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Stability analysis of water-lubricated journal bearings for fuel cell vehicle air compressor

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ABSTRACT

This paper presents a stability investigation of the step recessed, hydro-static and -dynamic hybrid water-lubricated journal bearings for fuel cell vehicle compressor. The dynamic coefficients and stability threshold speed of the bearings are numerically calculated by solving the turbulent Reynolds equation by using FEM. Two configuration bearings are comparatively studied and the different dynamic characteristics are revealed. The validity of the theoretical analysis was verified by the experiments on the built fuel cell compressor. Finally, the influences of the turbulence and geometrical parameters on the stability threshold speed are analyzed.

1. Introduction

Recently, new energy vehicles using hydrogen fuel cells have achieved an impressive development due to their advantages of quasi-zero emissions and high efficiency. As the core component in the vehicle, the fuel cell system needs an air compressor to supply pressured oxygen to the stack. Centrifugal type compressor directly driven by high speed motor is regarded to be an ideal solution for its low noise, high efficiency and compact structure.

The successful development of the compressor relies on several key technologies, such as bearing, motor, drive and impeller [1]. This paper discusses the bearing aspect only. Traditionally, the high speed rotor of automotive turbocharger is supported by oil lubricated bearing. The most notable limitation of oil lubricated bearings is the contaminants which can lead to pollution of the fuel cell proton exchange membrane. Therefore, air foil bearing solution has been demonstrated by several makers [1–4]. Meanwhile, the authors have shown the possibility and merits of using water lubricated hydro-static and -dynamic hybrid bearing in the fuel cell compressor [5].

In general, the centrifugal air compressor is required operating in high speeds in order to improve efficiency while decreasing in size. Therefore the rotor-bearing system faces the stability problem of water film, especially under non-laminar lubrication condition. The bearing stability performance is affected by many aspects, such as bearing configuration, supply pressure, Reynolds number, and geometrical parameters.

Ghosh and Statish found that the multi-lobe hybrid bearings with offset factors possess better dynamic behavior than circular hybrid bearings [6,7]. Franchek and Childs [8] experimentally studied square, circular, triangular recessed bearings and square recessed bearing with angled orifices. They reported that the square recess bearing with angled orifices has the most favorable overall performance. Brito and Miranda [9,10] experimentally studied the journal bearing with axial grooves and pointed out that the increase in supply pressure yields a slight increase in the attitude angle and minimum film thickness. A theoretical-experimental investigation taken by Gapone and Russo [11] showed that the ratio of the specific load and the supply pressure affects the bearing static characteristics and oil film instability threshold.

However, the above investigations were treated as the laminar lubrication. High rotating speed and the low viscosity of water make the lubrication film flow turbulently. The effect of turbulence should also be considered in the stability analysis of water lubricated bearing.

Capone [12] and Hashimoto [13,14] analyzed the dynamic characteristics of a water lubricated plain journal bearing with considering turbulent effect. They found that the bearing stability is overestimated if the influence of flow regime is neglected. However, their results are only suitable for hydrodynamic bearing with large clearance and eccentricity, i.e. small Sommerfeld number condition.

With fuel cell compressor application, the water lubricated bearing generally operates under large Sommerfeld number

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Nomenclature

b_0	circumferential width, mm
b_1	shallow recess circumferential width, mm
$\overline{b}_1$	dimensionless shallow recess circumferential width ($=b_1/b_0$)
b_2	deep recess circumferential width, mm
$\overline{b}_2$	dimensionless deep recess circumferential width ($=b_2/b_0$)
c_{ij}	damping coefficients (i, j=x, y), N s/mm
$\overline{c}_{ij}$	dimensionless damping coefficients (i, j=x, y)
D	bearing diameter, mm
d_0	orifice diameter, mm
e_x	bearing eccentricity in horizontal direction, mm
e_y	bearing eccentricity in vertical direction, mm
G_θ	circumferential turbulence coefficient
G_Z	axial turbulence coefficient
h	film thickness, mm
h_0	bearing clearance, mm
h_p	shallow recess depth, mm
$\overline{h}_p$	dimensionless shallow recess depth($=h_p/h_0$)
h_d	deep recess depth, mm
h_g	circumferential groove depth, mm
k_{ij}	stiffness coefficients (i, j=x, y), N/mm
$\overline{k}_{ij}$	dimensionless stiffness coefficients (i, j=x, y)
k_{eq}	equivalent stiffness coefficients
$\overline{k}_0$	mean direct stiffness coefficients
L	bearing length, mm
$\overline{L}$	slenderness ratio($=L/D$)
l_1	bearing axial seal width, mm
l_2	recess axial width, mm
$\overline{l}_2$	dimensionless recess axial width($=L_2/L$)
l_g	groove axial width, mm
m_R	bearing load, kg
N	recess number
P	film pressure, Pa
p_s	supply pressure, bar
p_s^*	critical supply pressure, bar
R	bearing radius, mm
Re	Reynolds number
S	Sommerfeld number($=\mu\Omega(RL/W)(R/\ h_0)^2(L/D)^2$)
W	load capacity, N
x	horizontal coordinate
y	vertical coordinate
z	axial coordinate
μ	water dynamic viscosity, Pa s
Ω	bearing rotating speed, rad/s
$\overline{\omega}_s$	dimensionless bearing stability threshold speed
ω_s	actual bearing stability threshold speed, krpm
$\overline{\omega}_s^*$	simplified dimensionless bearing stability threshold speed
θ	circumferential coordinate
γ_s	whirl-frequency ratio

condition. Furthermore, the recessed bearings with water supply are also indispensable for improving stability and cooling effect. For proper application of water lubricated bearing in turbulent condition, this paper analyzes the influences of bearing configurations, water supply pressure and geometrical parameters on the stability characteristics. A fuel cell compressor with water lubricated bearings was built and tested to show the validity of the theoretical study.

2. Governing equations

The modified Reynolds equation with considering turbulent effect is

$$\frac{\partial}{R\partial\theta}\left(\frac{h^3}{G_\theta\mu}\frac{\partial P}{R\partial\theta}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{G_z\mu}\frac{\partial P}{\partial z}\right)=\frac{U}{2}\frac{\partial h}{R\partial\theta}+\frac{\partial h}{\partial t} \tag{1}$$

$$G_\theta=12+0.0136Re^{0.9} \tag{2}$$

$$G_z=12+0.0043Re^{0.98} \tag{3}$$

$$h=h_0+\Delta h_p+e_x\cos\theta+e_y\sin\theta \tag{4}$$

where G_θ and G_z are turbulence coefficients given by Ng and Pan [15]. For laminar flow conditions, G_θ=G_z=12. h is the water film thickness, for the film in the recess, $\Delta=1$; $\Delta=0$ for the else.

The water film force components along the horizontal and vertical directions are calculated by

$$\begin{pmatrix}F_x\\F_y\end{pmatrix}=\int_0^L\int_0^{2\pi}P\begin{pmatrix}\cos\theta\\\sin\theta\end{pmatrix}d\theta dz \tag{5}$$

Linearizing the force components around the steady state equilibrium position under the assumption of small displacements of the journal, we have

$$\begin{cases}F'_x=F_x+k_{xx}x+k_{xy}y+c_{xx}x+c_{xy}y\\F'_y=F_y+k_{yx}x+k_{yy}y+c_{yx}x+c_{yy}y\end{cases} \tag{6}$$

The bearing stiffness and damping coefficients, k_{ij} and c_{ij} (i,j =x, y) respectively, are calculated by perturbation method. The perturbation Reynolds equations are as follows:

$$\begin{cases}\frac{\partial}{R\partial\theta}\left(\frac{h^3}{G_\theta\mu}\frac{\partial P_x}{R\partial\theta}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{G_z\mu}\frac{\partial P_x}{\partial z}\right)=-\frac{\partial}{R\partial\theta}\left(\frac{3h^2\cos\theta}{G_\theta\mu}\frac{\partial P}{R\partial\theta}\right)-\frac{\partial}{\partial z}\left(\frac{3h^2\cos\theta}{G_z\mu}\frac{\partial P}{\partial z}\right)-\frac{U}{2}\sin\theta\\
\frac{\partial}{R\partial\theta}\left(\frac{h^3}{G_\theta\mu}\frac{\partial P_y}{R\partial\theta}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{G_z\mu}\frac{\partial P_y}{\partial z}\right)=-\frac{\partial}{R\partial\theta}\left(\frac{3h^2\sin\theta}{G_\theta\mu}\frac{\partial P}{R\partial\theta}\right)-\frac{\partial}{\partial z}\left(\frac{3h^2\sin\theta}{G_z\mu}\frac{\partial P}{\partial z}\right)+\frac{U}{2}\cos\theta\\
\frac{\partial}{R\partial\theta}\left(\frac{h^3}{G_\theta\mu}\frac{\partial P_{\overline{x}}}{R\partial\theta}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{G_z\mu}\frac{\partial P_{\overline{x}}}{\partial z}\right)=\cos\theta\\
\frac{\partial}{R\partial\theta}\left(\frac{h^3}{G_\theta\mu}\frac{\partial P_{\overline{y}}}{R\partial\theta}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{G_z\mu}\frac{\partial P_{\overline{y}}}{\partial z}\right)=\sin\theta\end{cases} \tag{7}$$

The stiffness and damping coefficients are

$$\begin{cases}\begin{pmatrix}k_{xx}&k_{yx}\\k_{xy}&k_{yy}\end{pmatrix}=\int_0^L\int_0^{2\pi}\begin{pmatrix}P_x\\P_y\end{pmatrix}(\cos\theta\quad\sin\theta)d\theta dz\\
\begin{pmatrix}c_{xx}&c_{yx}\\c_{xy}&c_{yy}\end{pmatrix}=\int_0^L\int_0^{2\pi}\begin{pmatrix}P_{\overline{x}}\\P_{\overline{y}}\end{pmatrix}(\cos\theta\quad\sin\theta)d\theta dz\end{cases} \tag{8}$$

The dimensionless stiffness and damping coefficients, $\overline{k}_{ij}$ and $\overline{c}_{ij}$ (i, j =x, y), are defined as

$$\overline{k}_{ij}=k_{ij}\frac{(h_0/R)^3}{\mu L\Omega},\ \overline{c}_{ij}=c_{ij}\frac{(h_0/R)^3}{\mu L},(i,j=x,y) \tag{9}$$

To a rigid Jeffcott rotor supported horizontally by two identical journal bearings, the stability threshold speed (STS) of the fluid film bearing is related to the dimensionless stiffness and damping coefficients as

$$k_{eq}=\left(\overline{c}_{xx}\overline{k}_{yy}+\overline{c}_{yy}\overline{k}_{xx}-\overline{c}_{xy}\overline{k}_{yx}-\overline{c}_{yx}\overline{k}_{xy}\right)/(\overline{c}_{xx}+\overline{c}_{yy}) \tag{10}$$