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A CFD STUDY OF THE EHL LINE CONTACT PROBLEM WITH CONSIDERATION OF THE SURFACE ROUGHNESS UNDER VARIED LOADS

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ABSTRACT

Traditionally, the Reynolds equation is widely used to describe the flow of lubricants for the elastohydrodynamic lubrication (EHL) problem, though there are a number of limitations for this approach. In this work an advanced computational fluid dynamics (CFD) model has been developed for such EHL problem. The CFD model developed can predict the characteristics of fluid flow in the EHL problem, taking into consideration of the pressure distribution, minimal film thickness, viscosity and density changes. The cylinder is considered to be an elastic deformation according to the theory of Hertzian contact. Above all, the surface of the cylinder is defined to have an arbitrary roughness.

Reconstructing the object geometry, meshing and calculating the conservation of mass and momentum equations are carried out by using the commercial software packages ICEMCFD and ANSYS Fluent. In addition, the user defined functions (UDFs) for density, viscosity and elastic deformation of the cylinder as the function of pressure needs to be defined for this particular work. A number of simulation cases have been investigated, and detailed results of velocity, pressure and film thickness distributions are obtained. In particular, the effects of surface roughness on the EHL line contact problem are compared to the smooth surface case when the applied load is varied. It is found that the pressure profile at the center of the contact area directly relates to the roughness amplitude and the applied load. The roughness surface influences the fluctuated shape of pressure distribution. The pressure and the effect of surface roughness increase when the applied load is increased. At the same time, the film thickness of the lubricant will decrease with increasing the magnitude of the fluctuated pressure.

INTRODUCTION

The lubrication system is the heart of all mechanical machines and should be considered for constant improvement, as lubrication can reduce not only the friction force that is the cause of wasted energy, but also the wear that occurs on the contacting moving parts. Surface roughness has significant effects on the performance of lubrication. If the film thickness of a lubricant is lower than the surface roughness of the bearing, direct contact of both surfaces will occur, leading to a high friction coefficient and wear rate. Therefore, the study of surface roughness effects on the minimum film thickness is important for improving or solving the lubrication problems.

The EHL is defined as a thin film lubrication which takes into consideration the elastic deformation of materials under the action of high pressure. The EHL is developed from hydrodynamic lubrication by adding the function of elastic deflection in the film thickness equation. Conventionally, the fluid flow in the EHL problem has been represented by the Reynolds equation [1], developed by integrating the Navier-Strokes equations across the film thickness. In the last decades, the Reynolds equation has been the main equation used, improved by combining the film thickness, the viscosity equation, the density equation, the energy equation and the load balance. Many techniques are proposed to solve the Reynolds equation as a second-order non linear partial differential equation, which is difficult to solve by iterative method. However, it can be solved by applying the Newton-Raphson method [2] and the multigrid technique [3].

The effect of surface roughness on hydrodynamic lubrication problems was studied by creating a model with a general roughness pattern. Most roughness models are included in the film thickness equation. Then, in 1992, Venner and Napel [4] studied the effects of surface roughness on the EHL problem. They determined the roughness profile by measuring the actual surface of the material. They found that the surface texture significantly influences the pressure profile and the film thickness.

In 2002, a commercial CFD code, which is based on the full momentum equation, the continuity equation and the energy equation, was used to simulate the EHL line contact problem proposed by Almqvist and Larsson [5]. They compared the solution of an EHL problem from the Reynolds equation with that from using CFD techniques that are currently used in general commercial CFD software. The full momentum and continuity equations were calculated separately. The results of both methods have good agreement, though a small deviation is found in the case of thin film thickness. The full model of fluid flow was developed by using the CFX4 software [6], which has included the effects of temperature, surface roughness and time dependence. In the model, the fluid is considered as non-Newtonian and the upper surface is assigned to be sinusoidal roughness, while the lower plate is defined to a smooth surface. The results showed that the temperature and the surface roughness have significant influences on the pressure and film thickness.

In 2008, Hartinger et al [7] presented the CFD modelling of the thermo-elastohydrodynamic lubrication (TEHL) line contact problem. The shear in the thin film is studied and compared with the Reynolds equation approach. The free software OPENFOAM was used to solve the TEHL problem. The cavitation effect, which is a limitation for the Reynolds equation, is considered by modifying the pressure function to include the effect of the lubricant density. The results between the Reynolds equation and CFD model are similar, and there is only a small difference in the case of high viscosity.

Recently in 2011, Bruyere et al [8] presented the CFD model and full elastic model for an EHL sliding line contact problem. The finite-element method was used to discretise the Navier-Stokes equation in the form of linear equations. In addition, the Arbitrary-Lagrangian-Eulerian (ALE) method is used to control the mesh movement in conjunction with the particle and fluid location. This CFD model is more advanced than the Reynolds equation, providing solutions where the Reynolds equation cannot be used.

The generalised Reynolds equation is well understood, but is limited in the case of cavitations. In addition, the pressure is assumed to be constant across the film thickness in the Reynolds approach [9]. These reasons have led to attempts to use finite volume analysis to simulate and analyze the EHL problem. The main objective of this work is to develop an advanced CFD model, and use this model to study the effects of surface roughness on the EHL line contact problem. Some additional variables have to be applied in the CFD model such as the density equation, the viscosity equation and the film thickness equation, similar to the Reynolds approach.

NOMENCLATURE

E	[Pa]	Reduced modulus of elasticity
f	[-]	Volume fraction
h_o	[m]	The minimum film thickness
h_i	[m]	Film thickness
I	[-]	Unit tensor
L	[Nm^{-1}]	Applied load per unit width
p	[Pa]	Pressure
p_{sat}	[Pa]	Liquid saturation vapour pressure
R	[m]	Reduced radii of curvature
$R(i)$	[m]	Surface roughness term
SRR	[-]	Slide to roll ratio = $2(u_p-R_cw_c)/(u_p+R_cw_c)$
Δt	[s]	Time step
u	[m/s]	Velocity
V_{ch}	[-]	A characteristic velocity
x_i	[m]	Cartesian axis in *i* direction
z	[-]	Viscosity index

Special character

ρ	[kg/m^3]	Density
τ	[Pa]	Viscous stress tensor
η	[Pa·s]	Viscosity of Newtonian fluid
η_o	[Pa·s]	Viscosity at ambient pressure
ξ	[-]	Coordinate transformed range for numerical integration
σ	[kg·s^{-2}]	Surface tension of the liquid
ϕ	[-]	The net rate of flow in a fluid element

Subscripts

a	Average
c	Cylinder
p	Plate
in	At the inlet position
out	At the outlet position
i	At any node
o	Ambient or reference
l	Liquid phase
v	Vapour phase
m	Mixture phase
sat	Saturation vapour pressure
e,w,n,s	The neighbour cells of cell *P* in the east, the west, the north and the south direction

THEORY

Governing equations

The CFD approach is used to calculate the velocity and pressure of fluid flow in the EHL problem instead of the Reynolds equation. The characteristics of fluid flow can be explained by the conservation form of the fluid flow. This combines the continuity equation and momentum equation which can be written in the general form:

Continuity equation:

$$\frac{\partial \rho}{\partial t}+\nabla\cdot\left(\rho\vec{u}\right)=0 \tag{1}$$

Momentum equation:

$$\frac{\partial\left(\rho\vec{u}\right)}{\partial t}+\nabla\cdot\left(\rho\vec{u}\vec{u}\right)=-\nabla p+\nabla\cdot\tau \tag{2}$$

where $\tau=-\eta\left(\nabla\vec{u}+\left(\nabla\vec{u}\right)^T\right)+\frac{2}{3}\eta\bar{I}\nabla\cdot\vec{u}$

Film thickness equation

The distribution of film thickness is depended on the physical geometry of cylinder and the elastic deformation of contact which is governed by the pressure distribution over the contact [10].

Generally, the surface texture is considered in the EHL line contact problem by including the surface roughness in the equation of film thickness. Therefore, the film thickness equation can be written as

$$h_i=h_o+\frac{x^2}{2R}+R(i)-\frac{2}{\pi E}\int_{-\infty}^{x_b}p\left(\xi\right)\ln\left(x-\xi\right)^2d\xi \tag{3}$$

Load balance equation

In order to simulate the EHL problem, the gap between the cylinder and the bottom plate has to be corrected in each iteration and updated until the generated pressure is equal to the applied load:

$$h_o^{new}=h_o^{old}+defect \tag{4}$$

where

$$defect=L-\int_{-\infty}^{\infty}p_i dx$$

Density equation

The density of fluid is affected when the pressure of lubricants changes. There is a linear variation between pressure and density, as Dawson and Higginson presented [11]. The relation between pressure and density can be written as:

$$\rho_i = \rho_o \left(1 + \frac{0.59\times10^{-9} p_i}{1+1.7\times10^{-9} p_i} \right) \tag{5}$$

Viscosity equation

The viscosity of fluid depends on the pressure, as proposed by Roelands [12]. The Roelands model is approximately used to describe the behavior of Newtonian fluids as given below:

$$\eta_i = \eta_o \exp\left\{ [\ln \eta_o + 9.61]\left[-1 + \left(1 + 5.1\times10^{-9} p_i\right)^z \right] \right\} \tag{6}$$

Cavitation model

The Reynolds equation does not model cavitation region. All negative pressures calculated at the outlet are set to zero. However, the negative pressure can be solved by using a cavitation model, in conjunction with the CFD model for the EHL problem. The full cavitation model [13] used in this research is:

$$\frac{\partial(\rho_m f)}{\partial t} + \nabla(\rho_m \vec{v}_v f) = \nabla(\gamma \nabla f) + A - B \tag{7}$$

where A and B are given by

$$A = 0.02 \frac{V_{ch}}{\sigma} \rho_l \rho_v \sqrt{\frac{2(p_{sat} - p)}{3\rho_l}} (1 - f) \tag{8}$$

$$B = 0.01 \frac{V_{ch}}{\sigma} \rho_l \rho_v \sqrt{\frac{2(p - p_{sat})}{3\rho_l}} (f) \tag{9}$$

The density of a lubricant for a liquid phase is a function of pressure as defined in equation (5). Thus the density of the vapour phase should be calculated from the fraction equation. Therefore the density of a mixture phase can be written as:

$$\rho = \alpha_v \rho_v + \alpha_g \rho_g + (1 - \alpha_v - \alpha_g)\rho_l \tag{10}$$

Geometry and boundary conditions

In this research, the ICEMCFD software is used to create the CFD model of the EHL problem, as shown in Figure 1. Then meshes are generated in the geometry. The CFD model is solved using the commercial software ANSYS Fluent. The surface of the cylinder is assumed to be rough, while the surface of the plate is comparatively smooth.

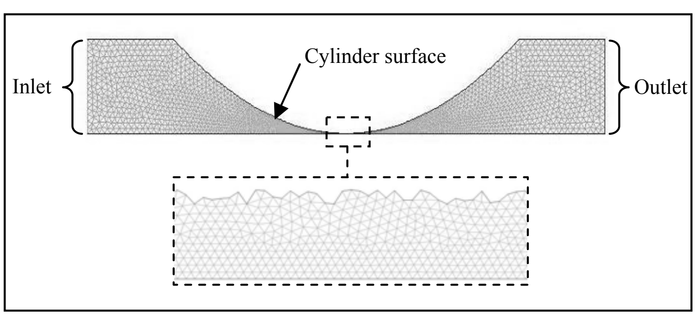

Figure 1 The CFD model for the EHL problem

In the numerical simulation, it is very important to specify the correct initial condition and boundary conditions; poorly defined boundaries would result in an incorrect simulation. In order to simulate the EHL line contact problem, the boundary conditions at the inlet and outlet are given as:

- Inlet, P=0 Pa at X=X_{in}
- Outlet, P=0 Pa at X=X_{out}

The parameters used in the simulation are listed in Table 1.

Table 1: Common parameters

Parameter	Value	Unit
Load	1.3×10^5	N/m
The velocity of plate	2.5	m/s
The angular velocity of ball	0	rad/s
Cylinder radius, R_c	10	mm
Roughness surface of plate, R_a	0.00125	μm
Modulus of elasticity of plate , E_p	345	GPa
Modulus of elasticity of cylinder, E_c	345	GPa
Time step, Δt	0.001	Sec.
Lubricant		
- Liquid dynamic viscosity, μ_l	0.02	m^2/s
- Vapour dynamic viscosity, μ_v	8.97×10^{-6}	m^2/s
- Liquid density, ρ_l	865	kg/m^3
- Vapour density, ρ_v	0.0288	kg/m^3
- Viscosity pressure index, z	0.55	-

NUMERICAL METHOD

The finite-volume method is used in this study. The integral form of the transport equation (1, 2) has to be applied to a discretized equation. The geometry domain is subdivided into a finite number of subdomain as shown in Figure 1.

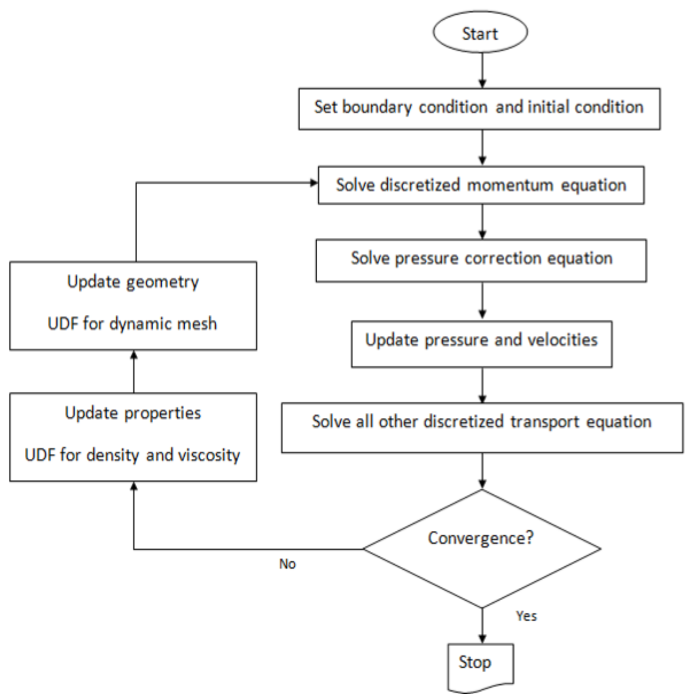

Figure 2 The flow chart for solving the EHL problem of the CFD model

The variables in the domain are calculated by using approximating methods, the initial and boundary conditions need to be defined. Then the transport equation in general form can be used to solve iteratively for all cells in the domain, as demonstrated below:

$$\left[\frac{(\rho\phi)^{new}-(\rho\phi)}{\Delta t}\right]\Delta V+\left[(\rho u\Delta A)\phi\right]_e-\left[(\rho u\Delta A)\phi\right]_w+\left[(\rho u\Delta A)\phi\right]_n$$
$$-\left[(\rho u\Delta A)\phi\right]_s=\left[\left(\Gamma\frac{\partial\phi}{\partial y}\right)\Delta A\right]_n-\left[\left(\Gamma\frac{\partial\phi}{\partial y}\right)\Delta A\right]_s+\left[\left(\Gamma\frac{\partial\phi}{\partial x}\right)\Delta A\right]_e-\left[\left(\Gamma\frac{\partial\phi}{\partial x}\right)\Delta A\right]_w \qquad (11)$$

In order to start the iteration process, the velocity and pressure fields are approximated for the first iteration. Then these parameters are used to solve the momentum equation and the pressure correction equation. These values will be corrected in each iteration, until acceptable convergence of pressure and velocity is achieved as shown in Figure 2.

RESULTS AND DISCUSSION

At this stage of the research, a 2D CFD model is created for the proposed EHL problem. The cylinder is assumed to be infinitely long and rotates on the plate, thus the pressure distribution in the z direction can be considered to be constant. Therefore only the pressure distribution in the front plane is investigated. To identify the roughness effects, a smooth surface case is considered first, then the second case of the roughness surface is modeled. The boundary conditions and geometry model are maintained the same in both cases.

1. The case of smooth surface

The aim of this case is to study the characteristics of fluid flow of the CFD model. The pressure, velocity, viscosity and density of the fluid are investigated. The pure sliding is considered in this case. The plate moves at 2.5 m/s in the x-direction meanwhile, the cylinder is stationary (*SRR=2*). The initial gap between the cylindrical surface and plate is 0.0001 mm. The fluid property is considered to be isothermal and Newtonian fluid. The results are shown in Figures 3-10.

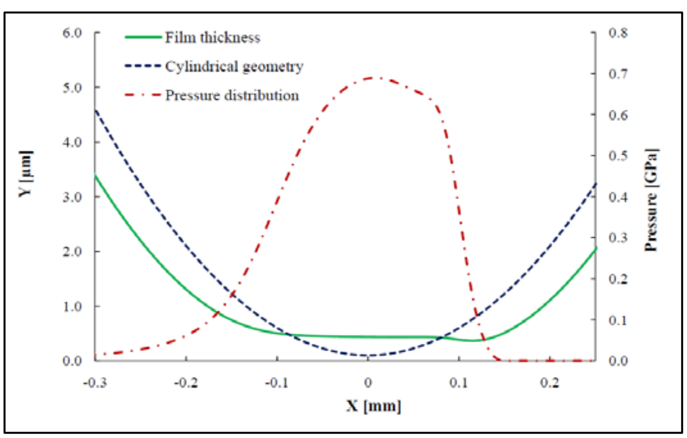

Figure 3 Pressure distribution and film thickness shape

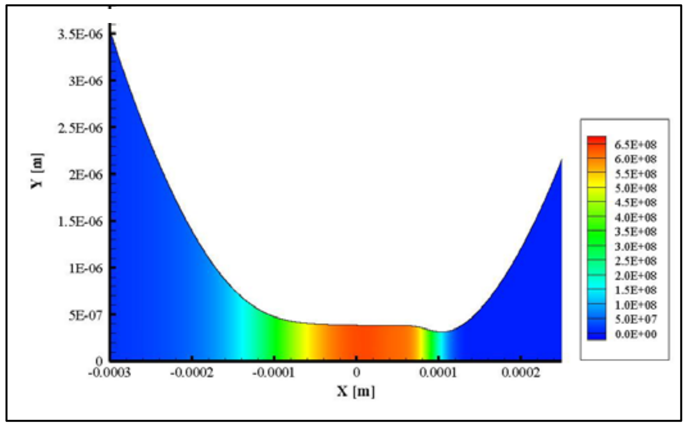

Figure 4 The contours of static pressure [Pa]

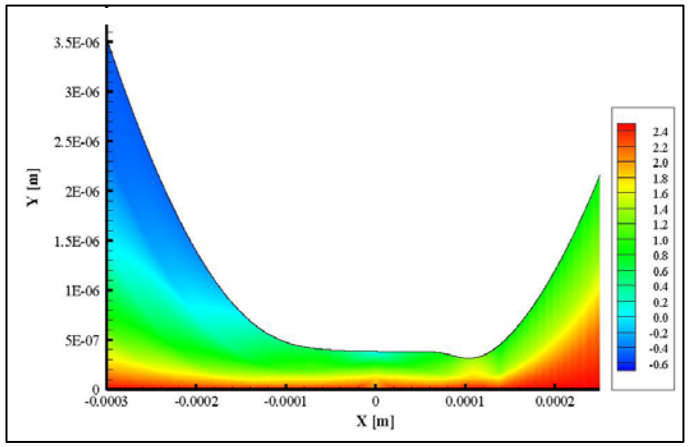

Figure 5 The contours of lubricant velocity [m/s]

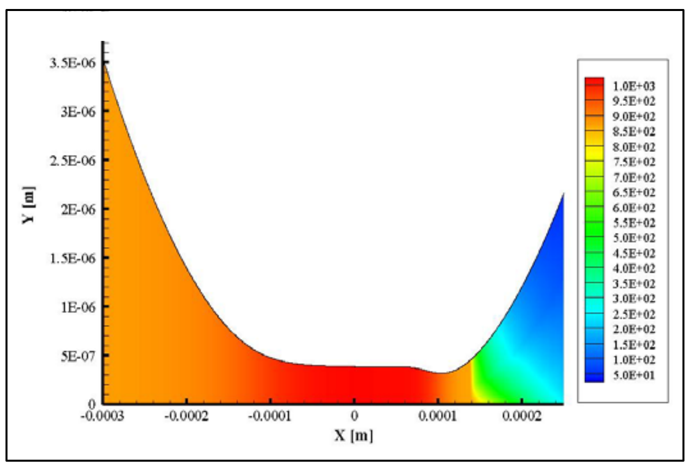

Figure 6 The contours of density [kg/m^3]

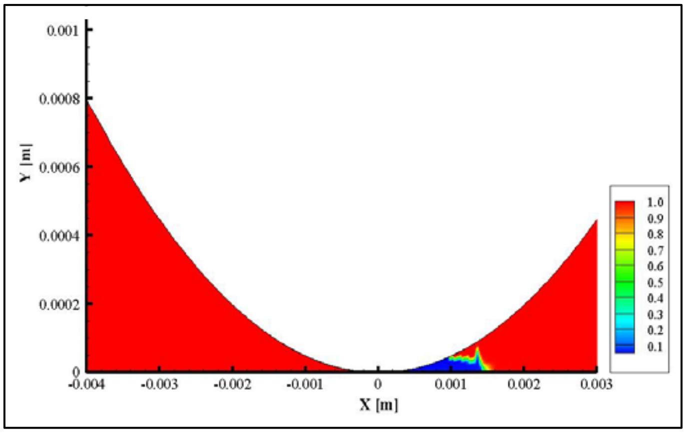

Figure 7 The contours of volume fraction (liquid phase)

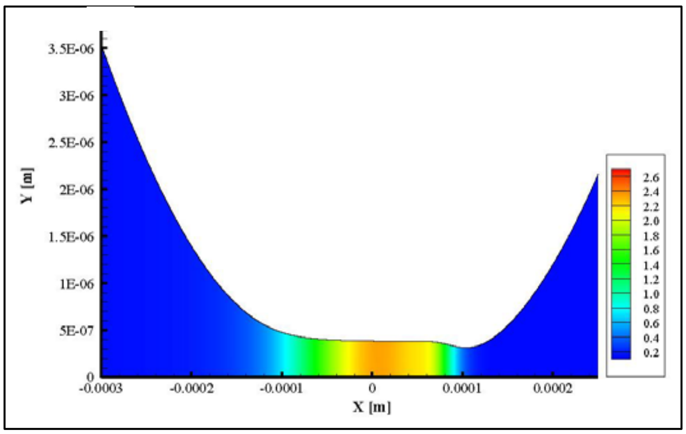

Figure 8 The contours of viscosity of the mixture phase [Pa· s]

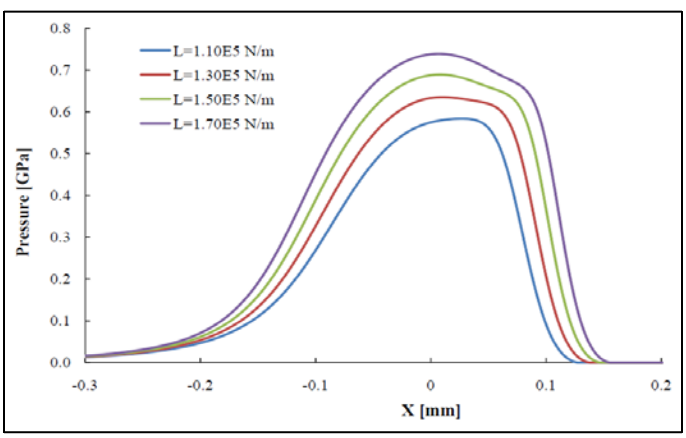

Figure 9 The pressure distribution with varied loads

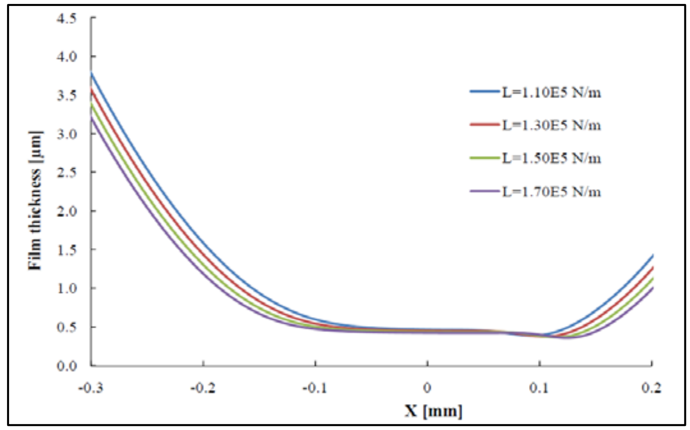

Figure 10 The film thickness with varied loads

2. The case of roughness surface

The aim of this case is to study the effect of surface roughness of the CFD model on the EHL problem. In this case, the boundary conditions are assumed to be the same as the first case, but the surface of cylinder is defined to be rough as shown in Figure 1. The computed results are presented in Figures 11-18.

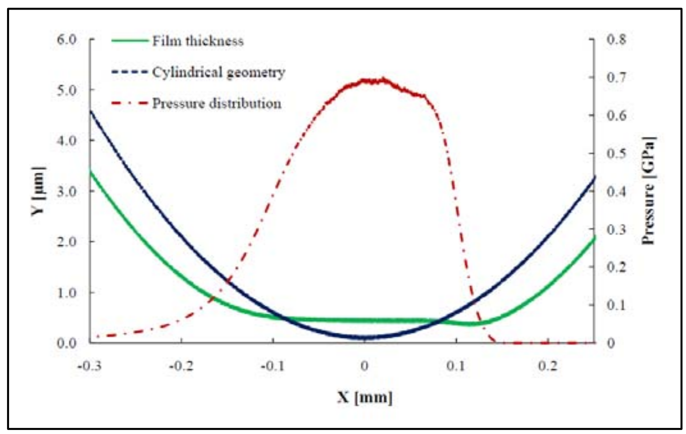

Figure 11 Pressure distribution and film thickness shape

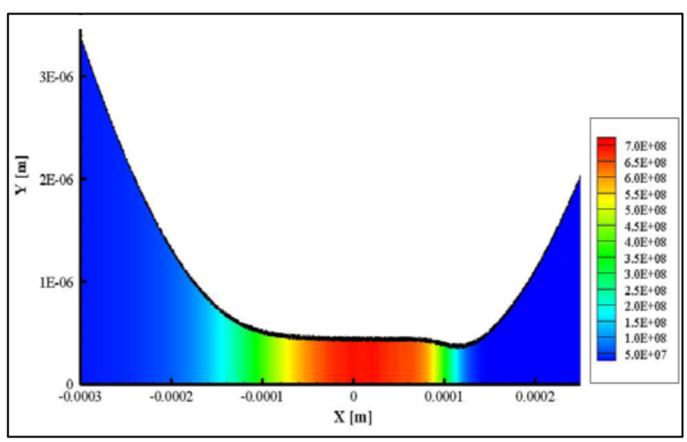

Figure 12 The contours of static pressure [Pa]

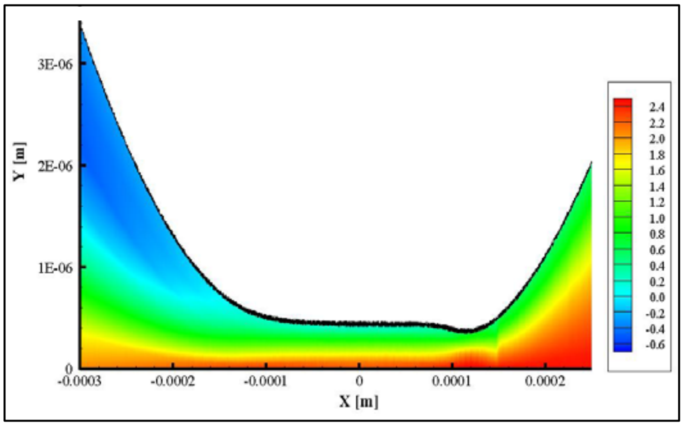

Figure 13 The contours of lubricant velocity [m/s]

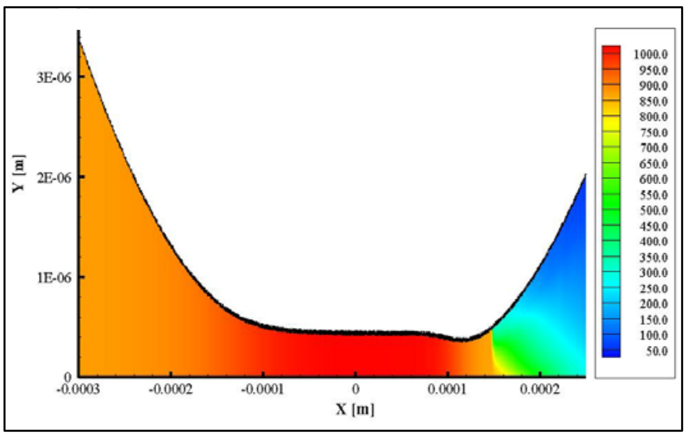

Figure 14 The contours of density [kg/m^3]

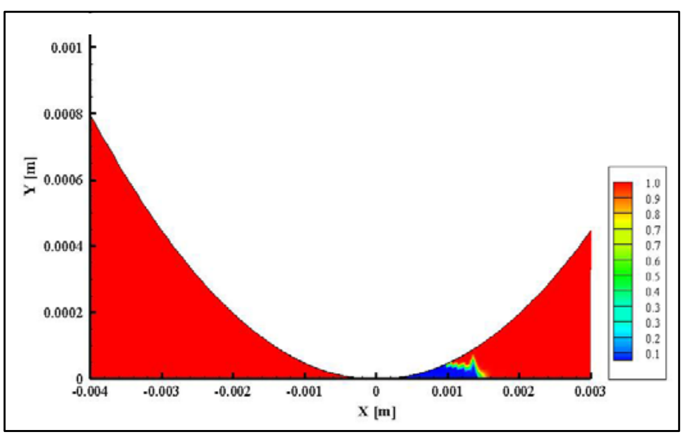

Figure 15 The contours of volume fraction (liquid phase)

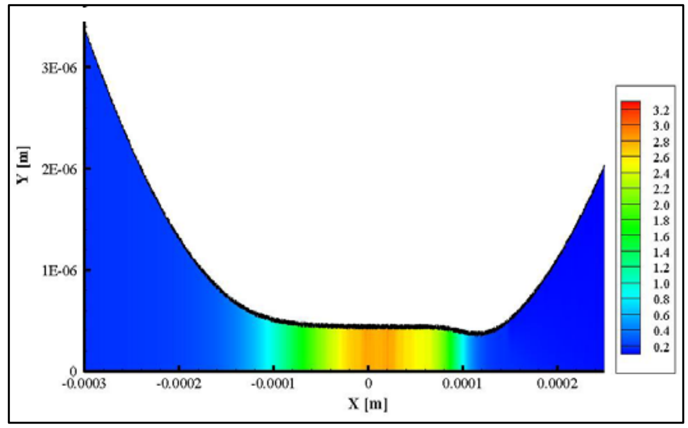

Figure 16 The contours of viscosity of the mixture phase [Pa·s]

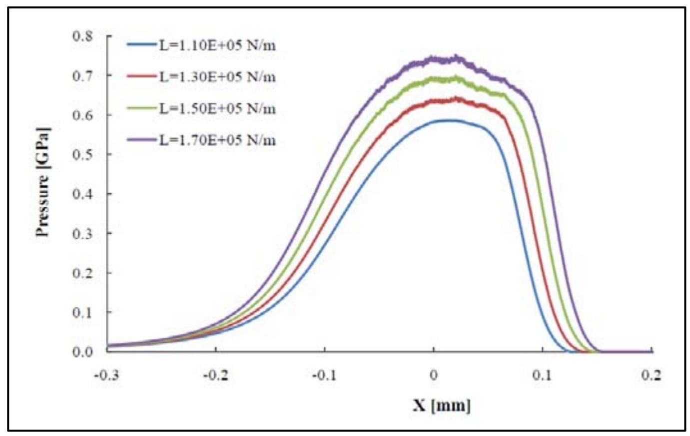

Figure 17 The effects of surface roughness on the pressure distribution with varied loads

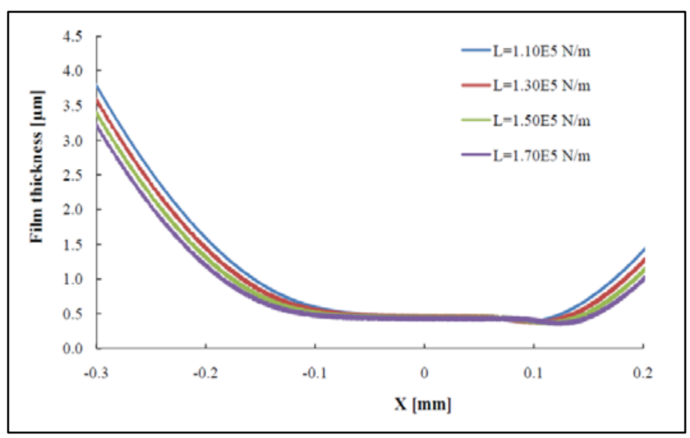

Figure 18 The effects of surface roughness on the film thickness with varied loads

Figures 3 and 11 present the elastic deformation of the cylindrical geometry and the pressure distribution at the contact area for the smooth surface case and the roughness surface case, respectively. The pressure distribution in the case of roughness surface is higher than the case of smooth surface around 0.05 GPa. This is also confirmed in Figures 4 and 12.

The contours of lubricant velocity of both cases have similar trend as presented in Figures 5 and 13. The velocity gradient is very high at the layer near the surface of the plate which is moved at 2.5 m/s. Then it decreases slightly when the film thickness increases.

The density property of lubricant, as shown in Figures 6 and 14, is maximum at the contact area zone, but it dramatically decreases from 1000 kg/m^3 to 100 kg/m^3 as the lubricant flows though the minimum film thickness. The effect of the cavitation model on the pressure distribution is shown in Figures 7 and 15. It can be seen that the negative pressure has been removed when the cavitation model is applied with the CFD model. This is the result of partial fluid phase changes from liquid to vapour when passing though the contact area, because the liquid is subjected to rapid expansion. Downstream of the passage throat, the fluid is transformed back to the liquid phase due to increased pressure.

The contours of viscosity of lubricant in Figures 8 and 16 have trends similar to the static pressure, because the viscosity of lubricant is the function of pressure in each node. In the case of roughness surface, the viscosity is higher than the smooth surface case around 0.6 m/s^2 and reaches the highest at x=0 m.

It can be clearly seen from Figures 9 and 17 that the roughness surface influences the fluctuated shape of pressure distribution, and the pressure increases when the applied load is increased. Meanwhile Figures 10 and 18 indicate that the effect of surface roughness on the film thickness will increase when the applied load is increased. This is due to the fact that the film thickness in the heavy load case is thinner than the light load case.

CONCLUSION

In this research, the CFD approach for the EHL line contact problem has been developed using the ICEMCFD and ANSYS Fluent software. This CFD model has been applied to isothermal cases with surface roughness effects under varied loads. The surface roughness significantly influences the pressure distribution in the case of thin film in the following ways:

- The pressure at the contact area in the roughness surface case is higher than the smooth case, while the film thickness is lower than the smooth surface case.
- The roughness surface effect on pressure distribution increases when the applied load is increased.
- The trend of the fluctuated pressure is similar to the roughness profile.

However, if the viscosity of lubricant is very high at the centre of the contact area, especially in the case of pure sliding, the thermal effect should be considered with respect to the viscosity and density of lubricant properties. The related simulation results will be reported somewhere else.

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