A Study on Bubbly Lubrication of High-Speed Journal Bearing Considering Live Surface Tension

S. M. CHUN [1]

[1] *Department of Automotive Engineering, Donghae University*
119 Jiheungdong, Donghae,Gangwondo, 240-713, KOREA

The influence of aerated oil on a high-speed journal bearing is examined by using the classical thermohydrodynamic lubrication theory coupled with analytical models for viscosity and density of air-oil mixture in fluid-film bearing including the live surface tension of aerated oil. Convection to the walls and mixing with supply oil and re-circulating oil are considered. The considered parameters for the study of bubbly lubrication are oil aeration level, air bubble size and shaft speed. The results show that, if the live surface tension is considered, the effect of air bubbles on the bearing load capacity is reduced due to temperature engagement comparing with that under the condition of a constant surface tension.

Keywords : Aerated oil, Flow mixing, Convective conditions on the walls, High-speed journal bearing, Turbulent Reynolds and energy equations

1. INTRODUCTION

In this paper, the Nikolajsen's viscosity and density models [1] are used together with classical Reynolds equation and energy equation to investigate and predict numerically the effects of oil aeration on the performance of a high-speed plane journal bearing with an axial groove.

Also, the convective conditions on the walls, the conditions of shaft alignment and misalignment, the contraction ratio at cavitation region, and the mixing between re-circulating oil and inlet oil are included.

2. GOVERNING EQUATION

The Reynolds' equation for a steadily loaded journal bearing for finite width may be written as

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{\mu} G_x \frac{\partial \overline{p}_g}{\partial x}\right) + \frac{\partial}{\partial z}\left(\frac{\rho h^3}{\mu} G_z \frac{\partial \overline{p}_g}{\partial z}\right) = \frac{U}{2}\frac{\partial(\rho h)}{\partial x} \quad (1)$$

The appropriate values of G_x and G_z are given by the references [2,3] in the range $1{,}000 \le \mathrm{Re} \le 30{,}000$.

The steady state two dimensional energy equation with heat transfer boundary conditions at the bearing walls may be derived under turbulence conditions as

$$\rho\left\{\left(\frac{Uh}{2} - \frac{h^3}{\mu} G_x \frac{\partial \overline{p}_g}{\partial x}\right)\frac{\partial\left(c_v \overline{T}\right)}{\partial x} - \frac{h^3}{\mu} G_z \frac{\partial \overline{p}_g}{\partial z}\frac{\partial\left(c_v \overline{T}\right)}{\partial z}\right\} =$$

$$\tau_c U + \frac{h^3}{\mu}\left\{G_x\left(\frac{\partial \overline{p}_g}{\partial x}\right)^2 + G_z\left(\frac{\partial \overline{p}_g}{\partial z}\right)^2\right\} - \left(q_{sT} + q_{bT}\right) \quad (2)$$

where, $q_{sT} = H_{sT}\left(\overline{T} - T_s\right)$ and $q_{bT} = H_{bT}\left(\overline{T} - T_b\right)$.

The values of H_s and H_b are chosen as shown in Table 1.

In the range $1{,}000 \le \mathrm{Re} \le 30{,}000$, appropriate values of $\overline{\tau}_c\ (= \tau_c / \frac{\mu U}{h})$ are given by the references [2,3].

The density (kg/m^3) and kinematic viscosity (cst) of pure oil can be expressed by equation (3) and equation (4) with constants, aa, bb and cc which vary depending on the kinds of the oil:

$$\rho_{oil} = 0.0361(aa - 0.000354 T_f) \bullet 27680. \quad (3)$$

$$\nu_{oil} = \frac{\mu_{oil}}{\rho_{oil}} = 10^{10^{(bb - cc\log_{10}(T_r))}} - 0.6. \quad (4)$$

where T_f and T_r represent the Fahrenheit temperature and Rankine temperature, respectively. And C_p is the specific heat (J/kg. °C) of oil that may be correlated with Celsius temperature T_c as equation (5).

$$C_p = 1796 + \frac{691}{160} T \quad (5)$$

The density (kg/m^3) and dynamic viscosity (Pa.s) of aerated oil is derived in [1] as shown below. Non-dimensional density can be described as

$$\overline{\rho} = \frac{\rho}{\rho_{oil}} = \frac{(1+\delta)\left(\overline{\overline{p}}_{oil} + 2\frac{\overline{\sigma}}{\overline{r}}\right)}{\delta + \overline{\overline{p}}_{oil} + 2\frac{\overline{\sigma}}{\overline{r}}}. \quad (6)$$

where, $\delta = \frac{m_{air}}{m_{oil}} = \frac{(\overline{\overline{p}}_{oil})_m + 2\overline{\sigma}/\overline{r}_m}{\frac{3}{4\pi}\left(\frac{r_m}{\overline{d}_m}\right) - 1}$, $\overline{\sigma} = \sigma / \left(\rho_{oil} \overline{R} T c\right)$.

$\overline{\overline{p}} = \overline{p}/\left(\rho_{oil}\overline{R}T\right)$. $\overline{r}$ is the real root between 0 and r_m of the polynomial equation $\overline{\overline{p}}_{oil}\overline{r}^3 + 2\overline{\sigma}\overline{r}^2 - \left[\left(\overline{\overline{p}}\right)_m + 2\sigma/r_m\right]r_m^3 = 0$.

And c is the bearing clearance. T is the absolute oil temperature. ρ_{oil} is the density of pure oil. σ is the surface tension of a bubble (N/m), and $\overline{R}$ is the gas constant (J/kg K).

The surface tension (N/m) of a bubble in gas-oil mixture can be expressed as [4]:

$$\sigma = \sigma_p[1/(1 + 0.02549\, P^{1.0157})] * 10^{-3} \quad (7)$$

Here $\sigma_p = AT(38.085 - 0.259 API)$. AT=1.11591-0.00305 T_c, API=141.5/SpGr − 131.5, SpGr=SpGr60-0.00035(T_f -60.). SpGr60 is a specific gravity of oil at 60 degree Fahrenheit.

Non-dimensional viscosity [1] can be expressed as:

$$\overline{\mu} = \frac{\mu}{\mu_{oil}} = \overline{\mu}_1 + \overline{\mu}_2 \tag{8}$$

where $\overline{\mu}_1 = \frac{\mu_1}{\mu_o} = \frac{\overline{\rho}}{1+\delta}$, $\overline{\mu}_2 = \frac{\mu_2}{\mu_o} = \Gamma \, \overline{h}_m \, \overline{r} / \sqrt{\overline{h}}$,

$\Gamma = \frac{\pi \sigma}{\sqrt{2}\mu_o U r_m} \left[\frac{r_m}{d_m}\right]$, $\overline{h} = \frac{h}{c}$, $\overline{d} = \frac{d}{c}$. And U is the bearing surface speed. d is the distance between bubbles. $\overline{\mu}_1$ represents the viscosity change due to the near-zero air viscosity within the bubbles. $\overline{\mu}_2$ represents the viscosity change due to bubble surface tension. $\overline{r}_m / \overline{d}_m$ represents oil aeration level illustrated in Fig. 1.

Air volume ratio can be expressed as equation (9) in geometrical aspects of oil aeration level.

$$V = \frac{4\pi}{3A^3} \tag{9}$$

where $A = 1/[2(\overline{r}/\overline{d})_o] + 1$.

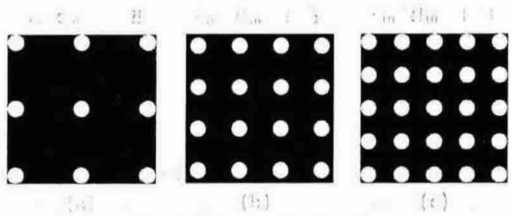

Fig 1. Oil aeration levels

Film thickness, h, can be defined by the expression using bearing coordinates [5]:

$$h = c(1 + \varepsilon \cos(\theta - \varphi)) \tag{10}$$

3. CAVITATION MODEL

The cavitation region in this study is assumed that several oil strips exist and the shaft is covered with oil as usual. It is assumed that the air bubbles mixed with oil are distributed orderly even though the size of air bubble can vary depending on pressure, and that the air bubbles still exist uniformly even inside oil strips. The temperature distribution at air in cavitation region is assumed as the temperature of aerated oil. The gage pressure in cavitation region is zero as mentioned as the boundary condition.

Therefore, for the turbulence treatment of fluid inside bearing, the mixed oil is treated as a sort of virtual oil whose physical properties appear as the mixed viscosity, the mixed density and the specific heat of oil itself. However, the heat transfer to the bearing walls is handled separately from both oil and gas.

4. OPERATING CONDITIONS

The journal bearing's operating conditions are illustrated in Table 1.

5. CONCLUSION

1. Considering the aerated oil film temperature field coupling energy equation with Reynolds equation, the bearing load capacity can be increased under the circumstance of increasing aeration level and reducing air bubble size. On the contrary, under the circumstance of reducing aeration level and increasing air bubble size, the bearing load capacity may be reduced only in small amount due to the effect of temperature involved.
2. As the results of the numerical analysis on thermohydrodynamic bubbly lubrication of a high-speed plane journal bearing, even if the aeration level is low like 1/8, the bearing load capacity may notably increase under the small air bubble radius like 0.002mm.
3. The difference in load capacity between aerated oil application and pure oil application with increasing shaft-speed is appeared notably. However, the difference in power loss is nearly not appeared.

Table 1. Journal Bearing Operating Conditions

Bearing Diameter	D = 73.6 mm
L/D Ratio	0.5
c/R Ratio	0.0039837
Eccentricity Ratio	ε = 0.65
Rotational Speed	N = 40,000 rpm
Lubricant Viscosity at 40 °C	μ_o = 0.0206Pa.s
Lubricant Density at 40 °C	ρ_o = 869.53Kg/m^3
Lubricant Specific Heat at 40 °C	C_o = 1968.75J/kg °C
Convective Heat Transfer Coefficient of Lubricant to Bush	H_{ob} =7700 W/m^2 °C
Convective Heat Transfer Coefficient of Gas(Air) to Bush	H_{ab} =2400 W/m^2 °C
Convective Heat Transfer Coefficient of Lubricant to Shaft	H_{os} =7700W/m^2 °C
Bush and Shaft Temperature	$T_{b,s}$ = 45 °C
Inlet Lubricant Temperature	T_{in} = 40 °C
Inlet Lubricant Pressure(gage)	P_{in} = 0.7 x 10^5 Pa
Axial Groove Width	17.1 ° (2 grids size)

6. REFERENCES

[1] Nikolajsen, J. L., "Viscosity and Density Models for Aerated Oil in Fluid-Film Bearings", *STLE, Tribology Transactions*, Vol. 42, no. 1, 1999, pp 186-191.
[2] Taylor, C. M., "Turbulent Lubrication Theory Applied to Fluid Film Bearing Design," *Proc. Inst. Mech. Engrs.*, Vol. 184, Part 3L, 1969-1970, pp. 40-47.
[3] Constantinescu, V. N., "Basic Relationships in Turbulent Lubrication and Their Extension to Include Thermal Effects," *Trans. of the ASME, J. of Lubrication Technology*, Vol. 95, 1973, pp. 147-154.
[4] Audul-Majeed, G. H. and Al-Soof, N. B. A., "Estimation of gas-oil surface tension", Journal of Petroleum Science and Engineering, Vol. 27, 2000, pp. 197-200.
[5] Chun, S. M. and Lalas, D. P., "Parametric Study of Inlet Oil Temperature and Pressure for a Half-Circumferential Grooved Journal Bearing," *STLE, Tribology Transaction*, Vol. 35, no. 2, pp. 213-224, 1992.