer [8] sealed liquid with low viscosity (such as propane and butane), the seal must be designed with lowest balance ratio of 0.7, so as not to lose the seal surface contact. According to Mayer [8] with water, balance ratio of 0.58 to 0.6 is sufficient to avoid parting of the seal faces. Referring to Lebeck [1] one can select stable balance ratio range for two phase single and multi-component liquid for given operating temperature of sealed fluid. Moreover his modelled decreases in stable balance ratio range with increase in operating temperature (Fig.2).

For an application satisfying conditions such as, seal face operates on low temperature such that two phase fluid film present between two mating faces and therefore one would have wider stable balance ratio range, Spring

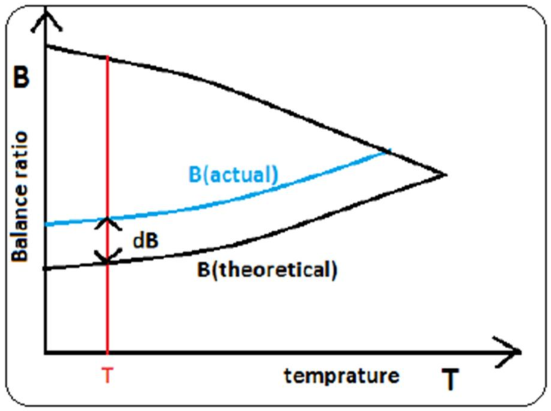

Fig.2 relation between balance ratio range and temperature

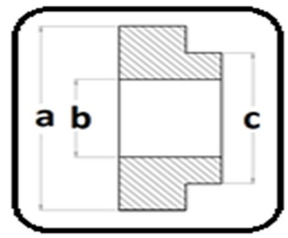

Fig.3. Design of rotary ring

pressure is negligible compared to pressure of sealed fluid, one can determine actual (minimum practically possible) balance ratio by using Equation 1.

$$B(actual) = B(theoretical) + dB + allowance \tag{1}$$

One can obtain empirical relation of dB as a function of parameters such as; seal face geometry, spring pressure, shaft axial run out, pressure fluctuation of sealed fluid, secondary seal friction, and sealed fluid viscosity.

IV. Design of Rotary Ring

Rotary ring is mounted to a shaft via a flexible element. As rotary ring wears out more frequently compared to stationary ring design of seal face area of rotary ring is more important. Once balance ratio (B) is selected dimension of rotary ring (Fig. 3) can be found using Equation 2.

$$B = \frac{a^2-b^2}{c^2-b^2} \tag{2}$$

Dimension b can be found by adding appropriate clearance to shaft diameter. Dimension c in Figure 2 is obtained considering crushing strength of rotary ring material (Equation 3).

$$\sigma_c = \frac{F_c}{\frac{\pi}{4}(c^2-b^2)} \tag{3}$$

Where σ_c= crushing strength of rotary ring material; F_c= net closing force on rotary ring
Net closing force can be determined using Equation 4.

$$F_c = F_p - F_o \tag{4}$$

Where F_p= pressure force on hydraulically loaded area of rotary seal ring; F_o=opening force on rotary ring exerted by fluid film present between two mating rings

$$F_p = P \times \frac{\pi}{4}(a^2 - b^2) \tag{5}$$

$$F_o = 0.5 \times P \times \frac{\pi}{4}(c^2 - b^2) \qquad (6$$

Where P = sealed pressure.
Here force applied by spring is neglected which in fact true in all practical application.

V. Design of Spring

Spring is essential to keep rotary and stationary ring face in contact at low or zero fluid pressure. Spring pressure is important factor as it impacts seal life. Optimum seal pressure maintains contact between faces at low pressure and simultaneously maintains a fluid film, which acts as a lubricant, between mating faces of seal rings. Since optimum spring pressure is dependent on wide range of variables it is calculated using empirical Equation 7.

$$P_s = (-1.0974d + 0.2105) \times 10^6 \tag{7}$$

Where P_s=spring pressure in Pa; d=shaft diameter in m
Stiffness of spring is calculated using Equation 8.

$$K = \frac{P_s \times \frac{\pi}{4}(c^2 - b^2)}{\delta} \tag{8}$$

Where K= stiffness of spring; δ = working deflection of spring; P_s= spring pressure
Working deflection of spring depends on type of seal.

VI. Prevention of Leakage Due To Centrifugal Force

Due to centrifugal force on two halves of a rotary ring, it's mating faces gets separated. Sealed fluid can leak from these openings. In order to prevent leakage from centrifugal opening, one must provide radially inward force which is greater in magnitude than centrifugal force. This is done by feature of rotary ring so called tapered rotary design [9]. Due to taper surface, radial component of reaction force of O ring provides necessary inward radial force (Fig.4).

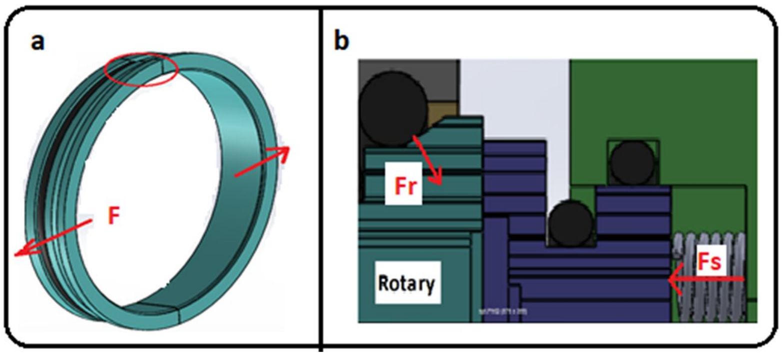

Fig.4 (a) centrifugal force on rotary ring halves; (b) tapered rotary design

Centrifugal force on rotary halves is calculated using Equation 9.

$$F = m \times \omega^2 \times r \tag{9}$$

Where F= Centrifugal force; m = Mass of rotary halve; ω = Shaft speed in rad per sec; r = Centroid radius of ring halve.
Radial force can be calculated using Equation 10.

$$F_r = F_s \times \cot(\propto) \tag{10}$$

Where F_r = inward radial force; F_s= spring force; ∝= taper angle of rotary ring.
If F_r is greater than F then design is safe against centrifugal leakage.

VII. Manufacturing and Alignment of Split Ring Halves

After seal rings are turned on lathe their faces are lapped on same lapping flat. This creates two equal surfaces. Two V notches are made at 180° on inner surface of seal ring [10]. The opposite force is applied on opposite sides of a ring. This creates stress Concentration near tip of V notch. Hence crack is produced at tip of V notch and propagates towards the outer surface. This produces two perfect halves of a ring. When to halves are mated, due to perfect hill to valley connection, no leakage is possible across the mating surfaces. Two halves of both the seal ring must be precisely aligned otherwise leakage would occur across the mating faces. During installation O ring provides necessary radially inward force which keeps mating faces of two halves in contact and properly aligned. Mating surface of two halves is rough, hence high static friction between mating surface compared to smooth surface. This prevents movement of two halves with respect to each other when they are subjected to small vibrations. Lapped surface is taken as reference plane (Fig.5), so that two halves of a ring does not move with respect to each other during movement of reference plane due to shaft axial run out.

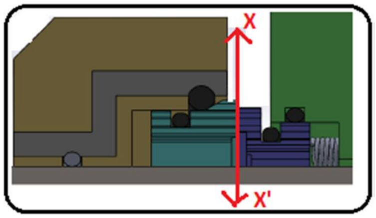

Fig.5 Lapped surface as a reference plane

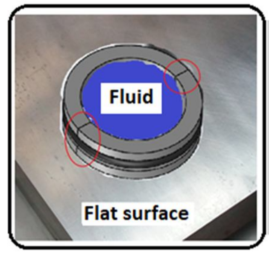

Fig.6 leakage testing setup

VIII. TESTING OF LEAKAGE FROM A RING

Two split halve of a ring is assembled using O ring. Then lapped surface of the ring is placed on same lapping flat on which the ring was lapped and water is poured inside the ring (Fig.6). During assembly of carbon ring halves, if by mistake mating surfaces of two halves moved relative to each other than some carbon particles are removed from surface. In that case water escapes across mating faces of two halves. But if assembly is done carefully than no leakage is possible.

IX. CONCLUSIONS

In this paper, design guidelines are presented for selecting value of balance ratio. Method to calculate rotary ring dimensions using value of selected balance ratio is described. Calculation of stiffness value of flexible element is presented. In split design, a method to prevent leakage due to centrifugal force on rotary ring halves is described. Manufacturing and alignment guidelines of split halves of a ring is described which is critical to avoid leakage between mating faces of two halves of a ring. A procedure to check for any leakage across mating faces of ring halves is described, which is important before installation of split mechanical seal.

REFERENCES

[1] Lebeck, A.O. 1988. Face Seal Balance Ratio Selection for Two Phase Single and Multicomponent Mixtures. In Proceedings of the 5th International Pump Users Symposium.

[2] Buck Jr, G. S. 1978. The Role of Hydraulic Balance in Mechanical Pump Seals. Proceedings of the Seventh Turbo machinery Symposium.

[3] Flitney, R.K.. 1977. Factors affecting mechanical seal design and application. Tribology international. 10(5):267-272.

[4] Lebeck, A. O. 1988.Contacting mechanical seal design using a simplified hydrostatic model. Tribology international 21(1): 2-14.

[5] Lebeck, A.O. & Chiou, B.C. 1982. Two-Phase Mechanical Face Seal Operation: Experimental and Theoretical Observations. In Proceedings of the 11th Turbo machinery Symposium.

[6] Goilkar, S., & Hirani, H.2010.Parametric study on balance ratio of mechanical face seal in steam environment. Tribology International. 43(5): 1180-1185.

[7] Karassik, I. & McGuire, J.T.2012. Centrifugal pumps. Springer Science & Business Media.

[8] Mayer, E.2013. Mechanical seals. Heinemann: Butterworth.

[9] Clark, M.S. & Azibert, H.V. 1998. Securing and centering devices for a split mechanical seal. Chesterton AW Co. U.S. Patent 5,725,220.

[10] Azibert, H.V.1986. Split mechanical face seal. , Chesterton AW Co. U.S. Patent 4,576,384.