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STUDY OF FEASIBILITY IN REPLACING STATIC WEAR RING BY FLOATING WEAR RING IN A CENTRIFUGAL PUMP

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Abstract. *Using floating wear ring instead of static ring in overhung centrifugal pump has some advantages. The floating wear ring reduces the erosion and increases the efficiency of the pump since it allows minimizing the radial clearance. In addition, because of the smaller clearance the stiffness and damping are higher, so that the unbalance response becomes lower. On other hand, one should pay attention at the cross coupling stiffness, because it increases as the clearance decreases and the correct design of the floating ring, otherwise, it may lock up and cause rubbing.*

Keywords: *floating wear ring, vibration, centrifugal pump*

1. INTRODUCTION

The major kind of pump used in refineries is overhung centrifugal pump. And some of them, which were manufactured before API 610-8°ed. (American Petroleum Institute – Centrifugal Pumps for Petroleum, Petrochemical and Natural Gas Industries Standard) are still being used in Brazil. These equipments have flexible rotors since the ratio L^4/D^2 (L=bearing span; D=largest shaft diameter at impeller) is high and the packing, that offers stiffness, was replaced by mechanical seal in order to get more safety. As result, the dynamic response of these pumps is very sensitive to any variation of the wear ring clearance. Vibration amplitudes have been identified of more than 20mm/s-rms at the bearing when the diametral clearance between static and rotational wear ring increases above 0.9mm. The high level of this vibration is because the frequency of the excited force is normally coinciding with the damped natural frequency of the pump which is induced by rubbing or fluid – see spectrum in figure 1.

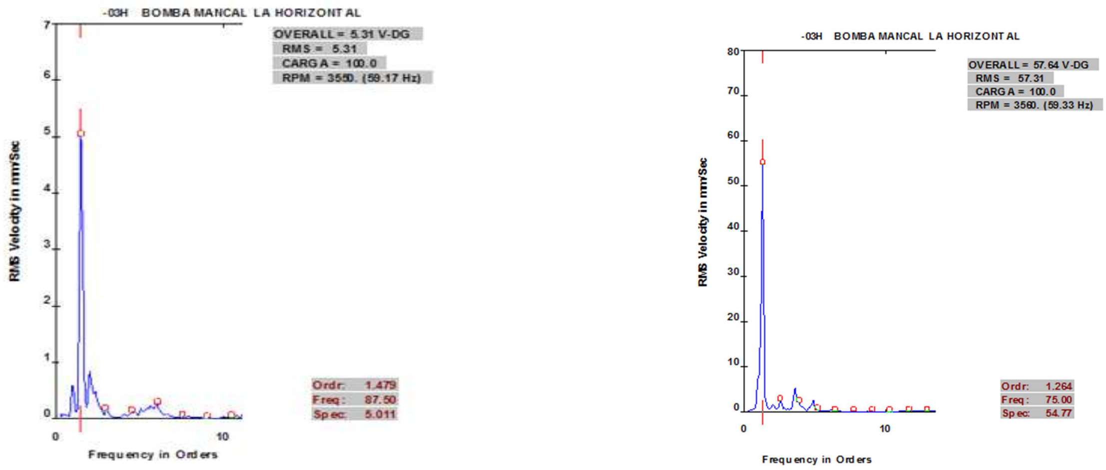

Fig.1. Natural frequency being excited when wear ring clearance becomes high

The main causes of the weariness of these components are the erosion between them and surfaces in contact. Beyond the vibration, the worn out condition of the wear ring reduces the efficiency of the pump since the flow through these annular seals increases.

So, in order to solve these two concerns (vibration and efficiency) this study proposes to investigate the feasibility of replacing the static ring by the floating wear ring in centrifugal pumps.

Floating wear ring may reduce the erosion and increases the efficiency of the pump since it allows minimizing the radial clearance without rubbing. In addition, because of the smaller clearance the stiffness and damping are higher, so the unbalance response becomes smaller.

The disadvantage of using floating wear ring is that each one has to be designed for a specific range operation condition and the cross coupling stiffness becomes higher as the radial clearance decreases.

2. FLOATING WEAR RING

The floating ring seal has been used in turbopumps of liquid rocket engine (LRE) systems and few multistage pumps with flexible rotors to increase the stiffness since the radial clearance could be minimized without or with minimal rubbing phenomenon.

The concept of floating wear ring is the same used in oil seal in compressors as depicted in figure 2. The ring is just able to move radially because its angular motion should be restrained by a rotational pin. Net differential pressure in axial direction times the contact face creates a normal force (N). And this normal force times the friction coefficient (μ) results the friction force that lock up the ring. When the net fluid force (F_L), which is generated like the case of fluid film bearing, plus the ring's weight (W) is higher than friction force, the ring floats radially to smaller eccentric position until the equilibrium condition is established ($N.\mu = F_L+W$). Therefore, the eccentricity ratio depends on the operating conditions, surface friction coefficient between ring and casing, ring geometric parameters and the influence of the O'ring.

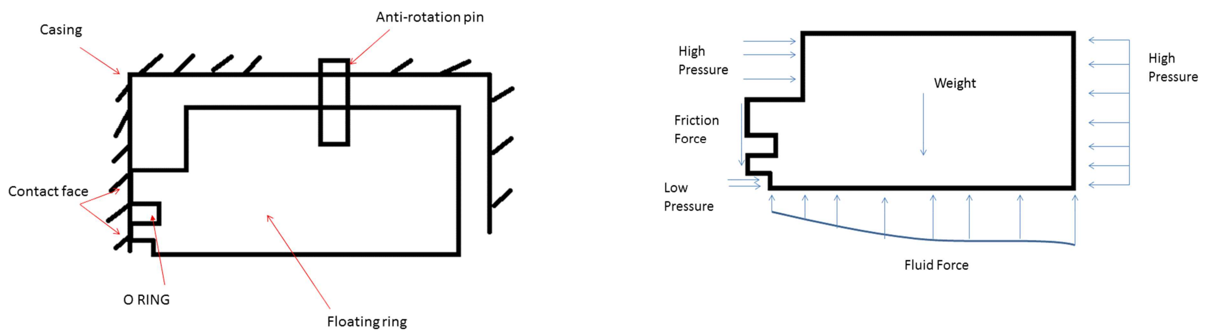

Fig. 2. Left: Floating ring concept. Right: Floating ring's force balance

While the floating ring is locked-up it acts like the fixed wear ring. Then, the analysis method of the eccentric annular seal is used to predict the static and dynamic characteristics of the floating ring seal. Although there are four degrees of freedom (DOF), the dynamics of seal will be represented by just 2 DOF, translational motions along perpendicular axis. Therefore, for small amplitude shaft translational motion (ε) the seal forces for lateral motions (X, Y) could be depicted by

$$\begin{bmatrix} F_x \\ F_y \end{bmatrix} = -\begin{bmatrix} K_{XX} & K_{XY} \\ K_{YY} & K_{YY} \end{bmatrix}\begin{Bmatrix} X \\ Y \end{Bmatrix} - \begin{bmatrix} C_{XX} & C_{XY} \\ C_{YX} & C_{YY} \end{bmatrix}\begin{Bmatrix} \dot{X} \\ \dot{Y} \end{Bmatrix} - \begin{bmatrix} M_{XX} & M_{XY} \\ M_{YX} & M_{YY} \end{bmatrix}\begin{Bmatrix} \ddot{X} \\ \ddot{Y} \end{Bmatrix} \tag{1}$$

where [K], [C] and [M] are the matrices of stiffness, damping and inertia force coefficients respectively.

Thus, the coupled system – rotor/fluid- is formulated by

$$\left[[M_r] + [M_f(\varepsilon)]\right]\begin{Bmatrix} \ddot{X} \\ \ddot{Y} \end{Bmatrix} + \left[[C_r] + \Omega[G].[C_f(\Omega)]\right]\begin{bmatrix} \dot{X} \\ \dot{Y} \end{bmatrix} + \left[[K_r] + [K_f(\varepsilon,\Omega)]\right]\begin{Bmatrix} X \\ Y \end{Bmatrix} = [F] \tag{2}$$

where **Ω** [G] is the shaft speed and gyroscopic effect and the subscript "*r*" and "*f*" refer to rotor and fluid.

3. METODOLOGY

In order to assess the feasibility of using floating wear ring, also known as annular seal ring, it was chosen a single stage overhung pump although it may be applicable also in multistage pumps. This pump is usually assembled with minimum diametral wear ring clearance of 0.7mm, otherwise, contact occurs between static and rotational seal ring, as tested on site. After running the pump less than two years, it is possible to observe a high radial vibration in supersynchronous whirl frequency, around 87Hz, caused by high diametral clearance of the wear ring, between 0.9 to 1.4mm. So, to avoid the premature weariness of annular seal the use a floating ring is going on.

The main basis for the preparation of this work was the hypothesis that the use of floating ring over the static ring allows its installation with a 50% lower clearance than specified by the manufacturer, with no risk of rubbing and/or locking as would happen with fixed rings.

In order to assess the rotordynamic behavior of replacing the fixed ring by a floating ring, the following procedure was carried out:

a. The code Seal2 from ROMAC was used to get the seal dynamic coefficient. The seal is modeled by bulk flow theory and to solve the Navier-Stoke equations the first order upwind finite difference method is used.
b. The shaft was modeled by Timoshenko beam elements using XLTRC2 code. The effects of rotatory inertia, gyroscopics and shaft weight were included. Shear stress was not considered since just the first and second vibration modes are point of interest.
c. The matrix of inertia of fluid was neglected since this is not relevant because the goal is to compare the difference of the results.
d. The entrance loss in annular seal is considered equal to 0.1.
e. The inlet swirl ratio considered was 0.5.
f. Calculation of the direct and cross coupling stiffness and damping of annular seal varying the radial clearance, from 0.20mm to 0.6mm and eccentricity from zero to 0.9. This procedure was carried out considering smooth seal and seal with groove.
g. The stiffness of the ball bearing considered was 1.75×10^8N/m and its damping was neglected.
h. Plotting the unbalance response with added weight in the impeller. The unbalance weight was 546.48g.mm. The predicted vibration amplitude in axial position of faces of the mechanical seal was determined.
i. Calculation of the flow through annular seal varying the wear ring radial clearance (c) from 0.20mm to 0.60mm.
j. Calculation of the resultant fluid force in annular seal. This information is important to design the floating ring.

4. SIMULATION DATA

The simulation considered a single stage centrifugal pump with the following characteristics:

- Discharge Pressure: 2227kPa
- Suction Pressure: 298.8kPa
- Flow rate: 203m³/h
- Fluid: light cicle oil
- Tempeturate: 217°C
- Dynamic viscosity: 0.000233N.s/m²
- Density: 775kg/m³
- Rated shaft power: 196kW
- Diameter of rotating annular seal: 184mm
- Dynamic coefficient of the bushing:
  - $Kxx=4.65 \times 10^5$N/m
  - $Kxy=5.1 \times 10^5$N/m
  - $Kyx=-5.1 \times 10^5$N/m
  - $Kyy=4.67 \times 10^5$N/m
  - $Cxx=4.75 \times 10^3$N.s/m
  - $Cxy=-4.25 \times 10^2$N.s/m
  - $Cyx=-4.23 \times 10^2$N.s/m
  - $Cyy=2.76 \times 10^3$N.s/m
- Length of annular seal: 32mm
- Shaft speed: 3550rpm
- Inertia Polar of Coupling: 0.006278kg-m²
- Transversal Inertia of Coupling: 0.005063kg-m²
- Mass of the Coupling 3.29kg
- Polar Inertia of rotor: 0.537572kg-m²
- Transversal Inertia of rotor: 0.268786kg/m²
- Mass of the rotor dry: 28.12kg
- Bearing stiffness ($K_{xx}=K_{yy}$): 175.11×10^6N/m

It was simulated the dynamic coefficient of annular seal and force response of the pump for 16 different wear ring parameters as displayed in table 1. These cases were chosen as following:

- Smooth and fixed seal with maximum radial clearance (25% higher than set by the API 610), operating in concentric position and maximum eccentricity.
- Smooth and fixed seal with minimum radial clearance achieved during assembly of the pump without occurrence of locking, operating in concentric and maximum eccentricity position.
- Fixed seal with groove at maximum radial clearance (25% higher than set by the API 610), operating in concentric and maximum eccentricity position.
- Smooth and fixed seal at minimum radial clearance achieved during assembly of the pump without occurrence of locking, operating in concentric and maximum eccentricity position.
- Smooth floating seal at 70% of the minimum radial clearance required by API 610, operating in concentric and maximum eccentricity position.
- Smooth floating seal at 57% of the minimum radial clearance required by API 610, operating in concentric and maximum eccentricity position.

Table 1. Annular Seal Parameters

CASE	Type of ring	Eccentricity	Radial Clearance (mm)	Land Length (mm)	Groove length (mm)	Groove Depth (mm)	Number of Grooves (0=plain seal)
Case 1	fixed	0	0.6	0.83	2.28	1	10
Case 2	fixed	0.9	0.6	0.83	2.28	1	10
Case 3	fixed	0	0.6	32	-	-	0
Case 4	fixed	0.9	0.6	32	-	-	0
Case 5	fixed	0	0.35	0.83	2.28	1	10
Case 6	fixed	0.9	0.35	0.83	2.28	1	10
Case 7	fixed	0	0.35	32	-	-	0
Case 8	fixed	0.9	0.35	32	-	-	0
Case 9	floating	0	0.25	0.83	2.28	1	10
Case 10	floating	0.9	0.25	0.83	2.28	1	10
Case 11	floating	0	0.25	32	-	-	0
Case 12	floating	0.9	0.25	32	-	-	0
Case 13	floating	0	0.20	0.83	2.28	1	10
Case 14	floating	0.9	0.20	0.83	2.28	1	10
Case 15	floating	0	0.20	32	-	-	0
Case 16	floating	0.9	0.20	32	-	-	0

Figures 3 and 4 below show the photo of the centrifugal pump and its model:

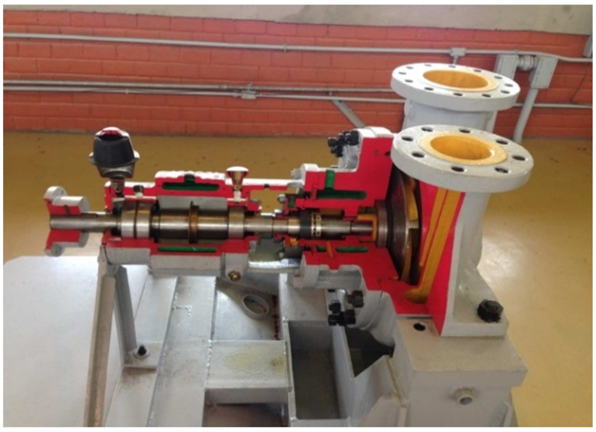

Figure 3. Photo of centrifugal pump modeled

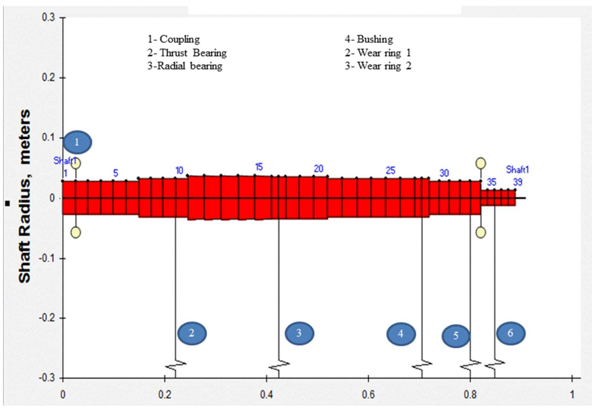

Fig. 4. Model of centrifugal pump

5. RESULTS

The plots in figure 5 to 8 depict how the behaviors of the dynamic coefficients are when the eccentricity, clearance and type of wear ring are varied. The subscribed “g” means the type of seal with grooves.

The variation of the dynamic coefficients with eccentricity is significant only when it is equal to or greater than 0.8. So, rotor response to the external forces would not be significantly influenced by eccentric position of rotating ring.

Unlike hydrodynamic bearings, the value of the direct stiffness is high even in a concentric position. This occurs because of the axial flow rate, that induces this stiffness even without rotation, creating a radial reaction force - a phenomenon known as the Lomakin effect.

In contrast to the value of the lower cross coupling stiffness, the seal with groove has a lower direct damping than smooth seal. Therefore, its application should be carefully evaluated since the instability problem related to cross-coupling is solved but, it can cause a reduction in damped natural frequency that, in turn, would approach of the machine's operating speed causing high vibration amplitude.

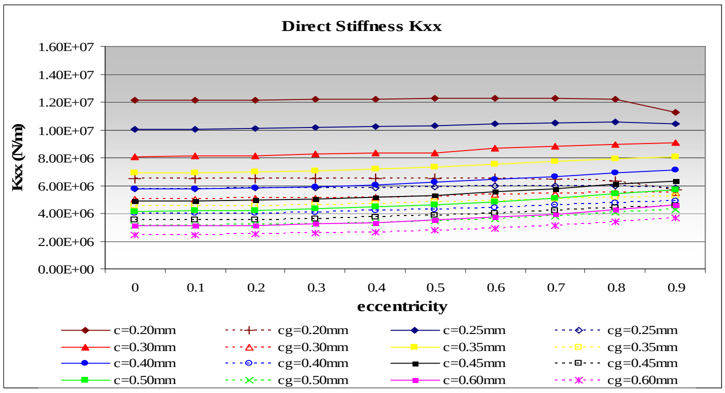

Fig. 5. Direct Stiffness K_{xx}

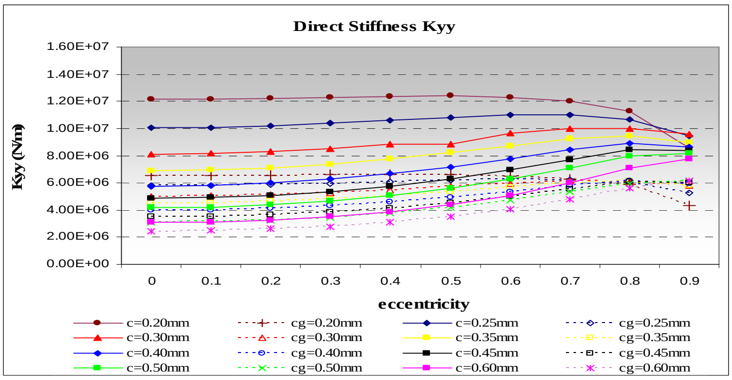

Fig. 6. Direct Stiffness K_{yy}

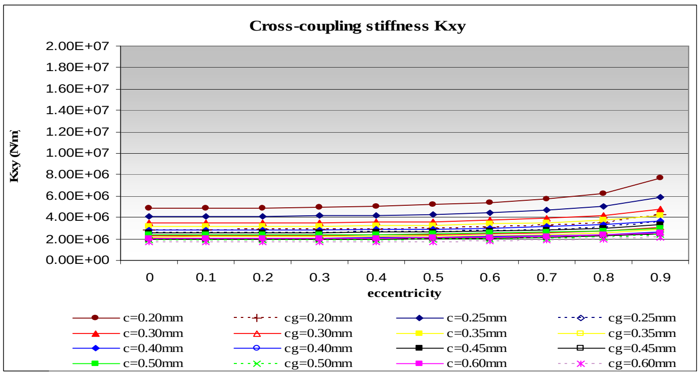

Fig. 7. Direct Stiffness K_{xy}

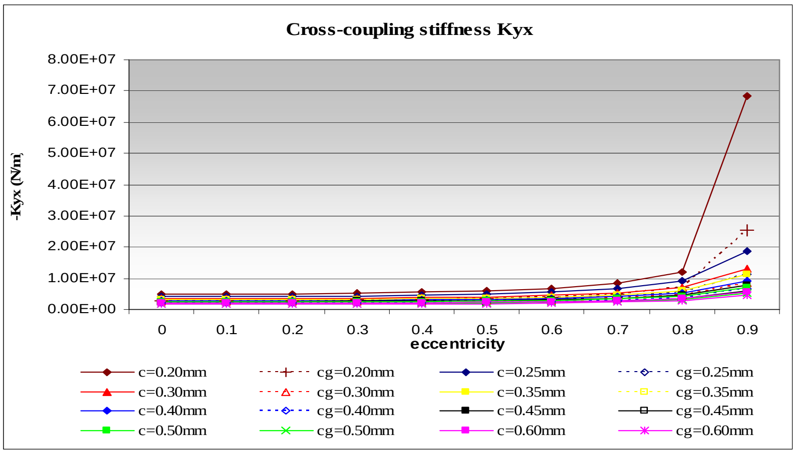

Fig. 8. Direct Stiffness K_{yx}

Figure 09 shows the damping ratio versus critical speed calculated for each case. Smooth seals have a higher damping ratio than those with groove. Although the first critical speed of the case 12 is 400% higher than operating speed, it has relatively low damping value so, the damping natural frequency could be excited in case of defects in rolling element bearings.

Depending on the wear ring clearance, this model of pump presents a high vibration at frequencies between 4260 rpm and 5325 rpm. Therefore, the probable cause of this vibration is the excitement of its damped natural frequency induced by contact between the rings or by fluid. However, in both cases it is expected that the floating seal minimizes or eliminates this problem, since the contact between wear rings would be avoided and the level of turbulence of the fluid would decrease.

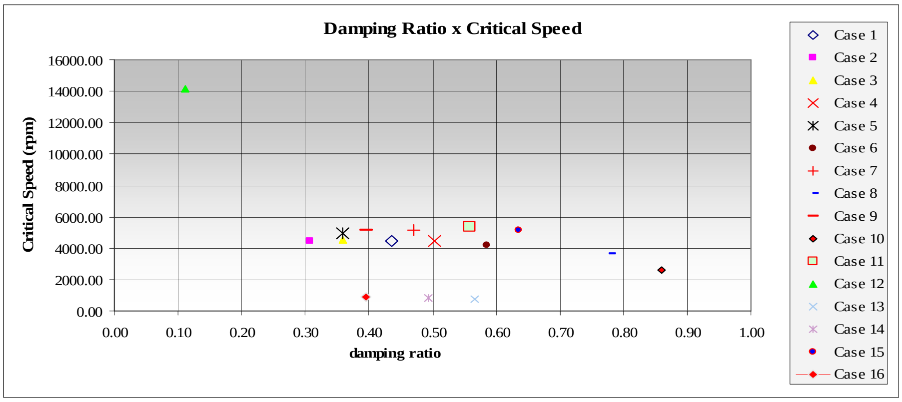

Fig. 09. Damping ratio versus critical speed

As the shaft deflection in the mechanical seal mating ring position is important for the reliability of the seal, it was plotted the rotor displacement in this region in figure 10. As displayed, the floating ring may reduce the vibration more than 300%.

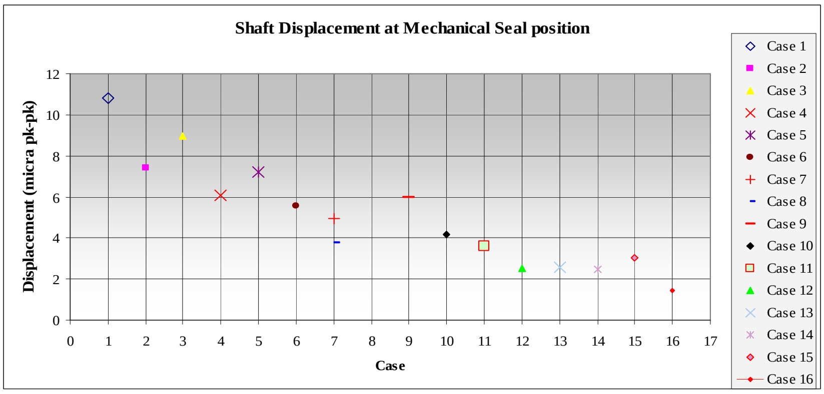

Fig. 10. Shaft Displacement at Mechanical Seal Mating Ring position

Regarding the recirculation flow through annular seals, floating ring with radial clearance of 0.25mm could reduce recirculation by 40% compared to a fixed ring with radial clearance of 0.35mm – see figure 11 . When worn, fixed wear ring with radial clearance of 0.6mm leads an increase in the recirculation flow at 100%, resulting in an excessive increase in the current of the driver motor that, in turn, may cause the trip of pumping system.

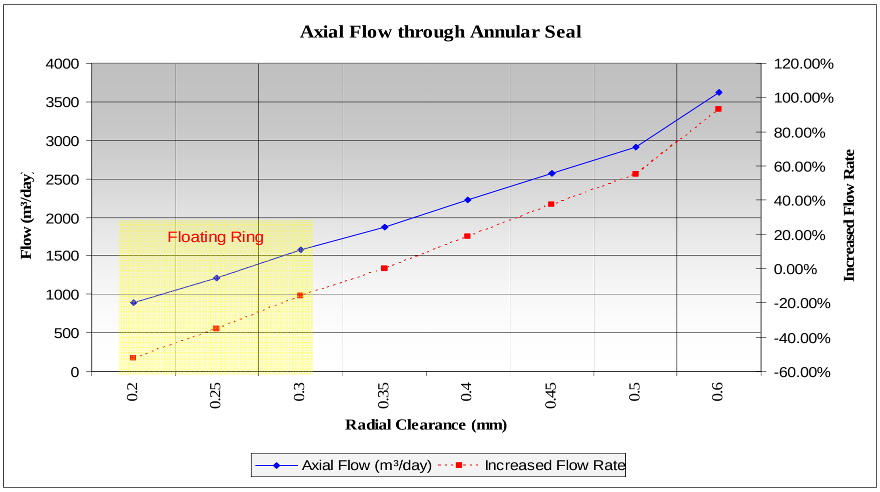

Fig. 11. Axial Flow through Annular Seal

The resulting fluid force on the ring was plotted in Fig. 12. In relation to the fluid forces, the most relevant variation is due to eccentricity and the type of the ring surface and less with radial clearance. This force should be equal to the friction force plus ring weight to lock up the ring.

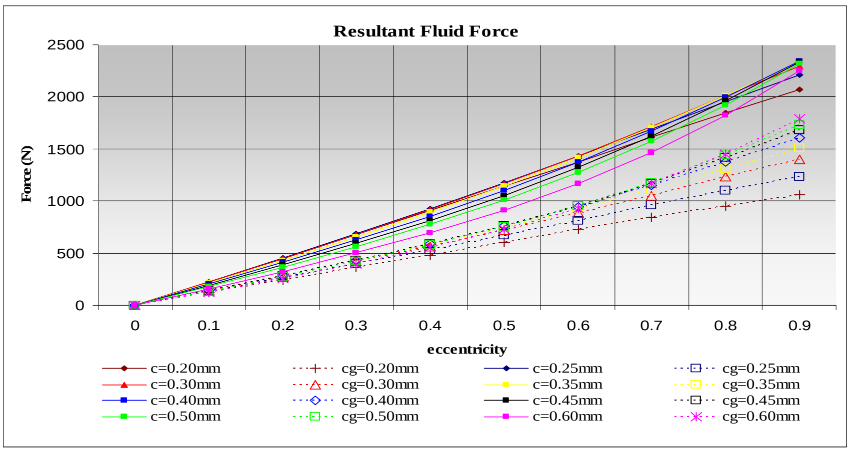

Fig.12. Resulting Fluid Force on the seal

6. CONCLUSIONS

After analyzing the values shown in figure 9, one is able to conclude that the cause of high amplitude vibration in supersynchronous frequency, around 4.260 and 5.500rpm, is due to the excitation of a damped natural frequency. This behavior normally starts when the radial clearance is higher than 0.45mm as observed on site. However, if the fixed ring is replaced by a floating ring, it is possible to increase the first mode or the damping ratio as predicted in cases 9 until 16. Thus, the application of a floating wear ring instead of a static ring in annular seal could be a solution to avoid the supersynchronous whirl frequency that usually excites the critical speed. Moreover, the floating ring could reduce the weariness rate by erosion as it allows assembling with smaller clearance.

The dynamic coefficients just vary significantly when the eccentricity is above 0.8. So, it becomes less important the lock-up position of the floating ring since it is designed to work with the eccentricity below 0.8.

Considering that this pump runs in low speed, 3550rpm, the instability caused by cross-coupling stiffness rarely happens. So, it may be better to design smooth ring than groove ring since the first one introduces more damping. But, if the friction coefficient is too small, the ring with groove could be a good choice due to lesser fluid force generated (Fig. 12), allowing less friction force to lock up the ring.

Figure 11 shows that it is predicted an increase in efficiency of pump around 50% when replacing the fixed ring by a floating ring. Similar result could be gotten using a nonmetallic wear ring, but the weariness rate is high due to contact between rotational and static ring.

In the future it should be tested this pump with floating wear ring on site condition to confirm the proposed presented in this paper. Similar floating wear ring was tested in a multistage pump and the results were satisfactory because this equipment ran with small amplitude vibration and erosion rate.

7. REFERENCES

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