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Lubrication characteristics of dual piston ring in bent-axis type piston pumps †

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Abstract

The bent-axis type of piston pump driven by the piston rod works by the piston rod driving the cylinder block; because of this the taper angle of the piston rod and the swivel angle between the cylinder block and the shaft are important design factors. If these factors cannot satisfy the conditions for optimum design, the friction loss between the cylinder bore and the piston increases, and the pump can fail to work under conditions of severe friction and wear. Since the piston reciprocates in the cylinder bore with high velocity, at the same time rotating on its own axis and revolving on the center of the cylinder block, a decrease of the volume efficiency is generated because of the leakage between the cylinder bore and the piston. Therefore, to prevent this, the piston ring is designed to be at the end of the piston, and the friction characteristics between the piston ring and the cylinder bore require further research due to their great influence on the performance of the piston pump. Thus, in this paper, the elastohydrodynamic lubrication (EHL) analysis of the film thickness, the pressure distribution, and the friction force, have been studied between the piston ring and the cylinder bore in the bent-axis type of piston pump. The analyzed results show that the friction force is influenced by the rotating speed and the discharge pressure.

Keywords: Oil hydraulic piston pump; Elasto-hydrodynamic lubrication (EHL); Piston ring; Friction force

1. Introduction

Recently, the oil hydraulic system, which is used in large and heavy equipment, has been at a disadvantage due to its restriction of performance, its environmental hazards, and its continuous noise. To solve these problems, a variety of research studies have been performed, such as the compact style oil hydraulic systems, high speed and high pressure electronic control systems, substitute oil, and reducing noise.

The bent-axis type of oil hydraulic piston pump acts as the core power source of oil hydraulic systems, which is typical in such applications. Such pumps are used as the main pump in heavy construction equipment due to their high speed and pressure, high total efficiency, and distinguished variable delivery.

Since the piston is tapered in a bent-axis type piston pump, to prevent leakage in the sliding part between the cylinder bore and the piston, the piston ring is designed to be at the end of the piston. Therefore, the friction and leakage characteristics in this sliding part have an influence on the performance of the piston pump.

Various research studies mentioned above have been carried out, for example, ElastoHydro-dynamic Lubrication [1] (EHL), the lubrication characteristics of a rotary compressor used for refrigeration and air-conditioning systems [2], a study of the tribological behavior of piston ring/cylinder liner interaction in diesel engines using acoustic emission [3], and so on.

In this paper, the EHL analysis of the film thickness, the pressure distribution and the friction force have been studied in the sliding part between the piston ring and the cylinder bore in the bent-axis type piston pump.

2. Theoretical analysis

Fig. 1 shows a diagram of the bent-axial type of piston pump where the mechanism is very complex because the piston is tapered in the bent-axis type piston pump. Therefore, the mechanical analysis between the piston ring and cylinder bore is accomplished, and then basic data for the EHL is obtained.

Fig. 2 shows the behavior of the piston and the piston ring in the cylinder bore. The boundary conditions are very complex because the pressure changes from the suction pressure to the discharge pressure during one rotating speed.

It is necessary to access the EHL case for direct lubrication

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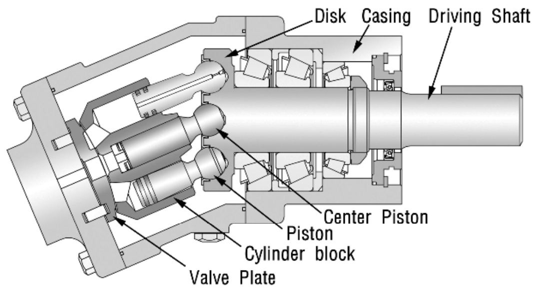
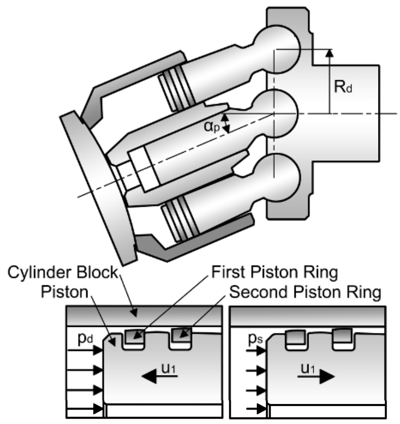


Fig. 1. Diagram of an axial piston pump.



(a) Delivery region (b) Suction region

Fig. 2. Behavior of the piston and the piston ring in the cylinder bore.

analysis, because the contact surface will incur elastic deformation at relative motion between the cylinder bore and piston ring.

Thus, in this paper, EHL analysis is carried out on the relative motion part between the cylinder bore and piston ring. The analysis model and boundary condition of the relative motion part are expressed in Fig. 3 [4], which shows the model between the piston ring and cylinder bore. In this figure, the piston and piston ring are moved by the sliding velocity u_1 on the fixed cylinder bore (u_2=0). R is the radius of the piston ring, P is the fluid pressure, h is the film thickness, and w is the load of the piston ring.

The assumptions for the analysis are as follows [5]:

(1) The flow in the y-direction of the lubricant is ignored because the contact film thickness of the elastic body is very small.
(2) The deformation of the elastic body is a plane deformation condition of the unlimited body.
(3) The lubricant is the uncompressed characteristic.
(4) The viscosity of the lubricant is expressed as an index coefficient of the pressure.
(5) The viscosity of the lubricant is not concerned with changes in temperature.

2.1 Basic equation of lubrication

Reynolds Equation in hydrodynamic lubrication is as follows [1]:

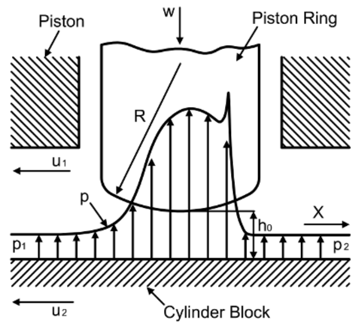


Fig. 3. Model between the piston ring and the cylinder bore.

$$h^3 \frac{dp}{dx} = 12u\eta(h - h_m) + 12\eta V_1(x - x_m) \tag{1}$$

This equation shows that the first term is the wedge effect, and second term is the squeeze effect. Here, x is the coordinate of the lubrication region, x_m is the x coordinate at the maximum pressure position (dp/dx=0), h_m is the film thickness at the maximum pressure position (dp/dx=0), and V_1 is the y-direction speed of the piston ring.

The average velocity u is expressed by the following equation:

$$u = u_1/2 = R_d w \sin\alpha_p \sin\theta/2 \tag{2}$$

Here $P_1=P_d$, $P_2=P_a$ at $u_1\geq 0$, and $P_1=P_a$, $P_2=P_s$ at $u_1<0$; also P_1 is the input pressure, P_2 is the output pressure, P_d is the discharge pressure, P_a is the atmospheric pressure, P_s is the suction pressure, θ is the rotating angle of the shaft, ω is the angular velocity of the shaft, α_p is the swivel angle, R_d is the pitch circle radius of the disk spherical surface part.

The viscosity coefficient η is expressed by the following equation:

$$\eta = \eta_0 \exp(\alpha p) \tag{3}$$

where η_0 is the viscosity of the lubricant at the atmospheric pressure, α is the pressure-viscosity coefficient. The film thickness h and the elastic deformation v are obtained as follows:

$$h = h_0 + \frac{x^2}{2R} + v \tag{4}$$

$$v = -\frac{2}{\pi E'}\int_{x_b}^{x_a} p(s)\ln(x-s)^2 ds + C_0 \tag{5}$$

where $\frac{1}{E'} = \frac{1}{2}(\frac{1-\sigma_1^2}{E_1} + \frac{1-\sigma_2^2}{E_2})$, h_0 is the film thickness at x=0, C_0 is the integral calculus constant, E' is the equivalent Young's modulus, σ_1, σ_2 are the Poisson ratios, and E_1, E_2 are the Young's modulus.

The unit load w on the piston ring is expressed by the following equation [6]:

Table 1. Geometrical data and the operating conditions.

Item	Values	Unit
Suction pressure	-0.05	MPa
Discharge pressure	30,35,40	MPa
Rotating speed	1500,1800,2100	rpm
Oil viscosity (at 40℃)	46	cP
Diameter of cylinder bore	25	mm
Diameter of pitch circle in piston head	73.5	mm
Width of piston ring	3.2	mm
Swivel angle	29	º
Curvature radius of equivalent cylinder	12.5	mm
Cylinder bore ・Young's modulus ・Poisson's Ratio	 97 0.311	GPa
Piston ring ・Young's modulus ・Poisson's Ratio	 210 0.3	GPa

$$w = \int_{x_a}^{x_b} p(x)dx \tag{6}$$

where x_a is the starting point of the analysis, and x_b is the ending point of the analysis. The boundary condition to solve Eq. (1) is expressed by the following equation:

$$p = p_1 \quad \text{at} \quad x = x_a$$
$$p = p_2, \quad \frac{\partial p}{\partial x} = 0 \quad \text{at} \quad x = x_b$$

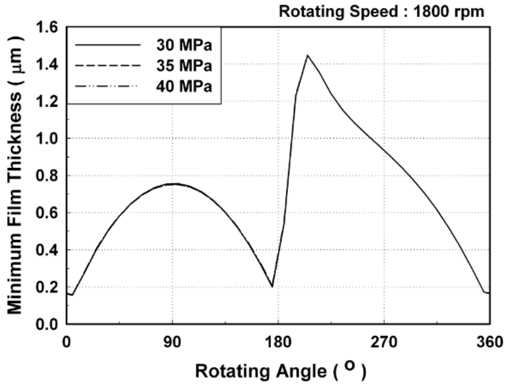
where the unit friction force is obtained as follows:

$$f = \int_{x_a}^{x_b} \frac{1}{2}\frac{\partial p}{\partial x}dx - \int_{x_a}^{x_b} 2\eta\frac{u}{h}dx \tag{7}$$

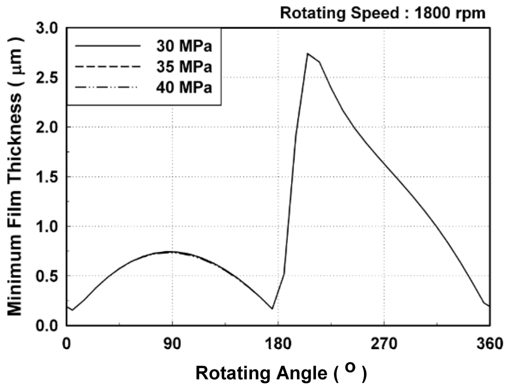
3. Theoretical results

The driving conditions presented in Table 1, have been used to perform the theoretical input conditions for the vertical load, the piston velocity, etc. Under these conditions, the EHL characteristics are studied, between the piston ring and the cylinder bore, at the bent-axis type piston pump.

Fig. 4 shows the minimum film thickness when the rotating speed is 1,800rpm and how the discharge pressure changes. In the figure, the discharge region is between 0º and 180º, and the suction region is between 180º and 360º. The minimum film thickness at the discharge region (0º-180º) is lower than the suction region (180º-360º) because the discharge pressure is acting on the piston ring at the delivery region. Comparing the first ring with the second ring, the minimum film thickness at the discharge region shows that the first ring is higher than the second ring, and at the suction region the first ring is lower than the second ring. The film thickness is seen to have no influence on the discharge pressure. Then, when the rotating angle is 0º and 180º, since the velocity of the piston becomes zero, but the minimum film thickness is not zero, this reason is considered due to the squeeze effect.



(a) First ring



(b) Second ringe

Fig. 4. Minimum film thickness for the discharge pressure.

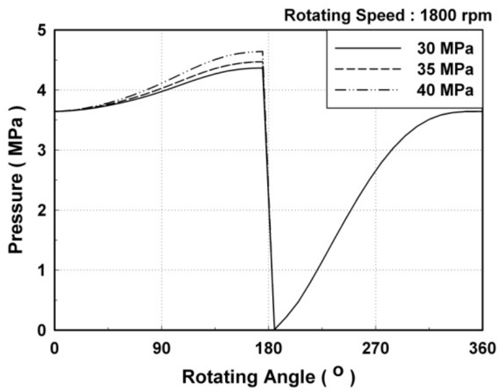
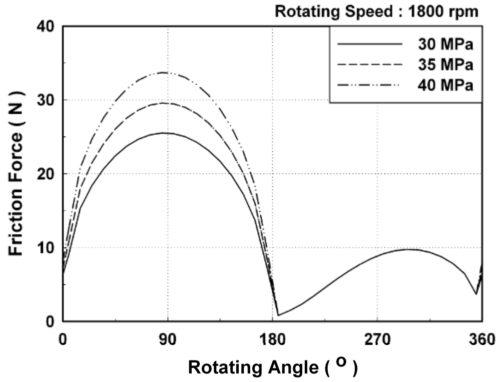
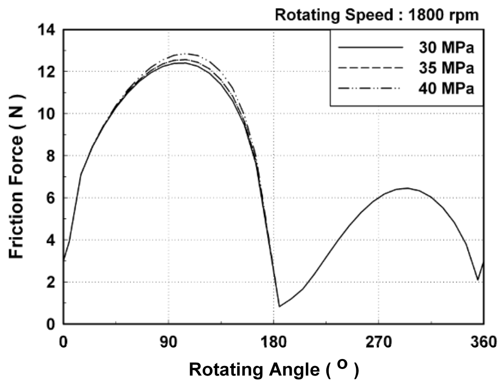


Fig. 5. Pressure between the first ring and the second ring for discharge pressure.

Fig. 5 shows the pressure between the first ring and the second ring when the rotating speed is 1,800rpm and the discharge pressure is changed. The pressure at the discharge region increases due to the discharge pressure, and the pressure at the suction region is consistent due to the consistent suction pressure. The pressure between the piston rings at the rotating angle 180º suddenly drops, since the location of the piston ring rapidly relocates on the other side.

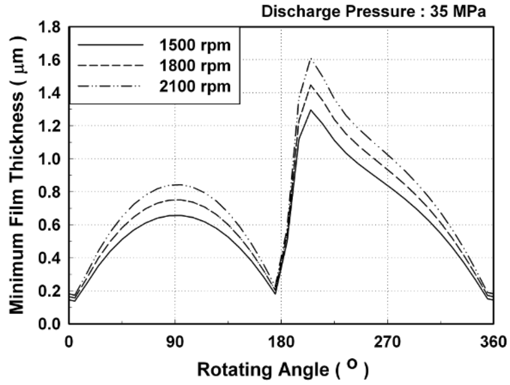


(a) First ring

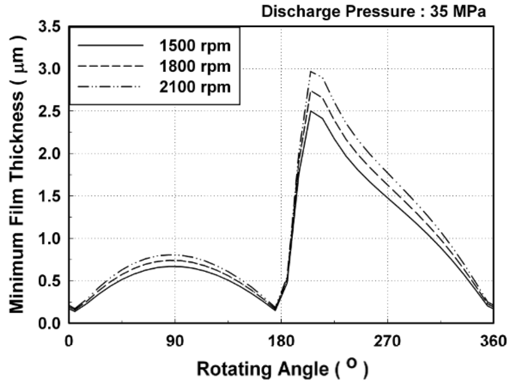


(b) Second ring

Fig. 6. Friction force for discharge pressure.



(a) First ring



(b) Second ring

Fig. 7. Minimum Film thickness for various rotating speeds.

Fig. 6 shows the friction force when the rotating speed is 1,800rpm and the discharge pressure is changed. The normal force is proportionally increased to the discharge pressures and the friction force is also increased in the discharge region, but the friction force is very small in the suction region because the suction pressure is small at -0.05Mpa. Comparing the first ring with the second ring, the friction force at the first ring is higher than the second ring. This is because the first ring is directly affected by the discharge pressure, but the second ring is affected by the pressure between the piston rings.

Fig. 7 shows the minimum film thickness when the discharge pressure is 35Mpa and the rotating speed is changed. In the figure, the minimum film thickness at the discharge region is lower than the suction region. There is no difference in the film thickness of the first ring and the second ring at the discharge region, but the film thickness of the second ring at the suction region is higher than the first ring by almost being double. As the rotating speed increases, based on the proportional increase of the minimum film thickness, the minimum film thickness shows a proportional relationship with the rotating speed. Due to the wedge effect, the increase in rotating speed will increase the film thickness more.

Fig. 8 shows the pressure between the first ring and the second ring, when the rotating speed is changed and the discharge pressure is 35Mpa. The increase in the rotating speed can be seen by the increase in the pressure between the piston rings.

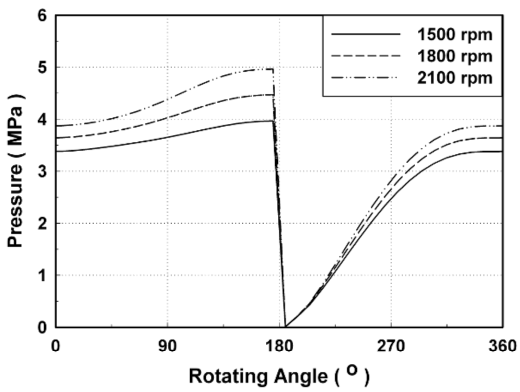
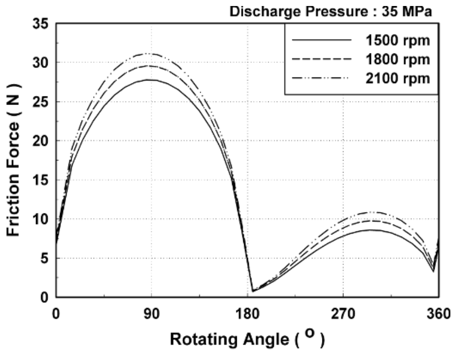


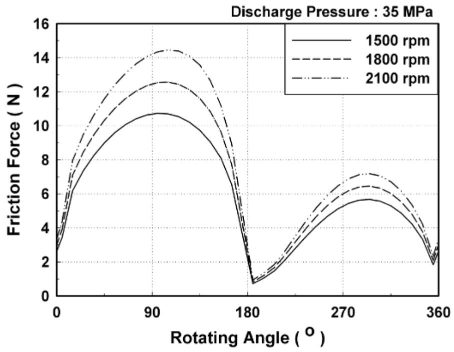
Fig. 8. Pressure between the first ring and the second ring for various rotating speeds.

The discharge pressure increases have an effect on the pressure of the discharge region, but not on the pressure of the suction region. Therefore, the pressure between the piston rings shows the increases by the growth in the inflow.

Fig. 9 shows the friction force when the discharge pressure is 35Mpa and the rotating speed is changed. As the rotating speed increases, the friction force shows the increases. Also, the friction force at the discharge region is higher than that of the suction region because the increase of the relative sliding speed increases the viscosity friction force. Comparing the

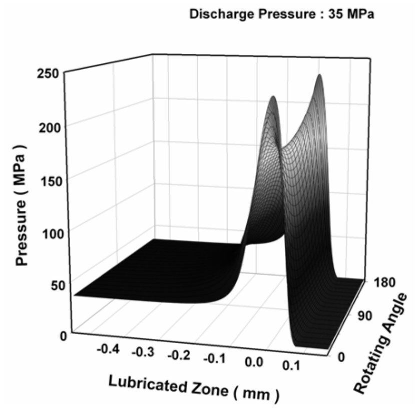


(a) First ring

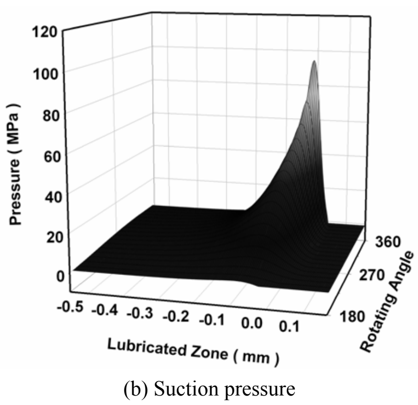


(b) Second ring

Fig. 9. Friction force for various rotating speeds.

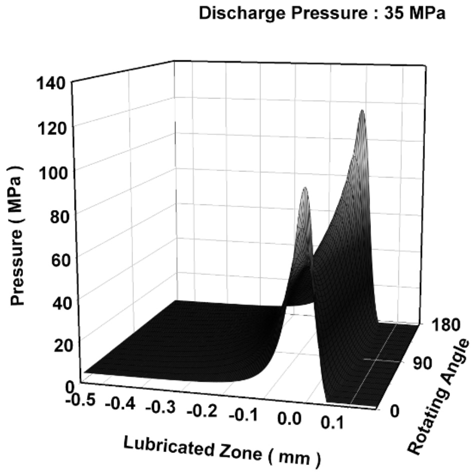


(a) Discharge pressure

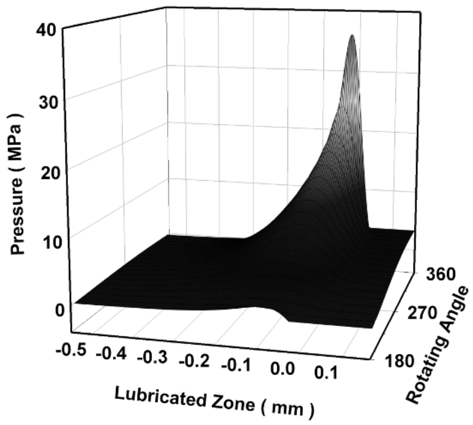


(b) Suction pressure

Fig. 10. Pressure distribution between the first ring and the cylinder bore (the delivery region).



(a) Discharge pressure



(b) Suction pressure

Fig. 11. Pressure distribution between the second ring and the cylinder bore (the delivery region).

first ring with the second ring, the friction force at the first ring is higher because the first ring is directly affected by the discharge pressure, but the second ring is affected by the pressure between the piston rings. Therefore, it is known that the friction force at the second ring is lower, and the pressure change at the second ring is smaller.

Fig. 10(a) shows the pressure distribution between the first ring and the cylinder bore at the discharge region, when the discharge pressure is 35Mpa and the rotating speed is 1,800rpm. The pressures of both ends appear to be more than at the middle point. The reason for this is that the decreases in the lubrication region, due to the decreases in the velocity of the piston, increase the maximum pressure. Fig. 10(b) shows the pressure distribution between the first ring and the cylinder bore at the suction region. As the rotating speed increases, the suction pressure can be seen to increase. The reason for this is that the suction pressure is the minus (-) value and the pressure between the piston rings are zero because the location of the piston rings is a conversion to the other sides. Soon after, as the rotating angle increases, the inflow of the flow rate increases and the pressure between the piston rings also increases. Therefore, the pressure between the piston ring and the cylinder bore increases as well.

Fig. 11(a) shows the pressure distribution between the second ring and the cylinder bore at the discharge region when the pressure is 35Mpa and the rotating speed is 1,800rpm. In this figure, as shown in Fig. 10, the pressures of both ends appear to be more than at the middle point. Fig. 11(b) shows the pressure distribution at the suction region. As the rotating speed increases, the suction pressure increases can be seen. As compared with the distribution pressure of the first ring, it can be seen that the maximum pressure of the first ring is higher than that of the second ring by almost two times, and the maximum pressure between the piston ring and the cylinder bore occurs locally in Figs. 10 and 11. Therefore, EHL is necessary for the analysis of the lubrication characteristics between the piston ring and the cylinder bore.

4. Conclusions

Based on theoretical analyses of the lubrication characteristics between the dual piston ring and the cylinder bore, the following conclusions can be obtained:

(1) Through the pressure distribution, the lubrication state between the piston ring and the cylinder bore shows that a lubrication region exists within the hydrodynamic and elasto-hydrodynamic lubrication region.
(2) In the given conditions, the film thickness between the piston ring and cylinder bore increases as the rotating speed increases, but is barely affected by changes of the pressure.
(3) As the discharge pressure and rotating speed increase, the friction force between the piston ring and cylinder bore increases.

In addition, through comparison and review of the results, it is known that the lubrication region between the piston ring and cylinder bore exists at the hydrodynamic lubrication and elastohydrodynamic lubrication region. Therefore, this study will be used as a basic foundation for future related research studies.

Nomenclature

C_0 : Integral calculus constant
f : Friction force per unit length
h : Film thickness at arbitrary x
h_m : Film thickness at dp/dx=0
h_0 : Film thickness at x=0
p : Pressure at arbitrary x
p_a : Atmospheric pressure
p_d : Discharge pressure
p_s : Suction pressure
R : Curvature Radius between equivalent cylinder bore and piston ring
R_d : Pitch circle radius of disk spherical surface part
u : Average velocity, $(u_1+u_2)/2$
w : Unit load
v : Elastic deformation
V_1 : Y-direction speed of piston ring
x : Coordinate of lubrication region
x_a, x_b : Stating point and ending point of analysis
x_m : X coordinate at maximum pressure position (dp/dx=0)
α : Pressure-viscosity coefficient
α_p : Swivel angle between cylinder block and shaft
η : Viscosity coefficient
η_0 : Viscosity coefficient at atmospheric pressure
θ : Rotating angle of shaft
σ_1, σ_2 : Poisson ratio
ω : Angular velocity of shaft
$E`$: Equivalent Young's modulus
E_1, E_2 : Young's modulus

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