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Investigation of Centrifugal Pump Face Seal

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***Abstract*—The rotating face mechanical seal principle is adaptable to serve a tremendous number of sealing needs standard mechanical seals can suit most requirements-including temperatures and shaft speeds through the choice of secondary seal and the combination of seal and seal face materials which are offered. Seals can be ordered in balanced configurations to seal pressures or used in a multiple for extremely high pressures or especially severe fluid services. Special mechanical seals can be furnished to meet the most demanding of industrial applications considering pressure, temperature, speed or fluid.**

Keywords — Seal; face material; pressure; speed; temperature

1. INTRODUCTION

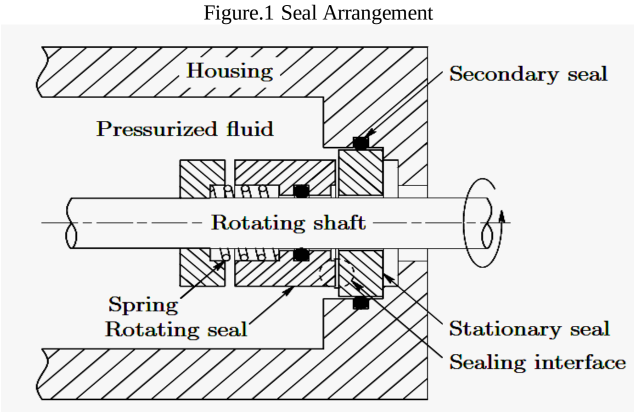
Almost everywhere pumps with rotating shafts are used a shaft seal is involved. The shaft seal forms a barrier between what is inside the pump and the atmosphere pump with a through-shaft is not completely sealed. It is a challenge to the entire pump industry to minimize leakage. There are countless variants of shaft seals, reflecting the diversity of the pump industry, and the need for specific solutions for individual situations. In its most basic form, a shaft seal combines a rotating part with a stationary part.

When properly designed and installed the rotating part rides on a lubricating film, only 0.00025 mm in thickness. Should the film become too thick, the pumped medium will leak if the film becomes too thin, the friction loss increases and the contact surfaces overheat, triggering seal failure.seal performance greatly influences pump performance. When functioning correctly, the seal remains unnoticed. As soon as it starts to leak, however, significant problems can arise with either the pump or the surrounding environment. The importance of the shaft seal must never be under estimated during pump design, operation, or maintenance

2. DETAILS OF PUMP AND SEAL USED

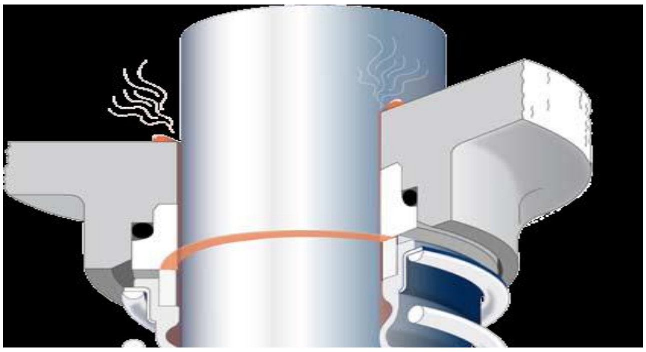
The pump selected for the process is .5 hp monoblock centrifugal pump with the specification of Voltage Range-200V-240V,50Hz, AC Single phase,speed-2750 rpm,head max 33m,discharge max 765 lpm,operating pressure of max 10 bar.

Figure.1 Seal Arrangement



A mechanical shaft seal consists of two main components a rotating part and a stationary part. See fig. 1. The rotating part is axially pressed against the stationary part. In the following, we shall focus on the mechanical shaft seal and its many construction possibilities and applications. As previously stated, a pump with a through-shaft is not leak proof. The mechanical shaft seal is essentially a throttle arranged around the shaft. It reduces leakage between the pump and the surroundings to an absolute minimum. The clearance between the stationary and rotating part of the seal must be small in order to reduce leakage.

The lubricating film formed in the sealing gap during pump operation results in the escape of some of the pumped medium to the atmospheric side. If the mechanical seal works well and
no liquid appears, the lubricating film has evaporated due to heat and pressure decrease in the sealing gap. Therefore, no liquid seeps out of the seal.



To maintain a good seal, the surface finish at the sealing interface must be quite smooth. Normally the surface roughness value of seal faces is less than 05 micron so it will make much influence on the result. The geometric values and other design parameters chosen for this case are representative of those used in a 0.5 hp monoblock centrifugal pump and are listed in the table shown below.

Table.1 Operating Condition

Parameter	Symbol	Value	Units
Shaft speed	ω	200	rad/sec
Spring force	F_{spring}	10	N
Seal outer radius	r_{outer}	10	mm
Seal inner radius	r_{inner}	7	mm
Fluid inlet pressure	P_{in}	1.0132	bar
Fluid outlet pressure	P_{out}	6.0	bar
Coefficient of friction	μ	0.07 to0.1	-

3. ANALYTICAL CALCULATIONS

A.Heat Generation:

The heat generation was estimated and the heat flux on the seal face was obtained assuming uniform heat flux on the surfaces due to the rubbing friction between the mating rings in the mechanical face seal. The pressure was assumed to be uniform over the face of the mating ring.

Heat is generated from friction due to high-speed sliding motion at the nose/runner interface. The heat generated is a function of axial force, sliding velocity, and friction coefficient.

To derive an expression for heat generation, the frictional work of the seal must first be determined. The friction force of the sealing pair is given by

Friction force $\mathbf{F_f = F_n\, f}$

Where f is the coefficient of friction between the mating rings. If it is assumed that 100% of the frictional work of the seal is expressed as heat, the total heat generation due to friction is

Heat generated $\mathbf{Q_f = F_f\, v}$

V= sliding velocity of rotating ring in m/s

Sliding velocity of rotating ring $= \boldsymbol{\omega\, \pi\, ((r_{outer} + r_{inner})/2)}$

The heat generated due to rubbing can be given as

$$\mathbf{Q_f = F_f\, \omega\, \pi\, ((r_{outer} + r_{inner})/2)}$$

Contact pressure = force/area

Contactpressure=$\mathbf{P_{contact}}$=**Fn/area=Fn/ π ((r^2$_{outer}$ + r^2$_{inner}$))**

Heat flux due to friction:

$\mathbf{q_f = P_{contact}\, \omega\, \pi\, f\, ((r_{outer} + r_{inner})/2)}$

Where Q_f - The calculated heat generated in a mechanical seal due to rubbing in between the mating rings, W

q_f – Heat flux, W/m^2

$P_{contact}$ - The mean pressure between the mating rings, N/m^2

ω - The angular velocity of the rotating ring, rad /sec

r_{outer} - Outer radius of the rotating ring, m

r_{inner} – Inner radius of the rotating ring, m

f - The coefficient of friction in between the rubbing parts.

This fluid pressure has to at least support some of the applied load. The spring force assures static equilibrium in the axial direction due to the hydrodynamic pressure or contact pressure in between the faces. The following mathematical formulation is derived for unbalanced seals.

The net axial force F_n on the sealing interface in a typical dynamic seal arrangement is the sum of the force due to the pressurized fluid and the force due to the spring.

$F_n = F_{spring} + F_{fluid}$

Where, F_{spring} is the spring force at its installed length

F_{fluid} is the force of fluid pressure acting to push the installed ring.

F_{fluid}= Area X pressure of fluid

Area= $\pi\, (r^2_{outer} - r^2_{inner})$

Heat Flux Values:

The heat flux is calculated by substituting the numerical values given in table in the heat flux equation

Table.2. Calculated Heat flux values at various coefficients of friction

Coefficient of friction value	Heat Generated Q_f W	Heat flux q_f W/m^2
0.07	3.7366	23333.33
0.08	4.2704	26666.66
0.09	4.8042	30000.00
0.1	5.338	33333.33

B. Deformation Calculation:

Normally the Deformation Is Caused By Axial Forces, Radial Forces, Thermal Gradient. In this case, the influence of axial is much higher and the other forces effect is negligible. The mean radii on the two rings where are the P_a force application points are

Mating ring

$$r_m = \frac{D + d}{4}$$

Primary ring

$$r_p = \frac{D + d_H}{4}$$

The rotation couple value is

$$\overline{M}_A = P_A (r_p - r_m)$$

Where $\overline{P}_A = Kp_1 b$

For a simple ring, the spinning angle is after Gamma:

$$\varphi = \frac{12\, M\, r_m}{E l^3 \cdot \ln \frac{r_a}{r_i}}$$

In all situations, the angle ϕ is very small so Sin ϕ ≈ϕ

The deformation in rotating ring is

$$\mathbf{S_{MA} = \phi\, b\, C_F}$$

By substituting ϕ & M relations, the equation obtained is

$$S_{AMa} = \frac{12 b^2 K p_1 (r_p - r_m) r_m c_F}{E_A l_A^3 . \ln \frac{D}{d}}$$

The obtained values are

- K= 0.8 (selected seal type is Balanced type)
- P_1= 0.6 MPa (fluid pressure)
- r_p= (10+7.5)/2=8.75mm
- r_m= (10+7)/2=8.5mm
- C_F=1
- E_A= 2.8×10^9 Pa =2800 MPa
- The following data are rotating ring dimensions
- l_A = 2.5 mm
- D= 20mm
- d = 14mm
- b=5mm

By substituting these values in deformation equation we can find the deformation;

The deformation obtained is

$$\mathbf{S_{ma} = 2.8746 \times 10^{-8}\ m}$$

In this case the deformation is for steady state condition.

C. Calculation Of Wear Rate:

The actual wear height can then be calculated using the following formula

$$\Delta h = (H - h)$$

Where, Δh- Decrease of the height caused by wear of the face [mm]

H- Height of the Groove before the wear test

h- Height of the Groove after the wear test

The specific wear rate, k_S, is calculated according to the following equation

k_s= ΔV/ Fn X S = A_{seal} X Δh / Fn X S mm^3 /Nm

Where ΔV - The worn volume of the seal,
Fn - The normal force,
S - The sliding distance
A_{seal} - The contacting seal area.

The sliding distance of the rotating distance can be found by using the following formula which is in terms of mean radius, Angular Velocity and time of run.

Sliding distance =S= π r ω t in m

Where r - Mean radius of the rotating seal in m
ω - Angular speed of the seal in rad/ sec
t - Time of the run in sec

A travelling microscope is an instrument for measuring length with a resolution typically in the order of 0.01mm. The precision is such that better-quality instruments have measuring scales made from Invar to avoid misreading due to thermal effects. The instrument comprises a microscope mounted on two rails fixed to, or part of a very rigid bed. The position of the microscope can be varied coarsely by sliding along the rails, or finely by turning a screw. The eyepiece is fitted with fine cross-hairs to fix a precise position, which is then read off the vernier scale. The groove heights were measured by using a Traveling microscope whose least count is 0.001cm.

After measuring the groove heights the mechanical face seal was fixed in monoblock centrifugal pump without any angular misalignment and the pump was allowed to run for 100 hours the mechanical face seal again removed from the centrifugal pump and the groove heights were measured.

Table .3. Heights of mechanical face seal grooves

Groove position	**Height before wear test H mm**	**Height after wear test h mm**	**Decrease of the height Δh=(H-h) mm**
First	1.32	1.25	0.07
Second	0.62	0.54	0.08
Third	0.31	0.26	0.05
Fourth	0.55	0.50	0.05
	Average		0.0625

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Wear rate:

Δh = 0.0625 mm

S= π r ω t

Mean radius = r = (outer radius + inner radius)/2
= (20+14)/4
=8.5 mm= 0.0085m

Angular speed= 250rad /sec (2400 rpm)

t = 20 hours= 20*60*60=72000 seconds

S=0.0085 × 250 ×72000 = 306000m

Area of seal face = π× (20^2-14^2) =640.885mm^2

By substituting the values in equation 6.2

ΔV/ Fn X S = A_{seal} X Δh / Fn X S

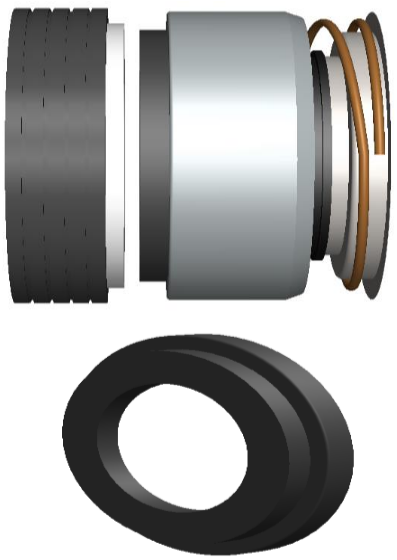
The specific wear rate of the silicon carbide and the carbon graphite material combination is

k_s **=2.08×10^{-7} mm^3/Nm**

4. FINITE ELEMENT ANALYSIS

The finite element method has become a powerful tool for the numerical solutions of a wide range of engineering problems. It has been developed simultaneously with the increasing use of the high- speed electronic digital computers and with the growing emphasis on numerical methods for engineering analysis. This method started as a generalization of the structural idea to some problems of elastic continuum problem, started in terms of different equations

Figure.2 Seal model



Analyzing the parameters like normal stress, Maximum principal stress, strain and deformation of the mechanical face seals for different coefficient of friction values and environmental temperatures during the startup and shutdown periods. The modeling of the mechanical face seal was done and the analysis was performed by using Ansys workbench 14.5 version. In Ansys Transient, structural analysis (Ansys solver) was selected to meet the transient operating conditions the model was imported into analysis as a IGES file. In the model, the face materials are going to be in contact and the remaining parts like spring, bellows and steel support are only to ensure the contact between the face materials in this different face material are selected and analyzed.

Figure.2 Model For Analysis

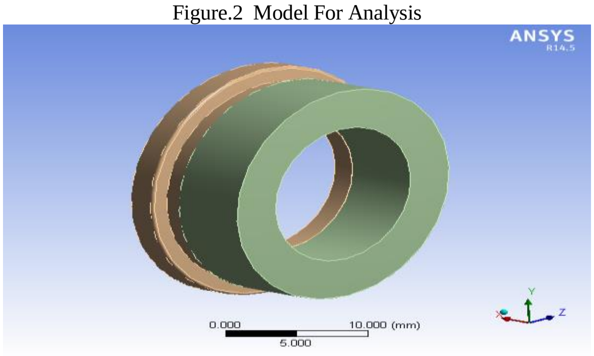


Table.4. Material Properties

Property	Silicon Carbide	Carbon Graphite
Density	3100 kg m^{-3}	1720 kg m^{-3}
Coefficient of Thermal Expansion	4.e-006 C^{-1}	1.2e-005 C^{-1}
Specific Heat	780 J kg^{-1}C^{-1}	434 J kg^{-1}C^{-1}
Thermal Conductivity	0.72 W m^{-1}C^{-1}	60.5 W m^{-1}C^{-1}
Compressive Ultimate Strength	3.9 x 10^{9} Pa	2.08 x 10^{9} Pa

Load Details For Mechanical Face Seal:
The following input data are used for the analysis, tin which speed of the pump was measured with a tachometer, the spring force was measured with the compression spring testing equipment and the fluid pressure was noted from pressure gauge.

Table.5.Load details

Parameter	Symbol	Value	Units
Shaft speed	ω	200	rad/sec
Spring force	F_{spring}	10	N
Seal outer radius	r_{outer}	10	mm
Seal inner radius	r_{inner}	7	mm
Fluid inlet pressure	P_{in}	1.0132	bar
Fluid outlet pressure	P_{out}	6.0	bar

A.Transient Conditions For Pump Start Up
For the analysis transient structural type was selected because the operating parameters were changing with respect to time. The input parameters like rotational velocity, spring force and the fluid pressure are varying from the startup to steady state.

Figure.3.Rotational Velocity Vs Time Graph For Startup

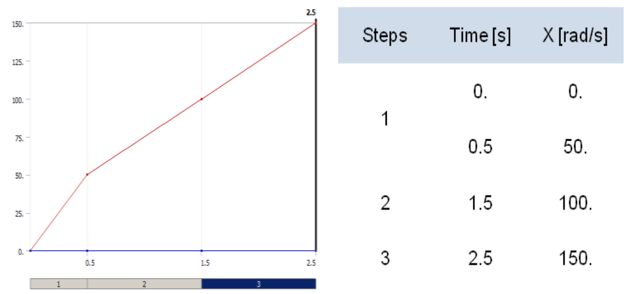


Figure.4.Spring Force (N) Vs Time (s) Graph for startup

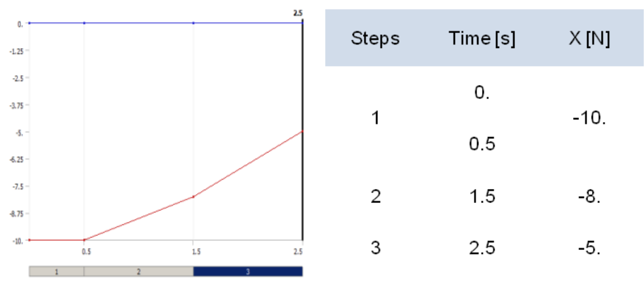
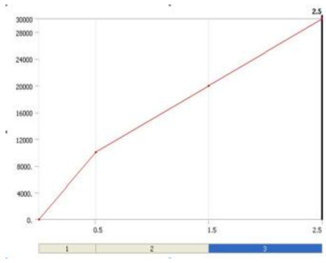


Figure.5.Fluid Pressure (Pa) Vs Time (s) for startup

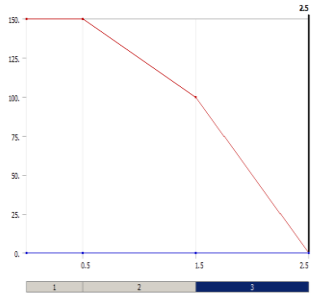


Steps	Time [s]	X [MPa]
	0.	0.
1		
	0.5	1.e-002
2	1.5	2.e-002
3	2.5	3.e-002

B. Transient Conditions For Pump Shut Down:

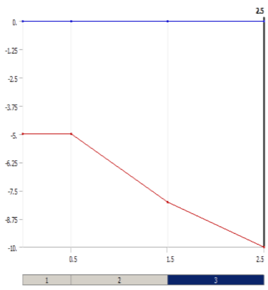
The transient condition was defined for initial 2.5 seconds from the startup of the pump to steady state (normal running) and for the shut up period the input values were defined as follows.

Figure.6.Rotational Velocity Vs Time Graph For Shut Down



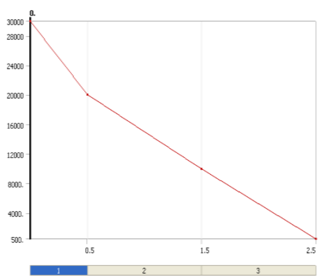
Steps	Time [s]	X [rad/s]
	0.	
1		150.
	0.5	
2	1.5	100.
3	2.5	0.

Figure.7.Spring Force (N) Vs Time (S) Graph For Shut Down



Steps	Time [s]	X [N]
	0.	
1		-5.
	0.5	
2	1.5	-8.
3	2.5	-10.

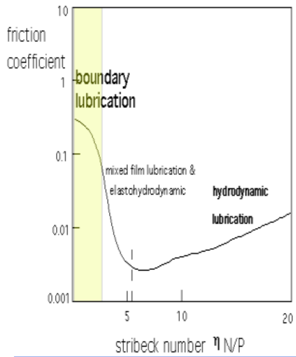
Figure.8.Fluid Pressure (Pa) Vs Time (s) for shut down

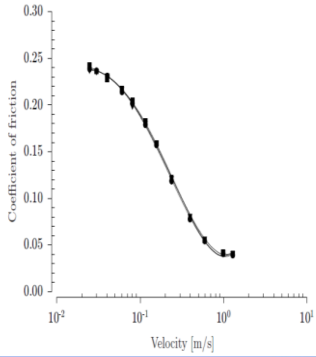


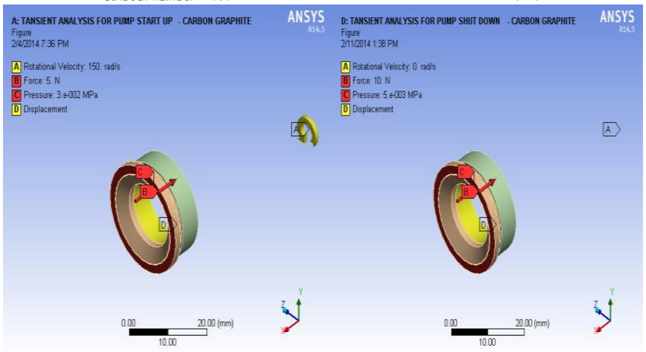
Steps	Time [s]	X [MPa]
	0.	3.e-002
1		
	0.5	2.e-002
2	1.5	1.e-002
3	2.5	5.e-003

Normal Force On The Rotating Ring:

The normal force is the force acting on the rotating ring to close against the stationery face. This force is mainly offered by the spring and by the fluid force acting on it. A frictional interaction is defined between the two seals. The coefficient of friction value varies based on the quality of the fluid and the lubrication between the face materials so this has been performed for various coefficient of friction values from 0.07 to 0.5. The lubrication regime is modeled in the contact region with different values of frictional coefficient. The chosen values were depicted in order to simulate the dry contact and hydrodynamic regime. The stress and stress strain state is insignificantly influenced by the value of the coefficient of friction. But in normal condition the coefficient of friction value between these material pair is 0.07.The spring force gets reducing when the sliding velocity increases this can be found by using strikbeck curve as shown below. The "Stribeck curve" or "Stribeck–Hersey curve" used to categorize the friction properties between two surfaces



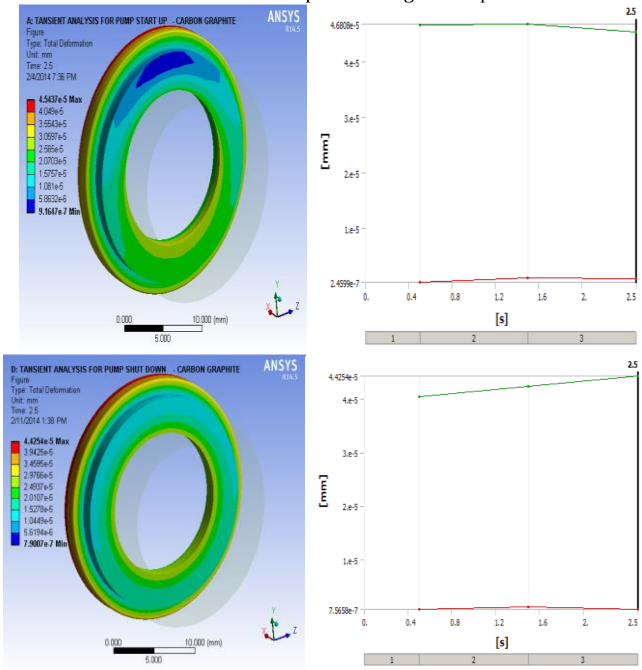




Effect on deformation:

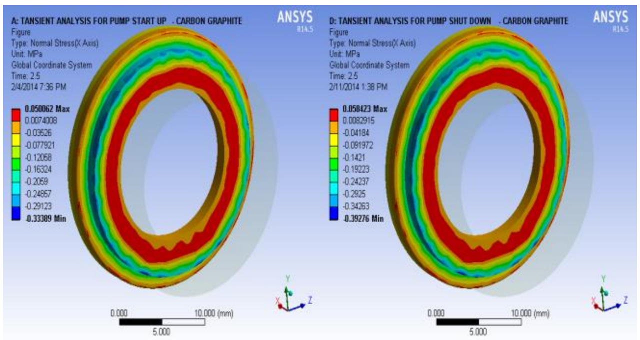
The deformation of the rotating ring was observed for various coefficients of friction. The deformation of the ring changes when the coefficient of friction varies one important thing observed is the environmental temperature doesn't have much influence on the deformation of the ring. It has been found by varying the environmental temperature alone for different coefficients of friction values. The external loads like rotational velocity, spring force and fluid pressure influence the initial deformation but during normal or steady running condition effect of external load is less on deformation.

Deformation Of Carbon Graphite During Start Up & Shutdown

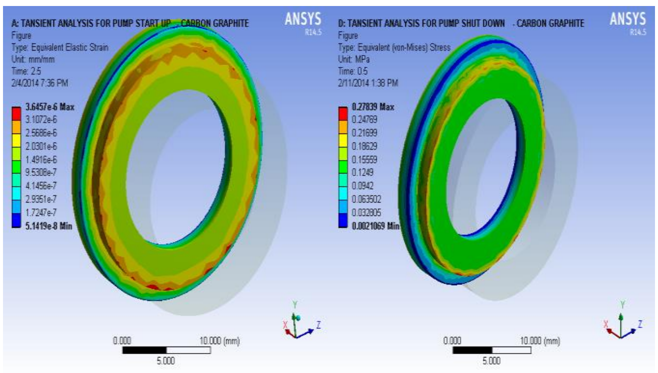


Effect on stress and strain:

The stress value obtained for various coefficients of friction values but the effect on stress and strain value was significantly affected by both environmental temperature and external load. A good similarity can be seen between Von Misses stress and strain.



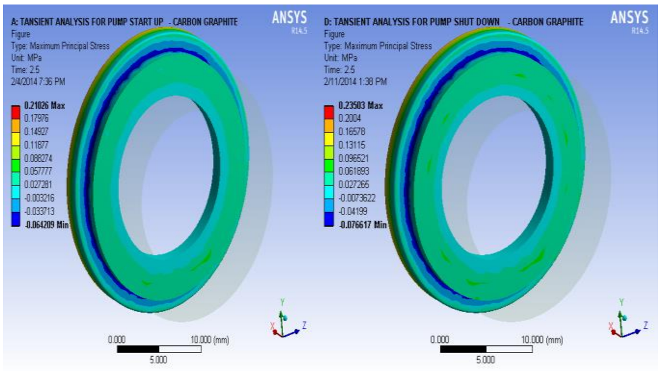
Equivalent Elastic Strain For Pump Start Up And Shut Down Carbon Graphite



Effect Of Maximum Principle Stress:

The maximum principal stress values are oriented in proper way on the circumference of the face seal contact surface and the values are increasing when the load and coefficient of friction value increases. The stress concentration is not distributed throughout the contact surface this characteristic may lead to the blisters. Blisters are shallow surface cracks that occur during startup, when surface layers of the seal face are lifted up by lubricant fluid films.

Maximum Principle Stress For Start Up And Shut Down-Carbon Graphite



The stress concentration is not distributed throughout the contact surface this characteristic may lead to the blisters. Blisters are shallow surface cracks that occur during startup, when surface layers of the seal face are lifted up by lubricant fluid films.

RESULTS AND DISCUSSION:

Using FEM analyses the stress and strain behavior was analyzed for the startup and shutdown period from the results the significant parameter which influence the changes during running condition is coefficient of friction values. So from this result allows us to investigate a proper model, with fewer simplifying hypotheses. The minimum stress and strain state occurs around ambient temperature. The results obtained by this analysis are useful for better understanding of the phenomenon that otherwise leads to contact surface damaging and fluid leakage. Considering the results it is obvious that every face seal has to be modeled and analyzed by FEM, under its own working conditions, in order to assure its good operation. It shows that the mechanical face seals used in monoblock centrifugal can work for wide range of coefficients of friction and environmental temperatures.
The wear rate of the carbon silicon carbide material combination was found at transient operating conditions. By knowing the specific wear rate and the wear geometry the selection of material combination and the life of the seal can be calculated.
Analytical method is used to find the heat generated during the running condition. With the knowledge of the heat generation and heat flux values the required cooling method can be implemented and the life of the seal can be increased.

CONCLUSION

This process helps us to investigate a proper model, with fewer simplifying hypotheses. Considering the results it is obvious that every face seal has to be modeled and analyzed by FEM, under its own working conditions, in order to assure its good operation. The film thickness distribution is directly influenced by face deformation, so it is very important to analyze the deformation during speed ramp up and down.

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