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RESEARCH

Effect of Surface Roughness and Temperature on EHL for Parallel Surfaces Subjected to High Acceleration

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A B S T R A C T

Frictional wear is one of the most common causes of material failure. One of the most common techniques to overcome this is to lubricate the surfaces in contact using an optimum amount of appropriate lubricant. Same technique is utilized in the present case to overcome the problem of frictional wear between two parallel surfaces subjected to high acceleration. The lubrication solution proposed is a modification of "Numerical model for mixed lubrication" presented in 1990s. Present research is an extension of author's previous work in the same. Here the smooth surfaces of the previous research have been replaced with rough surfaces having same surface roughness value. Also in order to study the effect of temperature the viscosity of lubricant is taken for different temperatures. Keeping in view the problem constraints as explained in section 3, certain modifications are proposed in the model of mixed lubrication. Outcome of the research is a graphical pattern showing the lubricant film thickness required at different positions between the contacting surfaces. Different grades of Krytox are used for the solution as it is nonflammable, anticorrosive Perfluoropolyether based grease, commonly used for high temperature cases. The results obtained by the numerical model are compared and found well-in-accordance with the experimental data available and with the analytical predictions made by scholars in the past, which verifies the applicability of the model chosen for solution.

1. INTRODUCTION

Friction is one of the most important natural forces which are encountered in everyday life. Depending on the type of application, it is often considered favorable or unfavorable (e.g. It is very important in case of driving a vehicle on road and mechanical fasteners whereas it is desirable to be diminished in case of shaft rotating inside a mechanical assembly, a piston sliding inside a cylinder in piston-cylinder assembly of an automotive engine).

For the case where it is necessary to avoid friction in order to reduce wear and energy loss, a fluid or precisely a viscous lubricant is used to facilitate the process. The field of Tribology provides us with principles and solutions to model such situations. One of the outcomes of such a solution is the prediction of an optimum lubrication layer which can help avoid the surface wear.

These lubrication solutions can be studied in two different lubrication regimes:

a. Fluid Film Lubrication
b. Boundary Lubrication

The fluid film lubrication regime also called as thick film lubrication can be further subdivided into:

a. Hydrodynamic Lubrication
b. Elastohydrodynamic Lubrication

Both these surfaces can be utilized in internally pressurized bearing if the surfaces in contact are converging and the presence of this internal pressure makes it possible for the bearing to support the external load [1]. The Hydrodynamic lubrication regime is also called as the thick film lubrication regime as the sliding surfaces are completely separated by the fluid film with thickness exceeding 1μm [1]. The basic difference between Hydrodynamic and Elastohydrodynamic Lubrication is that in the case of later, there is considerable elastic deformation in the contacting surfaces. The selection of lubrication regime depends on the circumstances under which the system is operating. If the system is working in such an environment where high factor of safety is required, the case of hydrodynamic lubrication should be considered. However if along with the safety factor criterion, the surface under study shows high values of elastic deformation then the regime of Elastohydrodynamic Lubrication should be considered. The selection of lubrication regime is very important for a particular Tribological solution as it further directs the study in different sub-fields of Tribology.

The present research proposes a lubrication solution while making use of different Tribological principles and mathematical relationships. The write-up is organized in 7 sections. The section II provides an overview of the material reviewed for the research and a brief introduction of each of the research work available in literature which is benchmarked for the present case. The problem under consideration is defined in Section III. Based on the problem constraints and requirements, a methodology is formed for the numerical solution which is explained in Section IV. The issues faced in previously employed techniques and modifications suggested through this research are also covered in this section. Section V provides an overview of the numerical procedure followed for the solution of the equations presented in section IV. The result deduced using the methodology and numerical procedures of section IV & V are presented and discussed in Section VI of the paper. Relevancy and comparison of work done by the author and previous research is also described in this section. Finally a conclusion is drawn in Section VII based on the above research and analysis.

2. LITERATURE SURVEY

The research in the field of Tribology dates back to the end of 19th century when Beauchamp Tower, an engineer, while performing an experiment noticed that the oil provided in between the surface of a journal bearing always leaks through a hole located on the side opposite to the load carrying side. He concluded that the primary cause of this phenomenon is the pressure generated between the two surfaces in contact.

Reynolds in 1886 provided analytical model for this pressure generation phenomenon. His theory was presented as a theory on Hydrodynamic lubrication, published in the Proceedings of the Royal Society [2]. This theory provided the first analytical proof that a viscous liquid can physically separate two sliding surfaces by hydrodynamic pressure, resulting in low friction and theoretically zero wear.

Further research showed that the principles presented by Reynolds in his theory can be utilized to predict thickness of lubricant required to develop such an amount of hydrodynamic pressure which can cause the two surfaces to not to come in contact with each other. After this, an extensive research is carried out in this regard in the field of Automotive to reduce the friction between sliding surfaces of a piston skirt and the cylinder.

The notable work in the field of engine Tribology includes the work done in 1974 by G.M Hamilton

[3] who measured film thickness between piston-liner while considering hydrodynamic lubrication regime.

In 1977, a detailed study of the Elastohydrodynamic case is submitted by Hamrock and Dowson who formulated the empirical relationships for the film thickness calculation [4].

Dong Zhu in 1991 [5,6], presented a numerical approach for the solution of mixed Lubrication in Piston-Skirt assembly. His analytical model was based on the model presented previously by Patir and Cheng in 1978 [7]. He utilized another relationship for lubricant film thickness calculations based on parameters like radial clearance between the piston skirt and cylinder surface and eccentricities involved, as he claimed that the empirical relationships of Hamrock fails in the extended range of parameters.

In 2002, he extended his work by considering broader range of parameters such as Load, Speed and Material properties. This work was presented in a series of research papers [8-10] in which he supported his theory by experimental data. All the work presented by the author was for Piston Skirt assembly in automobile engine.

The effect of lubricant viscosity on the film thickness is studied using the work [11]. Where they related the change in lubricant viscosity due to change in pressure with the film thickness values while working in EHL regime.

Surface roughness can be incorporated in the model by utilizing flow factors as presented in [12]. However keeping in view the simplicity of the model, the work done by Mistry K. N. and Priest M. in 2006 [13] is used who devised a relationship to relate the film thickness profile with the root mean square of surface roughness of surfaces under consideration. The surface roughness studies presented in [14] have also been helpful in validating the results obtained and to develop overall background in the field of Tribological solutions for rough surfaces.

The modification made in mixed lubrication model to incorporate surface roughness parameter along with other modifications is explained in the next sections.

3. PROBLEM DEFINITION

The problem studied in this research is encountered in a number of industrial equipment involving the contact between two conformal surfaces. For the case under consideration, two parallel surfaces are in contact with each other with one being stationary and the other sliding over it with high acceleration. This high acceleration is also the cause of high temperature and pressure between the contacting surfaces. The force driving the primary body is an impulse force provided by the burning of gas or any other fuel at its base. The main body is composed of a cylinder and a small fin attached to it. The secondary body is a hollow tube with rails inside, to support the linear movement of the primary body. The study is done for one complete slide of primary surface over the secondary surface (Fig. 1).

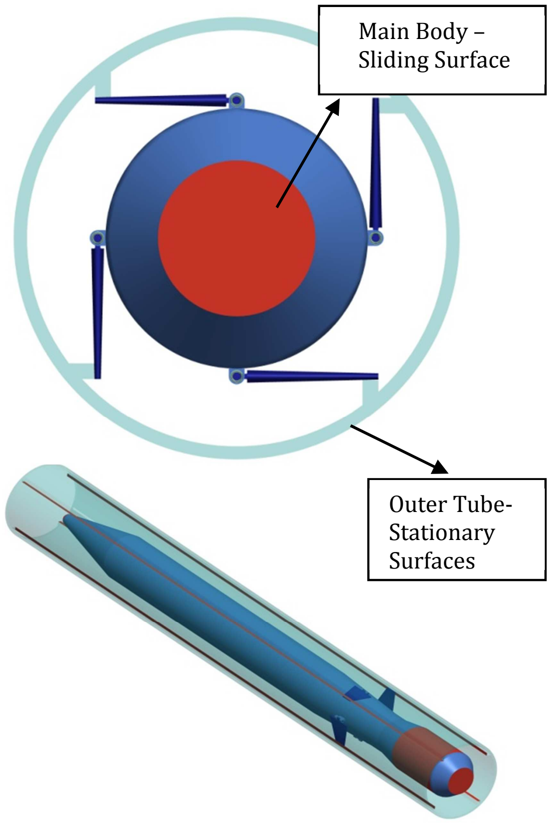

Fig. 1. Cross-sectional view and complete assembly under consideration.

Due to an impulsive force at the base, the body wobbles inside the tube. This wobbling phenomenon is very important as far as the Tribological solution is concerned as it provides the wedge shape contact which is necessary for the development of hydrodynamic pressure in the film provided for lubrication between the surfaces. The contact between the bodies is a line contact. If the bodies are left un-lubricated, the contact at high acceleration can cause some serious wear issues which not only result in power loss but also can create problems in the later stage of operation of the primary body.

The research is carried out to provide a lubricant solution for the area of contact so that the two surfaces do not come in contact during the complete time of operation. Due to high acceleration, it is assumed that there is considerable elastic deformation of the surfaces in contact.

As the surfaces are parallel to each other, therefore this becomes the case of plain slider bearing. Also the geometrical analogy can be established with the case of piston liner assembly for which a numerical model has already been proposed [5]. Therefore, the objective of this research is to modify the above mentioned mixed lubrication model with respect to application at hand and verify its applicability on the present case using experimental data and analytical prediction of various scholars. The methodology along with the proposed modifications is explained in the succeeding section.

4. METHODOLOGY AND MODIFIED NUMERICAL MODEL

In the current work, a two dimensional EHL model is presented for two sliding surfaces, while considering high acceleration. Finite difference method and inverse solution technique are adopted to solve the Reynolds equation and to generate film thickness for the complete slide of one surface over the other. Hydrodynamic friction force is also calculated in the process. To model the phenomenon, following logical assumptions are made:

- The lubricant is an incompressible newtonian fluid and the flow is laminar.
- Side leakage, oil starvation and surface roughness factors are neglected.
- No relative motion between the bodies under sliding motion.
- An iso-viscous case, that is, viscosity is same in the circumferential and sliding directions.
- The fully flooded inlet and reynolds exit conditions are applied.
- The surfaces of the ring and the liner are perfectly smooth.
- Thermal effects are neglected.

The model presented by Dong Zhu [5] uses following set of basic equations:

- Basic equation of motion
- Reynolds equation
- Lubricant film thickness equation
- Average shear stress equation
- Variable viscosity (w.r.t pressure) equation

The proposed modifications with brief justifications are:

4.1 Basic Equations Of Motion For Primary Body

The model for piston skirt assembly is considered as the case of mixed lubrication, however the present problem requires Elastohydrodynamic lubrication solution as previously discussed in Section 3. This means that the terms related to the contact forces be neglected.

Furthermore, the surfaces in contact are conformal surfaces in the same way as the surfaces in piston-skirt assembly, with the exception that geometrical parameters are different. This requires a modified structural model to incorporate the difference of geometrical parameters. In order to do this, the basic equation of motion for primary body is modified in following way:

The forces and moments acting around the centre of Gravity of the main body and fin are considered and equilibrium equations are applied at centre of gravity of fin. Final equations formulated by simplifying the equilibrium equation are:

$$F_h + F_S + F_{fh} = -F_{MB} - F_{Fin} \qquad (1)$$

$$M_h + M_{fh} + M_{MB} + F_{MB} * (x) = 0 \qquad (2)$$

The inertial forces of the body depend upon the primary body’s acceleration in transverse direction (which in turn are the derivatives of eccentricities calculated at the top and bottom of the tube i-e ($\ddot{e}_t$&$\ddot{e}_b$) and are given by the relationship:

$$F_{MB} = -m_{MB}\left(\ddot{e_t} + \frac{b}{L}(\ddot{e_b} - \ddot{e_t})\right) \tag{3}$$

$$F_{Fin} = -m_{Fin}\left(\ddot{e_t} + \frac{a}{L}(\ddot{e_b} - \ddot{e_t})\right) \tag{4}$$

$$M_{MB} = -I_B(\ddot{e_t} - \ddot{e_b})/L \tag{5}$$

$$F_S = F_G + \widetilde{F_{MB}} + \widetilde{F_{Fin}} \tag{6}$$

$$\widetilde{F_{MB}} = m_{MB} * aY \tag{7}$$

$$\widetilde{F_{Fin}} = m_{Fin} * aY \tag{8}$$

$$aY = \frac{F_G}{(m_{MB} + m_{Fin})} \tag{9}$$

Table 1. Nomenclature for equation (3)-(9).

Terms	Definition	Value
F_h and M_h	Hydrodynamic Force and moment	Calculated by integrating hydrodynamic pressures
F_{fh} and M_{fh}	frictional force and moment due to hydrodynamic lubricant film	Calculated by integrating the Shear stress
F_{MB}	Inertial force due to primary body	Calculated by Equation (3)
F_G	Gas Force or Thrust Force	Pre-defined term
F_{Fin}	Inertial force due to Fin	Calculated by Equation (4)
M_{MB}	Moment due to mass of primary body	Calculated by Equation (5)
$\widetilde{F_{MB}}$ and $\widetilde{F_{Fin}}$	Reciprocating inertial forces	Calculated by Equation (7 & 8)
aY	Acceleration of the body	Calculated by Equation (9)
I_B	Angular moment of Inertia about the body's center of mass	
x	Perpendicular distance from point of application of FMB to the center of gravity of Fin	

4.2 Lubricant Film Thickness with surface roughness

The film thickness equation used in the mixed lubrication is modified in a way that the deformation term $d\,(\theta, y, t)$ includes the deformation due to interaction between the two concerned surfaces only and the deformation due to thermal effect and combustion are neglected. The modified equation of film thickness for elastically deformed surfaces can be expressed as:

$$h = C_L + e_t(t)\cos\theta + [e_b(t) - e_t(t)] \times \frac{y}{L}\cos\theta + d\,(\theta, y, t) \tag{10}$$

As per the work done by Mistry et.al [13] the average lubricant film thickness can be related with the root mean square value of surface roughness using the following relationship:

$$h_a = h - \frac{\sigma}{\sqrt{2\pi}} \tag{11}$$

Here σ is the combined surface roughness variance given by the following formula:

$$\sigma = \sqrt{(\sigma_1^2 + \sigma_2^2)} \tag{12}$$

Table 2. Nomenclature for equation (10)-(12).

Terms	Definition	Value
C_L	Clearance between the fin surface and the tube wall	Pre-defined term
θ	angular coordinates	Calculated by integrating the Shear stress
$e_t(t)$&$e_b(t)$	eccentricities of the primary body at the top and bottom of tube	Assumed in the beginning and then calculated step wise
y	vertical distance travelled by the body in the tube	Changes at each time step
L	length of the tube	Pre-defined term
ha	Average film thickness	Calculated by Equation (11)
$\sigma_{1,2}$	Root mean square values of surface roughness	Pre-defined term

4.3 Average Shear Stress Factor

The calculation of average shear stress factor is important to calculate the hydrodynamic frictional forces and moments (F_{fh} and M_{fh}). The relationship for average shear stress is:

$$\tau = \frac{-\mu U}{h} \tag{13}$$

Here the U can be calculated wide relationship:

$$U = \frac{F_G}{m_{MB} + m_{Fin}} * t \tag{14}$$

The above equation is the modified relationship as compared to the one used by Dong Zhu due to difference of application and is based on the work of D.K. Kankane and S.N. Ranade [15].

Table 3. Nomenclature for equation (13)-(14).

Terms	Definition	Value
μ	lubricant viscosity	Pre-defined term
h	Film thickness	Calculated vide Equation (10)
t	Time step at which the velocity is calculated	

4.4 Variable Viscosity of the Lubricant

Basic model uses the Barus Equation to incorporate the effect of variable viscosity. The Barus equation gives a relationship between the viscosity and the pressure as [16]:

$$\eta_p = \eta_0 e^{\alpha p} \tag{15}$$

The studies have revealed that in case the pressure is greater than 0.5 GPa, the use of Barus equation can result in serious error in calculations [17]. For such cases, Chu et al. in 1962 [18] devised another relationship given as:

$$\eta_p = \eta_0 (1 + C \times p)^n \tag{16}$$

Table 4. Nomenclature for equation (15)-(16).

Terms	Definition	Value
α	pressure viscosity co-efficient	Calculated by plotting the natural algorithm of dynamic viscosity versus pressure. The slope of this graph is α[19]
C	Constant	Obtained through experimental data [19]
n	Constant	Approximately 16
η_p	Viscosity at Pressure 'p'	
η_0	Viscosity at atmospheric pressure	
p	Concerned pressure	

5. NUMERICAL PROCEDURE

Following procedure is followed for the solution of problem at hand:

- Assume eccentricities (e_t , e_b, $\dot{e}_t$&$\dot{e}_b$) for an initial time step and use them to calculate the film thickness (h) at the current time step by using equation (10).
- Squeeze film term ($\partial h/\partial t$) is calculated using derivative of equation (10).
- Using (h) and ($\partial h/\partial t$) shear stress factor τ is calculated through equation (11) which can be in turn integrated over the area of contact to findF_{fh} and M_{fh}.
- Solve Reynolds Equation using finite difference scheme and SOR (Successive over-relaxation) method to calculate hydrodynamic pressure P_h.
- The hydrodynamic pressure can be used to calculate hydrodynamic force and moments (F_h and M_h).
- Reciprocating inertial forces are calculated through equations (7&8).
- All the forces and moments calculated using above steps are utilized in the equation (1 & 2) to calculate inertial forces.
- These inertial forces are used in equations (7&8) to calculate new values of eccentricities ($\dot{e}_t$&$\dot{e}_b$).
- The steps from 2-8 are repeated until the convergence criterion is met.

Based on above numerical procedures following results are produced.

6. RESULTS AND DISCUSSIONS

Few parameters were initially plugged in the numerical solution which includes:

Table 5. Input Parameters.

Terms	Symbol Used	Value
Mass of primary body	m_{MB}	2000 Kg
Mass of fin	m_{Fin}	10 Kg
Viscosity of fluid (Krytox 226, 227)	η_0	Table 6
Diameter of primary body	D	0.45 m
Elastic Modulus	E	69 GPa
Radial clearance between fin and tube	C_L	0.00001 m
Thrust Force	F_G	17-32 KN
Surface Roughness	σ1,2	1.5 μm

Table 6. Viscosity Data.

Temperature	Lubricant Viscosity	
	Krytox 226	Krytox 227
50 °C	0.2707	0.4931
100 °C	0.0445	0.0748
150 °C	0.0142	0.0229
200 °C	0.0066	0.0101
250 °C	0.0038	0.0056

Based on the above data 40 cases were simulated and results were analysed.

Table 7. Parametric Study

Case #	Krytox Grade	Temp (°C)	Thrust Force	Roughness
1-5	226	50 - 250	40 KN	No
6-10	227	50 - 250	40 KN	
11-15	226	50 - 250	32 KN	
16-20	227	50 - 250	32 KN	
21-25	226	50 - 250	40 KN	Yes
26-30	227	50 - 250	40 KN	
31-35	226	50 - 250	32 KN	
36-40	227	50 - 250	32 KN	

It is important to mention here that the lubricant selected in the present case is a type of grease and not lubricating oil. The reason being, that our problem deals with high pressure and acceleration and in case of high pressure there is a chance that the lubricating oil will squeeze out, whereas lubricating grease will act as a lubricating oil reservoir that shall provide the lubricating oil whenever pressure is applied. This justification is supported by the study presented in [20]. Moreover the values of the lubricant viscosity are taken at a temperature of 100 °C [21] assuming this temperature at the surfaces in contact during the passage of the primary body through the secondary body as supported by [22].

MATLAB is used to implement the numerical solution proposed above. Following results were obtained for case 1-20 using the above data:

Table 8. Results for Case 1-10.

Thrust Force = 40 KN		
Temperature	**Krytox 226**	**Krytox 227**
50 °C	0.249 μm	0.345 μm
100 °C	0.172 μm	0.189 μm
150 °C	0.121 μm	0.131 μm
200 °C	0.098 μm	0.118 μm
250 °C	0.090 μm	0.093 μm

Table 9. Results for Case 11-20.

Thrust Force = 32 KN		
Temperature	**Krytox 226**	**Krytox 227**
50 °C	0.228 μm	0.301 μm
100 °C	0.128 μm	0.150 μm
150 °C	0.087 μm	0.106 μm
200 °C	0.079 μm	0.090 μm
250 °C	0.072 μm	0.078 μm

Following graphs have been plotted based on the above summarized results to describe the dependence of variation of film thickness with temperature change:

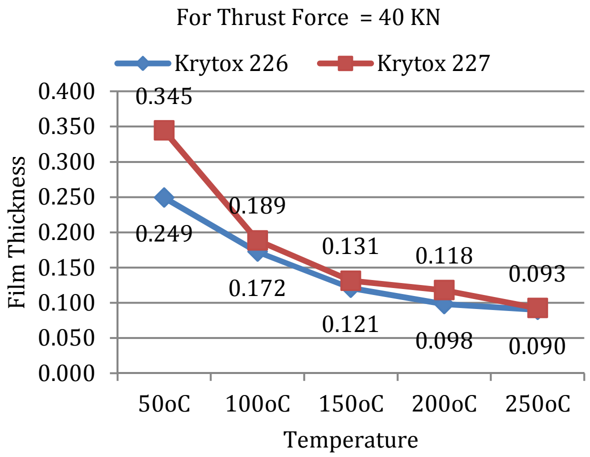

Fig. 2. Summarized results for case 1-10.

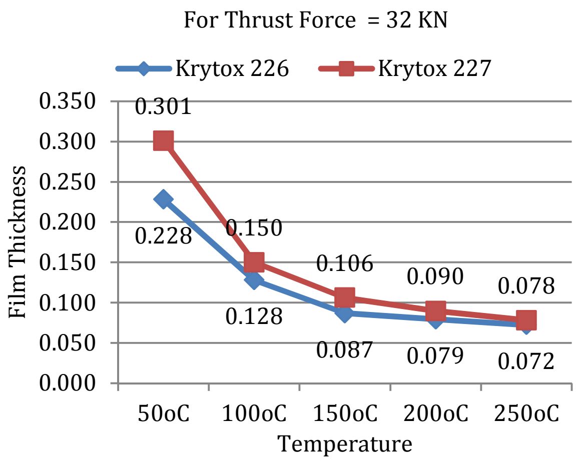

Fig. 3. Summarized results for case 11-20.

Table 10. Results for Case 21-30.

Thrust Force = 40 KN		
Temperature	**Krytox 226**	**Krytox 227**
50 °C	0.250 μm	0.345 μm
100 °C	0.191 μm	0.209 μm
150 °C	0.151 μm	0.166 μm
200 °C	0.134 μm	0.153 μm
250 °C	0.104 μm	0.125 μm

Table 11. Results for Case 31-40.

Thrust Force = 32 KN		
Temperature	**Krytox 226**	**Krytox 227**
50 °C	0.275 µm	0.302 µm
100 °C	0.178 µm	0.192 µm
150 °C	0.126 µm	0.137 µm
200 °C	0.100 µm	0.112 µm
250 °C	0.093 µm	0.105 µm

The graphical summary of the results is given below:

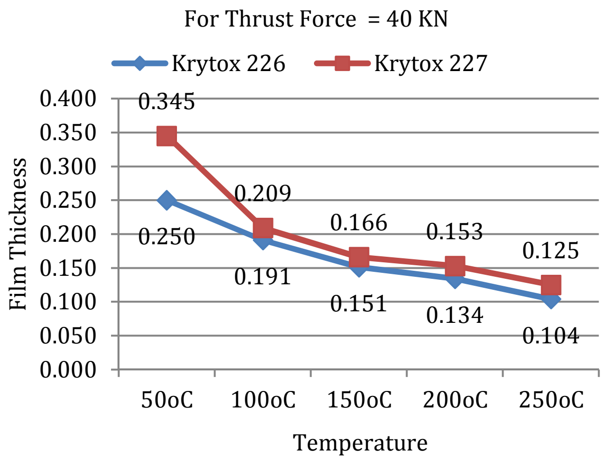

Fig. 4. Summarized results for case 21-30.

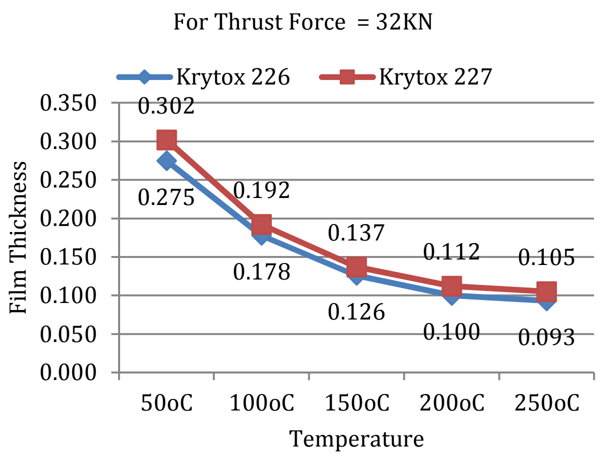

Fig. 5. Summarized results for case 31-40.

The graphs drawn by MATLAB are between the lubricant film thicknesses against the distance travelled by the body inside the tube. They are given at the end of the text for reference purpose. The above graphs are just the summary of the reference graphs. Following discussion is focussed on reference as well as the summary graphs:

- The graph between the film thickness and the length of the outer tube is almost a straight line with just a small peak in the beginning, it shows that the relationship between the two mentioned parameter is linear i-e with the advancement of the body in the tube the quantity of the fluid required to keep the surfaces apart from each other increases. The reason for this can be supported by the fact that as the thrust force increases during the motion of the body through the tube, the hydrodynamic pressure increases as it is directly proportional to the projectile velocity. As the hydrodynamic pressure increases, it tends to push the bodies in contact further apart from each other increasing the wedge between the surfaces which increase the thickness of the film between the surfaces.
- The lubricant thickness increases with the passage of time as the body moves through the tube.
- The spikes in the beginning of the graph are due to the wobbling of the primary body which is because of the initial thrust force.
- As the lubricant's viscosity increases, the maximum value of the lubricant film thickness increases. The results are supported by the experimental proofs obtained in the [22]. In this study, author employed an experimental technique to investigate the relationship between film thickness and the viscosity of the lubricant. The condition of pure rolling and rolling with sliding were considered. The value of viscosity was varied by changing the temperature of the source of lubricant. Author concludes that the viscosity of fluid to be taken for the calculation of film thickness is the viscosity of lubricant out of pressure zone and at the temperature of the surfaces in contact. Same criterion is used in this study to calculate the viscosity of the lubricant used. The experimental results also indicate that with increase of lubricant viscosity, the lubrication film thickness increases.
- Dong Zhu in his research [5] explained that if the deformation is taken into account or in other words if the case of Elastohydrodynamic lubrication is considered, the results would be non-linear. The reason for the linearity of present results can be justified on the basis of the difference of application to which the studies are employed. In the previous research for the case of piston skirt

assembly, four (04) strokes of the surfaces in contact were considered whereas in our case the bodies just come in contact for one stroke and during which the results remain linear.

If we focus on the summary graphs following can be concluded:

- Non-Linear variation of Film throughout w.r.t temperature
- Direct relationship between viscosity and film thickness (same as author's previous work [23])
- Slope of graph decreases throughout the temperature variation
- Relationship between thrust force and film thickness is observed to be the same as author's previous work.

While studying the effect of surface roughness on the individual cases of lubricant and thrust force following graphs can be plotted:

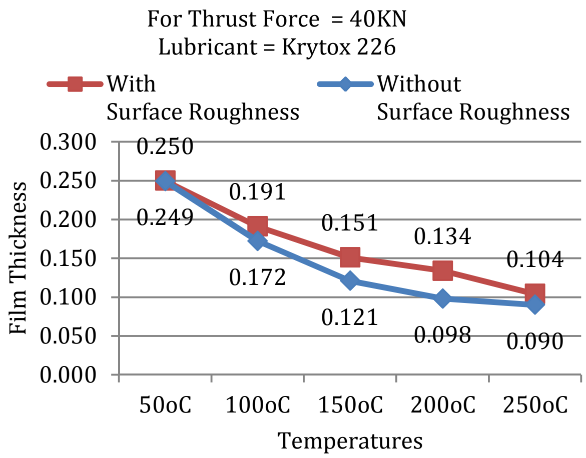

Fig. 6 Effect of surface roughness at 40KN Force for Krytox 226.

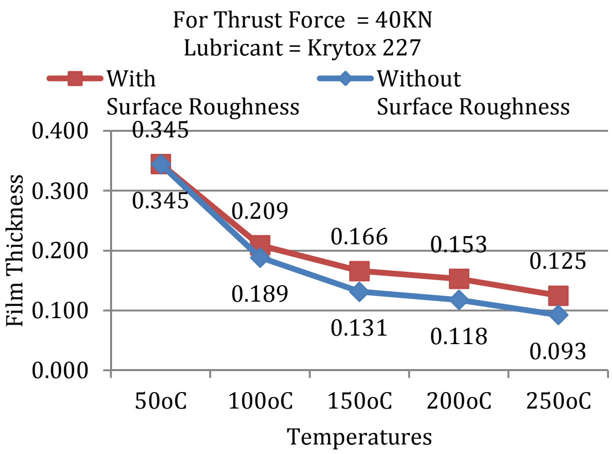

Fig. 7 Effect of surface roughness at 40KN Force for Krytox 227

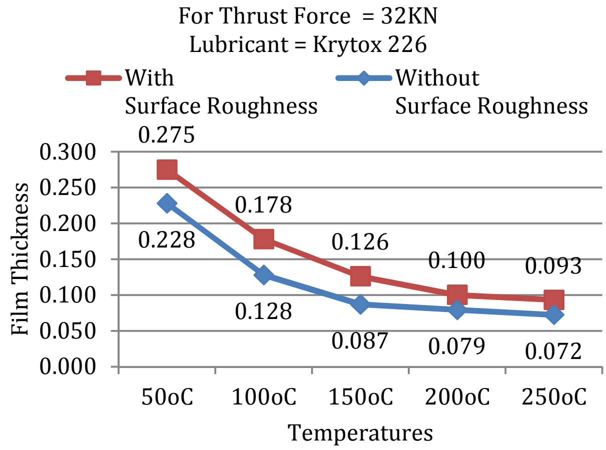

Fig. 8. Effect of surface roughness at 32KN Force for Krytox 226

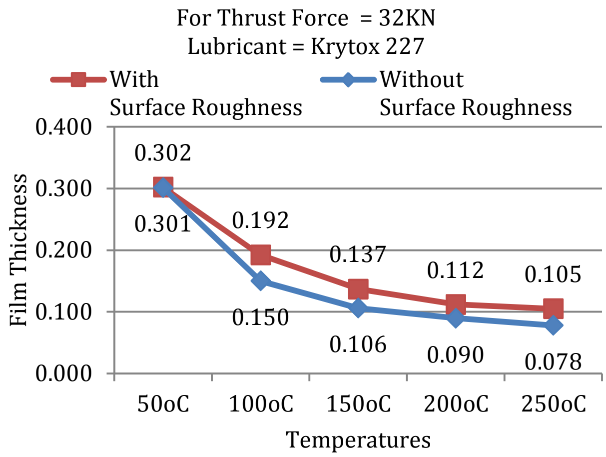

Fig. 9. Effect of surface roughness at 32KN Force for Krytox 227.

Based on above graphs following is concluded:

- The value of film thickness increases with the introduction of Surface roughness in the model.
- This increase in film thickness increases with the increase of temperature.
- Slope of graph decreases throughout the temperature variation
- The slope of the curve for case 31-40 the slope almost becomes constant after temperature is increased further from 250 °C.
- The slope of curve seems to be unaffected with the change of lubricant.
- The results found using the presently developed numerical model for film thickness prediction were in close accordance with the experimentally found value of 0.25 μm while using Krytox 226 for the experiment. An error of just 14.4 %

is observed. During the experiment a layer of 0.25 μm was applied to the contacting surfaces and no wear was observed.

7. CONCLUSION

Lubrication solution is one of the most effective techniques to avoid wear due to friction. The present model provides an effective numerical baseline to simulate different real life scenarios. The model provides a cost effective technique to make sure that the lubricant film thickness provided between the sliding surfaces is enough to generate such hydrodynamic pressures which can keep the surfaces separated.

Based on the above research and discussion, it is concluded that the distance travelled in the tube is directly proportional to the film thickness value. The increase in viscosity of the lubricant increases the amount of lubricant required to keep the surfaces parted. Final result of this analysis is a predicted value of lubricant film thickness. This predicted film thickness when supplied and kept between the two surfaces in contact will maintain such hydrodynamic pressures which can prevent wear of the surfaces. The results are well in accordance with the discussion presented in [6] and [20].

The effect of surface roughness and temperature has been studied and the results are very well in accordance with the predicted values and the previously published work of author.

The development of a numerical model for the problem stated in this research is very important as it lays down the basis for various parametric studies which can be carried out by altering the various parameters used in this particular paper.

The present work can be extended to draw a comparison between the hydrodynamic and Elastohydrodynamic lubrication regimes.

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