EFFECT OF STARVATION ON THERMO-ELASTOHYDRODYNAMIC LUBRICATION OF ROLLING / SLIDING CONTACT

Dr. Basim A. Abbas
Faculity member/College of Engineering/
Babylon University/Iraq

Dr. Somer M. Nacy
Assit. Prof. / Al-Khawarizmi
Engineering College/Iraq

ABSTRACT
A complete numerical solution of thermal compressible elastohydrodynamic lubrication of rolling / sliding contact was achieved to determine the effect of inlet boundary condition on the film shape, film pressure, and film temperature in an elastohydrodynamic line contact problem.
The direct iterative technique is used to solve the simultaneous system of Reynolds' ,elasticity , and energy equations for different locations of inlet oil fed . The effect of various load, speed , and slip conditions have been investigated . the results indicate that the effects of starvation are an increase of oil film temperature and decrease in oil film thickness so that the temperature effect are significant and can not be neglected.

الخلاصة
يتضمن البحث حلا" عدديا لمسألة التزييت الهيدروديناميكي المرن مع الأخذ بنظر الاعتبار التغير بدرجات الحرارة وذلك لدراسة تأثير منطقة الدخول على شكل طبقة الزيت, وتوزيع الضغط خلال طبقة الزيت وتوزيع درجة الحرارة خلال تلك الطبقة مع التركيز على المسائل ذات التلامس الخطي. تم استخدام تقنية الاختيار المباشر لحل المعادلات الحاكمة انيا والتي تتضمن معادلة رينولد ومعادلة المرونة ومعادلة الطاقة باستخدام طريقة الفروقات المحددة لمائع تزييت ذو علاقة لزوجة- ضغط تتبع قانون باروس. تمت مناقشة تأثير مختلف العوامل مثل الحمل والسرعة والانزلاق على شكل توزيع ضغط الزيت وتوزيع سمك طبقة الزيت ودرجة الحرارة خلال تلك الطبقة. أظهرت النتائج ان نقصان كمية الزيت تؤدي الى زيادة درجة الحرارة وتقليل سمك طبقة الزيت لذا يوصى بعدم اهمال تأثير درجة الحرارة عند العمل في مثل هذه الظروف.

KEY WORDS
Hydrodynamic lubrication , Porous, Journal Bearing

INTRODUCTION
Rolling sliding machine elements such as bearings , gears, cams and their followers are frequently subjected to high load, speed , and slip conditions. The problem of thermo-elastohydrodynamic lubrication had been treated by many workers, Cheng et.al. (1965), Daow et.al (1987) and Sadeghi et.al. (1990,1987).

Most the published works have not consider the effect of the starvation on pressure distribution , film thickness and oil film temperature so the location of the inlet meniscus is considered as known. However the reduction in film thickness due to starvation has been studied by Chiu (1972) . He conconcluded that the starvation effect in most rolling element bearing are considerably greater than the inlet heating effect.
Full solutions for estimating film thickness of point contacts were made by Hamrock and Dowson (1976,1977) for flooded as well as starved contact. They found a dimensionless central and minimum film thickness for the flooded contacts indicating that the film thickness decreases for the starved condition as compared to the flooded condition.
Johns- Rahngat and Gohar (1994) proved that the results obtained from the work of Hamrock and Dowson are quite realistic when employing starvation conditions at the contact.
The present work is an attempt to study the thermal effect on the performance of contacting element in line contact with starved inlet boundary condition.
Direct iterative method was adopted to solve the governing equations of the problem.

GOVERNING EQUITIONS

The governing equations describing the steady state , thermo-elasto hydrodynamic lubrication of the line contact using Newtonian lubricant can be discribed as follows:

Reynolds' Equation:

Reynolds' equation which govern the pressure distribution inside the oil film presented between two non conformal surfaces in line contact can be written here as follows:

$$\frac{d}{dx}\left(\frac{\rho h^3}{\mu}\frac{dp}{dx}\right) = 12U\frac{d(\rho h)}{dx} \qquad (1)$$

The pressure distribution in the contact zone is subjected to positivity constraint.The boundary conditions for Reynolds equation are given by
$P(x_i)=0$, $P(x_o)=dP_{xo}/dx=0$

Elasticity Equation:

Oil film thickness distribution can be evaluated in this case by using the following equation used by Wolff et.al. (1992)

$$h = h_o + \frac{x^2}{2R} - \frac{2}{\pi \overline{E}}\int_{x_i}^{x_o} p(s)\ln[(x-s)/s]^2 ds \qquad (2)$$

Equations of State

The lubricant viscosity is modeled by Baru's pressure viscosity formula. The equation was modified to include the thermal effect. It can be written as presented by previous reference

$$\mu = \mu_o e^{[\alpha p + \gamma(T - T_o]} \qquad (3)$$

The dependence of density on pressure and temperature can be expressed by the following equation:

$$\rho = \rho_o\left[1 + \frac{.58*10^{-9}P}{1+1.7*10^{-9}P}\right]*\left[1-\varepsilon_t(T-T_o)\right] \qquad (4)$$

Energy Equation

The temperature distribution a cross the oil film is expressed by solving the energy equation reported by Wolf (1992) and Gohr (1988)

$$\varepsilon_T uT\frac{\partial p}{\partial x}+\mu\left(\frac{\partial u}{\partial y}\right)^2=\rho Cu\frac{\partial T}{\partial x}-k\frac{\partial^2 T}{\partial y^2} \tag{5}$$

The boundary conditions used with the above equation are given by

At y=0 ; $T=T_o$

At y=h ; $$T_2=\frac{k}{\sqrt{\pi\rho_2 C_2 k_2 U}}\int_{-\infty}^{x}\frac{\partial T}{\partial y}\frac{d\zeta}{(x-\zeta)^{1/2}} \tag{6}$$

Where,

ρ_2 = density of the solid disk (Kg/m^3)=7800 Kg/m^3 for steel disk

C_2= heat capacity of the solid disk (KJ/Kg.K)=52 KJ/Kg.K for steel disc.

U= speed of the solid disk (m/s).

The mean oil film temperature can be expressed as follows

$$Tm=\frac{1}{h}\int_0^h Tdy \tag{7}$$

by making the suitable substitutions, the mean oil film temperature can be evaluated as:

$$T_m=\left\{\begin{array}{l}(T_1+T_2)/2+N_{UD}\,\eta^2\left[\frac{16}{15\eta^2U^2}\left(\frac{WH}{\pi}\right)^4\left(\frac{dp}{dx}\right)^2+\frac{Sp^2}{12}-\frac{h_f\,sp}{3}\right]+\\ N_{cu}\,\rho H^2\frac{dT_m}{dx}\left[\frac{2}{15\overline{\eta}\,\overline{U}}\left(\frac{WH}{\pi}\right)^2\frac{dp}{dx}-\frac{Sp}{24}-\frac{U_1}{12(U_1+U_2)}\right]\end{array}\right\}\Bigg/\left\{\begin{array}{l}1+N_{AC}\,H^2\frac{dp}{dx}*\{\frac{2}{15\eta\overline{U}}\\ \left(\frac{WH}{\pi}\right)^2\frac{dp}{dx}-\frac{Sp}{24}-\frac{U_1}{12(U_1+U_2)}\ \}\end{array}\right\} \tag{8}$$

Where,

$$N_{UD}=\frac{(E'\,R\overline{U})^2}{T_o\eta_o K} \tag{9}$$

$$N_{AC}=\frac{\varepsilon(E'\,R)^2\overline{U}}{4\eta_o K}\left(\frac{8W}{\pi}\right)^2 \tag{10}$$

$$N_{CU}=\frac{C\rho_o E'\,R^2\overline{U}}{\eta_o K}\left(\frac{8W}{\pi}\right)^{\frac{3}{2}} \tag{11}$$

$$h_f=\frac{8}{\eta\overline{U}}\left(\frac{WH}{\pi}\right)^2\frac{dP}{dx} \tag{12}$$

NUMERICAL TECHNIQUE

To obtain a complete numerical solution to the problem of compressible thermo-elastohydrodynamic lubrication of rolling/ sliding contacts the direct iterative procedure was followed to solve the simultaneous system of compressible Reynolds' Eq.(1), elasticity Eq.(2), and the energy Eq.(5) together with the Equations of state, Eqs.(3,4) . The flow chart shown in **Fig.(1)** illustrates the computational procedure used for calculating the pressure, film thickness , and temperature distribution within the lubricant film.

The isothermal pressure and film shape are obtained and these values are then used to arrive at the initial temperature field within the lubricant film. The influence of temperature is introduced on viscosity , and density and the new pressure and film thickness are calculated. The iterative procedure is continued until the resulted temperature and pressure satisfied the following convergence criteria .

$$\frac{\Sigma\left|P^n-P^o\right|}{\Sigma\left|P^n\right|}\le 0.0001 \quad \text{and} \quad \frac{\Sigma\left|T^n-T^o\right|}{\Sigma\left|T^n\right|}\le 0.0001$$

RESULT AND DISCUSSION

The analysis was carried out for three different nondimensional load parameters, namely (1.36E-7), (2.28E-7), and (3.42E-7). Three different slide to roll ratios are also been considered, namely (0.5,1.07,1.32). Five inlet positions were invistigated, one considered for the flooded condition, Xi=-4a while the other four account for the starved conditions in which Xi=-3.5a ,3a ,2.5a and 2a.The results in **Figs. (2,5,8)** show that the pressure spike decreases and diminshes for all the above cases. The pressure spike try to move outward towared the outlet with the increase of the applied load. The maximum pressure increses as the degree of starvation increases to maintaine a constant applied load.

The oil film thickness decreases with increasing the degree of starvation and the applied load as shown in **Figs. (3,6,9).** Also the nip that occurs in the nondimensional film shape at the fully flooded case diminishes as the amount of starvation increased and the film shape changes to that of flatened type. This is can be explained with refering to **Figs. (4,7,10)** since the oil film temperature increases with increasing the degree of starvation and the applied load which lead to low oil viscosity.

The results presented in **Fig. (12)** also shows that the oil film thickness decreases as the the slide to roll ratio increases. This can be explained with the aid of **Fig. (13),** since it can be shown from this figure that the oil film temperature increases with increasing the slide to roll ratio. The maximum pressure increases as the slide to roll ratio increases in order to maintain constant applied load.

CONCLUSIONS

The concluding remarks which can be withdrawn from the present results are :

1- There is a significant increase in oil film temperature as the degree of starvation and the slide/ roll ratio increases.

2- The pressure spike is decreased (nearly vanshes) as the amount of starvation and slid / roll ratio increases. The pressures pike tends to move toward the outlet as the applied load increases.

3- The oil film thickness decreases as the amount of starvation increased which is dangerous phenomenon leading to occurrence of film rapture due to metal to metal contact at the tip of the mating surfaces.

4- The results indicate that the temperature effect and the position of oil feed have significant effects and must be taken into consideration for proper design.

NOMENCLEATURE

a Hertizian half width = $\sqrt{2RW/((l/2)E^2(\pi/2))}$

E1 modulus of elasticity of roller (1) (N / m^2)

E2 modulus of elasticity of roller (2) (N / m^2)

E equivalent modulus of elasticity = $2/((1-\nu_1{}^2)/E1+(1-\nu_2{}^2)/E2)$ (N/m^2)

H oil film thickness (m)

h_o central oil film thickness (m)

P' oil pressure (N/m^2)

P non dimensional oil pressure = $p/P_{hertizian}$

$P_{hertizian}$ non dimensional oil pressure= $aE/4R$ (N/m^2)

R equivalent radius of the two rollers $= R_1R_2/(R_1+R_2)$ (m)

R_1 radius of roller (1) (m)

R_2 radius of roller (2) (m)

S_p slide to roll ratio=2(U_2-U_1)/ (U_1-U_2)

$\overline{U} = \eta_o(U_1+U_2)/4E'R$

$\overline{W} = W/E'LR$ Dimensionless load parameter

α pressure viscosity coefficient

ϵ oil thermal expansivity (K^{-1})
γ temperature viscosity coefficient
$\nu_{1,2}$ poisons ratio for the upper and lower disks=0.3

REFERENCES

Chin Y. P. , (1972), exploratory analysis of EHD. properties of lubrication , SKF. Technical Report No. Al. 72P010 ,.

F. Sadeghi , Thomas A. Daow, (1987), Thermal effect in rolling /sliding contact part (2) . analysis of thermal effect in fluid film, transection of the ASME , vol. 109, July, pp. 512-520.

F. Sadeghi , (1987), Thomas A. Dow ,Richard R .Johnson, Thermal effect in rolling / sliding contacts part 3 – Approximate method for prediction of mid film temperature and sliding traction, J. of Tribology , July, Vol. 109, PP. 519-524.

F. Sadeghi , Ping C. Sui , (1990), Thermal Elastohydrodynamic lubrication of rough surfaces , ASME journal of lubrication technology, vol. 112, pp. 341- 346.

Hamrock, B. J. and Dowson D., (1976), Isothermal elastohydrodynamic lubrication of point contacts part I – Theoretical formulation , J. of lub. Tech. Trans. of the ASME , vol. 98 ,.

Hamrock , B. J. and Dowson D. (1977), Isothermal elastohydrodynamic lubrication of point contacts part III – Fully flooded results, J. of Lub. Tech. Trans ASME vol. 99,.

H.S. Cheng and B. Sterlicht , (1965), A numerical solution for the pressure, temperature , and film thickness between two infinitely long lubricated rolling and sliding, cylinders under heavy loads . J. of basic engineering, sept..

Johns – Rahenjat , P.M. and Gohar, R , (1994), measuring contact pressure distribution under elastohydrodynamic point contacts, Tribo test Journal,1-1,.

R. J. Gohar, (1988), Elastohydrodynamic lubrication, Ellis Horwood Limited.

R. Wolff, T. Nonaka , A kubo, K. Matso ., (1992), Thermal elastohydrodynamic lubrication of rolling / sliding line contacts, Trans. of the ASME, vol.114, October, pp. 706 –713.

T. A. Daow R.D. Stockwell, J. W. Kannel, (1987), Thermal effect in Rolling / Sliding EHD contacts: part (1) experimental measurement of surface pressure and temperature. J. of tribology, July vol. 109, pp. 503-511.

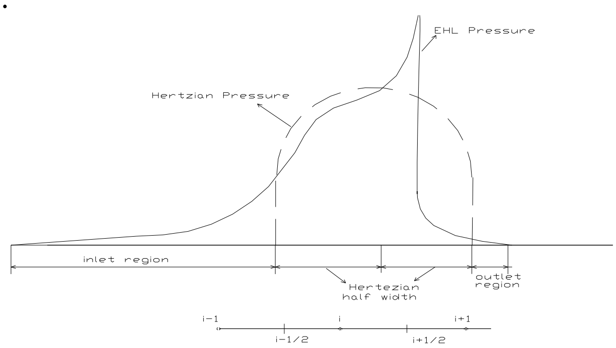

Fig.(1.a) Discretization Scheme

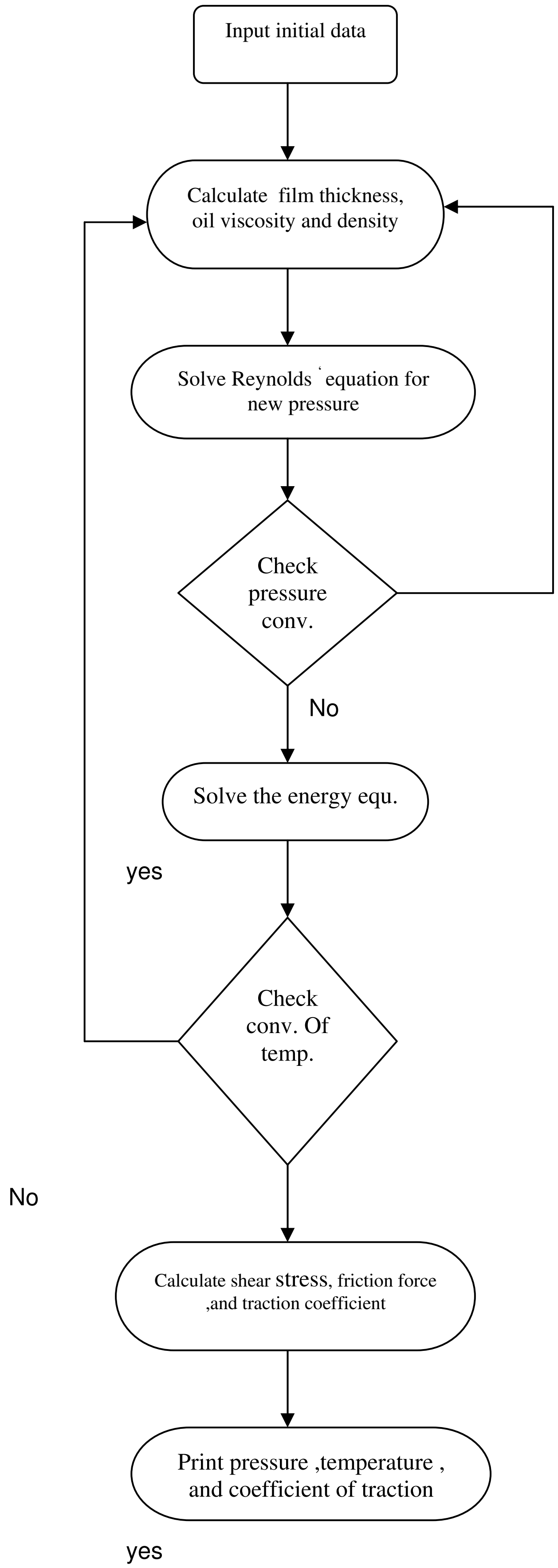

Fig.(1.b): Flow chart for computer program

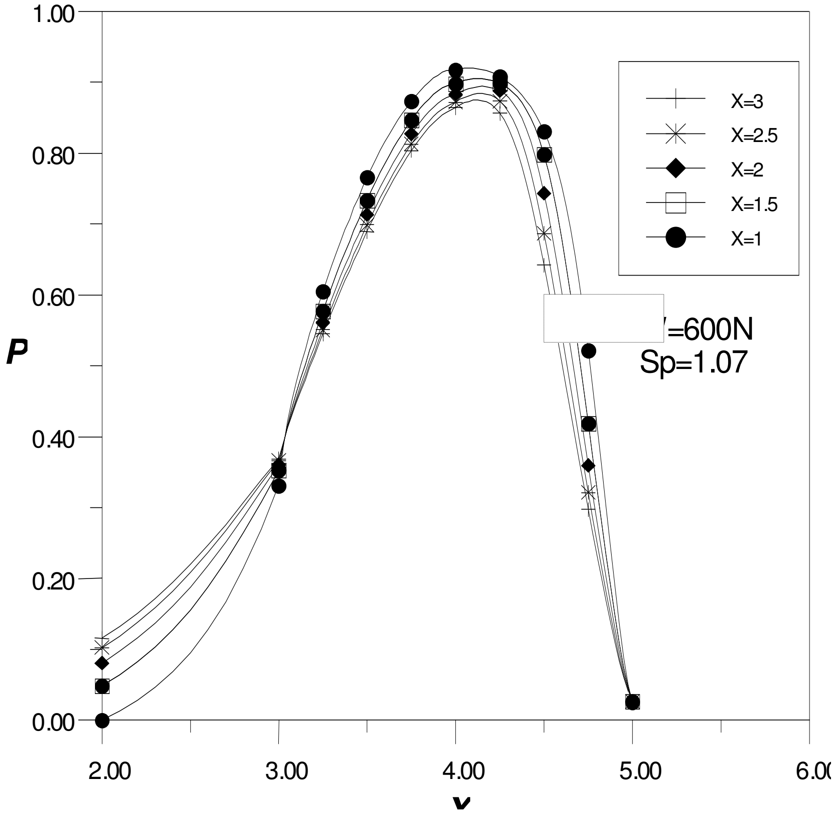

Fig.(2): Effect of starvation on oil pressure distribution
$\overline{W}=1.3*10^{-7}$

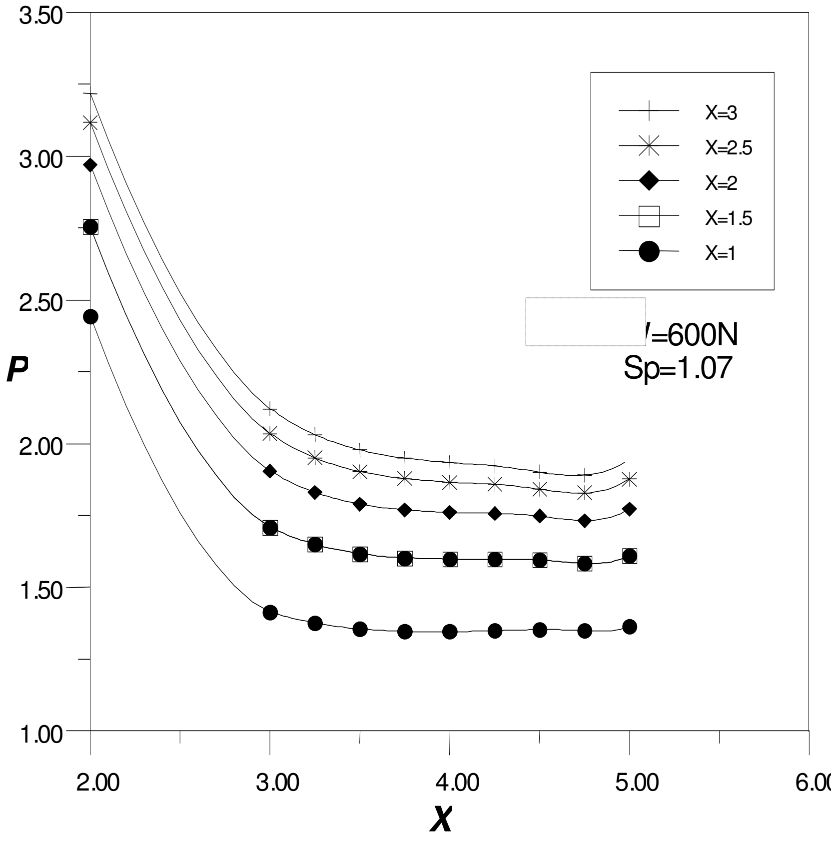

Fig.(3): Effect of load on oil film distribution for different oil inlet boundary condition. $\overline{W}=1.3*10^{-7}$

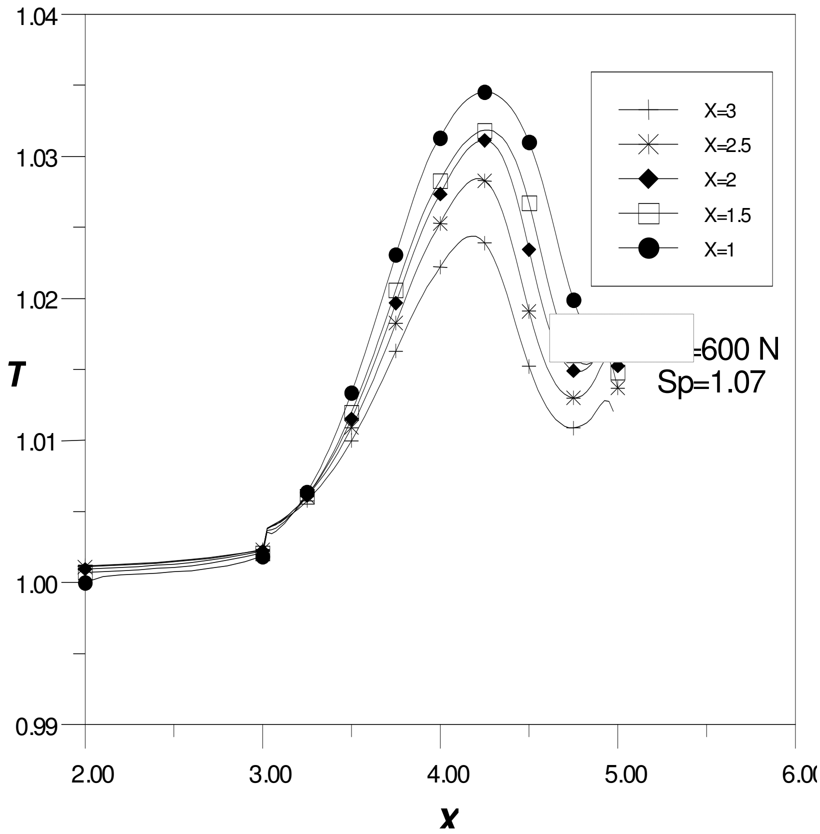

Fig.(4): Temperature distribution for different oil inlet
W=$1.3*10^{-7}$

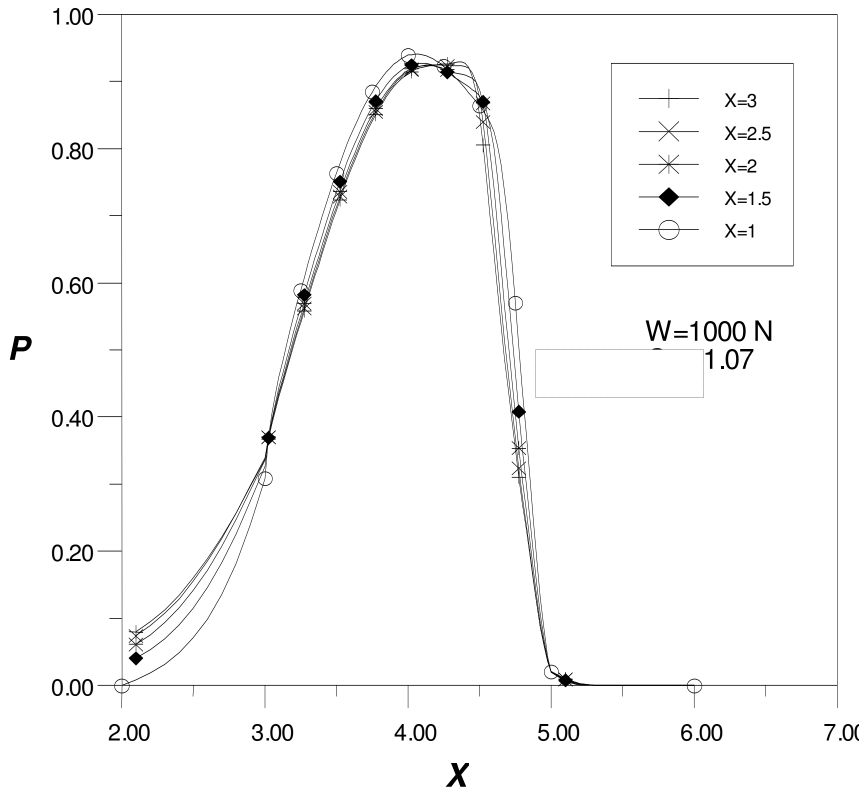

Fig.(5): Pressure distribution for different oil inlet
W=$2.28*10^{-7}$

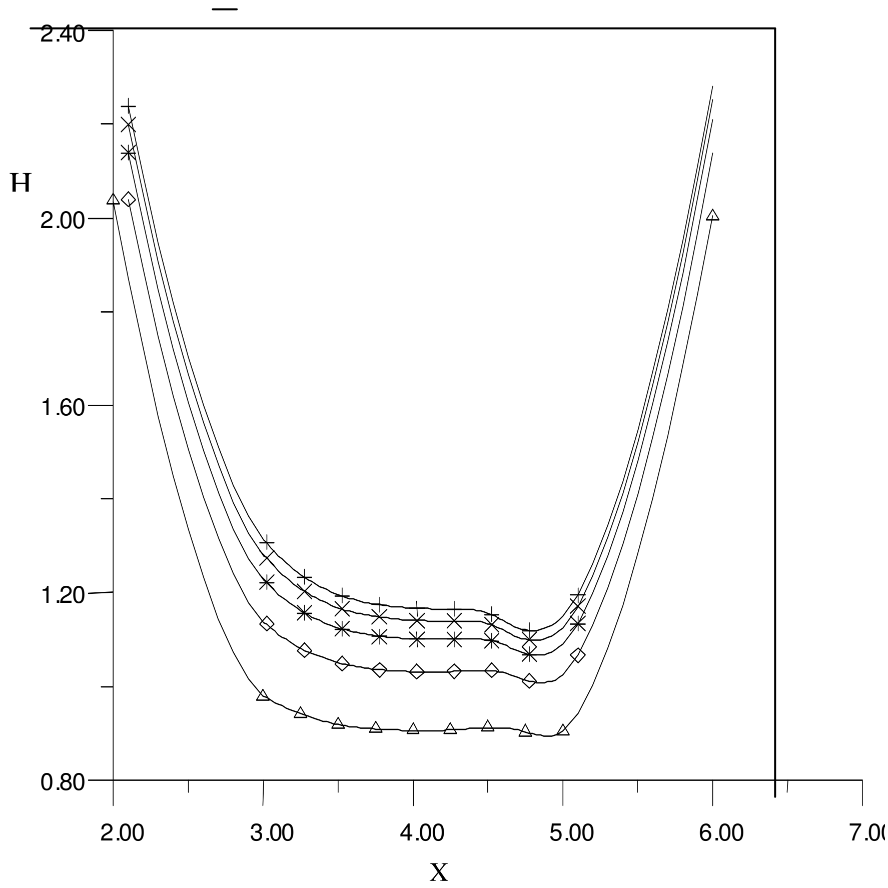

Fig. (6): Oil film thickness for different oil inlet

$\overline{W}=2.28*10^{-7}$

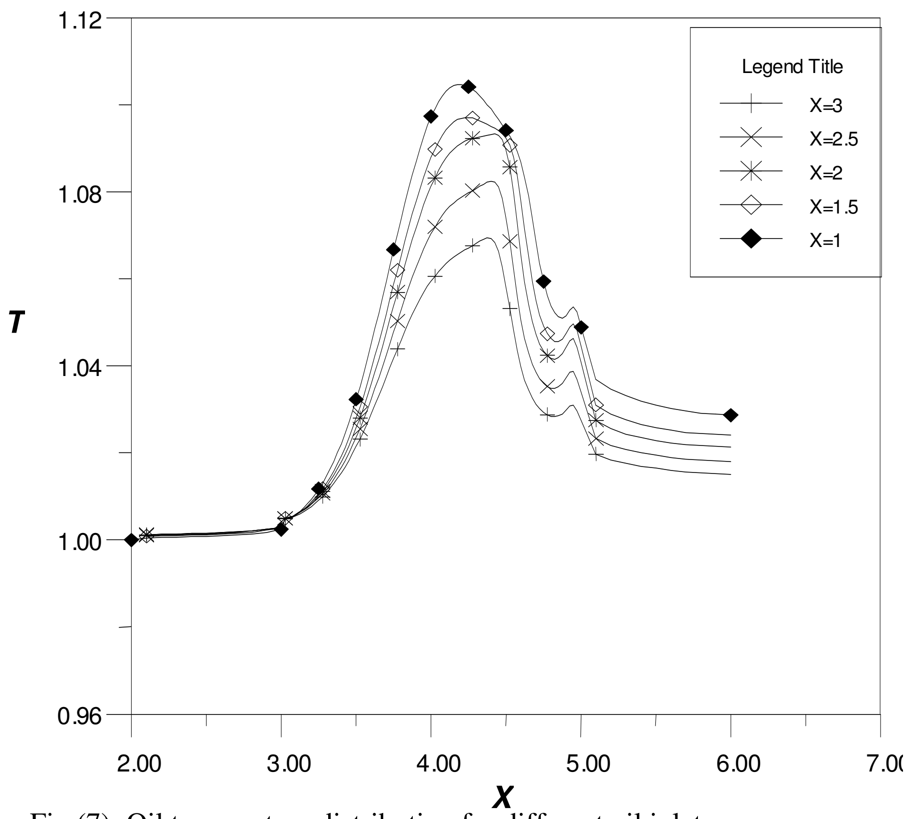

Fig.(7): Oil temperature distribution for different oil inlet

$W=2.28*10^{-7}$

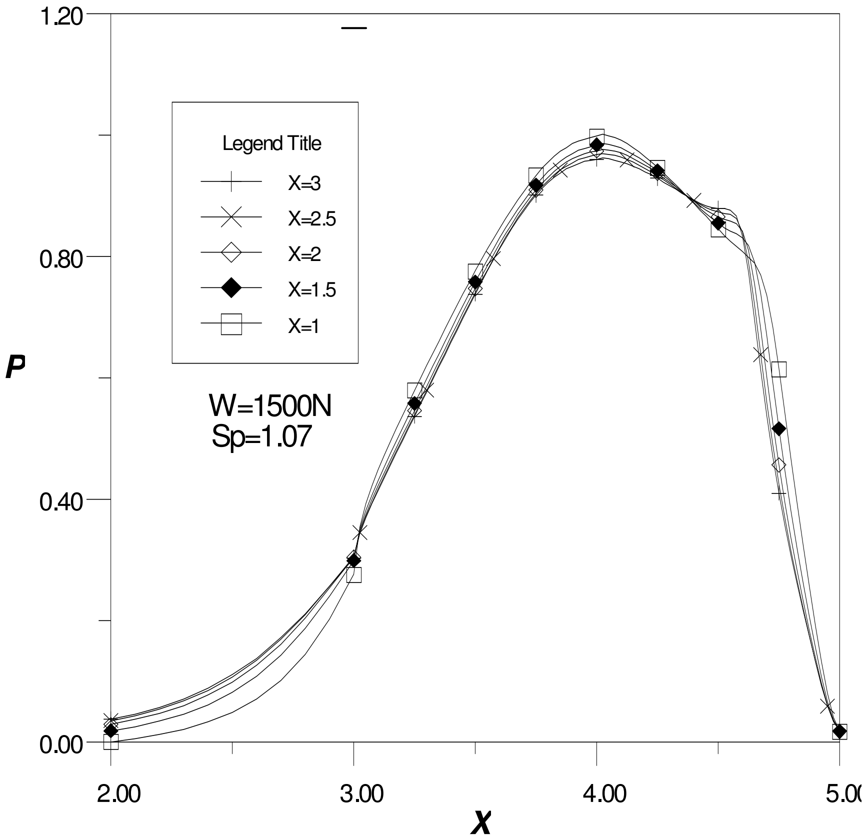

Fig. (8):Pressure distribution for different inlet boundary condition
$W=3.42*10^{-7}$

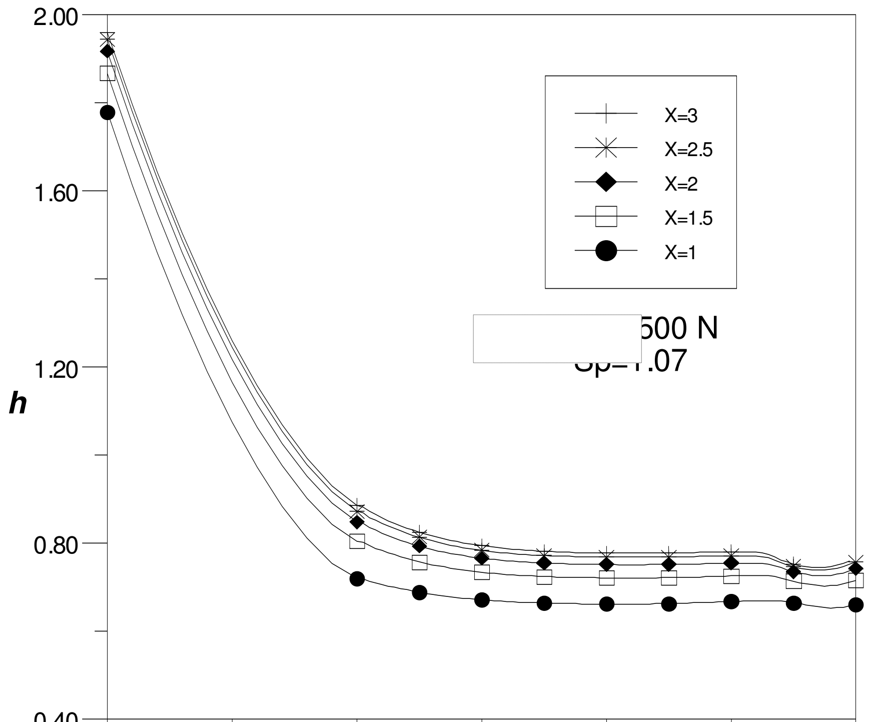

Fig. (9):Oil film thickness for different oil inlet
$W=3.42*10^{-7}$

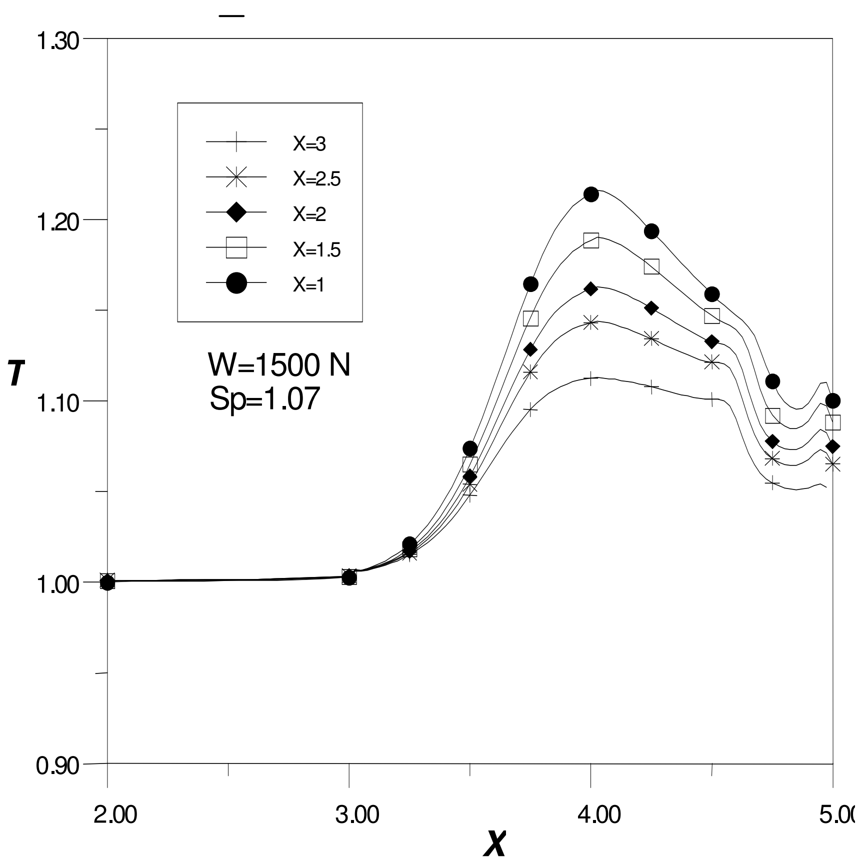

Fig. (10):Oil temperature distribution for different oil inlet
W=3.42*10^{-7}

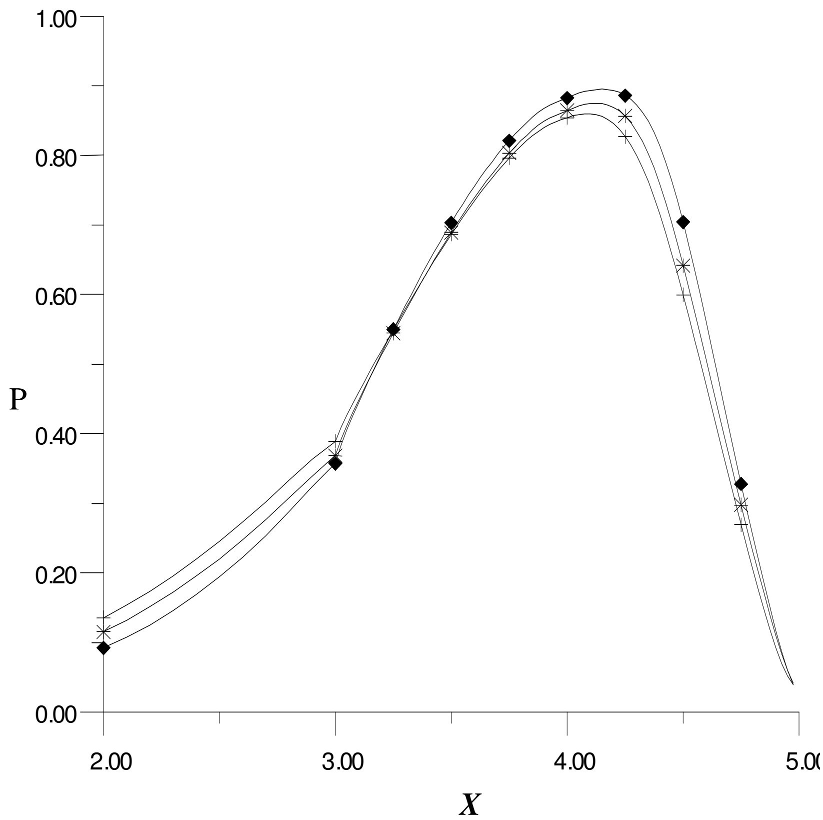

Fig.(11):Effect of slide to roll ratio on the pressure distribution for different oil inlet positions W=3.42*10^{-7}

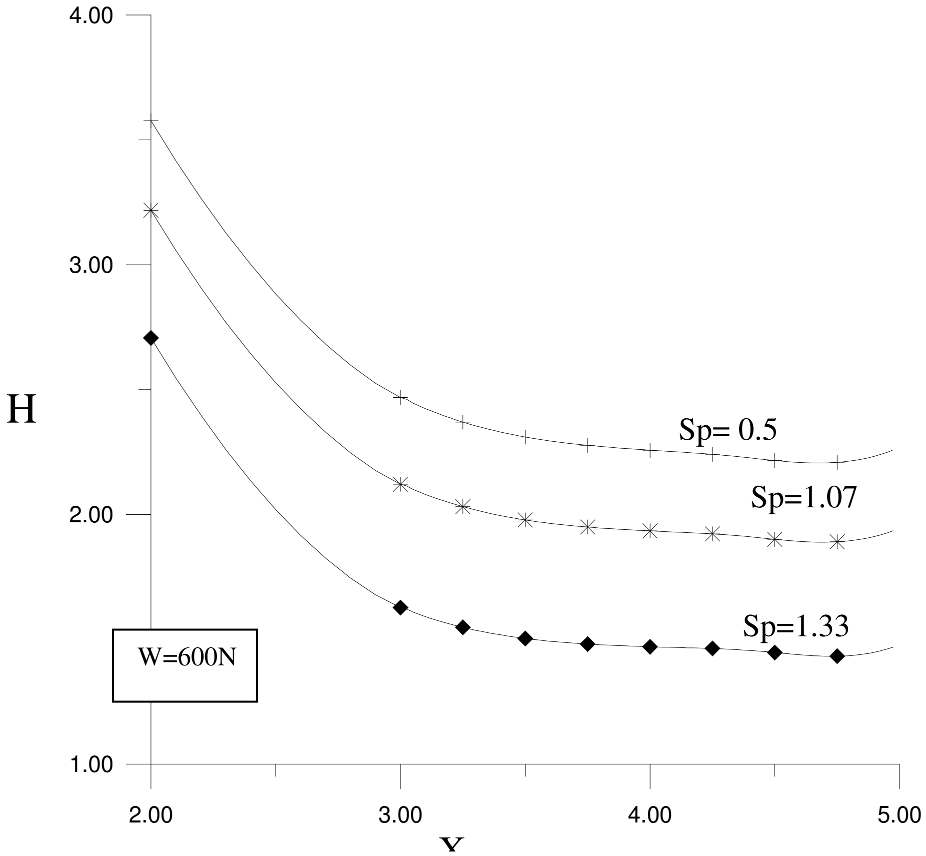

Fig. (12): Effect of slide to roll ratio on film thickness for different oil inlet

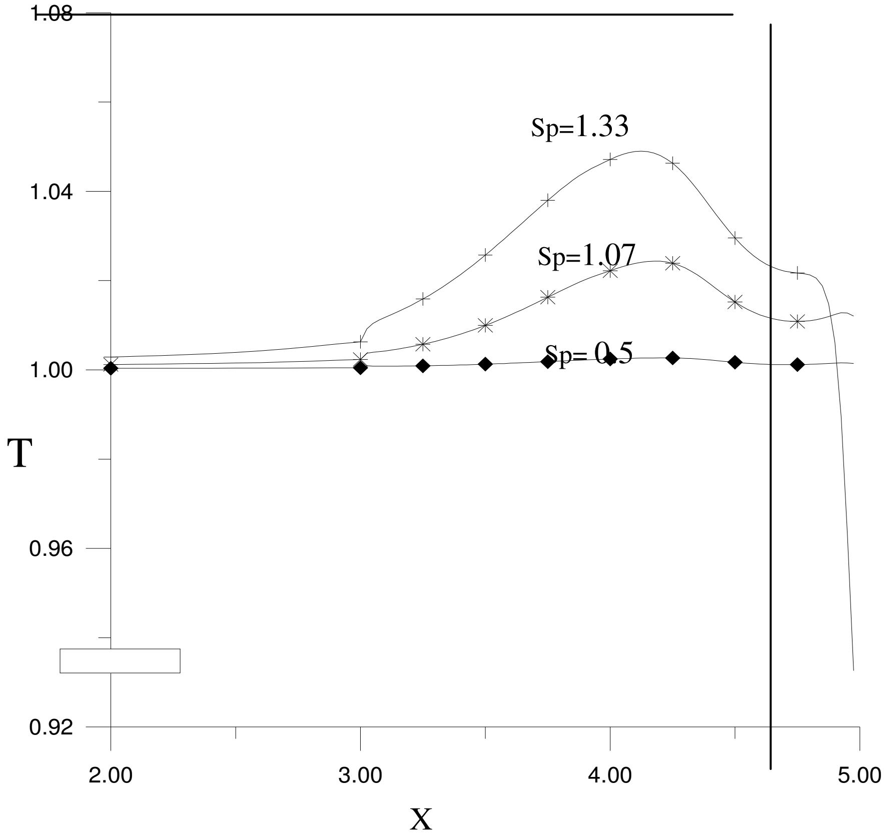

Fig. (13): Effect of slide to roll ratio on oil film temperature distribution for different oil inlet