A Review On Effect Of Lubricants On Frictional Force Developed On Seals For Turbo Power Appliances

NISHIKANT GOPAL PATIL[1], ANIL VERMA[2], DR. G.R. SELOKAR[3], DHANANJAY YADAV[4]

[1]RESERCH SCHOLAR, MECHANICAL ENGINEERING, SSSUTMS, M.P., INDIA
[2] ASSISTANT PROFESSOR, MECHANICAL ENGINEERING, SSSUTMS, M.P., INDIA
[3]REGISTRTAR, MECHANICAL ENGINEERING, SSSUTMS, M.P., INDIA
[4]ASSISTANT PROFESSOR, MECHANICAL ENGINEERING , SSSUTMS, M.P., INDIA

ABSTRACT

Mechanical seals were the machine aspect mainly designed to form a running seal between a rotating and stationary part, especially architect to prevent leakages of liquid or gases in centrifugal pumps, mixers, or other rotating equipment's. The execution and reliability of the mechanicals seals mainly depends on the wide range of factors such as equipment design, operation conditions, pumped or sealed fluid, pressure and pump speed, support system, material characteristics, choice, and pairing of seal faces. This research work presents an investigative study on the frictional characteristics of alumina, 316 stainless steel, and phosphor bronze materials against resin-impregnated carbon in the form of mechanical seals. To study the frictional characteristics of seal an experimental setup was designed for fluctuating normal load and speed. Among three combinations pairing of stationary and rotary seal alumina versus resin-impregnated carbon exhibited the superior frictional characteristics. Whereas in comparison with 316 stainless steel versus resin-impregnated carbon and phosphor bronze versus resin-impregnated carbon, the latter exhibited the better frictional characteristics. Phosphor bronze pair and 316 stainless steel pair resulted in low and stable friction coefficient in the range of 0.07–0.08 and 0.12–0.13, respectively. This work highlights that the phosphor bronze might be an alternate substitute for the applications involving 316 stainless steel as a seal face material, where the frictional characteristics was a major concern.

Lubrication plays a major role in reducing friction between the two surfaces. in mechanical seals lubrication reduces friction between interfaces to operate for extended period of time that operate for extended period of time. The use of organic based vegetables oils has increased worldwide due to biodegradable and nontoxic in nature instead of using mineral oil-based lubricants which mostly affected by environmental issues. In this work we go through with experimental study with pairs of seals material (tungsten carbide and resin impregnated carbon) with eco-friendly lubricant from class of vegetable oils— soybean oil and canola oil with an eco-friendly solid lubricant i.e. boric acid powder. The friction characteristics was studied under unlubricated conditions, independent paraffin oil, soybean oil, canola oil lubricating modes and finally 1 wt.%, 3 wt.% and 5 wt.% of boric acid powder mixed individually with soybean and canola oil. After all running-in test of all lubricating conditions, 5 wt.% of boric acid powder mixed with soybean oil had contributed a hybrid tribo film and resulted lowest friction coefficient value in the range of 0.06–0.07.

Keywords- *mechanical seal, Pump, Mechanical seal test rig, Lubricating unit, frictional force sensor, Proximity sensor, sensor disc, WINDUCOM 2010, Magnetic Stirrer.*

1. INTRODUCTION

Dynamic sealing technology is concerned with providing sealing solutions for machinery applications in which the conventional sealing ropes lack the required durability. This rule applies to all products in general and to mechanical seals in particular. Investigations have shown that the mechanical seal is critical component in pumps. The most frequent cause of damage to mechanical seals is still cause's damage to the sliding faces due to improper lubrication between mating rings.

1.1 MECHANICAL SEALS

A mechanical seal is a device which helps to join the rotating systems or mechanisms together using spring or hydraulic pressure. This type of seal can be used in plumbing systems to prevent leakage in the tubes and pipes. It can also be used for manufacturing industries and other high-pressure applications. The basic rotating face mechanical seal principle is adaptable to serve a tremendous number of sealing needs and standard mechanical seals can suit most requirements including temperatures to 500˚F and shaft speeds up to 3600 rpm through the choice of secondary seal and the combination of seal and face materials which are offered. Seals can be ordered in balanced configurations to seal pressures above 200 psi 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 for industrial applications considering pressure, temperature, speed or fluid.

1.2 SEAL INPUMPS

A mechanical face seal is an important component of variety of pumps used in chemical, petrochemical and process industry. The primary function of a mechanical seal is to prevent the leakage of the process fluid from the pump housing and shaft to an end face mechanical seal, also referred to as a mechanical face seal but simply as a mechanical seal, is a type of seal utilized in rotating equipment, such as pumps, mixers, blowers, and compressors. Figure 1.1 shows the pump operation with mechanical seal placements. The liquid may be able to leak out of the pump between the rotation shaft and the stationary pump casing. Since the shaft rotates, preventing this leakage can be difficult. Earlier pump models used mechanical packing (otherwise known as Gland packing) to seal the shaft.

Fig 1.1 Figure 1.1 Pump operations with mechanical seal

1.3 DESIGN MANIPULATION

In addition to the above terms and components, there is an additional terminology, which is worth mentioning. The rings in the mechanical face seals should possess the appropriate material properties. Generally, the primary ring is made of a hard material to allow the softer primary ring to run into it. Alternatively, in some applications, the primary rings are made of the hard material and the mating ring is made of softer materials. In special applications such as operation in some severe environments, the design may call for a hard material to run on another hard material.

As discussed earlier, in a mechanical face seal contact occurs in between the primary and the mating rings at the annulus. The essential character of the surfaces is that they form a type of sliding bearing through which the sealed fluid attempts to flow in the perpendicular direction to the sliding. The hard face sliding against a soft face has been found tribological desirable. This combination provides the best overall performance.

The secondary seal can take a wide variety of different shapes and mechanisms. It has the essential function of ensuring that the primary and mating ring faces are allowed to self-align and maintain close proximity under all operating conditions while at the same sealing between the ring and its mounting. There are several other O-rings that are also secondary seals. However, the secondary seal allows small relative motion for alignment at the same time. Many different types of springs are used.

1.4 MAJOR PARTS

1.4.1 Primary Ring:

The ring is mounted so as to provide flexibility to allow for small relative axial and angular motion for misalignment between the parts. The primary ring also provides one of the seal in surfaces.

1.4.2 Mating Ring

The ring is rigidly mounted to the shaft or to the housing but does not rotate. It provides the second sealing surface. This ring works as a surface guided ring.

1.4.3 Secondary Seal:

It allows the primary ring to have axial and angular freedom of motion while retaining the sealing integrity. The secondary seals are the O- rings in the case shown in Figure 1.2**.**

1.4.4 Spring

All mechanical face seals have some type of a spring mechanism to hold the annular surfaces together in the absence of fluid pressure. The fluid pressure provides a certain force that holds the surfaces together.

1.4.5 Drive Mechanism:

All mechanical face seals must have some type of a drive mechanism or rely on some other features to drive the primary ring in order to make certain that relative motion occurs only at the annular interface. The drive mechanism is designed so as not to reduce the self-aligning characteristics of the primary ring.

1.5 SELECTION OF MECHANICAL SEAL

The mechanical shaft seal should be selected according to the operating conditions at the shaft seal location. These important factors must be considered when selecting a mechanical shaft seal:

- Shaft seal diameter
- Type of pumped medium
- Temperature
- Sealing pressure
- Shaft speed of rotation

1.5.1 Shaft Seal Diameter

The shaft seal diameter must be selected to fit the pump shaft. If no seal with the required diameter is available, the shaft diameter can be changed with a bushing.

1.5.2 Type of Pumped Medium

The chemical resistance of the shaft seal materials to the pumped medium has to be considered. The viscosity of the pumped medium affects the lubrication and leakage of the seal. The viscosity of most media depends on the temperature. A single shaft seal can be used for a dynamic viscosity below 2500 cp (centipoise). For a higher viscosity, a back-to- back seal arrangement should be used.

1.5.3 Temperature

The elastomeric parts of the seal must be able to withstand the temperature of the medium around the seal. This might be different from the temperature of the pumped medium. If the temperature is above the boiling point of the pumped medium, lubrication is poor. This must be considered when selecting seal design and materials. The face temperature rise is a very important consideration for efficient frictional performance and efficient design of mechanical seal. The temperature of sliding surface should be kept minimum for efficient performance.

1.5.4 Sealing Pressure:

The sealing pressure is the pressure around the seal. For high pressures, a balanced seal should be used.

1.5.5 Shaft Speed of Rotation:

If the speed of rotation is low, shaft seals with hard/hard material pairings might produce noise because the lubricating film in the seal gap is extremely thin. At speeds above 15 m/sec, a balanced seal with a rotating seat must be used to reduce seal unbalance.
In addition to these operating conditions, the content of abrasives and additives in the pumped medium might be relevant to consider when selecting seal ring materials. In some instances, the space available for the shaft seal is also an important factor. When selecting the right sealing arrangement around the mechanical shaft seal, also take into account the content of abrasives and the risk of build-up of wearing particles on the atmospheric side as well as the health hazards, explosion risk and toxicity of the pumped medium.

1.6 OTHER FEATURES

Mechanical seals have some unique features other than the sealing function. Thus, the features are,
1.6.1 Invisible leakage
1.6.2 Less friction/power loss
1.6.3 Flexibility-to accommodate shaft deflections and “End play”
1.6.4 No periodical maintenance
1.6.5 Long-life
1.6.6 Low face temperature rise

1.7 MAJOR FAILURES

Here some failures can take place with operating the seals, thus the failures are listed and explained where it occurs. International analyses have clearly shown that mechanical seals are responsible for 39% of the damage to pumps and are therefore the most request cause. A breakdown of the costs accounted for by pump repairs also clearly shows that mechanical seals represent the largest single factor considered at 44%.

- The seal motion was restricted and the faces opened.
- Heat caused the rings to deteriorate.
- The seal materials are improper.
- The seal was installed incorrectly.

1.7.1 Motion Restricted

The spring-loaded (dynamic) seal face constantly moves to maintain full face contact with the stationary seal face. The main reasons for this movement are,

- The stationary face is not perpendicular to the pump shaft.
- The pump has bearing end play. This means that the shaft moves back and forth a few thousandths of an inch at frequent but random intervals.
- There is some impeller unbalance causing shaft whip.
- The pump is operated away from its BEP, causing side loads on the shaft.
- There is thermal shaft growth and pump vibration that affects the seal.

1.7.2 Thermal Degradation:

The rings are the one part of a mechanical seal that are sensitive to heat because of the way they are manufactured, when the two materials in contact with high pressure action upon operating condition the dissipated power due to viscous friction and asperities contacts in the sealing interface leads to a significant increase in temperature. Excessive heat can cause thermal distortions on the seal face and accelerate wear, and thus increase the leak path.

1.7.3 Improper Materials:

Improper materials are nothing but chosen hard materials can make some failures like,

- Some wear is inevitable while operating two rings,
- Friction between two rings is high
- The metal seal components may corrode, crack, fatigue failures may occur.
- High face temperature rise

1.7.4 Incorrectly Installed:

Many mechanical seals fail at initial start-up or prematurely because they were not installed correctly. Cartridge seals eliminate all measurement, protect the seal faces from contamination and are easy to install. With these seals, installation problem is minimized.

The outside seal is preset and requires no installation measurement. Only in-line seals require careful measurement to insure correct installation. By following the mechanical seals installation instructions, step-by-step correct seal installation is easily achieved.

1.8 APPLICATIONS

- Aerospace industries for motors and engines in rockets and turbojets.
- Water turbines, steam turbines, boiler feed pumps and nuclear reactor cooling pumps.
- Water pumps used in irrigation, pumping of fertilizers and insecticides.
- Swimming pool pumps and garbage disposal pumps, dishwashers and washing machines use small sealed pumps.
- The petroleum, chemical, textile and drug industry use mechanical seal pumps extensively in their respective processes.
- All automotive engines, compressor air conditioners.
- Ship propeller shafts as well as its auxiliary equipment in shipyards.

1.9 NEED FOR RESEARCH ANDDEVELOPMENT

The research and development for mechanical seals leads to enhancing the standard of the sealing technology in a various application of pumping systems.

1.9.1 Pollution Control Act:

Environmental Protection Agency (EPA) presently listed 189 substances as toxins under the section 112 of the Clean Air Act. Hence the mechanical seals are required by law to control emissions of toxic products. As per the EPA Regulations, Industries are obligated to design such seals that do not damage the equipment, handle higher pressure or vacuum, provides a wide range of environmental control options and are able to seal a wider range of chemicals. When handling the volatile organic compounds (VOC), seal selection is often determined by the specific gravity and maximum allowable emission levels.

1.9.2 Leakage Loss:

The use of proper mechanical seal not only reduces the electrical energy loss but also serves to save the expensive fluid being leaked from the pump. By taking a volume sample of leakage from a packed pump, measured in time, the amount of lost product each day is calculated. By these calculations, most plants have realized the value of effective mechanical seals over the compression packing. Volume of leakage per minute to per month is listed in the Table 1.2

Table 1.2 Leakage losses (Garlock Sealing Technologies)

S.NO	Loss	One drop per second	Three drop per second
1	1 Minute Loss	1/12 ounce	2 ounces
2	1 Hour Loss	6 ounces	1 gallon
3	1 DAY Loss	1 gallon	24 gallons

4	1 WEEK Loss	8 gallons	175 gallons
5	1 Month Loss	34 gallons	700 gallons

2. LITERATURE SURVEY

[1]. (**Praveen kumar et al. 2015).** The industrialized world uses mechanical seals in variety of applications. Various pieces of rotary or rotating equipment, pumps in particular, depend on mechanical seals to prevent leakage. Familiar rotary equipment devices include automobile water pumps, washing machines, dish washers, compressors, swimming pool pumps and farm service pumps. Mechanical seals are used anywhere that liquid and gases are transferred by rotating equipment.

[2]. (**Kavinprasad, Shankar et al. 2013).** The mechanical seal acts as a check valve and a slider bearing. The major objective of mechanical seal is to prevent liquid under pressure from leaking out of the pump or any other equipment, or from drawing air into the pump when under vacuum conditions. Since the mechanical seal must function as a slider or friction bearing, the mechanical seal has an unpredictable life span. The mechanical seal of a centrifugal pump is usually replaced many times during the life of a pump. All bearings need lubricant and for the application of mechanical seal.

[3]. **(Kong et al., 2012; Sebastiani et al., 2012**). lubricant is the liquid being pumped, sometimes external tailored liquids are used as lubricants, to improve its frictional properties and performance. It is specially designed to be used in centrifugal pumps, mixers, compressors, boiler-feed pumps, agitators and rotary unions. It incorporates various components such as rotating ring (softer material), stationary ring (harder material), and a spring, which ensures the actuation force for proper mating of ring at specific pressure. Over the past two decades, pioneering studies have been made in improving the bulk properties.

[4]. **(Kovalchenko et al., 2011).** Surface sensitive properties of the mechanical seals to enhance its performance and durability. In general, researchers had tried two approaches using surface engineering concepts.

[5]. **(Etsion and Halperin, 2016)** Surface texturing has emerged in the last decade as a viable option of surface engineering resulting in significant improvement in load capacity, wear resistance, frictions efficient etc. of tribological mechanical components. Various techniques can be employed for surface texturing but Laser Surface Texturing (LST) is probably the most advanced so far. LST produces a very large number of micro-dimples on the surface and each of these micro dimples can serve either as a micro-hydrodynamic bearing in cases of full or mixed lubrication, a micro-reservoir for lubricant in cases of starved lubrication conditions, or a micro-trap for wear debris in either lubricated or dry sliding, the state of the art in LST and the potential of this technology in various lubricated applications like mechanical seals.

[6]. (Kovalchenko et al., 2015). The performance and reliability of seals are known to be dependent upon the thermal characteristics of the rotating and stationary rings. A part of the interfacial heat is conducted into the rings. But a major portion of the heat generated at the rings interface is removed by means of convection to the flush fluid that wets their outer surfaces. Also, here a material which is used in the seal faces have important role for tribological inability of the whole system. So, chosen material also should consider the failure of the seal face. When overheating was noticed, under extreme operating conditions, during lab testing of one of Chesterton’s heavy-duty cartridge dual seals, engineers decided to study the situation. The goal was to find design changes that would improve the cooling performance of the seal, thereby broadening its performance envelope. Many researchers have applied advanced experimental and analytical tools to understand and quantify heat transfer behavior of mechanical seals also they found tribological behavior improvements on seal faces.

[7]. (Zhou *et al*., 2017). considered the spiral groove design made seal faces having good heat transferring rate than the flat faces. This design should regard as the annular ring thickness because that the entire groove over the ring thickness make a weakening effect of thermal deformation. Here the spiral groove design parameter such like spiral depth, angle, end radius and groove width optimization having should dependent on maximum bearing force, frictional heat of fluid film and the coupling effect between two seals.

[8]. (Isaev *et al.*, 2010). similarly conducted the experimental with spherical dimple-based design on surface enhancing the heat transfer. The spherical depth has been ranges from 0.13 to 0.26 mm at the corresponding value of based design on surface enhancing the heat transfer. The spherical depth has been ranges from 0.13 to 0.26 mm at the corresponding value of Re range from2*104 to 6*104. Even though this investigation is made with long channel for the temperature 373K, then the air flow can be achieved here for turbulent velocity around the channel. A good confidence in this investigation is the heat transfer rate can be attained without any losses from the material strength. Furthermore, the spherical shape rather than the rectangle section with simple, square section with the dimple and polygon section with the dimple design changes also should considered in the heat transfer rate improvements over the surfaces.

[9]. (Vite *et al.*, 2008). recently expressed with the experimental measurements and numerical means of cylindrically shaped dimples engraved circumferentially on the outside diameter of the mating ring. These designs can be enhancing the reducing a seal's interface temperature, it can work more effectively under severe operating conditions. The results can be validated with CFD analysis.

3. PROBLEM IDENTIFICATION AND METHODOLOGY

Problem-free operation of pumps is an essential requirement for cost-effective and reliable operation of plants and systems. Mechanical seals are used in the pumps to avoid leakages, where the high-pressure steam has great effect with seal rings due to having load effect. Due to improper lubrication between the mating rings, and improper material selection mostly failures accident takes places. Hence this problem was considered for the present study and possible solutions were suggested. Annually more than 40,000 new pumps installed in the process industries and needs to replace the failed units and over 80% of the failures are associated with the problems originated from the mechanical seals. The failure of seal mainly occurs due to improper lubrication, and improper material selection. So, a proper lubrication possibly with design modification and suitable material selection should provide a good result to increase the service life of seals.

3.1 HEAT GENERATION & FRICTION PERFORMANCE

The maximum heat generation depends on the frictional force between the stationary and rotating seals. According to dimensions and materials of seal ring, load and speed values are considered for the application of the pump system. Failure of materials is usually a sign of a discrepancy of material with environment. The consequential construction of seals eliminated major failure of some main component, some concentrate on the effects of environmental attack on sensitive components,

- Acid attack on carbon is directed against the impurities thus, the carbon will appear pitted. The reaction of the impurities to the acid solution causes holes and pits thus, weakening the structure and producing a porous carbon. A higher grade of carbon is required.
- Due to the effect of corrosion by chloride stress on stainless steel the spring can break Many single coil springs driven seals fail because the spring breaks. They are usually inexpensive and over-engineered, but they still fail. Plated seal faces are not corrosion resistant, so the plating material can be removed from the surface.

- Metals corrode. In seals where metal parts are designed to be thin due to flexibility requirements, metal bellows seals, welding techniques used in construction and material compatibility with mating components and pumped fluids are factors that affect the life of a seal.

3.2 METHODOLOGY

The experiment was conducted on the mechanical seal test rig and the microstructure image was captured by using optical microscope. Seal ring was analyzed under various load and constant speed conditions. The methodology for the proposed work is shown in Figure 3.1.

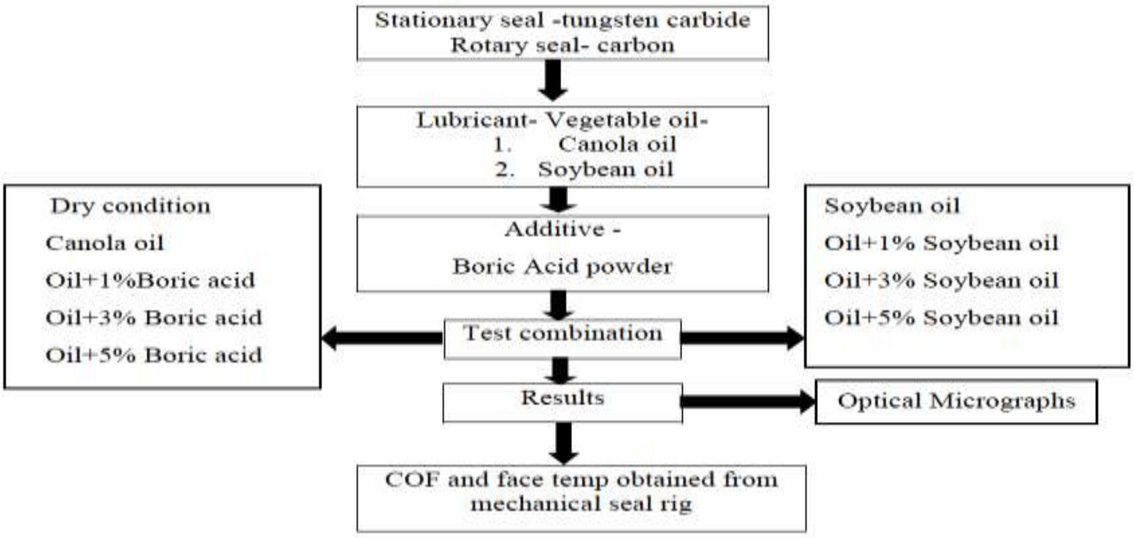
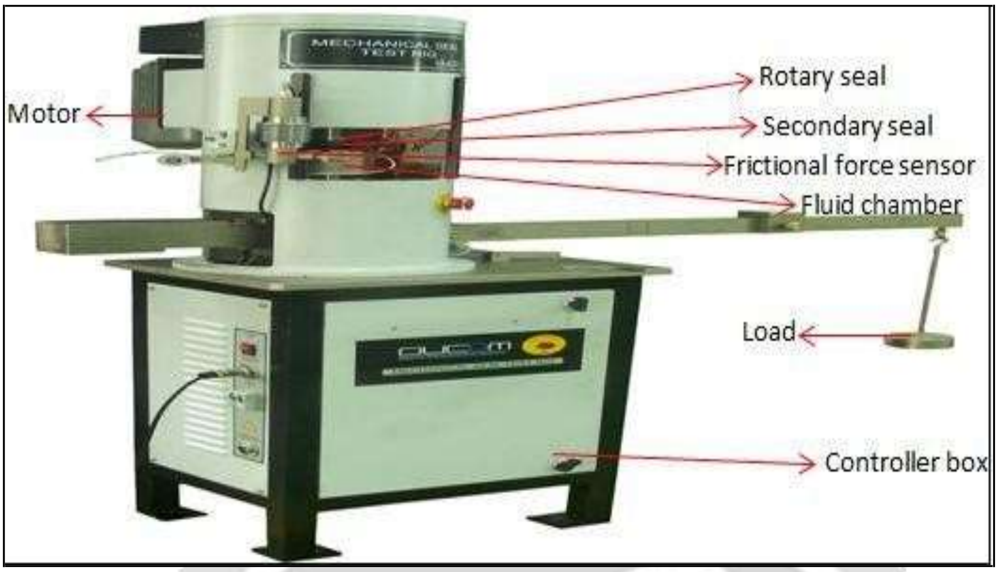


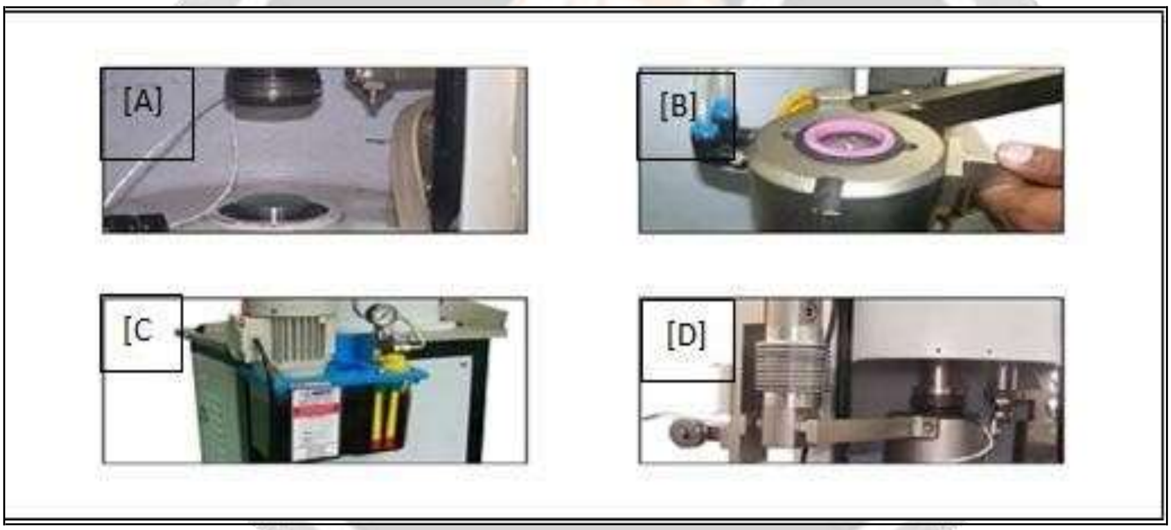
Figure 3.1 Methodology

4. EXPERIMENTAL METHOD

4.1 DESCRIPTION OF EQUIPMENT

The seal test rig shown in Figure 4.1 and 4.2, is a versatile equipment to conduct tests on multiple sizes and different kinds of seal materials, it is a stand- alone unit with holders for seals positioned at acceptable height for altering seals after test and easy handy for operation. It is armed with top holder fitted with rotary seal which made to rotate and press up the hard surface on the stationary seal, the stationary seal is locked in lower holder which press against rotary seal rotated by spindle, for full engagement proper spring pressure was maintained. Rotary seal was basket on spindle for rotating at 3000 rpm. housing is seated vertically on a frame and frame mounted above base plate. Spindle rotated on bearings inside stationary housing, the machine specification is shown in Table 4.1. spindle rotated by help of motor with belt drive, variable frequency drive-controlled speed of motors at all speeds to provide uniform torque. The cylindrical plunger is housed within a LM (linear motion) bush, seated on the frame below spindle in line with its axis, the bottom of plunger rest over a load cell on loading lever, the loading lever is hanging horizontally over base plate at an offset distance to get mechanical advantage for applying load manually. The loading lever swing in vertical axis at hanging position, load is applied by placing dead weights on pan at longer end of lever, shorter end of the lever with load cell pushes plunger vertically upwards while longer end of lever moves downwards to press against the rotary seal. with a force equal to the product of (dead weight & mechanical ratio). two holes are implemented on fluid chamber to inlet and to drain the fluid, for these two ports are provided for connection on the outer diameter of fluid chamber, one port is connected to pump and other to tank for draining fluid. gear pump issued to provide pressure to fluid inlet a lubrication, the lubrication unit is fitted with pressure gauge and pressure relief valve, motor gear pump sucked fluid inside the tank fitted with pressure gauge to supply fluid at a controlled pressure. The pumped fluid enters the chamber through inlet port to rise into the sealing zone through a hole on bottom stationary seal holder and flow out through the other hole with in the bottom specimen holder back into the tank through a throttle valve. The throttle valve is set to discharge drop by drop at 20 drops/sec, because of throttle valve the fluid pumped do not drain, pressure is retained with in contact zone. The free rotation of fluid chamber on plunger axis is arrested by an arm tightened horizontally on the outer diameter of fluid chamber, the free end of arm press on a load cell prevents the rotation of fluid chamber.





[A] Rotary seal clamped on spindle

[B] stationary ring seated on fluid chamber

[C] Lubricating unit

[D] frictional force sensor.

Figure.4.2 Parts of mechanical seal test rig

Table 4.1 Machine specifications

SR. NO	TEST PARAMETERS	SPECIFICATION
1	Speed	Min: 300 rpm, max: 3000 rpm
2	Load	75 -2000N
3	Frictional torque	Max 358Nm

3	Temperature	3 temperature measurements at fluid inlet, within sealed zone and on bottom sample holder.
4	Stationary & rotary seals of pump	Rotary seal = Ø 45x Ø 38x28.5mm. Stationary seal : Ø43x Ø33x8mm (id x od's x thick)
5	Power	230V / 50 Hz / 1 Ph / 1500 VA

4.2 CONTROLLERBOX

When Miniature circuit breaker on the machine panel is switched ON, the input supply is sent to AC drive, Instrumentation transformer shown in Figure 4.3. The transformer step downs to 15-0-15V and 0-10V, both these voltages are sent to data acquisition card (I-DAS) for rectification and the regulated output is sent to sensors shown in Figure 4.4. Depending on the input value of speed on pc screen, a corresponding voltage is sent to motor for rotation on clicking RUN icon on software. One beam type load cell of capacity 1100 kg is seated inside the loading arm firmly and over it a roller arrangement is fixed to exert pressure on the plunger in line contact. When dead weights are placed on loading pan for applying normal load, the loading arm being hinged moves down lifting the load cell portion of lever arm vertically upwards. This movement lifts the plunger & ball pot till it touches top ball, and further movement exerts pressure on the contact area. This exerted pressure is measured by the load cell and signal sent to controller for processing and display. To measure the temperature of inlet fluid with in contact zone Three resistance temperature detectors (RTD) sensors were used; it is mounted at bottom part of fluid chamber, the output of sensor is processed inside I-DAS card and displayed on software screen. Second RTD sensor is seated on the side of fluid chamber with the tip touching the outer diameter of stationary seal, the seal temperature measured is displayed on screen, to measure the fluid inlet temperature Third RTD sensor is fixed to the regulator of lubrication unit and measured temperature is displayed on the screen.

[1] I-DAS card [2] line filter [3] transformer [4] AC drive

Figure.4.3 Controller box

To measure spindle speed the proximity sensor is used, the least count of measurement is 1rpm, and for this an rpm sensor disc (indexing plate) with slots on circumference is fixed to top of spindle and rotates along with it. Proximity sensor is seated perpendicular to it on a bracket with equal gap for all slots on disc circumference; signal is generated when sensor disc approaches the active surface with in the specified switching distance. This sensor functions in contact less fashion and do not require any sensing mechanism

[1] Proximity sensor [2] sensor disc

Figure 4.4 View of proximity sensor.

Frictional torque is measured by a 200 kg beam type load cell fitted at a distance of 179 mm from the center of spindle, the load cell is fixed vertically on a bracket, with the load cell button engaging the ball pot handle. The handle side of fluid chamber is fitted with a taper bush to press the load cell button when test started; the force applied is measured as frictional force and converted to frictional torque by multiplying by 0.179. An inductive proximity sensor is selected as it has excellent means of detecting the presence of a wide range of metallic targets. This detection is accomplished without contacting target and is mechanically wear free. It is comprised of a high frequency oscillator circuit followed by level detector, a post amplification signal circuit and drives a buffered solid- state output. When the sensor disc is brought within effective range of the emitted field of the oscillator, a damping action results which reduces the amplitude of oscillator. This amplitude shift is converted to digital signal by the level detector, which drives a buffer stage. When the object is removed, the oscillator and digital output is turned to its former state.

The mechanical seal test rig is incorporated with the data acquisition card (I-DAS), which does the sampling of signals obtained through the above-mentioned sensors, and thus converts into digital numerical values which can be manipulated by the computer and the developed output screen is shown in Figure 4.5. The data acquisition card was controlled by the software WINDUCOM 2010, which was development by Ducom Instruments (INDIA).

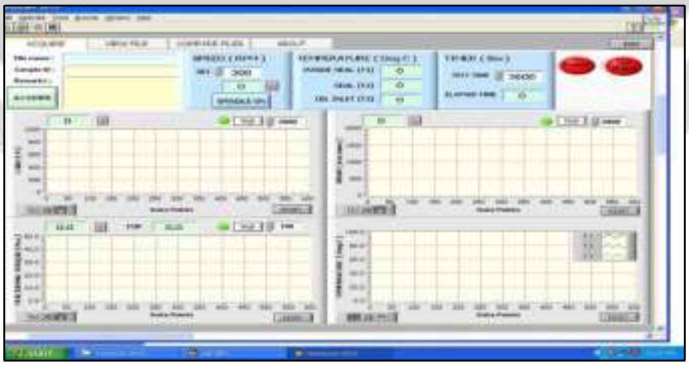


Figure.4.5 WINDUCOM 2010 software.

4.3 DESCRIPTION OF SEALRING

4.3.1 Tungsten Carbide:

Tungsten carbide (WC) is the designation of the type of hard metals based on a hard tungsten carbide phase and usually a softer metallic binder phase. The correct technical term of tungsten carbide is “cemented tungsten carbide”. However, the abbreviated term “tungsten carbide” is often used for convenience, “cemented” being understood. The hardness of WC is below that of most ceramics, whereas the wear resistance of the material is high, mainly due to its high toughness. Cobalt-bonded (Co) WC is only corrosion-resistant in water if the pump is made of a non-inert material such as cast iron. The corrosion resistance of some chromium-nickelmolybdenum-bonded WC types is similar to stainless steel EN 1.4401 (AISI 316). WC with less than 0.5 % binder phase has the highest resistance to corrosion, although the material is not resistant in media such as water containing hypochlorite. Mechanical seal has rotary and stationary seal; the rotary seal is carbon at inner diameter and the rubber portion. The rubber portion engaging the metal surface has od×id= Ø 45× Ø 38 total height of rotary seal including spring 28.5mm. The stationary seal is made of three kind of material tungsten carbide, ring has Ø 43× Ø 33× 8mm thick seal inside the step bore on cylindrical rubber outer cover having size Ø 52× Ø 33×10mm thickness. Figure 4.6 shows the machined seal rings.

[A] Tungsten carbide seal [B] resin impregnated carbon seal

Figure 4.6 Seal rings

4.4 LUBRICANT PROPERTIES

4.4.1 Properties of Canola Oil

Canola oils are becoming more attractive in a wide range of engineering applications (Deshmukh, et al). Organic-based oils are studied due to their potential for lubrication. But thermal and oxidation characteristics of these lubricants are quite limited. Canola oil with the properties are given in table 4.2

Table 4.2 Properties of canola oil

Properties	**Value**
Dynamic viscosity at 40^0C (mm^2/s)	33
Dynamic viscosity at 100^0C (mm^2/s)	7.34
Viscosity index (VI)	158
Density (g/cc)	0.914
Flash point (^{0}C)	275
Pour point (^{0}C)	-18

4.4.2 Properties of Soybean Oil

Table 4.3 shows the major properties of the Soybean oil which is used as an industrial lubricant to reduce wear and friction coefficient

Table 4.3 Properties of soybean oil

Properties	Value
Dynamic viscosity at 40^0C (mm^2/s)	38.6
Dynamic viscosity at 100^0C (mm^2/s)	8.52
Viscosity index (VI)	160
Density (g/cc)	0.934
Flash point (^{0}C)	324
Pour point (^{0}C)	-16

4.5 LUBRICANT ADDITIVE PROPERTIES

4.5.1 Properties Boric acid powder

Boric acid is the common term for orthoboric (or boracic) acid $H3BO3$, which is a hydrate of boric oxide $B2\ O3$. When in contact with water, boric oxide will readily hydrate, converting to boric acid. Boric acid is a weekly acidic white powder that is soluble in water (about 27% by weight in boiling water and about 6% at room temperature), soft, ductile, stable, free flowing, and easily handled. Finely ground technical grade boric acid powder (>99% pure) is readily available.

The Environmental Protection Agency has established that boric acid is benign and it is not classified as a pollutant under the Clean Water Act. Material safety data sheets for boric acid show no serious illnesses or carcinogenic effects from exposure to solutions or aerosols. The consumption of boric acid and boric oxide is distributed among glass making (78%), fire retardant (9%), agricultural fertilizer (4%), and industrial applications such as metal plating and finishing, paints and pigments, electroplating, and cosmetics (9%). (A dilute water solution of boric acid is also commonly used as a mild antiseptic and eyewash.

Boric acid powder is a viable technique for providing in situ lubrication for concentrated metal contacts. This technique can reliably produce friction coefficients less than μ=0.1 and can reduce wear rates by 100 times or more. The major properties of Boric acid powder are shown in Table4.4.

Table 4.4 Properties of acrylamide powder

Properties	Value
Molecular Weight	61.83
Melting Point	170^oC
Boiling Point	300^oC
Water Solubility	4.72 g/100ml at 20^oC
Density g/cm^3	1.435
Flash Point	Not flammable

4.6 MIXING OF LUBRICANTS

After investigation the frictional characteristics of each individual lubricant, the one which shown the better frictional performance is selected, and premixed with 1%, 3% and 5% of Boric acid powder individually, heated to 45°C and mixed using magnetic stirrer Fig 4.7 for 30min until the Boric acid powder is evenly spread throughout the lubricant and to interfere complete soluble.

Fig 4.7 Magnetic Stirrer

The kinematic viscosity of each lubricant was measured at 45°C and 100°C using say bolt viscometer Shown in Fig 4.8 (efflux cup viscometers).

Fig 4.8 Say bolt viscometer

4.7 TESTING PROCEDURE

The entire tests were carried out on a mechanical seal test rig, shown in Figure 4.1. The stationary ring was mounted in fluid chamber that allows the self-alignment of the ring and also permits axial loading of the stationary ring against the upper rotating ring. Initial load of 100N was applied at the end of the lever for the full engagement of the stationary and rotary ring. Each test includes a 3000s running-in period with a constant speed of 1500rpm for six loading conditions. After running-in, the load between the rings was varied from 100N to 500N (in a step of 100N, 200N, 300N, 4000N, 500N) (Engqvist et al., 2000). For each loading, the running-in period was 600s. After each loading conditions, the test was halted for 5min. The seal face temperature; friction torque and coefficient of friction were recorded for every 1s using WINDUCOM. The same procedure was repeated for all the stationary rings tungsten carbide. For the entire testing, resin impregnated carbon was used as rotary ring.

5. RESULTS AND DISCUSSION

5.1 FRICTION COEFFICIENT

5.1.1 Under Dry Running-in Test

Tests were repeated with three samples and computed average friction coefficients are shown in Figure 5.1. For each loading conditions, the coefficient of friction attains its stable value approximately after 200s, which was found to be in agreement with the previous studies. In case of dry running-in test, for tungsten carbide seal, the friction at the initial stage (at 100N load) of running-in was low and remains constant

(μ=0.134). for this low friction the responsible factors may be due to the presence of foreign materials on DLC coated seal layer such as (i) oxides (ii) moisture of metals and so on (Al-Samarai et al., 2014).
So there may be little or minimum metallic contact at the junction and also the oxide film had lower shear strength resulting in low coefficient of friction. After initial running, the surface layer crack and clean surface comes in contact, resulting the increases the value of frictional coefficient due to the inclusion of wear trapped particles resulted with the increase in roughness, face temperature and friction force. Hence the friction coefficient was found to be fluctuating and after certain duration of running-in approximately after 200s, the friction coefficient value attains a steady state. The average of those friction coefficients (μ=0.172) at the load of 100N had been calculated and shown in Figure 5.1. The same aspects were found for another loading conditions. The average friction coefficient at each individual load of 200N, 300N, 400N, and 500N was computed and shown in Figure 5.1. The microstructure of the sliding surface of the tungsten carbide and resin impregnated carbon under dry running-in test shown in Figure 5.6(a) and wear scar was observed.

5.1.2 Under canola oil test condition:

In unlubricated case, a series of experiments were conducted with canola oil as a base lubricant. When the lubricant was tested the friction, coefficient was found to be increase from 0.099 to 0.123 over the sliding range tested. Fig 5.1 represents the

variation of friction Coefficient. To reduce the friction Canola oil is used by the formation of temporary oil film with few molecules thickness. Oil film forms on the mating surface, due to the effect of applied load and temperature the oil film can be observed broken during working with canola.

A combination of the lubricant of canola oil and boric acid powder, the motivation of this combination of lubricant mainly focused on eco-friendly nature. Canola oil with Boric acid additive creates an oil film by penetrates through the surface cracks on the contact surface that is the reason to reduce the friction coefficient by the effect of atomic layers under shear loading (Duzcukoglu, et al 2010). So, the film maintains friction coefficient about 0.129 to 0.089, 0.18 to 0.078 and 0.17 to 0.074 Boric acid powder mixed with canola oil in the percentage of 1, 3 and 5 consecutively. Fig 5.1 shows variation of friction coefficient, when used 1% of boric acid with canola oil that can able to form a oil film on a metal surface, although initial friction is low compare to percentage of 3, 5. But when the final friction value is comparatively high, this shows the weak oil film formation on metal surface. When the boric acid powder mixed with canola oil with the percentage of 3, 5 did not degrade over entire range of sliding experiments. Initially both (3%, 5%) friction value is high this clearly shows the oil film could not able to form on the sliding surface at starting stage approximately after 300sec both 3%and 5% can able to form a strong protective oil film on the sliding surface (Brizmer et.al 2012), and once the film form on the sliding surface that will not degrade over entire range of time. After running in test for 3000s the surface roughness of WC seal found to be Ra=0.016to0.068μm, Ra=0.016 to 0.051 and Ra=0.016 to 0.048 respectively. The microstructure of the sliding surface of the tungsten carbide and resin impregnated carbon under Boric acid mixed with canola oil running-in test shown in Figure 5.6 (b) and wear scar was found.

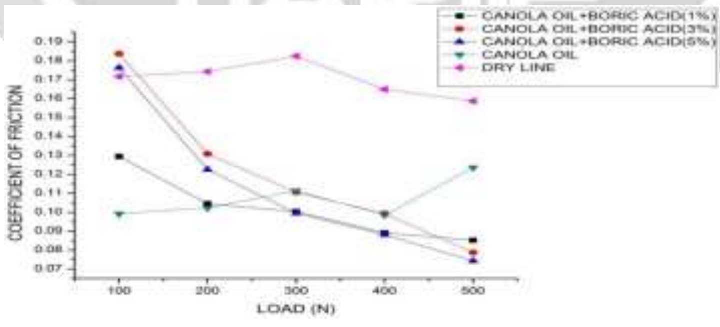


Figure 5.1 Variations of friction coefficient for dry and oil lubrication

5.1.3 Under soybean oil test condition

When soybean oil taken in to account the friction coefficient value comes between the range of 0.11 to 0.087. The rate of decrease and increase in Friction coefficient is largely influenced by oil structure and their ability to form thin lubricating film at the point of metal contact. Fig 5.2 shows the soybean oil continuously breakdown because of friction chemical process (Engqvist, H., et al 2000). Oil concentration level is very crucial consideration of friction. In order to increase the concentration level of soybean oil, boric acid powder combined with oil at the percentage of 1, 3, and 5 (Jianxin, D., et al 2012). According to the percentage of 1, For the Initial load of 100N the friction coefficient attains it stable value of $\mu=0.0953$ approximately after 300s the friction coefficient increased up to 0.1001. After the load 300N the friction coefficient is about 0.0903, and 400N 0.0806 and 500N 0.0764 this decrease of friction coefficient shows the soybean oil and the additive of boric acid powder has the ability to form a strong friction chemical film on the metal surface this friction chemical film is the reason for lowering the friction coefficient. When boric acid powder mixed with soybean oil with the percentage of 3, 5 the initial friction coefficient $\mu=0.0729$ and $\mu=0.0655$ according to the frictional chemical film formation on the metal surface the friction coefficient is gradually decrease in terms of $\mu=0.0803$, $\mu=0.0701$, $\mu=0.0694$, $\mu=0.0716$ and $\mu=0.0793$, $\mu=0.0734$, $\mu=0.0676$, $\mu=0.0611$ respectively. Compare to 1% of boric acid powder 3% and 5% friction coefficient value is low because of the oil concentration. Fig9 shows the frictional film formation on the sliding surface. After running in test for 3000s the surface roughness of WC seal ring found to be Ra =0.016 to 0.063 µm, Ra=0.016 to 0.044 µm and Ra=0.16 to 0.041 µm respectively. The microstructure of the sliding surface of the tungsten carbide and resin impregnated carbon under Boric acid powder mixed with oil running-in test shown in Figure 5.6(c) and frictional film was found and thereby no traces of wear scar were observed.

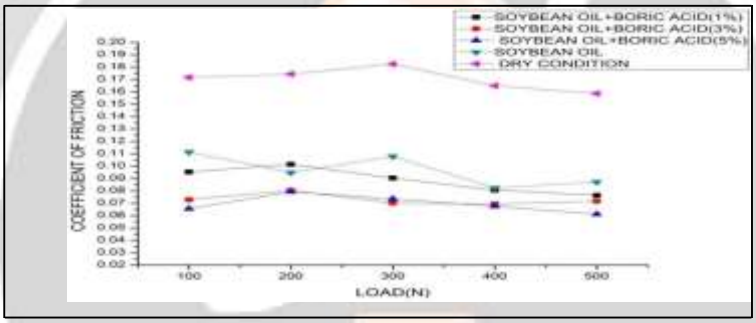


Figure 5.2 Variations of friction coefficient for Boric acid powder mixed with Soybean oil lubrication

5.2 FRICTION TORQUE

Fig 5.3 & 5.4 shows a frictional torque comparison between the canola oil and boric acid powder with canola oil in combination, soybean oil and boric acid powder mixed with soybean oil with the percentage of 1, 3 and 5. Friction coefficient has direct dependence of the frictional torque. The increase in surface roughness due to detachment of wear debris increases resulting in, by which the sliding resistance of the mating surface also increases. So hence the increase in normal load (N) will also increase the frictional torque which was confirmed from the Fig 5.3. Among the two lubricants (WC seal against resin impregnated carbon) soybean oil exhibits lowest friction torque

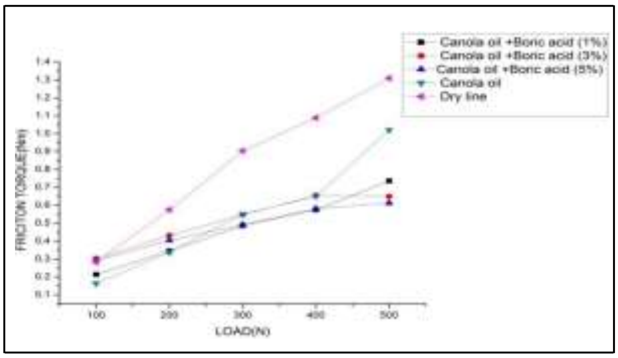


Figure 5.3 Variations of friction torque for dry and Canola oil lubrication

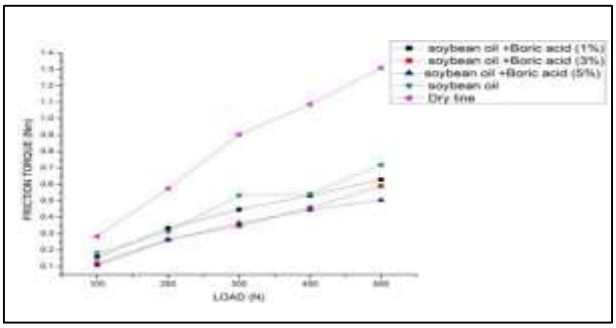


Figure 5.4 Variations of friction torque for dry and Soybean oil lubrication

5.2 FACE TEMPERATURERISE

The face temperature increase of the mechanical was the important parameter to be examined to investigate the frictional characteristics between the seals. In general, the face temperature rise was commonly observed in all two lubricants. Fig 5.5 & 5.6 shows the face temperature increase for two lubricants. Among which soybean oil mixed with boric acid powder with percentage of 5, exhibited the lowest face temperature increase of 89^{o}C. Whereas 1%, 3% of boric acid powder mixed with soybean oil exhibited 96^{o}C, 95^{o}C. Soybean oil alone exhibited 110^{o}C and dry condition 123^{o}C. In case of canola oil mixed with boric acid powder with the percentage of 1, 3 and 5 exhibited 105^{o}C, 103^{o}C, 92^{o}C respectively, and canola oil alone exhibited 114^{o}C.

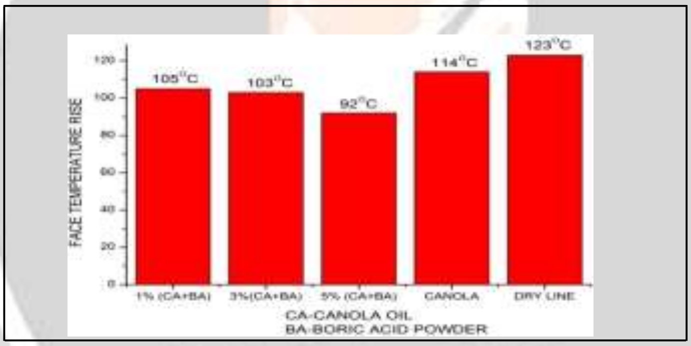


Fig 5.5 Variation of face temperature increase for Canola oil

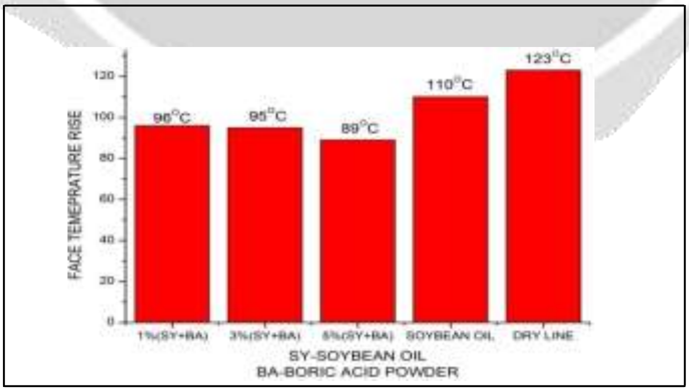


Fig 5.6 Variation of face temperature increase for Soybean oil

6. CONCLUSIONS

In this present study fundamental friction and interface temperature of carbon sliding against tungsten carbide were evaluated using the mechanical seal test rig and the results were summarized as follows

- From the above observed results, under dry running-in test condition, the life of the seal reduces due to high friction coefficient produced by seals, high friction torque and high face temperature.
- canola oil and soybean oil running-in test, there was a decrease in the friction coefficient, friction torque and face temperature than dry condition.
- Under 1wt%, 3wt%, 5wt% of boric acid powder individually mixed with canola oil and soybean oil as lubricants, Soybean oil mixed with 5wt% of Boric acid powder had exhibited the better frictional coefficient in the range of μ= 0.06 –0.05.
- The molecular structure of boric acid allows it to act as effective solid lubricant film. When crystallized, boric acid forms a weak Vander walls bonds between individual layers and strong hydrogen (covalent) bonds within a layer. Such a bonding structure makes the structural properties of boric acid highly anisotropic.
- When tangentially loaded, the individual lamellae slide relatively easily over one another. This was in contrast to the normal direction where the boric acid had a relatively high loaded carrying capacity. Hence, when properly aligned with a substrate, boric acid will exhibit minimal friction and provide effective separation between the surfaces

7. Future Scope

- Alternate composite material can be used as a stationary seal, which can increase the service life of the seal, reduce the cost of the seal and reduce the leakages drastically.
- Different lubricants in different speed variable conditions with different concentrations and combinations with eco-friendly behavior reduces the failures and increases the productivity efficiently.
- Design optimization of spring and drive mechanism can also improve the service life of the seal.
- Tribological behavior of PVD and CVD coated seals, composite seals can be studied by developing various eco-friendly lubricants.

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