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# Calculation of Oil Film Thickness from Damping Coefficients for a Piston Ring in an Internal Combustion Engine

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**Abstract:** In 1966 Jorgen W. Lund [1] published an approach to find the dynamic coefficients of a journal bearing by a first order perturbation of the Reynold's equation. These coefficients made it possible to perform a rotor-bearing stability analysis for a statically loaded bearing. In the mid seventies Jorgen W. Lund pointed out in lecture notes that the dynamic damping coefficients of the bearing could be used to find the shaft orbit for dynamically loaded bearings. In the present paper this method is further developed and utilized to determine the dynamic behavior of a piston ring in a combustion engine. The basic idea is to use the fluid film damping coefficients to estimate the film thickness variation for a piston ring under cyclic varying load. Reynolds Equation is solved for a piston ring and the oil film thickness is determined. In this analysis hydrodynamic lubrication is assumed and the effect of the squeeze motion is considered. Cavitation zones are taken into account and the piston ring is assumed to be fully flooded. Governing equations and numerical schemes are presented. Numerical computations show good correlation with analytical results.

**Keywords:** Piston Rings, Cavitation, Prediction of oil film thickness, Engine lubrication.

## 1    Introduction

Internal combustion engines are used for many purposes, e. g. ship propulsion. The classic method to analyze the interface between the piston ring and cylinder liner is to solve Reynolds Equation by integration and calculate e. g. oil film thickness. This is shown by [2]. In [1] is presented a method to determine the dynamic coefficients for a journal bearing. Volund [3] developed from Lund a method to calculate the orbit of the cross head in an internal combustion engine. This paper describes how this method is implemented for a piston ring.

## 2    Reynolds Equation

In this analysis endflow is neglected because $b \ll l$. Reynolds Equation for a one dimensional flow is shown in equation (1).

$$\frac{\partial}{\partial x}\left(\frac{h^3}{12\eta}\frac{dp}{dx}\right) = \frac{u}{2}\frac{\partial h}{\partial x} + \frac{\partial h}{\partial t} \tag{1}$$
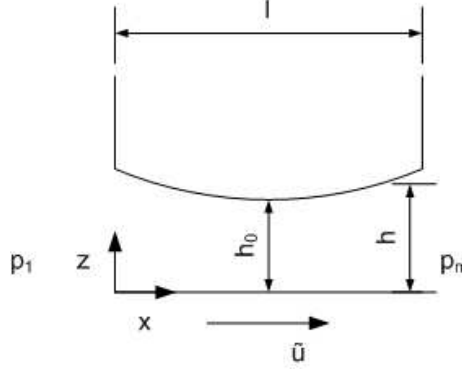
Figure 1: Shape of piston ring

## 3 Equilibrium of Forces

Equilibrium of forces is found by Newtons 2. Law. The forces from the mass of the piston ring is neglected, because this is considered to be small compared to the film forces. The forces acting on the piston ring are the external load and the film forces from the oil. This gives the equilibrium in force seen in equation (2).

$$F'_{ext} - W' = 0 \tag{2}$$

## 4 Geometry of Piston Ring

A piston ring in an engine is worn over time, and the initial shape of the ring changes to be roughly approximated by a circle or parabola. The piston ring shape used in the present paper is set to be parabolic. The piston ring shape are seen in figure 1.

## 5 Solution from damping coefficients

Solutions to equation 2 in time are found by perturbation approach. This can be done by introducing a perturbation in the velocity.
The piston ring is given a small perturbation in velocity $\Delta \dot{h}$. This leads to a perturbation in the pressure $\Delta p$. The perturbed pressure and velocity are given by equation (3).

$$\dot{h} = \dot{h} + \Delta \dot{h}$$
$$p = p_0 + p_{\dot{h}} \Delta \dot{h} \tag{3}$$

The perturbation pressure and perturbation velocity are introduced in Reynolds Equation. This is seen in equation (4).

$$\frac{\partial}{\partial x}\left(\frac{h^3}{12\eta}\frac{\partial}{\partial x}(p_0 + p_{\dot{h}}\Delta\dot{h})\right) = \tilde{u}\frac{\partial h}{\partial x} + \dot{h} \tag{4}$$

This equation is divided into two equations (5) and (6).

$$\frac{\partial}{\partial x}\left(\frac{h^3}{12\eta}\frac{\partial p_0}{\partial x}\right) = \tilde{u}\frac{\partial h}{\partial x} + \dot{h} \tag{5}$$

$$\frac{\partial}{\partial x}\left(\frac{h^3}{12\eta}\frac{\partial p_{\dot{h}}}{\partial x}\right) = 1 \tag{6}$$

Equations (5) and (6) are solved by the Finite Difference Method. The pressure distribution is found by applying the Modified Reynolds Cavitation criteria seen in (7).

$$p = p_1 \quad at \quad x = x_1 \qquad and \qquad \frac{dp}{dx} = 0 \quad at \quad x = x^* \tag{7}$$

Here the damping is calculated from the perturbation pressures in equation (8).

$$c' = \int p_h dx \tag{8}$$

The load carrying capacity is calculated from the pressure distribution, $p_0$ in equation (9).

$$W' = \int_0^l p_0 dx \tag{9}$$

A new value of the perturbation velocity is calculated from an equilibrium in forces seen in equation (10).

$$F'_{ext} = W' + c' \Delta \dot{h}_i \Leftrightarrow \Delta \dot{h}_i = \frac{F'_{ext} - W'}{c'} \tag{10}$$

The new $h_0$ is calculated by equation (11).

$$\dot{h}_{i+1} = \dot{h}_i + \Delta \dot{h}_i \tag{11}$$

When convergence between film forces and outer forces is obtained, the new position of the piston ring is calculated by assuming that the velocity and damping are constant over a time step. This is done by Eulers algorithm.

## 6    Performance Parameters

The flow is given by equation (12).

$$q'_x = -\frac{h^3}{12\eta} \frac{\partial p}{\partial x} + h\tilde{u} \tag{12}$$

The shear stresses are seen in equation (13).

$$\tau_{liner} = -\frac{h}{2} \frac{\partial p}{\partial x} - \frac{\eta(u_{ring} - u_{liner})}{h} \quad and \quad \tau_{ring} = -\frac{h}{2} \frac{\partial p}{\partial x} + \frac{\eta(u_{ring} - u_{liner})}{h} \tag{13}$$

The friction forces are then found by integrating the shear stresses. These are shown in equation (14).

$$F'_a = \int \tau_{liner} dx \quad and \quad F'_b = \int \tau_{ring} dx \tag{14}$$

The power loss is calculated from the friction force. This is seen in equation (15).

$$h'_{p,liner} = -F'_{liner} u_{liner} \quad and \quad h'_{p,ring} = -F'_b u_{ring} \tag{15}$$

The coefficient of friction at is calculated from equation (16).

$$\mu_{liner} = \frac{F'_{liner}}{W'} \quad and \quad \mu_{ring} = \frac{F'_{ring}}{W'} \tag{16}$$

The center of the pressure distribution is calculated by equation (17). This is the position where the resulting force is acting.

$$x_{cp} = \frac{\int px dx}{W'} \tag{17}$$

| Description | Size/Number/Type |
|:---:|:---:|
| $l$ | $9,5mm$ |
| $s_h$ | $9,5\mu m$ |
| $\eta$ | $0,01Pas$ |

Table 1: Piston ring geometry

# 7    Results from simulation

Data from MAN Diesels research engine, T50mx are used simulate the top ring in the piston. The data, shown in figure 2 contains information about the boundary pressure and velocity of the ring. Parameters to describe the piston ring shape and lubrication oil, type SAE 50 is presented in table 1.
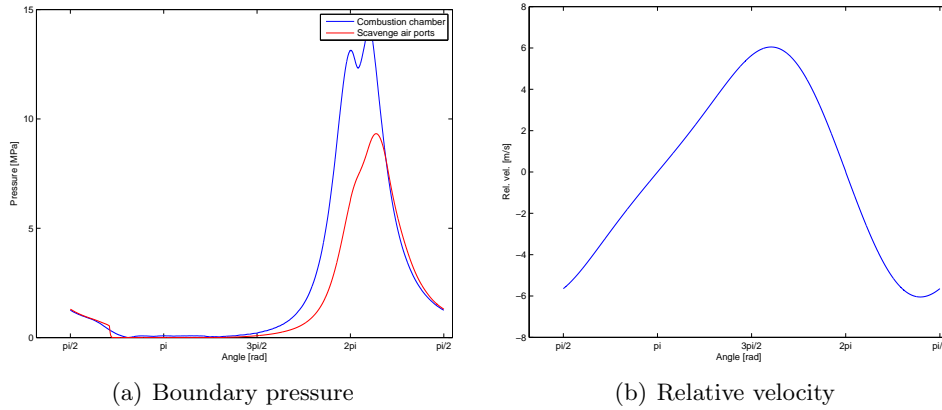


(a) Boundary pressure                    (b) Relative velocity

Figure 2: Input parameters

The simulation is started in the midpoint between TDC and BDC where the velocity has reached maximum and the boundary pressures are relatively small.

**Stroke from $\pi/2$ to $\pi$**
The velocity is decreasing which gives smaller load carrying capacity. At first the OFT is decreasing until the ring passes the scavenge air ports, where the pressure drops. The pressure drop leads to a smaller load behind the ring, which gives an increase in OFT. The steady decreasing velocity provides decreasing load carrying which makes the OFT decrease again. The fluctuations in OFT leads to oscillations in center of pressure and cavitation boundaries.

**Stroke from $\pi$ to $3\pi/2$**
In this part of the stroke the velocity is increasing. At first the boundary pressures are constant, which provides an increase in oil film thickness. Then the pressure increases because the combustion chamber volume gets smaller, which leads to decreasing oil film thickness.

**Stroke from $3\pi/2$ to $2\pi$**
The velocity gets smaller contemporary with a pressure increase. In TDC the pressure reaches maximum and the velocity minimum over the cycle. This combination results in a steady decreasing oil film thickness and the minimum oil film thickness is observed in TDC. The power loss gets larger very fast.

**Stroke from 0 to $\pi/2$**
The velocity increases and the overall pressure drops. An oscillation in combustion chamber pressure results in an oil film thickness peak of the concurrent increasing oil film. The peak can also be observed in the squeeze velocity and cavitation boundary.

# 8  Conclusion

From dynamic coefficients have the oil film thickness and other performance parameters been predicted for a diesel engine. The results shows that the minimum oil film thickness arises when the piston is in the TDC.

# 9  Acknowledgements

# References

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[2] D. Dowson M. Priest, R. I. Taylor and C. M. Taylor. Boundary conditions for reynolds equation with particular reference to piston ring lubrication. *Tribology series; 31*, pages pp. 441–452, 1996.

[3] Anders Volund and Peder Klit. Guide bearing charateristics. *9 th Nordic Symposium on Tribology, Nordtrib*, 2000.
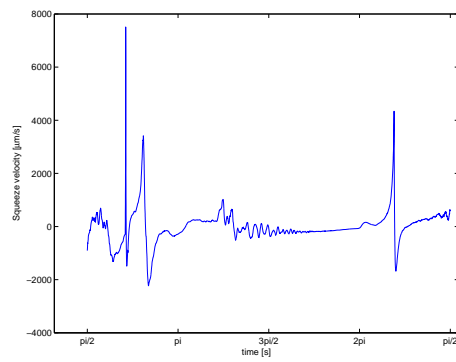
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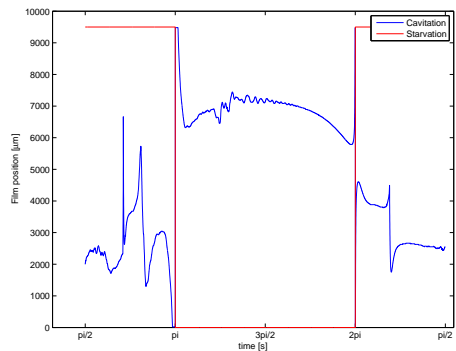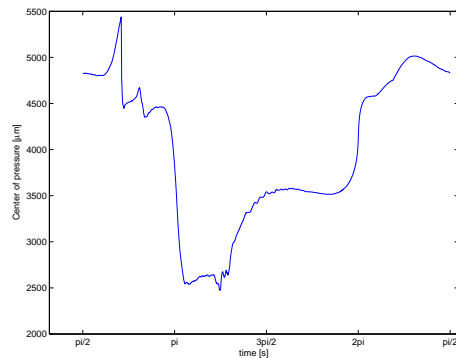Phone: +45 45256267, Fax: +45 45931577, E - mail: klit@mek.dtu.dk.
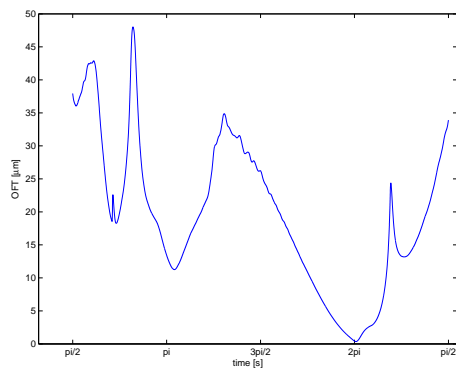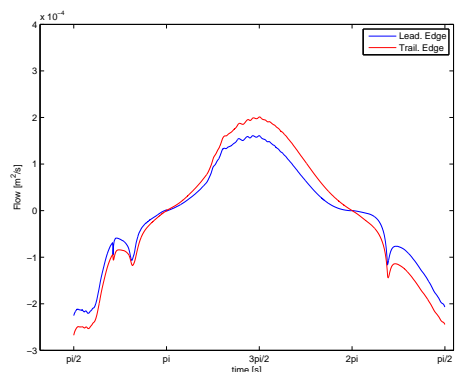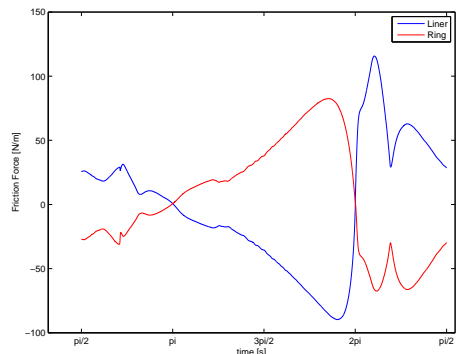
(a) Oil film thickness

(b) Squeeze velocity
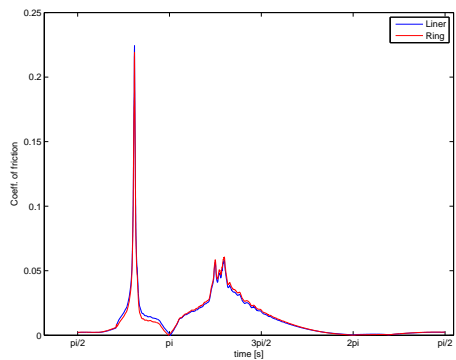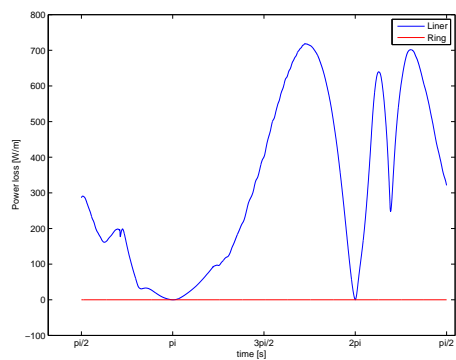
(c) Cavitation boundaries

(d) Center of pressure

(e) Flow

(f) Friction force

(g) coeff. of friction

(h) Power loss

Figure 3: Results

| | |
|---|---|
| $c'$ | Damping, Ns/m |
| $F'_{ext}$ | Load per unit width, $N/m$ |
| P | Oil pressure, $Pa$ |
| $q'$ | Volumetric flow rate per unit width, $m^2/s$ |
| $s_h$ | Shoulder height, m |
| $\tilde{u}$ | Mean surface velocity in x - direction, $m/s$ |
| W' | Load carrying capacity per unit width, $N/m$ |
| $\eta$ | Absolute viscosity, $Pas$ |
| $\rho$ | Density, $kg/m^3$ |

Table 2: Nomenclature