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PERFORMANCE CHARACTERISTICS OF MISALIGNED HYDRODYNAMIC JOURNAL BEARINGS

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ABSTRACT

The performance of two misaligned journal bearings are experimentally studied on a bearing test rig . The static characteristics, such as the distributions of oil film thickness, pressure and temperature of two misaligned bearings, are measured. The traditional harmonic excitation method is then extended to estimate the sixteen dynamic force coefficients of two misaligned journal bearings.

NOMENCLATURE

b^s_{ij} damping coefficient of s-th bearing, N/m/s

$B^s_{ij} = b^s_{ij}/Cb$ dimensionless

c, c_1,c_2 bearing clearance, m; $c=(c_1 + c_2)/2$

D bearing diameter, m

ε eccentricity ratio, dimensionless

f_i, F_e excitation force and their matrix, N; i=1, 2

h , H, Hmin film thickness, m ; H=h/c,

k^s_{ij} stiffness coefficient of s-th bearing

M, m equivalent mass matrix, m is the rotor mass , kg

p_i pressure in the oil film, N/m^2 or kPa; i=1, 2,...,

R bearing radius, m

S $= \dfrac{\mu\Omega RL}{\pi W}(\dfrac{R}{c})^2$ Sommerfeld number

W bearing static load, N

μ lubricant viscosity, $N.s/m^2$

ϕ, ϕ_0 angular coordinate, attitude angle and oil , rad

ω frequency , rad/s

Ω journal rotation speed, rad/s

γ misalignment angle to the load direction

τ misalignment ratio, dimensionless

INTRODUCTION

Due to manufacturing tolerances, deflections of journal and bearing supports, asymmetric bearing load, etc., journal bearings normally operate at a misaligned condition. Although there are considerable amount of experimental investigations on the misaligned bearings (Dubois et al, 1955, Ikeuchi et al, 1985; Bannister, 1976), the experimental research on the effects of misalignment on dynamic characteristics, have not been reported. To simulate the practical bearing application in the industrial turbine machines, the free journal configuration in the test rig is shown in Fig.1. To estimate 16-dynamic coefficients of two possibly misaligned bearings, the traditional harmonic method has to be modified.

This paper extends the traditional harmonic excitation method to estimate the 16 dynamic coefficients of two different bearings simultaneously. To avoid the effect of different running conditions on the estimated coefficients, two harmonic forces at different frequencies are excited on the rotor simultaneously and all the data are acquired during one running. The distributions of the static pressures, oil film thickness and temperature around two grooved bearings are shown in the first part of this paper. The dynamic coefficients obtained from the extended harmonic method will then be presented .

JOURNAL BEARING TEST RIG

Fig.1 shows the scheme of the journal bearing test rig, where the journal is freely alignable in two possibly misaligned bearings. Its main design is shown in Fig.2 . The journal is driven by a 16 kW DC motor through a universal coupling. An air bellows is employed to apply static load to the journal via a lever and a roller bearing. The oil temperature of the large oil reservoir (1300 L) can be controlled to within $\pm 0.2^{o}C$ by 3 electric heaters. Two 0.125 HP eccentric mass vibrators are used to generate the dynamic forces. The

force amplitude can be varied by adjusting the angle between two eccentric masses. The frequencies of the dynamic forces are controlled by two 0.75 kW frequency controllers. The test rig , supported on 6 rubber springs, has the natural frequency of 5.5 Hz Each test bearing has two 32^o x 50 mm wide grooves. Two grooves are located symmetrically in the horizontal direction.

The parameters of the test rig are listed as following: bearing diameter x length, 200 x 200 mm, clearances c_1=0.147mm, c_2=0.144 mm.

Journal speed	W= 100 — 4000 rpm
Equivalent rotor mass	m=192.5 kg,
Rotor natural frequency	1344 Hz
Test rig natural frequency	5.5 Hz

Lubricant viscosity

$\mu = 0.01204e^{-0.02574(T-40)}$ Ns/m^2, T in ^oC

Fig.3 shows the distribution of the transducers in the test rig. Eight proximity probes are used to measure the journal vibration and bearing oil film thickness. Four capacitance elements are installed in the journal to monitor the bearing clearance. Six pressure transducers in the journal are used to measure the dynamic pressure around the bearings. Three strain gauge load cells are employed to measure static and dynamic forces. There are 51 T-type thermocouples around bearings and shaft. Twenty-four thermocouples are buried in each bearing surface. A turbine flow transducer with digital flow and temperature read-out is used to measure the flow of the tested bearings.

Pressure distribution

The pressure distributions in two bearings can be monitored by 6 pressure transducers in the shaft, but limited by the number of channels on the slip ring, only 4 pressure transducers were connected to the computer in the experiment. Shown in Fig.5(a) are pressures measured by three transducers in bearing 2 at the same time. The oil-film thickness measured by the probe in the journal at the centre of bearing 2 is also plotted in Fig.5(a). From Fig.5(a) the maximum pressure in midplane lie in about 10 degree ahead of the minimum oil film thickness.

The pressure distribution in bearing 1 is also asymmetric to bearing's midplane. From the asymmetric axial pressure distribution, a significant misalignment existed in the bearing. By a nonlinear curve fitting technique, when t=0.18, $\gamma = -34.6^o$, $\Delta\dot{x}$=-0.039 Ωc and $\Delta\dot{y}$=0.12 Ωc, the theoretical pressure distribution (by the finite difference method, Qiu (1994)) correlates well with the measured pressure distribution as shown in Fig.5(b). Fig.5(c) shows the fitted pressure distribution in all bearing area.

Eccentricity and the attitude angle

The eccentricities at bearing centers are transferred from the eccentricities measured at two probe planes . Fig.6 shows the relations between the Sommerfeld number and eccentricity, and Fig.7 shows the experimental attitude angle at different loads. The solid lines are calculated results of a parallel bearing and the dashed lines are results of a misaligned bearing. In the theoretical calculation, the misalignment parameters are assumed according to the static pressure as shown in Fig.5(b). Referring to Fig.6, the theoretical Sommerfeld numbers of both bearings agree well with the experimental results when the misalignment is considered in the theoretical calculation. In the case of misaligned bearing, the variation of the attitude angle between the experimental and the theoretical values is about 5^o.

Supply flow

Fig.8 shows the measured dimensionless supply flow. It is the average flow to two bearings. If the misalignment is considered, the difference between the theory and experiment is smaller than that of the aligned bearing especially at moderate loads.

Temperature Distribution

The temperature distributions around two bearings are measured by 48 T-type thermocouples. Fig.9 shows typical temperature distribution around two bearings. At heavy load (S=0.091, Fig.9 (a) and (b)), the temperature reaches the highest points at down stream of the minimum oil-film thickness. The maximum temperature is 9 ^oC higher than the inlet temperature (Fig.9(a)). Axially, the temperature at the mid plane is lower than those measured at L/4 distance to the bearing ends. This is because the colder supply oil reaches the mid plane first and cools the bearing surface.

Estimated Bearing Coefficients

Fig.10 and 11 graphically present the estimated characteristics of bearing 1 and 2 at different loads. The solid lines are calculated results of a parallel (aligned) bearing and the dashed lines are results of a misaligned bearing.The misalignment parameters are approximately estimated according to the static pressure (Qiu & Tieu, 1994). The theoretical coefficients are calculated by the infinitesimal perturbation method (Qiu ,1994).

Critical Speeds

Employing the estimated bearing coefficients, the critical stable speeds at different loads can be predicted. Fig.12 shows the dimensionless stability contour compared with the theoretical predictions. If the misalignment is considered, the experimental results agree very well with the theoretical predictions. The experimental results confirm the conclusion that misalignments improve the stability of the journal bearing (Qiu & Tieu, 1994).

CONCLUSIONS

(1) A bearing test rig with free journal configuration is described in this paper. The static characteristics, such as the distributions of oil film thickness, pressure and temperature in two misaligned bearings, are measured.

(2) When the misalignment is considered in the theoretical calculation, the measured pressure distribution agrees well with the theoretical prediction.

(3) The estimated bearing dynamic coefficients of two tested journal bearings correlated well with the theory when the misalignment is considered in the theoretical calculation. The experimental results verify that the misalignment can improve the stability of the rotor-bearing system.

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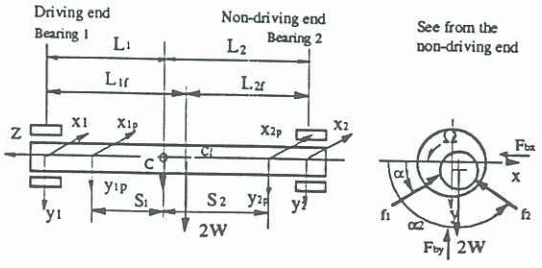


Fig.1 Scheme of the journal bearing test rig

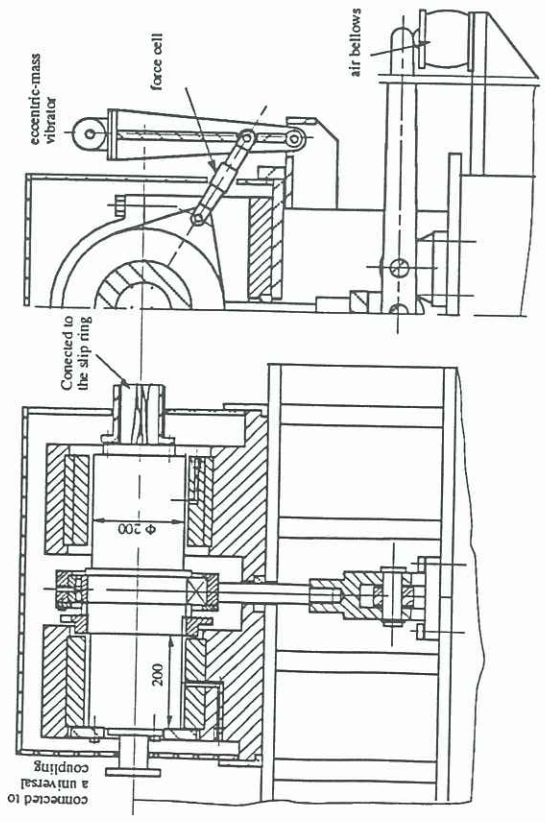


Fig.2 Design of journal bearing test rig

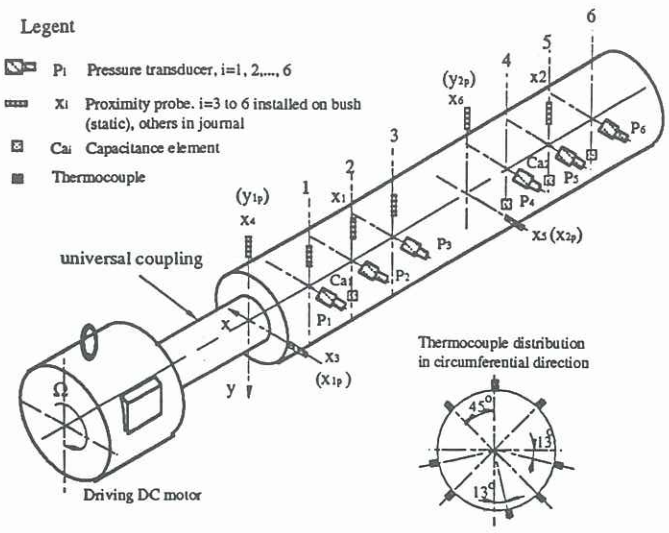
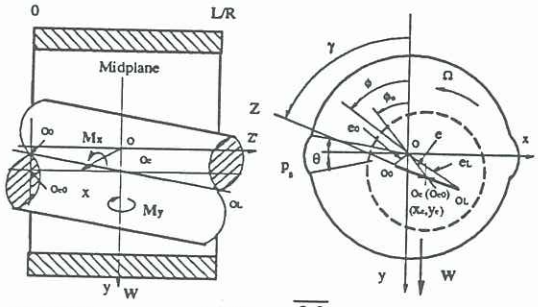


Fig.3 Distribution of transducers in the journal bearing test rig.



$Z' = Z - L/2R$ tilt ratio: $\tau = \frac{\overline{O_o O_L}}{2c}$

o centre of the bearing
o_o centre of the journal in the left end plane
o_c centre of the journal in the bearing midplane
o_{eo} cross point of the parallel line of the bearing axil in the bearing left end plane
o_L centre of the journal in the right end plane
$\overline{O_o O_L}$ projection of journal axis in the midplane

Fig. 4 Misaligned journal bearing

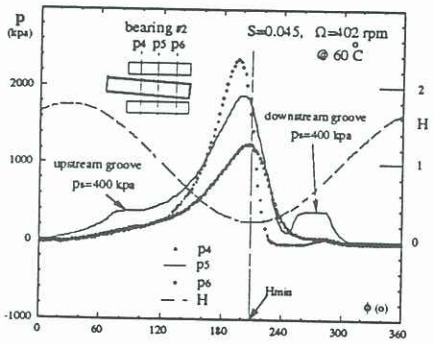


Fig.5 (a) Measured pressures and oil film thickness

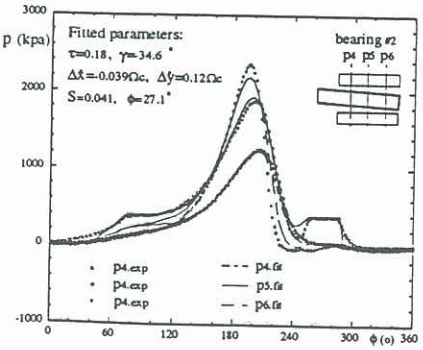


Fig.5(b) Theoretical pressures at the fitted misalignment parameters

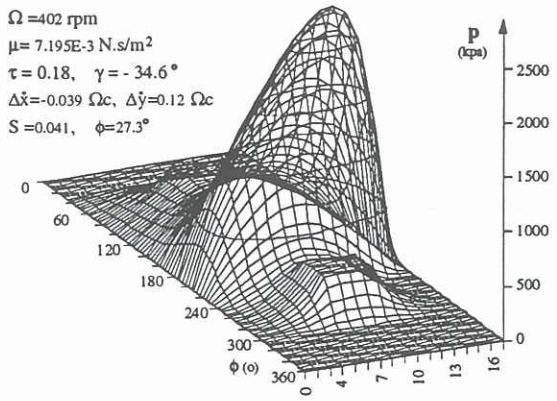


Fig.5 (c) Theoretical pressure distribution at the fitted misalignment parameters

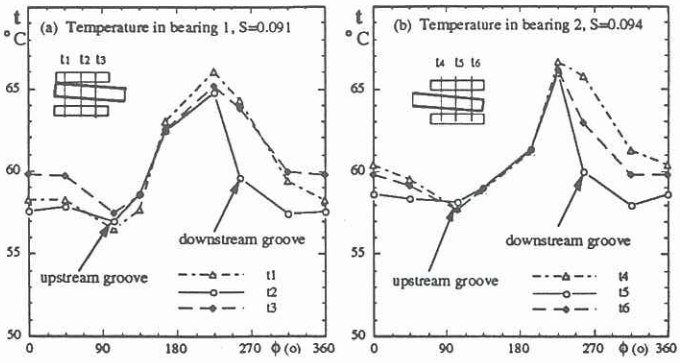


Fig.9 Temperature distribution around two bearings at different loads

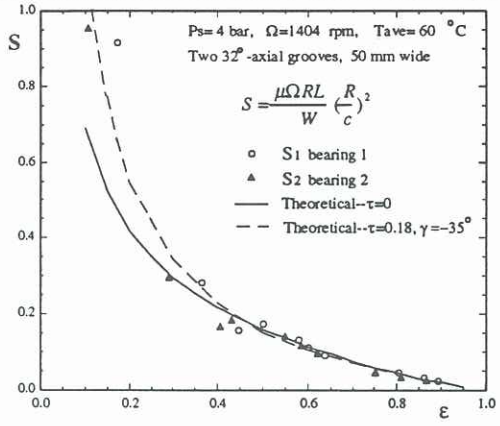
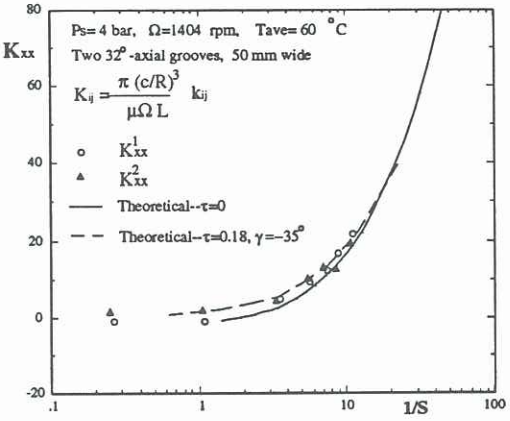


Fig.6 Measured Sommerfeld number vs eccentricity



$K_{ij} = k_{ij} / C_k$, C_k=2.569x10^7 N/m

Fig.10 Estimated bearing stiffness coefficient K_{xx}

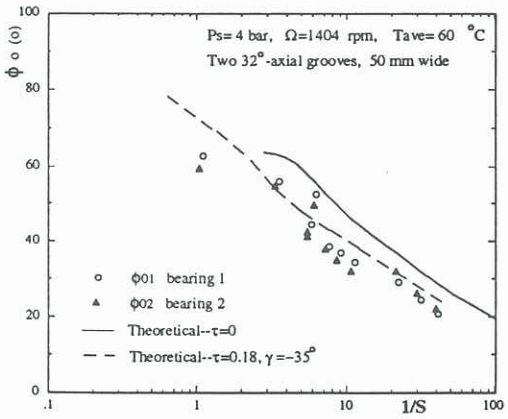


Fig.7 Measured attitude angle vs load

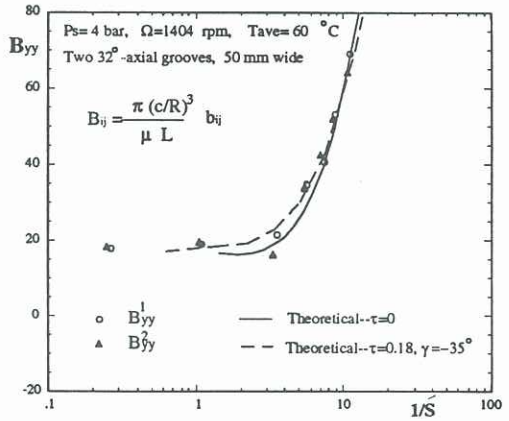


Fig.11 Estimated bearing damping coefficient B_{yy}

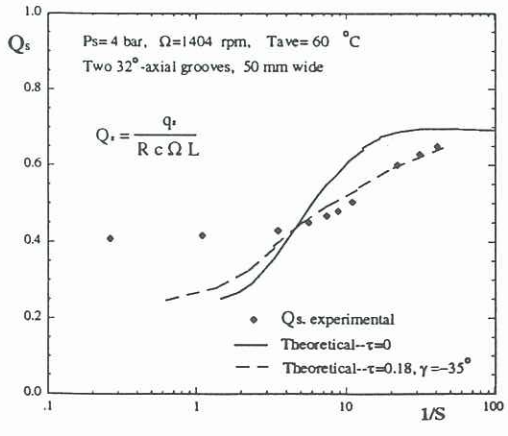


Fig.8 Experimental and theoretical supply flow

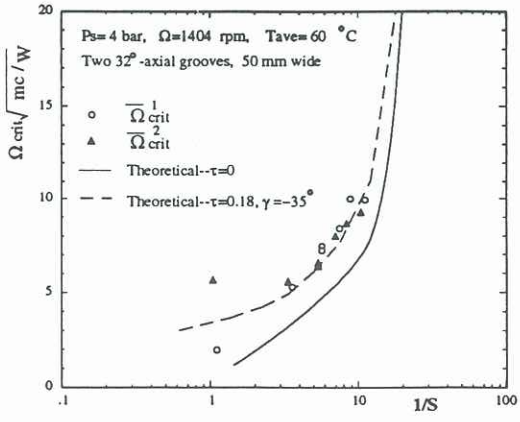


Fig.12 Critical speeds predicted from the estimated bearing coefficients