

Lubrication Analysis of a Con-Rod Bearing Using a Cycle Simulation of Gasoline Engines with A/F Variation

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ABSTRACT

In the case of engine bearings, pressure in a cylinder is necessary for the analysis of lubrication. In this study, a cycle simulation of gasoline engines has been developed to predict the pressure in the cylinder under the wide range of engine operation. In the cycle simulation, intake and exhaust processes are included and combustion process is calculated with flame propagation based on burning velocity. Here, the equation of ignition delay and the equation of burning velocity were determined with experimental results of a gasoline engine over wide A/F ratio. The pressure in the cylinder over the engine cycle is introduced into an elastohydrodynamic lubrication analysis of a con-rod bearing to calculate the load on the bearing in addition to the inertia force. Orbital movement, minimum film thickness, and power loss in the bearing were estimated over the wide range of engine operation. This method is useful to clarify the effects of the design and operation parameters of engines on the lubrication of bearings without experimental pressure data.

INTRODUCTION

Pressure in a cylinder over engine cycle is necessary for the analysis of tribology. In the ordinary cases, if possible, the pressure in a cylinder over engine cycle is measured experimentally ^{(1), (2)}. However, experimental apparatus and work upon an engine are necessary for measuring the pressure in a cylinder. The pressure in a cylinder changes with design and operation parameters of an engine so that there is no universality on the measured pressure.

It is well known that a cycle simulation is useful to estimate pressure in a cylinder and users can easily change design and operation parameters. Combination between the cycle simulation and a

tribology model is necessary to clarify the relation between engine dynamics and tribology.

In this study, a cycle simulation model of gasoline engines has been developed to predict the pressure in a cylinder under the wide range of engine operation. In the cycle simulation, intake and exhaust processes are included and combustion process is calculated with flame propagation based on burning velocity. Here, the equation of ignition delay and the equation of burning velocity were determined with experimental results of a gasoline engine over wide air fuel ratio, A/F. The pressure data obtained from the cycle simulation of gasoline engines are introduced into tribology model. In this case, the tribology model is an elastohydrodynamic lubrication analysis of a con-rod bearing.

Loads on the bearing were calculated with the pressure in the cylinder over an engine cycle in addition to the inertia force. Orbital movement, minimum film thickness, and power loss in the bearing were estimated over the wide range of engine operation.

NOMENCLATURE

A : area	[m ²]
A/F : air fuel ratio	
b : bearing width, namely length	[m]
c_p : specific heat at constant pressure	[J/(kgK)]
c_v : specific heat at constant volume	[J/(kgK)]
C_0 : radius of basic circle of bearing clearance	[m]
C_b : coefficient of burning velocity	[m ^{2.25} s ^{-0.5} K ^{-1.5} kg ^{-0.25}]
C_m : mean piston speed	[m/s]
C_s : flow coefficient at intake and exhaust system	
$[C]$: compliance matrix of bearing deformation	[m/Pa]
D : diameter of cylinder	[m]
E_x, E_y : x and y components of eccentricity of journal	[m]
h : specific enthalpy including heat of formation	[J/kg]
film thickness	[m]

l : length of con-rod between the centers of big and small ends	[m]
L : deviation from basic circle of bearing clearance	[m]
m : mass of gas	[kg]
n : engine speed [rps], n' : engine speed	[rpm]
p : pressure	[Pa]
Q : heat	[J]
Q_w : heat transfer to wall	[J]
r : radius of journal	[m]
R : gas constant [J/kgK], radius of bearing	[m]
S_b : turbulent burning velocity	[m/s]
t : time	[s]
T : temperature	[K]
u : specific internal energy including heat of formation	[J/kg]
U : tangential velocity of journal surface relative to bearing surface	[m/s]
V : cylinder volume	[m ³]
w : flow velocity	[m/s]
W_x, W_y : x and y components of load on bearing	[N]
α : angle between cylinder and con-rod axes	
α_w : heat transfer coefficient	[W/(m ² K)]
ε : eddy diffusion coefficient. scale $\times$ turbulence	[m ² /s]
θ : crank angle in cycle simulation	[deg.],
bearing angle in lubrication	
κ : specific-heat ratio	
μ : viscosity	[Pa s]
ρ : density	[kg/m ³]
λ : air excess ratio	
τ : ignition delay [s], shear stress in tangential direction	[Pa]
Subscripts	
0 : standard	1 : upward
2 : downward	u : unburned
w : wall	b : burned

CYCLE SIMULATION

Schematic diagram of engine is shown in Fig. 1. Engine is composed of an intake port, a cylinder, and an exhaust port. The intake port is connected with the cylinder through an intake valve and with atmosphere through a throttle valve. The exhaust port is connected with the cylinder through an exhaust valve and with atmosphere through resistance of flow between the exhaust port and atmosphere. In all the processes, gases are treated as ideal gas. The following differential equations are numerically calculated using the Runge-Kutta method.

INTAKE AND EXHAUST PROCESSES

Intake and exhaust processes relation to intake and exhaust strokes of an engine. The calculations of these processes were reported by Tazaka and Matuoka^{(3), (4)}. In this study, the model was made basically according to their reports. Mass flow rates at the intake, exhaust, and throttle valves and the mass flow rate between exhaust port and atmosphere were expressed by

$$dm = C_s A p w dt \quad (1).$$

Pressures at the intake port, the cylinder, and the exhaust port dynamically change so that pressures are checked every time and a higher pressure side

is decided as upper stream 1 and the lower pressure side is decided as down stream 2. The ρw is calculated with the following equations where the equations are selected with comparing the pressure ratio with critical ratio at every step of calculation.

$$\frac{p_2}{p_1} < \left(\frac{2}{\kappa + 1} \right)^{\frac{\kappa}{\kappa - 1}}$$

$$\rho w = p_1 \sqrt{\frac{2\kappa}{(\kappa - 1)RT_1} \left\{ \left(\frac{p_2}{p_1} \right)^{\frac{2}{\kappa}} - \left(\frac{p_2}{p_1} \right)^{\frac{\kappa + 1}{\kappa}} \right\}} \quad (2)$$

$$\frac{p_2}{p_1} \geq \left(\frac{2}{\kappa + 1} \right)^{\frac{\kappa}{\kappa - 1}}$$

$$\rho w = p_1 \sqrt{\frac{\kappa}{RT_1} \left(\frac{2}{\kappa + 1} \right)^{\frac{\kappa + 1}{\kappa - 1}}} \quad (3)$$

Energy conservation equation for the intake port and the exhaust port is

$$c_p T_1 dm_1 - c_p T_2 dm_2 = dQ_w + d(c_v m T) \quad (4).$$

Energy conservation equation for the cylinder is

$$c_p T_1 dm_1 - c_p T_2 dm_2 = dQ_w + d(c_v m T) + p dV \quad (5).$$

The areas, A , of the intake and exhaust valves are the cylinder surface areas having the diameter of valve, D_v , and the length of valve lift, l_v , at each time step, namely $A = \pi D_v l_v$.

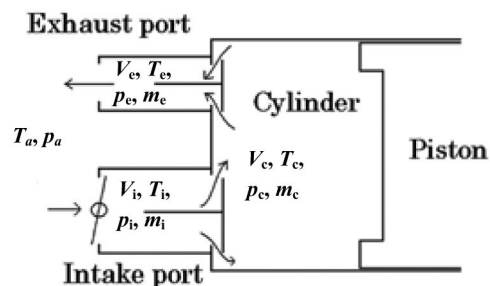


Fig. 1 Engine model.

COMPRESSION AND EXPANSION PROCESSES

Compression, combustion, and expansion processes relation to compression and expansion strokes of an engine. The calculations of these processes were reported by Saito and Oda⁽⁵⁾. In this study, the model was made basically according to their report. Gas in the cylinder under the compression process is mixture of air, fuel, and residual gas and gas in the cylinder under the expansion process is combustion gas. The combustion process is two zones, namely an unburned gas zone and a burned gas zone. The

thermodynamic equations for unburned gas and burned gas are described as follows where $m_b=0$ before combustion and $m_u=0$ after combustion. The equations of state of ideal gas are

$$\begin{aligned} pV_u &= m_u R_u T_u \\ pV_b &= m_b R_b T_b \end{aligned} \quad (6).$$

Volume and mass in the cylinder are

$$V = V_u + V_b \quad (7)$$

$$m = m_u + m_b \quad (8).$$

Till over 2% in mass of cylinder gas is burned after ignition, the gas is burned under constant pressure and then compressed adiabatically to satisfy Eq. (7). After the above process, combustion process is calculated with following equations based on the first law of thermodynamics.

$$\begin{aligned} d(m_u u_u) + p dV_u + dQ_{wu} &= -h_u dm_b \\ d(m_b u_b) + p dV_b + dQ_{wb} &= h_u dm_b \end{aligned} \quad (9)$$

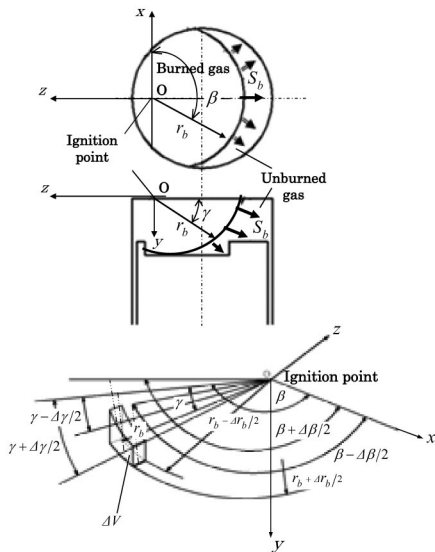


Fig. 2 Flame propagation under combustion process, two zone model.

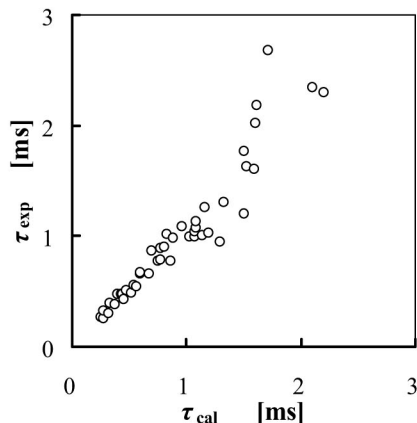


Fig. 3 Ignition delay, Experiment vs. Equation (10).

Combustion gas is composed of 11 components, namely N_2 , O_2 , CO , CO_2 , H_2O , H , H_2 , OH , O , N , and NO , under chemical equilibrium. Thermodynamic properties of each component are based on the Thermo-chemical Tables of JANAF ⁽⁶⁾. The thermodynamic properties of mixture are calculated from the properties of above components based on Dalton's law.

Ignition delay is the duration between spark timing and start of combustion. In the present study, the ignition delay, τ , is expressed by the following equation based on the experimental results of a single cylinder engine with a disc shape combustion chamber. In this report, λ is 1 at $A/F=14.4$.

$$\begin{aligned} \tau &= 1.069 \times 10^{-16} (A/F + 1)^{211.1} (A/F)^{-197.4} \\ &\quad \times n'^{-0.75} p^{-1.05} \exp(-882/T_u) \quad [s] \\ &= 0.0367 (\lambda + 0.06944)^{211.1} \lambda^{-197.4} \\ &\quad \times n'^{-0.75} p^{-1.05} \exp(-882/T_u) \quad [s] \end{aligned} \quad (10)$$

where engine speed is expressed by n' [rpm] and n [rps]. The relation between experiment and Eq. (10) is shown in Fig. 3 where the temperature and pressure used in Eq. (10) are mean values of those at spark timing and at start of combustion derived from the measured pressure and air flow.

Livengood-Wu integration is used in the cycle simulation considering the pressure and temperature under dynamic condition.

$$1 = \int_{t_s}^{t_s + \tau_0} \frac{dt}{\tau} \quad (11)$$

where τ_0 is the ignition delay in the cycle simulation and t_s is spark timing, and τ is ignition delay based on Eq. (10).

As shown in Fig. 2, combustion process is assumed that flame propagates spherically in radial direction from the spark point with turbulent burning velocity according to unburned gas condition. Mass reduction of unburned gas by means of combustion, dm_b , is calculated from the volume of spherical flame propagation at each time step, namely at each crank-angle. In the calculation, burned gas volume and propagation volume are divided into small elements and each element is integrated till flame front or burned gas region attach to the wall. Therefore the shape of a combustion chamber and the location of a spark could be considered. In this study, turbulent burning velocity reported by Kim ⁽⁷⁾ is adopted.

$$\begin{aligned} S_b &= S_{b0} \left(\frac{T_u}{T_0} \right)^{1.5} \left(\frac{p_0}{p} \right)^{0.25} \sqrt{\left(\frac{\rho_u}{\rho_0} \right) \left(\frac{\mu_0}{\mu_u} \right) \left(\frac{\varepsilon}{\varepsilon_0} \right)} \\ &= C_b T_u^{1.5} p^{-0.25} n \sqrt{\rho_u \left(\frac{\mu_0}{\mu_u} \right)} \end{aligned} \quad (12)$$



where T_u , ρ_u , and μ_u are temperature, density, and viscosity of unburned gas respectively. Subscript 0 expresses the standard values. ε is proportional to engine speed squared, n^2 . Viscosity of air at 653[K] is used for μ_0 and other standard values are included in the coefficient, C_b . In the present study, the C_b is expressed by the following equation based on the experimental results of a single cylinder engine with a disc shape combustion chamber. The C_b is determined in comparison between the mass fraction burned obtained from analysis of the experimental pressure and that obtained from the cycle simulation. The comparisons are shown in Fig. A1 in Appendix A1.

$$C_b = 12.93(A/F)^{73.04} \{(A/F)+1\}^{-78.11} [m^{2.5} MPa^{0.25} K^{-1.5} kg^{-0.5} s^{-1} rpm^{-1}] = 0.03295(\lambda)^{73.04} \{\lambda + 0.06944\}^{-78.11} [m^{2.25} s^{-0.5} K^{-1.5} kg^{-0.25}] \quad (13)$$

The C_b s determined with the experiment are compared with those calculated with Eq. (13) in Fig. 4.

EGR or residual gas in the cylinder could be considered using the partial pressure and density of fresh mixture in Eqs. (10) and (12). Effect of stroke of engines could be considered as the mean piston speed, therefore n is modified with $n=n_0(STROKE[mm]/85)$. n_0 is the engine speed of 85 [mm] stroke engine used in the experiments.

Comparisons of the calculation with the cycle simulation and the experimental results are shown in Figs. A2 through A5 in Appendix A2.

HEAT TRANSFER

Heat loss to wall dQ_w is expressed with the area of wall, A_w , by

$$dQ_w = \alpha_w A_w (T - T_w) dt \quad (14).$$

At an intake and an exhaust ports, heat transfer coefficient of heat loss to wall is according to Tazaka and Matuoka⁽⁴⁾. In the cylinder, Woschni's equation⁽⁸⁾ is used for heat transfer coefficient of heat loss to wall.

$$\alpha_w = 1.299 \times 10^{-2} D^{-0.2} p^{0.8} T^{-0.53} \times \left\{ C_1 C_m + C_2 \frac{V_s T_0}{p_0 V_0} (p - p_m) \right\}^{0.8} \quad (15)$$

where C_m is mean piston speed [m/s], p_m is pressure under motoring, p_0 and T_0 are pressure and temperature at standard volume, V_0 , and V_s is stroke volume respectively. $C_1=2.28$ and $C_2=3.24 \times 10^{-3}$ [m/(sK)] under compression and expansion strokes and $C_1=6.18$ and $C_2=0$ [m/(sK)] under intake and exhaust strokes. In the case of burned gas, wall area, A_w , is the contact area of spherical burned zone and

chamber wall. In the case of unburned gas, wall area, A_w , is difference between total chamber wall and area of burned gas. Therefore heat loss to wall can be calculated for each burned and unburned zones respectively.

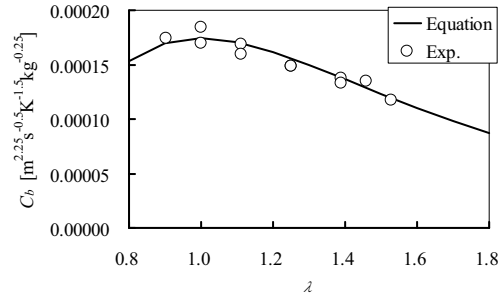


Fig. 4 C_b , Experiment vs. Equation (13).

LUBRICATION MODEL OF CON-ROD BEARING

Lubrication model of journal bearing is according to the elastohydrodynamic, EHD, lubrication model of con-rod bearing reported by Ozasa et al.^{(1),(9)}. Model of journal bearing is shown in Fig. 5. The x , y , z and θ are the abscissa, the ordinate and the transversal coordinate on the con-rod, and the angular coordinate on big-end bearing respectively. The Reynolds equation,

$$\frac{1}{R} \frac{\partial}{\partial \theta} \left\{ \frac{h^3}{12\mu R} \frac{\partial p}{\partial \theta} \right\} + \frac{\partial}{\partial z} \left\{ \frac{h^3}{12\mu} \frac{\partial p}{\partial z} \right\} = \frac{1}{R} \frac{\partial}{\partial \theta} \left\{ \frac{hU}{2} \right\} + \frac{\partial h}{\partial t} \quad (16),$$

is solved in the $p>0$ region. In the case of a big-end bearing, the hydrodynamic pressure and shear stress in the $p>0$ region equilibrate with the load assuming that the moment of force on the small-end bearings is zero. The load in the x and y directions, W_x and W_y , are expressed by

$$W_x = \int_{-\frac{b}{2}}^{\frac{b}{2}} \int_0^{2\pi} \left\{ p \cos \theta - \tau \left(\sin \theta - \frac{R}{l} \right) \right\} R d\theta dz \quad (17).$$

$$W_y = \int_{-\frac{b}{2}}^{\frac{b}{2}} \int_0^{2\pi} \left\{ p \sin \theta + \tau \left(\cos \theta - \frac{R}{l} \tan \alpha \right) \right\} R d\theta dz$$

Friction is estimated with integrating shear stress over the film region. The shear stress is expressed by

$$\tau = -\frac{h}{2R} \frac{\partial p}{\partial \theta} + \mu \frac{U}{h} \quad (18).$$

Tangential velocity of journal surface relative to bearing surface, U , must be calculate considering engine speed and angular movement of con-rod. The load, W_x and W_y are calculated from inertia force