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A review of mechanical seals tribology and condition monitoring

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ABSTRACT

Mechanical seals have become one of the most popular sealing systems for rotating machinery because of low leakage and absence of a requirement for routine maintenance. Generally, a mechanical face seal should operate with a fluid film as thin as possible, to reduce the leakage and to restrict friction and wear. Recent advances in a system of computer software based on finite element modelling and analytical approaches help in understanding of the working conditions of the mechanical face seals. This paper reviews tribological bahavior and condition monitoring of mechanical seals based on the literature of the recent years. It covers friction, wear and thermal characteristics of mechanical seals and the application of computational methods and other techniques to give good understanding of the tribological behavior and condition monitoring of seal faces.

Keywords: Mechanical seal, tribological behavior, condition monitoring.

Introduction

Mechanical seals are leakage control devices, which are found on rotating equipment such as pumps, propeller shafts in ships and submarines, compressors, for conditioners, of cars and turbo jet engines and liquid propellant rocket motors in aerospace industry to prevent, or reduce to a minimum acceptable level leak of gas or liquid from between component surfaces. Mechanical seals also prevent dirt from entering through those surfaces. The seal is made between the very smooth, very flat faces of two rings, one is attached to and rotates with the shaft, and the other is attached to the housing and is stationary. A schematic illustration of the components of a conventional mechanical seal is shown in Fig.1. Due to increasing technical requirements, mechanical seals are widely used under various operating conditions. Face seals have to operate at higher pressures and higher speeds, so the failure of mechanical seals may be attributed to various factors. Typical failure conditions of seals are shown in Fig.2.

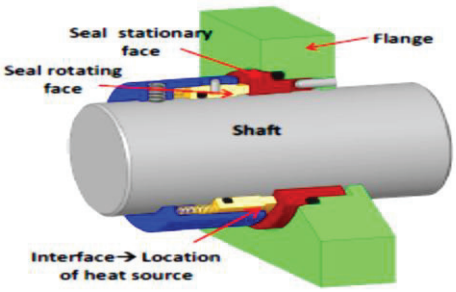


Fig.1.Two primary parts of a mechanical seal [1]

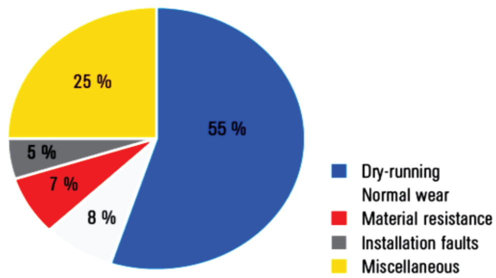
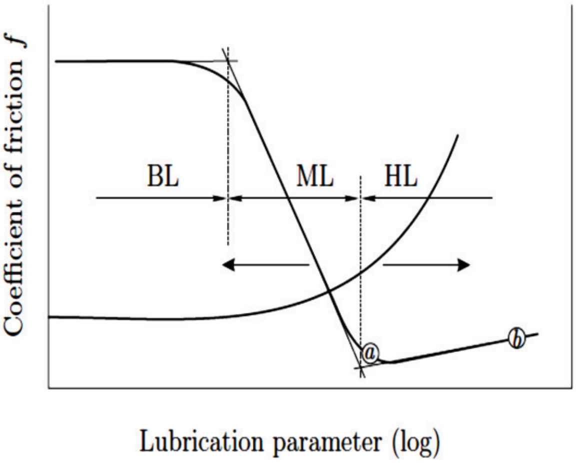


Fig.2. Typical failure conditions of seals are [2]

Although the production costs of seals are low, any failure of these components can damage the hydraulic system heavily and often leads to excessive leakage of fluid to the environment [3]. For this purpose, a lot of researches have been reported so far, this paper deals with review of tribological behavior, Finite Element Method (FEM) analysis and Condition monitoring (CM) of mechanical seals.

Secction1: Tribological behavior of seals

As can be seen in Fig. 2, the most important reason for mechanical seal failures is related to tribology of the contact area. Therefore, a reliable understanding of tribological behavior of the sealing system is necessary to assure safe usage during all operating states of hydraulic components. Tribology is defined as the science of interacting surfaces in relative motion dealing with the friction and wear occurring between components. To keep leakage as low as possible, the seal gap must be very small. As a result, the lubricating film is very thin. Consequently, the seal face materials must be able to withstand rubbing against each other at high load and speed. The materials choice for mechanical seals is highly depends on the working conditions, however, from the analyses conducted by Popa and Onescu [4], the graphite based on materials and carbides are almost 60% and 20 % of the cases respectively. G.A. Jones [5], carried out a careful investigation on the tribological behaviour of mechanical seal face materials in dry line contact and showed that because of its direct relation to wear, the *PV* (pressure×velocity) value at the interactive surfaces is crucial in determining the performance of a mechanical seal. He concluded that the *PV* capability of face materials is dependent on the development of a carbon graphite contact film. Qiu [6] studied the tribological properties of face seal material coatings with different metal Ti components on disks of same material under dry running condition. He concluded that the coating worn out quickly under unlubricated condition and the wear mode of the coatings is thought to be a combination of adhesive wear and abrasive wear. He also developed a finite element model to evaluate the heat transfer characteristic of the seal prototype. From the above discussion it can be concluded that in order to control friction and wear, seal faces can be lubricated. If the lubricant is not able to separate the surfaces, contact between the surfaces takes place at asperity level. However, a too thick fluid film is unfavourable with regard to leakage, as this is proportional to the cube of the film thickness [7]. In Fig.3 the coefficient of friction is schematically plotted as a function of a lubrication parameter (The lubrication parameter is defined in many ways in the literature) which yields the generalized Stribeck curve. Fig.3 also shows the separation *h*. In this graph, three lubrication regimes can be distinguished, i.e. Hydrodynamic Lubrication (HL), Mixed Lubrication (ML) and Boundary Lubrication (BL). The different lubrication regimes are schematically represented in Fig.4.



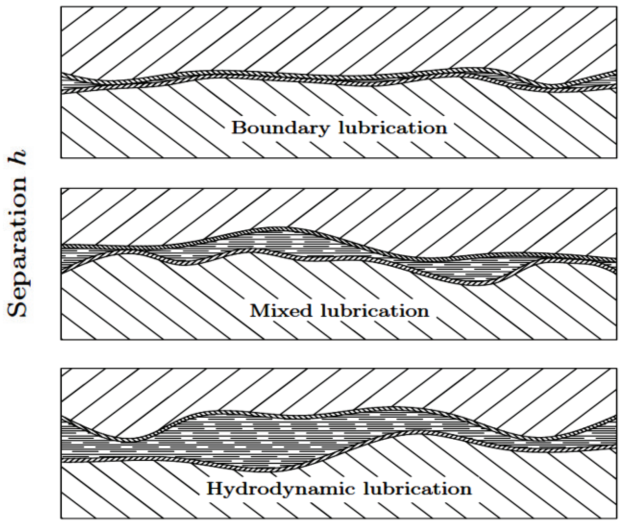
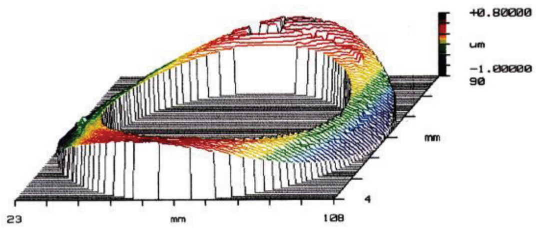


Fig.3. Generalized Stribeck curve [8]

Fig. 4. Lubrication modes [8].

For mechanical face seals an optimum operational region would be around the transition from hydrodynamic to mixed lubrication, indicated by position *a* (see Fig.3). In this region a low coefficient of friction is accompanied by a low wear rate and a low leakage, as the separation is rather small. Position *b*, where face seals may operate as well, will also show a low coefficient of friction and hardly any wear as the faces are fully separated by a fluid film. However, as shown in the graph, position *b* is accompanied by a much larger separation and hence, a large leakage. In the work of H. Lubbinge [8], an isothermal model was developed which predicts the frictional behaviour of mechanical face seals as a function of the operational conditions. He designed and built a test rig in order to verify the theoretical model. Lapping and grinding techniques were applied to make different waviness amplitude and coining angle in seals as shown in Figs.5 and 6.



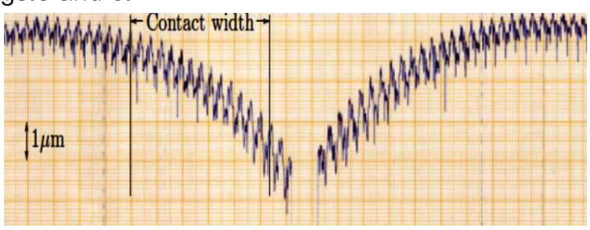
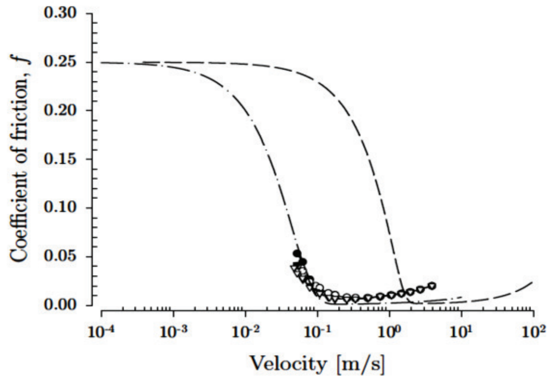


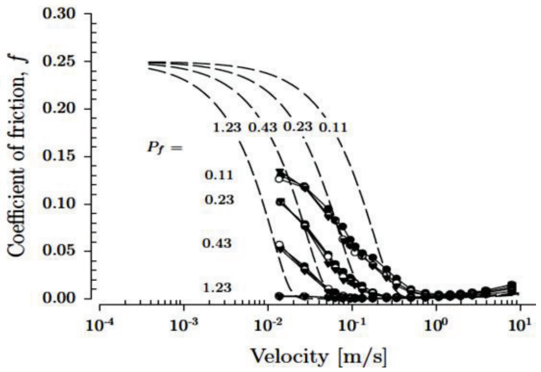
Fig.5. Lapped waviness on the face seal [8]

Fig. 6. Coning applied on the face seal [8]

Based on the work of H. Lubbinge, obtained Stribeck curves with coning and fluid pressure vs. calculated Stribeck curves are presented in Figs 7 (a) and (b) respectively.



(a) Curves with coining



(b) curves with fluid pressure

Fig.7. Measured Stribeck curves vs. calculated Stribeck curves (dashed line and dash-dot line) as a function of the velocity [8].

Secction2: Finite Element Method (FEM) analysis

Prediction of mechanical seal performance has been proved to be a great challenge for many years. It is quite reasonable to say that the mechanical seal is one of the most unpredictable machine components [9]. Computer software based on FEM helps in understanding of mechanical face seal performance. The history of development of analytical tools for mechanical seal design at John Crane EMA can be traced back as early as 1968 [9]. To date most published work, Minet et.al [10] presented a numerical deterministic study of mechanical seals operating in mixed lubrication regime. They have investigated the influence of working conditions in terms of friction and surface distance for different surfaces. The geometrical configuration of their work and their Measured Stribeck curves are shown in Figs 8 and 9 respectively.

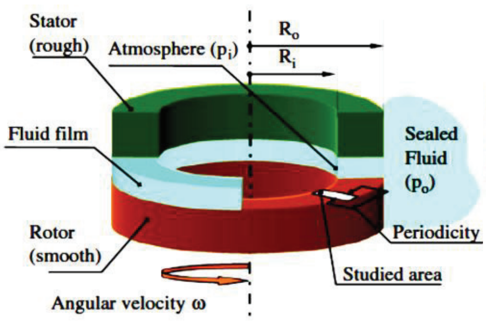
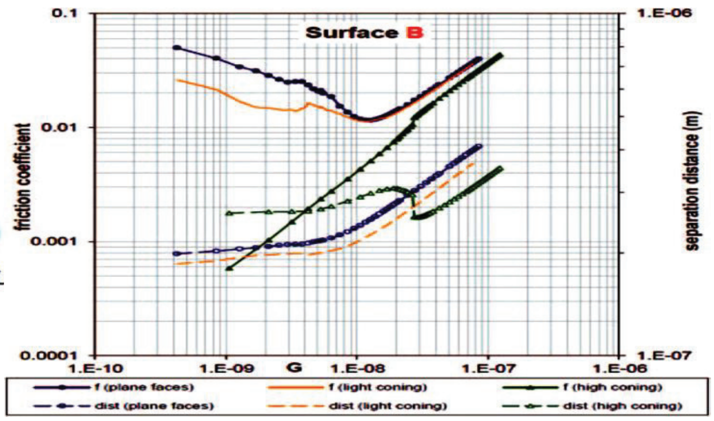


Fig .8. Model description [10].



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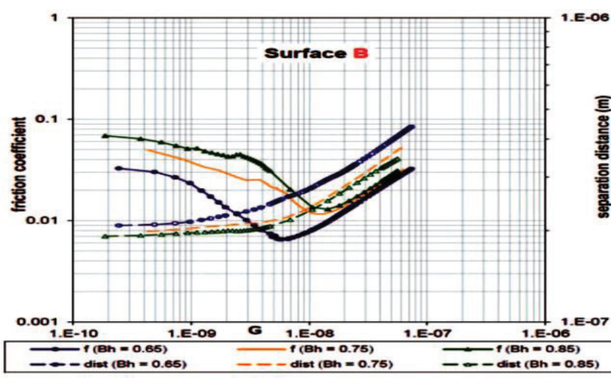
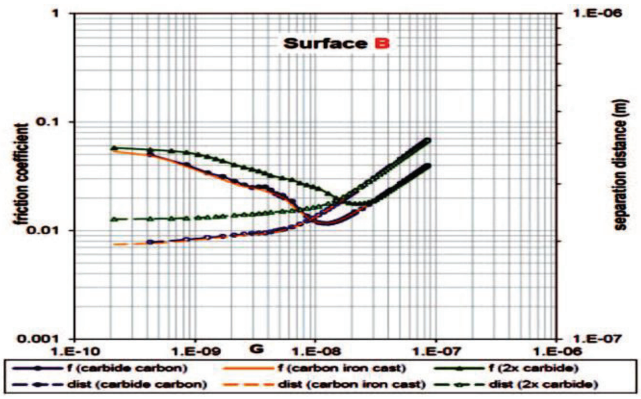
(a) For different coning angle values



(b) For different balance ratio values



(C) For different face materials

Fig.9. Friction and average face distance versus the duty parameter, G [10].

As can be seen in Fig.9 (a), the effect of coining angle has not a general trend. It can be explained by additional hydrostatic load which is generated by coning angle and provides a significant friction reduction at low values of the duty parameter. At higher speeds, an interaction between the asperities induced hydrodynamic pressure and the hydrostatic effect lead to more complex behavior which is not considered in the work of H. Lubbinge [8]. The results of this study are illustrated in table1 briefly.

table1

	Balance ratio↑	Feeding pressure↑	Material hardness↓	Conning angle↑	Surface roughness↑
Film Thickness	↓	↓	↓	Not clear	↑
Friction	↑	↓	↓	↓	↑

Sticlaru [11] presented a FEM analysis for different temperature and fluid pressure. He defined a frictional interaction between the two seals and then modelled the lubrication regime in the contact region with different values of frictional coefficient. The mesh configuration in his is presented in Fig.10.

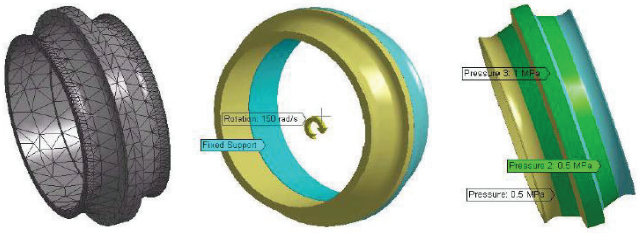
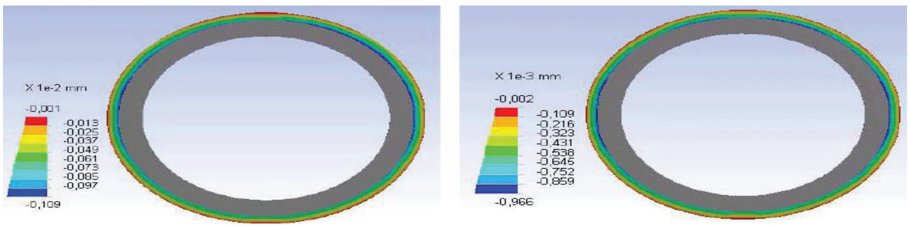
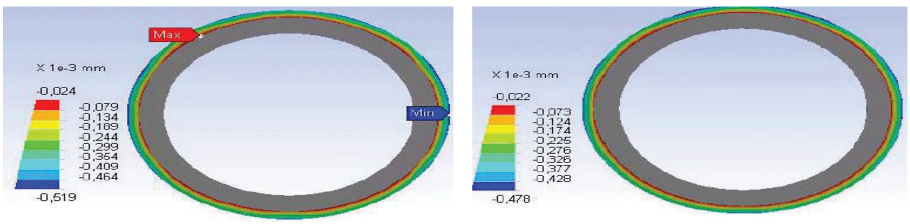


Fig.10. Mesh, Constrains and External Loads [11]

The simulation results of this work for contact gap between the stationary seal and rotating one are shown in Figs. 11 and 12. It can be seen that all gap contact values are negative, demonstrating the presence of penetration, but the results values are very small. The maximum penetration occurs at the inner side of the contact region at higher temperature, while at lower one it appears on the outer side of the contact region, and, in condition of a dry regime, the penetration increases, which develops wear in the contact surface[11].



(a) Coefficient of Friction f = 0.7; (b) Coefficient of Friction f = 0.2;
Fig.11. Gap at t = 125º C [11]



(a) Coefficient of Friction f = 0.7 (b) Coefficient of Friction f = 0.2
Fig. 12. Gap at t = −25º C [11]

Secction3: Condition monitoring of seal faces

As shown in Fig.2, for most mechanical seal failures, a precursor is the collapse of the lubricating film between the rotating and stationary ring. Several attempts have been made to monitor the condition of mechanical seals using Acoustic Emission (AE) technique. This technique is based on observation the relative sliding motion of contacting asperities leads to generation of AE signals. Interested readers may refer to work of Fan et.al for a review of condition monitoring of mechanical seals using AE technique [12]. Since acoustic methods are sub-optimal due to external noise in industrial environments, such a technique has had varying degrees of success in controlled laboratory tests and in a limited field trial. In this regard, Fan designed and built a test rig employed the industrial products and was able to simulate the realistic operation of mechanical seals under different working conditions [13]. Based on the literature, the main problem with the technique is the difficulty in distinguishing AE generated by the seal from emissions by other sources and from background noise (e.g. from motors). Fan applied advanced signal processing methods to reduce the effect of background noises. He reported that using advanced signal processing methods such as time-frequency analysis, the background noise never has been observed as a problem. The relationship between AE RMS value and sliding speed achieved by Fan and a Comparison the AE signals between the static experiment and running experiment in the time frequency domain are shown in Figs. 13 and14 respectively.

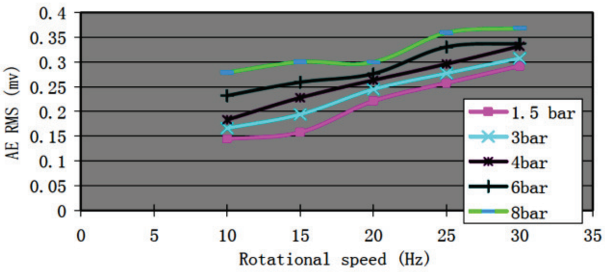


Figure.13. AE RMS value vs. sliding speed [13]

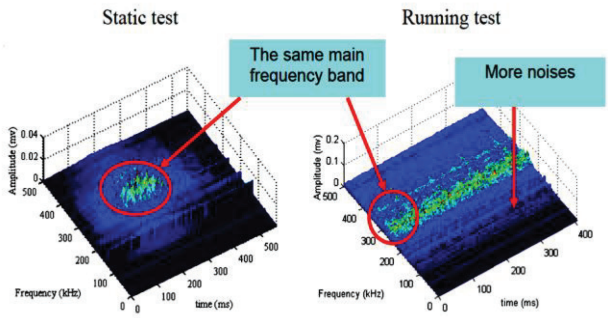


Figure.14.T-F analysis of AE signal [13]

Since wired sensors are difficult to use in small spaces that exist in mechanical seal faces, attempts were made to develop wireless sensors for temperature monitoring of face seals. Lokesh et.al [1] introduced the first RF temperature sensor for condition monitoring of mechanical face seals. The sensor comprised a resonant inductor-capacitor circuit that was inductively coupled to a separate interrogating coil as shown in Fig.13. The resonant circuit was mounted on the stationary face of the mechanical seal. A temperature rise was sensed as a resonant frequency increase, caused by a decrease in the sensor capacitance value. Using Agilent Advanced design system (ADS), they simulated the circuit shown in Fig.15. As can be seen in Fig.16, a good agreement was observed between simulated and measured resonant frequencies at low temperature.

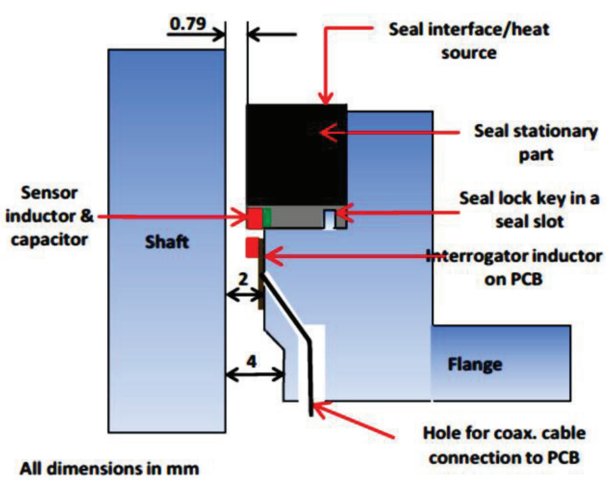


Fig.15. Simplified cross section diagram [1]

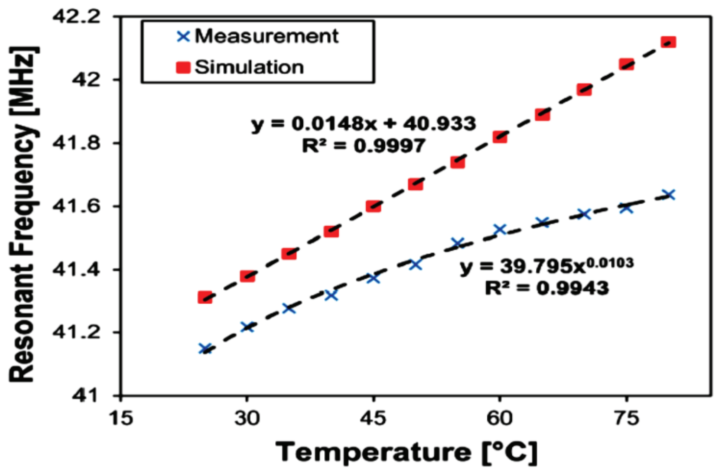


Fig.16. Resonant frequency vs. temperature [1]

Conclusion

This article presents a review on the tribological behaviour and condition monitoring of mechanical seals. Based on this study it can be concluded that:
1- The materials used in the face seals must prove some proprieties such as thermo-mechanical, physical and chemical characteristics to satisfy the harsh working conditions. The graphite based on materials is widely used in seal materials. From this group, the most used is the graphite allied with antimony. Besides the materials choice, their tribological behaviour is the most important part of a mechanical seal project.
2- A good agreement between the simulation and the experimental results was reported when a coolant is supplied, however in dry running condition, it is necessary to take seals' wears into account.
3- It has been understood the surface roughness leads to the generation of a hydrodynamic load able to fully separate the surfaces after a duty parameter threshold. This corresponds to the transition from mixed to hydrodynamic lubrication regimes (optimum point).
4- The working temperature has a major effect upon the contact surface, penetration and frictional stress. All these aspects generate problems in functioning which have to be known in order to explain the leakage of fluid.
5- The effect of working conditions on the lubrication of mechanical seals is well understood; it is necessary to pave a way for developing AE techniques and signal processing strategies, as well as computational fluid dynamics (CFD) analysis to investigate the tribiological behavior of mechanical seals which has not been reported in previous works.

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