Plain journal bearings under steady loads: design guidance for safe operation


Research and Development Organization, The Glacier Metal Company Ltd.,
Wembley, Middlesex, England.
PLAIN JOURNAL BEARINGS UNDER STEady LOADS: DESIGN GUIDANCE FOR SAFE OPERATION

F. A. MARTIN, CEng, FI MechE
Manager, Design Techniques, Glacier Metal Company Limited,
Ealing Road, Alperton, Wembley, Middlesex.

D. R. GARNER, BSc (Eng), MSc
Research Engineer, Glacier Metal Company Limited,
Ealing Road, Alperton, Wembley, Middlesex.

The MS of this paper was received at the Institution on 31 October 1972 and accepted for publication on 7 May 1973.

SYNOPSIS

Bearing performance is dependent upon operating within certain limits which, when exceeded, may result in varying degrees of unavailability. Two limits of safe operation which are important when considering the load carrying capacity of a journal bearing are attributable to: a) thin oil films (low speed operation), b) high temperatures—bearing material and oil (high speed operation). A bearing may be further derated within these limits by adverse conditions such as misalignment of the shaft. Graphical aids are presented which incorporate all these factors, enabling designers to assess quickly the suitability of their design. Guidance is also given on oil flow requirements and power loss. The information presented is based on theoretical and service knowledge.

INTRODUCTION

1. Nearly four decades ago Taterinoff (Ref. 1) presented design charts giving safe loads for journal bearings with lengths ranging from 1.5 to 3 times the bearing diameter. The general trend over the years is to design with shorter bearings (now typically with length/diameter from 0.25 to 1.0) and this has been accomplished by the advance in knowledge of bearing behaviour, enabling reliable designs to be made with smaller margins of safety. However, the designer still has difficulty combining his own practical 'feel' for the problem with the often highly mathematical procedures which are currently being made available.

2. This can lead him to use a simplified approach to everyday bearing design, with only the occasional problem justifying a more sophisticated hand or computer solution. The danger of an over-simplified approach is that the designer may extrapolate his rule-of-thumb experience beyond that which is practicable. The difficulty in most full design procedures is the need to unravel dimensionless groups, usually involving an operating viscosity term. The determination of the viscosity should be based upon a heat balance, equating work done to heat removed. This reiterative procedure is long and tedious if calculated by hand (Refs. 2, 3) with only slight improvement using a graphical approach (Ref. 4). Many authors omit this heat balance and assume that trends in dimensionless groups portray trends in actual performance. This spurious ignores the interaction between some parameters; for example an increased clearance may apparently reduce film thickness if the viscosity is artificially held constant, but in practice the operating viscosity may increase, possibly reversing the trend.

3. Any calculations, however, are only a means to an end, the designer primarily being concerned with whether his bearing will operate satisfactorily; will the minimum film thickness be adequate and the bearing temperature acceptable? General guidance on the limiting conditions of safe operation is therefore required and some method must be devised to show how these limits of operation are affected by the many design variables involved.

DESIGN LIMITS:

- a) thin oil films (danger of metal to metal contact)
- b) high bearing temperatures (danger of material softening)
- c) high temperature of oil leaving bearing (danger of excessive oil oxidation)

* Limit for Satisfactory Bearing Operation under Hydrodynamic Conditions, which make for a reliable design; some of these are illustrated diagrammatically in Fig. 1. At low speed and high load conditions (line a) there is a danger of metal contact due to small oil films in the bearing. At high speed and high load (line b) there is a danger of oil film breakdown. At high speed and low load (line c) the relatively high temperature of the mineral oil leaving the bearing may possibly result in a high bulk temperature in the lubrication system and cause excessive oxidation of the oil. All these lines bound a region of safe operation. It is advisable to work well within such limiting lines especially where there is a chance of having two modes of failure at the same time (at the apex of lines 'a' and 'b').

* The additional limitation of oil film instability is usually associated with high speed, high load applications, has not been quantified in this paper.

Fig. 1

JOURNAL SPEED

BEARING LOAD

Region of safe operation:

(a) (b) (c)

© I Mech E 1974
Ensure that the design point of speed and load lies below the appropriate guidance curve, otherwise OIL FILM MAY BE INADEQUATE.

Fig. 6(a) Load Capacity Slide Chart: Small Oil Film Limit (transparency).
This chart may not be applicable at high speeds if non-turbulent conditions prevail (see FIG. 4).

Ensure that the design point of speed and load lies to the left of the appropriate guidance curve, otherwise full lines - there is a danger of EXCESSIVE OIL OXIDATION.
Broken lines - the BEARING MATERIAL may be TOO HOT.

**Fig. 6(b) Load Capacity Slide Chart: High Temperature Limits (transparency)**
Place transparency on this sheet such that appropriate DIAMETER and OIL GRADE are coincident with this point.

2 axial groove journal bearing
 groove length 0.8 of bearing length
 groove width 0.25 of bearing diameter

Oil feed conditions at bearing
0.1 MN/m² (= 15 lbf/in²) & 50°C

Bearing clearances as shown in Fig. 4.

Fig. 6(c) Load Capacity Slide Chart: Load-Speed grid (backing sheet).
of the chart in question, Fig.6(a) or (b), should be placed on the backing sheet, Fig.6(c), using the guide lines on the top (transparent) sheet to keep the two sheets square. The transparency should then be moved to a position where a point in the grid, defined by the diameter and oil grade values, is coincident with the cross on the backing sheet. The relevant length/diameter line then shows the limit of load against speed for the conditions considered.

21. By using the two transparencies the relative position of a design point (defined by its speed and load values) to the limiting lines can now be seen. If this point is within the limits then the bearing, under reasonable environmental conditions, should operate satisfactorily. If it is outside the limits then the slide charts provide a quick method of determining which variable or variables can be changed to improve the design. For instance increasing the bearing length will raise the limit line for film thickness, but may worsen a high temperature situation. The latter effect was noted experimentally by Brown & Neuman (Ref.7) at these high speeds the charts demonstrate and quantify the advantage of thinner oil.

22. Fig.6(b) assumes a limiting bearing outlet temperature in the range 75-90°C (associated with oil oxidation). If there is good reason to change this limit, or to fix a more precise value, then use may be made of the method of predicting the oil outlet temperature which is given in the section on performance prediction.

23. If misalignment is unavoidable (every effort should be made to reduce it to a minimum) and can be quantified, then allowance for it should be made in the slide charts by considering a correction factor \(\frac{W}{W_0}\) obtained from Fig.7. \(W_0\) represents the load on an aligned journal giving a particular minimum oil film thickness \(h_{min}\), and \(W\) represents the reduced load on the misaligned journal system which gives the same value of minimum oil film thickness, in this case at the end of the bearing. Thus the slide chart Fig.6(a) and (c) giving limiting load conditions for safe operation (based on small oil film conditions) may be used for a misaligned bearing, by displacing the limiting load curves, on the assembled slide chart, by a factor \(\frac{W}{W_0}\). Instead of this procedure it is often more simple, although less accurate, artificially to increase the actual design load (on the backing sheet of the slide chart Fig.6(a)) by the ratio \(\frac{W}{W_0}\), and this revised \(W\) should then lie below the original limiting curves of the aligned case. The misalignment chart (Fig.7) shows the reduction in load carrying capacity for any minimum oil film thickness value. However, when determining the load ratio to be applied to the slide chart the allowable value of minimum film thickness (taken from Fig.5) should be used.

24. Finally, it is advisable, wherever possible, to work away from the actual limiting lines; this allows an extra safety margin for the unintentional adverse conditions such as small misalignment, contaminated oil, etc.

**PERFORMANCE PREDICTION**

25. Having determined that a given bearing is likely to operate satisfactorily, the designer then often needs to know the power loss and oil flow requirement of the bearing. As well as being of interest as far as the overall efficiency of the machine is concerned, these quantities have a direct influence on the lubricant supply system - the size of oil pump and supply lines, the need for coolers etc. This system must be capable of adequately supplying sufficient cooled oil (heat is the tangible form of bearing power loss) for any bearing within the limit of the manufacturing tolerances on clearance. When determining the oil pump capacity, the bearing flow at maximum possible clearances (within the tolerance range) should be considered since this has the maximum flow requirement. Maximum power loss on the other hand, important when considering the heat dissipation from the overall system, can occur anywhere within the tolerance range of clearance. The clearance has therefore been left as a variable in the prediction charts for power loss and oil flow. However, the clearance range should preferably still have a minimum value corresponding with Fig.4, since the slide charts (which inherently consider these clearances) can then be used to check for safe operation prior to predicting bearing performance.

26. Under some critical conditions the need to control the variations in film thickness, power loss or oil flow might require the range of the clearance tolerances, but usually, for a given manufacturing detail the designer may fix a value of clearance which he feels is adequate (see for instance). The size of this tolerance range is mainly a function of economic considerations, and may vary considerably. However, as a rough guide a tolerance on clearance of about \(0.004 \sqrt{d}\) mm (where \(d\) is the nominal shaft diameter in mm) represents an attainable figure for bearings bored 'in situ', but for pre-finished bearings this tolerance may typically be more than doubled.

27. The minimum film thickness ratio \(h_{min}/h_{crit}\) proved to be a significant term when predicting the power loss and oil flow in a bearing. Using this ratio, prediction charts in monograph form were devised which again permitted quick and accurate use with very little calculation. This minimum film thickness ratio may be obtained from Fig.8, for known operating conditions*. Basically these guides are used to define the problem, and by linking these along the appropriate guide lines, as indicated by the arrows on the chart, a point in the fourth, answer, chart is obtained.

28. Power loss and oil flow may then be determined from Figs.9 and 10 respectively, again the method of use is shown on each chart. The power loss given is a bearing requirement flow, and should be increased when determining pump capacity to allow for bearing wear, unequal flow to multi-supplied bearings etc. Typically the figures should be increased by 20-50%, and a pump chosen which can make more than supply this amount.

29. The value of the oil outlet temperature may be estimated from the power loss and oil flow values as read from the charts, using the equation:

\[
\text{outlet temp.} = \text{inlet temp.} + \text{temp. rise} = 50 + \frac{A}{Q} (\text{C})
\]

where \(A = 0.0005\) for \(kW\) and \(Q = m^3/s\)

\(A = 5\) for \(hp\) and \(Q = \text{gal/min}\)

* A useful by-product is the ability to predict minimum film thickness for a bearing which has a different clearance to the minimum value shown in Fig.4, i.e. a bearing which cannot be considered on the slide charts.
To maintain the same minimum film thickness in a misaligned bearing the load must be reduced in the ratio \( \frac{W_m}{W_a} \) found by:

From co-ordinate \( \left( \frac{\delta}{c_r}, \frac{h_{\min}}{c_r} \right) \) follow guide lines and stop at appropriate \( \frac{b}{d} \). At this point read off value of \( \frac{W_m}{W_a} \).

**Fig. 7** The Effect of a Misaligned Journal on Load Carrying Capacity (Developed from Ref.10).
Fig. 8 Prediction of Minimum Oil Film Thickness.

Oil feed conditions at bearing:
0.1 MN/m² (15 lb/in²) and 30°C.
Fig. 9  Prediction of Bearing Power Loss.

Fig. 10  Prediction of Bearing Oil Flow Requirement.
CONCLUSIONS

30. Design information has been presented in such a manner that the effect of change in operating conditions on the suitability of a bearing design may be quickly and easily determined. Advice is given on the likely limiting conditions for safe bearing operation based on service experience of the two axial groove bearings considered. The framework of acceptability given can be moulded on the basis of any more pertinent operational experience which is thought applicable.

31. Throughout, the approach has been to allow the designer to consider directly the real operating parameters (e.g., load, speed, oil grade) so that he need not be concerned with the much more quoted dimensionless terms. The need for extensive numerical calculation has been obviated.

ACKNOWLEDGEMENTS

The authors wish to thank the Directors of The Glacier Metal Company Limited for permission to publish this paper. They would also like to record their appreciation of the help received from other members of the staff, in particular the assistance of Mr. K. Rolls in preparing the design guidance charts.

They are grateful for the permission given by the Director General Ships, Ministry of Defence to use unpublished experimental data from high speed bearing tests carried out at the Admiralty Engineering Laboratories.

The illustrations representing the load capacity 'slide chart' (fig.6), together with Figs 8, 9 and 10 are the copyright of The Glacier Metal Co. Ltd.

REFERENCES


DISCUSSION

T. I. Foulke, Fellow

The authors are to be congratulated for their extremely interesting and useful paper. However, would it be possible to incorporate into the charts Professor Vogelpohl's extremely practical criterion that one must cool excessive wear during starting and stopping 'touch down' would not occur at journal surface speeds greater than about 1 m/s?

J. K. Patrick, Member

The paper by Martin and Garner is of considerable interest as it presents a method of bearing design with less calculation than ESOI 60523 which we have been using for teaching purposes at Strathclyde for the last six years.

Considerable attention is given in 60523 to the establishment of the actual oil flow in the bearing and to the flooded/starved condition as defined. In practice we have found that the majority of bearings run in the 'starved' condition with resulting increases in effective and maximum temperatures. (In fact, 60523 Fig.1 considers the establishment of the flooded/starved condition as a secondary effect where in most cases it is a primary effect and should be included in the calculation of effective viscosity from the start.)

I would like to ask the authors if, i) in preparing Fig.1 they considered the different conditions of each case and calculated the temperature accordingly or did they assume the existence of the theoretical fluid $Q_0$ (as defined in 60523).

One naturally compares a new approach with the established method and in this case I was impressed with the ease with which one can use the charts to produce a bearing design, although they do not apply to sample oils 3 and 4 of 60523.

Using one of our test rig bearings for comparison I found that the author's method does not indicate the effect of shaft slope in the journal Fig.7, until the procedure is well advanced. In many cases the diameter of the shaft in a given set-up is not controlled by maximum stress deflection but by the maximum tolerable slope in the bearing. 60523 indicates this situation clearly at the approximate design stage, perhaps the authors could guide the user towards the early selection of suitable shaft diameters to minimise the derating effect of misalignment.

Completing the design procedure the bearing considered ran in a 'starved' condition as defined in 60523. This gives a running condition with lower flow and correspondingly higher outlet temperature, friction, horse-power, and eccentricity ratio. Do the authors consider that the starved condition indicated by 60523 is not significant as they base their own design on the flooded condition at all times?

In paragraph 28 we are advised to virtually double the theoretical flow when selecting pump capacity. This excess oil must flow from the thick film of the bearing and unless it can be encouraged to cool the housing it can have little effect on the effective and maximum bearing temperature although it will reduce the drain temperature. In a large system, can the increased cost of pumps, piping, tanks and coolers, be justified by such a generous provision of lubricating oil capacity?

In using Fig.8 to establish eccentricity ratio, should the unit bearing load be based on the actual load on the bearing or the load the bearing would carry without misalignment?

AUTHORS' REPLIES

F. A. Martin and D. R. Garner

The authors would like to thank Mr Foulke for introducing the important concept of 'start up' and 'run down' operation of journal bearings. There is no doubt that this is an area where design information is badly required. Mr Foulke quotes Vogelpohl's limit of a maximum peripheral speed of 2 m/s at 'touch down'. Unfortunately it is very difficult to quantify 'touch down'.

To this can be done by checking the calculated hydrodynamic film thickness against some limiting value. The hydrodynamic film thickness for any speed and load can be quickly estimated using Fig.8 in the paper, but the limiting 'touch down' value is difficult to define. The calculated film thickness which we quote in the paper (Fig.5) is designed to be a safe operating value, and touch down will not occur until a significantly lower film thickness is reached. In fact the 'failure value' given in ref.5 (when the film thickness is equal to the peak to valley height of the surface roughness) is just one third of the allowable (Fig.5) value.

It is open to discussion whether this 'failure value' can be thought of as representing touch down or whether it should be increased, say by a factor of two, to allow for slight unknown misalignment, lubricant contamination etc. Using this limit of twice the 'peak to valley' height we have calculated the permissible specific loading under the start-up/run-down condition of 1 m/s, for a wide range of bearing sizes and lubricant grades. These loads are very sensitive to change in length/diameter ratio and lubricant grade, and for practical conditions, vary from below 0.5 to above 9 kN/m² (75-750 lb/in²); the higher end of this range of start-up loads is generally acceptable in current practice. The basis of this criterion is not fully understood by the Authors. It may be that other variables have to be taken into account and this requires further study.

In reply to Dr Patrick, when preparing Fig.3 (and indeed all the other figures which depend on a heat balance) we have considered the appropriate 'starved' or 'flooded' conditions (as defined in ESOI item 60523). We concurred that for many practical applications the supply of fresh cool lubricant to the bearing is not sufficient to remove all the hot 'carry over' lubricant at the feed grooves; this so-called 'starved' condition can raise the overall temperature of the lubricant considerably. The term 'starved' may be misleading in this context, since it is often used elsewhere to denote an operating condition in which the lubricant film is situated in the loaded region (ref.17). The effective temperature calculated by using the appropriate flow has then been used to predict the maximum bearing metal temperature. The inset graph in Fig.3 is certainly not in a form suitable for the designer but from it we have now produced a chart Fig. B, which is quick and easy to use. The method of use is illustrated at the bottom of the figure, and simply involves moving around the various grids; both the effective and the maximum temperature can be found. The chart covers the same range of variables as the design aids given in the paper and goes some way towards completing the set.

Comment was made that sample oils 3 and 4 of 60523 cannot be considered in our paper. Oil 4 has a viscosity of about 1.5 5000 cP at an ambient temperature of 20°C and would have to be preheated for easy handling. It is not very representative of the more commonly used industrial mineral oils.
and we therefore intentionally did not include it. Oil 3 (75 cSt at 60°C) corresponds to an SAE 50 oil and has been included in the design charts. However we consider that this grade of oil would only be used for large bearings and in the slide charts it has been included only for bearing diameter greater than 250mm. Fig. D2 illustrates the temperature/viscosity characteristics which we have considered; the normal range of turbine oils can be seen to have much lower viscosities than both of the above mentioned oils.

![Viscosity-Temperature Characteristics](image)

**FIG D2** Viscosity-Temperature characteristics for the range of oils considered in the design charts.

When considering safe operating limits for misaligned bearing it is obviously advantageous to be able to start from aligned bearing design procedures. Thus, using the given applied load, an aligned film thickness can be determined. If the bearing is now misaligned the load capacity will be reduced and it is convenient to picture this in one of two ways:

(i) the load may be (artificially) reduced to maintain the same minimum film thickness. This is the method adopted if Fig.7, which is used in conjunction with Fig.6. Since the slide chart automatically uses a safe minimum film thickness, the modified loads obtained from Fig.7 will also be safe.

(ii) the minimum film thickness may be reduced and the applied load left unchanged (as used in ESDU 66023). We have now produced a chart enabling this reduction from the aligned values of Fig.6 to be estimated. This chart, Fig. D3 is based on hydrodynamic considerations, unlike ESDU 66023, which merely considers the geometry of the change from the aligned condition. The reduced film thickness must be checked against the allowable values of Fig.5.

The power loss, oil flow and maximum temperature prediction charts are strictly only applicable to the aligned case. As an approximation they may be used for the misaligned case, but using the minimum film thickness ratio for the aligned condition.

![Derating Effect of Misalignment](image)

**FIG D3** The derating effect of misalignment on journal bearing film thickness.

We would agree with Dr Patrick that a doubling of the bearing oil flow requirement cannot be justified, and we are not suggesting this. However, differences between theoretical and measured oil flow do exist, as indicated by considerable experimental evidence, and for this reason we recommend the safety factor of 20-50% when sizing pumps, coolers etc. We are not suggesting that a bearing with a known (measured) lubricant requirement should be deliberately over-sponsored.

References

Ref T7 - 'Tribology Handbook'
(Section 7 - Grease, wick and drip fed journal bearings)