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# Thermohydrodynamic analysis of airfoil bearing based on bump foil structure

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### KEYWORDS

Foil bearings; Thermohydrodynamic analysis; Bump foil structure **Abstract** The load carrying capacity of the gas foil bearing depends on the material properties and the configuration of the underlying bump strip's structure. This paper presents three different cases for selecting the dimensions of the foil bearing to guarantee the highest possible load carrying capacity. It focuses on three main parameters that affect the compliance number; these parameters are the length of bump in  $\theta$  direction, the pitch of bump foil, and the thickness of bump foil. It also studies the effect of changing these parameters on load carrying capacity according to both isothermal and thermohydrodynamic approaches.

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#### Nomenclature

а	viscosity constant of air, Pa s/°C
b	bearing width, m
С	radial clearance, m
$C_p$	specific heat at constant pressure, J/kg K
ď	distance between bumps, m
D	bearing diameter, m
$D_h$	housing diameter, m
е	eccentricity, m
Ε	modulus of elasticity, Pa
h	fluid film thickness, m
$\overline{h}$	dimensionless air film thickness, $\frac{h}{C}$
$h_b$	bump height, m
$h_{\min}$	minimum film thickness, m
$\bar{h}_{\rm cr}$	dimensionless film thickness at $\theta = \theta_{cr}$
J	number of iterations of the matlab program
Κ	constant reflecting the structure rigidity of the
	bumps, m <sup>3</sup> /N
$K_a$	conductivity of air, W/K m
l	half length of bump in $\theta$ direction, m
L	bearing length, m
п	number of bumps
р	fluid film pressure, Pa
$\bar{P}$	dimensionless hydrodynamic pressure, $\bar{P} = \frac{p}{p_{e}}$
$p_a$	ambient pressure, Pa
$Q_{\rm rec}$	recirculating flow rate, kg/s
$Q_{\rm suc}$	suction flow rate, kg/s
r	radius of bump, m
$r_o$	radius of spindle, m
R	radius of shaft, m
S	pitch of bump foil, m
$t_b$	thickness of bump foil, m
Т	temperature of gas, °C
$T_a$	ambient temperature of the air, °C
$T_{\rm in}$	inlet temperature of the gas film, °C
$T_{\rm ref}$	reference temperature, °C
и	linear velocity of shaft speed, m/s
v	linear velocity of gas flow in axial, $y$ , direction, m/s
W	linear velocity of gas flow in $z$ direction, m/s
W	bearing load carrying capacity, $W_x$ , $W_z$ , compo-
	nents, N

steady-state aerodynamic pressure, m non dimensionless load component in the direction of foil movement, $\bar{W}_x = \frac{W_x}{P_x RL}$ $\bar{W}_z$ non dimensionless load component in axial length direction, $\bar{W}_z = \frac{W_z}{P_x RL}$ Cartesian coordinate in the direction of motion Cartesian coordinate across the film thickness coordinate in axial length direction, m dimensionless coordinate in axial length directio $\bar{Z} = \frac{Z}{L/2}$ Freek symbols bump foil compliance number coefficient of cubic expansion wrap angle, ° bearing liner deformation, m eccentricity ratio, $\varepsilon = \frac{\varepsilon}{C}$ angular coordinate in the direction of motion or critical value of $\theta$ at subambient hydrodynam pressure bearing number, $\Lambda = \frac{6\mu_x \omega}{P_a} \left(\frac{R}{C}\right)^2$ absolute viscosity of the fluid, N s/m <sup>2</sup> viscosity of air, N s/m <sup>2</sup> Poisson ratio density, kg/m <sup>3</sup> attitude angle, ° bump arc angle, ° bump arc angle, ° angular velocity of the shaft, rad/s wiscoripts ambient, bearing entrance conditions $\theta$ quantities in the x, y, or z directions $\theta$ quantities in the r or $\theta$ directions volume time	$W_t$	radial deformation of the bump foil due to the
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$\frac{bubscripts}{ambient, bearing entrance conditions}, y, z  ext{ quantities in the } x, y, \text{ or } z  ext{ directions} \\ \theta  ext{ quantities in the } r \text{ or } \theta  ext{ directions} \\ \text{volume} \\ \text{time} \\ \frac{bverbar}{vymbol}  ext{ non-dimensionalized parameters} $	ω	angular velocity of the shaft, rad/s
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<u>everbar</u>	t	time
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#### 1. Introduction

In recent years, foil bearings have gained more attention than any other types of bearings because of their unique mode of operation and diversity of applications. They also have various advantages compared to the conventional rigid journal bearings in terms of higher load carrying capacity, lower power loss, better stability, and greater endurance. These bearings are self-acting, and can operate with ambient air or any processing gas as the lubricating fluid.

Their assembly includes a first thin smooth compliant sheet facing the shaft, one or more corrugated foil, a second sheet between the foils and a compliant sheet for preventing sagging of the first sheet between ridges of foil. Under the action of the hydrodynamic pressure, the foil structure deforms. Therefore, the fluid film pressure must be coupled to the deformation of the foil structure in order to know the characteristics of the foil bearing performance. From this point of view, many analytical studies have been conducted based on a range of structural models.

The concept of a foil bearing was first described in a report over 50 years ago by Blok and Van Rossum [1]. In 1957, Patel and Cameron [2] followed this work with another experimental investigation using steel tape and oil by introducing a more elaborate differential equation for finite width and derived a less restrictive differential equation for the gap thickness than Blok and Van Rossum [1].

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