LB 456/80

Stability of profile bore bearings: influence of bearing type selection

D.R. Garner, C.S. Lee and F.A. Martin*

Fluid film bearings play an important part in determining and controlling the vibrations of a rotor system. Under certain conditions, however, the bearings can effectively cause vibrations. This paper presents information on the resistance of several common bearing types to self-excited whirl, and on the dynamic coefficients of the oil film which influence the rotor system. Certain simplifying assumptions are necessary in producing generalised information, but the data allow realistic bearing selections to be made

The factors which influence the vibrational behaviour of rotating machinery are numerous, diverse and sometimes difficult to quantify. An analytical model of the system should allow for the mechanical configuration and properties of the rotor, pedestals, foundations etc, and incorporate the operational influences of such components as bearings, seals, glands and couplings. A properly constructed model allows the response of the machine to various forcing mechanisms to be determined. Simplified calculations, however, can still provide a useful guide to possible system problems. For example it is relatively easy to calculate the first critical speed of a rotor on rigid supports, and for initial design purposes it is often sufficient to assume that the influence of actual bearings will be to reduce this speed by 20–30%. Similarly, by making certain simplifying assumptions the resistance of the bearings to self-excited vibrations can be judged in isolation from the rest of the system, but such information still is not easy to obtain since it involves a detailed study of hydrodynamic action within the bearings.

At slow speeds hydrodynamic bearing design is concerned mainly with keeping the shaft and bearing surfaces completely separated by a lubricant film. As speeds increase greater attention has to be paid to keeping acceptable thermal conditions within the bearing, ensuring that the heat generated in shearing the thin lubricant film is adequately removed by conduction through the components or, more realistically, by the quantity of cool oil passing through the clearance space. The influence of the bearings on the vibrational behaviour of the rotor also becomes more important. For rotors operating below the first critical speed the oil film stiffness can play a large part in determining the vibration levels resulting from unavoidable out of balance, hydraulic, or other forces. Equally this stiffness can significantly affect the value of the rotor critical speed, and the damping from the oil film, normally larger than other sources of damping within a machine, tends to control the vibration amplitude at the critical speed. For rotors operating above the first critical speed the bearings may or may not have a significant effect on vibrational behaviour, depending on the mode shape of the vibrations, ie the position of the bearings relative to the nodes.

Four stiffness and four damping terms are needed to describe the linearised performance of a bearing oil film; in each case two 'direct coupled' terms to quantify the change of oil film force in the same direction as a displacement or velocity, and two 'cross coupled' terms to describe the change of oil film force in a direction

*Research and Development Organisation, The Glacier Metal Co Ltd, Alperton,

Notation

b	Bearing length, m
B	Oil film damping, Ns/m
c_b	Bearing radial clearance, m, see Fig 1(a)
c_d	Bearing diametral clearance ($=2\,c_b$), m
c_l	Lobe radial clearance see Fig 1(a), m
d	Bearing diameter, m
K	Oil film stiffness, N/m
M	Rotor mass per bearing, kg
N	Shaft rotational speed, rev/s
N_C	Critical speed of rotor on rigid supports, rev/s
N_F	Flexible rotor threshold speed of instability, rev/s
N_R	Rigid rotor threshold speed of instability, rev/s
N_W	Oil film whirl frequency, Hz
W	Applied load on bearing, N
δ	Sag of rotor under static conditions, m
η	Dynamic viscosity, Ns/m^2

Dimensionless Terms

$B'_{xx}, B'_{xy}, B'_{yx}, B'_{yy}$ damping components, $\left(=\dfrac{c_d N B}{W}\right)$

For suffix notation see Fig 1(b); B_{xy} is the rate of change of the x component of oil film force with velocity in the y direction

$K'_{xx}, K'_{xy}, K'_{yx}, K'_{yy}$ stiffness components, $\left(=\dfrac{c_d K}{W}\right)$

M'_F Flexible rotor critical mass, $\left(=\dfrac{M c_d N_F^2}{W}\right)$

M'_R Rigid rotor critical mass, $\left(=\dfrac{M c_d N_R^2}{W}\right)$

W' Load number, $\left(=\dfrac{W}{\eta N b d}\left(\dfrac{c_d}{d}\right)^2\right)$

m Preset, see Fig 1(a)

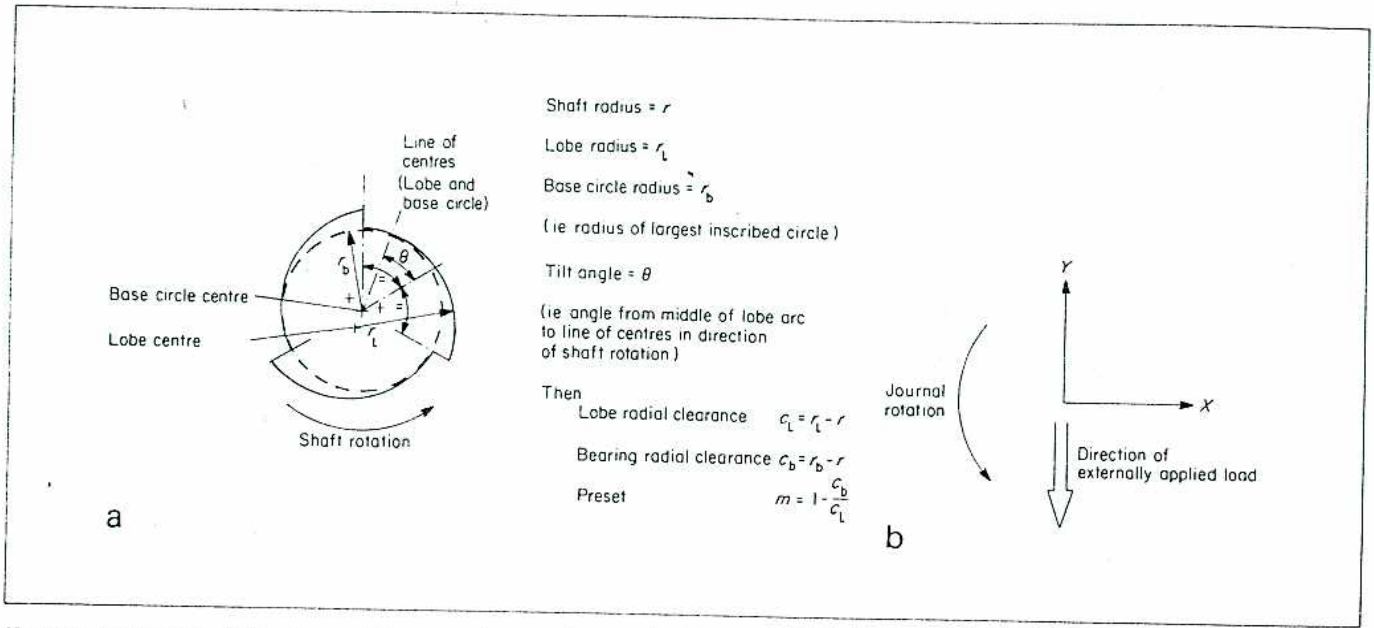


Fig 1(a) Definition of bearing profile. (b) Coordinate system for dynamic characteristics

perpendicular to a displacement or velocity. It is the cross coupling effects which promote bearing instability. A tilting pad journal bearing has very low cross coupled terms, perhaps two orders of magnitude less than the direct coupled terms, and in all practical cases can be thought of as an inherently stable bearing. This certainly does not mean that a machine running on tilting pad bearings will not vibrate, since critical speed, balance, or hydraulic or aerodynamic forces may still be beyond the capabilities of the bearing to control. Additionally, there are often advantages to be obtained in terms of space and complexity by using a fixed profile bearing instead of a tilting pad.

The purpose of this article is to present information on stability, stiffness and damping for some commonly used fixed profile types, enabling a realistic choice of bearings for high speed machinery, to be made at an early stage in the design process. Data are restricted to bearings having a length to diameter ratio of 0.5, and assume laminar conditions within the oil film.

Oil film whirl

When a bearing breaks into classical oil film whirl the shaft centre precesses around the bearing centre at a frequency close to half the rotational speed. Such vibrations may be unacceptable themselves, they may excite some other fundamental frequency of the system (oil whip), or they may reduce the load capacity of the bearing to such an extent that overheating and consequent break-up of the bearing lining material occurs.

The tendency to whirl can be suppressed by de-rating a cylindrical bore bearing by reducing the length, adding grooving in the loaded region etc, or by incorporating a shallow, blind, circumferential recess in the unloaded half. Because of lubricant entrainment in the recess this produces a parasitic load in the top half of the bearing, which has a stabilising effect. The depth of the recess can be quite critical; for highest stability it should be of the same order as the diametral clearance[1]. Alternatively some form of bore profiling in the bearing can be used.

The direct result of profiling is to provide additional convergent regions within the bearing clearance space and therefore additional regions of hydrodynamic pressure. There are many ways in which this can be done: Fig 1(a) shows a general profiled bore. As drawn it is constructed from a series of circular arcs, as are all of the profiles considered in this article. Non-circular lobes are quite permissible, but impose additional difficulties in specification and analysis. The higher stability profiles all have some disadvantages, for example high lubricant requirements, an inability to accept bi-directional rotation of the shaft, complication of manufacture and/or assembly; generally, the simpler the profile used the better. With a tilt angle (θ) equal to zero the bearing will have the same characteristics for either direction of rotation.

Table 1 Bearing profiles considered

Bearing type	Code used in figs	Load direction (direction 4 in Fig 8)	Oil gutterway angle (see Fig 8), degrees	Preset, m, used in Fig 3(a) Fig. 9	Preset, m, used in Fig 3(b) Fig. 10	Comments
Cylindrical	CY	Between gutterways	30			
Lemon bore	LB	Into lobe	30	0.6	0.5	
Offset halves	OH	Into lobe	15	0.6	0.5	
Four lobe	4L	Into gutterway	20	0.8	0.7	Symmetrical lobing
Three lobe	3L	Into lobe	20	0.8	0.7	Symmetrical lobing

Rigid rotors

By considering a symmetrical, simply supported rotor, the resistance of any bearing to self-excited whirl can be expressed as a function of the oil film stiffness and damping[2]. Under given operating conditions a dimensionless critical mass can be predicted[1,3] which is on the limit of stability (see notation). Information on five profiles, plus a cylindrical bore bearing (Table 1), is given in Fig 2 as a function of a dimensionless load number (note that all the dimensionless groups use the bearing clearance, not the lobe clearance). Various presets (the amount of lobing) are considered. To use these graphs, calculate a value of W', a value of M'_R, and plot this point to ensure that it lies below the appropriate curve. For rotors operating under gravitational weight loading only, the mass and load terms are directly related, but external forces such as gear loads can have significant stabilising effects because of the reduction in the ratio of M/W. Note that it is not possible directly to obtain a threshold speed of instability from the dimensionless critical mass number, since the terms on both axes involve the rotational speed, N. If a threshold speed is required then an operating line must be plotted on these graphs, and the point where this crosses the limit line gives the required speed value.

Experience has shown that different profiles have distinct ranges of preset that are desirable and practical. Preset cannot be fixed absolutely at the design stage for any lobed bearing, since it will vary due to manufacturing tolerances on the two clearances. A variation in preset of approximately 0.1 is realistic (Table 1). Fig 3 shows a direct comparison between the various profiles for the chosen maximum and minimum presets. Fig 3(a) should be used at minimum clearance conditions, Fig 3(b) at maximum clearances.

In any real machine there will be damping mechanisms external to the bearing which will tend to improve the stability situation. These may come from couplings, gear meshes or, particularly strongly, hydraulic components. Using the graphs presented here should, therefore, lead to a conservative design.

Although the assumption has been made that the rotors are symmetric, experience has shown that the normal non-symmetrical rotors found in turbines, compressors etc can be reasonably examined using these graphs.

The frequency of whirl can also be expressed simply (Fig 4). Vibration measurements taken on real machines in which bearings are tending to whirl

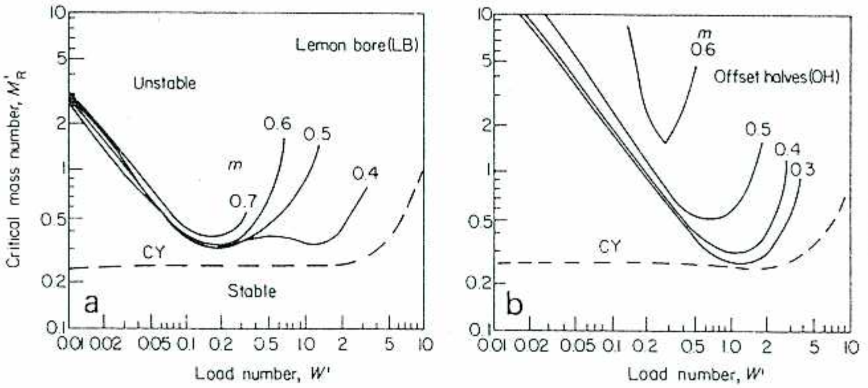


Fig 2 Bearing instability limits for various profile bores; including cylindrical (b/d=0.5, rigid rotor)

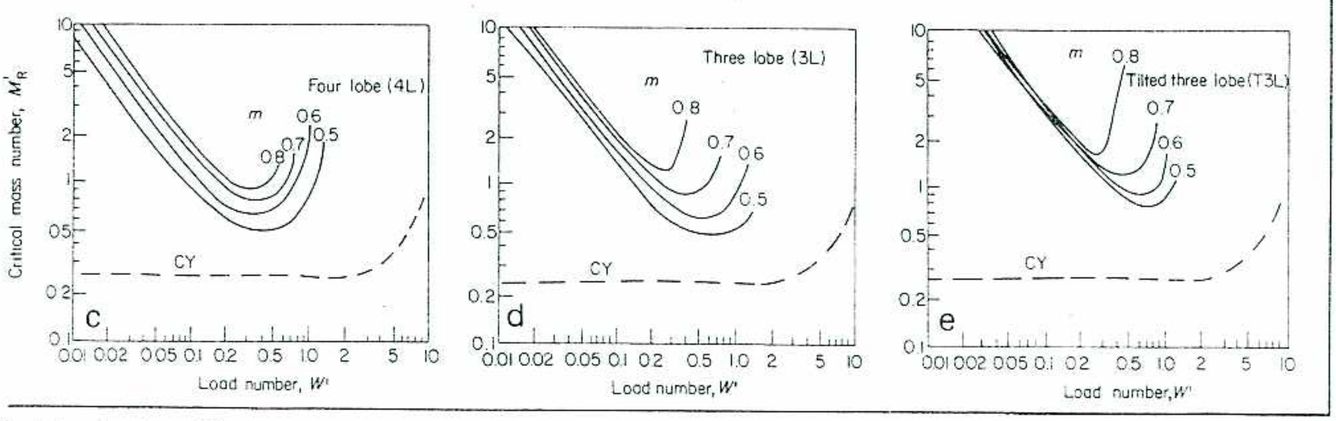
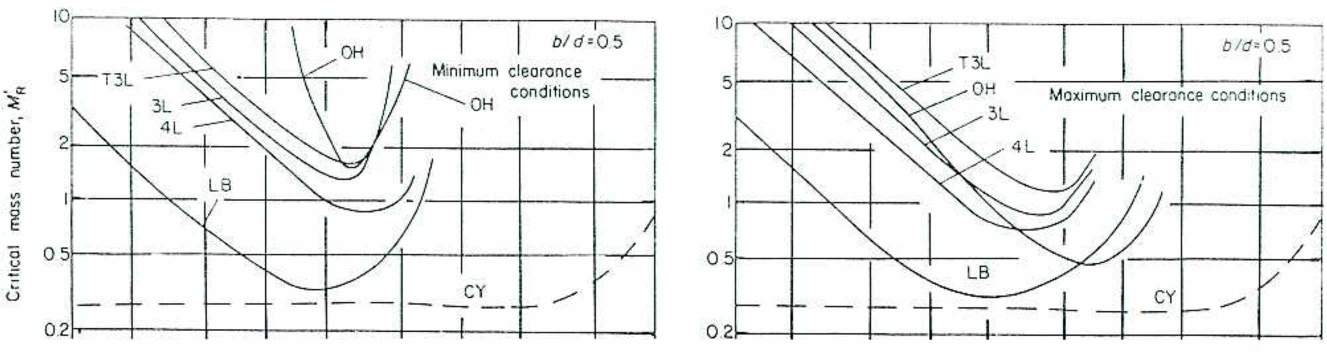
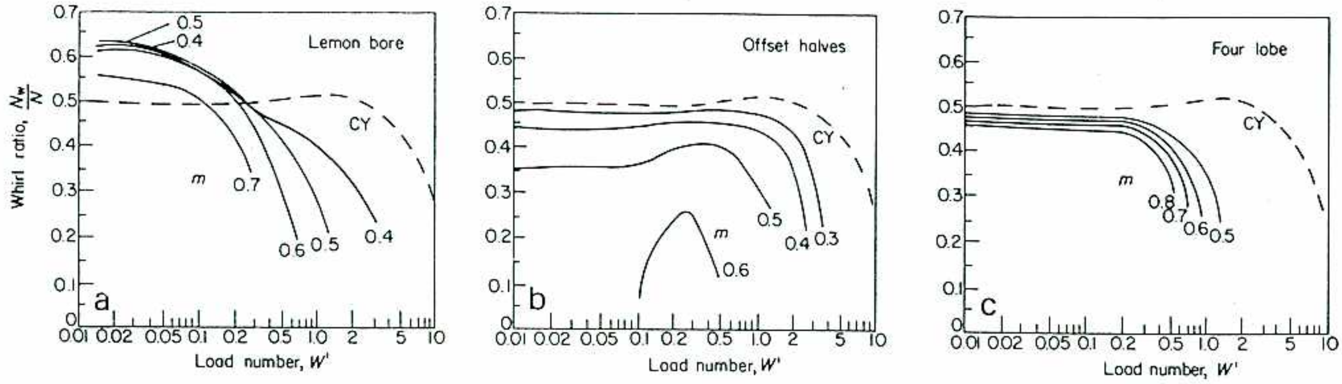


Fig 3 Bearing instability limits at recommended presets – rigid rotor (see Table 1)





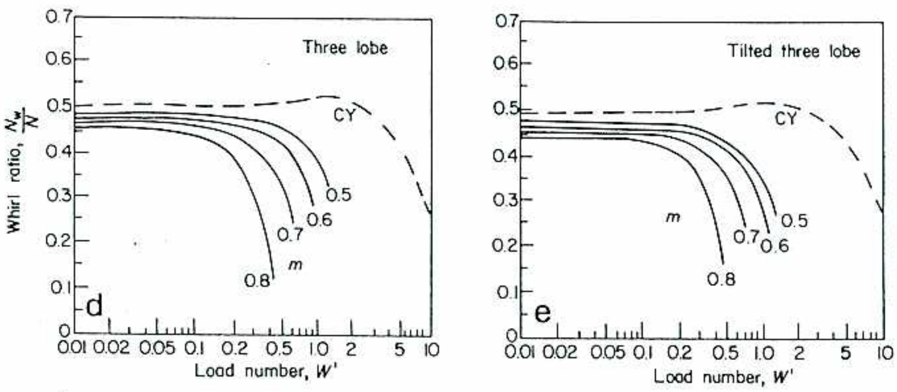


Fig 4 Frequency of whirl at bearing instability for various profile bores including cylindrical (b/d=0.5)

normally could be expected to show frequencies slightly below these values. For example with cylindrical bore bearings it is unusual in practice for self induced vibrations to occur outside a frequency range 0.42 to 0.47, although predicted values (the dashed line throughout Fig 4) may be higher than this. In many cases the vibration amplitudes are so severe that no thought is given to exactly what the frequency is! Two particular points of interest in these curves are the possibility of seeing above half frequency whirl in a lightly loaded lemon bore bearing, and at or below one third frequency in an offset halves bearing; both of these extremes are seen in practice. For a bearing which is running stable but close to whirl, imposed (external) frequencies near to the whirl frequency ratio can cause vibration problems. If, for instance, a machine has some forcing mechanism at one third of rotational speed then an offset halves bearing may give higher vibration amplitudes than a cylindrical bore.

Flexible rotors

Flexibility in a rotor will make the stability situation worse, and the information contained in Figs 2 and 3 cannot be applied directly. For a symmetrical, flexible rotor with a single lumped mass at mid-span, Fig 5 (developed from reference 2) quantifies the lowering of the rigid rotor limiting dimensionless critical mass as a function of the rotor sag. The ratio of flexible to rigid dimensionless critical masses obtained from this chart (as shown by the dotted lines) is used to plot new limit lines on Figs 2 or 3.

Since the critical speed on rigid supports, N_C, may be related directly to the rotor sag, δ, then the instability threshold speed for a flexible rotor may be expressed in terms of the shaft critical. Fig 6, for a cylindrical bore bearing, has on the axes terms similar to the dimensionless critical mass and dimensionless load terms, but using the N_C instead of the actual operating speed. From this chart the ratio of instability speed to critical speed may be obtained directly.

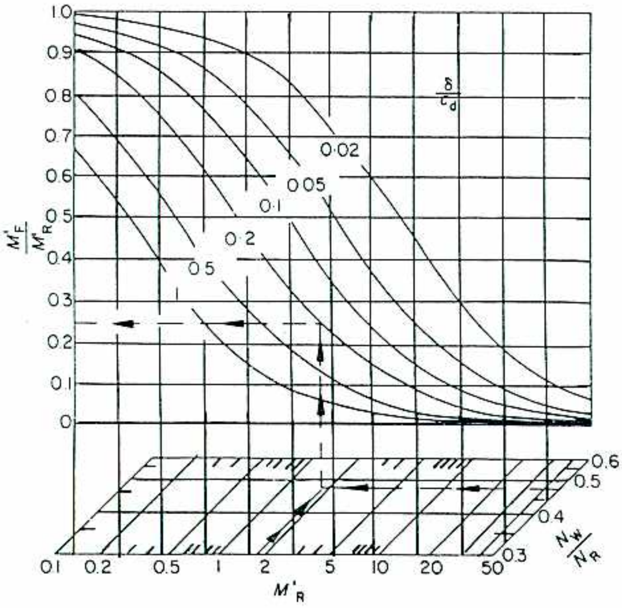


Fig 5 Modifying effect of rotor flexibility on rigid rotor bearing instability limits

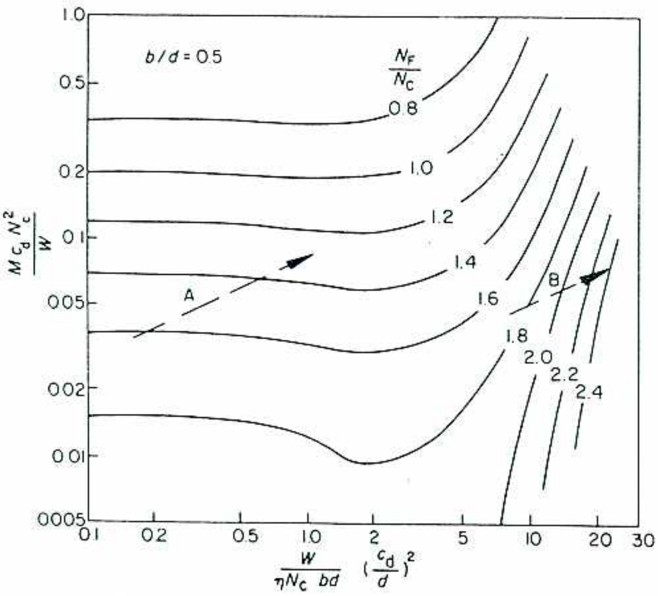


Fig 6 Bearing instability limit for a flexible horizontal rotor with cylindrical bore bearings

The arrows on this diagram illustrate the direction one would move, and the consequential effect, when increasing clearance. For example, at 'A' an increase in clearance lowers the threshold speed, N_F, while at 'B' it raises the threshold speed.

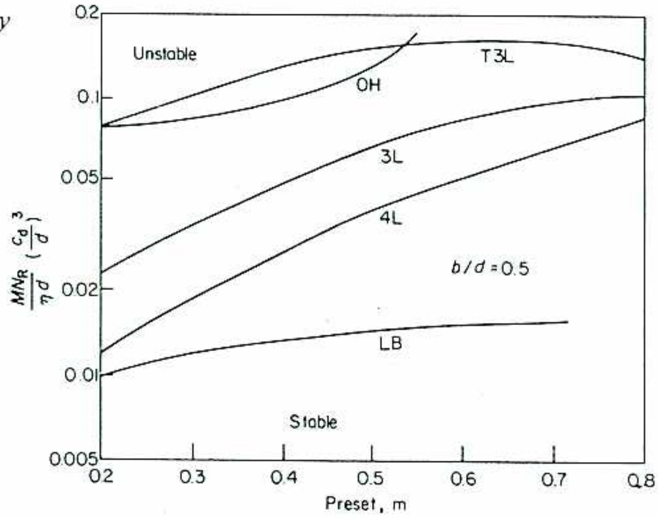
Vertical rotors

With a vertical rotor, when the applied load can be nominally zero, stability cannot be checked on the basis used so far. For a rigid rotor Fig 7 gives the limit of stable operation as a function of preset for various profiles. The offset halves bearing has a different trend from the other profiles, and at high presets is predicted to be inherently stable. The cylindrical bore bearing is not shown on this graph since in theory it is susceptible to whirl under any no-load conditions. In practice they are sometimes used in vertical applications, normally at low speeds, but they probably rely for stability on damping within the machine or the existence of small but finite loads.

Effect of load direction

In any bearing which has grooving or profiling the operation is dependent upon the direction as well as the magnitude of the applied load. Fig 8 gives some indication of the loading directions which are to be avoided. Within the exaggerated clearance space of each profile has been plotted the inherently stable boundary. When the shaft centre is outside this boundary the bearing is not susceptible to oil film whirl; inside the boundary it may or may not be stable depending upon the precise operating conditions. The journal locus paths for various directions of applied load are also plotted. Taking the lemon bore as an example this shows that loading in direction 4 (the normal load direction into the crown of the bearing) moves the shaft into the inherently stable region much more quickly than loading direction 5. Other factors, such as minimum oil film thickness, will also influence the acceptability of any load direction. It can be seen that the offset halves bearing at low loads operates in a region of inherent stability, confirming the results presented in Fig 7. It is easy to place too much importance on these boundaries; they are certainly not automatic limits to stable operation, and satisfactory operation inside a boundary is quite feasible.

Fig 7 Bearing instability limits for a vertical (zero applied load) rigid rotor



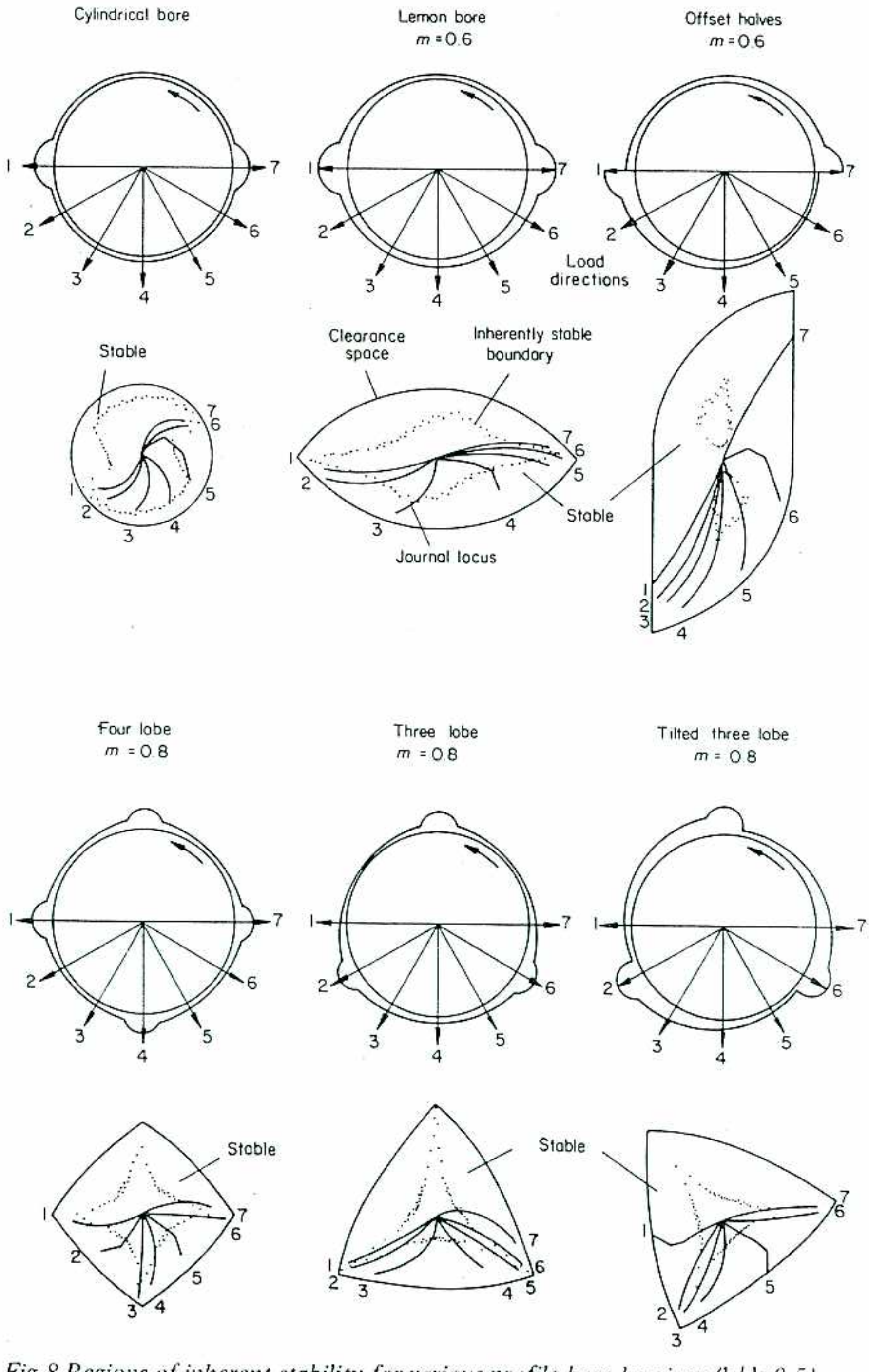
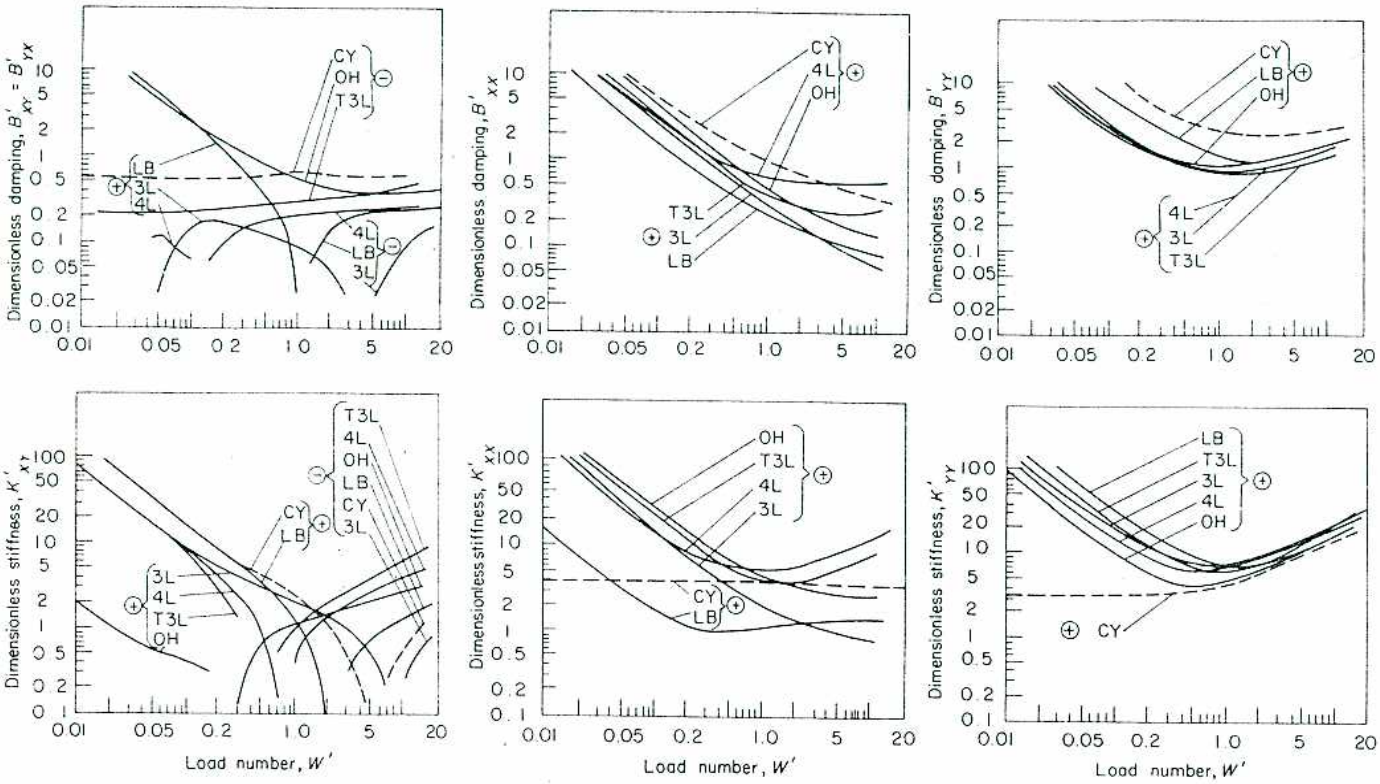


Fig 8 Regions of inherent stability for various profile bore bearings (b/d=0.5)



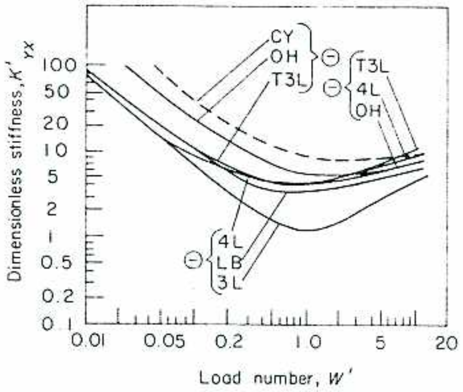


Fig 9 Linearised stiffness and damping coefficients at recommended presets. Minimum clearance conditions (see Table 1). b/d=*0.5*

Dynamic characteristics

The stiffness and damping characteristics of the oil film are required in any detailed rotor dynamics study[4], and are plotted for the various profiles at typical maximum (Fig 9) and minimum (Fig 10) presets. The cross coupled damping terms, B_{XY} and B_{YX}, in the analysis here are identical. If particularly high (or low) stiffness or damping are required then these charts enable the relative merits of the various profiles to be judged.

Choice of profile

Many factors should be considered when deciding on the bearing type to be used in a specific application. These include oil film thickness, operating temperatures, power loss, oil flow requirement, stability and response to dynamic forces. The following summary of the strengths and weaknesses of various profiles, however, may be useful as a guide.

Cylindrical bore: The simplest bearing, use wherever possible; poor stability, but grooving can be used to give some improvements.

Lemon bore: Better stability than the cylindrical bore with little additional manufacturing complexity; vertical stiffness and damping very good, but horizontal rather poor; load capacity good; suggested maximum preset 0.6; presets up to 0.7 quite common for increased stability but then suffers increasingly from poor horizontal characteristics and high oil flow requirement.

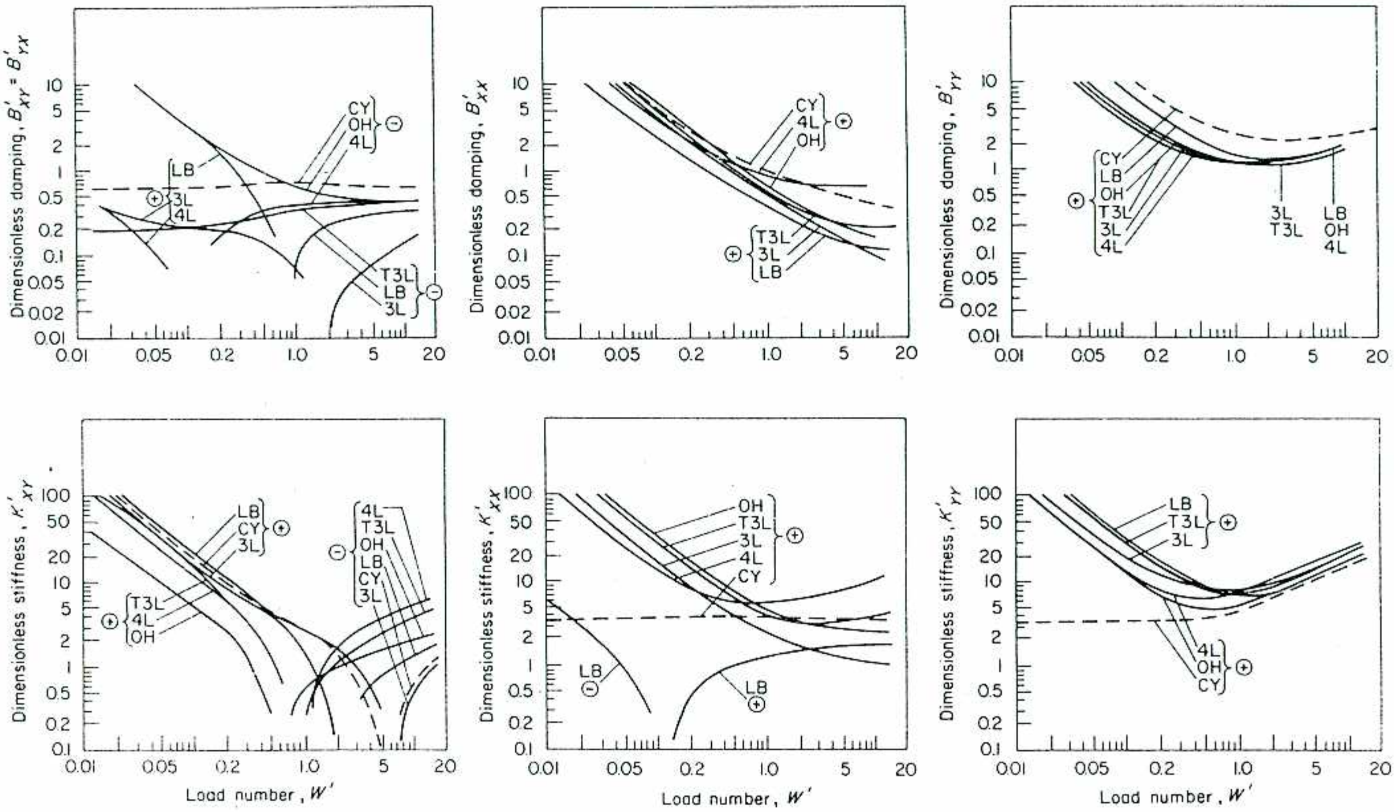
Offset halves: Extremely good stability characteristics and high stiffness and damping; careful choice of preset and manufacturing tolerances can give essentially equal direct coupled stiffnesses; high oil flow requirement, which can be an advantage at high speeds or when large heat soak into the bearing is expected; load capacity good if loaded into or near to the crown, but avoid loading towards step; only suitable for one direction of rotation; suggested maximum preset 0.6.

4-lobe: Useful where moderate stability and stiffness and damping are required; load capacity rather poor; large lobe clearances are normally needed, with the result that this bearing tends to operate with non-laminar flow conditions at lower speeds than other profiles, and thus has higher power loss and higher temperatures; suggested maximum preset 0.8–0.85.

3-lobe, symmetrical: Good for high speed applications with high stability and stiffness and damping; difficult to use satisfactorily where (for assembly reasons) bearings have to be in halves; suggested maximum preset 0.8–0.85.

3-lobe, tilted: Improved stability over the symmetrical 3-lobe; high oil pumping characteristic allows for high speeds, or operation with heat soak into the bearings; some reverse rotation possible depending on amount of tilt used, but certainly not a bi-directional bearing; suggested maximum preset 0.8–0.85

With any profiled bore bearing care must be taken over the choice of (base circle) clearance. A tight clearance which gives acceptable running temperatures may nevertheless cause problems if rapid starts from cold are required. In most machines the shaft heats up much faster than the bearing housing or casing, and consequent differential thermal expansions cause a transient reduction in clearance. This in turn leads to higher rates of



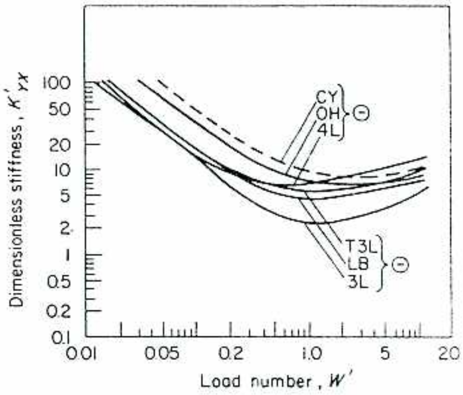


Fig 10 Linearised stiffness and damping coefficients at recommended presets. Maximum clearance conditions (see Table 1). b/d=*0.5*

shear, and if temperatures do not rapidly equalise then an almost total loss of clearance can result, with damage to the high spots of the bearing bore. As a general rule the minimum clerance should not be taken below 0.001 mm per mm of shaft diameter, but even then on high speed machines careful starting procedures may still be required. These could include circulating pre-heated oil before start-up, or specifying gradual speed increases. This damage mechanism is unusual in cylindrical bore bearings perhaps because of the larger clearances used. A cylindrical bore bearing is also better at tolerating significant dynamic loads, such as out of balance.

Conclusions

While the vibrational characteristics of any rotor/bearing system result from the interaction of all the components within that system, the influence of individual components may still be studied in isolation provided that certain simplifying assumptions are made. Information has been presented on the stability of hydrodynamic journal bearings, with both rigid and flexible, but symmetrical, rotors. The use of profiled bore bearings can significantly increase the speed at which oil film whirl occurs, relative to a simple cylindrical bore bearing. Oil film stiffness and damping, important in controlling or suppressing the critical speed vibrations of the rotor, are also greatly influenced by bore profile. Each bore shape has its own strengths and weaknesses, and there is no single optimum profile for high speed bearing applications.

References

1. **Allaire P.E.** Design of Journal Bearings for High Speed Rotating Machinery. *ASME Design Engineering Conference May 1979: Fundamentals of the Design of Fluid Film Bearings*
2. **Lund J.W.** Self-Excited, Stationary Whirl Orbits of a Journal in a Sleeve Bearing. *PhD Thesis, Rensselaer Polytechnic Institute, Troy, New York, 1966*

Acknowledgements

The authors wish to thank the Directors of The Glacier Metal Co Ltd for permission to publish this article, and to recognise the assistance of various colleagues in its writing.

3. **Akkok M. and Ettles C.M.** The Effect of Grooving and Bore Shape on the Stability of Journal Bearings. *ASLE Annual Meeting May 1979. Preprint 79-AM-6D-3*
4. **Lund J.W. and Thomsen K.K.** A Calculation Method and Data for the Dynamic Coefficients of Oil-Lubricated Journal Bearings. *ASME Design Engineering Conference April 1978: Fluid Film Bearings in Rotating Machinery and Bearing Design Optimisation*