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The effect of manufacturing tolerances on the stability of profile bore bearings

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SYNOPSIS Profile bore journal bearings are used to suppress rotor instability caused by oil film whirl. The effect of manufacturing tolerances on the clearances in a tilted 3-lobe bearing and an offset halves bearing is demonstrated. Results show the effects of these dimensional variations and how these changes affect the instability threshold speed.

1 INTRODUCTION

Fixed profile, fluid-film journal bearings are used in many different types of turbomachinery and when properly designed they offer a comparatively simple, long life support for the rotor. Furthermore, the dynamic characteristics of the oil film may be used in many cases to control the vibrational behaviour of the rotor. Oil film stiffness can be varied to change the frequency of critical speeds and oil film damping will reduce the amplitude of rotor response to forced vibrations, particularly at critical speeds. The dynamic characteristics of other parts of the machine (e.g. seals, foundations, couplings and impellers) will also influence the rotor dynamics but it is frequently the case that the dynamic interaction between the rotor and the bearing oil films will have the most significant influence on the rotor vibrations.

One problem that can arise with hydrodynamically lubricated bearings, under conditions of high speed and low loads, is the onset of self-excited vibrations. In essence, the fluid film acts as a mechanism for the transfer of rotational energy from the rotor to translational or whirling motion. In this condition, the shaft centre precesses around the bearing centre at a frequency close to half the rotational speed. Hence the names half-speed whirl or oil film whirl are frequently used to describe this phenomenon. High levels of vibration nearly always occur, with the possibility of overheating and consequent damage to the bearing lining, and conditions may be made even worse by the excitation of some other fundamental frequency of the system (e.g. oil whip).

This self-excitation is an instability of the rotor-bearing system. The onset speed of high levels of rotor vibration, associated with this bearing induced instability is known as the instability threshold speed. This speed will be affected by many factors in a real machine; these include rotor flexibility, bearing support flexibility, the rate of acceleration of the rotor, residual unbalance and forces acting on the rotor arising from impellers, seals etc.

Fortunately, it is possible to judge the resistance of the bearings to self-excited vibrations in isolation from the rest of the system. A detailed study of the hydrodynamic action within the bearing, when combined with simplifying assumptions to overcome the complexity of interaction in the complete machine, allows the bearing designer to predict the instability threshold speed. Care is required when interpreting such an analysis, but experience has shown that bearing induced instability can be avoided with the use of profile bore bearings.

Engineering components can never be manufactured to an absolute dimension. Bearing dimensions must be toleranced to allow for the limits in accuracy of the machining process and measurement techniques. The resulting small variation in dimensions, altering the shape and size of the clearance space, can have a significant effect on the performance of fluid-film journal bearings. The operating limits on the design of a fixed profile journal bearing include minimum allowable film thickness, maximum allowable metal temperature, maximum allowable oil outlet temperature and a check on stable operation. All these parameters are affected by manufacturing tolerances. However, in the case of high speed turbomachinery, loads are not normally high and film thickness is not generally a problem. The major concerns are firstly to ensure sufficient oil passes through the clearance space to remove the heat generated in shearing the thin lubricant film and secondly that bearing induced self-excited vibration is avoided.

The purpose of this paper is to investigate how manufacturing tolerances affect the clearances in two common forms of profile bore journal bearing, the tilted 3-lobe and the offset halves. The effects of changes in clearances on the instability threshold speed are then illustrated, taking into account the change in effective viscosity for different speeds and clearances for the study case conditions.

2 NOTATION

2.1 Dimensional Notation

		Units for Dimensionless Terms
b =	Bearing length	m
C_b =	Bearing radial clearance	m
C_d =	Bearing diametral clearance (base circle)	m
C_L =	Lobe radial clearance	m
d =	Bearing diameter	m
M =	Rotor mass per bearing	kg
N =	Shaft rotational speed	rev/s
P =	Specific load = $W/(bd)$	N/m^2
r_b =	Base circle radius (see Fig 1) (largest inscribed circle radius)	m
r_L =	Lobe radius (see Fig 1)	m
r_s =	Shaft radius	m
W =	Applied load on bearing	N
δ =	Tolerance on shaft radius (mm)	-
η =	Dynamic viscosity	Ns/m^2

2.2 Dimensionless Terms (in consistent units)

Load Number $W' = \frac{P}{\eta N}\left(\frac{C_d}{d}\right)^2$

Critical Mass $M' = \frac{M}{W}\frac{C_d}{2}N^2$

Speed term $\frac{\eta N}{P}\left[\frac{M}{W}d\left(\frac{P}{\eta}\right)^2\right]^{0.4}$

Clearance term $\frac{C_d}{d}\left[\frac{M}{W}d\left(\frac{P}{\eta}\right)^2\right]^{0.2}$

Preset $m = 1 - \frac{C_b}{C_L}$

3 BEARING GEOMETRY

In the profile bore bearings considered here, the geometry is partly defined by two clearances, a bearing radial clearance C_b and a lobe radial clearance C_L.

The bearing clearance is directly related to the base circle shown dotted in Fig 1, where the base circle is the largest possible inscribed circle with radius r_b about centre O_b. In terms of radii, the bearing radial clearance C_b is defined as:-

$$C_b = r_b - r_s$$

This clearance is sometimes known as the 'shake' clearance or 'assembled' clearance. (The corresponding diametral clearance C_d is also used in the dimensionless terms).

The lobe clearance C_L relates to circular lobe arcs of radius r_L, Fig 1, and is defined as:-

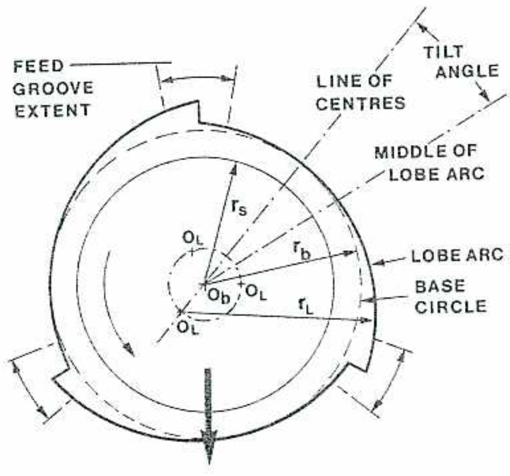
$$C_L = r_L - r_s$$

It is apparent that when the lobe clearance, C_L, is equal to the bearing clearance, C_b, and the centres of the circular lobe arcs and base circle are coincident, the complete bearing profile is cylindrical. Increasing C_L relative to C_b or offsetting the lobe arc centre with C_b constant, progressively increases the film convergence within the lobes. This convergence, commonly considered as a geometric preloading effect, may be defined as a preset m:-

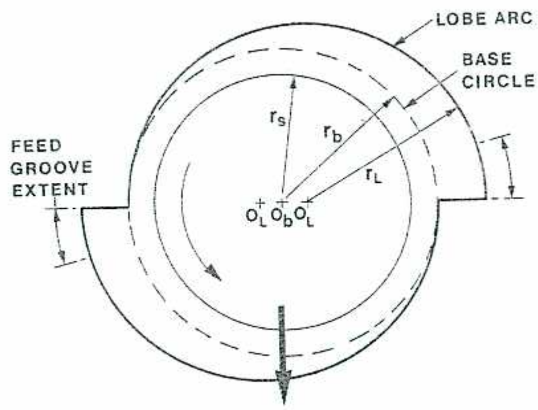
$$m = 1 - \frac{C_b}{C_L} = \frac{\text{distance } O_L \text{ to } O_b}{C_L}$$

Values of preset up to 0.6 are normally used for offset halves bearings, and presets up to 0.85 are used for 3-lobe bearings.

Apart from preset, another important geometric factor describes the position of the smallest local clearance when the shaft is at the bearing geometric centre. For the offset halves



a) TILTED 3-LOBE BEARING



b) OFFSET HALVES BEARING

Fig 1 Bearing geometry

bearing, this position occurs at the end of the lobe. In tilted 3-lobe bearings, the tilt angle Fig 1a defines the angular position of this point as measured downstream from the mid-lobe arc position. A 'symmetrical' 3-lobe bearing, which has the same operating characteristics for either direction of rotation, has zero tilt angle. Values of tilt angle of zero, 20 and 40 degrees are commonly used by the authors' company; increasing tilt angle gives improved stability (1) and higher rates of oil flow through the bearing.

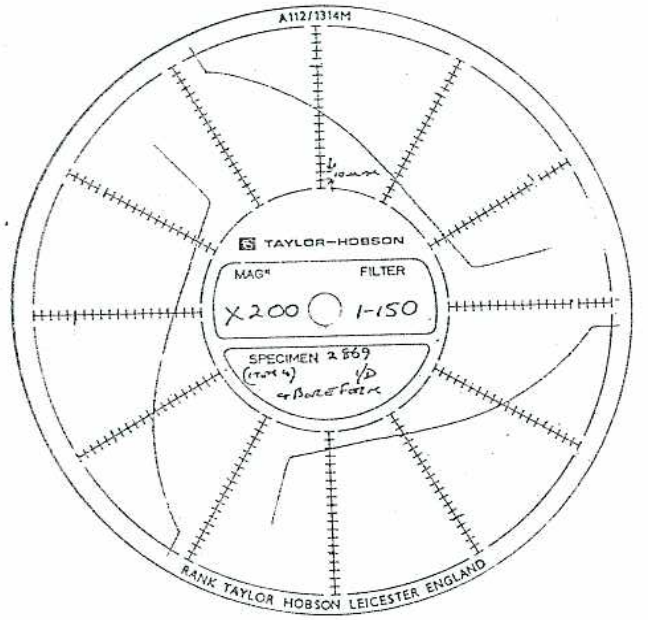


Fig 2 Talyrond trace of a typical tilted 3-lobe bearing profile

Manufacturing tolerances and the method of production will affect the various dimensions in the bearing and the shaft diameter. Thus variations in the shape of the bearing and changes in the clearances C_b and C_L can occur and such variations can significantly affect the performance of multilobe bearings. However, with advanced manufacturing techniques and attention to housing design, together with the correct location of the bearing components, the 'shape' of the profile in terms of similarity between lobes and accuracy of the circular arc form can be closely controlled. The 'Talyrond' profile of a typical production tilted 3-lobe bearing, manufactured using a computer controlled machine tool, is shown in Fig 2. This figure demonstrates the similarity between lobes and the quality of the profile which can be achieved. The extremely small differences that do exist between one lobe and another will not significantly affect the hydrodynamic action of the bearing. Other manufacturing methods can produce very much less satisfactory profiles (2). Apart from the profile it is still necessary to consider the effects of manufacturing tolerances on the dimensions that cause C_b and C_L to vary.

4 DIMENSIONLESS STABILITY ANALYSIS

In the present work, the pressure distribution in the oil film was calculated using a finite difference solution of Reynolds' Equation. A perturbation analysis was then carried out to predict the eight dynamic coefficients, (3) to (representing the threshold of instability) for various journal positions in the clearance space. This analysis, forming the basis to the study case, considers a symmetrical rigid rotor. The effect of rotor flexibility (for a symmetrical shaft) is to reduce the rigid rotor instability threshold (more unstable) especially at high speeds.

The dimensionless groups of critical mass M' and load number W' in slightly different forms, have become the standard method of presenting a journal bearing stability map. Such a map is shown in Fig 3 (ignore inner grid for now). A typical boundary line showing where the rotor-bearing system becomes unstable is shown for a tilted 3-lobe bearing with a preset of 0.7. When considering an instability threshold speed below which the bearing is stable it is not useful to consider M' and W' in isolation since speed is contained in both of these terms. For example if the operating speed and viscosity was such that M' was equal to two and W' equal to one, then this point in Fig 3 would be to the right of the boundary curve, in the stable region. There would be some temptation to say that this condition was inherently stable as there is no upper limit to M' at this point. However, this is not the case as M' does not uniquely define the instability threshold speed (W' also being a function of speed). Alternative dimensionless groups have therefore been derived;

A speed term

$$\left[\frac{\eta N}{P}\right]\left[\frac{M}{W} d \left(\frac{P}{\eta}\right)^2\right]^{0.4} \quad \text{(independent of clearance)}$$

and a clearance term

$$\left[\frac{C_d}{d}\right]\left[\frac{M}{W} d \left(\frac{P}{\eta}\right)^2\right]^{0.2} \quad \text{(independent of speed)}$$

The relationship between the proposed clearance and speed terms and the better known critical mass and load number terms is shown by the tilted grid in Fig 3. This is a purely mathematical relationship and will apply to any

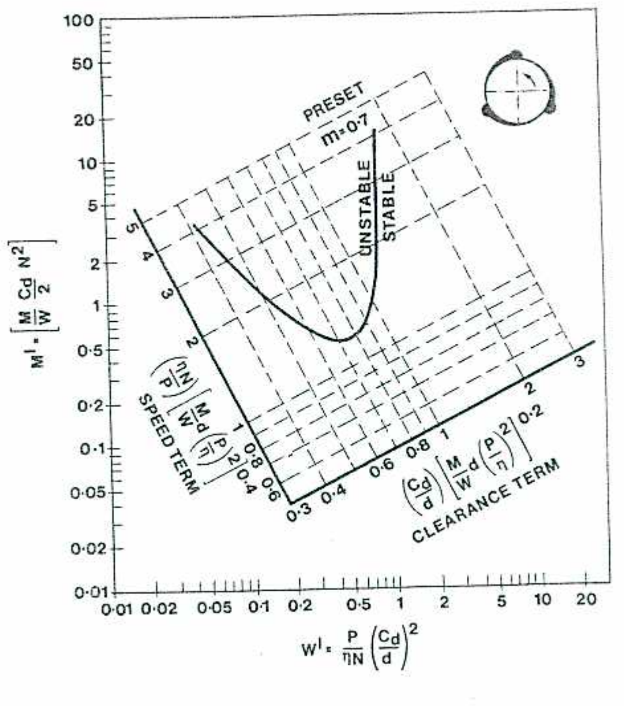


Fig 3 Instability threshold related to unique dimensionless

bearing. For the particular 3-lobe example case, using the same stability curve but now with the new axes, the bearing is seen to be unstable if the speed is increased by 20%.

The variation in instability threshold for different values of preset demonstrates the different characteristics between the 3-lobe bearing with a tilt angle of 20 degrees, Fig 4a, and the offset halves bearing, Fig 4b. In general, the higher the preset, the smaller is the region of instability, although in the case of the tilted 3-lobe bearing at low values of the clearance term, the effect of preset is small. The region of instability for the offset halves bearing at a preset of 0.5 is significantly larger than the region of instability at a preset of 0.6, Fig 4b. This difference is important when considering the effect of manufacturing tolerances and how small changes in clearance may affect bearing stability.

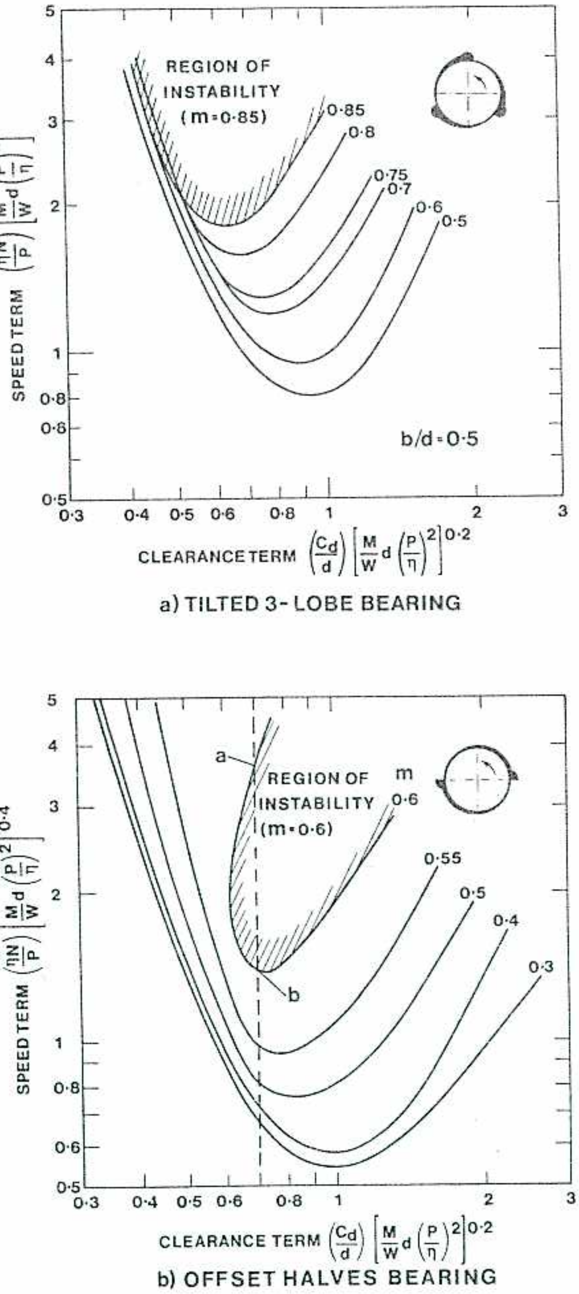


Fig 4 Instability threshold curves (dimensionless)

5 INSTABILITY THRESHOLD SPEEDS FOR CASE STUDY CONDITIONS

Real dimensions must be considered if the effects of manufacturing tolerances on the instability threshold speed of a journal bearing are to be demonstrated. Conditions used in this paper and which are typical of a profile bore type application are listed in Table 1. A heat balance and the effect of turbulence in the oil film (4) were included in the analysis to predict the operating viscosity for each case considered.

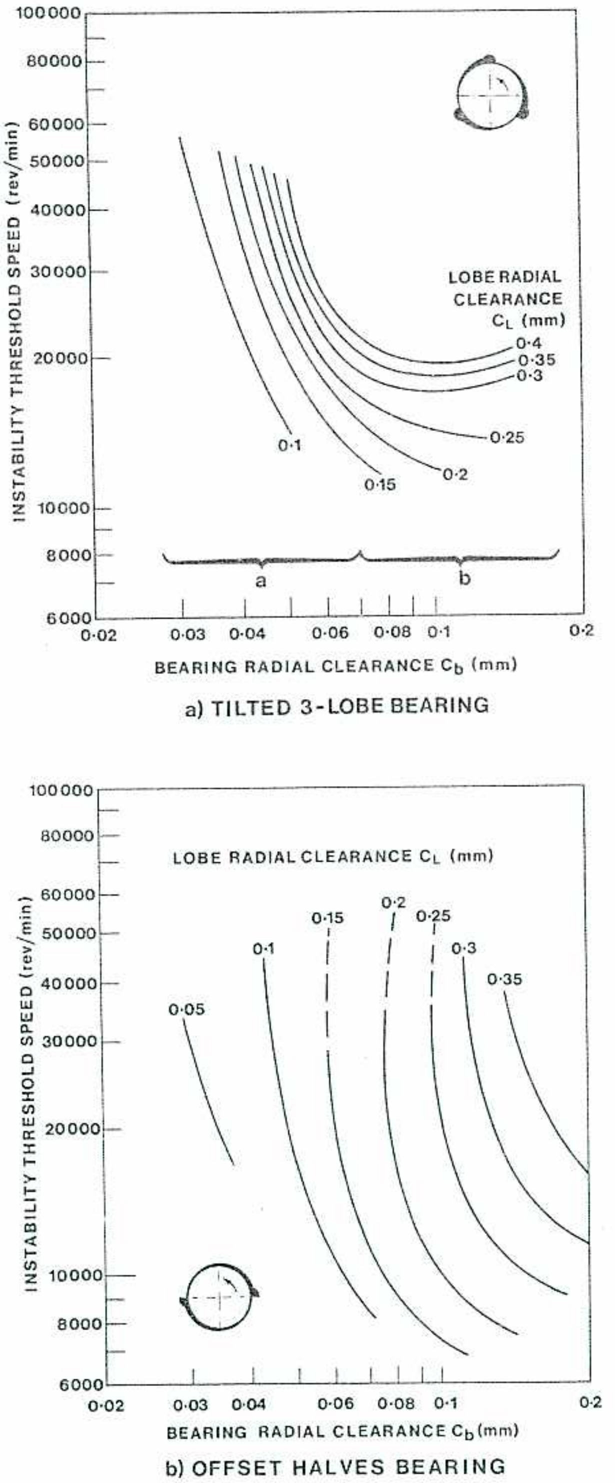


Fig 5 The effect of bearing clearance on instability threshold for various lobe clearances

Table 1 Bearing dimensions and operating conditions

Bearing diameter	=	70 mm
Bearing length	=	35 mm
Diametral bearing clearance		0.07 to 0.3 mm
Specific load P	=	0.5 MPa
Speed range 5000 to 30000 rev/min		
Oil ISO VG 46 Turbine oil		
Oil supply pressure	=	1 bar
Oil supply temperature	=	50°C
Tilted 3-lobe bearing:- (see geometry definitions Fig 1) Tilt angle	=	20 degrees
Feed groove-		
angular extent	=	20 degrees
axial length	=	28 mm
presets m	=	0.5 to 0.85
Offset halves bearing:-		
Feed groove -		
angular extent	=	15 degrees
axial length	=	28 mm
presets m	=	0.3 to 0.6

Manufacturing tolerances affect the clearances in the bearing and it is important to show directly the effect of changes in either clearance, Fig 5. In the case of the 3-lobe bearing, Fig 5a, the instability threshold speed increases with lobe clearance for a given bearing clearance; such a change corresponds to increasing preset. At low bearing clearance (in region 'a' for example) the instability thresholds speed is very sensitive to changes in bearing clearance. The instability threshold speed is less sensitive to changes in large bearing lobe clearances (in region b for example) especially for large lobe clearances.

The offset halves bearing, Fig 5b, shows rather different characteristics. Instability threshold speeds are sensitive to small changes in bearing clearance, as shown by the wide range of threshold speeds for a given lobe clearance.

6 EFFECT OF MANUFACTURING TOLERANCES

The tolerances that affect the bearing and lobe clearances can be separated into two components; the tolerance on the shaft diameter and the tolerances on the bearing itself. Shaft diameter tolerance is normally Grade IT6 giving a tolerance of 0.019 mm, on a 70 mm shaft diameter. Typically, shaft tolerance will acount for nearly 50% of the tolerance on the bearing clearance. The bearing tolerances depend upon the type of bearing, eg thickwalled or thinwalled, the profile and the manufacturing method.

The range of possible bearing and lobe clearances for one particular thickwalled bearing case may be plotted as shown in Fig 6 to form a 'tolerance box'. This figure uses a framework of bearing clearance, lobe clearance and preset.

Minimum bearing clearance, C_b, requires the shaft to be at maximum diameter, whereas maximum lobe clearance, C_L, requires the journal to be at minimum diameter. One cannot have both a maximum and minimum shaft diameter at the same time for a specific case. Hence the tolerance box cannot extend to all four corners bounded by maximum and minimum clearances. This is shown in Fig 6 by the regions dimensioned δ in the tolerance box, where δ is the tolerance on the shaft radius.

It can be seen in Fig 6 that preset is represented by a series of straight lines passing through the origin of the axes. When the lobe clearance is equal to the bearing clearance the preset is zero (line at 45 degrees). The preset will vary over the tolerance box and a single design value, at say minimum clearances (bottom left hand corner of the tolerance box) is only one of a range of possible values.

The effect of manufacturing tolerances on the instability threshold speed for the study case bearing can be demonstrated by plotting tolerance boxes onto a map of threshold speeds considering a wide range of bearing and lobe clearances; Figs 7 and 8. The main point of this paper is brought out in these two figures showing how both bearing and lobe clearance affect the instability threshold speed. The figures relating to a tilted 3-lobe and offset halves bearing give quite different trends for the two bearing types.

These 'maps' also form the background for investigating the behaviour within tolerance boxes and one can focus down within a typical box, related to actual geometry and dimensions, placed at any position on the map. In practice any instability threshold speed shown within a particular tolerance box has a chance of being the threshold speed, depending on actual manufacturing size, above which the bearing system would be unstable. The spread of results can be quite considerable as shown within the tolerance boxes on Figs 7 and 8 and also in Table 2. In actual bearings the clearances are affected by both dimensional tolerances and thermal distortion and therefore there may be considerable scatter in the observed instability onset speed when compared to the theoretically predicted value.

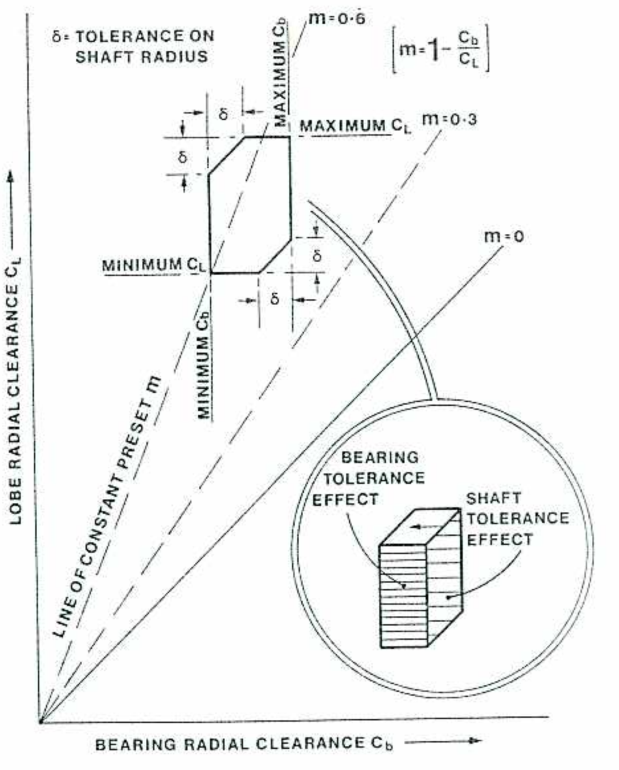


Fig 6 The effect of manufacturing tolerances on the clearances in a thickwalled multi-lobed journal bearing

Table 2 Percentage change in instability threshold speed due to tolerance effects

BEARING TYPE	TOLERANCE BOX	$\left[\frac{\text{LOWER THRESHOLD}}{\text{THRESHOLD@MinC}_L\text{\&C}_b}\right]$		$\left[\frac{\text{UPPER THRESHOLD}}{\text{THRESHOLD@MinC}_L\text{\&C}_b}\right]$	
TILTED 3-LOBE	A	17600/17900	(98%)	19500/17900	(109%)
	B	14000/14000	(100%)	18200/14000	(130%)
	C	13500/20000	(68%)	26000/20000	(130%)
OFFSET HALVES	D(CNC)	12600/13900	(91%)	17700/13900	(127%)
	D(conv)	11500/13900	(83%)	19500/13900	(140%)
	E(CNC)	9750/17500	(56%)	Stable/17500	(-)
	E(conv)	8000/17500	(46%)	Stable/17500	(-)

6.1 Tilted 3-Lobe Bearing

Real tolerance boxes for the thickwalled 3-lobe bearing manufactured on a computer controlled boring machine are shown in Fig 7, assuming an IT6 tolerance on the 70 mm shaft. Three tolerance boxes, with very different characteristics within each box were chosen to illustrate the likely range of possible instability threshold speeds at three nominal bearing clearance conditions and different values of preset.

It can be seen from Fig 7 that the effect of manufacturing tolerance on the instability threshold speeds, for the particular operating conditions considered, depends very much upon the location of the tolerance box and its size. Box A shows an increase in the lobe clearance tolerance range (height of box) due to the large lobe radius considered. In box A, the minimum instability threshold speed is approximately 17600 rev/min, increasing to 19500 rev/min at maximum preset. In this case, the effects of manufacturing tolerances are moderate.

In general, increasing the bearing clearance and allowing the preset to decrease (ie from left to right in Fig 7) reduces the instability threshold speeds. However, this is not always the case, as shown by box B, where the threshold speed may increase with bearing radial clearance. In this tolerance box the maximum threshold speed is approximately 18200 rev/min and this does not occur at maximum preset but at the maximum bearing clearance condition. More important, the lowest possible instability threshold speed has been reduced to 14000 rev/min at minimum bearing clearance.

At low values of both clearances, tolerance box C, there is a considerable range of threshold speeds across the box, with a maximum of 26000 rev/min at maximum preset, reducing to 13500 rev/min at minimum preset. Tolerance box C has the largest range of presets for the three boxes shown and the significant variation in instability threshold speed within tolerances box C demonstrates clearly how necessary it is to consider manufacturing tolerances when investigating the stability of profile bore journal bearings.

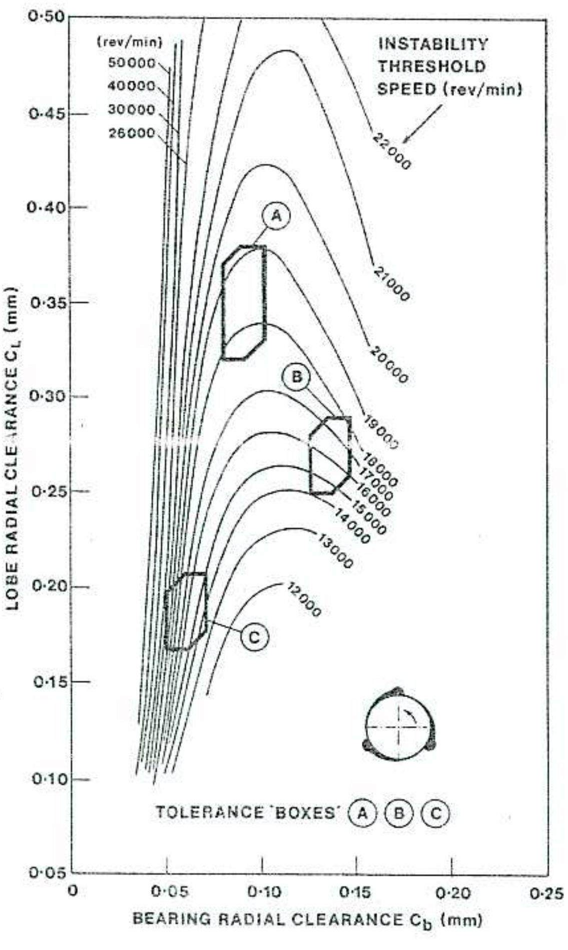
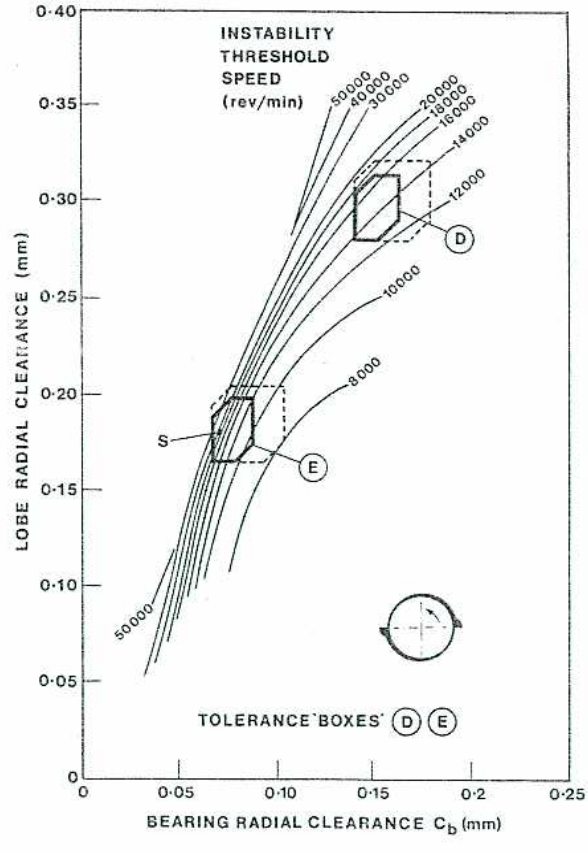


Fig 7 The effect of clearances and manufacturing tolerances on instability threshold speed — tilted 3-lobe bearing



ȝ 8 The effect of clearances and manufacturing tolerances on instability threshold speed — offset halves bearing

6.2 Offset Halves Bearing

The offset halves bearing can be manufactured in a number of different ways and the variation in the size of the tolerance box is significant, particularly for bearings of less than 100 mm diameter. Conventional boring of a split bush as two half bearings is one common manufacturing method and the corresponding tolerance boxes for the study case bearings are shown dotted on Fig 8. An improved method of manufacture, using computer controlled profile boring of the complete bush allows tighter dimensional tolerances. Tolerance boxes for this latter method are shown as full lines and the reduced spread of possible clearances, for identical minimum clearance conditions, is clearly visible.

Tolerance box D relates to a comparatively large clearance condition. The computer controlled manufacturing method produces a bearing with a range of instability threshold speed varying from 12600 rev/min to 17700 rev/min. When using conventional machining, this range is expanded from 11500 rev/min to 19500 rev/min.

In the case of tolerance box E the minimum instability threshold speed is increased from 8000 rev/min to 9750 rev/min by improving the manufacturing method. The maximum threshold speed, corresponding to the maximum preset condition at minimum bearing clearance, is complicated by the shape of the region of instability for a preset of 0.6, (curve b to a Fig 4b). With reference to Fig 8, the instability threshold speed line of 50000 rev/min 'folds' back over tolerance box E, and passes through point S. At the maximum preset condition, top left hand corner of box E, the bearing is inherently stable, and no threshold speed exists. At clearance conditions to give a preset of approximately 0.6, there exists a higher speed, above the instability threshold speed, at which the bearing would be predicted as being stable by the linear analysis.

Comparing Figs 7 and 8, see Table 2, it can be seen that it is possible to design a 3-lobe bearing for this application with a comparatively small spread in threshold speeds caused by manufacturing tolerances. The threshold speed lines are more tightly grouped for the offset halves bearing and manufacturing tolerance will have a significant effect whatever the values of clearance chosen.

7 CONCLUDING REMARKS

A new set of dimensionless groups are introduced, a speed term independent of clearance and a clearance term independent of speed. These new terms have more potential to indicate trends in the instability threshold speed than the conventional terms M' and W', especially when changing clearance conditions. Even more realistic trends, however, are attainable when further considering specific cases by carrying out a heat balance and, when applicable, allowing for turbulence in the oil film.

Both bearing clearance and lobe clearance have a significant influence on the instability threshold speed in profile bore bearings. This is illustrated in Figs 7 and 8 for the tilted 3-lobed bearing and an offset halves bearing. The general trends of instability threshold for these two types of bearing geometry differ considerably.

The importance of considering manufacturing tolerances is illustrated by the fact that the spread of possible instability threshold speeds within a 'tolerance box' can be very large (depending on shape, size and position of the tolerance boxes in Figs 7 and 8). In general, tighter bearing clearances tend to result in higher instability threshold speeds. However, with tighter bearing clearances one must be cautious, making sure that the bearing does not overheat or that there is no chance of sudden loss of the bearing clearance due to transient thermal gradients during rapid start-up from cold.

The tolerance boxes considering computer controlled profile boring machines, are smaller than the tolerance boxes considering conventional manufacturing methods, thus giving a smaller spread in the instability threshold range. A further advantage with this type of manufacture is the repeatability of each lobe profile shape and its excellent surface profile.

8 ACKNOWLEDGEMENTS

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