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Investigation on Grease Elastohydrodynamic Lubrication

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Abstract

Lubrication as a friction reduction technique has been used in variety of mechanisms and machines. The contacting surfaces depart from each other by the pressure produced in lubricant due to surface wedge shape. In some applications pressure is such high that deforms the contacting surfaces and more space is provided for the lubricant. Increasing the film thickness leads to misestimating of friction coefficient. Greases are usually used when the lubricating area is unreachable easily or there is not enough space for oil recirculation. In this paper, Herschel-Bulkley's model was used for isothermal non-Newtonian grease lubrication under point contact elastohydrodynamic condition. A good agreement between experimental and simulation results is shown. The effect of different type of grease is compared according to lubricant film thickness and friction coefficient. Results show that threshold yield stress does not significantly affect tribological parameters but the power law exponent does. Higher load and lower entraining velocity cause thinner film as well as higher friction coefficient in spite of the grease type.

Nomenclature

α	Lubricant constant	β	Lubricant constant
ϕ	Plastic viscosity (Pa.s)	ϕ_0	Viscosity at standard condition
n	Power law exponent	θ	Temperature (°C)
p	Pressure	ρ	Density, (kg/m3)
R	Curvature radius, (m)	t	Time, (s)
τ	Shear stress, (Pa)	τ_0	Threshold yield shear stress, (Pa)
u	Velocity in x direction, (m/s)	u_b	Surface velocity, (m/s)
u_p	Center core velocity, (m/s)	v	Surface deformation, (m)
h	Film thickness, (m)	h_0	Central thickness of lubricant film, (m)
h_p	Plug flow thickness, (m)	x	Coordinate, (m)
y	Coordinate, (m)	z	Coordinate, (m)

1. Introduction

Lubrication is a way of preventing friction and wear of moving surfaces that contact together. If the contacting surfaces have non-conformal geometry (the center of curvature of the two surfaces is not on one side of the contact line), a large load is applied to the small contact area, thereby increasing the contact area pressure. As a result of this pressure, the surface undergoes elastic deformation. In such cases, the total elastic

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deformation is as large as the thickness of the lubricant film, hence it is called elastohydrodynamic regime. In elastohydrodynamic lubrication, characteristics such as surface hardness, non-conformality radius, and the rate of change of viscosity with pressure are important. The high viscosity of the lubricant and the shortness of happening time make it impossible for the lubricant to escape through the gap of mating surfaces, so the surfaces are separated from each other. Until the operating conditions such as velocity, load and temperature do not change drastically, the oil starvation and direct surface-to-surface contact do not occur.

In most EHL analyses, the lubricant is considered as Newtonian fluid. However, in the case of a high-velocity roller bearing, the effects of non-Newtonian viscosity are remarkably effective and are required for more accurate prediction of the minimum film thickness. Grease is a lubricant that uses filler in its structure to adhere to moving parts and is not separated from the work piece by the force of gravity or under operation pressure. Lubrication with grease is semi-permanent; this is an advantage when it hard to access the bearing or lubrication area of machines, such as motors mounted on roofs, drive lines, and etc.

The most important issue in the grease lubrication from the point of view of modeling and estimation of operation conditions is its non-Newtonian behavior, especially under high contact pressure. This non-Newtonian behavior can completely change the predictions that result from conventional modeling methods. Therefore, modeling and examining lubrication conditions can empirically lead to the increased accuracy of models and better understanding of this problem.

Numerous attempts have been made to numerically model the elastohydrodynamic lubrication with grease and several parameters such as lubrication layer squeeze, cavitation boundary condition, grease type and lubricant layer degradation [1–5]. Although grease's shear stress at a specified shear rate is not lower than that of its base oil, the thickness of the grease layer can be less than that of the base oil, due to fluid heat flow. Jonkisz and Krzeminski-Freda [6] numerically confirmed that extrapolated values of lubricant film thickness in elastohydrodynamic lubrication formulas for base oil also have good approximations when greases are used as lubricants. Cheng [7] also provided comprehensive research on the behavior of greases using a numerical method. He proposed the possibility of solidification of lubricants under stress and discussed its different characteristics with oil flow. Sugimura and Akiyama [8] experimentally investigated this situation by using fluorescent greases on the laboratory and pin on disk apparatus. Lu and Khonsari [9] experimentally investigated an oscillating journal bearing lubricated with grease. The oscillatory motion, as a transient motion, causes a momentary shift of the lubrication regime from the hydrodynamic to the mixture and results in friction increment. They also investigated the effect of grease lubrication in journal bearings in an earlier study [10]. The effect of parameters such as load, lubricant type, and bearings material were investigated. The results indicated that the lubricant layer was too thin and the lubrication regime was mixed. In a relatively new experimental study, Cousseau et al. tested three different types of grease with a pin-on-disk laboratory apparatus to find a relationship between the thickness of the grease layer and its properties [11]. In recent years, more attention has been paid to grease lubrication. Zhang and et al. investigated a numerical model of grease lubrication in roller bearing element to provide an estimation of friction force and film thickness [12]. Wang and et al. studied the elastohydrodynamic lubrication of the grease-lubricated surface experimentally to improve the film forming capacity [13]. They found that for the aluminum based grease, the fibers tended to accumulate in the middle of the contact area rather than at the edges and the urea-based grease could be easily sheared into smaller particles. Agusto and et al. carried out a numerical study of grease lubricant labyrinth seals and found the proper friction losses equation in this type of seals [14].

In this paper the grease lubrication behavior under point contact elastohydrodynamic condition is investigated experimentally to validate a numerical model. Then the effects of several factors such as grease type, load, and speed on the contact of pin and disk are considered.

2. Grease Pressure Distribution Modeling

Consider the grease flow in the inlet the area of a pair of rollers. As the grease is pulled in by the moving roller surfaces, a complex flow pattern with a forward and backward flow pattern is seen at low pressure in the far at inlet. In the area near the load line where the pressure is significant, a flow velocity profile is formed exclusively forward. In Hertzian pressure region, the flow is assumed to be parallel. Along the center line of the flow field, where shear stress is less than that of lubricant threshold yield stress, a constant velocity is seen at each transverse section known as the plug flow area. The plug flow around the center line of the rollers is where the pressure gradient is zero. To model the grease flow, we need to modify the governing relationships of lubrication known as the Reynolds equation [1] in order to take into account the grease plug flow and pseudoplastic behavior. The grease lubricant rheological behavior is expressed on the basis of Herschel-Bulkley's three-parameter model [2]:

$$\tau = \tau_0 + \phi \left| \frac{\partial u}{\partial z} \right|^n \tag{1}$$

where τ_0 is threshold yield shear stress, ϕ is called plastic viscosity, and n is power law exponent. For theoretical analysis, the flow is assumed to be incompressible, slow, steady and isothermal. Due to the thin layer of lubricant, the inertial forces are negligible and in the grease layer the balance between the compressive and viscous forces is in equilibrium. If the gap of contact between the two mating bodies and the lubricant between them is assumed to be the same as the schematic shown in Fig. 1, and h denotes the thickness of the lubricant layer, ρ the lubricant density and p the lubricant pressure, Karthikeyan et al., Modified Reynolds equation as relation (2) [15]:

$$\frac{\partial}{\partial x}\left(\frac{\rho h_a^3}{\phi}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{\rho h_a^3}{\phi}\frac{\partial p}{\partial y}\right) = 12\left[\frac{\partial}{\partial x}\rho\left(u_b h - u h_p\right)+\frac{\partial\left(h\rho\right)}{\partial t}\right] \tag{2}$$

where the plug flow thickness, h_p, is obtained by assuming Bingham's fluid behavior (Eq. (1)) for grease as Eq. (3).

$$h_p = \frac{2\tau_0}{\partial p/\partial x} \tag{3}$$

and $h_a = h - h_p$. The velocity u is the difference between the surface velocity, u_b, and the center core velocity, u_p, which is calculated as follows:

$$u = u_b - u_p = u_b - \left(\frac{\tau_0}{\phi h_p}\right)\frac{1}{4}(h-h_p)^2 \tag{4}$$

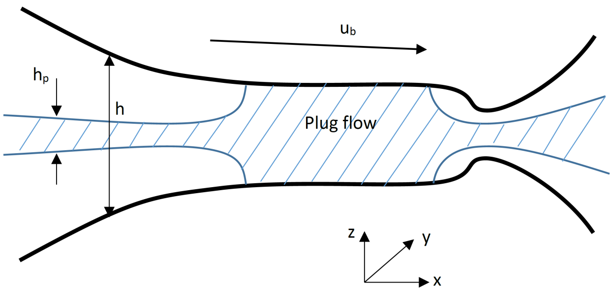

Fig. 1. Contact geometry in elastohydrodynamic lubrication and non-Newtonian grease lubricants.

As shown in Fig. 1, the contact conditions of the non-conformal lubricated surfaces cause very high pressure that results in an elastic deformation of the surfaces. In the simulation, the contact geometry usually considers as two overlapping cylinders and this deformation makes the geometry like two parallel surfaces, and the lubricant flow is somewhat similar to the Couette flow. When two long cylinders are pressed together, their surface deformation is obtained by the following relation:

$$v\left(x\right) = -\frac{4}{\pi E'}\int_{x_i}^{x_e} p(s)\ln|x-s|\,\mathrm{d}s \tag{5}$$

where x_i is the coordinate of the lubricant entrance, x_e is the coordinate of the discharge and E' is the Young's modulus. The lubricant layer thickness relationship is as follows:

$$h = h_0 + \frac{x^2}{2R} + v(x) - v(0) \tag{6}$$

where h_0 is the central thickness of lubricant film.

The influence of pressure in addition to surface deformation also results in the change of the lubricant properties such as viscosity and density. Roelands proposed the following relationship for changing the basic lubricant viscosity [16]:

$$\phi = \phi_0 \exp\left\{(\ln\phi_0 + 9.67)\left[-1+(1+5.1\times 10^{-9}p)^{z'}\right] \times \left(\frac{\Theta-138}{\Theta_i-138}\right)^{-S_0}\right\} \tag{7}$$

where $\Theta = \theta_i + \Delta\theta + 273$ and the temperature of the contact area is in Kelvin and S_0 is a constant, calculated as follows:

$$S_0 = \beta\left(\frac{\Theta-138}{\ln\phi_0 + 9.67}\right) \tag{8}$$

and z' is calculated as following:

$$z' = \frac{\alpha}{5.1\times 10^{-9}(\ln\phi_0 + 9.67)} \tag{9}$$

The following equations are commonly used to vary the density with pressure [16]:

$$\rho = \rho_0\left(1+\frac{0.58p}{1+1.7p}\right) \tag{10}$$

3. Numerical Solution

Since the desired quantity of solving the modified Reynolds equation is pressure and then the thickness of the lubricant layer, the finite difference method is a good option for discretizing this equation. To solve the equations, the pressure distribution is first assumed and then based on this supposal pressure distribution, the surface elastic deformation, lubricant layer thickness, lubricant viscosity and density are calculated. Then the new pressure distribution is obtained by solving the equation set obtained by Newton-Raphson method. The new pressure distribution is compared with the previous pressure distribution. The pressure solution convergency criterion is defined as following

$$\frac{\sum_i\sum_j\left(P_{ij}^{new}-P_{ij}^{old}\right)}{\sum_i\sum_j P_{ij}^{old}} \leq 1\times 10^{-6} \tag{11}$$

The boundary conditions are set as zero pressure at far inlet and far outlet. In EHL problem usually four times of semi-axes of hertzian contact ellipse in inlet and outlet is a proper choice for solution domain. If their difference is in the convergence range, the final obtained solution is tested in the load Eq. (12).

$$W = \int_{x_i}^{x_e} p(x)\,\mathrm{d}x \tag{12}$$

When the load equation is satisfied, the solution is completed. If it is not, the minimum film thickness is modified and the solution restarts. The solution flowchart is shown in Fig. 2.

4. Experimental Setup

One of the elastohydrodynamic contact condition test apparatus used to empirically model two non-conformal surface contact is the Pin-on-Disk device. The pin is selected from a hard, wear-resistant material compared to the disk. In this study, the pin head has a cylindrical cross-section to provide linear contact conditions. The device can measure the amount of frictional force when the pin slides on a rotating disk. It is possible to put a lubricant between the disc and the pin. Then, by modeling the lubricant flow, the frictional force between the lubricant and the surface can be obtained and compared with the experimental data.

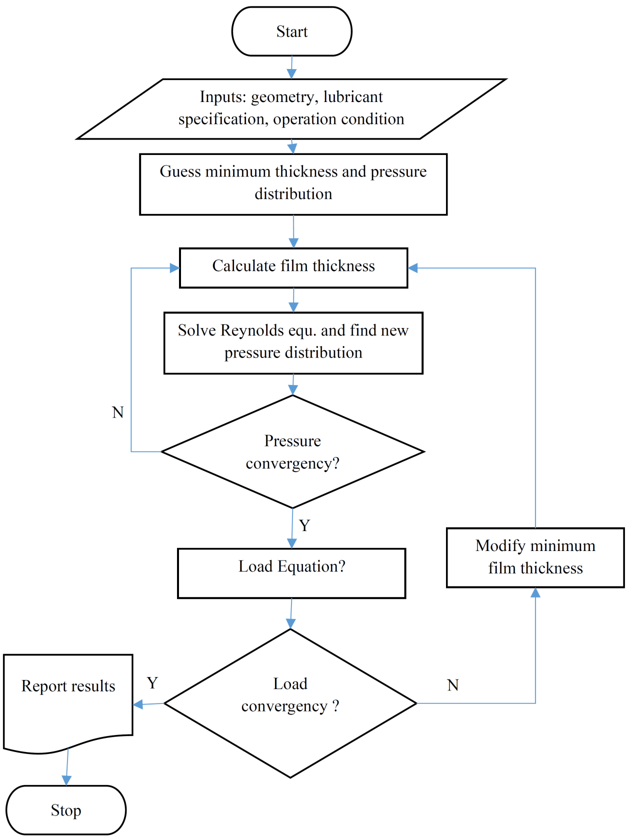

Fig. 2. Numerical method flow chart.

5. Results

The numerical results of the model for the contact of a cylinder with 5mm diameter and a flat rotating disk was compared with experimental data to confirm the accuracy of model. The tests were accomplished in operation condition of load 20N and rotational speed 0.23m/s. The grease used for the tests was lithium based Parsoil product with NLGI grade of 2. The base oil of this type of grease has a viscosity of 310cp at 20°C. Five fully polished disks were tested and the average results are presented here as experimental data. Fig. 3 shows the friction coefficient achieved from experiment, which the mean friction coefficient is achieved as 0.064. Friction force is calculated as following when the lubricant film thickness is determined.

$$F = \int \tau_h \,\mathrm{d}A_h \tag{13}$$

where τ_h is hydrodynamic shear stress and can be numerically found based on Herschel Balkley's model.

Compared to the calculated value from numerical simulation which is equal to 0.055, there is 14% error. In estimation of friction coefficient only the shearing of lubricant film is considered and the interaction of mating surface asperity are ignored. If the film thickness is thick enough that asperity contact probability is too much low, this assumption would be reasonable. For the lubricated contact of non-conformal surface such as pin and disk, high entraining speed and light load form a thick lubricant film at contact area. Therefore, the test we arranged included such an operation condition to better match the results. In the case of mixed to boundary lubrication which asperity contact become significant, lubricant shearing can be neglected with respect to asperity friction.

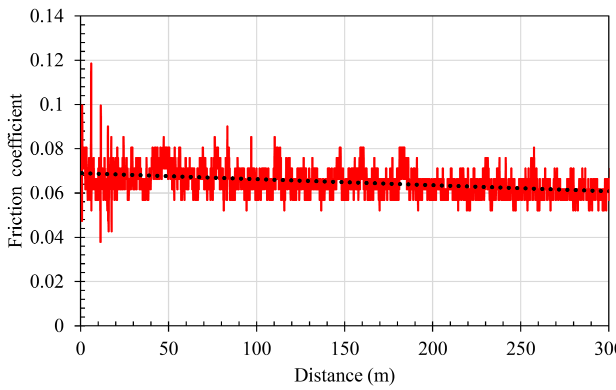

Fig. 3. Experimental results of friction coefficient in 20N load and 0.23m/s velocity condition.

As shown in the Fig. 3 the simulation estimations have a good correlation with experimental data. The error is in the range of 1 to 20 percent with average of 8 percent. Therefore, the proposed model is acceptable in hydrodynamic and elastohydrodynamic lubrication condition which the asperity contact does not occur.

Table 1
Some type of grease and their specification.

Grease No.	Base of grease	Base oil viscosity at ambient temp. (20°C)	τ_0 in Eq. (1)	ϕ in Eq. (1)	n in Eq. (1)	Reference
1	Sodium	0.095	600	1.5	0.84	[2]
2	Barium	0.095	1000	0.55	0.84	[2]
3	Calcium	0.03	230	4.3	0.77	[2]
4	Calcium	0.03	570	15	0.68	[2]
5	Sodium	0.12	1500	269.4	0.47	[4]
6	Calcium	0.075	2800	32.4	0.62	[4]
7	Lithium	0.26	5000	85.19	0.62	[4]

Fig. 4 shows the pressure distribution and film thickness variation at the contact area. The main characteristics of elastohydrodynamic lubrication are surface deformation to form a parallel contact area and very high pressure in order of Hertzian pressure magnitude. At the outlet region of contact area, there is a decreasing pressure which accelerates the flow rate and causes a continuity problem. A closing gap and steep rise of pressure called spike is consequence of flow continuity solution. The minimum film thickness happens at this point, so its location and amplitude are recognized as the most important finding in the flow problem. It is shown that pressure spike is a feature of viscosity variation with high pressure and there is not any spike in an isoviscous flow assumption [17]. Compressibility condition increases the magnitude of spike amplitude with respect to the fixed density solution. Detection of pressure spike in numerical results need a fine grid domain especially for compressible fluid.

As shown in Fig. 4, the flat contact area, closing gap, and pressure spike are detected properly and it shows that simulation details such as grid size, discretization, solution algorithm and numerical findings are acceptable.

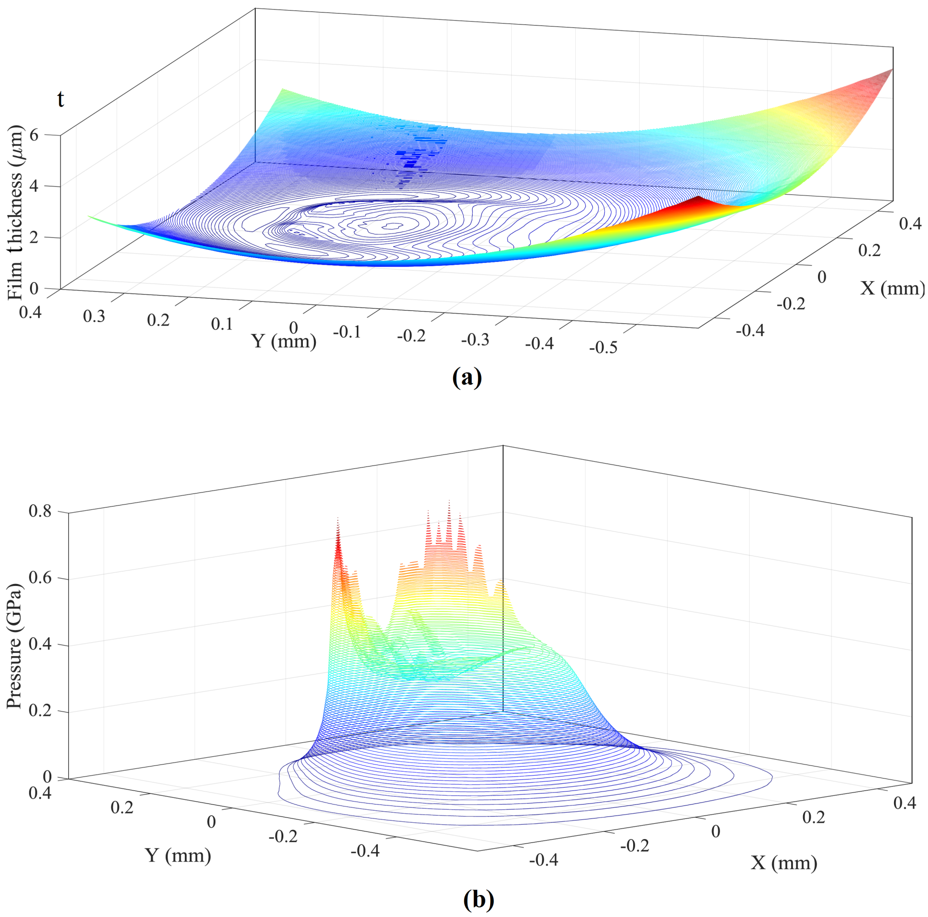

Fig. 4. The pressure distribution and film thickness in contact area for 200N load and 0.8m/s velocity.

Since the accuracy of the model is validated, the effect of various parameters such as grease type, operational speed, and load can be investigated. The rheology equation of several types of greases with different bases according to the Herschel-Bulckly model presented in different references is given in Table 1.

Fig. 5 compares the friction coefficient and minimum film thickness for various type of lubricants that is listed in Table 1. The difference between threshold yield stress of greases is considerable but their performance does not have any relation to this stress. For instance, grease 2 has a yield stress 1.5 times larger than that of grease 1, but their average and minimum film thickness and friction coefficient is definitely similar. Grease number of 6 and 7 show almost the same outputs while their yield stress is different. Kauzlarich and Greenwood also confirm the low dependency of film thickness and grease yield stress [5]. It is obvious that the main parameters that have a significant effect on the results are base oil viscosity, plastic viscosity, and rheological exponent. Grease number of 1 and 4 have an almost the same yield stress but different exponent and their outputs are not identical. Grease 3, 4 and 5 have very thin film thickness which for latter small exponent is an effective factor and for two former low base oil viscosity is governing. These observations show that threshold yield stress has a negligible effect on the film thickness. In general, Greases with lower viscosity show the thinner film and higher friction force.

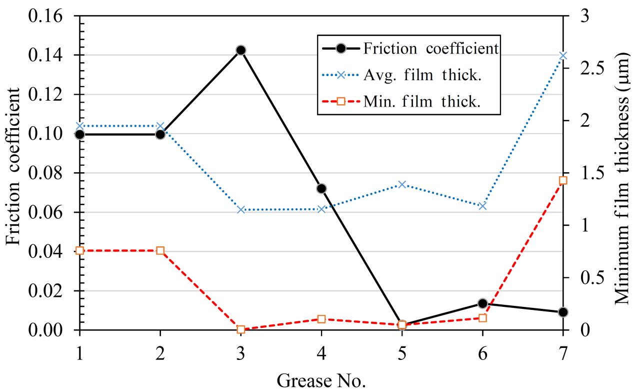

Fig. 5. Friction coefficient and film thickness for various type of lubricants, at 100N load and 1m/s entraining velocity.

Figs. 6 and 7 show the dependency of grease film thickness and maximum pressure on rheological exponent, n, and viscosity, ϕ. The results show that film thickness and maximum pressure is increasing with n and decreasing with ϕ. The influence of grease thickener does not affect film formation estimation due to simplicity of rheological model. Kaneta et al. experimentally found that the grease elastohydrodynamic lubrication films is influenced by the thickener structure and base oil viscosity [18]. Cyriac et al. conducted a series of tests that their results showed that increase in the film thickness due to entrainment of the thickener was proportional to the ratio of thickener volume fraction to the size of the particle [19]. They found that at medium speeds, the grease film thickness is closer and a bit thicker to the base oil film thickness. The increase in film thickness can be modelled by an increase in viscosity. It can be concluded that our simulation is appropriate for the most grease applications due to the fact that proper viscosity model was carefully chosen in this research.

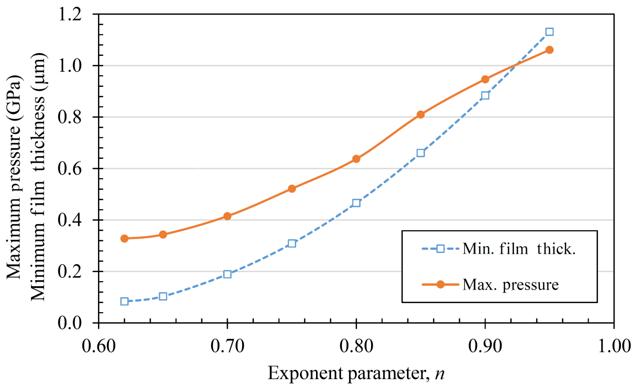

Fig. 6. Grease film thickness and maximum pressure to rheological exponent, n.

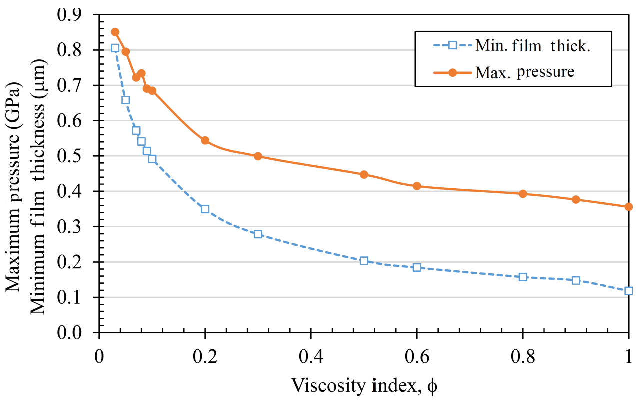

Fig. 7. Grease film thickness and maximum pressure to plastic viscosity, ϕ.

Figs. 8 and 9 show the load and speed effect on film thickness and maximum pressure, respectively. Higher load causes higher pressure and lower minimum film thickness. But the rate of pressure increment is much higher than film squeeze. On the other hand, higher speed results in thicker film and moderated pressure. When the load is high and entertaining velocity is relatively low, the film thickness becomes thinner and asperity contact governs the friction, and hydrodynamic friction is not considerable. But at higher speed or lower load mating, the surface gap becomes thick enough so that asperity friction is neglected.

Load and speed influence on film thickness is considerably different. The effect of speed on film thickness is dominant but it does not increase the maximum pressure due to essential load equilibrium. On the other hand, load drastically affects the film thickness and pressure spike. This is because of pressure load counterbalance and film formation lubrication mechanism.

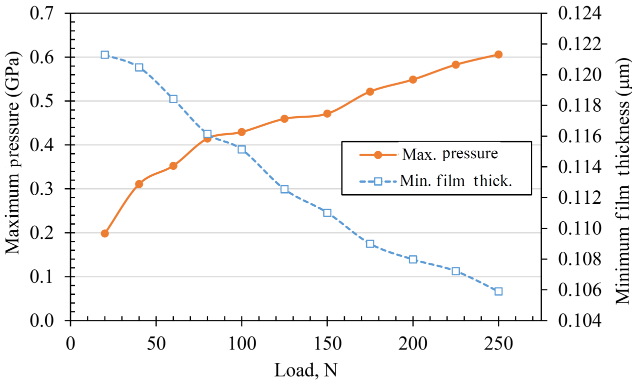

Fig. 8. Load effect on film thickness and maximum pressure.

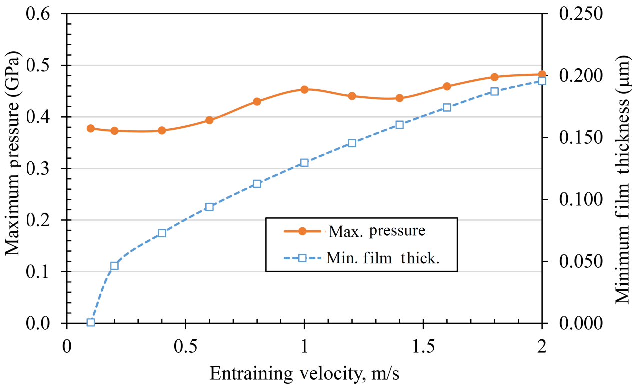

Fig. 9. Speed effect on film thickness and maximum pressure.

6. Conclusions

A numerical simulation for point contact elastohydrodynamic of non-Newtonian grease lubrication was presented based on Herschel Balkley's rheological model. The model was validated by an experimental test on pin on disk apparatus.

Film thickness and pressure distribution were calculated and the existence of the parallel contact region due to surface deformation under pressure induced in fluid was confirmed. The load and entraining velocity effect on pressure spike and minimum film thickness was investigated. Higher load causes sharper spike and lower film thickness.

Grease type effect according to Herschel Bulkley's parameters was studied and it was shown that viscosity and rheology exponent is the most significant parameter. Although the effect of a thickener was not considered in the rheological model directly but increasing the viscosity compensated this deficiency.

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