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# Turbulent Journal Bearings: Design Charts for Performance Prediction

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*Under conditions of high sliding speed the lubricant film in hydrodynamic bearings becomes turbulent, and large errors can occur if analysis and design are then based on the commonly used laminar theories.*

*As an extension to one such set of published laminar design charts, information is presented here which allows the film thickness, power loss, and oil flow requirement to be obtained for cylindrical bore journal bearings operating in the turbulent regime. A wide range of conditions is covered, and mathematical manipulation is kept to a minimum.*

*A new turbulent variable, a function of the normal Reynolds and Sommerfeld numbers, is used throughout the charts as a basic parameter; this has the advantage of not being a function of the (unknown) operating viscosity, and is, therefore, readily calculable for any given conditions.*

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## INTRODUCTION

The current trend in the design and use of rotating machinery is for operation at higher and higher sliding speeds, as a result of either a straight increase in rotational speed or the use of larger diameter rotors. Inevitably, this continual uprating introduces its own problems, since scaling up of existing designs is not always possible. The two main problems encountered concern operation with turbulent oil films and the stability of oil lubricated bearings. This latter difficulty can usually be overcome with the use of lobed or tilting pad bearings, but there are still numerous cases where it is advantageous to use a normal cylindrical bore journal bearing in the turbulent regime. In itself, this presents no difficulty, but the machinery designer does have the problem of having to predict and examine bearing performance in a regime where little published information is readily to hand. The need for this becomes apparent when it is realized that laminar theory, if applied to turbulent bearings, may

NOMENCLATURE

For any system of consistent units—SI units given

$b$ = bearing length, m
$c_d$ = diametral clearance, m
$c_r$ = radial clearance, m
$d$ = bearing diameter, m
$h_{min}$ = minimum oil film thickness in the bearing, m
$H$ = power loss, $W$
$N$ = rotational speed, rev/s
$P$ = specific load $\left(\frac{W}{bd}\right)$, Pa or N/m$^2$
$p_f$ = oil feed pressure, Pa or N/m$^2$
$Q$ = oil flow requirement, m$^3$/s
$W$ = applied load, $N$
$Y$ = dimensionless group (see below)
$\rho$ = oil density, kg/m$^3$
$\eta$ = effective viscosity in the bearing (dynamic), Ns/m$^2$
$\eta_f$ = oil feed viscosity (dynamic), Ns/m$^2$

DIMENSIONLESS GROUPS

a) Turbulence Variable

$$Y = \frac{2}{\pi} Re\, S = \frac{d^2N^2\rho}{P}\left(\frac{d}{c_d}\right)$$

where $Re$ = Reynolds number = $\frac{Uc_r\rho}{\eta}$ and U is the surface velocity

$$S = \text{Sommerfeld number} = \frac{\eta N}{P}\left(\frac{d}{c_d}\right)^2$$

b) Power loss variable

$$\frac{H}{WNb}\left(\frac{d}{c_d}\right)$$

c) Oil flow variable

$$\frac{Q}{bc_d dN}$$

d) Oil feed variable

$$\frac{p_f}{\eta_f N}\left(\frac{c_d}{d}\right)^2$$

BEARING GEOMETRY

Cylindrical Bore

Two axial grooves at ±90 degrees to the load line
axial length of groove 0.8 of the bearing length
circumferential width of groove 0.25 of the bearing diameter (i.e. approximately 30 degrees angular extent).

predict only half of the actual power loss. While a great deal of work has been done on turbulent lubrication, the majority of published material is concerned with the basic mathematics of turbulence, and thus is of little immediate help to the machine designer. The purpose of this paper is to present design information for cylindrical journal bearings operating under turbulent conditions, in a form which is readily applicable to everyday design, and without the need for anything but the simplest of mathematics.

## TURBULENT LUBRICATION THEORY

The basic analysis of fluid film bearings is, of necessity, a complex mathematical procedure, involving, in its simplest form, a numerical solution of the governing Reynolds equation. For laminar conditions, this is reasonably well defined, but still involves the usual simplifying assumptions of isothermal conditions, Newtonian lubricants, rigid (circular) surfaces etc. As soon as the film conditions become turbulent, however, a major modification to the Reynolds equation is required, to take account of the local small scale motions of the oil flow within the clearance space. Several methods of allowing for this have been proposed, usually involving a modified Reynolds equation (*1*) (*2*), but sometimes as a modification to the dimensionless data obtained from a normal laminar solution (*3*). Taylor (*4*) has compared several of these basic methods, and, on the basis of the results which he presented, and because of the ease of computation, it was decided here to use the method of Ng and Pan. Now, it is accepted that this does not give a complete answer to the problem of turbulent lubrication, in particular no account is taken of the effects of inertia of the oil film and of the transition region between laminar/turbulent conditions; both these aspects are currently receiving attention (*5*), (*6*), (*7*). However, easy-to-use design information is needed, and until these further refinements are fully developed and proved, the reasonably accurate turbulence model of Ng and Pan enables such information to be produced.

Ng and Pan's theory introduces into the Reynolds equation axial and circumferential flow factors which are functions of the local Reynolds number. The resulting Reynolds equation has been solved using finite difference techniques to produce dimensionless performance data. These results are exactly equal to the laminar case at a Reynolds number of zero, and depart from laminar for any higher values. Guidance is given on an operating region where nonlaminar effects need to be considered.

Dimensionless data is often presented in graphical form as design aids, but, before it can be applied to actual problems, some form of heat balance procedure must be employed, equating the heat generated by the shearing of the film to that carried away in the oil; this is complicated by the nonlinear relationship between the viscosity of a mineral oil (or even a synthetic lubricant) and its temperature. Such a procedure is conceptually quite simple, but, in practice, is very tedious unless programmed for use on a computer. A particularly comprehensive method relating to pressure-fed journal bearings operating in the laminar regime was published by the Engineering Sciences Data Unit (*8*). Again, this involves a certain amount of mathematical dexterity if used in its "raw" form, but Martin and Garner (*9*) presented some of the data relating to specific but practical bearing configurations in chart form, which all but removed the need for calculation. The turbulent information given here is related to this latter work, and enables higher operating speeds to be considered.

Basically, a computer program was set up to use the dimensionless data calculated on Ng and Pan's theory and to perform the necessary heat balance. In order to cover a reasonable range of conditions about 12,000 cases were computed, varying such parameters as bearing diameter and length, oil grade, clearance, speed, and load. To make the problem more manageable, certain conditions had to be fixed throughout; these mainly involved oil feed conditions and grooving arrangements in the bore. The information presented is applicable only to oil feed pressures of 0.1 MN/m$^2$ (approximately 15 lbf/in$^2$) at a temperature of 50 C. The bearing considered had two axial gutterways, with the proportions given in the nomenclature.

## DESIGN CHARTS

A set of design charts has been presented by Martin and Garner (*9*) for a two-axial groove bearing operating under laminar conditions. One of these charts, that on minimum film thickness prediction, is reproduced here as a "stepping stone" in determining a turbulent oil film thickness. Additionally, charts are given for power loss and oil flow prediction, together with associated information on oil grades, recommended clearances, etc.

### Turbulence Variable

A "new" variable, which is independent of the unknown operating viscosity, is introduced and called the turbulence variable $Y$. This term is a function of Reynolds number and Sommerfeld number as shown in the nomenclature and in its final form can be expressed by

$$Y = \frac{d^2N^2\rho}{P}\left(\frac{d}{c_d}\right)$$

This viscosity independent turbulence variable is used together with an appropriate film thickness ratio to form the framework of the design charts presented here for predicting performance under turbulent conditions. It should be noted that $Y$ is a dimensionless term and should, therefore, be formed from a consistent set of units (e.g., the SI units given in the nomenclature). A calculator for the turbulent variable $Y$ is given in Fig. 1 (corrections for rev/min to rev/s and mm to m etc., are inherent in this chart giving $Y$ in consistent units).

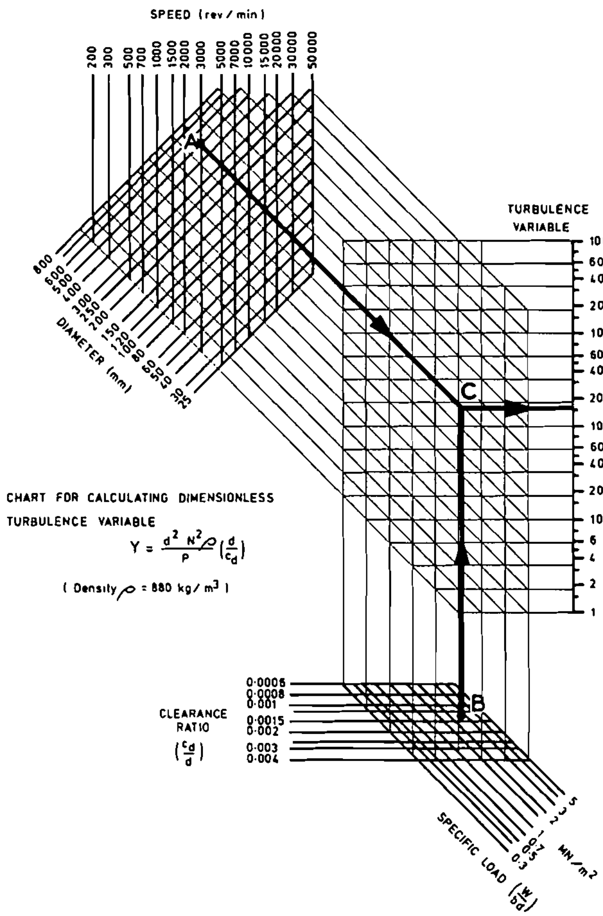


Fig. 1—Calculator for turbulence variable

## Oil Characteristics

The viscosity/temperature characteristics of the range of oils considered are given in Fig. 2 and are classified by the numerical value of viscosity (centistokes) at 60 C. This temperature is not in itself significant, but is merely a method of classifying an oil more precisely than ISO and SAE ranges, etc.

## Laminar or Turbulent?

This paper deals with nonlaminar conditions, and, therefore, in any particular case one must first decide whether such conditions apply. The influencing factors are oil film thickness, Reynolds number, and clearance ratio.

The first of these may be obtained from Fig. 3, a design chart for laminar conditions. Three grids define the problem (at A, B, and C) and the minimum film thickness ratio ($h_{min}/c_r$) is read from the fourth grid (at F) after following the appropriate guide paths. The oil film thickness in the bearing can be found for any clearance ratio, speed, length to diameter ratio, oil, and specific load.

The influence of the Reynolds number is brought in by using the turbulence variable, $Y$, and this, together with the clearance ratio, the length/diameter ratio and the laminar oil film thickness ratio appear in Fig. 4. This figure gives a limit to laminar operation defined by the onset of Taylor Vortex flow, and is based on the work of Frêne (*6*). If nonlaminar conditions are indicated, then the design charts in this paper are applicable; if not then a laminar design procedure Ref. (*9*) can be used.

## Minimum Film Thickness

If Fig. 4 indicates that the bearing will be non-laminar then the laminar oil film thickness (from Fig. 3) must be modified using the slide chart Fig. 5(a) and 5(b). It should be noted that this method is used for expediency. In preparing the data for turbulent conditions, a complete heat balance was performed for each case. No reference was made to a laminar film thickness, which has been incorporated here only as a means to an end in data presentation; even then it was necessary to resort to slide chart techniques to accommodate all the data.

Figure 5(a) and (b) shows the slide chart for obtaining the minimum film thickness ratio under turbulent conditions, and Fig. 6 illustrates how to use it. The use of the charts is explained in more detail in the Appendix, where a numerical example is considered.

The further influence of four variables has had to be allowed for over and above the turbulence variable ($Y$); these are rotational speed, oil grade, length/diameter ratio, and clearance ratio. The first three of these appear in the upper grid of the backing sheet, Fig. 5(b), and enable a point (B) to be marked for any particular case (see Fig. 6). The transparency, Fig. 5(a), is then placed on

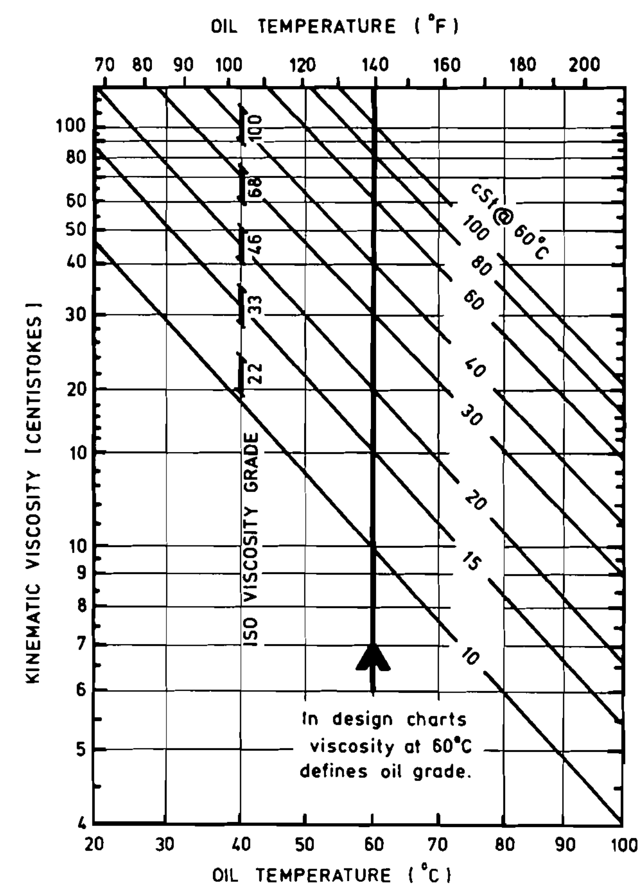


Fig. 2—Viscosity-temperature characteristics for the range of oils considered in the design charts.

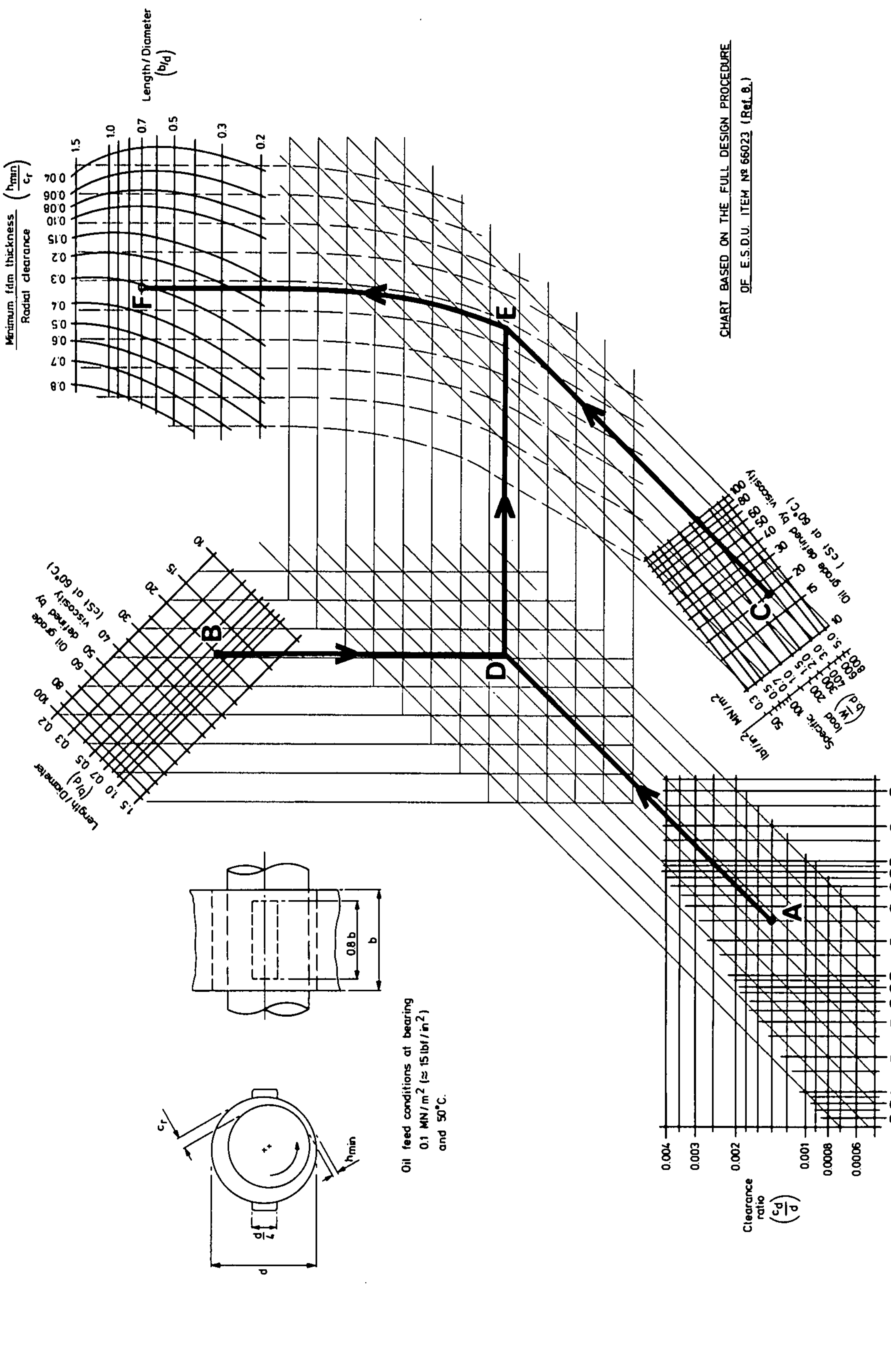


Fig. 3—Prediction of minimum oil film thickness (laminar)

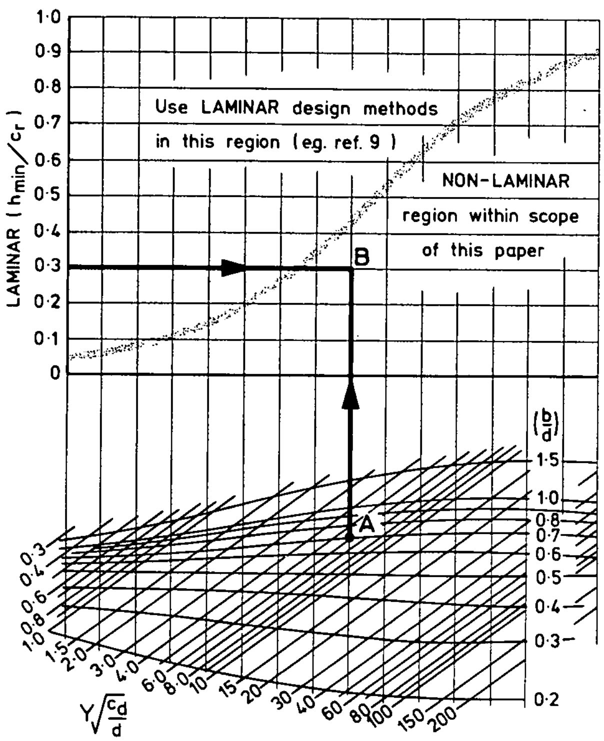


**Fig. 4—Guidance on the choice of laminar or nonlaminar design methods.**

top of the backing sheet with the guide lines superimposed, and slid vertically until the appropriate value of clearance ratio is over the marked point on the backing sheet. The curved lines on the transparency, showing laminar film thickness ratio, then give the relationship between film thickness under turbulent conditions and the turbulence variable. For any particular case, the value of this turbulence variable is calculated (Fig. 1 may be used if required) and a laminar film thickness ratio obtained from Fig. 3. This film thickness, together with the appropriate value of $Y(b/d)^2$ then defines a point (C on Fig. 6). The value of $h_{min}/c_r$ (read off vertically below point C) is that given by turbulent theory.

In high speed turbulent bearings it is unlikely that oil film thickness would present a problem. However, a guide to a safe, allowable, minimum value is shown in Fig. 7, which has been reproduced from Ref. (*9*). This is based on surface finishes produced by good engineering practice, and assumes that the bearing is operating under favorable environmental conditions without gross misalignment.

The charts leave clearance ratio as an independent variable and the designer is free to select an appropriate value. Guidance on minimum values is given in Fig. 8.

The film thickness ratio for turbulent conditions and the turbulence variable $Y$ are important parameters in the power loss and oil flow charts Figs. 9 and 10.

## Power Loss

Turbulent power loss may be obtained from Fig. 9. The value of $h_{min}/c_r$ from Fig. 5 (i.e. that given by turbulent theory), together with the $b/d$ ratio fixes a point on the left hand grid. One then follows the guide lines to intersect a vertical line at $Y\frac{d}{b}$ and the power loss variable is read off on scale B. It is interesting to note that, if the appropriate laminar film thickness is used in the left hand grid, then scale A gives the laminar power loss.

## Oil Flow

The oil flow prediction chart Fig. 10, is slightly more complicated to use, but the example guide paths should illustrate the general method. A dimensionless pressure term associated with oil feed conditions has to be determined. The oil feed pressure in the film thickness charts was 0.1 MN/m$^2$; however, it is not envisaged that feed pressure will make a great difference to film thickness, and while it is strictly only correct to substitute this value into the pressure parameter, it is felt that the flow could be predicted with reasonable accuracy for other oil feed pressures ($p_f$). The feed viscosity $\eta_f$, will depend on the oil used, and for guidance oil feed viscosities are given in Table 1.

## Oil Temperature

It is advisable to check on the bulk oil outlet temperature as the higher the value the faster is the thermal degradation of the oil. "Limiting" values of 75–80 C are recommended in Ref. (*9*) as a reasonably safe guide for normal turbine oils. Higher temperatures can be tolerated if appropriate oxidation resistant oils are used. The outlet temperature can be calculated from the power loss $H$ (watts) and oil flow $Q$(m$^3$/s):

$$\begin{aligned}\text{oil outlet temp} &= \text{inlet temp} + \text{temp rise}\\ &= 50 + \frac{0.5}{10^6}\frac{H}{Q}\,(^\circ\text{C})\end{aligned}$$

Another limit relating to temperature is the maximum bearing metal temperature especially occurring with small $b/d$ ratios. There is little published information on the prediction of bearing temperature under turbulent conditions and it is a field open to further study.

Table 1

| Oil Grade Defined by Viscosity (cSt) @ 60 C | 10 | 20 | 30 | 40 | 60 | 80 | 100 |
|---|---|---|---|---|---|---|---|
| Oil viscosity ($\eta_f$) at 50 C inlet temperature (Ns/m$^2$) | 0.011 | 0.025 | 0.038 | 0.053 | 0.083 | 0.12 | 0.15 |

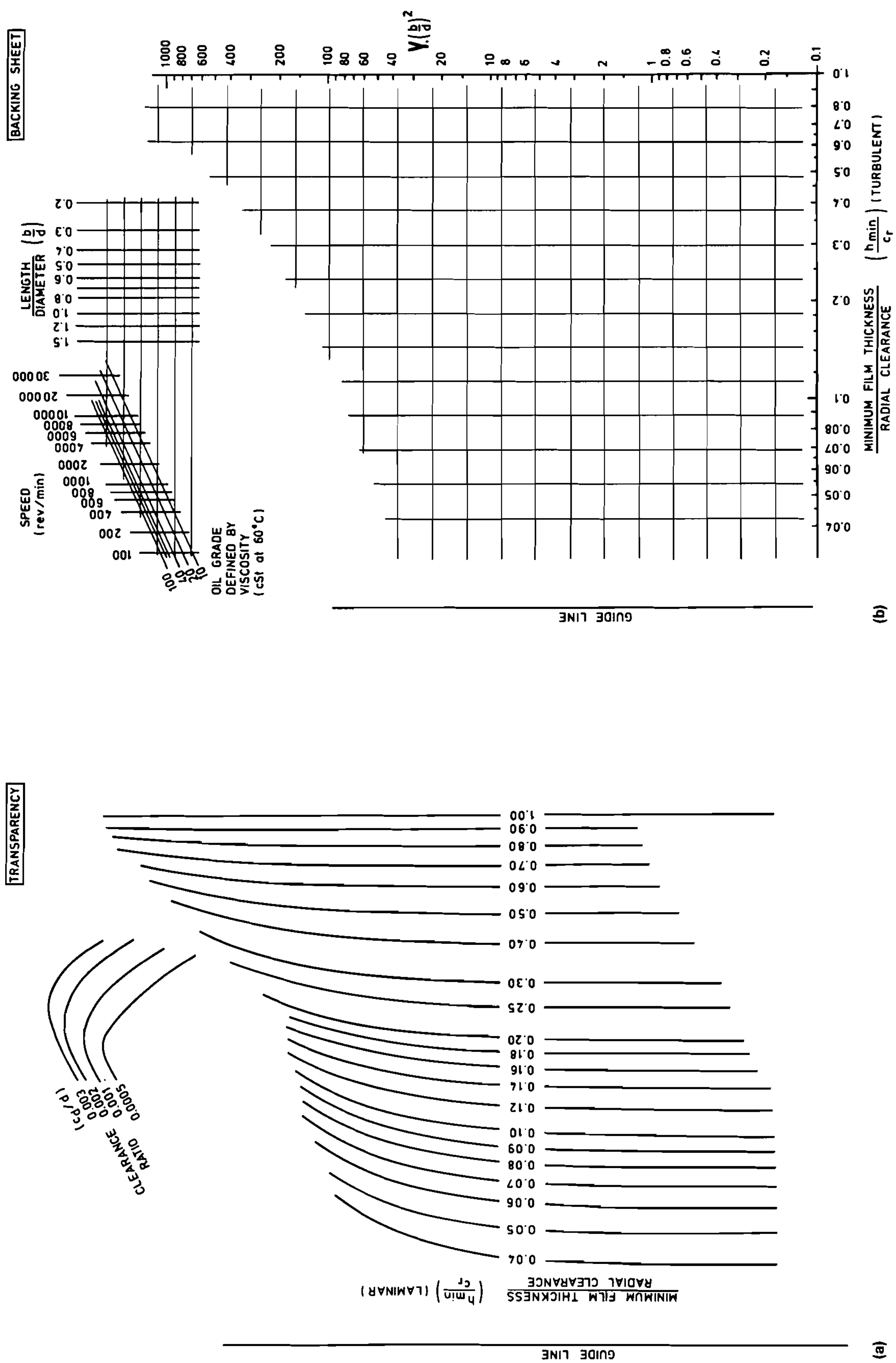


Fig. 5(a) and 5(b)—Prediction of minimum oil film thickness under turbulent conditions. 5(a) transparency—5(b) backing sheet

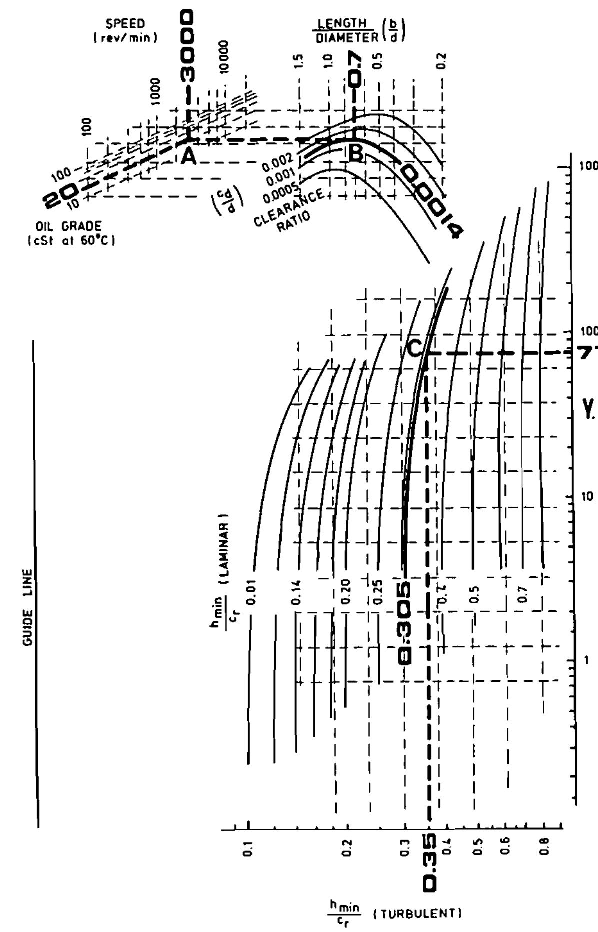


Fig. 6—Method of use of slide chart—transparent overlay [Fig. 5(a)] sliding on backing sheet [Fig. 5(b)].

## CONCLUSIONS

Design charts are presented which enable the performance of cylindrical journal bearings under turbulent conditions of operation to be predicted. Oil film thickness, power loss, oil flow requirements, and bulk outlet temperature can be obtained for a wide range of operating conditions. A turbulence variable $\left(\frac{d^2N^2\rho}{P}\right)\left(\frac{d}{c_d}\right)$ is introduced, which is directly amenable to calculation by the designer (unlike Reynolds number with its unknown

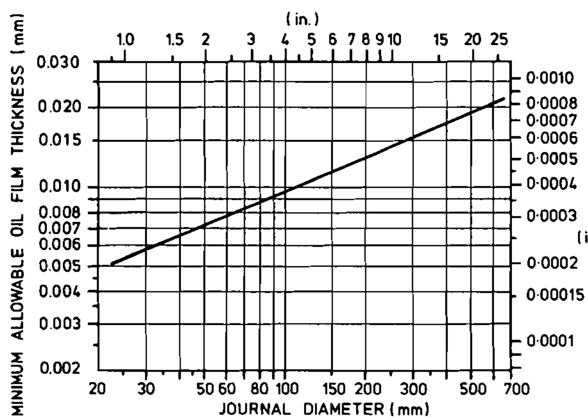


Fig. 7—Minimum allowable oil film thickness

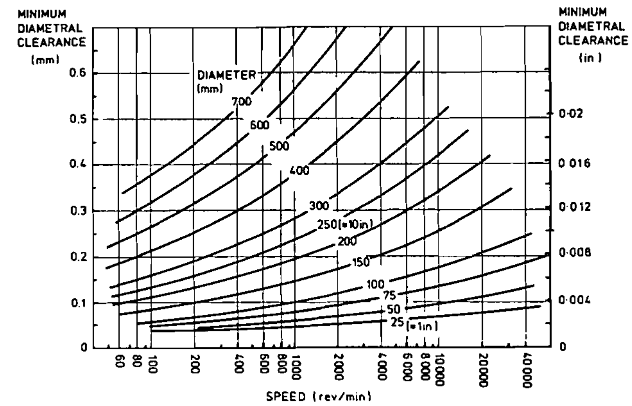


Fig. 8—Guidance on minimum clearance

operating viscosity). This variable was used throughout to form a basic framework for turbulent conditions. The design charts supplement similar charts for laminar conditions, Ref. (*9*).

Two further areas which this paper ignores, but where information is required, are the prediction of maximum bearing temperature when the oil film is turbulent and the effect of turbulence on the dynamic characteristics of the bearing and hence on rotor/bearing stability.

## ACKNOWLEDGMENTS

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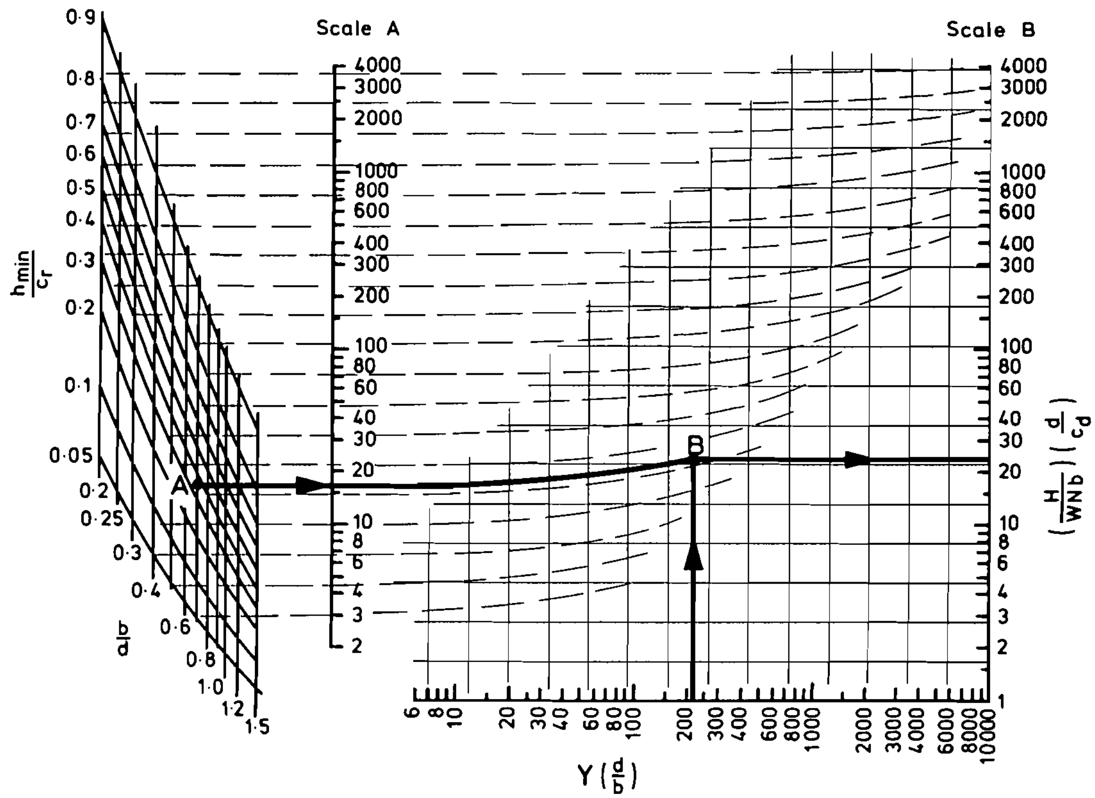


Fig. 9—Prediction of bearing power loss (turbulent)

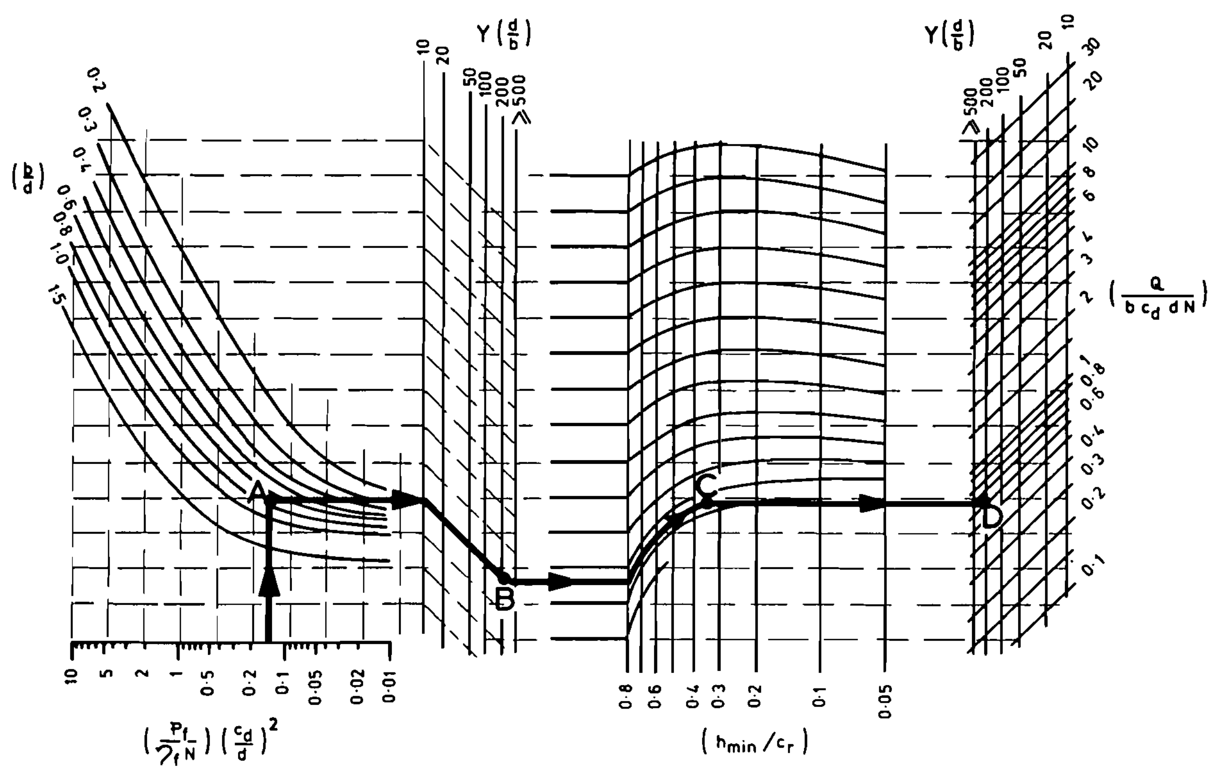


Fig. 10—Prediction of oil flow requirements (turbulent)

Steady Loads: Design Guidance for Safe Operation." First European Tribology Congress. Sept 1973 paper C313/73, The Inst. of Mech. Engrs., London.

## APPENDIX

### EXAMPLE

A gas turbine bearing has a diameter of 500 mm and a length of 350 mm; it has to carry a load of 0.44 MN at 3,000 rev/min. The diametral clearance is 0.7 mm and the bearing has two axial grooves extending 0.8 of the bearing length; the grooves are situated at ±90 degrees to the load line. The oil used is a medium turbine oil. Estimates are needed of the film thickness, power loss and oil flow.

#### Procedure

a) **Turbulence variable $Y$**

This may be calculated directly from $Y = \left(\frac{d^2N^2\rho}{P}\right)\left(\frac{d}{c_d}\right)$ in consistent units or obtained from Fig. 1: (assuming density = 880 kg/m$^3$)

On Fig. 1: Fix point A on grid (diameter 500 mm, speed 3,000 rev/min)
Fix point B on grid ($c_d/d = 0.0014$; $P = 0.44/(.5 \times .35) = 2.5$ MN/m$^2$)
Draw lines from A to B following guide lines to intersect at C
At which point read off a value of $Y$

Turbulence variable $\underline{Y = 157}$

b) **Oil classification** (from Fig. 2)
Medium Turbine Oil = <u>20 cSt at 60 C</u>

c) **Check if laminar or turbulent**
First find laminar film thickness ratio:

On Fig. 3: Fix point A on grid ($c_d/d = 0.0014$; speed 3,000 rev/min)
Fix point B on grid ($b/d = 0.7$; oil 20 cSt at 60 C)
Fix point C on grid $\left(\frac{W}{bd} = 2.5 \text{ MN/m}^2;\right.$ oil 20 cSt at 60 C$\left.\right)$
Follow guide lines (via points of intersection D and E) and read off $h_{min}/c_r$ at point F (at $b/d = 0.7$)

$h_{min}/c_r$ (laminar) = 0.305

This value is required for use in Fig. 4 to check if the bearing is turbulent and will be further used in the slide chart Figs. 5(a) and (b) to find $h_{min}/c_r$ (turbulent).

Calculate $Y\sqrt{\frac{c_d}{d}} = 157 \times \sqrt{0.0014} = 5.87$

On Fig. 4: Fix point A on grid $\left(Y\sqrt{\frac{c_d}{d}} = 5.87;\ b/d = 0.7\right)$
Follow vertical guide line from A to intersect with $h_{min}/c_r$ (laminar) of 0.305 at point B.

Point B is in NON-LAMINAR region and, therefore, the design charts in this paper are applicable to this example.

d) **Oil film thickness**

Calculate $Y\left(\frac{b}{d}\right)^2 = 157 \times 0.7^2 = 77$

From slide chart Fig. 5(a) and (b) (illustrated in Fig. 6 where the grid of the backing sheet is shown dotted) proceed:

Fix point A on grid on backing sheet
(oil 20 cSt at 60 C; speed 3,000 rev/min)
Fix point B on grid of backing sheet
(horizontal line from A; $b/d = 0.7$)

Place transparency [Fig. 5(a)] over backing sheet [Fig. 5(b)] and slide vertically (with "guide lines" superimposed) until clearance ratio "line" of 0.0014 on the transparency lines up with point B. The two sheets are now positioned correctly.

Fix point C by using $Y\left(\frac{b}{d}\right)^2 = 77$ and $h_{min}/c_r$ (laminar) = 0.305
Read off $h_{min}/c_r$ (turbulent) from vertical line through C giving $h_{min}/c_r$ (turbulent) = <u>0.35</u>

($h_{min} = 0.35 \times 0.7/2 = 0.123$ mm and is well above the minimum allowable value given in Fig. 7).

e) **Power Loss**

Calculate $Y\frac{d}{b} = \frac{157}{0.7} = 224$

On Fig. 9: Fix point A on grid ($h_{min}/c_r = 0.35$; $b/d = 0.7$)
Fix point B on grid $\left(\text{curved line from A; } Y\frac{d}{b} = 224\right)$

From B read off $\left(\frac{H}{WNb}\right)\left(\frac{d}{c_d}\right) = 23$

$$H = 23\frac{WNbc_d}{d} = 23 \times 440000 \times 50 \times 0.35 \times 0.0014$$
$$= 248000 \text{ Watts} = \underline{248 \text{ kW}}$$

Note—Using the laminar $h_{min}/c_r$ value of 0.305, the laminar power loss from scale A is 150 kW. This is about 60 percent of the more realistic turbulent value.

f) **Oil Flow**

Calculate value of oil feed parameter

$$\left(\frac{p_f}{\eta_f N}\right)\left(\frac{c_d}{d}\right)^2 = \left(\frac{0.1 \times 10^6}{0.025 \times 50}\right)(0.0014)^2 = 0.157$$

Assuming $p_f = 0.1 \times 10^6$ N/m$^2$
$\eta_f = 0.025$ Ns/m$^2$ (from Table 1)

On Fig. 10: Fix point A on grid $\left(b/d = 0.7;\ \left(\frac{p_f}{\eta_f N}\right)\left(\frac{c_d}{d}\right)^2 = 0.157\right)$
Fix point B on grid $\left(\text{guide lines from A; } Y\frac{d}{b} = 224\right)$

Fix point C on grid (guide lines from B; $h_{min}/c_r = 0.35$)
Fix point D on grid (guide lines from C; $Y\frac{d}{b} = 224$)

Read off $\frac{Q}{bc_a dN} = 0.63$

$$\begin{aligned}\text{Flow } Q &= 0.63\ bc_a dN \\ &= 0.63 \times 0.35 \times 0.0007 \times 0.5 \times 50 \\ &= 0.0039\ \text{m}^3/\text{s}\end{aligned}$$

The bulk oil outlet temperature can be estimated from

$$\begin{aligned}\text{oil outlet temp} &= 50 + \frac{0.5}{10^6}\frac{H}{Q}\ (^\circ\text{C}) \\ &= 50 + \frac{0.5 \times 248000}{10^6 \times 0.0039} = 82\ \text{C}\end{aligned}$$

This appears to be on the high side and an oil with appropriate oxidation resistance would be required.

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## DISCUSSION

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The goal of this paper is certainly commendable as turbulence is an operating condition faced by the bearing and machine designer more and more frequently. A journal bearing design procedure which includes the effects of turbulence, yet does not require the designer to employ a computer, is thus welcome.

The design procedure presented here is limited to cylindrical bore journal bearings with a specific groove configuration in the bore and specific oil inlet conditions. However, these specified conditions are relatively common in bearing design and it is anticipated that minor variances would not significantly affect the operating characteristics.

The value of the procedure presented is, of course, dependent on the degree of accuracy to which it predicts turbulent bearing performance characteristics. No comparisons to test data are provided but such information could give confidence in the use of the procedure.

In this respect, the discussant has taken design specifics from a 482.6 mm (19 inch) cylindrical bore journal bearing and calculated the power loss using the procedure and plots in this paper. This is then compared to the actual losses as determined from test and previously reported (*A1*). Film thickness test data were not obtained so no comparison of this could be made.

The specifics of this particular bearing design and application are listed below.

(a) – b = Bearing Length = 406.4 mm (16 in)
(b) – $C_d$ = Diametral Clearance = 0.7366 mm (0.029 in)
(c) – d = Bearing Diameter = 482.6 mm (19 in)
(d) – p = Bearing Loading = 1.28 MPa (185 psi)
(e) – $p_f$ = Oil Feed Pressure = 0.103 MPa (15 psi)
(f) – $\rho$ = Oil Density = 858 kg/m³ (0.031 lb/in³)
(g) – $T_f$ = Oil Feed Temperture = 49 C (120 F)
(h) – Oil – 14 cST @ 60 C

The axial feed slots in the bore were slightly longer circumferentially and slightly shorter axially than specified in this paper. This was not believed to be significant for this comparison. The power loss was calculated at two shaft speeds, 2,800 rpm and 4,200 rpm, both in the turbulent region (as determined from the test data and confirmed by the authors' Fig. 4).

At 2,800 rpm, the comparison is as follows:

| TEST | CALCULATED |
|---|---|
| 224 kW (300 HP) | 210 kW (282 HP) |

The agreement is excellent, possibly in part the result of some good fortune in using the limited size plots available at the time this comparison was made.

At 4,200 rpm, the comparison is:

| TEST | CALCULATED |
|---|---|
| 701 kW (940 HP) | 488 kW (654 HP) |

The agreement here is rather poor, the reasons not readily apparent. One reason may well be errors in working with the plots, as mentioned. In this respect, possibly the authors could review this condition, checking it against their master charts.

The authors state that the information is applicable only for an oil inlet temperature of 50 C. In this respect, a point of interest from the tests on this specific 482.6 mm (19 in) bearing was that an essentially constant power loss was obtained in a test at 3,600 rpm where the only variable was the oil inlet temperature, this being varied from 38 C (100 F) to 60 C (140 F).

The authors note that oil film thickness is unlikely to be a problem in high speed turbulent bearings. Figure 7 is provided, however, as a guide to a safe, allowable, minimum film thickness depending on journal diameter, and assuming favorable environmental conditions. An additional emphasis on this note of caution with respect to this plot may be in order, particularly for the generally high speed bearings considered here. This is, the bearing operating temperature may well be the limiting factor before these limiting values of film thickness are reached.

## REFERENCE

(*A1*) Gardner, W. W. and Ulschmid, J. G., "Turbulence Effects in Two Journal Bearing Applications," *ASME Transactions, Journal of Lubrication Technology*, Jan., 15-21 (1974).

## AUTHORS' CLOSURE

The authors are grateful to Mr. Gardner for his comments, in particular for the reference to the 483 mm (19 in) diameter bearing and associated test data. It is pleasing to know that the discussant has worked through the various design charts in the paper and made comparisons between his experimental power losses and those predicted.

The differences that he found at 4,200 rev/min are very large, and surprising in view of comparisons such as those presented in Fig. B1, where experimental data from Wilcock and Rosenblatt (*B1*) is compared with our predictions.

The authors do not think that the reason for Mr. Gardner's large differences lies in the inaccuracy of reading the charts since, on their master copies, they arrive at similar predicted values. The charts themselves inevitably lose some accuracy over the computed data which was used to generate them since, to avoid a plethora of graphs, considerable condensing had to take place. The authors have, therefore, run the basic computer program for Mr. Gardner's cases, and the power losses at speeds of 2,800 and 4,200 rev/min are respectively 14 percent and 10 percent lower than his predictions using the paper.

The large differences are, thus, not directly attributable to the charts presented in the paper, but occur between Mr. Gardner's results and the authors' use of Ng and Pan's turbulence theory. The basic dimensionless data from this theory, for Mr. Gardner's geometry of a length/diameter ratio of 0.85, is presented in Fig. B2; this shows the relationship between load number, power loss, eccentricity ratio, and Reynolds number.

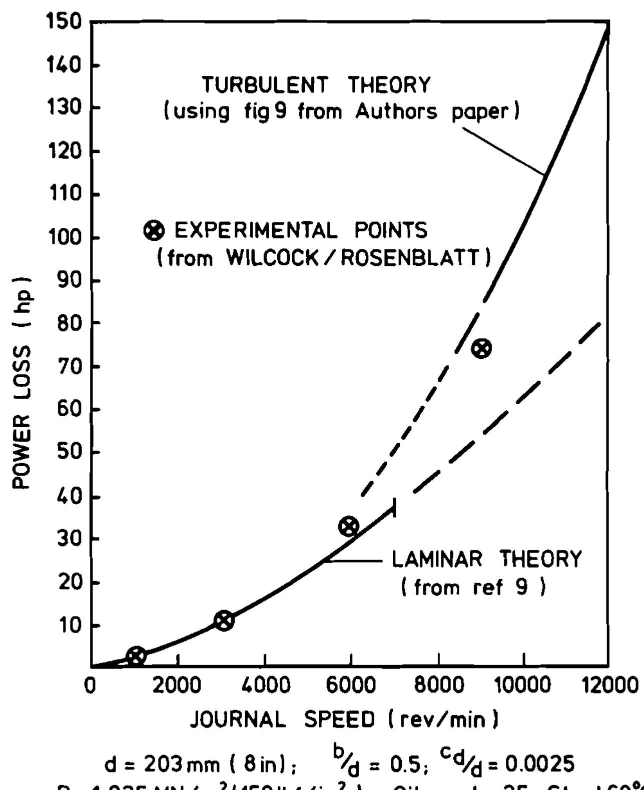


**Fig. B1—Trend in predicted power loss compared with experimental results for a 203 mm (8 in) diameter bearing.**

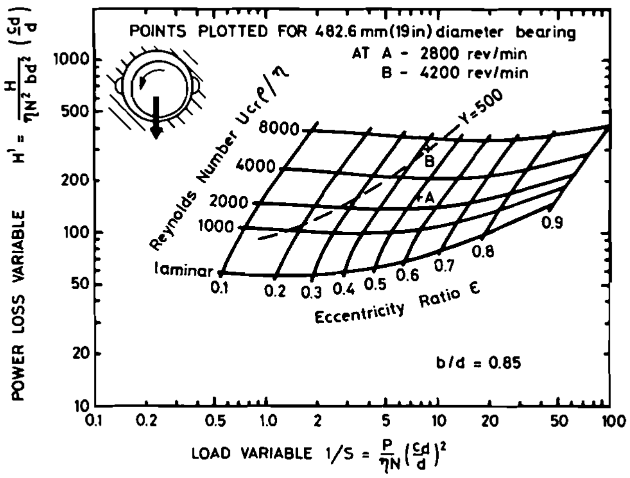


**Fig. B2—Dimensionless performance characteristics for a bearing with length to diameter ratio of 0.85, based on the theory of Ng and Pan.**

Now, the turbulence variable used in the paper ($Y$) is a function of Reynolds number ($Re$) and the Sommerfeld number ($S$), and, in Mr. Gardner's high speed case, has a value of about 500. This value has been plotted on Fig. B2 as the dotted line. The actual operating point must fall somewhere on this line, its precise position being determined only by the value of effective viscosity. Therefore, at a given value of viscosity, there is also a given value of the power loss variable ($H'$), and, consequently, of dimensional power loss. This relationship, taken from the dotted line shown, has been plotted in Fig. B3. The computed result gives an effective operating viscosity of 0.0047 Ns/m², which can be seen to correspond to a power loss of 440 kW (point B). The top scale shows the temperature relationship with viscosity for the particular oil used, and indicates an effective temperature of 95 C in this case.

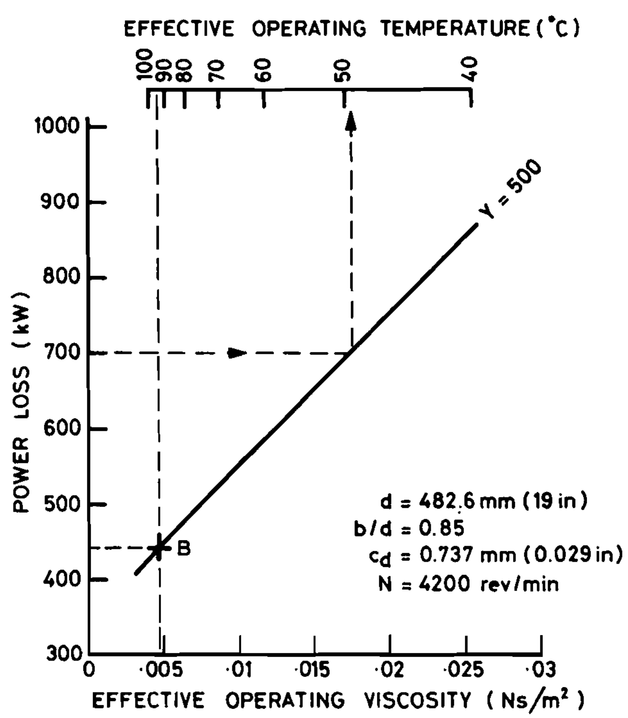


**Fig. B3—Variation in predicted power loss with effective viscosity for Gardner's test bearing.**

In order to obtain a power loss of 700kW, the effective viscosity needs to be about 0.0175 Ns/m², implying an effective temperature of about 50 C. Clearly, this is totally unrealistic, being of the same order as the feed temperature, and, therefore, the Ng and Pan theory cannot be made to tie in with Mr. Gardner's results. This theory has been used extensively for many years without any reports in the literature of substantial differences from experimental evidence, making Gardner's findings all the more surprising.

Referring to his original paper [his Ref. (A1)], it appears that power losses were obtained from heat balance calculations, based on measured oil flow rates and temperature rises, and there is, therefore, a possibility that churning and seal losses in the rig could be included in the total. Two sets of results from this reference have been replotted in Fig. B4; the "o" symbols refer to the results used by Mr. Gardner in his discussion, and the "x" symbols to, basically, a similar bearing but with an orificed oil flow of 5 gal/min to the trailing edge inlet groove. In the latter case, the total flow rates were given as 100 US gal/min at 4,000 rev/min and 50 US gal/min at zero speed. The authors' corresponding computed values (allowing the trailing edge groove to take all the flow it wants) are 75 US gal/min at 4,000 rev/min and 10 US gal/min at zero speed; again, this reinforces the view that somehow the same bearing conditions are not being modelled, and it may be that the experimental bearing has substantial additional oil flow from chamfers or reliefs. With experimental power losses obtained from a heat balance calculation, it is necessary for the total flow to be "perfectly" mixed on outlet, and large chamfer flows may cause problems with this technique.

In conclusion, the authors are satisfied that the charts presented in the paper adequately provide design information based on the turbulence theory of Ng and Pan and in a form which a designer can easily use. Mr. Gardner's results appear to contradict this theory, but the authors know of no other experimental evidence which indicates that the theory can seriously underestimate the power loss. It is now to be hoped that other researchers can present results which will indicate whether or not current turbulence theories are realistic.

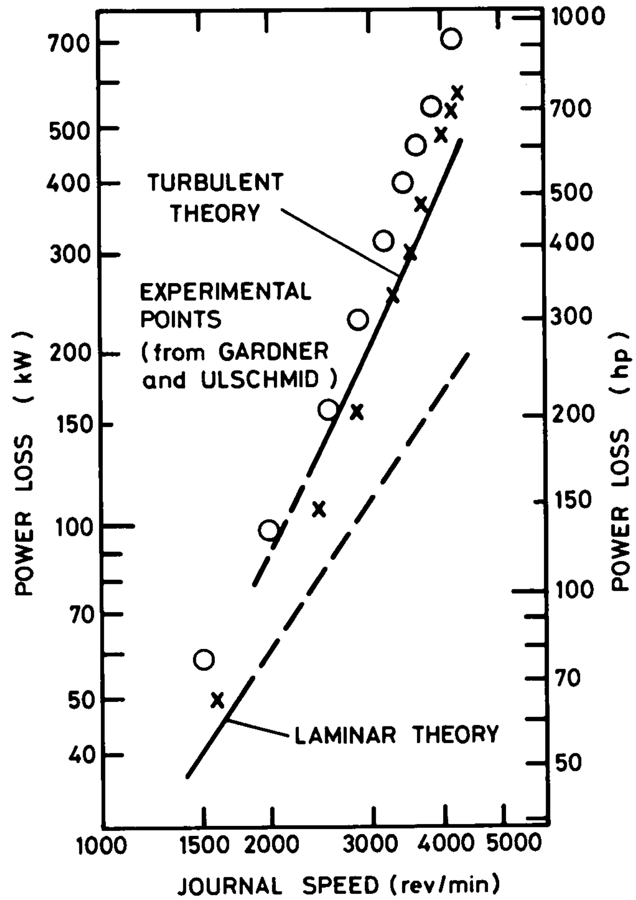


**Fig. B4—Trend in predicted power loss compared with experimental results for a 483 mm (19 in) diameter bearing.**

## REFERENCE

(B1) Wilcock, D. F. and Rosenblatt, M., "Oil Flow, Key Factor in Sleeve-Bearing Performance," *Trans. ASME*, 74; 849-865 (1952).