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Hydrodynamic Lubrication of Linear Hydraulic Engines

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Abstract

This paper is a study of the hydrodynamic lubrication between the cylinder liner and the piston ring of a linear hydraulic engine. The influence of geometric, material and working conditions as well as the temperature and shear-rate dependence of lubricant viscosity on the oil film minimum thickness and power loss is presented in the paper. These results can be used in the linear hydraulic engines design.
KEYWORDS*: linear hydraulic engines, lubrication, temperature*

1. Introduction

Numerical modeling of linear engines is a domain of study that received great attention in the last decades all over the world [1÷12]. Research studies of the internal combustion engine performances in different working conditions analyze both the steady-state running of the engines [1, 11] and their non-stationary running [2, 3, 6, 7]. These models are influencing the engine parts design reducing, in this way, its time and cost. The aim of these studies is the power maximization, the emissions minimization and fuel economy. Their complexity level depends on the number of aspects taken into consideration. They may analyze the gases thermodynamics, combustion or flow, the heat transfer, the power losses due to friction, ea.

The power (energy) losses due to friction appear on the lubricated surfaces of the engine and the biggest losses are registered on the lubricated surfaces at the piston rings area. The factors that are affecting the lubrication characteristics in the area mentioned above are [7]: the oil film thickness (the lubrication type) and viscosity, the piston rings shape, the working conditions, ea. As a function of the thickness of the oil film situated between the liner and the piston ring, Froelund [7] defines three lubrication types:

- *hydrodynamic lubrication*, when the oil film thickness is four times bigger then the combined roughness;
- *mixed lubrication*, when the oil film thickness is bigger than the combined roughness and smaller than four times the combined thickness;
- *pure friction* at the interface, when the oil film thickness is not bigger enough in order to have a mixed lubrication.

The hydrodynamic lubrication appears between two surfaces in relative motion with a viscous fluid between them. If the surfaces are forming a convergent angle, the hydrodynamic pressure that appears has load-carrying capacity. This mechanism is the base of slider and journal bearing functionality and it appears, also, at the piston ring passing over the liner at hydraulic engines (Fig. 1) offering small friction, wear and fatigue resistance.

The way in which the researchers treated the hydrodynamic lubrication evolved in time. Their models varied as a function of the precision with which the viscosity temperature variation was treated, the governing equation complexity, ea. Over the years, the viscosity was considered as being constant or it was calculated as a function of cylinder temperature, the average temperature of oil film ([6], [7]) or the local temperature and shear rate ([1]). Theoretically, the last version is the most correct one even if experimental data are suggesting that the average temperature of the oil film is the best reference for the calculation of the temperature variation of the oil film viscosity [3]. Considering a temperature and shear-rate variation of viscosity, the researchers eliminated the old constraints of constant thickness oil film and, thus, the lubricant film thickness variation could be calculated. As a result, the power loss (the engine efficiency) could be established, a loss that is higher at higher speeds and powers.

This paper is a study of the hydrodynamic lubrication of an linear hydraulic engine with an TPM (TPM/P) type piston ring. The oil viscosity is a function of temperature and shear-rate and the energy conservation equation is transient; the governing equations were solved with the finite differences method using an iterative procedure. The results are giving a better insight on the hydrodynamic lubrication of a linear hydraulic engine by giving variations of the oil film minimum thickness and the power loss. These results can be used as guidance in the design process and in the technological decisions regarding the linear hydraulic engines.

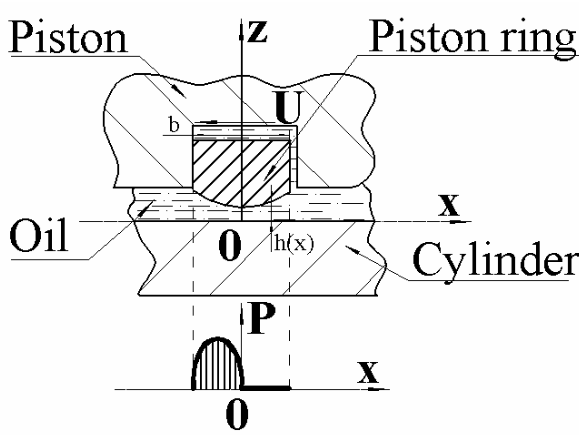


Fig. 1. *Cylinder, piston and piston ring of a linear hydraulic engine. Pressure variation in the oil film.*

2. Mathematical model

The equations defining the behavior of the oil film between the piston ring and the cylinder liner (Fig. 1), the mass, momentum and energy conservation equations with Reynolds equation, are describing completely the velocity, pressure and temperature fields of the lubricant. The following dimensionless variables were defined for the system of equations adimensionalization:

$$\bar{z} = \frac{z}{h};\ \bar{h} = \frac{h}{h_{ref}};\ \bar{\mu} = \frac{\mu}{\mu_0};\ \bar{T} = \frac{T}{T_0};\ \bar{x} = \frac{x}{b};$$

$$\bar{u} = \frac{u}{U};\ \bar{w} = \frac{w}{U}\cdot\left(\frac{b}{h_{ref}}\right);\ \bar{t} = \frac{t}{T_c};$$

$$\bar{h}_{min} = \frac{h_{min}}{h_{ref}};\ \bar{P} = \frac{P\cdot h_{ref}^2}{\mu_0\cdot U};\ \bar{L} = \frac{L}{b}. \tag{1}$$

where (x, z) is the system of equation defined by Fig. 1, (u, w) is the velocity field, T is the temperature, t is the time, μ is the oil dynamic viscosity while μ_0 is the oil dynamic viscosity at initial temperature T_0, P is the pressure, U is the piston speed, h is the thickness of the lubricant in the z direction while h_{min} is the oil film minimum thickness and h_{ref} is the reference value (1μm), b is the ring width, L is the piston course, T_c is the piston course time. The “–” sign is used for the dimensionless variables.

The dimensionless forms of the governing equations are:
the momentum conservation equation:

$$\frac{\partial}{\partial \bar{z}}\cdot\left(\bar{\mu}^*\frac{\partial \bar{u}}{\partial \bar{z}}\right) = \bar{h}^2\cdot\left(\frac{\partial \bar{P}}{\partial \bar{x}}\right); \tag{2}$$

Reynolds equation:

$$\frac{\partial}{\partial x}\left(\overline{F_2}\,\bar{h}^3\frac{\partial \bar{P}}{\partial \bar{x}}\right) = U\frac{\partial}{\partial \bar{x}}\left(\bar{h}\frac{\overline{F_1}}{\overline{F_0}}\right) + \frac{b}{R}\cdot\frac{\partial \bar{h}}{\partial t} \tag{3}$$

where:

$$\overline{F_0} = \int_0^1 \frac{1}{\bar{\mu}^*(I_2)}d\bar{z};\ \overline{F_1} = \int_0^1 \frac{\bar{z}}{\bar{\mu}^*(I_2)}d\bar{z};$$

$$\overline{F_2} = \int_0^1 \frac{\bar{z}}{\bar{\mu}^*(I_2)}\cdot\left(\bar{z} - \frac{\overline{F_1}}{\overline{F_0}}\right)d\bar{z}. \tag{4}$$

In equation (3), the (n+1) unknowns are the pressure (in each point of discretization along the x axis) and the time variation of the oil film thickness. There are (n+1) unknowns. A supplementary equation is needed and it is obtained from the hypothesis that the piston ring can be modeled as a thin wall cylinder [1].

The boundary conditions for the Reynolds equation solution, (3), are:

$$P(x = -b/2) = P_0; \tag{5}$$

$$P(x = b/2) = P_1, \tag{6}$$

where P_0 is the trial pressure and P_1 is the output pressure, respectively.
the energy conservation equation:

$$\bar{u}\frac{\partial \bar{T}}{\partial \xi} + \frac{1}{\lambda_1}\cdot\frac{\partial^2 \bar{T}}{\partial \xi^2} + \frac{\bar{w}}{\bar{h}}\cdot\frac{\partial \bar{T}}{\partial \bar{z}} =$$

$$= \left(\frac{\lambda_2}{\lambda_1}\right)\frac{1}{\bar{h}^2}\cdot\frac{\partial \bar{T}}{\partial \bar{z}^2} + \left(\frac{\lambda_3}{\lambda_1}\right)\frac{\overline{\mu^*}}{\bar{h}^2}\cdot\left(\frac{\partial \bar{u}}{\partial \bar{z}}\right)^2, \tag{7}$$

is solved in the computational domain, $(\xi, \bar{z})$, where,

$$\frac{\partial^2}{\partial\xi^2}=\frac{\partial^2}{\partial\overline{x}^2}+\left(\frac{\overline{z}}{\overline{h}}\right)^2\cdot\left(\frac{\partial\overline{h}}{\partial\overline{x}}\right)^2\cdot\frac{\partial^2}{\partial\overline{z}^2}-2\frac{\overline{z}}{\overline{h}}\cdot\frac{\partial\overline{h}}{\partial\overline{x}}\cdot$$
$$\cdot\frac{\partial^2}{\partial\overline{x}\cdot\partial\overline{z}}+2\frac{\overline{z}}{\overline{h}^2}\cdot\left(\frac{\partial\overline{h}}{\partial\overline{x}^2}\right)^2\cdot\frac{\partial}{\partial\overline{z}}-\frac{\overline{z}}{\overline{h}}\cdot\frac{\partial^2\overline{h}}{\partial\overline{x}^2}\cdot\frac{\partial}{\partial\overline{z}};\tag{8}$$

$$\frac{\partial}{\partial\xi}=\frac{\partial}{\partial\overline{x}}-\frac{\partial\overline{h}}{\partial\overline{x}}\cdot\frac{\overline{z}}{\overline{h}}\cdot\frac{\partial}{\partial\overline{z}};\tag{9}$$

$$\lambda_1=\frac{\rho c_p}{k}L\omega b;\lambda_2=\left(\frac{b}{h_{ref}}\right)^2;\lambda_3=\frac{\mu_0(U)^2}{k\cdot T_0}\tag{10}$$

with ρ - the density and c_p - the specific heat.

The temperature and shear-rate dependence of viscosity is given by:

$$\mu=\mu_i\cdot\left(\frac{k+\mu_2\cdot\left|\frac{\partial u}{\partial z}\right|}{k+\mu_1\cdot\left|\frac{\partial u}{\partial z}\right|}\right),\tag{11}$$

where

$$\mu_1=\mu_{i1}e^{-\beta_1(T-T_l)};\tag{12}$$

$$\mu_2=\mu_{i2}e^{-\beta 2(T-T_l)};\tag{13}$$

$$k=k_ie^{\beta_3(T-T_l)},\tag{14}$$

and μ_1 and μ_2 are the viscosity at small and high shear-rate, respectively, while μ_{i1} and μ_{i2} are their values at the initial temperature T_0. k is a parameter that allows the curve fitting of the experimental data.

The boundary conditions for equation (7) are:

$$T(-b/2,z)=T_0;\tag{15}$$

$$\frac{\partial T}{\partial x}(b/2,z)=0;\tag{16}$$

$$T(x,0)=T_0;\tag{17}$$

$$\frac{\partial T}{\partial z}(x,h)=0.\tag{18}$$

the mass conservation equation:

$$\frac{\partial\overline{w}}{\partial\overline{z}}=-\overline{h}\left(\frac{\partial\overline{u}}{\partial\xi}-\overline{z}\frac{\overline{h}'}{\overline{h}}\frac{\partial\overline{u}}{\partial\overline{z}}\right),\tag{19}$$

is, also, solved in the computational domain.

The system of equations (2), (3) and (7) are discretized in the computational domain using centered finite differences ([15]) or backward/forward finite differences for the boundary points. The horizontal velocity, pressure and temperature fields are calculated simultaneously, while the vertical velocity of the oil film can be found integrating (19).

3. Results and discussions

The results presented here were obtained for the conditions of Table 1. They analyze the influence of geometric, material and working conditions on the lubricant minimum thickness (h_{min}) and the power loss (P_p) [1]:

$$P_p=\frac{\mu_{i1}U^2\pi D_0 b}{h_{ref}}\cdot\left[\int_A\left(\frac{\overline{\mu}}{\overline{h}}\right)\overline{U}^2d\overline{x}+\int_A\left(\frac{\overline{h}^3}{12}\right)\left|\nabla\overline{P}\right|^2d\overline{x}\right]\tag{20}$$

Table 1. *Process conditions.*

Name	Units of Measurement	Value
Trial pressure	P_0[bar]	250
Output pressure	P_1[bar]	1.0
Piston speed	U[m/s]	0.10
Active course	L[m]	0.30
Interior diameter	D_0[mm]	45.0
Piston ring radius	R[m]	0.001
Initial oil film minimum thickness	h_0[μm]	0.5
Initial temperature	T_0[°C]	20.0
The oil viscosity	μ[Pa.s]	0.011
Temperature and shear-rate viscosity dependence	β_1[K^{-1}]	0.0214
	β_2[K^{-1}]	0.0214
	β_3[K^{-1}]	0.0270
	μ_{i1}[Pa.s]	0.0111
	μ_{i2}[Pa.s]	0.0063
	k_i[Pa]	1500
Oil density	ρ[kg/m^3]	884.0
Young modulus	E[GPa]	3.0

Figure 2a presents the time variation of the minimum thickness of the oil film, h_{min}. A constant “steady-state” value is noticed after a period of time of $2x10^{-4}$ of the course time, T_c, where T_c=L/U. The power loss due to friction and squeeze phenomenon is presented by Fig. 2(b). An equilibrium value (P_m) of almost 1W (0,005% of the nominal power) is noticed.

The equilibrium value of the oil film thickness is not very much influenced by the initial minimum thickness, Fig. 3(a), or the piston ring radius, Fig. 3(b). The influence of the distance ring/liner, D, and the working velocity on the equilibrium value of the oil film thickness is presented by Fig. 4(a) and Fig. 4(b), respectively. The equilibrium values remain insignificantly influenced by these parameters; their influence is important for the power loss [16], [17].

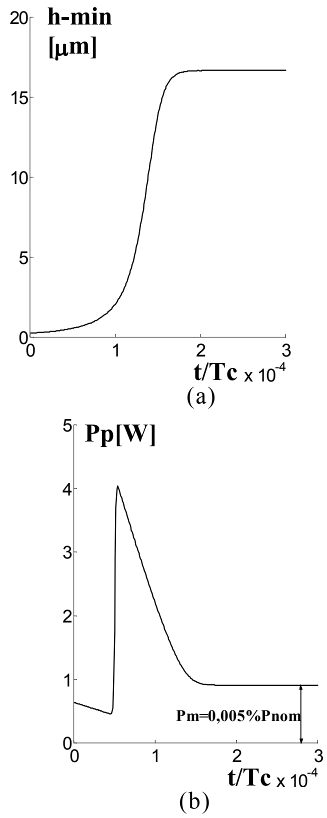


Fig. 2. *Time variation of lubricant minimum thickness (a) and the power loss (P_p)(b).*

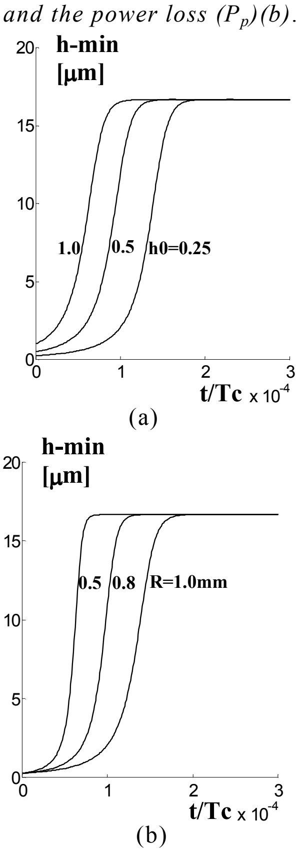


Fig. 3. *Time variation of the minimum thickness of the oil film for three values of the initial minimum thickness, h_0 (a) and the ring profile radius, R (b).*

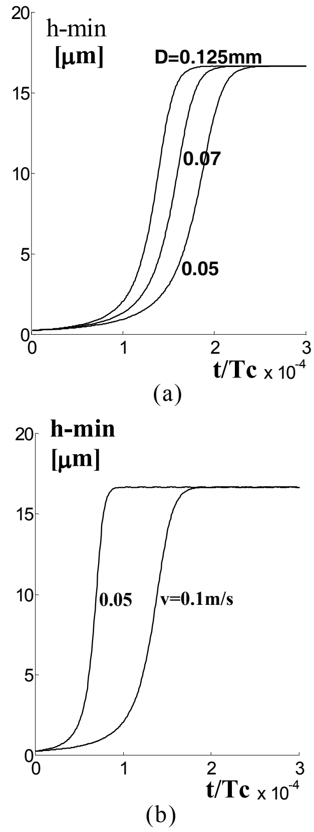


Fig. 4. *Time variation of the minimum thickness of the oil film for three values of the liner/ring gap, D, (a) and for two values of the piston velocity (b).*

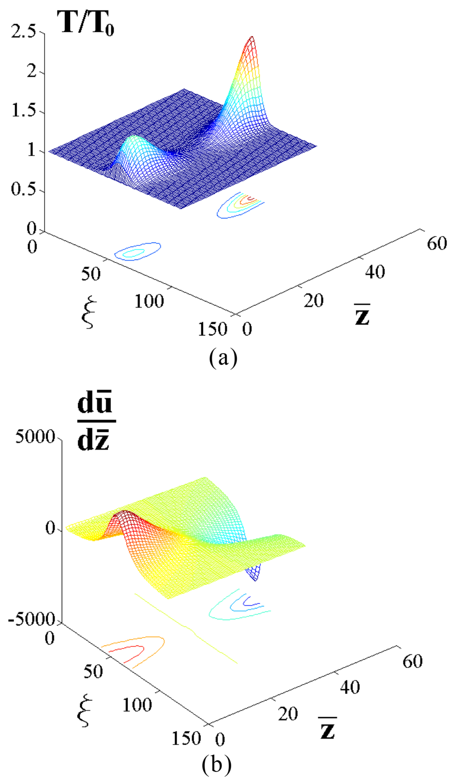
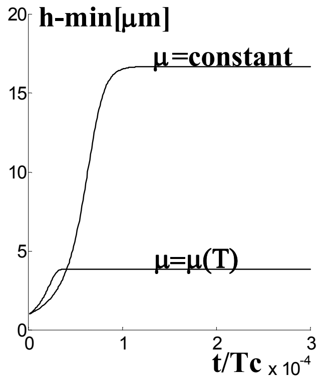


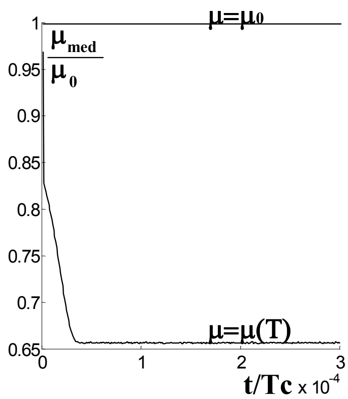
Fig. 5. *Dimensionless temperature field (a) and $\partial u/\partial z$ field (b) for T_0=20°C, t=3X10$^{-4}T_c$ and h_0=1μm.*

Temperature field of the oil film situated between the abscises x=-b/2 and x=b/2 (Fig. 1) is presented by Fig. 5(a) in dimensionless form for T_0=20°C and t=3X$10^{-4}T_c$.

The maximum values of the temperature field appear in the vicinity of the liner and the piston ring where the shear rates are maximum, the viscous dissipation is maximum, in other words, where the du/dz field has maximum values, (Fig. 5b).



(a)



(b)

Fig. 6. *Time variation of maximum, T_{max}, and average, T_{med}, temperature (a) and average viscosity (b) for T_0=20°C and h_0=1µm.*

Figure 6a presents the time variation of the medium, T_{med}, and maximum, T_{max}, temperature for the case when the fluid viscosity is a function of local temperature and shear rate. The equilibrium values obtained by both values are noticed around the 2.0 and 1.1 values, respectively. Figure 6b presents the time variation of the average viscosity of the oil film for the case when the viscosity does not vary with temperature and for the case when viscosity varies with the temperature and the shear rate at every point. The process parameters are those presented by Table 1, T_0=20°C and h_0=1µm. It can be noticed that the "equilibrium" value of the average viscosity in the second case decrease to ≈65% of the initial value, μ_0.

The reduction of the minimum thickness of the oil film can be noticed in Fig. 7 in the same time with the temperature increase (Fig. 6a) and the viscosity decrease (Fig. 6b). This reduction is important and it shows a good agreement with the results presented in the literature [1÷3], results that are suggesting a maximum decrease for the temperature and shear-rate dependent viscosity case.

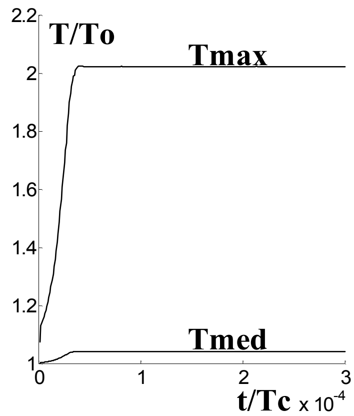


Fig. 7. *Time variation of the minimum thickness of the oil film: constant viscosity and local temperature and shear rate dependent viscosity. T_0=20°C and h_0=1µm.*

Concluding Remarks

This study analyses the hydrodynamic lubrication that appears between the piston ring and the cylinder of a linear hydraulic engine. It is based on the governing equations: the mass, momentum and energy conservation equations and Reynolds equation of the oil film situated between the piston ring and the liner.

The dimensionless form of the equations are offering generality and clarity to the results obtained using the finite difference method.

The temperature, pressure and velocity fields, the minimum thickness and the power loss offer the possibility for a complete understanding of the hydrodynamic lubrication process that is taking place between the liner and the piston ring.

The influence of different geometric, material and functional factors as well as the temperature and shear-rate dependence of the viscosity on the minimum thickness of the oil film and power loss is analyzed. The results obtained agree with the trends presented in the literature [1, 2, 3].

The results of this study can offer guidance for the design activity and for the technological decisions regarding the geometry or the functionality of the linear hydraulic engines.

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Ungerea hidrodinamică a motoarelor hidraulice lineare

Rezumat

Această lucrare este un studiu al ungerii hidrodinamice dintre cilindru şi inelul pistonului la motoarele hidraulice lineare. Influenţa caracteristicilor geometrice, de material şi funcţionale precum şi a dependenţei vâscozităţii lubrefiantului de temperatură şi forfecare asupra grosimii minime a peliculei de ulei este analizată. Aceste rezultate pot fi utilizate în proiectarea motoarelor hidraulice lineare.

Hydrodynamische Schmierung von linearem hydraulischem Engines

Auszug

Dieses Papier ist eine Studie der hydrodynamischen Schmierung zwischen die Zylinderzwischenlage und den Kolbenring einer linearen hydraulischen Maschine. Der Einfluß von geometrischem, das Material und die Arbeitsbedingungen sowie die Temperatur und die Scherenrate Abhängigkeit der Schmiermittelviskosität auf dem Öl filmen minimale Stärke und Leistungsabfall wird im Papier dargestellt. Diese Resultate können bei linearem hydraulischem Maschinen Design verwendet werden.