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Analysis of Mechanical Seals for High-Speed Centrifugal Gas Compressors

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Abstract: A study aimed at seal selection efficiency for centrifugal pumps in the oil and gas industry is presented. A detailed analysis of mechanical seals in use in exploration and production activities of the oil and gas sector was undertaken. The approach of analysis was using seal design equations as mathematical models for simulating the performance of the mechanical seal. The results showed a mechanical seal with balance value of 0.5, an increased surface area between mating surfaces; provided with a flush system to enhance cooling and with seal face gap of 50 mm or less between the mating surfaces for minimal or zero leakage. The obtained results can aid the industries in seal selection and seal manufacturers in seal specifications.

Keywords: Centrifugal compressor, mechanical seal, seal face

INTRODUCTION

A mechanical seal, otherwise known as mechanical end face seal, can be defined as a machine element that prevents fluid from escaping a container at a rotating shaft extending through its housing utilizing axial force to keep the end faces of the primary sealing elements, which are 90^o to the shaft axis and moving relative to each other, in contact (Schoenherr, 1963). The distinguishing characteristic is that the dynamic sealing takes place perpendicular to the shaft axis not parallel to it such as in lip seals, throttle bushings or conventional packing. Figure 1 is a simplified version of a typical end face seal.

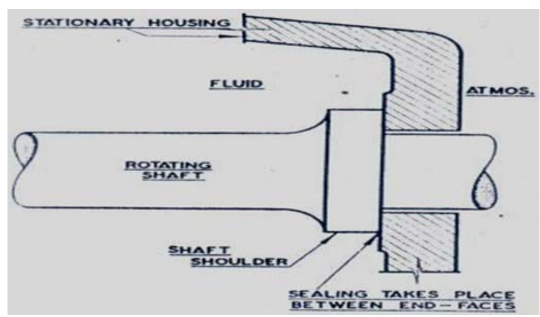

Fig. 1: A simplified version of an end face seal

A compressor is a machine that increases the total pressure of the gas stream to that required by the cycle while absorbing the minimum shaft power possible. In centrifugal compressors, flow is radial. A centrifugal compressor consists essentially of a rotating impeller followed by a radial diffuser. The impeller inlet is called the inducer, or eye and the outlet the exducer. The impeller has a tip clearance relative to a stationary shroud and has seals relative to a plate. Centrifugal compressors are the second most widely used compressor after reciprocating machines. A centrifugal compressor functions by increasing the velocity of the gas as it passes through an impeller. The velocity is then reduced in the diffuser at the same time as the temperature and pressure increase. Very high speed compressors are available, with speeds up to 30,000 RPM and discharge pressures of up to 700 bar are achievable. Centrifugal compressors offer compactness, simplicity and ease of maintenance, but require a high level of sophistication in design and selection (Shell Internationale Petroleum Maatschappij, 1991). According to Shell Internationale Petroleum Maatschappij (1991), in EP (Exploration and Production) duty, the mechanical seal is the most common. Other seals usually used include lip seals and packed gland seal. A mechanical seal depends for its satisfactory operation on the maintenance of a 10 micron lubricating film of the sealed fluid between the rotating and stationary seal faces, so the principal limitations on satisfactory seal performance are:

- Seal temperature
- Sealing pressure
- Presence of abrasives
- Gas bubbles
- Vibration or shaft misalignment

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Mechanical seals first appeared (in a crude form) at the turn of the century. Turbine designers were looking for a more positive sealing device. A more sophisticated form is shown in a patent dated 1913 (Wilkinson, 1913). The inventor shows a so-called double-seal with cooling in the seal chamber and mentions the concept of seal balance. The seal is considered of the controlled leakage type. There is no axial mechanical loading device such as a spring to keep the faces in intimate contact. By 1919 a patent appeared in which springs were shown (Doran, 1919). They push a ring-shaped piece against a shaft shoulder. The inventor calls it packing for steam turbines, although in present-day terminology it is a single-mechanical end face seal. In the late 1920's designers and engineers particularly in the refrigeration and automotive fields began demanding more positive sealing devices (Schmitz, 1947). The mechanical seal responded to the challenge. By 1945 its reputation had been established. Pump manufacturers and the chemical industry have developed a great demand for such a seal. Recent studies include Seals classification by Schoenherr (1965), Shaft Sealing Systems and Dry Gas Seals (Shell Internationale Petroleum Maatschappij, 1991).

High-speed centrifugal compressors are characterized by Froude numbers greater than unity. Froude number (Fr) is the ratio of stream velocity to the velocity of propagation of wave. The objective of this research is to study mechanical seals with a view to predicting the performance of seals in its operating conditions in a High-Speed Centrifugal Gas Compressor.

Frequent failure of mechanical seals used in gas compressors, particularly high-speed centrifugal compressors and the need to correct this trend is posing enough challenges to design engineers. Considering that these compressors run at high velocities and being radial in nature, there is frequent and constant interaction between the seals and the shroud in one case and the back plate in the other. Another major reason is the need to eliminate leakages in gas compressors. A proper design of mechanical seals will help achieve this objective. Seals are primarily meant to eliminate leakages.

The compressor shaft seal is designed to prevent the process gas from escaping along the shaft to atmosphere. In non-toxic and non-hazardous environments, a labyrinth seal is utilized which allows a small leakage of gas to the atmosphere. In the EP environment with the hazard of allowing hydrocarbons escape to the atmosphere, a seal arrangement is utilized to contain the process gas within the compressor. The traditional method has used one of two types of seal, which require the injection of sealing oil at a pressure greater than that of the gas pressure in the sealing chamber. However, the recent developments in dry gas seals have effectively eliminated the requirement for seal oil systems from new installations (Shell Internationale Petroleum Maatschappij, 1991). The prime motivation for the introduction of dry gas seals instead of using the traditional seal oil arrangement has been on the grounds of safety. Information from both manufacturers and users indicates that over 70% of all centrifugal compressor operational faults are attributable to lube and seal oil systems. These problems can lead to fires and leakage of toxic gas, both with major safety and environmental implications. A seal oil system involves significant hardware, is often difficult to troubleshoot and provides a weight and space impact, which may be important in an offshore environment. A dry gas seal system for a new installation is less expensive and user experience to date indicates that the systems are at least as reliable as oil systems, but have a much lower risk of fire or gas leakage.

Therefore justification of dry gas seals on new installations can also be made on the basis of capital savings, especially if carried out in conjunction with magnetic bearings as the complete compressor lube oil system can be eliminated. On existing machines with seal oil systems, retrofit of dry gas seals can be difficult to justify unless serious reliability problems exists with the seal oil system which cannot easily be resolved, or there are possible major consequences resulting from a gas leakage. Dry gas seals are now considered standard for all new EP equipment purchases. Detailed analyses of mechanical seals for high speed gas centrifugal compressors will therefore, help the compressor engineer select an appropriate seal.

The need to design seals that can withstand the high operating speeds gas compressors run in, while properly eliminating gas leakages motivated this research. The analysis problem will be studied using seal design equations as mathematical models for predicting the performance of industrial mechanical seals using simulation.

FORMULATION AND SOLUTION

The sealing surfaces are perpendicular to the shaft, with contact between the primary and mating rings to achieve a dynamic seal. The primary ring is flexibly mounted in the seal head assembly, which usually rotates with the pump shaft and the mating ring is usually fixed to the pump gland plate. Each of the sealing planes is lapped flat to eliminate any visible leakage. Wear occurs at the seal faces from sliding contact between the primary and mating rings. The amount of wear is small, as a film of the liquid sealed is maintained between sealing faces. Normally the mating surfaces of the seal are of dissimilar materials and held in contact with a spring. The preload from the spring is required to produce the initial seal. The spring pressure holds the primary and mating rings together during shutdown or when there is a lack of liquid pressure.

Any seal installation is made up of two assemblies. The seal head assembly includes the primary ring and its associated component parts. The mating ring assembly includes those parts required for the mating ring to function. During operation, one of the components is stationary (Netzel, 1986).

There are three points of sealing common to all mechanical seal installations:

- At the mating surfaces of the primary and mating rings.
- Between the rotating component and the shaft or sleeve.
- Between the stationary component and the gland plate.

When a seal is installed on a sleeve, there is an additional point of sealing between the shaft and sleeve. Certain mating ring designs may also require an additional seal between the gland plate and the stuffing box. The secondary seal, between the rotating seal component and the shaft or sleeve, must be partially dynamic. As the seal faces wear, the primary ring must move slightly forward. Because of vibration from the machinery, shaft run out and thermal expansion of the shaft against the pump casing, the secondary seal must move along the shaft. This is not a static seal in the assembly. Flexibility in sealing is achieved from such secondary seal as bellows, O-ring, wedge, or V-ring. Most seal designs fix the seal head to the shaft and provide for a positive drive to the primary ring.

The mating ring is usually a separate replaceable part. A static seal is used to prevent leakage between the mating ring and the gland plate. The static seal and the mating ring form the mating ring assembly. Although mechanical seals may differ in various physical aspects, they are fundamentally the same in principle. The wide variation in design is a result of the many methods used to provide flexibility, ease of installation and economy (Netzel, 1986).

Seal balance/mechanical seal hydraulic balance: The greatest concern for a seal user is the dynamic contact between the mating seal surfaces. The performance of this contact determines the effectiveness of the seal. If the seal load at the faces is too high, the liquid film between the seal rings could be squeezed out or vapourised. An unstable condition would result, with a high wear rate of the sealing planes. The power would increase as the friction increased with solid contact. Seal face materials also have a bearing limit, which should not be exceeded

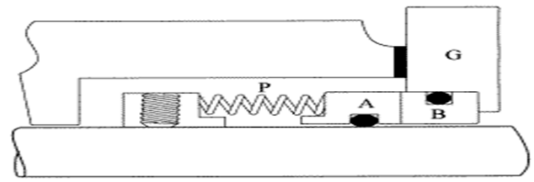

Fig. 2 : Seal hydraulic balance

(Netzel, 1986). Seal balancing can avoid these conditions and lead to a more efficient installation.
Hydraulic balance is very easy to understand, using Fig. 2:

A = The spring loaded face with an area of say 6 cm^2
B = The stationary face held to the front of the stuffing box by gland "G"
P = The hydraulic pressure in the stuffing box is given as say 10 Kg/cm^2

To understand hydraulic balance you must know that:

- Pressure (Kg/cm^2) × Area (cm^2) = Force (Kg.*)

*: Multiplying this number by gravity (9.8 m/sec^2) gives Newtons of force.
There are at least two forces closing the seal faces:

- The mechanical spring force.
- The hydraulic force caused by the stuffing box pressure acting on the seal face area.

There are at least three forces trying to open the seal faces:

- A hydraulic force is created any time there is fluid between the seal faces.
- A centrifugal force created by the action of the fluid being thrown outward by the rotation of the pump shaft.
- A hydrodynamic force created because trapped liquid is, for all practical purposes, non-compressible.

Closing forces:

- A spring load of 2 Kg/cm^2 is an industry standard when the seal face is new and a load of 0.7 Kg/cm^2 should still be available when the carbon seal face has worn away. We need this minimum load to prevent normal vibration from opening the lapped faces. This load is set by installing the mechanical seal with the proper amount of compression as shown on the mechanical seal installation print. A tolerance of plus or minus 0.8 mm is typical.
- Since the definition of hydraulic force was given as pressure X area :
- 10 Kg/cm^2 × 6 cm^2 = 60 Kg of closing hydraulic force.

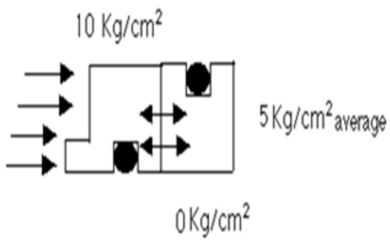

Fig. 3: Linear pressure drop in a seal face

Opening forces:
Hydraulic force:

- Testing shows that sometimes there is a film of liquid between the faces, sometimes there is only vapor, sometimes there is nothing at all and sometimes there is a combination of all three. This means that if there is liquid or vapor between the faces, it is under pressure trying to force the lapped faces apart. The stationary face (B) cannot move because it is being held by gland "G", but the spring loaded face (A) will respond to this force.

  1Using Fig. 3, if we assume a straight line or linear pressure drop across the seal faces, we would get an average of:
  5 Kg/cm^2 × 6 cm^2 = 30 kg of force trying to open the seal faces.

- Centrifugal force is acting on the spring loaded face (A) trying to spin it perpendicular to the rotating shaft.
- Stationary face (B) is not perpendicular to the shaft because it is referenced against the stuffing box face which is a casting that is not perpendicular or square to anything. A gasket located between the gland and the stuffing box further compounds the problem. Testing has shown that a surface speed of 25 m per sec centrifugal force is powerful enough to open most mechanical seal faces (McNally, 2004).
- Seal faces are lapped to within three helium light bands or slightly less than one micron. This slight waviness is enough to generate hydrodynamic lifting forces as we try to compress non-compressible liquid that is trapped between the lapped faces.

While two forces are acting to close the seal faces, three forces are acting to open the seal faces. If the closing forces are the greater forces the seal will generate heat that is often destructive, but always a waste of energy and pump efficiency. If the opening forces are the greater forces the seal will leak and that is never desirable.

A balanced seal, by definition, balances these opening and closing forces so that the seal will not get hot and it will not leak. Since the hydraulic closing forces were twice the opening forces (10 kg/cm^2 vs. 5 kg/cm^2)

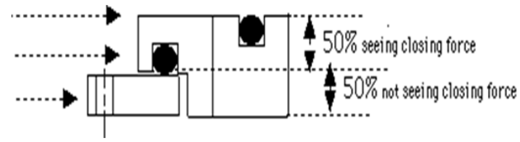

Fig. 4: Force balancing on a seal face

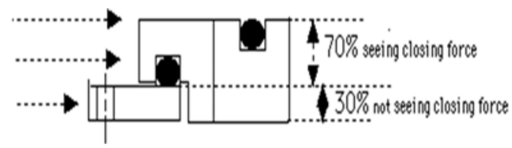

Fig. 5: Percentage balance of seal face

we install a sleeve inside the seal to reduce the closing area and thereby reduce the closing force. Figure 4 explains the phenomena:

The 10 Kg/cm^2 is now pushing on only 3 cm^2 because the inner sleeve is attached to the shaft and cannot move. The opening force remains the same. The numbers look like this:

- 10 Kg/cm^2 × 3 cm^2 = 30 kg. Closing
- 5 Kg/cm^2 × 6 cm^2 = 30 kg. Opening

We have eliminated the hydraulic forces from acting to open or close the seal faces. This leaves only the spring force to close the seal and the hydrodynamic and centrifugal forces to try to open the seal faces.

The final design solved the problem of balancing the other forces and linearity by overbalancing the closing hydraulic forces to compensate for:

- The nonlinear pressure drop across the seal faces.
- The hydrodynamic opening forces.
- Centrifugal opening force.

Figure 5 shows the final result:

Seventy percent (70%) of the seal face area is exposed to the hydraulic closing force instead of the 50% shown in the previous drawing. This is the standard 70-30 balance used by most mechanical seal companies. The seal designer can increase or decrease the percentage of over balance by changing the stepped sleeve diameter. This is done to:

- Decrease the face loading for low specific gravity fluids and higher speed shafts.
- Increase the face loading for higher viscosity liquids.

All that was required to hydraulically balance the seal was the simple low cost sleeve, but it is this additional cost that is keeping the original equipment manufacturer from adopting the design as his standard. The "bottom line" is that with an un-balanced seal design you either suffer the consequences of adding heat to the stuffing box

area, or having to provide cooling to remove the heat that is being generated by the un-balanced seal.
The closing force in Newton on the seal face is:

$$F_c = pa_c \qquad (1)$$

Where p = stuffing box pressure, N/m^2
a_c = hydraulic closing area, m^2

The pressure in Newton per square metre between the primary and mating rings is:

$$Pf = \frac{F_c}{a_o} = \frac{pa_c}{a_o} \qquad (2)$$

where a_o = hydraulic opening area (seal face area), m^2
The ratio of hydraulic closing area to seal face area is defined as seal balance b:

$$b = a_c/a_o \qquad (3)$$

The actual face pressure P_f in Newton per square meter is the sum of the hydraulic pressure P_h and the spring pressure P_{sp} designed into the mechanical seal:

$$P_f = P_h/P_{sp} \qquad (4)$$

where, $P_h = \Delta p[b\text{-}k]$, N/m^2 (5)

Δp = Pressure differential across seal face, N/m^2
b = Seal balance
K = Pressure gradient factor (Table 1)

The mechanical pressure for a seal is given by:

$$P_{sp} = \frac{F_{sp}}{a_o}\ N/m^2 \qquad (6)$$

where, F_{sp} = Seal spring load, N
a_o = Seal face area, m^2

Then the actual face pressure can be expressed as:

$$P_f = \Delta p[b\text{-}k] + P_{sp} \qquad (7)$$

The product of the two, pressure times velocity, is referred to as PV and is defined as the power N_f per unit area with a coefficient of friction of unity:

$$PV = \frac{N_f}{a_0} \qquad (8)$$

For seals, the equation for PV can be written as follows:

$$PV = P_f V_m = (\Delta p\,(b - k) + P_{sp})V_m \qquad (9)$$

where, V_m = velocity at the mean face diameter $d_{m,}$ m/s.

Therefore:

$$Q_S = C_1 N_f = C_1[(PVfa_o] \qquad (10)$$

Where Q_s = heat input from the seal, Watts:

C_1 = 1 for SI units
N_f = Power consumption
= $PVfa_o$

where f is the coefficient of friction.
The flow rate for cooling can be found by:

$$G_{pm} = \frac{Q_S}{C_2(sp.ht)(spgr)\Delta T} \qquad (11)$$

where, Q_s = Seal heat, W
C_2 = 1000 in SI units
Sp.ht = Specific heat of coolant, J/kg.K
Spgr = Specific gravity of coolant
ΔT = Temperature rise, K

When handling liquids at elevated temperature, the heat input from the process must be considered in the calculation of coolant flow. Then:

$$Q_{net} = Q_p + Q_S \qquad (12)$$

An estimate for seal leakage in cubic centimeters per hour can be made from the following equations (Lobanoff and Ross, 1992):

$$Q = \frac{C_3 h^3 (P_2 - P_1)}{\mu In(R_2 / R_1)} \qquad (13)$$

where, $C_3 = 1.88 \times 10^9$ in SI units:

h = Face gap, m
P_2 = Pressure at face ID, N/m^2
P_1 = Pressure at face OD, N/m^2
μ = Dynamic viscosity, $N.s/m^2$
R_2 = Outer face radius, m
R_1 = Inner face radius, m

SIMULATION RESULTS AND DISCUSSION

The results of simulation of the above design equations are presented and discussed below.

From Fig. 6, it can be seen that hydraulic pressure is zero at b = 50% i.e., when the seal is at balance position the hydraulic pressure is exerting null effect. Therefore, face pressure is relieved at the balance point. Resultant pressure is now mechanical pressure.

From Fig. 7, it is evident that face pressure is a very low value at b = 5.0. This agrees with literature as hydraulic pressure is relieved, leaving the spring pressure acting alone.

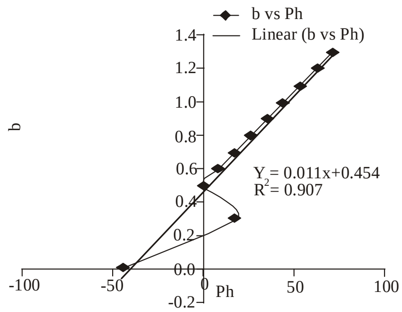

Fig. 6: Seal balance/hydraulic pressure (b vs Ph)

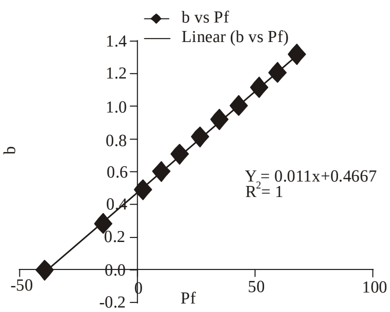

Fig. 7: Seal balance/seal face pressure (b vs Pf)

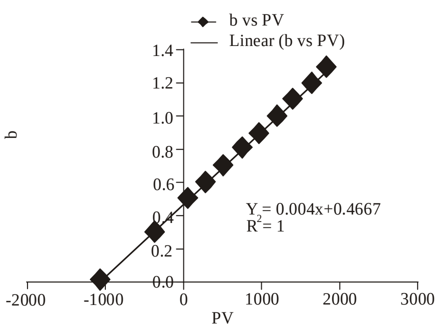

Fig. 8: Seal Balance/Power per unit area (b vs PV)

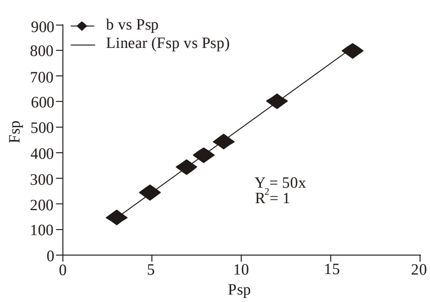

Fig. 9: Seal spring load/mechanical pressure (Fsp vs Psp)

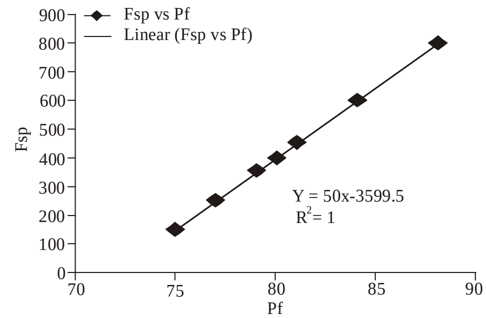

Fig. 10: Seal spring load/seal face pressure (Fsp vs Pf)

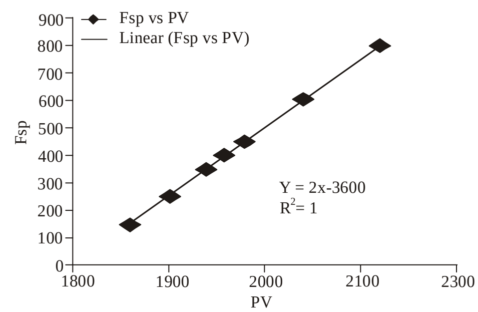

Fig. 11: Seal spring load/power per unit area (Fsp vs PV)

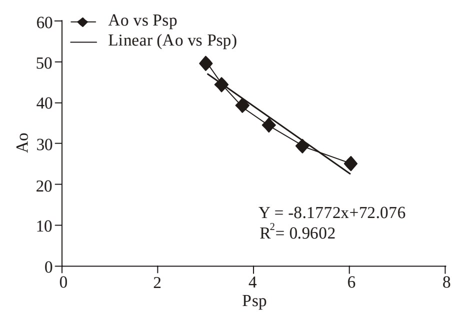

Fig. 12: Seal Face Area/Mechanical Pressure (Ao vs Psp)

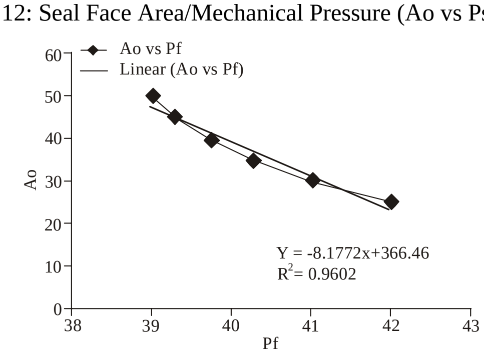

Fig. 13: Seal face area/seal face pressure (Ao vs Pf)

From Fig. 8, power per unit area is also lowest for a balanced seal at b = 5.0. For an unbalanced seal (b = 100% or slightly more), power per unit area requirement is very high.

From Fig. 9, as seal spring load increases the mechanical pressure for a seal or spring pressure increases. Therefore care must be taken to ensure that a balanced seal is used so as to relieve the face pressure on the seal. However, the face pressure similarly increases due to the increasing mechanical pressure (Fig. 10). Also power per unit area increases (Fig. 11).

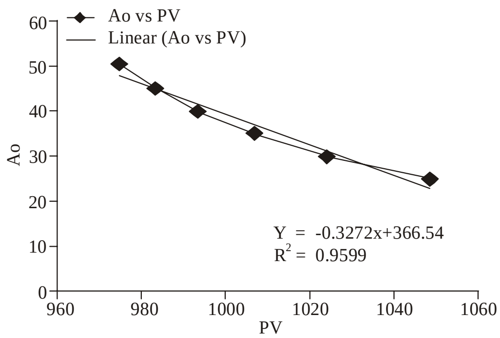

Fig. 14: Seal Face Area/Power per unit area (Ao vs PV)

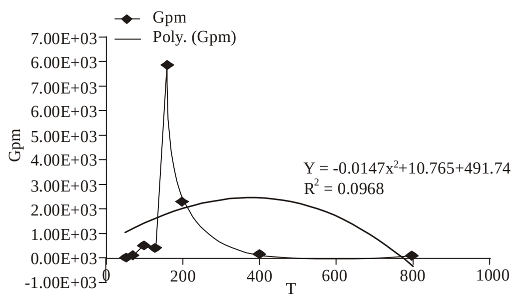

Fig. 15: Cooling flowrate/temperature rise (Gpm vs T)

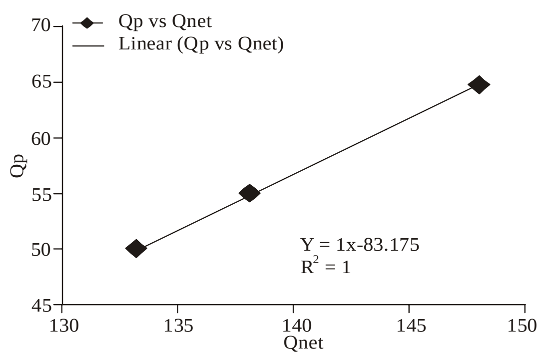

Fig. 16: Process heat load/coolant heat flow (Qp vs Qnet)

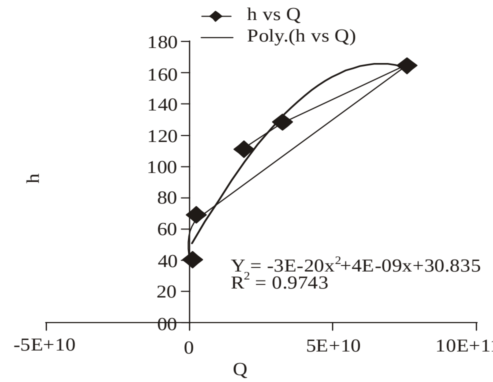

Fig. 17: Seal face gap/seal leakage (h vs Q)

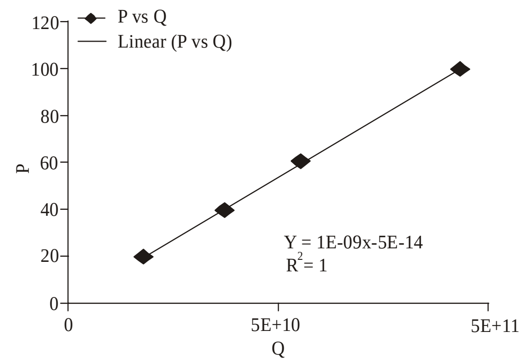

Fig. 18: Pressure difference/seal leakage (P vs Q)

From Fig. 12 and 13, as seal face area is reduced, the mechanical pressure on the seal and the face pressure increases. Increasing the seal face area is another way of relieving face pressure on the seal. Use of seals with wider surface area reduces failure rate of mechanical seals. In the same vein, the power per unit area increases with reduction in seal face area (Fig. 14).

Cooling rate is very poor at very high temperatures (Fig. 15). Care must be taken to use seals at moderate temperatures. Due to the poor cooling rate, heat removal

Table 1: Regression equation and R^2 values for plots

Plot title	Regression equation	Regression type	R^2 value	Error $(1 - R^2)$
b vsPh	Y = 0.0114x + 0.4547	Linear	0.9079	0.0921
b vs Pf	Y = 0.0111x + 0.4667	Linear	1	0
b vs PV	Y = 0.0004x + 0.4667	Linear	1	0
F_{sp}vsP_{sp}	Y = 50x	Linear	1	0
F_{sp}vs P_f	Y = 50x – 3599.5	Linear	1	0
F_{sp}vs PV	Y = 2x – 3600	Linear	1	0
$A_o$vsP_{sp}	Y = -8.1772x + 72.076	Linear	0.9602	0.0398
$A_o$vs P_f	Y = -8.1772x + 366.46	Linear	0.9602	0.0398
$A_o$vs PV	Y = -0.3272x + 366.54	Linear	0.9599	0.0401
T vsGpm	$Y = 100.4x^{0.0904}$	Power	0.0735	0.9265
G_{pm}vs T	$Y = -0.0147x^2 + 10.765x + 491.74$	Polynomial	0.0968	0.9032
$Q_p$vsQ_{net}	Y = x – 83.175	Linear	1	0
Q vs h	$Y = 5E+06x^2 – 4E+08x + 7E+09$	Polynomial	0.9977	0.0023
h vs Q	$Y = -3E – 20x^2 + 4E-09x + 30.835$	Polynomial	0.9743	0.0257
P vs Q	Y = 1E-09x – 5E-14	Linear	1	0
R_2/R_1vs Q	Y = 6E-11x – 0.0003	Linear	1	0

at high temperatures is usually lower than the heat generation rate hence the seal is likely to fail. Seal flushing is usually a way of reducing heat generation and increasing cooling at seal faces (fig. 16).

From Fig. 17, it can be seen that the gap between the seal faces is unique at 50 and 100 mm. These values represent a balanced and an unbalanced seal respectively. At 50 mm gap, the leak rate is very low whereas it is relatively higher at 100 mm. the implication is that a balanced seal is less likely to leak. Similarly as pressure change and radius ratio increases, as seen in Fig. 18, the seal leakage increases. Therefore, it is best to exert low pressure changes between the inner and outer diameters of the seal; and to ensure very little differences between the outer and inner radii of the seal face.

Below is a table of regression equation and R^2 values of the various plots.

From Table 1, it can be seen that the mathematical models that generated the above plots are reliable from the standpoint of the error in the regression. The only exception is Eq. (3.11), which has a very high degree of error. It is difficult to say that the model is completely unreliable as it could be subjected to other reliability tests; however from the standpoint of error analysis, it is safe to say that the model is not reliable.

CONCLUSION

From the above analysis, a seal with balance value of 0.5, increased surface area, provided with a flush system to enhance cooling and seal face gap of 50 mm or less for minimal or zero leakage respectively is recommended. According to API 610 and 682, mechanical seals suitable for high speed centrifugal gas compressors must be designed to allow for safe sealing of the gas and reduce/eliminate contamination of the sealed gas. Similarly, leakages should be minimized by using balanced cartridge seals with the needed characteristics. Various types and their characteristics have been highlighted. The selection now depends on intended use and desired results.

Since failure is a common phenomenon in seal usage, care must be taken in designing the seal for flushing and cooling. Similarly, temperature requirements should be carefully studied before seal selection.

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