N86-30171

INFLUENCE OF DESIGN PARAMETERS ON OCCURRENCE OF OIL WHIRL IN ROTOR-BEARING SYSTEMS

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Oil whirl instability is a serious problem in oil lubricated journal bearings. The phenomenon is characterised by a sub-synchronous vibration of the journal within the bush and is particularly apparent in turbogenerators, aeroengines and electric motors. This paper presents a review of previous papers on the subject of oil whirl, and describes a simple theory which has been used to aid the design of an oil whirl test rig.

Predictions of the onset of oil whirl made by the theory presented in this paper were found to agree with those of previous researchers. They showed that increasing the shaft flexibility, or the lubricant viscosity, and decreasing the bearing radial clearance tended to reduce the oil whirl onset speed thus making the system more unstable.

NOTATION

A_{ab}	Net stiffness coefficient: Force in a direction per unit displacement in b direction
B_{ab}	Net damping coefficient: Force in a direction per unit velocity in b direction
c	Bearing radial clearance.
C_{1-5}	Coefficients of system characteristics equation.
D	Diameter of bearing.
F_s	Static force actiing on bearing
{F}	Force Vector.
$[K_j]$	Bearing stiffness matrix
$[K_s]$	Shaft stiffness matrix
L	Length of bearing.
M	Mass of rotor.
S_{1-4}	Roots of system characteristic equation.
{U}	Displacement vector for journals relative to bearing bushes
{V}	Displacement vector for mass relative to journals

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$[K_e]$	Equivalent stiffness matrix.
x	Displacement in horizontal direction
X	Function of displacement amplitude in horizontal direction.
y	Displacement in vertical direction.
Y	Function of displacement amplitude in vertical direction.
{Z}	Displacement vector of rotor mass relative to bushes
$\alpha_{1,2}$	Real parts of roots of system characteristic equation
$\beta_{1,2}$	Imaginary part of roots of system characteristic equation
ω	Vibration frequency

Suffices

x	Refers to horizontal direction
y	Refers to vertical direction

INTRODUCTION

The phenomenon of oil whirl was first discussed and investigated in the 1920's, the most important paper of that time being by Newkirk and Taylor (1). It was found that whipping of the shaft ceased when the oil supply was halted which led to the conclusion that it was due to vibrations within the oil film. A simplified explanation was proposed that relied upon the fact that there is a pressure difference around the journal surface.

Work by Robertson (2) and Hagg (3) led to an improved theory to predict the presence of oil whirl. Robertson solved Reynolds equation to determine the oil film forces and then equated these with body forces to predict the motion of a rotor mounted on both a stiff and an elastic shaft. In contrast, Hagg determined oil film force by examining the continuity of the oil flow and then determined the system stability by assuming the bearing oil film to behave as a spring and a damper. Both methods were found to agree with the results of Newkirk and Taylor. Hagg also determined, experimentally, that tilting pad bearings were unable to sustain oil whirl vibrations and hence were more stable.

In 1956 Hagg and Sankey (4) suggested that by assuming the oil film to behave as pairs of springs and dampers, hence having two displacement coefficients and two velocity coefficients, the coefficients may be added to the rotor system properties and a more reliable estimate of oil whirl may be calculated. To improve stability estimates further, Morrison (5) proposed that four further coefficients are required which link forces in the horizontal directions to velocity and displacement in the vertical direction, and vice-versa (the "cross-coupling coefficients"). In 1959, Hori (6) noted that in certain cases the displacements could well be comparatively large compared to the oil film and so the analysis shoud be essentially non-linear. Hence, Holmes (7) and Bannister (8) were able to argue the case for utilising non-linear coefficients when modelling the oil film and therefore improving the estimate of oil whirl onset.

Bannister (8), Newkirk and Lewis (9), and Downham (10) have all examined the effect of misalignment of the journal within the bush. Bannister and Newkirk and Lewis have determined experimentally that misalignment was a stabilising factor, and Downham suggested that misalignment effectively stiffens the oil film amd that this was the stabilising factor. Later, Mukherjee and Rao (11) carried out a numerical analysis to determine the values of damping and stiffness coefficients for an inclined journal bearing but no comparison was made between these coefficients and those for an aligned bearing. Dhagat et al (12) investigated the stability of journal bearings with lateral and skew motions by determining a total of sixteen coefficients. However, neither stated the effect misalignment had upon stability. More recently Gupta (13) found that misalignment of the bearing caps, which he termed preset, had an advantageous effect upon stability.

Simmons and Marsh (14) showed experimentally and theoretically that increased pedestal flexibility has a detremental effect upon whirl onset. They also postulated the possibility of a cessation of oil whirl as rotor speed increases and a subsequent secondary whirl onset; this however, has not been validated in practice. Lund (15,16) both calculated, and illustrated experimentally, the effect of shaft flexibility on whirl onset. As with pedestal flexibility it was found that a more flexible shaft has a detremental effect upon stability. More recently Ruddy (17) has modelled a rotor-mounted on a flexible shaft with various bearing designs to investigate system stability. Ruddy, however, does not investigate the effect of varying shaft flexibility.

The effect of imbalance has been examined by Barret et al (18) who used a time transient non-linear analysis to examine the motion of the journal. They found that if a rotor operated above whirl onset an increase of imbalance can reduce the magnitude of the whirl orbit to below that of a fully balanced rotor. Akers et al (19) have also utilised a non-linear analysis to solve the equation of motion for the shaft in both the vertical and horizontal directions. Although they examined the effect of an out of balance load there is much confusion over the interpretation of results the conclusions being that imbalance may or may not increase stability, and it was unclear what the deciding factor was. More recently Bannister (20) used a finite difference technique to solve Reynolds equation and hence produce the oil film force, and then proceeded to solve the equations of motion, which was a similar method to that of Akers. Bannister states that the bearing operation goes through a transition stage, before entering a region of instability, where he states that an examination of the magnitude of the non-synchronous component of vibration can indicate when the system is about to go unstable. Bannister also agrees with Barrett that imbalance does produce a degree of stability.

It was earlier stated that the oil film may be represented by a series of springs and dampers. Much work has been carried out to determine these coefficients empirically, especially by Lund (21), Morton (22), Bannister (9) and more recently Parkins (23). Unlike previous researchers Bannister and Parkins allowed for oil film non-linearity and so their coefficients may be used to produce a more accurate estimate of oil whirl. More recently, Bentley and Muszynska (24,25) have used the application of a sinusoidal perturbing force to examine the oil film dynamic stiffness and stability of journal bearings. It was found that an increase in oil supply pressure or a decrease in oil temperature causes a subsequent increase in dynamic stiffness. A method for determining whirl onset experimentally was also described, although the effect of oil temperature and feed pressure on whirl onset was not examined.

In general, the conclusions reached by most of the previous investigators are that reducing the flexibility of the shaft and pedestal does tend to promote instability, whereas the inclusion of misalignment tends to stabilise the system. There is also a general concensus that when determining whirl onset and whirl orbits the use of non-linear theory leads to greater accuracy. However, there is confusion over the effect of imbalance and further research is needed in this area.

This paper presents a theoretical method of predicting stability based on linear oil film coefficients derived by Lund (21) and is similar to the method suggested by Cameron (26). The analysis is developed further to allow for the presence of shaft flexibilities using matrix notation to express the equations which model the bearing and shaft properties. The effects of varying shaft flexibility, radial clearance and oil viscosity on whirl onset have been examined and the results were found to agree in general with those of previous researchers.

THEORY

This analysis applies to the case of a single mass M mounted at the centre of a light flexible shaft of stiffness K which runs in two identical oil film journal bearings as illustrated in Figure 1. The oil film in the journal bearings is assumed to be described by the eight linearised oil film force coefficients. The force-displacement relationships for the journal bearings are, in matrix form:

$$\left\{ F \right\} = \left[K_j \right] \left\{ U \right\} \tag{1}$$

where the matrix K_j contains the oil film force coefficients for the two oil films acting in parallel. Similarly, a force displacement relationship may be written for the shaft itself

$$\left\{ F \right\} = \left[K_s \right] \left\{ V \right\} \tag{2}$$

Inverting the stiffness matrices of equations (1) and (2) and adding together the resulting displacements gives

$$\left\{ U \right\} + \left\{ V \right\} = \left[[K_j]^{-1} + [K_s]^{-1} \right] \left\{ F \right\} \tag{3}$$

where $[K_s]$ is of the form

$$\begin{bmatrix} K_{xx} & 0 & 0 & 0 \\ 0 & K_{xx} & 0 & 0 \\ 0 & 0 & K_{yy} & 0 \\ 0 & 0 & 0 & K_{yy} \end{bmatrix} = [K_s]$$

since both damping and cross-couple terms have been considered as negligible for the shaft itself or simply

$$\left\{ Z \right\} = \left[K_e \right]^{-1} \left\{ F \right\} \tag{4}$$

which on re-inverting gives

$$\left\{ F \right\} = \left[K_e \right] \left\{ Z \right\} \tag{5}$$

where K_e contains equivalent stiffness and damping coefficients, A_{xx}, B_{xx}, etc., which allow for both oil film and shaft flexibility. The matrix $[K_e]$ is, however, more complicated since it is itself an inversion of the summation of two inverted matrices. However, it is still of the form of the matrices $[K_s]$ and $[K_j]$ and may be written as

$$\begin{bmatrix} K_{xx} & -\omega C_{xx} & K_{xy} & -\omega C_{xy} \\ \omega C_{xx} & K_{xx} & \omega C_{xy} & K_{xy} \\ K_{yx} & -\omega C_{yx} & K_{yy} & -\omega C_{yy} \\ \omega C_{yx} & K_{yx} & \omega C_{yy} & K_{yy} \end{bmatrix} = [K_e]$$

The equations of motion of the rotor mass in the horizontal and vertical directions may now be written as

$$A_{xx}.x + A_{xy}.y + B_{xx}.\dot{x} + B_{xy}.\dot{y} = -M\ddot{x} \qquad (6)$$

and

$$A_{yx}.x + A_{yy}.y + B_{yx}.\dot{x} + B_{yy}.\dot{y} = -M\ddot{y} \qquad (7)$$

Assuming sinusoidal displacements of the form

$$x = Xe^{st} \qquad (8)$$

and

$$y = Ye^{(st + \Phi)} \qquad (9)$$

in the horizontal and vertical directions respectively, these may be differentiated to obtain expressions for velocity and acceleration. Substituting back into equations (6) and (7), dividing throughout by e^{st} and eliminating the ratio X/Ye^{Φ} leads to the system characteristic equation

$$M^2 S^4 + (B_{xx} + B_{yy})M S^3 + [(A_{xx} + A_{yy})M + B_{xx} B_{yy} - B_{xy} B_{yx}] S^2 +$$

$$(A_{yy} B_{xx} + B_{yy} A_{xx} - A_{xy} B_{yx} - B_{xy} A_{yx}) S + (A_{xx} A_{yy} - A_{xy} A_{yx}) = 0 \qquad (10)$$

The roots of equation (10) usually occur in conjugate pairs of the form

$$S_{1,2} = \alpha_1 \pm j\beta_1 \text{ and } S_{3,4} = \alpha_2 \pm j\beta_2 \qquad (11)$$

For motion of the mass to be unstable at least one value of α must be greater than zero, the value of β indicates the frequency of the resulting vibrations.

RESULTS

A computer program based on the theory presented was written to investigate the stability of journal bearing-rotor systems. The bearing stiffness and damping coefficients used in the analysis were calculated from dimensionless groups determined by Lund (21) for plain journal bearings.

The analysis was used to investigate the effects of shaft stiffness, journal diameter, bearing clearance, and lubricant viscosity, as indicated by the enclosed graphs. Unless otherwise stated as a variable, the system parameters used were shaft stiffness 85 kN/m, journal diameter 58.4 mm, bearing L/D ratio 1, bearing radial clearance 0.1 mm and lubricant viscosity 0.046 Ns/m^2.

DISCUSSION

Figures 2 and 3 show stability maps for the cases when shaft stiffnesses are 85 kN/m and 85 MN/m respectively. It can be seen from Figures 2 and 3 that the occurence of whirl onset corresponds to a much lower value of system operating parameter for the stiffer shaft; this means that a much larger value of journal rotational speed is necessary for whirl onset and so the system is more stable. To further illustrate the effect of shaft stiffness, Figure 4 shows whirl onset boundaries for a series of shafts having different stiffnesses. It is clear that as shaft stiffness increases stability increases; these conclusions are in agreement with those reached by Simmons and Marsh (14) and Lund (15, 16) as discussed previously.

Generally, the whirl onset boundary associated with a particular system, as illustrated in Figures 2 and 3 may be approximated to a horizontal line, for example, in Figure 3, below an operating parameter of 0.8×10^{-3} the roots α are either positive or only marginally negative. Hence the area between the horizontal line and the whirl onset boundary may be discussed as a transition stage; this tends to agree with Bannister's findings (20) that a system reaches a transitional stage of instability before finally becoming unstable.

The effect of varying viscosity on the system is illustrated in Figure 5. It can be seen that by decreasing the viscosity of the lubricant the whirl onset boundary attains a lower operating parameter, thus a decrease in viscosity tends to cause an increase in stability. This has been demonstrated in the past on several large machines, where it has been found that when initially running a rotor up to operating speed oil whirl may exist, but after a period of running, that is after the oil increases in temperature and hence its viscosity is reduced, the oil whirl may often disappear (27).

Figure 6 illustrates the effect of bearing radial clearance upon system stability. It may be seen that as clearance is reduced the whirl onset boundary coincides with a larger operating parameter. However, it is unclear how much a particular change in clearance affects the system stability, this is because bearing clearance forms a part of the operating parameter to which the vertical axis refers. In this case it is more meaningful to plot whirl onset speed against radial clearance as shown in Figure 7. From this graph, it may be seen that for relatively large clearances an increase in clearance gives greater stability. However, for relatively small clearances this may not be the case and in fact the converse may occur. In this light it is interesting to note the conflicting opinions of Newkirk and Taylor (1) who determined experimentally that a decrease in clearance will increase stability and that of Barwell (28).

CONCLUSIONS

The authors agree with past researchers that design parameters do have an effect on rotor bearing stability. These may be summarised as follows:-

(i) Journal/bush-misalignment tends to promote stability.

(ii) Reducing shaft/pedestal stiffness reduces system stability.

(iii) Reducing lubricant viscosity increases system stability.

(iv) Reducing radial clearance decreases stability.

(v) The effect of imbalance upon stability is not clear and needs further investigation

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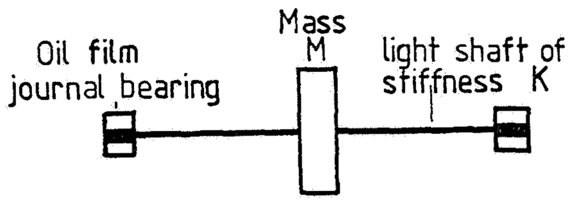
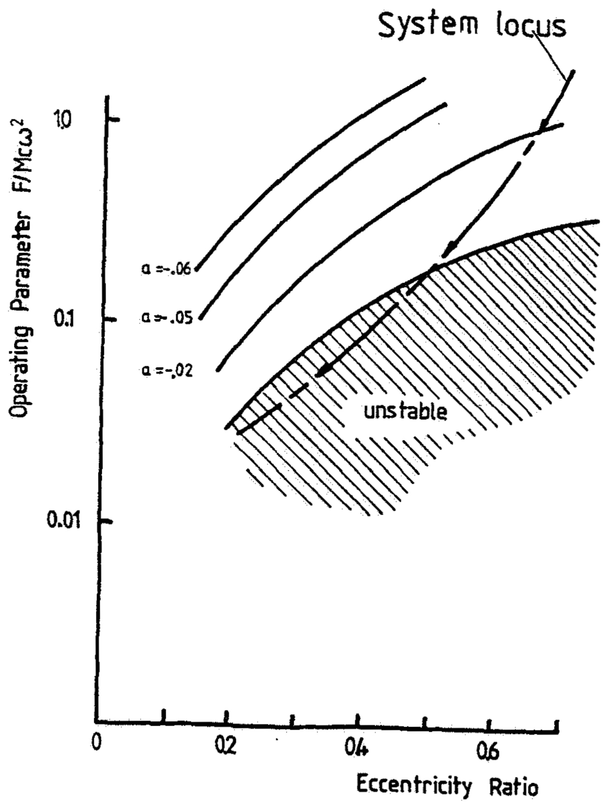


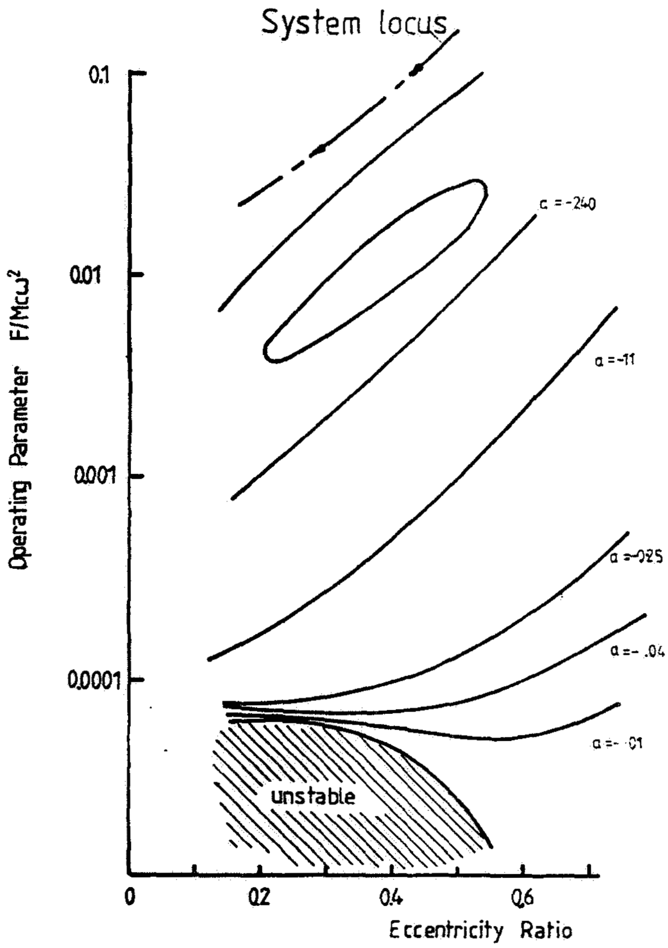
Rotor-Bearing Configuration

Figure 1



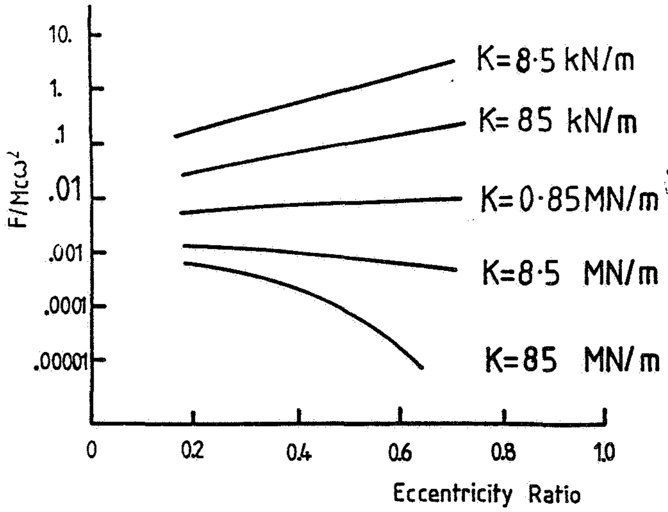
Stability map for shaft of stiffness 85 kN/m

Figure 2



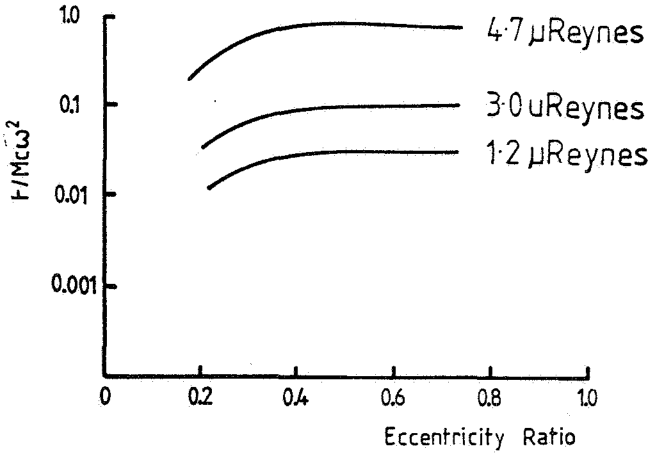
Stability map for shaft of stiffness 85 MN/m

Figure 3



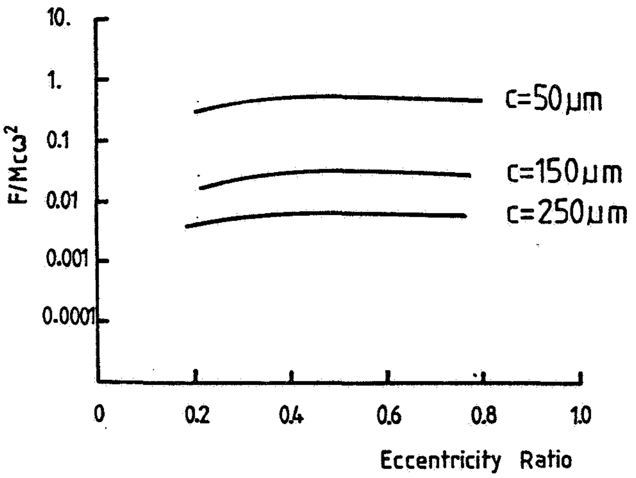
Variation of whirl onset boundary
with shaft stiffness

Figure 4



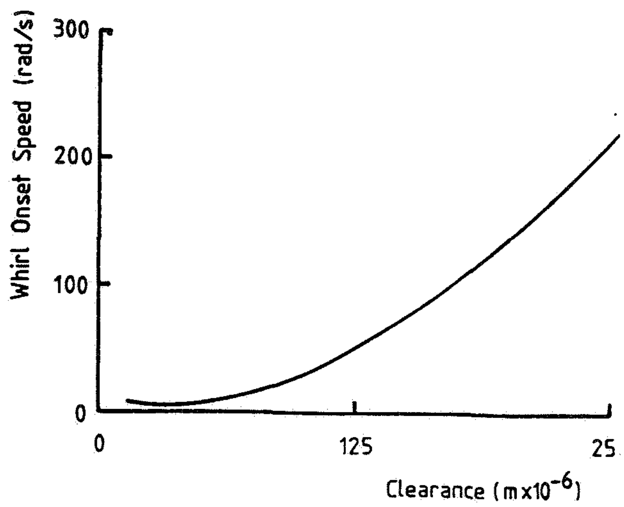
Variation of whirl onset boundary
with lubricant viscosity

Figure 5



Variation of whirl onset boundary
with radial clearance

Figure 6



Variation of whirl onset speed with
clearance

Figure 7