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EFFECT OF VISCOSITY ON ELASTOHYDRODYNAMIC LUBRICATION BETWEEN PARALLEL SURFACES SUBJECTED TO HIGH ACCELERATION

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Abstract:*Material wear due to friction is one of the most commonly experienced causes of material failure in any mechanical industry. Various studies have been conducted and as a result of these studies various lubrication solutions have been proposed. Present work is an attempt to propose an Elastohydrodynamic lubrication solutio20n for a frictional wear problem experienced in an industrial application involving the sliding contact between two parallel surfaces subjected to high acceleration. The "Numerical model for mixed lubrication" developed by Dong Zhu in 1990s has been modified to accommodate the constraints of problem at hand. The solution proposed predict the lubricant film thickness that when maintained between the contacting surfaces can avoid full metal contact, which in turn shall avoid the material wear. The present research is an attempt to comprehend the effect of change of viscosity and change of clearance between the surfaces on the value of film thickness. Different grades of nonflammable, anticorrosive Perfluoropolyether based grease (Krytox) are used. The results obtained are in the form of graphs which calculates the value of film thickness for one complete slide of one surface over the other. 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.*

Keywords: *Tribology, Friction, Elastohydrodynamic lubrication, Krytox, Shear stress factor, hydrodynamic pressure*

1. INTRODUCTION

Wear is the major cause of material wastage and loss of mechanical performance and any reduction in wear can result in considerable savings. Friction is a principal cause of wear and energy dissipation. Considerable savings can be made by improved friction control. It is estimated that one-third of the world's energy resources in present use is needed to overcome friction in one form or another. Lubrication is an effective means of controlling wear and reducing friction. Principles studied under the field of tribology helps to analysis such problems of frictional wear. Lubrication phenomenon is being used to avoid fricational wear since the time of its discovery.

For any tribological studies the lubrication regime in which that particular machine/ application is working is very important as it steers the later research in the field. The outcome of most of these researches is a lubrication solution that if maintained between the contacting surfaces can avoid dry metal-metal contact and inturn diminishes the chances of any mechanical wear. A lubrication solution can be studied under two main types of regimes:Fluid Film Lubrication and Boundary Lubrication.

Out of these regimes, the fluid film lubrication is considered in case the required factor of safety is high so it is desired that the two contacting surfaces have minimum chances of coming in contact with each other. This requires a thick lubrication layer (exceeding a thickness of more that 1um) [1] between the surfaces so that enough hydrodynamic pressure exists to keep the surfaces apart.

Table 1: Nomenclature of Symbols.

Terms	Definition	Value	Terms	Definition	Value
F_h and M_h	Hydrodynamic Force and moment	Calculated by integrating hydrodynamic pressures	M_{MB}	Moment due to mass of primary body	Calculated by Equation (5)
F_{fh} and M_{fh}	frictional force and moment due to lubricant film	Calculated by integrating the Shear stress	$\widetilde{F_{MB}}$ and $\widetilde{F_{Fin}}$	Reciprocating inertial forces	Calculated by Equation (7)
F_{MB}	Inertial force due to primary body	Calculated by Equation (3)			Calculated by Equation (8)
F_G	Gas Force or Thrust Force	Pre-defined term	aY	Acceleration of the body	Calculated by Equation (9)
F_{Fin}	Inertial force due to Fin	Calculated by Equation (4)	I_B	Angular moment of Inertia about the body's center of mass	
x	Perpendicular distance from point of application of F_{MB} to the center of gravity of Fin		C	Constant	Obtained through experimental data
η_p	Viscosity at Pressure 'p'		p	Concerned pressure	
η_0	Viscosity at atmospheric pressure		n	Constant	Approximately 16
α	pressure viscosity co-efficient	Calculated by plotting the natural algorithm of dynamic viscosity versus pressure. The slope of this graph is α[16]			

For the case of boundary lubrication no hydrodynamic film is sustained. The coefficient of friction is very high and friction is proportional to the applied load for a certain range of sliding velocity and temperature.

Fluid film lubrication is further classified as hydrodynamic and Elastohydrodynamic lubrication, based on the fact that the surfaces under consideration show remarkable elastic deformation or not. As evident from above discussion the selection of lubrication regime is very important and is completely dependent on the circumstances under which the surfaces come in contact.

The present research is focussed on the lubrication solution for problem under the regime of elastohydrodynamic lubrication. The present research is devided into 7 sections. Section 2 focusses on the literature studied and benchmarked for the present research. The problem under consideration is explained in the section 3.

The constraints explained in the senction 3 laid down the basics for developement of a methodology for the solution, the same is explained in the section 4. Section 5 explains the results obtained on the basis of methodology of section 4. Section 6 concludes the work by drawing the conclusions based on discussions of section 5.

2. LITERATURE SURVEY:

The studies in the field of tribology initiated in 19th century with the experiment of Beauchamp Tower when he noticed that the oil film provided in between the surfaces of a journal bearing tries to pump out of a hole provided on top of the bearing. He conculded that the pumping phenomenon can be explained with the generation of hydrodynamic pressure between the bearing surfaces.

Reynolds in 1886 [2] provided the mathematical solution for the generation of above mentioned pressure. The theory presented by Reynolds provided the long waited analytical proof of the hydrodynamic pressure generated between the surfaces. This hydrodynamic pressure helps keep the surfaces apart and thus avoiding the metal-metal contact which inturn avoid the mechanical wear.

After Reynolds, many scholars tried to predict the lubricant film thickness based on his work, for problem related to different types of bearings including thrust bearing, journal bearing and slider bearing etc. The most notable work in this reagrd was done in the field of automobiles to predict the lubrication solution for piston-liner assembly.

G.M. Hamilton in 1972 [3] predicted the film thickness solution for piston-liner problem while considering the regime of hydrodynamic lubrication. Further in 1977 Hamrok and Dowson [4] formulated the empirical formulae for film thickness calculation.

For Elastohydrodynamic film thickness solution, the first realistic model was provided by Ertel and Grubin [5]. After Grubin other significant contribution in the field was made by Dowson and Higginson [6]. They described an iterative procedure that not only yielded a wide range of solutions during the next decade, but also enabled them to derive an empirical minimum-film thickness formula for line contacts.

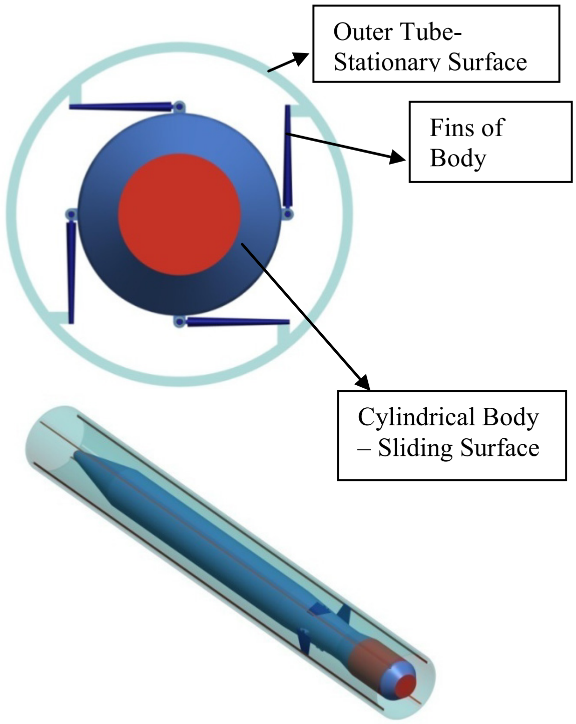
One of the most considerable works in this regard was presented by Dong Zhu in 1991 [7, 8]. He presented a model namely "Numerical Model for Piston-Skirt assembly under mixed lubrication". He developed a new relationship for film thickness based on the eccentricities and clearance between the piston and cylinder. It was also claimed that the empirical relationship developed previously by Hamrok is not applicable for cases of high pressure and acceleration. Further in 2002 he extended his work [9-11] by considering a wider range of parameters like Load, Speed and Material properties. He also supported his theory with experimental data in a series of research papers.

The aim of the present research is to verify the applicability of Dong Zhu's model on our present assembly. However our present problem, described in detail in the next section, has different geometrical and environmental constraints, therefore it requires certain modification in the original model.

3. PROBLEM DEFINITION:

The problem at hand involves a cylindrical body sliding inside a hollow tube. The sliding action of the cylidrical body is under high accelearation and therfore involves high pressure environment. It is desirable that the body does not have any wear at its surface as it can alter the trajectory followed by it during its further operation. Based on high factor of safety required and the deforamtion of the body due to high pressure and high acceleration, elastohydrodynamic lubrication regime is considered. The assembly is shown below.

The force which forces the cylindrical body to slide is a thrust force at the base of the body. This further guides the research to be done in the field of thrust bearing. The thrust force is also responsible for the wobbling action of the cylinder inside the tube this wobbling is in itself responsible for the generation of hydrodynamic pressure in the lubricants' layers provided between the surfaces that keeps the surfaces apart and avoid mechanical wear as will be explained in more detail in later sections.



The present research tries to find a lubrication solution in term of film thickness for the assembly explained earlier. The film thickness predicted through the results of the research, if maintained in between the contacting surfaces shall be able to withstand the thrust force avoiding the surfaces to come in contact.

The model preseneted by Dong Zhu is modified based on the above mentioned constarints as shall be described in the following section.

4. MODIFIED NUMERICAL MODEL:

To model the phenomenon, following 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.

Based on the problem constraints and the assumptions made following modification are made in the Piston skirt model.

4.1. Modifying mixed lubrication model to EHL Model

As previsouly explained the lubrication regime to be considered for our present case is Elastohydrodynamic lubrcation whereas the regime of Dong Zhu's work [7] was mixed lubrcaiton. In order to incorporate this change, the terms (forces and moments) related to the contact between the two bodies are neglected.

$$F = F_h + \cancel{F_c}$$

$$F_f = F_{fh} + \cancel{F_{fc}}$$

$$M = M_h + \cancel{M_c}$$

$$M_f = M_{fh} + \cancel{M_{fc}}$$

4.2. Modification in Basic Dynamic Model

Due to different geometrical constraints the dynamic model is modified thorugh following set of equations. These equation are based on summation of forces and moments. The forces and moments acting around the Center of Gravity of the cylindrical body and fin are considered and equilibrium equations are applied at center of gravity of fin. Final equations formulated by simplifying the equilibrium equation are:

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

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

Where

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

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

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

$$F_S = F_G + \widetilde{F_{MB}} + \widetilde{F_{Fin}} \quad (6)$$

Where e_b & e_t are eccentricities at the top and bottom of the tube.

4.3. Modification in Acceleration Profile

The acceleration profile is modified based on the research by D.K. Kankane et. al. [12] while calculating the in-bore velocity of a projectile:

$$\widetilde{F_{MB}} = m_{MB} * aY \quad (7)$$

$$\widetilde{F_{Fin}} = m_{Fin} * aY \quad (8)$$

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

4.4. Modification in Pressure-Viscosity Relationship

The numerical model for mixed lubrication uses the Barus eqution to cater for the change in viscosity due to pressure. Studies have shown that in case of high pressure problems the use of Barus equation can result in serious error in film thickness and hydrodynamic pressur calculations and therefore a new relationship developed by Chu et al. [14] in 1962 shall be used.

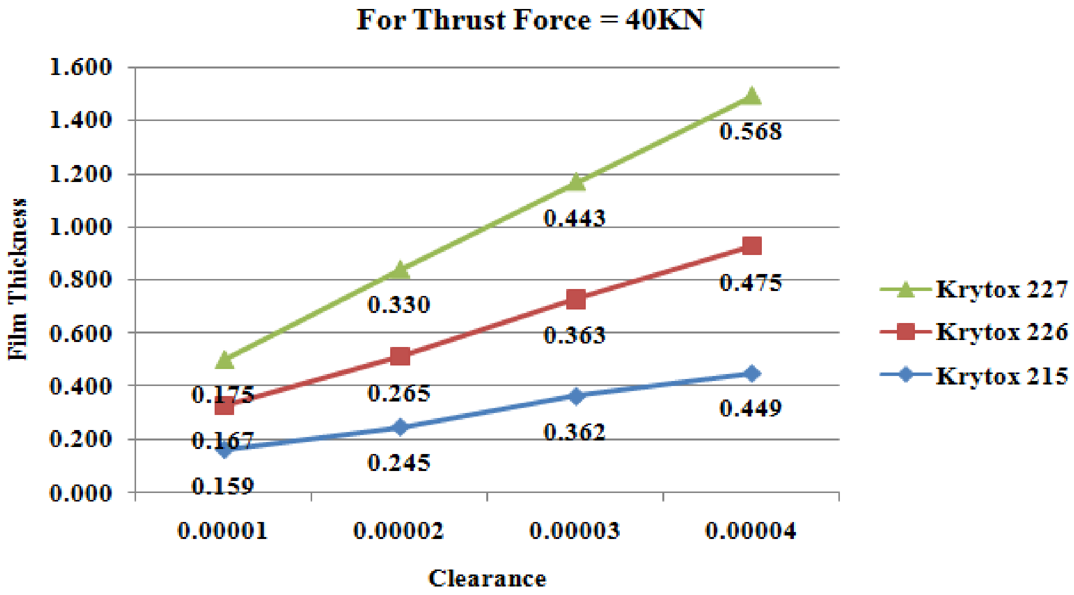
Barus Equation [13]	$\eta_p = \eta_0 e^{\alpha p}$	(10)
Equation by Chu et. al	$\eta_p = \eta_0(1 + C \times p)^n$	(11)

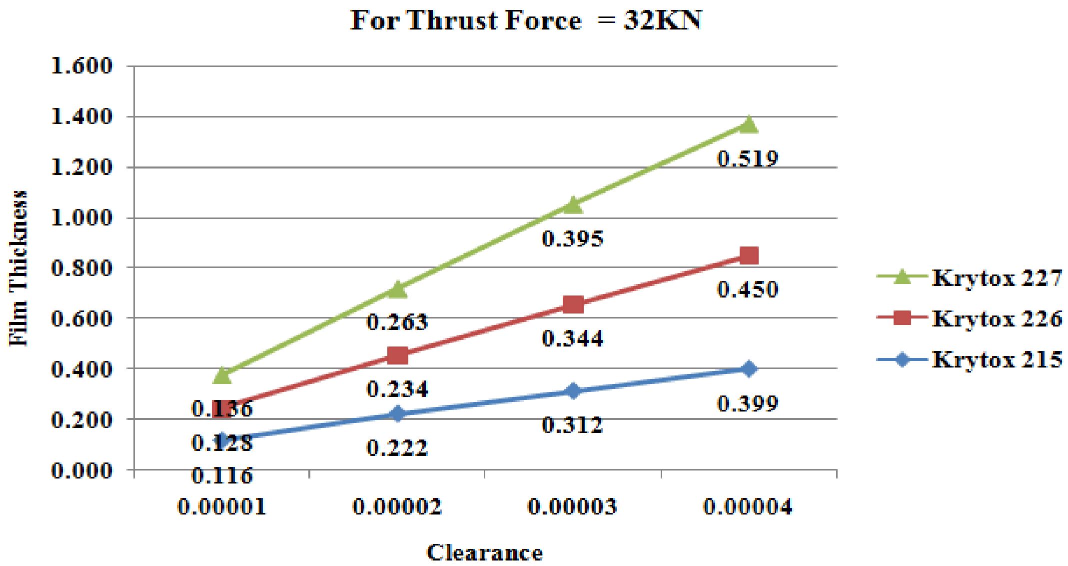
5. RESULTS AND CONCLUSION:

Following data is used as input for the numerical model.

Terms	Symbol Used	Value
Mass of primary body	m_{MB}	2000 Kg
Mass of fin	m_{Fin}	10 Kg
Viscosity of fluid (Krytox 215, 226, 227)	η_0	0.03204Pa.s 0.04550 Pa.s 0.07476 Pa.s
Diameter of primary body	D	0.45 m
Elastic Modulus	E	69 GPa
Thrust Force	F_G	17-32 KN

Following results have been plotted using MATLAB. The results are graphical, showing the effect of change in clearance between the contacting surfaces, viscosity and the thrust force. The graph is drawn between the film thickness and the length of the outer tube, this gives us the value of the film thickness that if maintained between the surfaces throughout the length of the tube shall avoid any metal contact between the surfaces and therefore will help diminish the mechanical wear due to it.





The results concluded for different grades of Krytox at different values of clearance have been attached at the end of the paper for reference purpose however a summarized graph for both case of thrust forces are plotted in MS Excel and are shown above.

Based on the graphs attached at the end of the paper and the summary graph displayed above, following is concluded:

- The film varies almost linearly throughout the length of the tube. This can be explained by the fact that during the sliding of one surface over the other the thrust force increases and this increase in the thrust force causes an increase in the hydrodynamic pressure which is directly proportional to the velocity of the sliding surface. This increase in the hydrodynamic pressure requires an increased amount of lubricant to avoid metal-metal contact.
- The small peaks in the beginning of the graph can be explained with the help of wobbling phenomenon that takes place due to the impulsive nature of the force provided.
- It is clear from the above graphs that the value of film thickness required, increases with increase in the viscosity of the liquid keeping the clearance between the surfaces constant. This conclusion is also supported by the experimental results provided in the research carried out by Crook, A. W in 1961 [15].
- It is also concluded that with increase in the clearance between the surface the value of film thickness increases this can also be

- explained with the direct relationship between the film thickness and the clearance as provided by Dong Zhu [7, 8].
- The experimental results already available for the present problem are for the case of Krytox 226 while keeping the clearance level of 0.00002 are 0.25 µm that is close in accordance with the numerically calculated value of 0.265 µm with an error of 6%.
- The summarized graph also show that the greater the value of the viscosity the greater the slope of the line which show that the effect of clearance between the contact surfaces increases with increase in the viscosity of the lubricant.

6. CONCLUSION:

Based on the above discussion it is concluded that the effect of clearance between the sliding surfaces under Elastohydrodynamic lubrication regime increases with increase in the viscosity of the lubricant and also that the increase in the thrust force requires a more thick layer of lubricant to be provided between the surfaces to avoid mechanical wear. The comparison of results with the experimental data and the analytical data of scholars from past, the applicability of the modified Dong Zhu's model for present case of sliding surfaces subjected to high acceleration is verified. The present work can be extended to incorporate the roughness of the contacting surfaces as well as to draw a comparison of Elastohydrodynamic and hydrodynamic lubrication regime's circumstances.

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For Thrust Force of 40KN
Krytox 215 (μ = 0.03204 Pa.s)

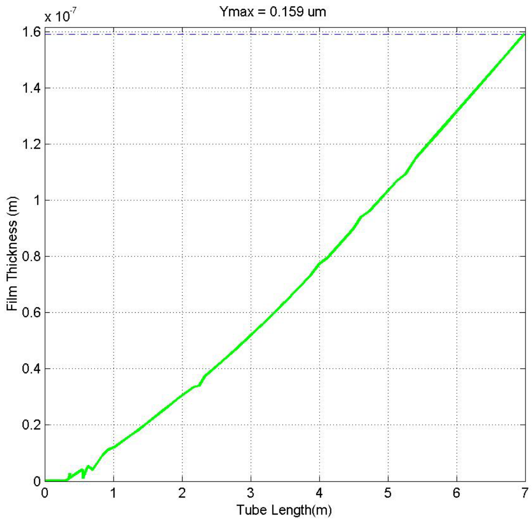


Figure 1. Clearance = 0.00001

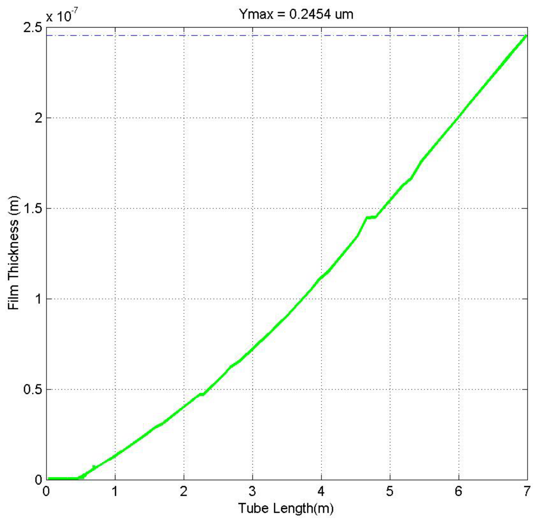
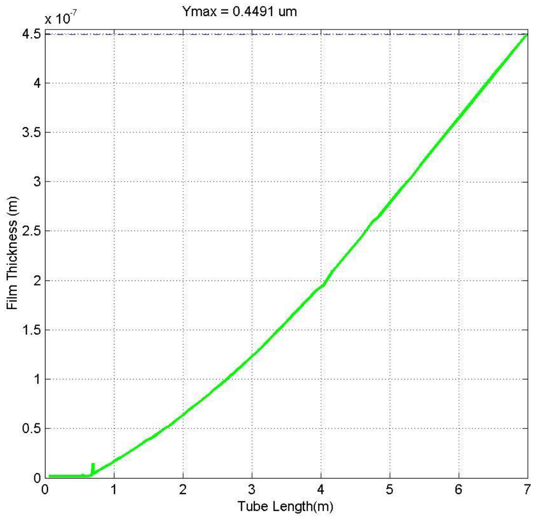


Figure 2. Clearance = 0.00002



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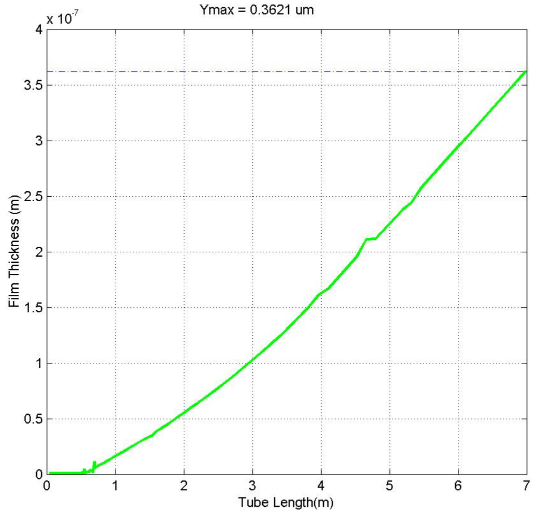
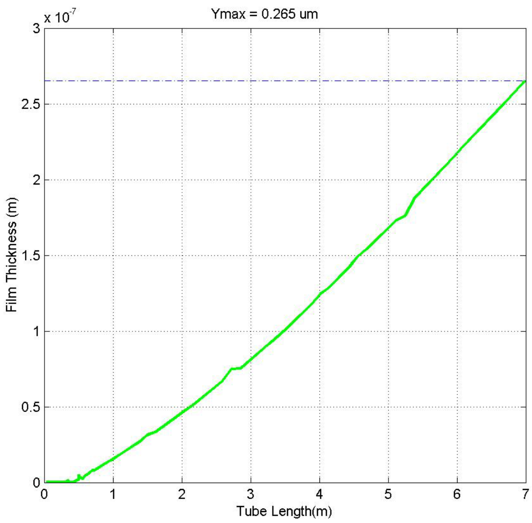


Figure 4. Clearance = 0.00004

Krytox 226 (μ = 0.0445 Pa.s)



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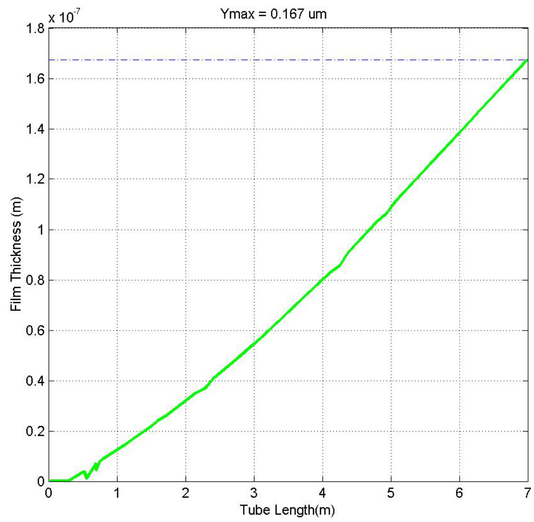

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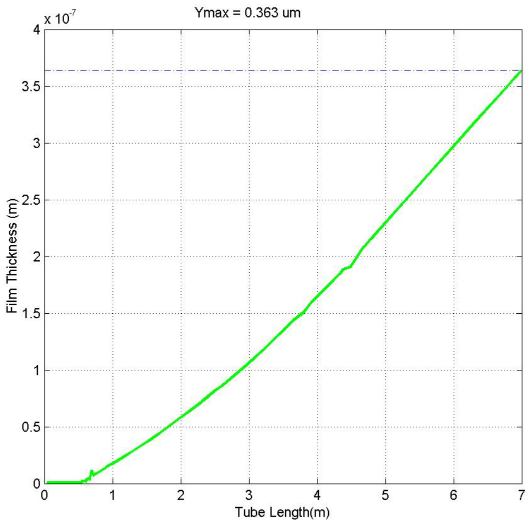

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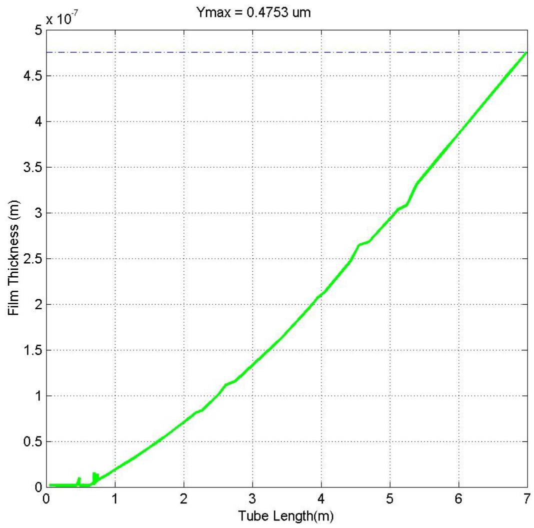
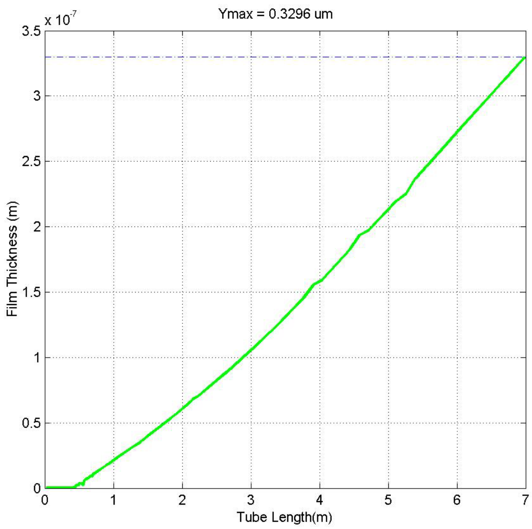


Figure 4. Clearance = 0.00004

Krytox 227 (μ = 0.07476 Pa.s)



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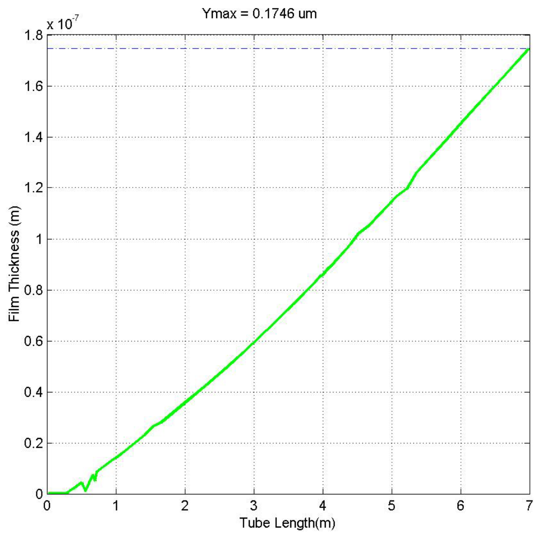

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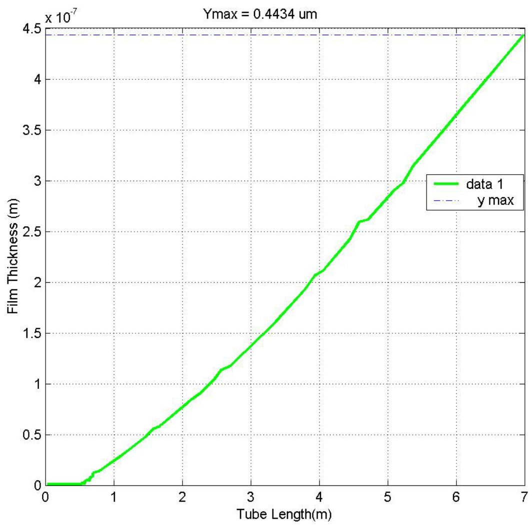

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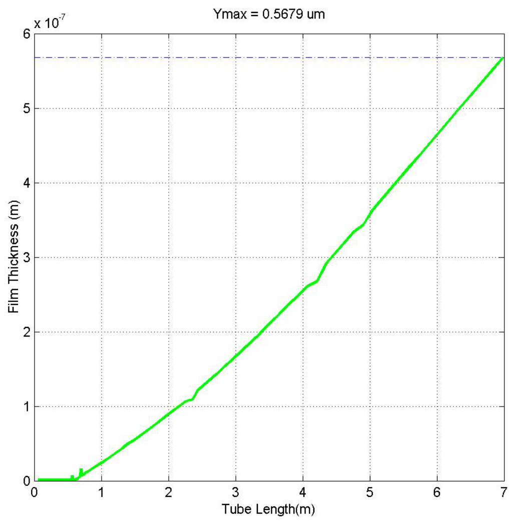
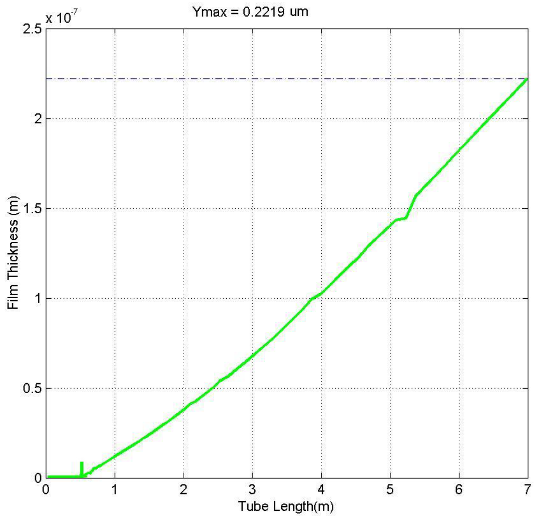


Figure 4. Clearance = 0.00004

For Thrust Force of 32KN

Krytox 215 (μ = 0.03204 Pa.s)



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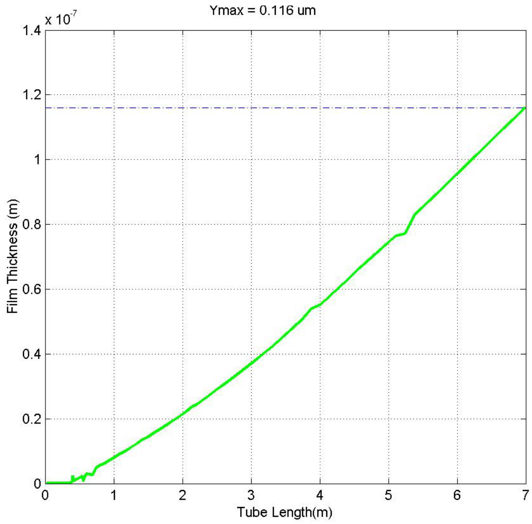

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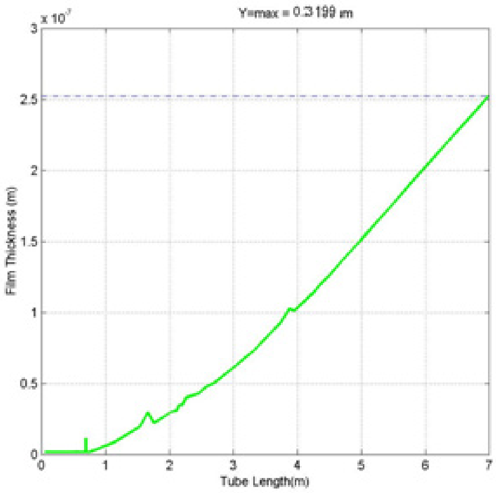

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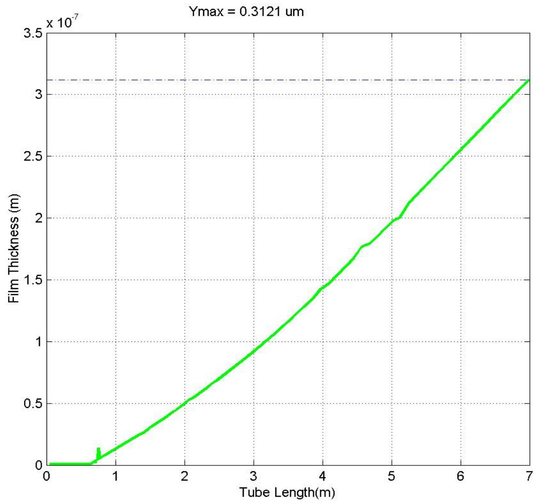

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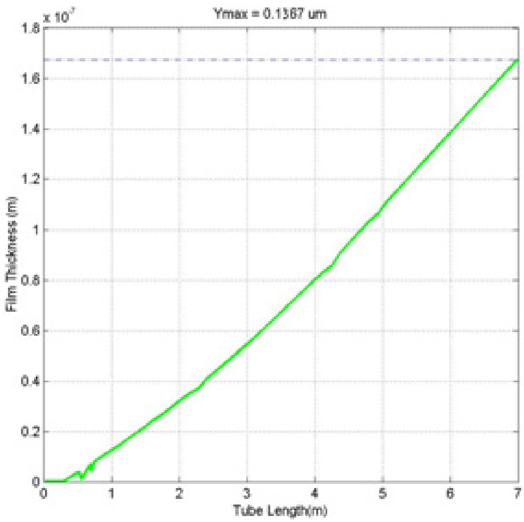
Krytox 226 (μ = 0.0445 Pa.s)



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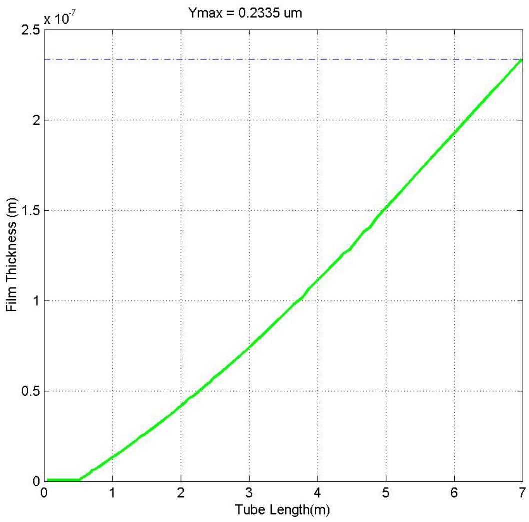

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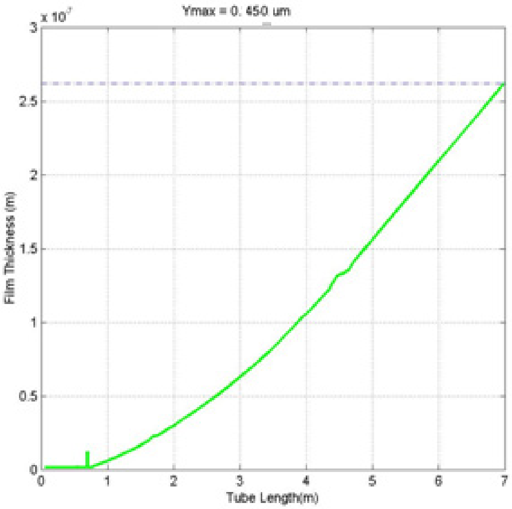

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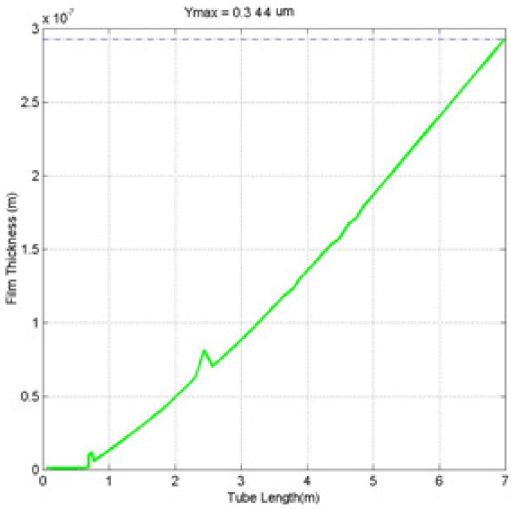
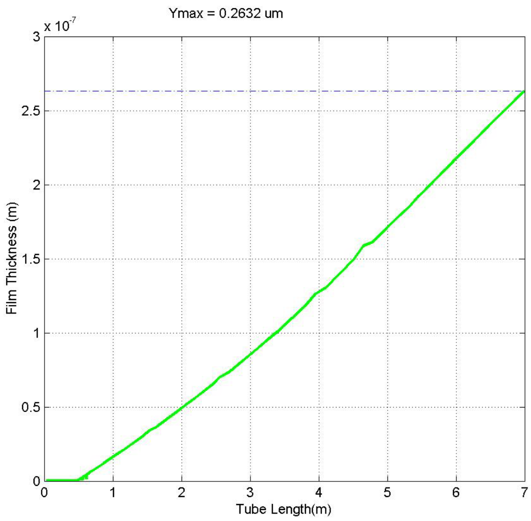


Figure 4. Clearance = 0.00004

Krytox 227 (μ = 0.07476 Pa.s)



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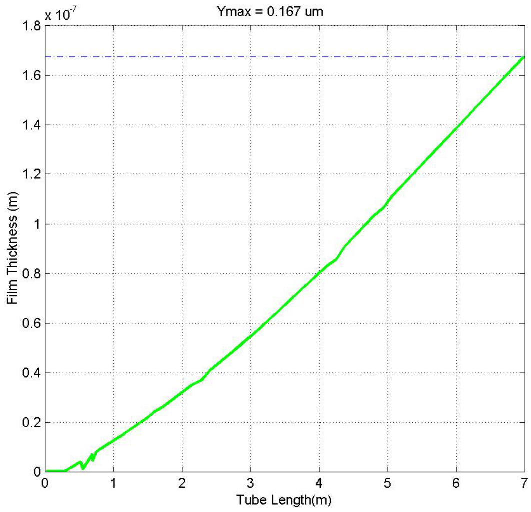

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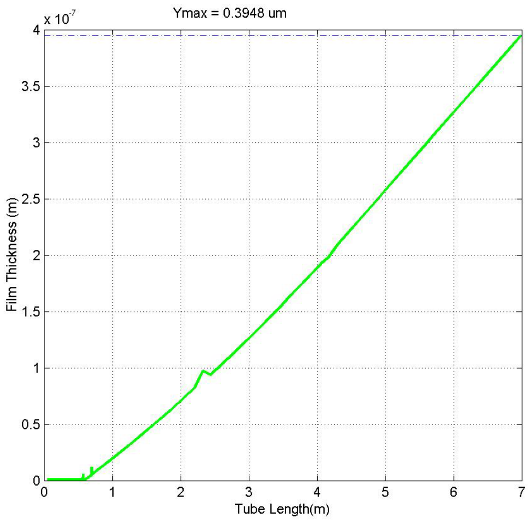

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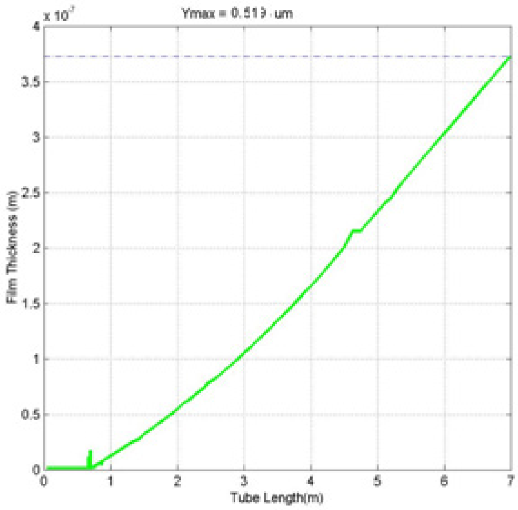

Figure 4. Clearance = 0.00004