International Journal of Engineering and Advanced Technology (IJEAT)
ISSN: 2249-8958 (Online), Volume-9 Issue-3, February 2020

Structural Analysis of Mechanical Seal Holding for Mixer Pumps

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***Abstract:** A mechanical seal is a device that used to reduce leakage containing pressure or ignoring contamination. Mechanical seal is the device used in centrifugal Pumps, Mixers, Agitators and Compressors to arrest or reduce the leakage. The mechanical seal standard API 682 provides the design guidelines for the seals which are fits below 110mm shaft diameter and 4.2 N/mm² seal chamber pressure. Any seal more than these parameters required special engineering or guidelines needs from technical service. The main aim is to design the mechanical seal for more than 110mm shaft diameter and more than 4.2 N/mm² pressure. Since the basic seal components are mostly standardized and already tested than the seal gland connections are seems to be critical for large shaft diameter and high or heavy pressure applications. Hence the stress and deflection of the mechanical seal gland is analyzed based on barrier pressure and seal chamber pressure with the help of Inventor and another critical area is the seal drive arrangements.*

***Keywords:** Mechanical seal, Mixer Pump, Analysis, Barrier pressure, Seal chamber pressure.*

I. INTODUCTION

Mechanical face seals are used to seal the pressurized fluids in rotating apparatus such as compressors, agitators and pumps. These seals mostly consist of a rotating part mounted on the shaft and a rigidly fixed stationary part is fastened to the housing. The two parts kept in contact by the action of springs and of the pressurized fluid. Better operating conditions are achieved when the seal faces are partly separated by a thin lubricating fluid film, avoiding wear on the faces while reducing leakage rate to tolerable value. When a pump is running, the liquid could leak out of the pump between the rotating shaft and the rigidly fixed pump casing. As the shaft revolves, restricting this leakage can be difficult. Previously pump models used mechanical packing which is also known as Gland Packing to seal the shaft. Since Second World War, mechanical seals have changed packing in plenty applications. The design, structural and thermal analysis of the four wheeler disc break has been carried out by the researcher [8], [11].

Revised Manuscript Received on February 05, 2020.

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Even in small modification in the design leads to major change may happen in results and it is proved with Inconel718 material [6], automobile suspension [10], Passenger cabin comfort [7] and tractor tow pin [9].

1.1 Working of Mechanical Seal

An end faced mechanical seal uses both the rigid and as well as the flexible elements which helps in maintaining the contact at sealing interface and sliding over each other, allowing a rotating element to pass through the sealed case. These elements are hydraulically and also mechanically loaded with spring or with other devices to give the required contact As per Nau.B to hold the two primary sealing surfaces in contact with one another, an actuating force is necessary and it is usually given by the spring. Along with a spring, it's also provided with pressure of the sealed fluid [3]. The primary sealing surfaces must be the parts of the seal which should be allowed only to rotate relative with one another and they should not rotate relative to the parts of the seal that dominance them in place [4]. To maintain this non-rotation a method of drive should be provided. The primary seal is obtained by two flat lapped faces which create a hard leakage path perpendicular to the shaft. Rubbing unit between these two flat contact surfaces decrease the leakage. As in all seals, only one face is held stationary in housing and the second other face is fixed and rotates with the shaft as shown in Fig.1 Among one of each faces there can be a non-galling material like the carbon-graphite. The other material used is generally a relatively hard material like silicon-carbide. Different types of materials are normally used for the stationary insert and revolving searing face in order to prevent sticking of the 2 faces [5]. The smooth face usually has the little mating surface usually called as wear nose.

1.2 Seal Classifications

Mechanical seals are classified in API 682 based on the applications, types and configurations. The seal selection is based on the classifications provided in API The seal configurations covered by the international standard can be divided into three categories, three types and three arrangements. Further, Arrangement 2 and 3 seals can be in three orientations: "face-to-back", "back-to-back" and "face-to-face".

1.3seal Categories

Mechanical seals are categorizes based on application which means the equipment conditions and the stuffing box envelop. There are three seal categories, as follows.

1.3.1 Category 1

One Seals are expected to be used in non-ISO 13709 pump seal chambers, alternatively to meet the dimensional necessity of ASME B73.1, ASME B73.2 and ISO 3069

Type C seal chamber dimensions and their application which are finite to seal chamber temperatures that range in between – 40 °C to 260 °C and absolute pressures till 2.2 N/mm^2.

1.3.2 Category 2

Seals are expected for use in seal chambers meeting the chamber envelope dimensional Requirements of ISO 13709. Their application is limited to seal chamber temperatures from – 40 °C to 400 °C and absolute pressures up to 4.2 N/mm^2.

1.3.3 Category 3

Gives the most extremely tested and seal design that is documented. It is required that the entire seal cartridge is qualification tested as an assembly in the required fluid. They meet seal chamber envelope's requirement of ISO 13709 or the equal. Their application is limited to seal chamber temperatures from – 40 °C to 400 °C and absolute pressures up to 4.2 N/mm^2.

1.4 Seal Types

The three types of seals are as follows.

1.4.1 Type A

TypeA seal is a balanced, inside-mounted, cartridge design, pusher seal with the double springs and in which the flexible elements normally rotates. Secondary sealing elements are elastomeric O-rings.

1.4.2 Type B

Type B Seal is a balanced inside-mounted, cartridge design, non-pusher (metal bellows) seal in which the flexible element normally turns. Secondary sealing elements are of elastomeric O-rings.
If multiport flush design provided it improves heat dissipation for uniform face cooling.

1.4.3 Type C

The Seal also can be balanced, inside-mounted, cartridge-design non-pusher metal bellows seal in which flexible element is normally at rest state. Secondary sealing elements are flexible graphite. Bellows seals are inherently balanced. Stationary metal bellows seals are the primary choice for high temperature service. The Type C stationary bellows configuration is chosen as per standard because of its advantage if the gland plate and shaft lose their perpendicular alignment. In this arrangement, the bellows can deflect to a fixed position to match the rotating face. In a rotating arrangement, Type B, the bellows have to flex and change positions once per shaft revolution to accommodate the run out of the stationary face; however, the rotating metal bellows tend to throw out particulate in middle of the bellows in coking or other particulate-bearing services.
The user should make a note that the rotating bellows seals mostly have the ability to vibrate. Therefore, they are equipped with dampening tabs or devices to control vibration. Stationary bellows seals largely avoid this issue. Metal bellows seals offer benefit of having only the stable secondary seals. This allows their application in high-temperature services where suitable O-ring elastomers are not available [1]. Metal bellows seals are also a cost-effective alternative for services where chemical resistance or cost of O-ring materials is an issue. Type A and Type B seals are suitable for the temperatures that range till 176 °C. Type C seals are for the high temperatures that range till 400 °C.

1.5 Seal Arrangements

The seal arrangements are determined based on the sealant fluid character. Based on the seal arrangements the number of basic seals differs. There are three seal arrangements, as follows.

1.5.1 Arrangement 1

Seal configurations having one seal per cartridge assembly there is no external fluid supply to maintain the sealed environment.

1.5.2 Arrangement 2

Seal structure having two seals per cartridge assembly, with the space between the seals at a pressure less than the seal chamber pressure. Normally this seal arrangement is referred to as the unpressurized seal and used for category 2 application. In this arrangement, if the inboard seal fails then the product will leak into the buffer chamber and diverted to the safe drain

1.5.3 Arrangement 3

Seal structure which is having two seals per cartridge joining utilizing an externally supplied barrier fluid at a pressure greater than that of the sealed chamber pressure. Normally this seal arrangement is referred to as the pressurized seal and used for category 3 application. In this arrangement, if the inboard seal fails then the product will not leak into the barrier chamber because of the pressure differential

II. DESIGN AND CALCULATIONS OF MIXER PUMP

Because of the high operating pressure of 6.4 N/mm^2 Barrier pressure and the shaft size 140mm, the risk is considered high. Due to the High pressure and size of the seal, the seal parts have been designed with large cross-sections for ruggedness. Large chamfers have been incorporated into the seal faces. This will allow better fluid movement down to the seal face contact area. It will also help to reduce the risk of chipping the seal faces. Basic mechanical seal selected as per API 682 type A seal since the seal Chamber pressure is very high. Type A consists of pusher flexible elements. The pressure capability of type A seal is greater than type B & C because these type B & C are having flexible elements with metal bellows whereas type A seal have springs. The seal glands are designed to have pipe port connections for barrier Inlet and outlet to cool the mechanical seal faces. Bearings are selected based on shaft sleeve size since the seal needs to be tolerating high pressure. API 610 seal chamber requirements are considered for all obstructions and Openings

2.1 3D Assemblies of Mechanical Seal

The three-dimensional models of mechanical seal assembly drawings are created with the help of Inventor software. Autodesk Inventor developed by U.S.-based. The three-dimensional models of mechanical seal assembly drawings are created with the help of Inventor software. Autodesk Inventor developed by U.S.-based. The three-dimensional models of mechanical seal assembly drawings are created with the help of Inventor software. Autodesk Inventor developed by U.S.-based.

The three-dimensional models of mechanical seal assembly drawings are created with the help of Inventor software. Autodesk Inventor, developed by U.S.-based software company Autodesk, is 3D mechanical CAD design software for creating 3D digital prototypes used in the design, visualization, and simulation of products. Inventor includes an integrated motion simulation and assembly stress analysis environment. Users can input driving loads, friction characteristics, and dynamic components, then run dynamic simulation tests to see how a product will work under real-world conditions. The simulation tools can help users optimize strength and weight, identify high-stress areas, identify and reduce unwanted vibrations, and size motors and actuators to reduce energy consumption. Finite element analysis (FEA) lets users validate the component design by testing how parts perform under loads (using actual load information instead of estimates). Inventor's Parametric Studies and Optimization technology lets users modify design parameters from within the assembly stress environment and compare various design options, then update the 3D model with the optimized parameters as shown in Fig.1.

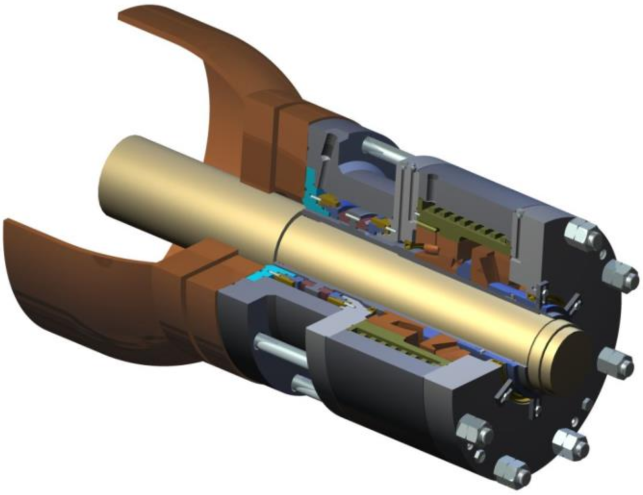

Fig.1 3D cut sectional view of mechanical seal

2.2 Barrier Pressure Acting Area Calculation

Barrier pressure is acting on the basic seal area. That means the entire basic seal soaked with barrier pressure [2]. But due to seal balancing, it is not necessary to calculate pressure for basic seal as shown in Fig.2. The barrier pressure acting area calculated and presented in Table 1.

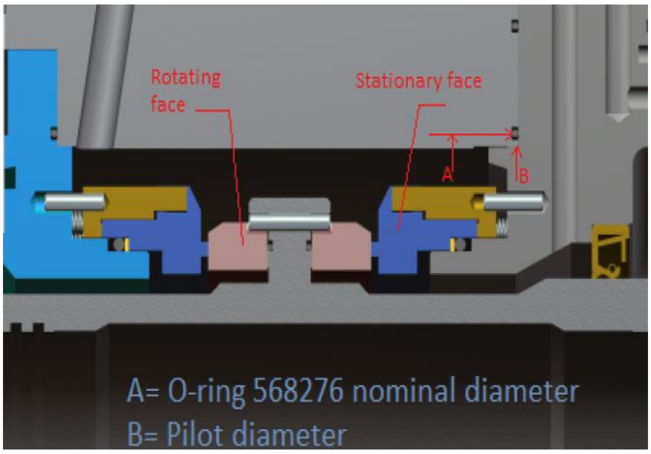

Fig.2 Barrier pressure acting area on the mechanical seal

Table 1 Barrier pressure acting area calculation

Barrier Pressure Acting Area				
S. N	Description	Notation	Values	Unit
1	O-ring Diameter	OD	280.6	mm
2	Pilot Diameter	ID	271.0	mm
3	Area	A= $\frac{\pi}{4} X (O.D^2 - I.D^2)$	4148.6	mm^2

As per the American petroleum institute, standard maximum mechanical seal withstand capacity is 4.2 N/mm^2. By taking the mixer seals to report the pressure limitations for the mixer is Considered 20-40 N/mm^2 only. So our design pressure must be higher than existing design values. The barrier pressure is taken as 6.4 N/mm^2 is given Table 2.

Table 2Barrier force on the gland

Barrier force acting on the gland				
S.N	Description	Notation	values	units
1	Barrier Pressure	P1	6.4	N/mm^2
2	Area	A	271	mm^2
3	Barrier Load	N1=P1 X A	26651	N

2.3 calculation of Force on End Plate Due to Barrier Pressure

The box pressure acting area on seal assembly values are given in Table 3 and the pressure acting on end plate is given in Table 4. The load acting on seal gland as shown in Fig.3

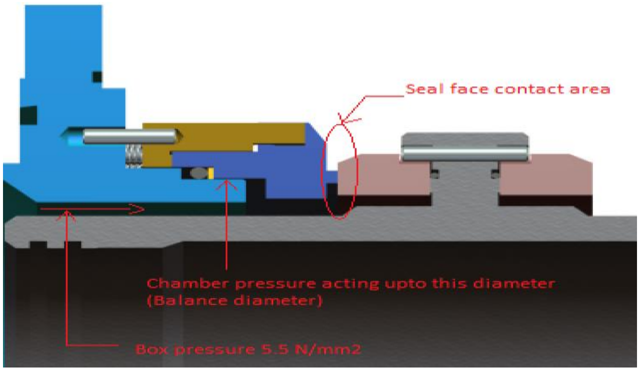

Fig.3 Chamber pressure acting area on the mechanical seal

Table 3 chamber force on the gland

Force on gland				
S.N	Description	Notation	Values	Units
1	Balance Diameter	D1	196.85	mm
2	Max Vessel Pressure	P2	5.5	N/mm^2
3	Load Due To Box Pressure	N2=P2 × A	167388	N

Table 4 Pressure acting area on end Plate

Pressure acting area on end Plate				
S. N	Description	Notation	Values	Units
1	Fore on End Plate	N	167388	N
2	Load Acting ID	ID	226.2	mm
3	Load Acting OD	OD	320	mm
4	Load Acting Area	$\frac{\pi}{4} \times (O.D^2 - I.$	40238.71	mm^2

2.4 Minimum Thickness for Flat Heads, Cover and Blind Flanges

The gland thickness calculations are calculated as per ASME class Vii and presented in Table 5.

Table 5 Gland thickness calculation

Minimum Thickness Calculation (AS PER ASME CLASS VII)				
S.N	Description	Notation	Values	Units
1	Factor	C	0.7	No unit
2	Pressure	P= (P1+P2)	10.56	N/mm^2
3	Max Allowable Stress For S316	S	292.9	N/mm^2
4	Joint Efficiency	E	0.5	No unit
5	Max Press Acting Diameter	D	320	mm
6	Thickness	T =D x √(CP/SE)	71.92	mm

Table 6 Gland thickness

S.N	Minimum thickness	Provided thickness
1	71.92 mm	78.10 mm

Table 6 shows our design is safe since the actual thickness greater than the minimum required

2.5 Gland Thickness Calculation

Table 7 Pressure acting area on end Plate

Evaluate the Gland (Cooling Jacket) thickness by Radial load				
S.N	Description	Notation	Values	Units
1	Inner Diameter	D	290	mm
2	Load Acting Length	L	106.6	mm
3	Load Acting Area	A=πxDxL	971192	mm^2
4	Radial Load of FAG 23232 – E1A-M	N	15000	N
5	Pressure Acting at ID of Cylinder	P=LOAD/AREA	0.15	N/mm^2
6	Allowable Stress for SS316	σt	205	N/mm^2
7	Factor to be Considered	FOS	0.05	-
8	Minimum Thickness	T=P x d/2xFOSσ_t	2.12	mm

The contact area between the gland and cooling jacket is very critical because the pressure acting on the cooling jacket is much enough to distort the gland. Refer to table 7 for minimum thickness. Actual thickness provided is very minimum required thickness.

2.6 Connecting Bolt-Calculations

Table 8 connecting bolt specifications

The equation for Tensile Stress Area of Screw			
S. N	Description	Values	Units
1	Pitch Diameter	11.43	mm
2	For 1/2-13 UNC Tensile Stress Area	102.6	mm^2
3	Maximum Allowable Tensile Stress	500	N/mm^2
4	Factor of Safety	2.5	-
5	Barrier Pressure	6.4	N/mm^2
6	ID	196.85	mm
7	OD	275.97	mm

Table 9Equation for Tensile Stress Area of Screw

The equation for Tensile Stress Area of Screw				
S.N	Description	Formula	Values	Units
1	The capacity of the one Screw	Area of screw X max.allow.stress = 102.6×1× (500/2.5)	20520	N
2	Total Pressure Acting Area	$A=\frac{\pi}{4}\times(O.D^2-I.D^2)$	29381.4	mm^2
3	Force Due to Barrier Pressure	6.4×Total Pressure Acting Area	188040.9	N
4	Minimum No of 1/2-13 UNC Screws Required	(188040.9/205201)+1	10.2	Numbers

Currently, No of Screws used is 12 Nos and Table 9 shows the minimum requirement

2.7 Seal Gland Analysis

Table 10 properties of the stainless steels materials

Property	Values
Density ρ	8.03 g/cm³
Yield strength	2050 N/mm^2
Tensile strength	5150 N/mm^2
Young's modulus E	2000 KN/mm^2
Thermal conductivity	14.6 W/m-K
Specific heat	485 J/kg-K
Poisson's Ratio	0.27-0.30

The above-mentioned table 10 shows mechanical properties are taken from standards and the values keyed in the inventor. The mechanical seal gland is tightened using stud and nut. The torque values of the stud can be taken from standards and applied in boundary conditions. Following assumptions are taken for Autodesk Inventor analysis below assumptions doesn't provide any negative impact on the analysis, Gravity or self-weight of components is ignored, Fluid characteristics are not considered, another seal component's not considered and spring force not considered.Pressure inputs and boundary conditions is given in table 11.

Table 11 Pressure on seal glands

Load Type	Pressure
Pressure on seal faces	6.4 N/mm^2
Pressure on gland	5.5 N/mm^2
Pressure on the cooling jacket	0.15 N/mm^2

It is very important to define the pressure acting places in Inventor analysis because the result or outcome from the analysis is based on the input we provided refer table 11 for pressure values provided to mechanical seal assembly drawing. The mechanical seal must be constraint with the stuffing box to fixed contact. The barrier and seal chamber pressure pressures and the bearing loads are provided. To corresponding glands with fixed no sliding contact type

III. RESULT AND CONCLUSION

3.1 Seal Design Results

The possibility of the seal design for 110mm shaft diameter is more in pusher seal when compared to the bellow seal because the basic seal (Stationary and rotary face) availability is more in pusher seal and the pressure capability is high in pusher seal design refer to table 12 for the seal design result

Table 12 Seal design result

S. N	Parameters	Minimum required gland thickness	Provided gland thickness
1	Gland thickness	71.92 mm	78.10mm
2	Cooling jacket lip	2.12mm	5.6mm
3	No.of binding screws required	10 Numbers	12 Numbers

The thickness of the gland and gland outer diameter is designed to match API requirements and the correct allowances provided for pipe ports.

IV. ANALYSIS RESULTS

The mechanical seal glands are very critical to the performance on seal since the primary sealing element is mounted with the seal gland. Any deflection on seal gland is provided large impact on seal performance the stress and deflection on the gland is depending on pressure acting inside mechanical seal as shown in Fig.4 for Von mises result on seal gland

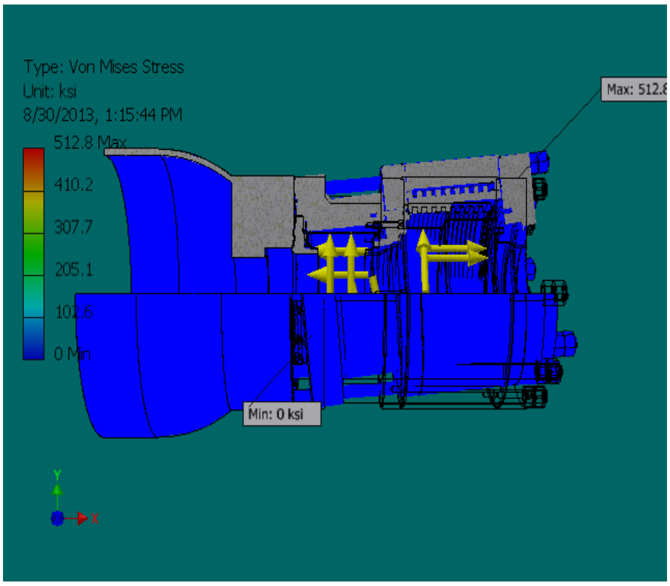

Fig.4 Von Mises result on seal glands

The above figure shows Von-mises stress on flanges after the application of seal chamber pressure and barrier pressure. The maximum Von-mises stress observed in the seal gland is 24230 psi which is within yield strength of the gland material 33091 psi.

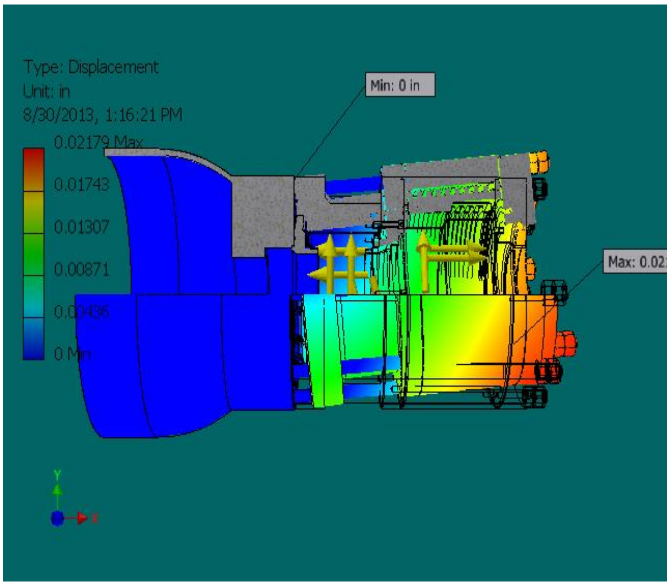

Fig.5 Displacement on seal glands

The above figure shows displacement on flanges after the application of seal chamber pressure and barrier pressure. The maximum displacement observed is 0.508 mm which is within allowable tolerance of 1.5mm and seal setting dimension as shown in Fig.5 for displacement on seal glands.

V. CONCLUSIONS

Custom engineered mechanical seal design developed for mixers. Analyzed the design in Inventor and calculated the gland thickness and fasteners requirements. Three-dimensional model created to find the leakage paths and to analyze the displacement. Possible drive arrangements for mixers are identified and designed the three types of drive arrangements based on size availability and selected the optimum drive arrangements

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