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Analysis of Dissipated Power Caused by Lubrication in Ringless Reciprocating Systems

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Abstract— **The purpose of this paper was to evaluate the electromotor input power loss caused by oil viscosity between piston and cylinder in reciprocating systems such as compressors, presses and pumps with crank and slider driver without oil ring or ringless pistons. Using the numerical and analytical approaches respectively for nonlinear and linear oil velocity profiles assumed between piston and cylinder, dissipated power caused by oil viscosity was calculated and results of these two approaches were compared to validate finite difference results. Finally, the effect of vertical or horizontal position of piston and cylinder were compared in the case of nonlinear oil velocity profile for different applications.**

Keywords- ***Dissipated power, Lubrication, Linear and nonlinear oil velocity profiles, Ringless piston, Analytical and numerical methods.***

I. INTRODUCTION

Piston and cylinder systems are widely used in power engineering applications. Calculation of dissipated power caused by lubrication between piston and cylinder in reciprocating systems is essential for installation, design process, analysis, and optimization. The main applications of these systems are in reciprocating compressors, reciprocating presses, reciprocating pumps, and so many other systems driven by crank and slider mechanism [1].

In reciprocating compressors which are used in domestic refrigerators in which very low friction loss is needed, ringless pistons are being utilized to minimize the friction between piston and cylinder. To lessen the frictional losses of the piston and cylinder systems, the cylinder bore's length is reduced. In early studies, Li et al. [2] and Zhu et al. [3-4] tried to minimize the friction loss in reciprocating pistons of automotive engines. Afterwards, Gommed and Etsion analyzed gas lubrication of a ringless piston in a series of papers [5-7] in a low heat rejection (LHR) engine. Parata et al. [8] performed a dynamic analysis for the oil film between piston and cylinder in small refrigerating compressors.

To diminish both friction loss and refrigerant gas leakage through the piston and cylinder clearance in the high efficiency compressors of domestic refrigerators, it is required to perform a dynamic analysis of the secondary motion of the piston. Kim [9] has presented a formulation for the piston dynamics considering hydrodynamic forces and moments between piston and cylinder and the variation in bearing length of the piston. Jeng [10] indicated that the predominant source of friction in reciprocating engines is the piston ring assembly and estimates represent that 40% of the mechanical losses are due to piston ring. In addition, to simulate a piston ring, Ronen et al. [11] presented a hydrodynamic computational model in reciprocating parallel surface bearings. They concluded that appropriately sized dimples can considerably reduce frictional loss. This effect was confirmed in the experimental work of Zhao et al. [12] and later in a work by Ryk et al. [13]. They have shown that pressurization of trapped lubricant caused by deformation of a pocket under load and surface roughness in boundary and mixed lubricated contacts decrease solid contact pressure. Using numerical and experimental approaches, Nathan et al. [14] investigated the potential for surface patterning and features to reduce friction at the piston ring cylinder liner interface. Meng et al. [15] analyzed the influence of oil film inertia on piston skirt lubrication in a high speed engine using an iteration method. The result has shown that the ratio of piston skirt's length to its diameter increases the effect of oil film inertia on the friction force. Mixed lubrication analysis for the piston ring pack performed by Young et al. [16] with considerations of average Reynolds equation and asperity contact model. Also, to determine the asperity contact forces, Choi et al. [17] presented a complete one-dimensional mixed lubrication model for the piston ring in which the average flow model is used to calculate the mean hydrodynamic film pressure. In this model of the piston ring, the effects of the roughness height, patterns, and engine speed on the nominal minimum oil film thickness (MOFT), frictional force, and power losses were investigated.

Therefore, that piston ring plays the key role in friction between piston and cylinder wall; however, in this paper, ringless piston which is going to be used widely in reciprocating systems is analyzed. In reciprocating systems, depending on the size of clearance between piston and cylinder walls, nonlinear velocity profiles could be approximated as linear velocity profiles [18]. Thus, in presented paper, the dissipated power caused by the lubrication between ringless piston and cylinder has been estimated using the linear velocity profile as verification for finite difference results of the nonlinear velocity profile. It has been concluded that the dissipated power in the case of nonlinear velocity profile is 41.1% more than that of linear one. The most dominant factor to control this difference is clearance between piston and cylinder and this fact is physically explained in this study.

Because of lots of applications of ringless piston and cylinder in vertical or horizontal positions of reciprocating systems, the effect of vertical or horizontal position of piston and cylinder on dissipated power has been compared in the case of nonlinear velocity profile. As a case in point, reciprocating presses are mostly vertical, while reciprocating compressors [19] are mostly horizontal. Hence, in the case of nonlinear velocity profile, different cases have been considered consisting of vertical position of cylinder with downward and upward motions of piston and the horizontal position of piston and cylinder. In fact, dissipated power in horizontal position of piston and cylinder is nearly equal to the average dissipated powers of upward and downward motions of piston.

II. GEOMETRY, PEROBLEM DEFINITION, AND GOVERNING EQUATIONS

The schematics of linear and nonlinear oil velocity profiles are given in Fig. 1. Also, crank and slider mechanism and its notations used in this work are illustrated in Fig. 2 and to compare linear and nonlinear oil profiles, the constant parameters are evaluated in Table 1.

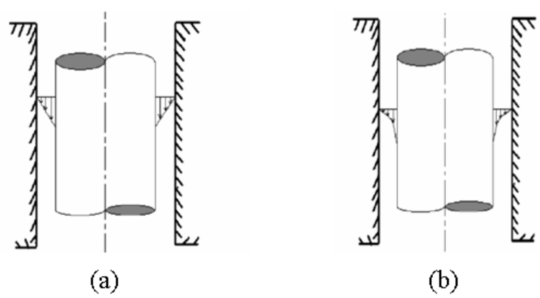

Figure 1. (a) linear and (b) nonlinear oil velocity profiles between piston and cylinder

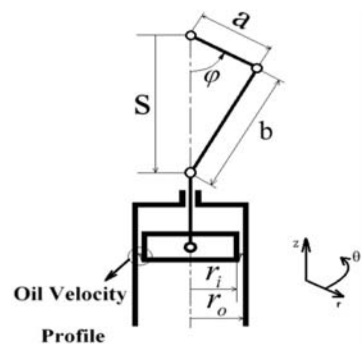

Figure 2. Schematic of a crank and slider mechanism and notations

TABLE I. CONSTANT PARAMETERS

Parameter	a (cm)	b (cm)	r_i (cm)	r_o (cm)	L (cm)	$\dot{\varphi}$ (rpm)	υ (cm^2/s)
Value	8	10	4	4.1	10	1500	0.417

The governing equation of the problem in cylindrical coordinate system has been derived using a number of assumptions. At first, the fluid flow is assumed to be laminar and in each crank angle, velocity profiles are assumed to be homogenous along the piston. Additionally, the oil particles are transferred only in the direction of piston movement and the movement of particles in other directions is negligible. Finally, oil temperature and viscosity are nearly constant since the dissipated power is calculated for one cycle during the steady state performance of the reciprocating compressor or other facilities. Thus, the simplified form of Navier-Stokes equation [20-22] can be derived for the problem in one dimension as follows:

$$\frac{\partial^2 u}{\partial r^2} + \frac{1}{r}\frac{\partial u}{\partial r} - \frac{\dot{\varphi}}{\upsilon}\frac{\partial u}{\partial \varphi} = \frac{g}{\upsilon} \tag{1}$$

The third term of (1) is local acceleration, while convective acceleration of piston is trifling. According to Fig. 2, for downward motion of piston, boundary conditions are [23]:

$$u(r_i) = \dot{s} = -a\,\dot{\varphi}\sin\varphi\left[1 + \frac{\cos\varphi}{\sqrt{\left(\frac{b}{a}\right)^2 - \sin^2\varphi}}\right] \quad ; \quad u(r_o) = 0 \tag{2,3}$$

Then, substituting s for φ gives

$$\frac{\partial u}{\partial \varphi} = \frac{\dot{s}}{\dot{\varphi}}\frac{\partial u}{\partial s} = -a\sin\varphi\left[1 + \frac{\cos\varphi}{\sqrt{\left(\frac{b}{a}\right)^2 - \sin^2\varphi}}\right]\frac{\partial u}{\partial s} \tag{4}$$

III. SOLUSION METHODS

A. Estimation of dissipated power using nonlinear oil velocity profile

The discretized computational domain is shown in Fig. 3. In this figure, the different crank angles show the different positions of piston along cylinder and the piston course is divided into eight equal parts only to show the velocity profiles clearly in Fig. 4 and 5 in specified crack angles. However, in numerical solution of the problem, the piston course is divided into 8000 equal parts. Then, discretized equation for each node of the grid can be derived using finite difference method for equally-spaced grid as:

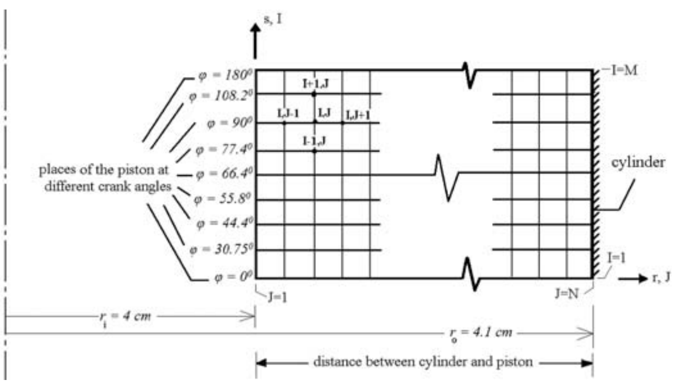

Figure 3. Coordinates and grids in the domain between piston and cylinder

$$\frac{u_{I,J+1}+u_{I,J-1}-2u_{I,J}}{(\Delta r)^2}+\frac{1}{r_{I,J}}\frac{u_{I,J+1}-u_{I,J}}{\Delta r}+$$

$$\frac{a\dot{\varphi}}{\upsilon}\sin\varphi_I\left[1+\frac{\cos\varphi_I}{\sqrt{\left(\frac{b}{a}\right)^2-\sin^2\varphi_I}}\right]\frac{u_{I+1,J}-u_{I,J}}{\Delta s}=\frac{g}{\upsilon} \qquad (5)$$

In which φ_I is the crank angle corresponding to the position 'I' along the piston course. The boundary conditions given by (2,3) can be written in the discretized form as:

$$u_{I,N}=0 \quad ; \qquad \text{for I=1,2,...,M} \qquad (6)$$

$$u_{I,1}=-a\dot{\varphi}\sin\varphi_I\left[1+\frac{\cos\varphi_I}{\sqrt{\left(\frac{b}{a}\right)^2-\sin^2\varphi_I}}\right]; \quad \text{for I=1,2,...,M} \quad (7)$$

There is no need to the initial condition or boundary condition for φ or s since when (5) is applied on nodes at φ_I =0 and φ_I =180, it can be solved independent of φ. After calculating velocity profile, shear stress over the entire outer surface of piston can be estimated using the law of viscosity for Newtonian fluids as follows: (Fig. 3)

$$\tau=\mu\frac{u_{I,1}-u_{I,2}}{\Delta r} \qquad (8)$$

Then, dissipated power which is product of shear force and piston velocity can be calculated as:

$$P_I=-a\dot{\varphi}\tau_I(2\pi r_i L)\sin\varphi_I\left[1+\frac{\cos\varphi_I}{\sqrt{\left(\frac{b}{a}\right)^2-\sin^2\varphi_I}}\right] \qquad (9)$$

Therefore, by integrating the dissipated power in each crank angle, the average dissipated power for half of a cycle becomes:

$$\overline{P}=\frac{1}{\pi}\int_0^\pi P\,d\varphi=\frac{1}{180}\sum_{I=1}^{M}\left[\frac{P_I+P_{I+1}}{2}\right](\varphi_{I+1}-\varphi_I) \qquad (10)$$

Finally, the dissipated power in vertical state of piston and cylinder is calculated by averaging the dissipated powers in the upward and downward motions of piston.

B. Estimation of dissipated power using linear oil velocity profile

For validation of results of nonlinear oil profile, the linear oil profile is used. According to Fig. 4, the linear oil profile is given by

$$u=\left(\frac{r_o-r}{r_o-r_i}\right)\dot{s} \qquad (11)$$

Also, using the law of viscosity for Newtonian fluids, the dissipated power is calculated as follows:

$$P=\tau(2\pi r L)\dot{s}=\left[\mu\left(-\frac{\dot{s}}{r_o-r_i}\right)\right](2\pi r_i L)\dot{s}=-\frac{2\pi\mu r_i L}{r_o-r_i}\dot{s}^2 \qquad (12)$$

Integrating (12) for half of a cycle gives the average dissipated power as following:

$$\overline{P}=-\frac{2\mu r_i L a^2\dot{\varphi}^2}{r_o-r_i}\int_0^\pi \sin^2\varphi\left[1+\frac{\cos\varphi}{\sqrt{\left(\frac{b}{a}\right)^2-\sin^2\varphi}}\right]^2 d\varphi \qquad (13)$$

To calculate the dissipated power in case of linear oil profile, there is no difference between upward and downward motions of the piston because the gravity term does not exist.

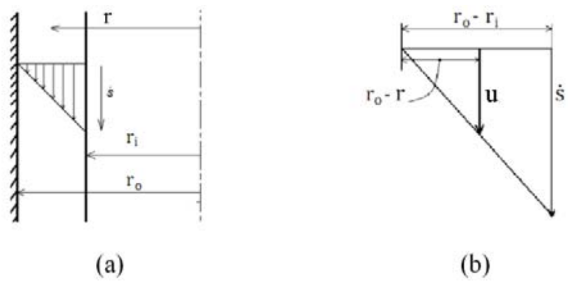

Figure 4. Linear velocity profile

IV. DESCUSSION AND RESULTS

For half of the cycle, when crank angle changes between 0° and 180° (Fig. 2), the nonlinear velocity profiles are shown for downward and upward motion of piston in Fig. 5 and 6 respectively.

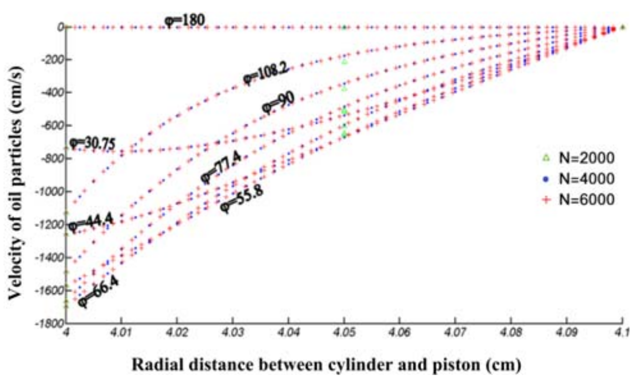

Figure 5. Velocity profiles for downward motion of piston (angle φ is in degree)

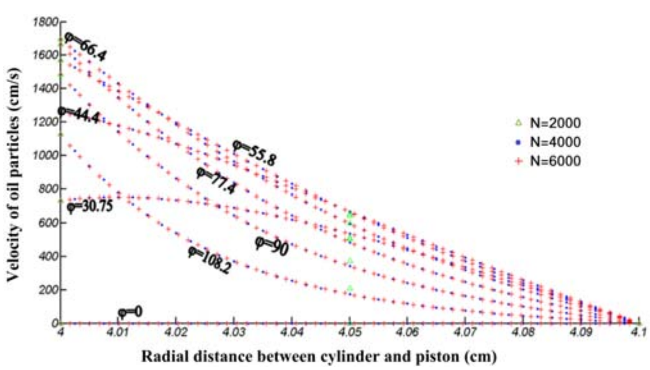

Figure 6. Velocity profiles for upward motion of piston (angle φ is in degree)

In these figures, crank angles are correspond to equally divided piston course to illustrate the rate of change of the velocity profile during a piston course. When the piston and cylinder are in horizontal position, the values of the velocity profiles are between the values of velocity profiles in upward and downward motions of piston. The dissipated power was found to be 150.03 watts for downward motion of piston and 157.77 watts for upward motion of piston and 153.94 watts for horizontal state of piston and cylinder which is roughly the average of the values of dissipated power in downward and upward motions of piston.

In Fig. 5, when $\varphi = 30.75°$, it can be seen that the diagram has a minimum which is due to high momentum of fluid when piston is going to be stopped at $\varphi = 0$ (Fig. 2).

Fig. 7 shows grid independency of the numerical results. The velocity profiles in this figure are given for the piston course at $\varphi = 66.4°$ as a test case. It can be seen that the velocity profiles for M=6000 and M=8000 are nearly coincident and the maximum difference between these two cases is less than 2%.

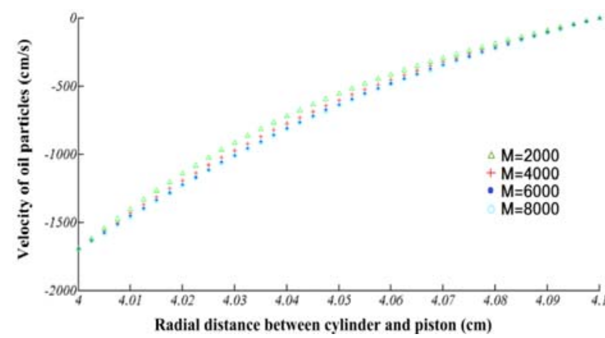

Figure 7. Grid independency of the results in radial direction

Results show that the dissipated power calculated using nonlinear velocity profile is more than that of linear velocity profile. As illustrated in Fig. 8, for a specified φ and $\dot{s}$, $\Delta r_N < \Delta r_L$ where Δr_N denotes Δr for nonlinear velocity profile and Δr_L denotes Δr for linear velocity profile. The law of viscosity for Newtonian fluids implies that Δr has inverse relationship with shear force, and dissipated power. As shown in Fig. 8, in case of linear velocity profile, Δr is higher than that of nonlinear velocity profile. Therefore, the shear force and power for linear velocity profile should be less than those of nonlinear velocity profile. In fact, the dissipated power is estimated 90.58 watts for linear velocity profile and 153.9 watts for nonlinear velocity profile.

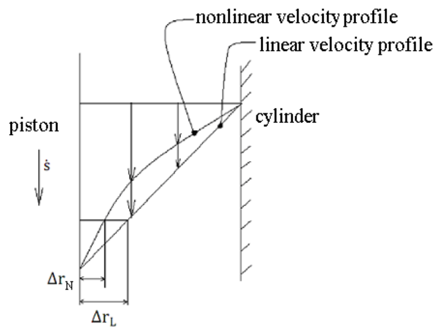

Figure 8. Comparison of the results of linear and nonlinear oil velocity profiles

V. CONCLUSION

Dissipated power, which is an imperative factor in installation, design, analysis, and optimization of reciprocating compressors, presses, pumps, and other facilities, caused by oil viscosity has been evaluated in this paper. To find the amount of dissipated power, nonlinear and linear oil velocity profiles assumed between piston and cylinder and using numerical and analytical methods respectively, results of these two approaches were compared with each other. In addition, using the finite difference method, the effect of the geometrical parameters and oil properties on dissipated power can be analyzed.

In the case of nonlinear oil velocity profile, the shear stress and dissipated power in upward motion of piston were higher than those in downward motion of piston. Indeed, the dissipated power for downward motion of piston (150.03 watts) was less than that of horizontal state of piston and cylinder (153.94 watts) and it was less than the dissipated power for upward motion of piston (157.77 watts). Furthermore, the dissipated power for a cycle in horizontal position of piston and cylinder was nearly equal to the average of dissipated power in upward and downward motions of piston. These slight differences were due to the low weight of oil film and all of the differences between dissipated powers were physically acceptable. Moreover, according to Fig. 8, dissipated power which is calculated using linear velocity profile should be less than that of nonlinear velocity profile which was consistent with the results. The difference between the results of linear and nonlinear velocity profiles is primarily function of clearance between piston and cylinder and this fact is described in Fig. 8.

VI. NOMENCLATURE

a crank length (m)
b connecting rod length (m)
L piston length (m)
M number of grid points in s direction
N number of grid points in r direction
P dissipated power (W)
p pressure (pa)
$\overline{P}$ average dissipated power for half of cycle (W)
r radial coordinate (m)
r_i radius of piston (m)
r_o radius of cylinder (m)
s piston path (m)
$\dot{s}$ linear velocity of piston (m s^{-1})
u oil velocity profile in z direction (m s^{-1})
$u_{I,J}$ IJ th node velocity (m s^{-1})
z vertical coordinate (m)
$\Delta\varphi$ finite difference in φ direction (deg)
Δr finite difference in r direction (m)
φ crank angle (deg)
$\dot{\varphi}$ angular velocity of crank (rad s^{-1})
θ angular coordinate (deg)
μ oil dynamic viscosity (N s m^{-2})
υ oil kinematic viscosity (m^2 s^{-1})
τ shear stress (pa)

REFERENCES

[1] M. Razaghi, Analysis, computation and solution of Navier-Stokes Equations for the study of dissipated power that caused by consumed oil and also rate of oil consumption in reciprocating systems without oil ring and with crank and slider driver. Bachelor thesis, KIAU, 2009.

[2] D.F. Li, S.M. Rohde, and H.A. Ezzat, "An automotive piston lubrication model,". ASLE Trans., vol. 26, 1983, pp. 151-160.

[3] D. Zhu, H.S. Cheng, T. Arai, and K. Hamai, "A numerical analysis for piston skirts in mixed lubrication part: basic modeling," ASME J. of Tribol., vol. 114, 1992, pp. 553-562.

[4] D. Zhu, Y.Z. Hu, H.S. Cheng, T. Arai, and K. Hamai, "A numerical analysis for piston skirts in mixed lubrication part 2: deformation consideration," ASME J. of Tribol., vol. 115,1993, pp. 125-133.

[5] K. Gommed and I. Etsion, "Dynamic analysis of gas lubricated reciprocating ringless piston: basic modeling," ASME J. of Tribol., vol. 115, 1993, pp. 207-213.

[6] K. Gommed and I. Etsion. "Parametric study of the dynamic performance of gas lubricated ringless piston," ASME J. of Tribol., vol. 116, 1994, pp. 63-69.

[7] I. Etsion and K. Gommed, "Improved design with non-cylindrical profile of gas lubricated ringless piston," ASME J. of Tribol., vol. 117, 1995, pp. 143-147.

[8] A.T. Prata, J.R.S. Fernando and F.Fagotti, "Dynamic analysis of piston secondary motion for small reciprocating compressor," ASME J. of Tribol., vol. 122, 2000, pp. 752-760.

[9] T.J. Kim, "Numerical analysis of the piston secondary dynamics in reciprocating compressors," KSME Int. J., vol. 17, 2003, pp. 350-356.

[10] Y. Jeng, "Theoretical analysis of piston-ring lubrication part 1: fully flooded lubrication," STLE Tribol .Trans., vol. 35, 1992, pp. 696-706.

[11] A. Ronen, I. Etsion, and Y. Kligerman, "Friction-reducing surface-texturing in reciprocating automotive components," STLE Tribol. Trans., vol. 44, 2001, pp. 359-366.

[12] J. Zhao, F. Sadeghi, and H.M. Nixon. "A finite element analysis of surface pocket effects in hertzian line contact". ASME J. of Tribol., vol. 122, 2000, pp. 47-54.

[13] G. Ryk, Y. Kligerman, and I. Etsion, "Experimental investigation of laser surface texturing for reciprocating automotive components," STLE Tribol. Trans., vol. 45, 2002, pp. 444-449.

[14] N.W. Bolander and F. Sadeghi, "Surface modification for piston ring and liner," Solid. Mech. Appl., vol. 134, 2006, pp. 271-283.

[15] M.F. Ming and Z.Y. Yun. "Influence of oil film inertia on piston skirt lubrication in high speed engine," Shanghai Univ., vol. 7, 2003, pp. 285 - 288.

[16] K.J. Young, K.J. Woon, C.M. Rae, and H.D. Chul, "Friction characteristics of piston ring pack with consideration of mixed lubrication parametric investigation," KSME Int. J., vol. 16, 2002, pp. 468-475.

[17] M.R. Choi, J.K. Choi, and D.C. Han, "Calculation of mixed lubrication at piston ring and cylinder liner interface," KSME Int. J., vol. 15, 2001, pp. 859-865.

[18] F.M. White, Viscous Fluid Flow, 2nd ed., Singapore: McGraw-Hill, 1991, pp. 104-217.

[19] J.R. Cho and S.J. Moon, "A numerical analysis of the interaction between the piston oil film and the component deformation in a reciprocating compressor," Tribol. Int., vol. 38, 2005, pp. 459-468.

[20] H.S. Joseph and A. Nuri, Fluid Mechanics, Berlin: Springer, 1997, pp. 229-259.

[21] T. Xu, W.J. Zuo, T.S. Xu and C.J. Ge, "Numerical simulation of fluid-solid coupling for lubricating oil extrusion process between piston and cylinder with micro-pits," J. Tribol., vol. 29, 2009, 340-344.

[22] I.H. Shames, Mechanics of Fluids, 4th ed, New York: McGraw-Hill Inc, 2003, pp.163-243.

[23] J.E. Shigley and J.J. Uicker, Theory of machines and mechanisms, 2nd ed, New York: McGraw-Hill Inc, 1995, pp. 89-148.