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The Static and Dynamic Performance of a MEMS/MST Based Gas-Lubricated Journal Bearing with the Slip Flow Effect

H. D. KWAK, Y. B. LEE, C. H. KIM [1], N. S. LEE and D. H. CHOI [2]

[1]*Tribology Research Center, Korea Institute of Science and Technology*
P.O. Box 131, Cheongryang, Seoul 130-650, KOREA
[2]*Department of Mechanical Design and Production Engineering, Hanyang University*
17 Haengdang-Dong, Seongdong-Ku, Seoul 133-791, KOREA

The influence of the slip flow on the MEMS/MST based gas-lubricated journal bearing is investigated. Based on the modified Reynolds equation, the numerical analysis of the finite difference method was developed by applying the first order slip flow approximation. The numerical prediction of bearing performance provides the significant results concerning the slip flow effect in micro scale gas-lubricated journal bearing. The result indicates that the load-carrying capacity as well as the rotordynamic coefficients were significantly reduced due to the slip flow. Through this work, it is concluded that the slip flow effect could not be ignored in the micro gas-lubricated journal bearing.

Keywords : Gas-Lubricated Bearing, Knudsen Number, Slip-Flow, MEMS(Microelectromechanical Systems), MST(Micro Structure Technology)

1. INTRODUCTION

In recent years, MEMS or MST based gas-lubricated bearings for micro-rotating machinery such as micro gas turbine (see Fig. 1) has made their emergence. In general, the rarefaction effect on the gas-lubricated bearings has been ignored. However concerning the gas-lubricated bearings for micro rotating machinery, the rarefaction effect arises because the local Knudsen number at the minimum film thickness may be greater than 0.01 as well as the surface is rough due to their manufacturing method. In this situation, the gas starts to slip on the bearing surface, i.e. slip flow.[1] In addition, some micro rotating machinery such as micro gas turbine operates at very high temperature condition. In such condition, the slip flow effect could be especially significant because the molecular mean free path increases with temperature. Thus in this paper, the slip flow effect is considered to estimate the performance of the gas-lubricated journal bearings. The modified compressible Reynolds equation with rarefaction effect is utilized as a governing equation and solved numerically.

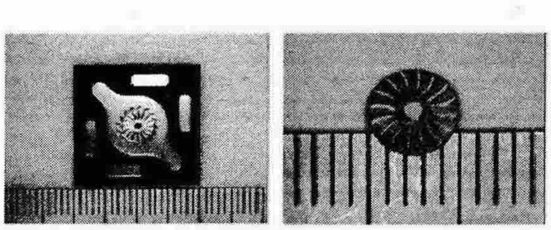

Fig. 1 Micro gas turbine manufactured by KIST

2. ANALYSIS

In this paper, the lubricant is modeled as an isothermal perfect gas. The pressure distribution within the clearance of the bearing is governed as the following modified Reynolds equation.

$$\nabla\cdot\left(-\frac{1}{12\mu}\varphi^{P} p h^{3}\nabla p+\frac{\bar{U}}{2}ph\right)+\frac{\partial}{\partial t}(ph)=0 \qquad (1)$$

$\varphi^{P}(p,h)$ denotes the molecular rarefaction coefficient in the following form.

$$\varphi^{P}(p,h)=\sum_{k=0}^{3}\left\{c_k(p)h^{-k}\right\} \qquad (2)$$

The rarefaction coefficient, equation (2) is valid for high Knudsen numbers using power series expressions in terms of inverse Knudsen number.[2] A first order slip flow boundary condition[3] is one of the possible candidate approximated equations for thin gas film lubrication and is widely used in the case that the film thickness is greater than 0.1 μm . For the first order slip flow approximation, the coefficients c_k can be chosen as $c_0 = 1$, $c_1 = 6a\lambda$ and $c_2 = c_3 = 0$. The molecular mean free path in the coefficient c_1 is calculated according to the reference[1] with accommodation coefficient(a) set to 0.5

For small amplitude journal motion measured from the static equilibrium position, the pressure and the film thickness can be expressed as a first order Taylor series:

$$p = p_0 + p_x\Delta x + p_y\Delta y + p_{\dot{x}}\Delta\dot{x} + p_{\dot{y}}\Delta\dot{y} \qquad (3)$$

$$h = h_0 + h_x\Delta x + h_y\Delta y + h_{\dot{x}}\Delta\dot{x} + h_{\dot{y}}\Delta\dot{y} \qquad (4)$$

As the rarefaction coefficient is a function of the pressure and the film thickness, it may be expressed as follows:

$$\varphi^{P} = \varphi^{P}_{0} + \left(\frac{\partial\varphi^{P}}{\partial p}\right)_0\cdot\Delta p + \left(\frac{\partial\varphi^{P}}{\partial h}\right)_0\cdot\Delta h \qquad (5)$$

Substituting Equations (3)-(5) into Equation (1), neglecting the higher-order terms, it yields the zeroth-order and first-order equations. In the numerical procedure, the finite difference scheme was applied. The accuracy of solution was examined by the doubling a mesh density. Once the steady-state and perturbed pressures are known, the static and dynamic coefficients are readily calculated from proper integrations. Herein, these equations are omitted.

The analysis model was focused on the micro gas-lubricated journal bearing in Orr's[4] and Piekos'[5] work, which uses in micro gas turbine. Its properties are as follows: L/D ratio is 0.075, temperature of gas is 300 K and 1600 K, nominal clearance is 12 μm .

3. RESULTS AND DISSCUSSION

Fig. 2 gives the comparison of pressure distribution without slip and with slip effect at the axial mid-plane of bearing. The figure shows that the solution of modified Reynolds equation without slip is virtually identical to the D.J. Orr's[4] solution, which implies that the numerical analysis in this work is accurate. However, Fig. 2 indicates the significant differences of pressure distribution between slip and no-slip condition. The predicted results herein show that the pressure near the minimum film thickness under the slip condition is less than that under no-slip condition. Fig. 3 clarifies influence of slip flow. It is evident that load-carrying capacity decreases under the slip condition. In the figure, the absolute differences of load-carrying capacity between slip and no-slip condition increase with the bearing number. In addition, an increment of temperature also increases the slip flow effect on load-carrying capacity because the molecular mean free path is proportional to temperature.

Fig. 4 and 5 demonstrate the direct stiffness and direct damping coefficients in a range of the bearing number, respectively. In Fig. 4, the direct stiffness coefficients of bearing between no-slip and slip condition are almost identical in a low bearing number region. As the bearing number increases, where high-speed region, the difference of stiffness between no-slip and slip condition becomes larger. Especially, slip condition significantly reduces the stiffness of bearing at high temperature because the molecular mean free path becomes greater in proportion to temperature. The direct damping coefficients exhibit some different manner compared to the direct stiffness. Before the direct damping coefficients reach peak, it is true that the slip condition reduces the value of damping coefficients compared to the no-slip condition. However, the peak values of damping coefficients in slip condition are greater than that in no-slip condition. In addition, slip condition removes the location of peak value to higher bearing number region.

4. CONCLUSION

The slip flow effect was considered to estimate the static and dynamic performance of micro gas-lubricated journal bearings. Based on the modified compressible Reynolds equation with 1st order slip flow approximation, the static and dynamic characteristics of micro gas-lubricated journal bearings were demonstrated and showed a difference scheme between slip and no-slip condition. In conclusion, these results strongly support that the slip flow has substantial influence on the bearing characteristics and should not be ignored in estimation of performance of micro gas-lubricated bearings.

5. ACKNOWDGEMENT

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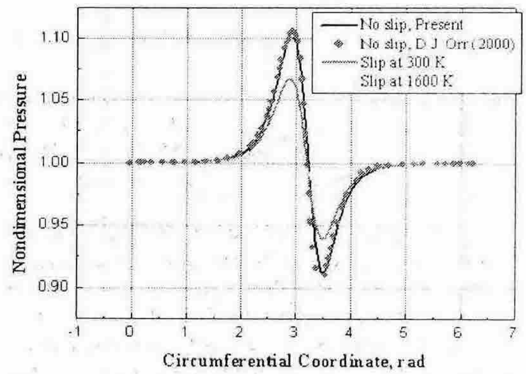

Fig. 2 Comparison of pressure distribution at the axial mid-plane, $\Lambda = 1.0$, $\varepsilon = 0.85$

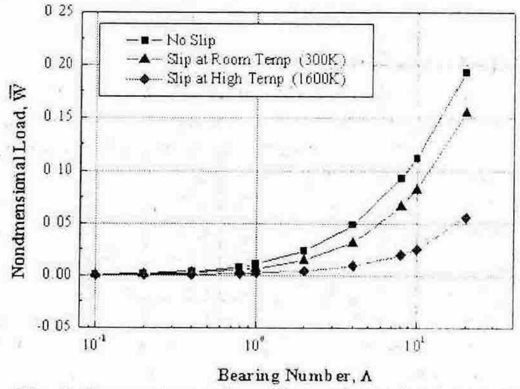

Fig. 3 Comparison of nondimensional load-carrying capacity, $\varepsilon = 0.9$

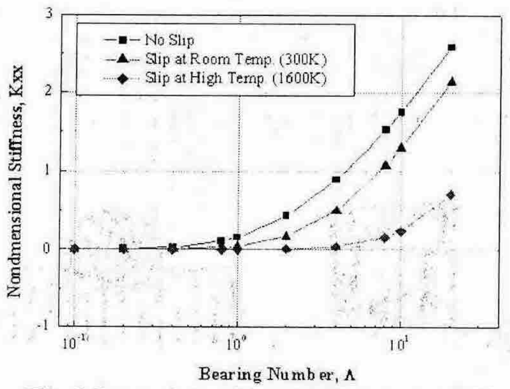

Fig. 4 Comparison of direct stiffness, $\varepsilon = 0.9$

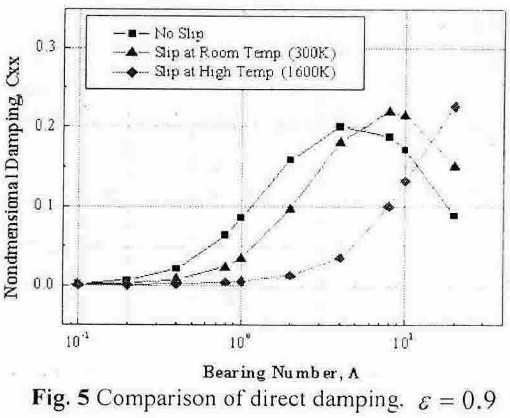

Fig. 5 Comparison of direct damping, $\varepsilon = 0.9$