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Improving Centrifugal Pumps Seals to Prevent Leakage during Operation

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Abstract

C-Frame centrifugal pumps are fitted with a Stuffing Box Seal and is designed to allow only a small amount of drops per minute. It is used to cool down the packing gland when the pump shaft is rotating during pumping operations. It has been a normal problem that there are severe corrosion of a metal casing of a centrifugal pump due to excessive slurry water leaking through a pump seal.

The leakage rate can be adjusted by tightening and loosening the packing gland. Excessive leakage failure of a pump seals which must be thoroughly investigated with the force diagram that relates to pump seal operating parameters that impacts the pump seal during operation is the focus on this study. Solid-works, CFD and Mathcad were tools used to design and simulate flow discharge rate versus the pressure of seal. The three simulation results revealed the optimal design that will be more efficient during pump operation are revealed in the study. For accurate simulation results it was shown that, the reactive force of the seal must be taken into consideration throughout the three-simulation test.

Keywords: Discharge leakage rate, pressure, compression, flow velocity, design limits efficiency.

1. INTRODUCTION

The research paper is about improving a shaft seal of a centrifugal pump that is used to pump a slurry in a platinum mine. Currently, the platinum plant has a problem of slurry leaking from the shaft of a pump during pumping operations due to the poor design of a static part (seal) [1-4]. It has been discovered that the seal has not been effectively doing its intended purpose of preventing a leakage and the problem is caused by the poor design [1-6]. Therefore, a need to correctly utilize knowledge gained in studies to model a better design of seal to prevent excessive slurry leakage from 1200 drops per minute (5 lit/minute) to approximately 190 drops per minute (0.5 lit/minute) to comply to the requirements of the occupational health and safety act [1-9].

In almost all centrifugal pumps, the rotating shaft that drives the impeller penetrates the pressure boundary of the pump casing [1-5]. The pump must be designed properly to control the amount of liquid that leaks along the shaft at the point that the shaft penetrates the pump casing [1-5]. There are many different methods of sealing the shaft penetration of the pump casing [1-3]. Factors considered when choosing a method include the pressure and temperature of the fluid being pumped, the size of the pump, and the chemical and physical characteristics of the fluid being pumped [1-4].

One of the simplest types of shaft seals is the stuffing box. The stuffing box is a cylindrical space in the pump casing surrounding the shaft [1-5]. Rings of packing material are placed in this space.

Packing is material in the form of rings or strands that are placed in the stuffing box to form a seal to control the rate of leakage along the shaft. The packing rings are held in place by a gland. The gland is, in turn, held in place by studs with adjusting nuts. As the adjusting nuts are tightened, they move the gland in and compress the packing. This axial compression causes the packing to expand radially, forming a tight seal between the rotating shaft and the inside wall of the stuffing box.

The high-speed rotation of the shaft generates a significant amount of heat as it rubs against the packing rings [1-6]. If no lubrication and cooling is provided to the packing, the temperature of the packing increases to the point where damage occurs to the packing, the pump shaft, and possibly nearby pump bearings [1-4]. Stuffing boxes are normally designed to allow a small amount of controlled leakage along the shaft to provide lubrication and cooling to the packing. The leakage rate can be adjusted by tightening and loosening the packing gland as shown in Fig.1.

Figure 1. Centrifugal pump seal under investigation.

The type of material for the packing in the stuffing box is a graphite fibre because of its favorable properties, such as name; it is not abrasive and has improved lubricity. This type of material is used to seal rotating shafts at higher speeds with less leakage. In this paper, the parameters that impacts the performance of the seal during operation are investigated for optimal operation of the shaft seal during operation.

4. METHODOLOGY

Excessive leakage failure of a pump seals and which is thoroughly investigated with the force diagram that relates to pump seal. For shaft leakages to be prevented, it is important to understand the forces that affects the shaft seal during operation.

4.1 Forces acting on a seal (Diagram)

Hydrodynamic seals and squeeze-film dampers constitute a category of devices that involve the flow in an annulus between two cylinders; the inner cylinder is generally the shaft (radius, R) which is rotating at a frequency, Ω, and may also be whirling with an amplitude or eccentricity, ε, and a frequency, ω. The outer cylinder is generally static and fixed to the support structure normally referred to as the seal (contact between the support and the shaft). The mean clearance (width of the annulus) will be denoted by δ and the axial length by L.

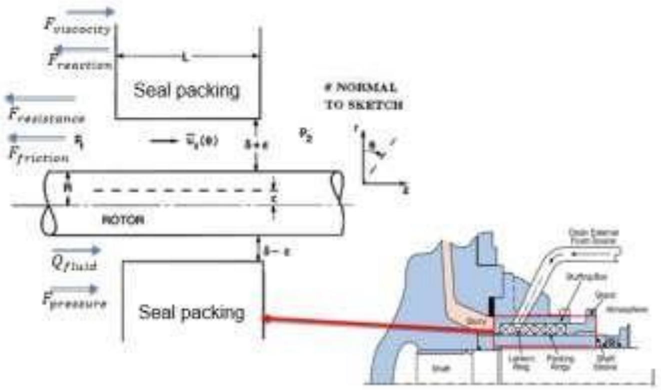

Figure 2. The seal and parameters that impacts the seal during operation.

From first principles, the forces that acts on the shaft seal are revealed as,

F_{resist} = Force of resistance given in N

$F_{friction}$ = Force due to friction between seal and shaft contact during operation given in N

$F_{pressure}$ = Force due to pressure given in N

$F_{reaction} = Force\ of\ reaction\ due\ to\ bearing\ in\ N$

$F_{viscocity}$ =Force due to moving fluid in the system

Q_{fluid} = Leakage discharge rate given in m^3/s

P_1 = Pressure behind a seal given in Pa

P_2 = Pressure outside a seal given in Pa

Total forces acting on the shaft seal are given as

$$(-Fresistance + Fpressure - Ffriction + Fviscocity + Freaction) = Ftotal\ in\ sysem \quad [1]$$

F_{total} is a function of force due to resistance, force due to friction between seal surface contact with shaft during operation, and the force due to fluid flowing into the system. The force of resistance acts against the surface of the shaft and the fibre material of the packing during the shaft rotation, this force is minimized by the lubricity of the packing, process water, and the smoothness of the shaft. Force of friction is due to the movement of water against the surface of the seal gland, the hydraulic force acts against the seal due to the pressure applied by water against the seal during pumping operation, water will always move from the high potential to lowest potential. (due to the difference in pressures; inside the pump, the hydraulic pressure is greater than the pressure measured in between the packing's as seen in figure 3.5), this is the same for the force due to viscosity. The reactive force (force of reaction) is the force applied by the seal gland to the forces applied during the hydraulic movement of water during pumping operation.

Each force is defined as;

$F_{resistance} = m \times g \times \mu$

$Fpressure = Pfluid \times ASeal$

$Ffriction = m \times g \times f$

$$F_{viscocity} = \delta \times A \times \frac{dv}{dx}$$

$$-(m \times g \times \mu) + [P_{fluid} \times A_{seal}] - (m \times g \times f) + (\delta \times A \times \frac{dv}{dx}) - F_{reaction} = Ftotal\ in\ system$$

In both hydrodynamic seals, the basic fluid motion is caused by the rotation of the shaft. In a seal, there is an additional axial flow due to the imposed axial pressure difference. In a squeeze film damper, there is no rotational motion, but forces are generated by the whirl motion of the shaft.

Bernoulli Equation: Effects of Pressure & Resistance

The Reynolds number is an important parameter in these flows, and it is useful to evaluate three different Reynolds numbers based on the rotational velocity, on the mean axial velocity, V (given by V=Q/ 2π Rδ where Q is the volumetric axial flow rate), and on the velocity associated with the whirl motion.

The Bernoulli Effect of inertia effect can be simply explained as follows. When an eccentricity is introduced, the fluid velocities will be increased over that part of the shaft circumference where the clearance has been reduced. At

Reynolds numbers much larger than unity, the Bernoulli equation is applicable, and higher velocities imply lower pressure. Therefore, the pressure in the fluid is decreased where the clearance is small and, consequently, there will be a net force on the shaft in the direction of the displacement. Pressure is expected to decrease as the number of packing's increases as seen on the graph below;

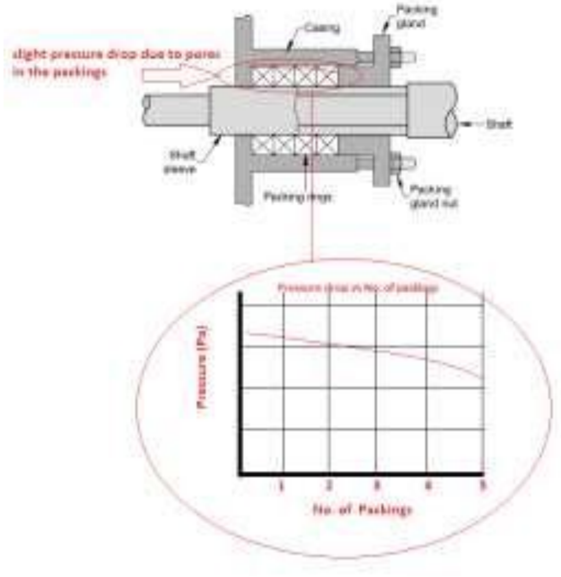

Figure 3. The dynamic forces on the seal during operation.

From energy equation the change of pressure and velocity through the system can be established. The change in pressure and velocity during the system operation is given as,

$$P_1 - P_2 = \rho g \left(\frac{V_2^2}{2g} - \frac{V_1^2}{2g}\right)(gZ_2 - gZ_1)$$

The discharge leakage rate Q, rewriting the Bernoulli's' equation such that the fluid velocity becomes the subject and the discharge rate Q= VA then the expression becomes

$Q_{fluid} = 2\pi R \frac{\sqrt{(P_2-P_1)}\, C^3}{f A_p}$ (E. Storteig, 1999)[8]

As seen on the expression above, the discharge rate is a function of pressure, gravitational acceleration, discharge area, and the fluid specific gravity when omitting the elevation difference.

Below is a relationship between the Pressure difference and the fluid leakage on the seal.

[6]

Where

$P1$ & $P2$ = $fluid\ pressure\ inside\ the\ pump\ casing\ and$

$atmospheric\ pressure\ outside\ seal$

v_1 = $fluid\ velocity\ inside\ the\ pump$

g = $gravitational\ force\ due\ to\ gravity$

Therefore the fluid pressure difference will be re-written as;

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + gZ_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + gZ_2$$

$$P_1 - P_2 = \rho g \left(\frac{V_2^2}{2g} - \frac{V_1^2}{2g}\right)(gZ_2 - gZ_1)$$

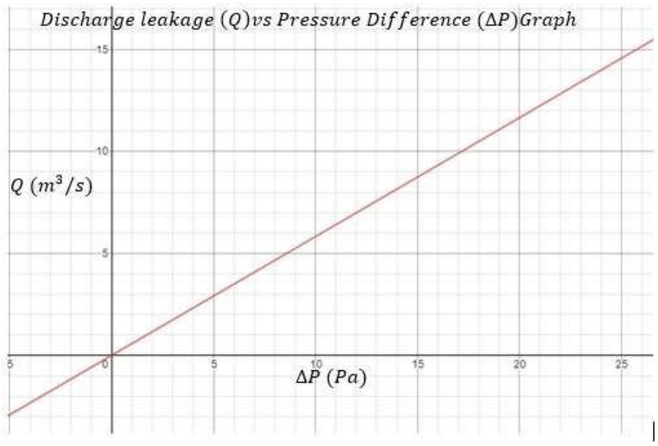

Figure 4. Change in pressure and system discharge during operation.

Face specific pressure has an important effect on sealing situation, face specific pressure will make the dry friction surface heating, increase the wear, and power consumption increased at the same time, and end face specific pressure is too small, easy to leak and pollution, so want to have good sealing performance and enough life, to ensure the end face specific pressure in a suitable range, so the working efficiency of the pump will also increase. Produced by elastic elements on the face of pressure, and sealing structure, medium type, and pressure, to ensure that the sealing during the start and stop, compensation face wear and tear, stuffing should generate a certain amount of pressure.

According to the working principle of the centrifugal pump, the pressure of the liquid decreases gradually from the suction pipe to the suction inlet of the pump. When the pressure of a certain point in the pump is low and the vapor pressure of the liquid is saturated, some of the liquid will vaporize, and the resulting bubble is brought into the impeller by the liquid flow, and the pressure in the impeller is broken. Due to the instantaneous vacuum generated by the condensation point, the surrounding liquid is subjected to high-velocity impact, resulting in violent water strikes, which is called the cavitation phenomenon around the faces of the seal as seen on the diagram. Adverse consequences caused by cavitation are Increased noise and vibration, Performance degradation and the service life of the packing seal decreases.

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5. RESULTS AND DISCUSSION

Solid Works was used to make a technical design drawing of a stuffing box, with a sectioned view to show the internal parts of a stuffing box. It was also used to demonstrate the exploded view of the assembly and how the parts of the stuffing box will be assembled on the final assembly drawing. The simulation was made using finite element analysis and CFD where relationships of parameters were studied and demonstrated using tables and graphs where discussions, conclusions, and recommendations were drawn.

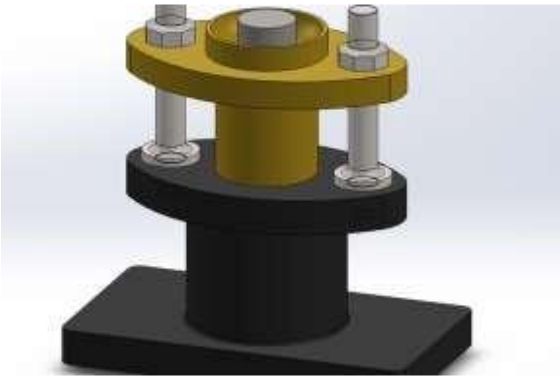

Figure 5. Stuffing box seal of the system

The image above is a 3D model drawing of a stuffing box seal with each part assembled to form one complete unit. The simulation of the seal will be conducted to analyze its performance to meet the design objective which is to reduce leakage. This part focuses on the simulation of the pump seal to observe the parameters such as porosity by some parameters were taken into consideration when designing and modeling the shaft seal (stuffing box) which will be analyzed using the following engineering tools such as; CFD and Finite Element Analysis.

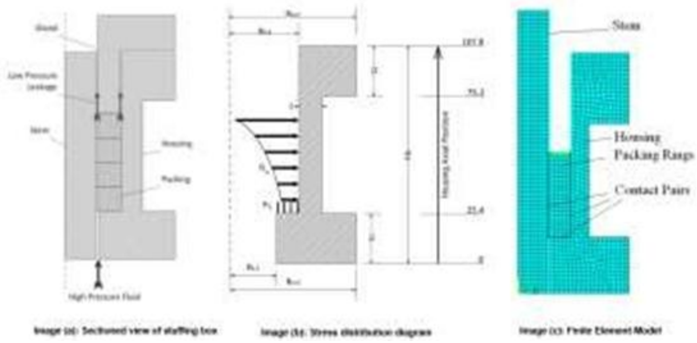

Figure 6. (a) Section of seal under high pressure, (b) stress distribution due to external forces during operation and (c) stuffing box packing using a Finite Element Analysis.

The first diagram on the left (image (a)) shows a sectioned view of a seal with a high-pressure fluid flowing through the packing and the second diagram (image (b)) shows the stress distribution diagram of the packing to the packing housing and the diagram on the right (image (c)), shows the model of a stuffing box packing using a Finite Element Analysis. The mechanism of sealing off the packed stuffing box is ensured by adequate compression of the packing material which reduces the pores and fills the leak paths at the contact interface to prevent fluid from escaping to the outer boundary. Therefore, it is expected that, when increasing compression of the gland of the stuffing box, the pores becomes small, thus allow a minimum amount of liquid to flow through the packing's, there are 3 sets of simulations done as seen on the graph of Flow rate vs Pressure Graph.

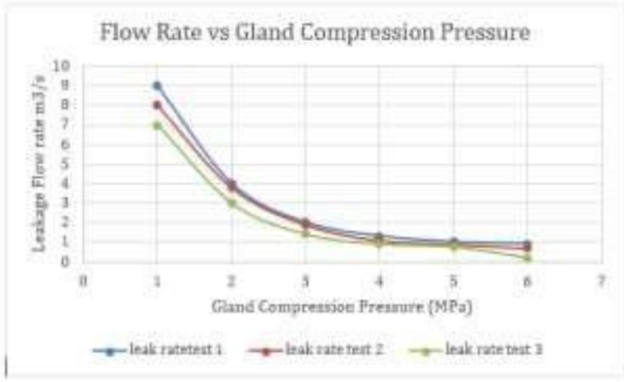

Figure 7. The relationship between Gland compression pressure and leakage flow rate during operation

The analytical development of the complex behavior of a packed stuffing box is limited to the distribution of the stresses in the packing rings. The simple model used is based on the ratio of the radial contact pressure to the axial gland stress known as the lateral pressure coefficient.

As discussed, the discharge leakage rate Q, when rewriting the Bernoulli's' equation such that the fluid velocity becomes the subject and the discharge rate Q= VA then the expression becomes;

$$Q_{fluid} = 2\pi R \frac{\sqrt{(P_2 - P_1)}\, C^3}{f A_p}$$

It is clear that for a given pressure difference across the seal, the seal clearance is the dominating factor for the leakage $Q_{leakage}$ since halving the clearance C has the same effect as increasing the length of the seal L or the friction factor f by a factor of eight.

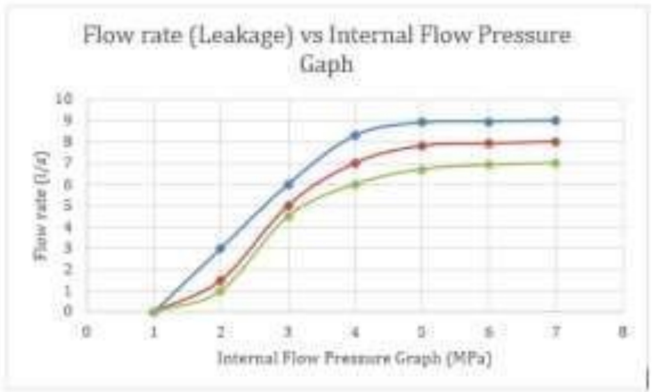

Figure 8. The relationship between internal flow pressure and flow rate during seal operation.

As seen on the expression, the discharge rate is a function of pressure, gravitational acceleration, discharge area, and the fluid specific gravity when omitting the elevation difference. It is therefore seen that the discharge rate will always be proportional to the internal pressure, as the pressure increased, the flow of water increased, and this is true for the same three simulations conducted as seen on the graph above. This can also be seen on the velocity vs pressure simulation of a centrifugal pump.

6 CONCLUSION

Since the main aim of the current paper was to prevent leakages in a centrifugal pump seal during operation, the parameters that impact the pump seal during operation were studied for optimal operation. In the current design, the correct compression pressure to best match the leakage rate of water from the pores had to be further investigated for optimal system performance. In the this paper, the correct design and simulation of the stuffing box seal, the correct best matching pressure and leakage rate was met, this is seen in the discharge rate versus compression pressure graph that, at an applied pressure of 7MPa, the discharge leakage rate is at 0.009m3/s. This value of the discharge rate obtained is the best thus far compared to the initial discharge rate of 0.039m3/s at a pressure of 12MPa. Therefore, through the abovementioned factors, the objective of the design project was achieved as there is an improvement in terms of discharge rate (being lesser than the previous one of the existing designs of the stuffing box seal).

7. RECOMMENDATION

Following the results obtained from simulation and having noted the following as discussed in it is hereby recommended without reservation that all forces be taken into consideration during simulation for accurate results.

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