OPTIMUM COMPUTER DESIGN OF HYDRODYNAMIC JOURNAL BEARINGS

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By
MOHAMED ABDEL AZIZ AHMED KHATTAB
B.Sc., AIN SHAMS UNIVERSITY, EGYPT
Dipl., POLYTECHNIC OF TURIN, ITALY

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AUTHOR: Mohamed Abdel Aziz Ahmed Khattab
B.Sc., Ain Shams University, Cairo, Egypt
Dipl., Polytechnic of Turin, Turin, Italy

SUPERVISOR: Professor J.N.Siddall

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SCOPE AND CONTENTS:

A user - orientated computer program for an optimum solution of the hydrodynamic journal bearings is developed. The computer package is formulated in such a way to determine the optimum solution using any of the following optimization techniques adapted from OPTISEP: DAVID, SIMPLEX, SEEK1, and SEEK3 .

A user guide and a complete documentations for the computer package are included in the thesis.

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I INTRODUCTION

Historically, the problem of having a successfully working bearing dates back hundreds of years, Slider-type bearings, in which there is one surface sliding over the other seem to be the first known type of bearings.

A look about at the numerous engineering components, devices, and machines that typify our today's civilization illustrates how it is important to have successfully operating bearings. Many variations of bearing designs have been developed, each satisfying a particular function. Journal bearing types may be classified from the standpoint of cooling conditions, as (a) Self-contained bearings, where the heat developed by friction is dissipated from the bearing housing surface to the surrounding air by conduction, convection, and radiation; (b) Pressure-fed bearings, where the frictional heat is removed by the circulating oil which is introduced to the bearing, in this case, under a controlled pressure and temperature.

The full solution of a hydrodynamic journal bearing design may require much data and many theoretical computations in order to get a successfully operating bearing within the practical limitations of bearing design. In general, such a satisfactory solution is possible after a long series of trials, even from the practicing engineer. Therefore, a computer program to carry out this design problem and to cut down the effort, time and money will be very useful.

The primary objective of this book is to provide a computer package for an optimum design of hydrodynamic journal bearings. The package that has been developed in this thesis is highly user oriented and requires a minimum of knowledge of the theory of bearing design and no knowledge of computer programming, FORTRAN, or optimization theory. However, it is considered very important that the user have a good understanding of the principles of bearing operation and lubrication.

No successful design can be guaranteed by any theoretical analysis if the design assumptions do not match the real life situation. Practical limitations which are inherently unavoidable in real problems have been taken into account in the optimum design procedure, thereby minimizing the possibility of getting geometrically unreasonable or impractical designs. Chapter VI, gives a brief description of these limitations.

To satisfy the primary objective of the thesis, chapter II to chapter V give a discussion of the Hydrodynamic Theory of Lubrication starting from the fundamentals of viscosity.

Chapter VII, deals with the optimum design procedure, design criteria and constraints.

A user's manual and complete documentation is provided .

II FUNDAMENTALS

2-1 Viscosity

Definition of "viscosity" or the coefficient of viscosity seems to be have been first given in 1866. It may be best defined by considering a film of oil between two parallel horizontal plates as shown by figure 2-1a. The upper plate is set in motion by a force of F pounds, and the force is just sufficient to move the plate with a constant velocity V_2 in a horizontal direction.

Since the velocity is constant, the force F will depend on the condition of the lubricant oil and not on the mass of the plate. If the velocity of the lower plate is V_1, then the change of velocity between the plates is $V_2 - V_1 = dv$. Due to the difference in velocity a horizontal shearing stresses in the oil will develop, and an element of oil outlined by the rectangle a,b,c,d will be displaced to a new position a',b',c',d'.

According to Newton's law of viscous flow -"at any point in a fluid the shearing stress is directly proportional to the rate of shear" . If the plates are only a differential distance (dy) apart, then the rate at which shear stress takes place is

$$dv/dy$$

If A is the shear area of the element of oil, the shear stress according to Newton will be

$$s = F/A = \mu . dv/dy \qquad \text{psi} \qquad (2\text{-}1)$$

or

$$\mu = s/(dv/dy) \qquad \text{lb.sec/in}^2 \ \ (\text{REYN})$$

in which μ is called the "coefficient of internal friction" or the "absolute viscosity coefficient" or more simply "absolute viscosity" .

In the metric system, the absolute viscosity is

$$z = s/(dv/dy) \qquad \text{dyne. sec/cm}^2 \ (\text{POISE})$$

z always denotes the absolute viscosity in centipoise (1/100 of a poise), and μ denotes the absolute viscosity in reyn .

2-2 Kinematic Viscosity

Kinematic viscosity is another form of expressing viscosity, and this is defined as;

$$\text{Kinematic viscosity}, \nu = \frac{\text{absolute viscosity}}{\text{mass density of fluid}}$$

$$= \mu / \rho \qquad \text{in}^2/\text{sec.}$$

$$\text{or} \quad = z / \rho \qquad \text{cm}^2/\text{sec. (stoke)}$$

The equation of the kinematic viscosity for the Saybolt viscosimeter is

$$\nu = 0.22\ t - \frac{180}{t} \qquad \text{centistoke} \qquad (2\text{-}2)$$

where t is the time in seconds taken for a given volume of oil at a given temperature to flow out of a container

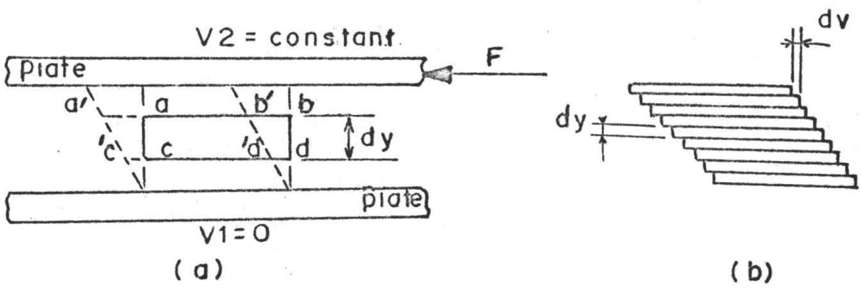


Fig 2-1 Horizontal shear in film

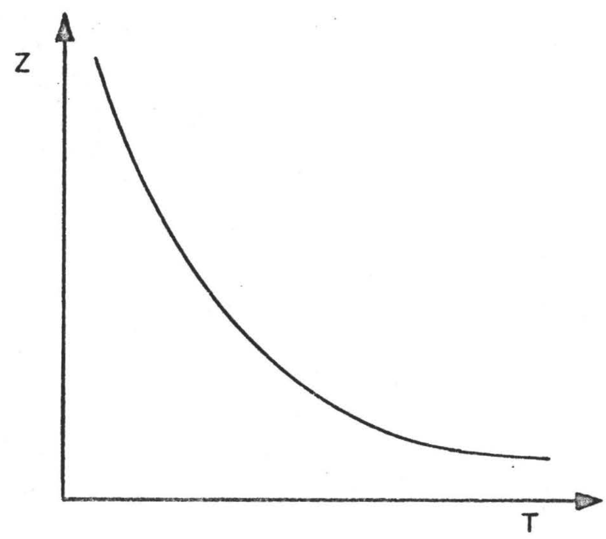


Fig 2-2 Viscosity-Temperature Curve

through a capillary tube .
Usually t is expressed in SUS (Saybolt Universal Seconds).

If t is given in SUS at temperature T °F, then the kinematic viscosity ν at the same temperature can be directly determined from equation(2-2), and to get the absolute viscosity in centipoises ($z = \rho . \nu$), the mass density ρ, at the same temperature T, must be known. How to find it, is the subject of the following section.

2-3 Specific Gravity

According to the specifications of the American Petroleum Institute (API), the specific gravity of oil is measured at 60 °F by a calibrated glass hydrometer, The following equation is used to determine the specific gravity at 60 °F

$$\rho_{60} = \frac{141.5}{131.5 + \text{degrees API at } 60\,^\circ\text{F}} \qquad (2\text{-}3)$$

From the specific gravity at 60 °F, (ρ_{60}), the specific gravity at any other temperature T, (ρ_T), can be obtained as follows .

$$\rho_T = \rho_{60} - 0.00035\,(T - 60) \qquad (2\text{-}4)$$

2-4 Relation Between μ and z

If z is given in centipoise, then

$$\mu = \frac{z}{6.9\times10^6} \quad \text{reyns} \qquad (2\text{-}5)$$

2-5 Viscosity—Temperature Relation

Viscosity of a liquid lubricant is a function of its temperature. It decreases by increasing the lubricant temperature. The relation between viscosity and temperature is different in different grades of oils, and two oils having the same viscosity at a certain temperature, might have different viscosities at another temperature.
Figure 2-2 shows the manner in which the viscosity changes with temperature, while Appendix 2 , shows a chart that indicate how the viscosity of a number of SAE-classified lubricating oils changes with temperature.

2-6 SAE Classification of Oils

This classification is based upon the viscosity without considering any other characteristics of lubricating oils.
Lubricating oils are classified by numbers in terms of SUS. The larger the SAE number, the more viscous is the oil.
There are two series of viscosity grades; one for motor oils, and other for gear oils .

SAE classification for crankcase oils	SAE classification for gear oils
5W	75
10W	80
20W	90
20	140
30	250
40	
50	

III HYDRODYNAMIC LUBRICATION OF JOURNAL BEARINGS

3-1 Introduction

Osborne Reynolds developed the first mathematical relationship between load, speed, fluid properties of the lubricant, and clearance that constitutes the hydrodynamic theory of lubrication. Sommerfeld and W.J. Harrison have subsequantly simplified the work of Reynolds .

Although the hydrodynamic theory of lubrication is based essentially on a complicated mathematical analysis, the results are easily interpreted. A knowledge of the hydrodynamic theory of lubrication will help to understand bearing performance .

The following section describes the development of oil pressure in a bearing .

3-2 Pressure Development Mechanism in an Oil Film, [30] .

Consider two parallel horizontal plane surfaces with an oil film inbetween, Fig 3-1. If the upper plate moves in the direction shown with a constant velocity V, and the lower plate is stationary. Then the velocity will vary uniformly from zero at the lower plate to its maximum value V at the upper surface. In this case, as mentioned in section 2-1, the rate of shear stress is constant throughout the oil film.

If we assume that the plates surfaces are wide enough in the direction perpendicular to motion so that the flow in that direction can be neglected, then the rate of oil flowing across any section is constant and the velocity of flow of the oil film at any point is proportional to the distance of the point from the stationary surface. This explains how a bearing with parallel surfaces can not support any load by its fluid film, and if any load is applied, the lubricant will be squeezed out of the bearing and metal to metal contact will exist under such condition.

Consider next the case in which the lower plate AB is stationary while the upper plate A'B' is moving in the perpendicular direction to the surface AB, as shown by Fig 3-2 . Surfaces AB and A'B' do not have in this case any relative motion in the horizontal direction. Considering the same assumption that the plates surfaces are wide enough in the direction perpendicular to the paper, so that the flow in this direction may be neglected . As the upper plate starts to move toward the lower plate as indicated by the arrow, the oil is squeezed out of the plates in both directions to the left and to the right of the midway section CC' and increases gradually to reach

the maximum value at sections AA', BB' at both sides. Inany cross section the velocity of oil flow is a maximum midway between the surfaces and zero at the surfaces. Velocity distribution curves in the oil film at different sections are shown in Fig 3-2 .
Thus under these conditions [30] ,"the oil in each layer (with the exception of the two layers directly on the surfaces, where the oil is adhering to the surfaces) increases its velocity as the distance from the section midway between AA' and BB' increases. Such a type of flow is possible if there is a pressure gradient along the surfaces, with a maximum pressure at the middle section CC' falling gradually to zero at the end sections AA' and BB' . Figure 3-2 indicates also the character of pressure distribution for the case described above.

It has been found that the pressure developed in different sections of the film depends upon the viscosity of the lubricant and the difference in the rate of flow of lubricant through these cross-sections. However, in this particular case , this difference depends upon the velocity of the upper plate A'B', the areas of surfaces AB and A'B', and the instantaneous clearance h. This type of flow of liquid lubricant due to the difference in pressure in different cross sections is known as "pressure induced flow" .

Another example of this kind of flow is given by figure 3-3. Where the lubricant flows from the left to the right due to the difference in pressure between sections AA' where the pressure is P_1, and section BB' where the pressure is P_2. The velocity distribution across the fluid film is exactly the same in different sections and is given by

$$U = \frac{1}{2\mu} \cdot \frac{dp}{dx} (t^2 - y^2)$$

where μ is the absolute viscosity, lb.sec/in^2,

$\frac{dp}{dx}$ is the pressure gradient,

t = h/2(half the clearance), and

2y is the thickness of an element of fluid having length dx and width b .

According to these three examples, there are two main types of velocity distribution across the oil film. The first type of velocity distribution is a straight line and it occurs when the rate of shearing stress across the lubricant film is constant. In the second type, the velocity distribution is a curved line, so that the rate of shear in the different layers across the oil film is different. It is clear that the first type of velocity distribution takes place where there is a relative motion between two parallel surfaces in a direction parallel to each other, as illustrated by the first example. The

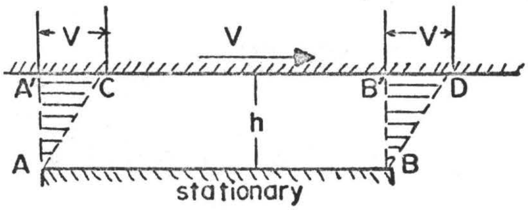


Fig 3-1 Flow between two parallel surfaces
A'B' is moving, AB is stationary

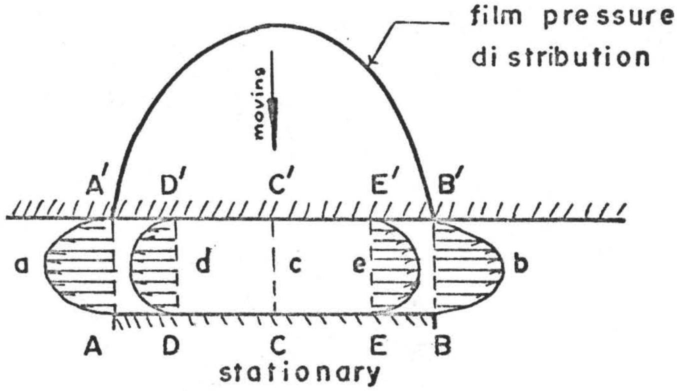


Fig 3-2 Flow between two parallel surfaces
A'B' is moving downward, AB is stationary

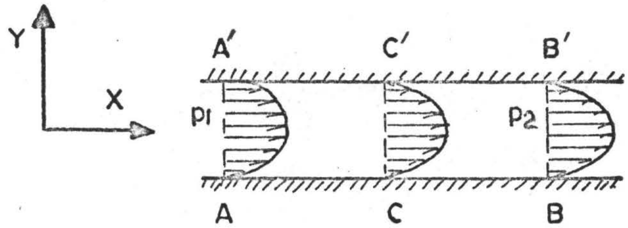


Fig 3-3 Pressure-induced flow

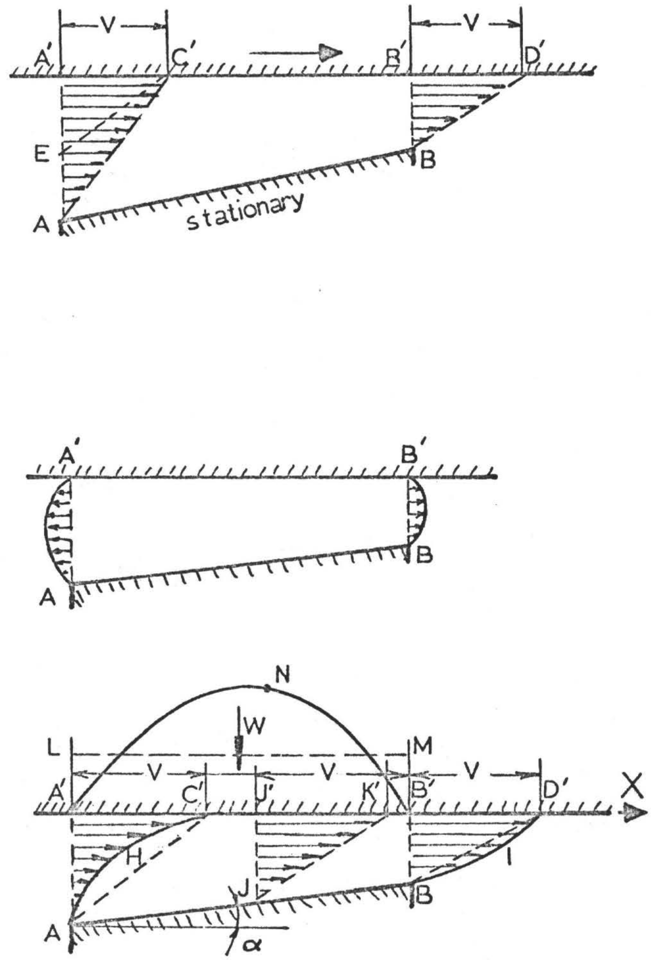
second type occurs if pressure exists in the oil film due to a change of the volume between the surfaces, as explained in the second example figure 3-2, or due to other means that do not depend upon the motion of the surfaces as in the third example of figure 3-3, or due to either factor. Therefore, it is important to note that pressure in the oil film always exists whenever the velocity distribution across the film is represented by a curved line.

Consider now the case in which the two plates are not parallel while the upper plate is moving to the right with velocity V and the lower plate is stationary, Fig3-4a. Again the lubricant flow in the direction perpendicular to the paper is neglected. Surface A'B'tends to carry a certain volume of lubricant per unit time, through section AA', to the space between the surfaces. Assuming a width of unity, this rate of flow may be represented as shown in figure 3-4a by the triangle AC'A'. The volume of lubricant in this section that tends to discharge from the space between the surfaces through section BB' during the same period of time is represented by BD'B'. Since there is a difference in the cross sectional area of section AA' and BB', a surplus volume represented by AEC' will be present. Because there is no flow in the perpendicular direction, as assumed before, and because the lubricant is assumed incompressible, then the rate of lubricant flow to the film space should be equal to the rate of flow out of the space; and the excess volume of lubricant carried into the space is squeezed out through sections AA'and BB' causing a constant "pressure-induced flow" through these sections .

Figure 3-4b indicates the character of the velocity distribution across these sections due to "pressure induced flow" only . Figure 3-4c shows the actual velocity distribution in sections AA' and BB' . This actual velocity distribution is the result of the combined flow of lubricant due to "pressure-induced flow" and viscous drag.

Studies of the actual velocity distribution curves at the different sections of figure 3-4c indicate that the maximum pressure exists in a section, JJ', somewhere in-between sections AA' and BB', but nearer to BB' than to AA' . Also it has been indicated that at section JJ' there is no pressure induced flow and the velocity distribution across this section is represented by the straight lineJK'.

The character of the pressure distribution in the oil film in this case is represented by the curve A'NB'. Line LM shows the mean pressure in the film. Due to the pressure developed in the oil film, the upper plate A'B' can support a vertical load W preventing in turn metal-to-metal contact between the surfaces. If the load W is increased , the space clearance between the surfaces will be



(a)

(b)

(C)

Fig 3_4 Flow between two inclined plane surfaces decreased and consequently the ratio between the distances AA' and BB' will increase, which means that the pressure developed in the oil film will also increase and a new equilibrium between the load W and the pressure developed in the oil film will be established .

From the foregoing discussion, it can be seen that the load carrying capacity of a bearing of such an arrangement can be increased by any of the following .

a) Increasing lubricant viscosity.

b) Increasing surface area of the bearing.

c) Increasing the relative motion between the surfaces.

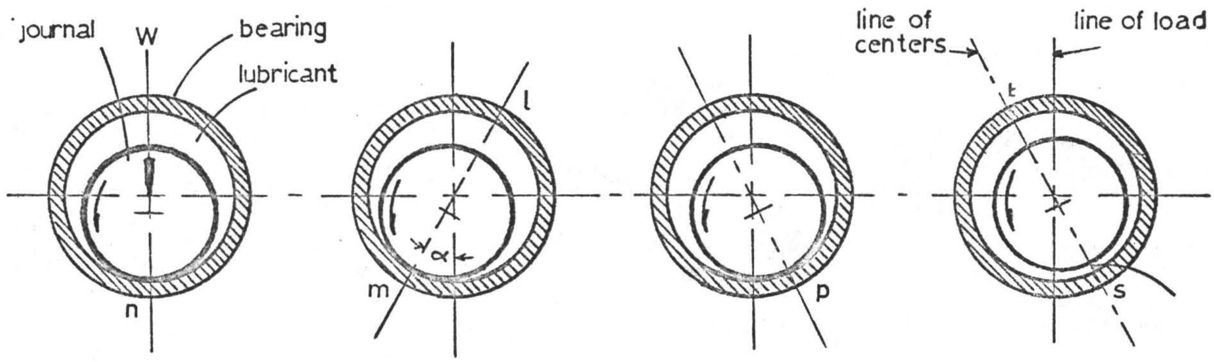
d) Increasing, in certain limits, the angle between the surfaces.

e) Decreasing the space clearance.

It can be seen also that a positive pressure in the oil film is developed when the direction of motion of the upper plate is toward the convergence of the two surfaces AB and A'B'. On the other hand a negative pressure (less than the atmospheric) may be expected if the motion of the upper plate is reversed without changing the inclination of the lower plate. In this case the bearing will be unable to support any load by its lubricant film .

3-3 Behavior of a Journal in its Bearing

A full journal bearing is shown diagrammatically in figure 3-5, assuming that the clearance space is



(a) (b) (c) (d)

Fig 3_5 Formation of continuous oil film in a full journal bearing completely filled with lubricant fluid all the times. If the shaft is at rest as in figure 3-5a while it is loaded by a vertical load W, a metal-to-metal contact between the surfaces of the journal and the bearing will occur. If the journal starts to turn in the direction shown in figure3-5b it will roll up on the bearing wall due to friction between the surfaces. The point of contact at that time is moved from n to m. The friction in this case corresponds to boundary or extreme boundary conditions. The angle α shown in figure 3-5b may correspond to the angle of sliding friction between the surfaces if the pressure developed in the film is neglected, perhaps due to a very slow starting speed. In this position we can define the oil film condition, as a converging film above the line ml and a diverging film below this line. Therefore, as the speed increases, the moving journal surface tends to drag oil into the wedge shaped area (converging film) between the journal and the bearing. As mentioned in section 3-2, under such conditions a positive pressure is developed in the converging part of the oil film. This pressure forces the journal to move to the right in the position shown in figure 3-5c .

As the journal speed increases , an oil pressure is created, sufficient to separate the surfaces, and the journal is forced into the position shown in figure 3-5d. This is the equilibrium position where the surfaces are separated by a film of oil whose minimum thickness is now at point s. The magnitude of the minimum film thickness at point s and the position of the line of centers st depend on the load W, properties of the oil, size and speed of the journal , the clearance, and the bearing length, (as will be seen in the following sections and in chapter VI) .

3-4 Reynold's Equation in Two Dimensions

The phenomenon of pressure development in a converfing oil film discussed in the preceding sections was first analysed mathematically by O. Reynolds in 1866. He obtained a differential equation defining the relationship between the pressure gradient, the lubricant viscosity and the form of the oil film.

Certain basic assumptions, which simplify the conditions existing in the actual bearing, are the foundation of the entire analysis. The assumptions are:

1- There is no oil flow in the perpendicular direction to motion, that is, no end leakage from the bearing to the outside. This means that the bearing is of infinite width in the Z direction, see figure 3-6.

2- The viscosity of the lubricant is constant throughout the oil film. A number of authorities have shown that this is a reasonable assumption since little error will be introduced, and there is a great simplification in the bearing analysis.

3- The flow of the lubricant in the film is laminar.

4- Inertia forces in the oil film resulting from acceleration of the fluid are so small that their influence on the pressure developed in the film may be neglected.

5- The fluid is incompressible, therefore the mass rate of flow past adjacent sections is constant.

6- Shear stress is proportional to the rate of shear . This means that the liquid is Newtonian .

7- There is no change in the pressure across the clearance space, so that the pressure in the film is only a function of x.

8- No slip occurs between the lubricant and the bearing surfaces.

It is defined by many authors [22 , 30 , 33],and others, that a bearing in which there is no end flow (end leakage) and in which the viscosity of the lubricant is constant throughout the film is called an"idealized bearing" . Hence the results of calculations based upon these theoretical assumptions have only a qualitative value and they must be corrected for quantitative studies. An important correction factor is one that takes care of the end leakage effect in the actual bearing.

Consider the part of the converging oil film shown in figure 3-6, where the bearing surface is fixed and the journal rotates with a constant speed.

Let X, Y, and Z be the directional axis as indicated in the figure, then;

x is the distance in the tangential direction.

y is the distance normal to the bearing surface.

z is the distance along the axis of the journal.

p is the oil film pressure in psi. above the bearing.

s the shearing stress in oil in psi.

We assume that the journal rotates in the direction indicated in figure 3-6. According to the basic assumption of the analysis, summing the four indicated forces in the X-direction gives,

$$(p + dp)\, dy\, dz + s\, dx\, dz = p\, dy\, dz + (s + ds)\, dx\, dz$$

or

$$\frac{ds}{dy} = \frac{dp}{dx} \tag{3-1}$$

Consider now figure 3-7, where the clearance space around the journal is filled with oil and there is no end leakage out of the bearing. According to these two assumptions, the oil flows only in the X-direction while the oil flow past any section in that direction in constant. The velocity of oil flow is zero at $y = 0$ and it is equal to the journal velocity at $y = h$, where h is the clearance thickness .

Let; Q = the rate at which the oil circulates around the journal in the X-direction.

L = the bearing length, inches.

v = oil flow velocity at a distance y above the bearing surface.

From figure 3-7, then

$$Q = \int_0^L \int_0^h v \, dy \, dz$$

$$Q = L \int_0^h v \, dy \qquad \text{in}^3/\text{sec} \qquad (3\text{-}2)$$

From equation (1-1), we have

$$s = \mu . dv/dy$$

Differentiating with respect to y, then

$$ds/dy = \mu . d^2v/dy^2 \qquad (3\text{-}3)$$

But, from equation (3-1), equation (3-3) becomes

$$d^2v/dy^2 = \frac{1}{\mu} dp/dx$$

Integrating we get

$$v = \frac{1}{\mu} (dp/dx) \frac{y^2}{2} + c_1 y + c_2 \qquad (3\text{-}4)$$

From the boundry conditions c_1 and c_2 can be determined;

$v = 0$ at $y = 0$ giving, $c_2 = 0$

and $v = V$ (the peripheral speed of the journal) at $y = h$ (the oil film thickness), giving

$$c_1 = V/h - \frac{h}{2\mu}(dp/dx)$$

Finally equation (3-4) becomes

$$v = V.y/h - \frac{1}{2\mu}(dp/dx)(hy-y^2) \qquad \text{in/sec} \qquad (3\text{-}5)$$

This is the velocity in equation (3-2). It consists of

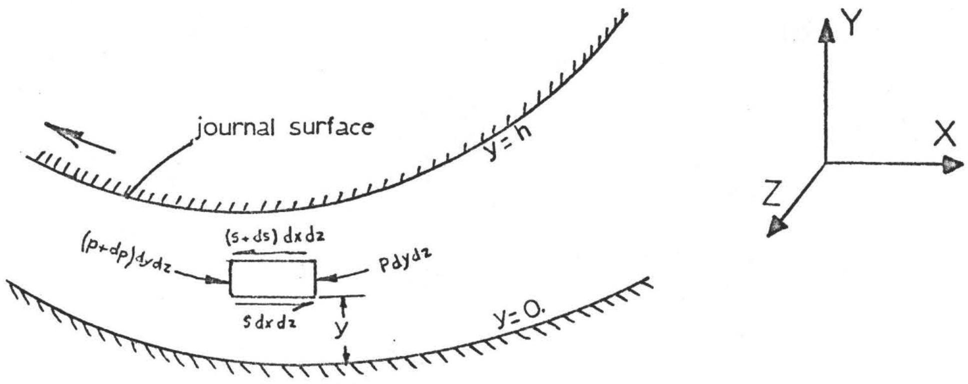


Fig 3_6 Forces on an oil particle

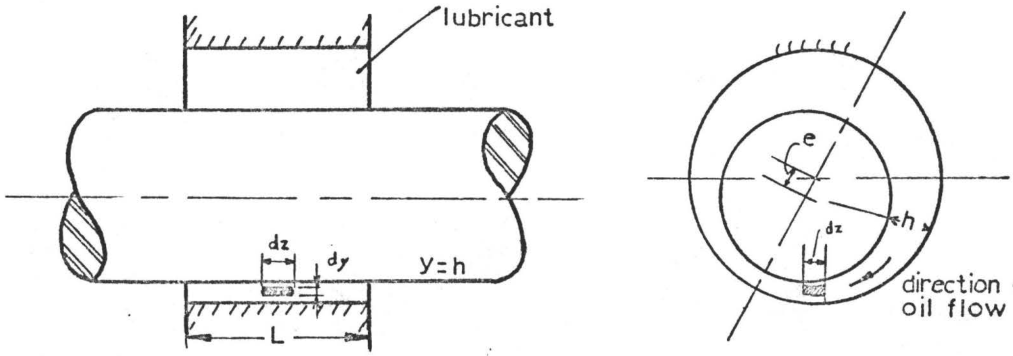


Fig 3_7 Oil circulating around a journal

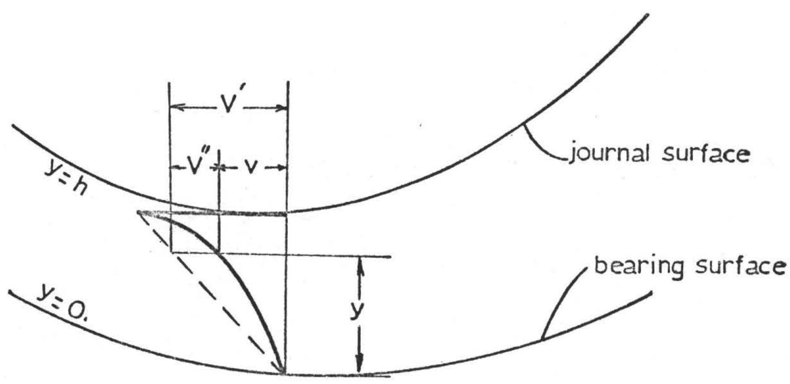


Fig 3_8 Velocity of oil around a journal

two parts. The first part is $V' = \frac{V}{h} y$ shown as a straight line in figure 3-8 and it is caused by the velocity of the journal with respect to the bearing, as already discussed in section 3-2. The second part, $V'' = -\frac{1}{2\mu}(dp/dx)(hy - y^2)$, is a parabola and it depends upon the lubricant viscosity and the pressure gradient dp/dx.

Figure 3-8, gives a graph representing equation (3-5). Notice that this graph is the same as that of figure 3-4c represented by (AHC').

Knowing the oil velocity as a function of y as given by equation (3-5), then the rate at which oil circulates may be found from equation (3-2) to be

$$Q = L\left[\frac{Vh}{2} - \frac{h^3}{12\mu}(dp/dx)\right] \quad in^3/sec \qquad (3\text{-}6)$$

If h_1 is the oil film thickness where $dp/dx = 0$, and Q_1 is the corresponding flow rate, then

$$Q_1 = VLh_1/2 \qquad in^3/sec \qquad (3\text{-}7)$$

Since the rate of flow across any section is constant, thus

$$Q = Q_1$$

Combining equation (3-6) and equation (3-7), the pressure gradient becomes

$$dp/dx = (6\mu V/h^3)(h-h_1) \qquad (3\text{-}8)$$

This is the Reynold's differential equation in two dimensions for the pressure gradient in a converging oil film with no end leakage. This equation may be used for determining the load-carrying capacity.

3-5 Harrison Equation for Oil-film Pressure Around a Journal Bearing, [33].

Consider the full journal bearing in figure 3-9. The angle θ is measured from the line of centers indicated where the oil film thickness is h at $\theta = 0$. The angle θ is measured positive in the direction of motion of the journal. If R is the radius of the journal, then $dx = Rd\theta$ a differential distance around the journal.

Let e represent the journal eccentricity, as defined previously in figure 3-5 .

$$e = C.\epsilon$$

where; C is the radial clearance, and

ϵ is the eccentricity ratio.

The oil film thickness h at any angle θ as shown in figure 3-9 becomes

$$h = e \cos\theta + \sqrt{(R+C)^2 - e^2\sin^2\theta} - R$$

Since, $e^2\sin^2\theta \ll (R+C)^2$, it can be neglected, then the oil film thickness as a function of θ is

$$h = C(1 + \epsilon \cos\theta) \qquad (3\text{-}9)$$

Similarly, if the oil film pressure is a maximum at angle θ_1 then the corresponding oil film thickness is

$$h_1 = C(1 + \epsilon \cos\theta_1) \qquad (3\text{-}10)$$

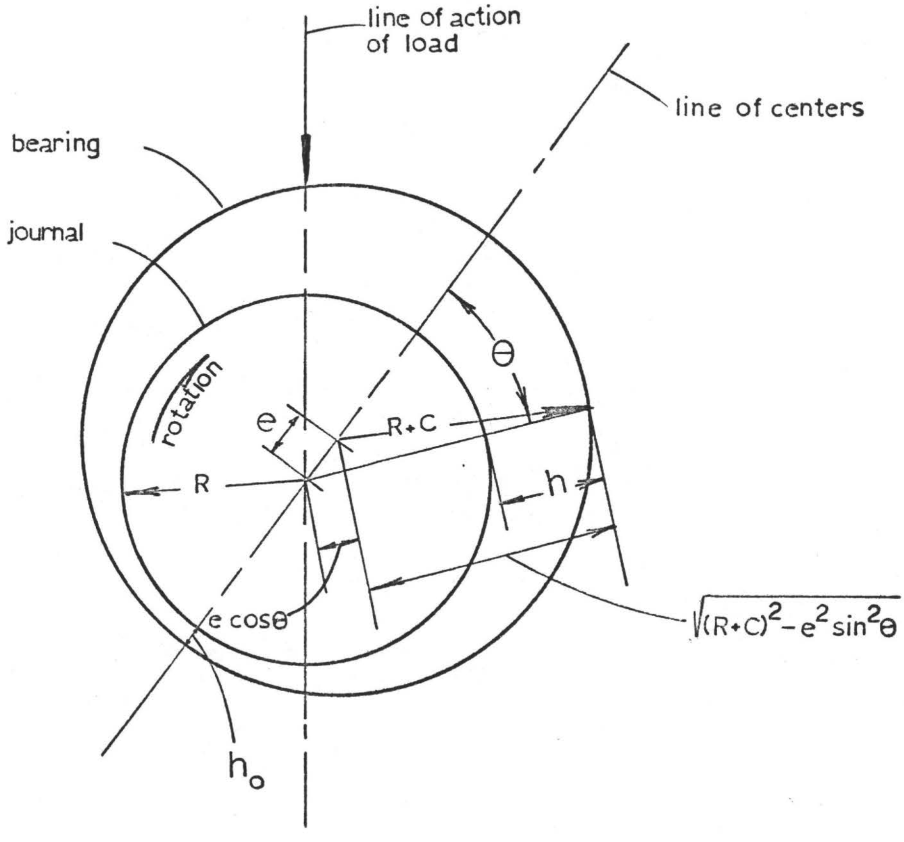


Fig 3_9 Position of journal in a full bearing

Accordingly, equation (3-8) will take the following form

$$dp/d\theta = (6\mu VR\epsilon)/C^2 \left[(\cos\theta - \cos\theta_1)/(1+\epsilon\cos\theta)^3\right] \qquad \text{.............. (3-11)}$$

then, the oil film pressure at any angle θ is

$$p = (6\mu VR\epsilon)/C^2 \int((\cos\theta - \cos\theta_1)/(1+\epsilon\cos\theta)^3)d\theta + p_o \qquad \text{.............. (3-12)}$$

where p_o is the oil film pressure at the line of centers where θ = zero.

Assuming that $p - p_o = 0$ at $\theta = 0$, and at $\theta = 2\pi$, then integrating equation (3-12) will give

$$\cos\theta_1 = -3\epsilon/(2+\epsilon^2) \qquad \text{.............. (3-13)}$$

the oil film pressure becomes

$$p - p_o = \frac{6\mu VR\epsilon}{C^2} \cdot \frac{\sin\theta\,(2+\epsilon\cos\theta)}{(2+\epsilon)^2(1+\epsilon\cos\theta)^2} \qquad \text{.............. (3-14)}$$

This is Harrison's equation for a full journal bearing, based on the assumption of no oil flows from the ends of the bearing.

Figure 3-10 indicates the variation in oil film pressure, in a polar diagram using equation (3-13) and equation(3-14), taking; $(6\mu VR/C^2) = 1$,

$\epsilon = 0.8$, and

p_o = the journal radius.

From the plot it is easy to see that :

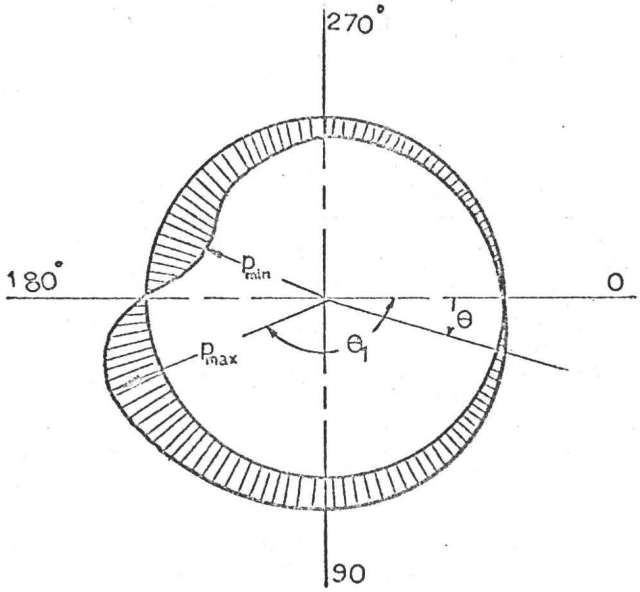


Fig 3_10 Theoretical oil film pressure distribution

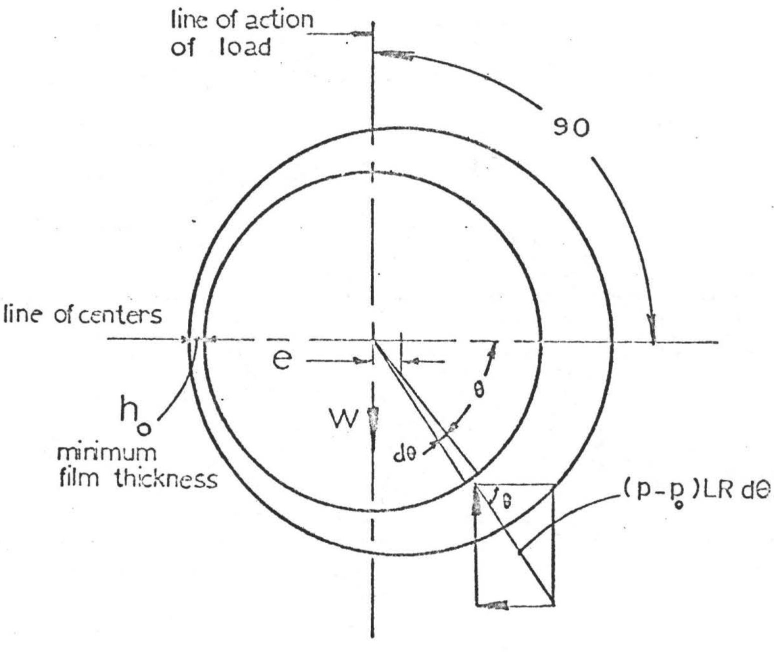


Fig 3_11 Theoretical position of a full journal bearing

a) The pressure distribution on one side of the line of centers is the negative reflection of that on the other side .

b) The vertical components of both the negative and positive pressure act in the same direction, whereas the horizontal ones act in opposite directions. Accordingly the resultant force able to carry a vertical load on the journal will act perpendicular to the line of centers. Hence the journal theoretical position will be as shown in figure 3-11. In an actual bearing, this position of the journal with respect to its bearing can never exist.

In the actual problem the effect of end leakage is such that negative pressure will not exist and the positive pressure will not cover 180^{o} arc, The angle between the line of action of the load and the line of centers is always less than 90^{o} as shown in figure 3-5.

3-6 Load-Carrying Capacity, Harrison-Sommerfeld Equation, [33]

If W is a vertical load acting on the journal, then from figure 3-11, W can be written as follows:

$$W = LR \int_0^{2\pi} (p - p_o) \sin \theta \, d\theta \qquad (3\text{-}15)$$

From Harrison equation (3-14), we substitute for $(p - p_o)$ and integrate

$$W = 12\pi L \mu V (R/C)^2 \frac{\epsilon}{(2+\epsilon^2)\sqrt{1-\epsilon^2}} \quad \text{lbs} \qquad \text{................ (3-16)}$$

This is the Harrison-Sommerfeld equation for the load - carrying capacity of an idealized journal bearing.

Considering that :

$$V = \omega R$$

$$W = 2RLP$$

where;

ω is the angular velocity of the journal $\frac{2\pi N}{60}$ rad/sec

N is the rotational speed of journal, revs/min, and

P is the unit load, psi of projected bearing area.

equation (3-16) takes the following form

$$\frac{\mu N}{P}\left(\frac{R}{C}\right)^2 = \frac{5(2+\epsilon^2)\sqrt{1-\epsilon^2}}{\pi^2 \epsilon} \quad \text{sec/min} \qquad \text{................ (3-17)}$$

The group $\frac{\mu N}{P}\left(\frac{R}{C}\right)^2$ is called the Sommerfeld number, designated by S, thus

$$\left.\begin{aligned} S &= \frac{\mu N}{P}\left(\frac{R}{C}\right)^2 \\ \text{or} \quad S &= \frac{5(2+\epsilon^2)\sqrt{1-\epsilon^2}}{\pi^2 \epsilon} \end{aligned}\right] \qquad \text{....... (3-18)}$$

The reciprocal of the Sommerfeld number is called the " capacity number ", designated by C_n .

Thus,

$$C_n = \frac{P}{\mu N}\left(\frac{C}{R}\right)^2$$

$$\text{or} \qquad C_n = \frac{\pi^2 \epsilon}{5(2 + \epsilon^2) . \sqrt{1 - \epsilon^2}} \qquad (3\text{-}19)$$

Although the capacity number, given by equations (3-19), represents the load-carrying capacity of a bearing only in an ideal case, it is very important in any real bearing analysis .

3-7 Minimum Oil-Film Thickness

If the capacity number is known, then from equation (3-19), the eccentricity ratio ϵ can be determined. The minimum film thickness h_o is at $\theta = 180^o$, see Fig 3-9. Therefore, we substitute $\theta = 180^o$ in equation (3-9), and the minimum film thickness will be

$$h_o = C(1 - \epsilon) \quad \text{inch} \qquad (3\text{-}20)$$

3-8 Friction Torque in Full Journal Bearing

To determine the frictional torque in this case, the shearing stress in the oil at the journal surface must be known. Therefore, using equation (3-1),

$$ds = \frac{dp}{dx} dy$$

in which dp/dx is a constant. Integration gives

$$s = \frac{dp}{dx} y + s_o$$

where s_o is the shear stress in the oil at the bearing surface where $y = 0$.

But $$s = \mu \frac{dv}{dy}$$ from equation (1-1), thus

$$\mu \frac{dv}{dy} = \frac{dp}{dx} y + s_o$$

or $$\mu\, dv = s_o dy + \frac{dp}{dx} y\, dy$$

after integration

$$\mu v = s_o y + \frac{dp}{dx} \frac{y^2}{2} + \text{constant}$$

but constant = 0, as $y = 0$ if $v = 0$

Then

$$\mu v = s_o y + \frac{dp}{dx} \cdot \frac{y^2}{2}$$

If $y = h$ (the oil film thickness), then the journal speed v becomes V, so that s_o will be

$$s_o = \frac{\mu V}{h} - \frac{dp}{dx} \cdot \frac{h}{2}$$

Consequently,

$$s = \frac{\mu V}{h} + \frac{dp}{dx} \left(y - \frac{h}{2}\right)$$

Let $y = h$, and $dx = R\, d\theta$

The shear stress at the journal surface will be

$$s = \frac{\mu V}{h} + \frac{h}{2R} \cdot \frac{dp}{d\theta} \quad \text{lb/in}^2 \qquad (3\text{-}21)$$

Examining equation (3-21), it can be seen that it is made up of two parts; the first part depends on the journal speed and the second part on the rate of change of film pressure with the angle around the journal.

Multiplying the shear stress given by equation(3-21) by the area under shear, L dx = LR dθ, then by the journal radius R, we get the differential torque dM as

$$dM = \frac{\mu V R^2 L}{h} d\theta + \frac{RL}{2} h \, dp$$

Substituting dp from equation (3-11), and h from equation (3-9) gives the total frictional torque.

$$M = \frac{\mu V R^2 L}{C} \int_0^{2\pi} \frac{d\theta}{(1 + \epsilon \cos \theta)} +$$

$$\frac{3\mu V R^2 \epsilon L}{C} \int_0^{2\pi} \frac{\cos \theta - \cos \theta_1}{(1 + \epsilon \cos \theta)^2} d\theta$$

or

$$M = M_v + M_p \qquad \text{lb.inch}$$

where; M_v is the velocity torque caused by viscous friction, and

M_p is the film pressure torque.

We let $V = \omega R$ and integrate to get the velocity torque.

$$M_v = \frac{2\pi}{\sqrt{1-\epsilon^2}} (\mu R^3 L \omega / C) \qquad \text{lb.inch}$$

and the pressure torque

$$M_p = \frac{6\pi \epsilon^2}{(2 + \epsilon^2)\sqrt{1 - \epsilon^2}} \cdot \frac{\mu R^3 L \omega}{C} \qquad \text{lb.inch}$$

Therefore the total torque will be

$$M = \frac{\mu R^3 L \omega}{C} \cdot \frac{2\pi}{\sqrt{1-\epsilon^2}} \cdot \left(1 + \frac{3\epsilon^2}{(2 + \epsilon^2)} \right) \qquad \text{lb.inch}$$

Rearranging the last equation gives

$$\frac{MC}{R^3LN\mu} = \frac{\pi^2}{15\sqrt{1-\epsilon^2}} \cdot \left(1 + \frac{3\epsilon^2}{(2+\epsilon^2)}\right) \quad \text{min./sec.}$$

.... (3-22)

The quantity $\frac{MC}{R^3LN\mu}$ is the "torque number" .

IV INFLUENCE OF END LEAKAGE ON THE BEHAVIOR OF BEARINGS

4-1 Introduction

Several theoretical and unrealistic-assumptions were considered in deriving Reynold's differential equation in section 3-4, and consequently in deriving the capacity equation in section 3-6. The assumption that has the greatest effect on the results of the bearing analysis is that there is no end flow in the direction perpendicular to the direction of rotation of the journal . Actually the oil flows out of the bearing ends and the shorter the bearing , the more oil runs out.

Experiments prove that the actual oil film pressure in a full journal bearing carrying a steady load in one direction is about that shown in figure 4-1. The supporting pressure exists only over about 120° of arc, and the load line is shifted off the bearing center in such a way that the line of centers makes with it an angle less than the 90°, stated by the theoretical analysis in section 3-5 .

In this chapter the determination of the load capacity of an actual bearing, considering end flow, is discussed.

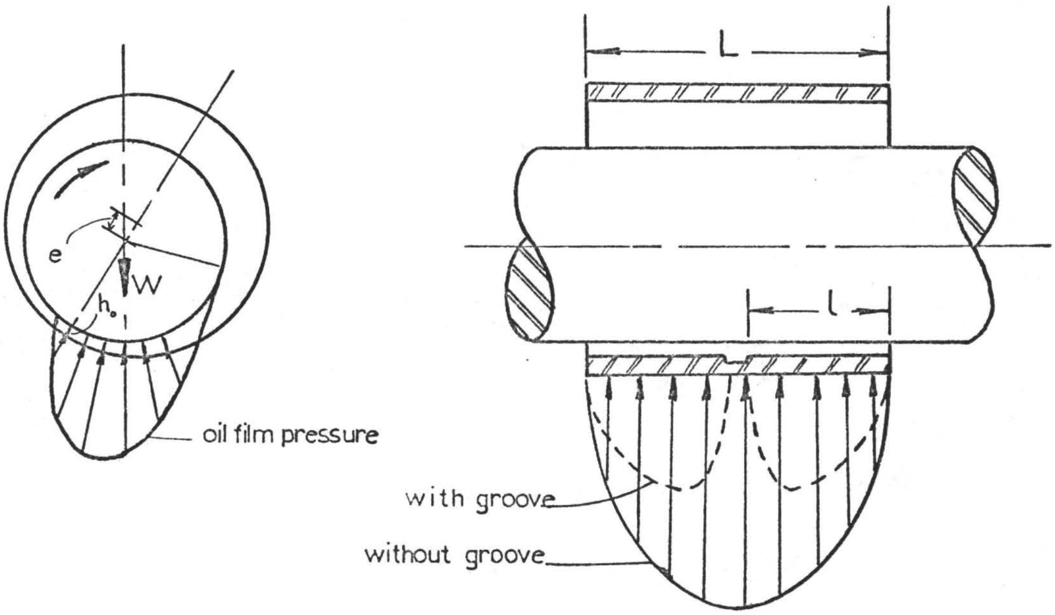


Fig 4_1 Diagram of full bearing showing oil film pressure

4-2 Reynold's Differential Equation in Three Dimensions, [30] .

The eight basic assumptions explained in section3-4 and used in the derivation of Reynold's equation in two dimensions are the basis of the present derivation, however the end flow or the flow in the Z-direction will be considered as occurring.

Without going into mathematical details, Reynold's equation in three-dimensions may be written

$$\frac{\partial}{\partial x}\left[\frac{h^3}{\mu}\cdot\frac{\partial p}{\partial x}\right] + \frac{\partial}{\partial z}\left[\frac{h^3}{\mu}\cdot\frac{\partial p}{\partial z}\right] = 6\ V\frac{dh}{dx} \qquad (4\text{-}1)$$

4-3 Kingsbury's electrical analogy

Kingsbury [21,22], and Needs [27], applied electrical analogy methods to solve Reynold's differential equation in three dimensions, and to come out with some leakage factors for the actual bearing analysis.

4-4 Dennison correlation

The experimental work of Kingsbury, Needs, Howarth, McKee and McKee, and others has been correlated, with reference to the theory of lubrication, by Dennison [9]. He has determined values for the capacity number given by equation (3-19) as a function of the eccentricity ratio ϵ , and tabulated these values of C_n for various length to diameter ratios (L/D).

i.e

$$C_n = \frac{P}{\mu N} \left(\frac{C}{R}\right)^2 = f(\epsilon) \qquad \text{.......(4-2)}$$

The same thing has been done for the torque number, where the right side of equation (3-22) was replaced with experimental data, which includes the effect of end flow and deals with that part of the bearing arc which can support the load. The experimental values of the torque number T' are given as a function of the eccentricity ratio ϵ ,i.e.

$$T' = \frac{MC}{\mu R^3 LN} = f(\epsilon) \qquad \text{.......(4-3)}$$

The group of curves shown in Dennison's design chart figure 4-2 express the results in terms of C_n and T' as a function of ϵ and L/D ratio. The data of figure 4-2 has been tabulated for reference in Appendix 3 .

The actual frictional torque can be determined from Dennisons design chart. The chart is entered by the value of C_n to the appropriate L/D ratio. From the chart T' is taken. Then the actual torque loss by friction can be determined, using equation (4-3) by transposition, i.e.

$$M = \frac{\mu N R^3 L}{C} T' \qquad \text{lb.inch ....(4-4)}$$

where M is the actual total frictional torque (i.e considering end leakage).

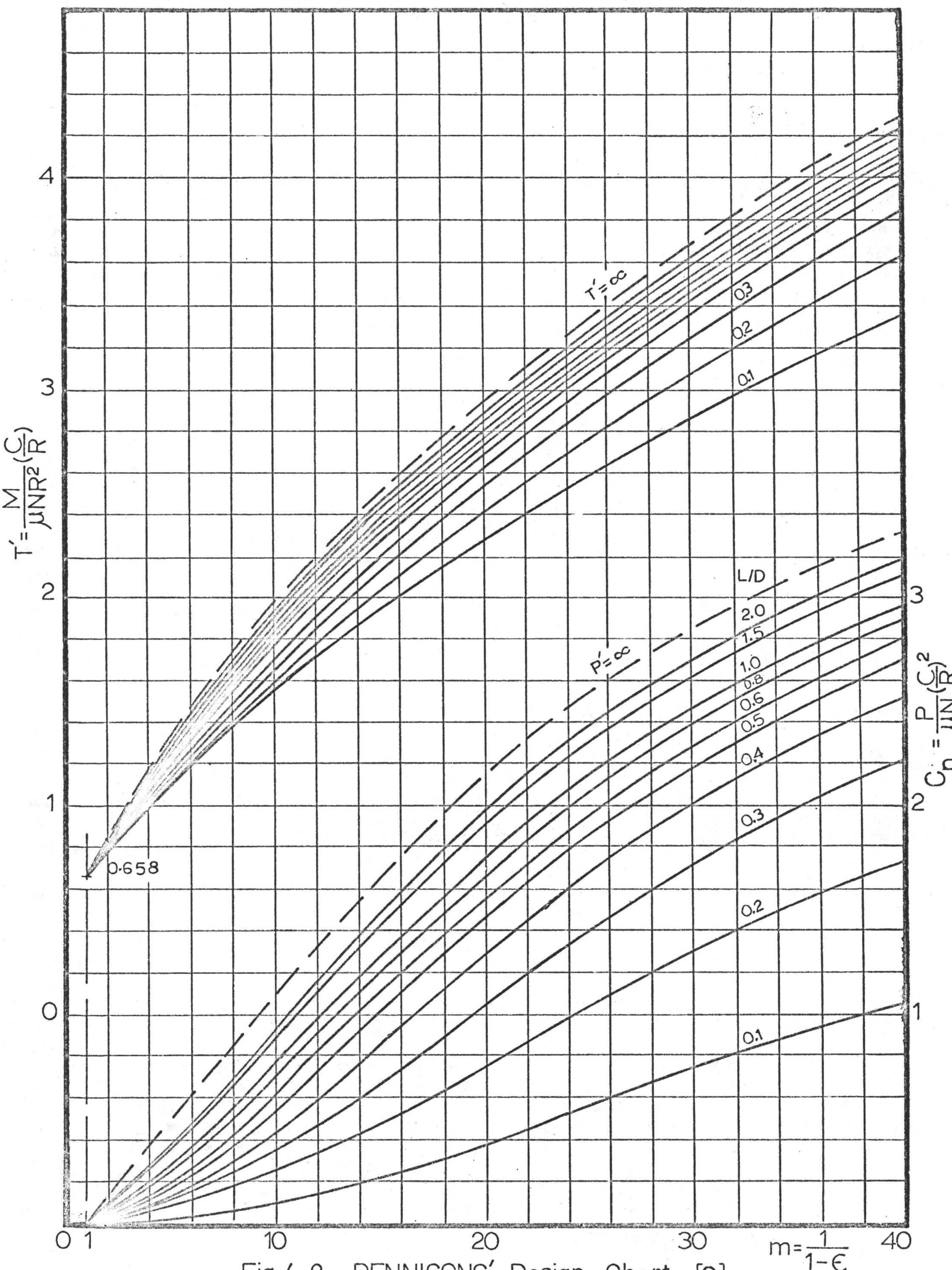


Fig 4_2 DENNISONS' Design Chart, [9]

V JOURNAL BEARING DESIGN

5-1 Introduction

Journal bearings may be classified according to the method of lubricant feed to them, roughly, to three main classes; Non-pressurized (self-contained) bearings, Pressurized-fed bearings, and Externally pressurized bearings.

This thesis deals only with the first two classes; the non-pressurized, and the pressure-fed bearings.

5-2 Self-Contained Bearings

This class of journal bearings is, in general, used at low speeds and moderate loads, and therefore in application where the duty is less severe or less critical.

Self-contained bearings have a wide field of application in industry. They are capable of getting rid of the frictional heat generated by direct conduction, convection, and radiation to the surrounding atmosphere from the housing. A bearing of this class reaches thermal equilibrium from one to three hours after starting [11].
For heavy loads the equilibrium may be reached at a high level of temperature - 200 ^{o}F or more. Such high temperatures are not always acceptable for industrial use, because of the deterioration of the lubricant itself and the formation of

harmful acids due to a chemical breakdown. High temperatures also initiate softening of the bearing materials.

Therefore, many authorities [11], [30], and others, recommend the limit of acceptable equilibrium film operating temperature for usual industrial applications to range from 160 - 180 ^{o}F. and if there is any tendency to exceed these limits, an auxiliary cooling method should be used.

In self-contained bearings, it can be assumed, as a good approximation, that the heat generated by friction is entirely dissipated from the bearing housing surface to the surrounding air .

The rate of dissipation of heat $= K\ A\ \Delta t_W$ Btu/hr.

where;

K = the heat transfer coefficient, Btu/hr.^{o}F.ft^2,

A = the hot outside area of the bearing housing, ft^2,

and Δt_W = the temperature rise of the housing outside surface above ambient air temperature, ^{o}F.

The temperature gradient from the operating temperature of the lubricant to the outer surface of the bearing housing depends upon the method of oil supply to the bearing such as; oil supply by oil-rings, waste pack, wick, or by oil-bath .

Bearings supplied by oil rings or oil-bath have a copious supply of lubricant and there is not much difference in temperature along the circumference of the bushing as well as along the axial length of the bearing, due to axial distribution oil grooves that are usually used .

Bearings supplied by wick, waste-pack, or by drop-feed are somewhat starved and the bearing clearance is substantially empty. Consequently, the heat generation is more concentrated than for the other bearings fed with a copious supply of oil. Since, there is not enough supply of lubricant in those bearings and there is risky to heat concentration that tends to distort the bearing shell, the total heat must be transfered to the heat dissipating surface by conduction through the body of the bearing .

The class of non-pressurized journal bearings includes the following types of bearings that have been considered in the thesis for optimum design .

5-2.1 Oil-ring Bearings

The oil ring bearing plays an important part in the design of rotating machinary; they are probably more widely used than any other. The oil ring lubricated bearing is a self-contained unit, requiring very little attention or maintenance. Like all bearings, of course, it does have certain limitations to satisfy its function. Limitations in oil ring lubricated bearings are associated with high speed and high temperature .

The oil ring, as shown in figure 5-1, rides on the top of the shaft through a slot cut in the upper half of the bearing,while it passed below the bearing dipping into an oil reservoir to a depth of about one inch during operation. The oil ring rotates by friction with the rotating journal, so that it carries the oil up from the reservoir to the top of the journal. Spreader grooves are usually used for the distribution of oil from the slot along the length of the bearing.

On this type of self-contained bearing , loads should be vertical, since the slot must occupy about 180^{o} of bearing arc.

Ring design

In designing the oil ring, the main object is to maximize the quantity of oil delivered by the ring to the bearing. For a given ring cross section shape, the oil delivary increases as the ring speed approaches the rotational speed of the journal. The ring speed can be increased by increasing the friction between the ring and the shaft, that is, increasing the driving force on the ring , and this can be increased by increasing the total weight of the oil ring. Therefore, the oil ring must be made of a heavy material. Brass is recommended because of its high density and chemical stability plus good boundary-lubrication characteristics. For small rings sizes, die-cast zinc find a considerable use.

The oil ring is considerably larger in diameter than the journal. The inside diameter is about 1.5 times the

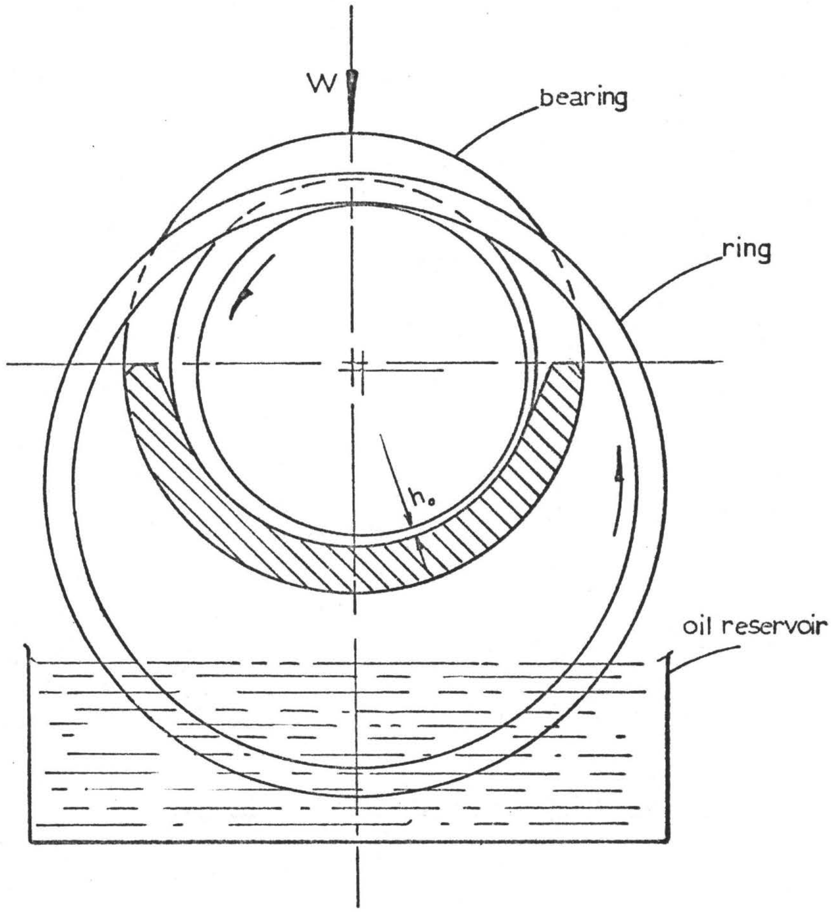


Fig 5_1   Oil ring representation

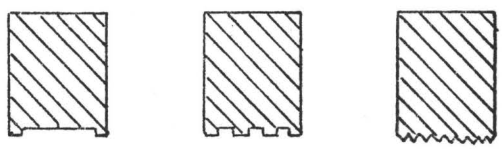
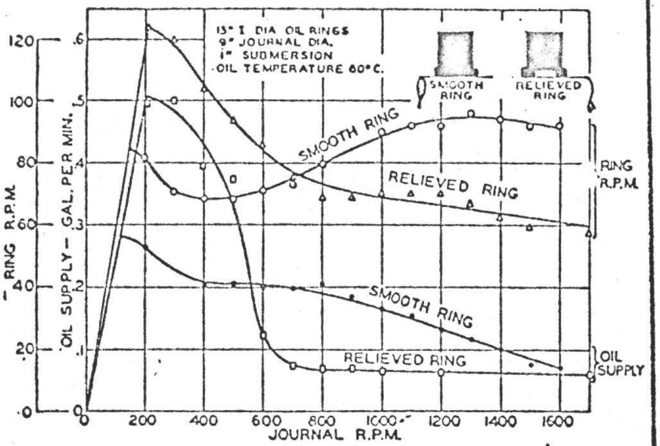
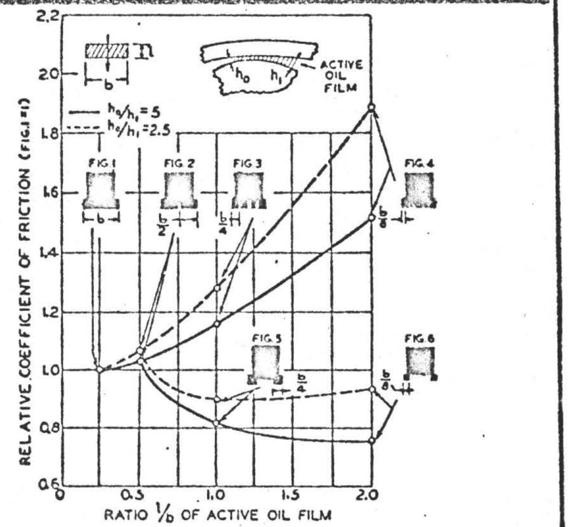


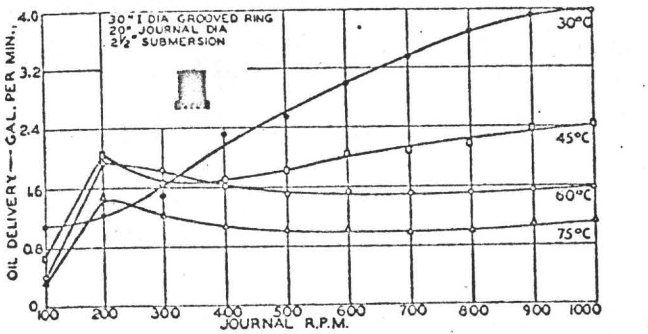
Fig 5_2   Some oil ring cross sections



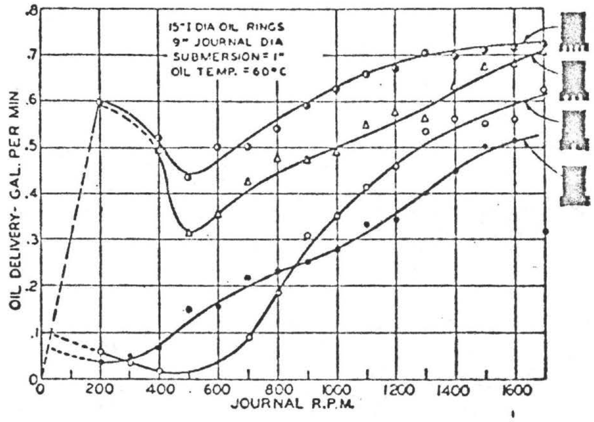
OIL-RING PERFORMANCE



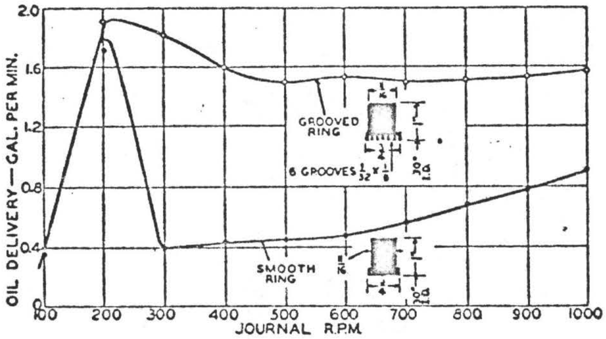
VARIATION OF COEFFICIENT OF FRICTION WITH DIFFERENT TYPES OF GROOVING



AMOUNT OF OIL DELIVERED AT VARIOUS TEMPERATURES AND JOURNAL SPEEDS

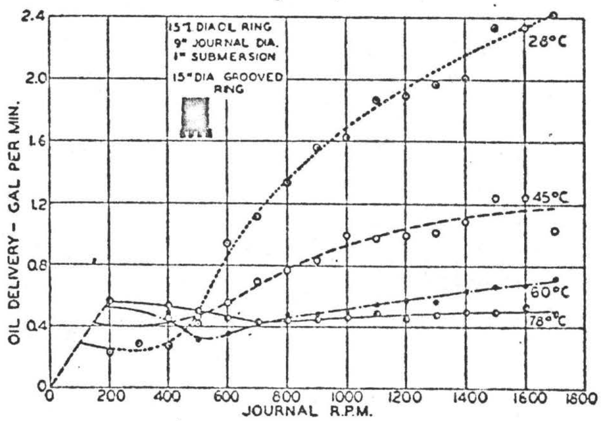


AMOUNT OF OIL DELIVERED BY RINGS WITH DIFFERENT NUMBERS OF GROOVES



AMOUNT OF OIL DELIVERED BY SMOOTH AND GROOVED RINGS AT VARIOUS JOURNAL SPEEDS

(Oil temperature = 60 C; submersion = 3 in.; viscosity = 15 centipoises.)



AMOUNT OF OIL DELIVERED AT VARIOUS TEMPERATURES

Fig 5-3

Experimental results explain oil ring performance, [4] journal diameter. The most suitable cross-section found is, in general, a rectangular one, where the width of the ring frequently from one half to one fourth the radial height.

Ring Grooves

Experimenters [4] , [11] , [40] , agree that shallow grooves on the inside surface of the ring increase the oil delivary, see figures 5-2 and 5-3 . These grooves hold a large quantity of oil on the inner surface of the ring and deliver it to the rotating journal.
The grooves are usually about 1/16" wide and from 1/32" to 1/16" deep [40]. They should be spaced from 1/16" to 1/32" apart .

Figures 5-3, indicate the performance of oil rings, from reference [4] .

5-2.2 Waste-Pack Bearings

This type of self-contained bearings is lubricated with oil-saturated waste . The waste is pressed against some part of the rotating journal and transfers oil to it by capillary action. Two best known bearings of this type are; waste-packed railroad journal bearings shown schematically in figure 5-4, and waste-packed electric motor armature bearings shown schematically in figure 5-5 .

The performance of a waste-packed bearing [18], is subject largely to the same hydrodynamical laws that govern

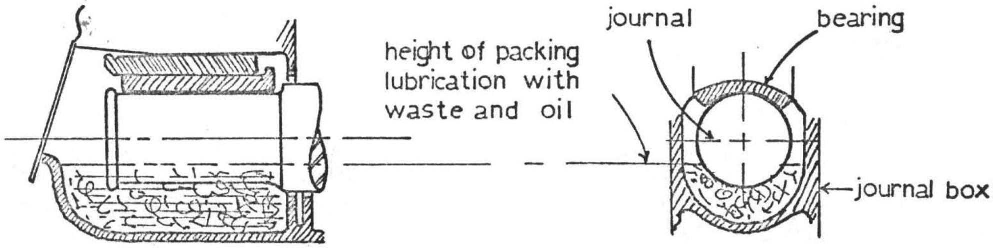


Fig 5-4 Railroad bearing in a standard journal box

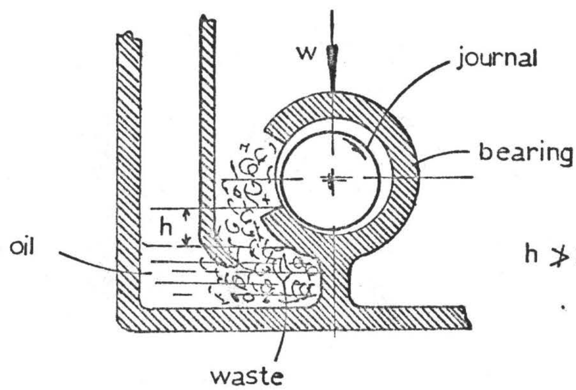


Fig 5-5 Waste-packed armature bearing

the operation of the perfectly lubricated bearing, but in contrast with it, the load in waste-packed bearings is carried partly by hydrodynamic pressure and partly by direct contact.

In waste-packed bearings a window is cut along the circomference in such a way to provide enough angle between the window and the line of action of the load under any conditions of service, so that an efficient load-carrying oil film can be stablished.

The oil-saturated waste packing should cover the whole window. A wick long enough to reach the bottom of the oil well can be placed across the window and extended so as to fill the waste chamber behind the wick with waste. This design, see figure 5-6, helps to completely seal the window, but the pack must be tamped very tight to force the wick against the journal and to prevent any loosening of the waste . .

Grooves

According to KARELITZ [18], "in general, the usefulness of grooves in waste-packed bearings for oil transmission along the bearing is doubtful, and that when properly designed and manufactured a waste-packed bearing will function just as satisfactorily without grooves.". Because the designer is not always certain whether the bearing is properly designed or properly manufactured, grooves should be used and they must be carefully chamfered or rounded to avoid disrupting the continuity of the oil film or scraping the oil from the journal. Moreover, it is very important to finish the surfaces of the journal and shell as well as possible . This increases the life of the bearings.

Material

Wool waste is recommended as packing material. Lamp wick or cotton thread should be used when the mechanical properties are of secondary importance, as in auxiliary oil drippers, syphon feeds,... etc.

5-2.3 Wick-Oiled Bearings

In wick-oiled bearinga, a wick is used to deliver the oil to the journal through a window in the top of the bearing. Oil flow by end leakage is replaced by oil fed through the wick material by capillary action. Figure 5-6 indicates a good way of providing good oil-storage capacity and oil retention features, [40]. In this design the packing and wick act as a filter for the oil passing to the bearing. In wick-oiled bearing, retention of the oil is quite important since only a small amount of oil can be stored.

The felt wick has a cylindrical shape. The packing used in the design shown in figure 5-6 is a soft felt or mass of fibers which is more deformable than the wick felt.

The packing has large pores and holds oil more loosely than the wick. It should fill the cavities around the wick to provide good oil storage and to decrease leakage and oxidation.

In some cases no wick is used and a long-wool-sliver packing is pressed against the shaft by a wick spring as shown in figure 5-7 .

Material

Although, the wick and packing materials each has a different function in oil delivery, experience has shown that both are best made from high quality wool fibers. These should not contain any alkaline or acidic materials in order to decrease the rate of oil oxidation. A packing material should have a good oil storage and absorption capacity.

Grooves

Operation of wick-oiled bearings is improved by oil grooves. The most effective type is the spiral groove on the shaft, but it must be used for a single direction of rotation and it is costly. An X-shaped groove is used on the bearing surface. The groove leads diagonally away from the bearing window and stops short of the ends of the bearing.

5-2.4 Oil-Bath Bearings

In this type of bearings, the journal is partly submerged in an oil reservoir as shown in figure 5-8.
This method of oil supply is particularly suited to bearings

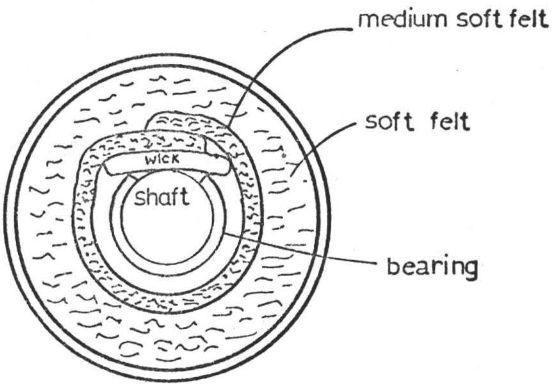


Fig 5-6 Roll shaped wick with soft felt packing

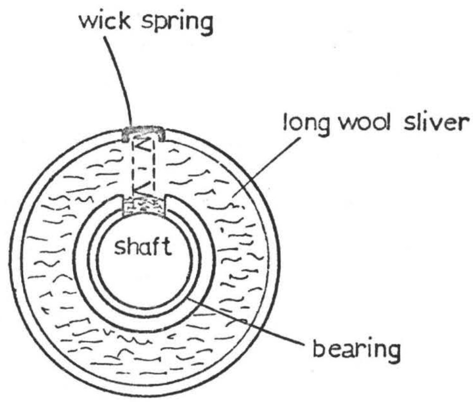


Fig 5-7 Long-wool sliver packing

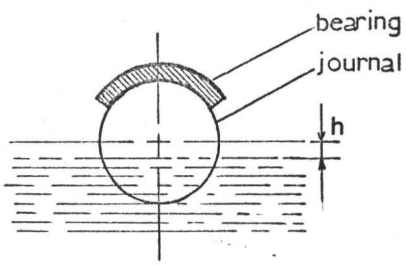


Fig 5-8 Oil bath representation

that carry the load on the top half.

Design procedure of oil bath bearings is considered the same as that of oil ring bearings.

5-3 Pressure - Fed Bearings

In pressure-fed bearings the lubricant is fed to the bearing under a positive pressure. The primary function of the pressure feed of the lubricant in this class of bearings is to ensure its reliability by a positive feeding of the lubricant at all times.
The lubricant is fed to the bearing not only at a given pressure, but also at a controlled temperature. Feeding the lubricant at a controlled pressure and temperature, then, provides more effective cooling to the bearing and uniformity in its performance, specially those bearings that work under heavy duty conditions.

This type of bearing is commonly used in modern high speed turbines, compressors, and gears as well as in heavy large machinery and internal combustion engines.

Heat generated in the bearing oil film is dissipated partly by the surface of the bearing housing and partly by the oil circulating through the bearing. However, the amount of heat dissipated by the surface of the bearing housing is relatively small in comparison with that generated in the bearing . Because of this, it is a good approximation to assume that all heat generated by friction is removed by lubricant cooling. Accordingly,

$$\text{The rate of heat dissipation} = Q.\ C_p.\ \Delta t \quad \text{Btu/hr.}$$

where; Q is the total oil flowing through the bearing, Gpm,

C_p is the heat capacity of lubricant, Btu/gal.^{o}F,

and Δt is the oil temperature rise, ^{o}F .

Pressure-fed bearings include two main types; split cylindrical bearings, and full cylindrical bearings.

5-3.1 Split Cylindrical Bearings

The split cylindrical bearing is one of the most common types of bearings. It is commonly used in the field of internal combustion engines.

An optimum design solution for a split bearing of the type illustrated in figures 5-9 and 5-10 is available. This bearing, as shown, is split into two halves and has a cylindrical bore. The bearing has large axial oil-distribution grooves located at the split, each oil groove subtends an angle of, approximately, 30 degree.

5-3.2 Full Cylindrical Bearings

For simplicity and rigidity a fullcylindrical bearing is, generally, used. The bearing is normally made in one cylindrical piece, although, it is sometimes made from two halves without chamfering the edges . This type of pressure-fed bearings finds its application in light machinery and where the direction of the load cannot be predicted.

Figures 5-11 and 5-12 illustrate two types of this category of bearings. These are the crcumferential-groove, and the single oil-hole type. In the first type, the lubricant is fed under pressure into a circumferential groove midway along the bearing length. Therefore the lubricant is induced to run completely around the bearing. In order to prevent oil circulation within the groove, a small dam at one point in the groove is used. In the second type, the oil is fed through a hole at the top of the bearing. The rate of lubricant flow through this type is smaller than that through a comparable bearing with a circumferential groove. Therefore it is used where the cooling effect is not a critical factor.

Section 5-4, gives the rate of oil flow through a pressure-fed journal bearings .

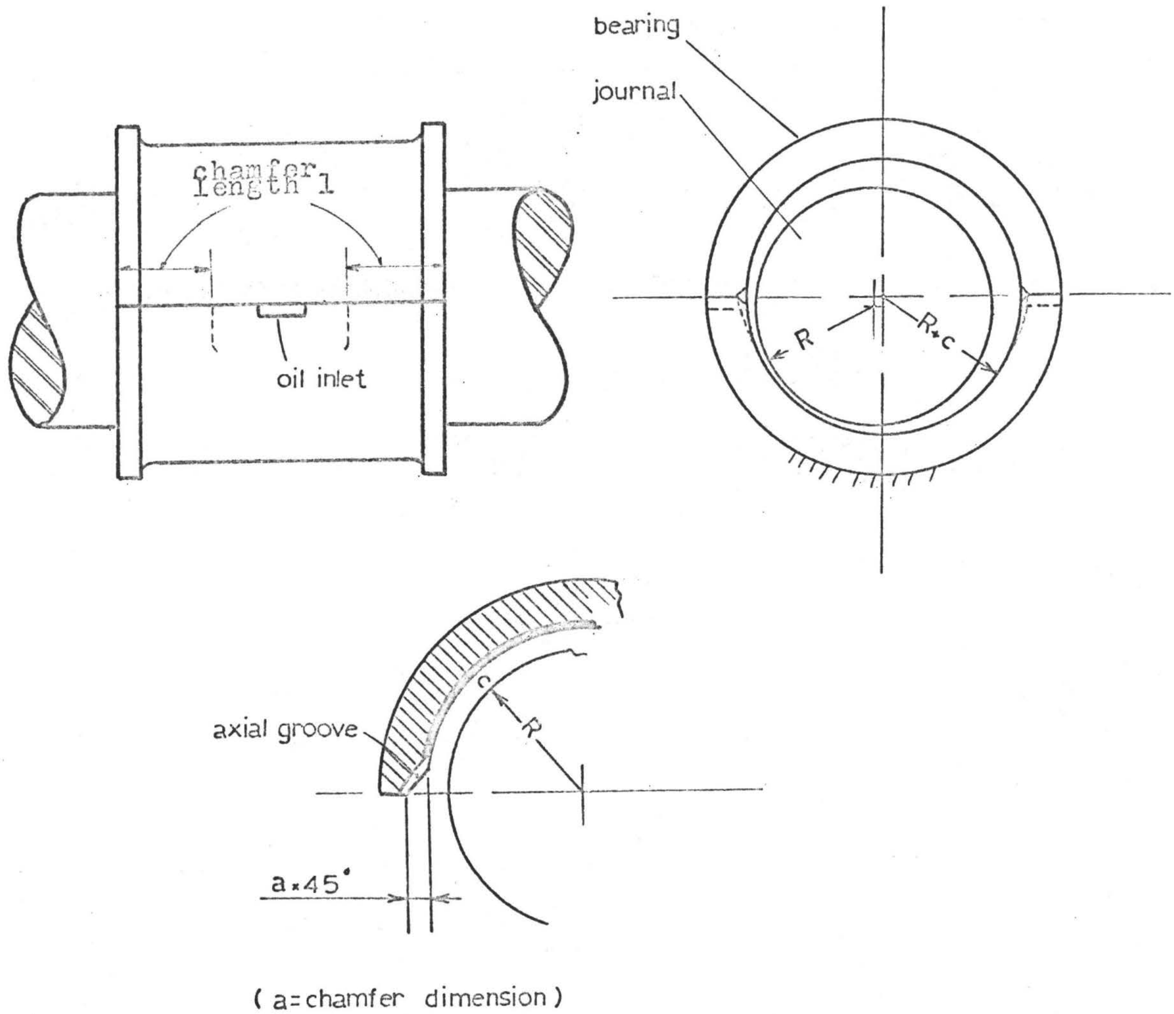


Fig 5_9 Cross section at chamfer opening
for split cylindrical bearing

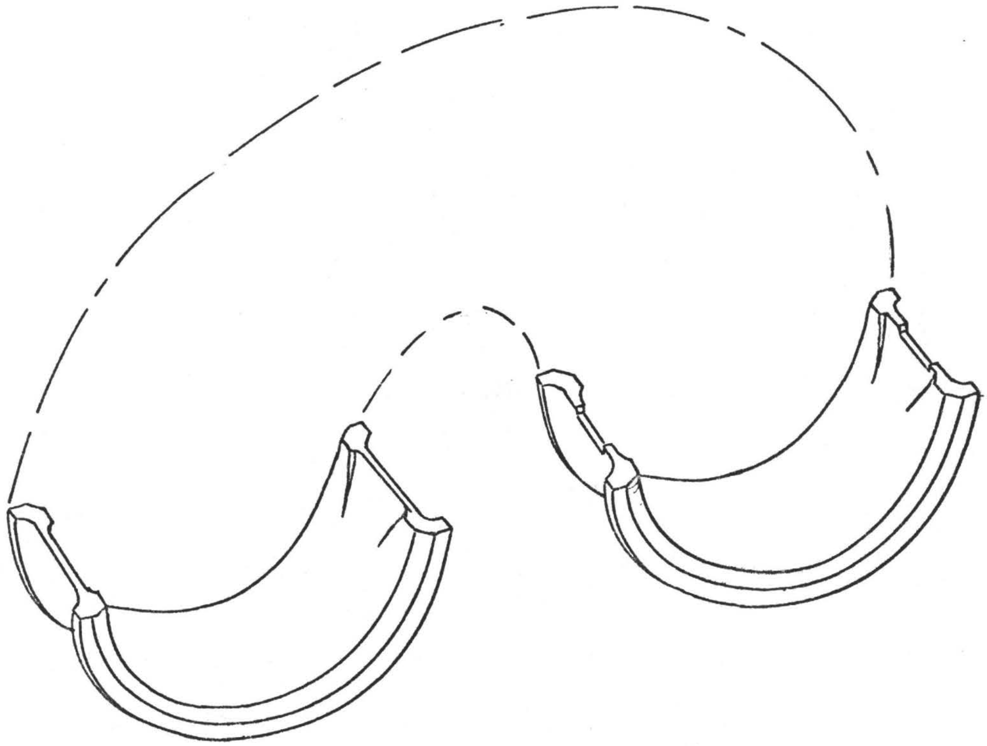


Fig 5.10 Lower and upper half of split bearing with axial oil grooves

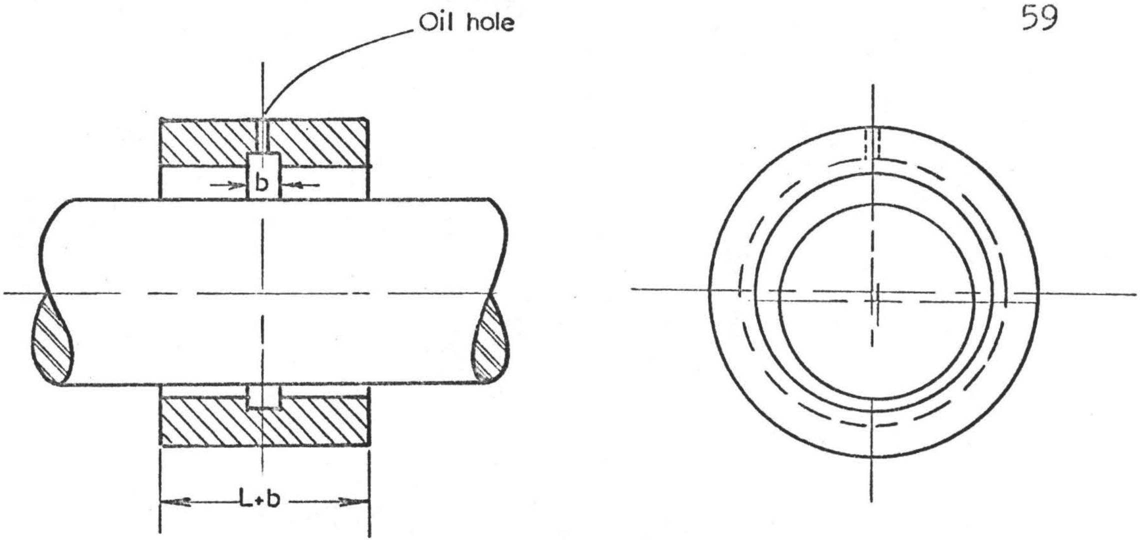


Fig 5_11 Full_cylindrical journal bearing with a circumferential oil groove

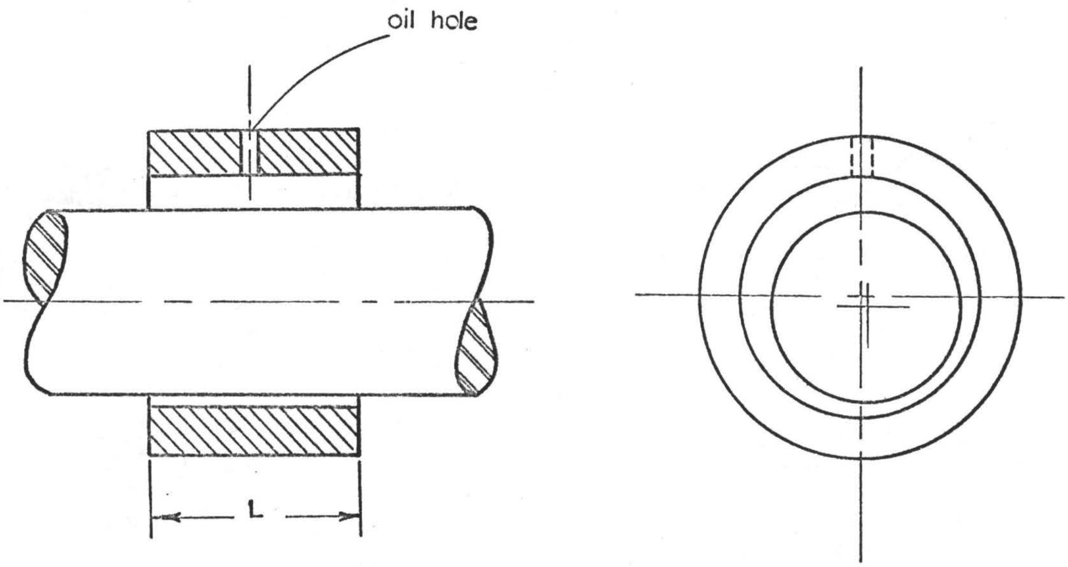


Fig 5_12 Full_cylindrical journal bearing with a single oil hole

5-4 Oil Flow Through a Pressurized Journal Bearing

As will be mentioned in chapter VII, the heat dissipation, and consequently the thermal equilibrium of pressurized journal bearings, depends primarily upon the quantity of oil flow through the bearing. So in order to predict the bearing performance, the oil flow must be known .

According to Wilcock and Murray Rosenblatt [41], there are two kinds of oil flow. The first is called Zero-speed flow (Q_o) and the second is Hydrodynamic-flow (Q_h) . Zero-speed flow depends upon the oil inlet pressure in the oil feed grooves in the bearing, whence it occurs whether the journal rotates or not. It consists of two kinds, that oil which flows through the bearing clearance space (Q_f), and that oil which flows through the chamfer openings(Q_c) in split type of bearings, see figure 5-9.
Hydrodynamic oil flow, on the other hand, is the oil flow out of the bearing ends (end flow) that results from the generating hydrodynamic pressures within the bearing.

5-4.1 Oil Flow Through a Split Journal Bearing with Axial Groove, [41]

A- The oil flow through the bearing clearance is given by the following formula :

$$Q_f = 4.6 \times 10^6 \frac{C^3 \; P_1}{z} \qquad \text{Gpm.} \qquad (5\text{-}1)$$

where; C = radial clearance, inch.

P_1 = oil inlet pressure, psi.

z = oil viscosity, centipoise.

B- Flow through the chamfer openings is given by :

$$Q_c = i \cdot \xi \cdot Q_c^o \quad \text{Gpm.} \qquad (5\text{-}2)$$

where; i = number of chamfer openings.

Q_c^o = the uncorrected chamfer flow, gpm.

ξ = correction factor for chamfer flow.

The uncorrected chamfer flow is given by

$$Q_c^o = 47200 \frac{(a + C)^4}{lz} P_1 \quad \text{Gpm.}$$

where; a = chamfer dimension, inch.

l = axial length of chamfer, inch.

C = radial clearance, inch.

P_1 = oil inlet pressure, psi.

z = oil viscosity, centipoise.

The correction factor for chamfer flow ξ is given as a function of the uncorrected chamfer flow Q_c^o as follows

$$\xi = \frac{-1 + \sqrt{1 + 4A1}}{2A}$$

where; $A = Q_c^o \cdot (40/lz)$

The chart in figure 5-13 gives the correction factor as a function of Q_c^o and lz .

C- Hydrodynamic oil flow can be determined from the following equation

$$Q_h = \frac{N\,R\,L\,C}{73.5}\ q \qquad \text{Gpm.} \qquad (5\text{-}3)$$

where; N = shaft rotational speed rpm.

R = journal radius, inch.

L = bearing length, inch.

C = radial clearance, inch.

q = oil-flow coefficient, the chart in figure 5-14 gives q as a function of the eccentricity ratio (ϵ) and L/D ratio, [40] .

The total oil flow is

$$Q = Q_f + Q_c + Q_h \qquad \text{Gpm.}$$

5-4.2 Oil Flow Through a Full Cylindrical Bearing with Circumferential Groove [40]

A- The oil flow Q_f is given by the following relation

$$Q_f = 3.76\ 10^6\ \frac{R\,C^3 P_1}{Lz}\,(1 + 1.5\,\epsilon^2) \qquad \text{Gpm.} \quad (5\text{-}4)$$

where, L, in this equation is the total active length of the bearing (see figure 5-11).

B- Hydrodynamic oil flow is given by

$$Q_h = \frac{N\,R\,L\,C}{73.5}\ \bar{q} \qquad \text{Gpm.} \qquad (5\text{-}5)$$

where , in this case, $\bar{q}$ is an oil coefficient defined as a function of $S\ (L/D)^2$. The chart in figure 5-15

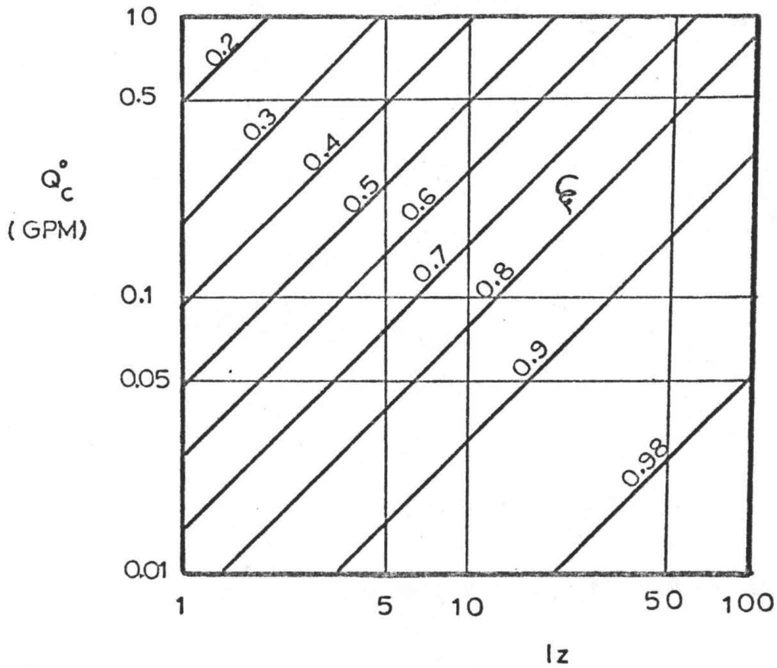


Fig 5_13 Chamfer flow correction factor ξ as a function of Q_c° and Iz

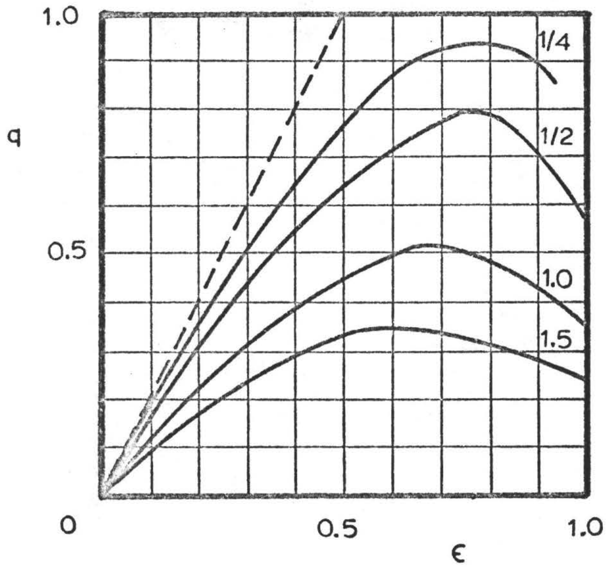


Fig 5_14 Oil flow cofficient q for different L/D ratio

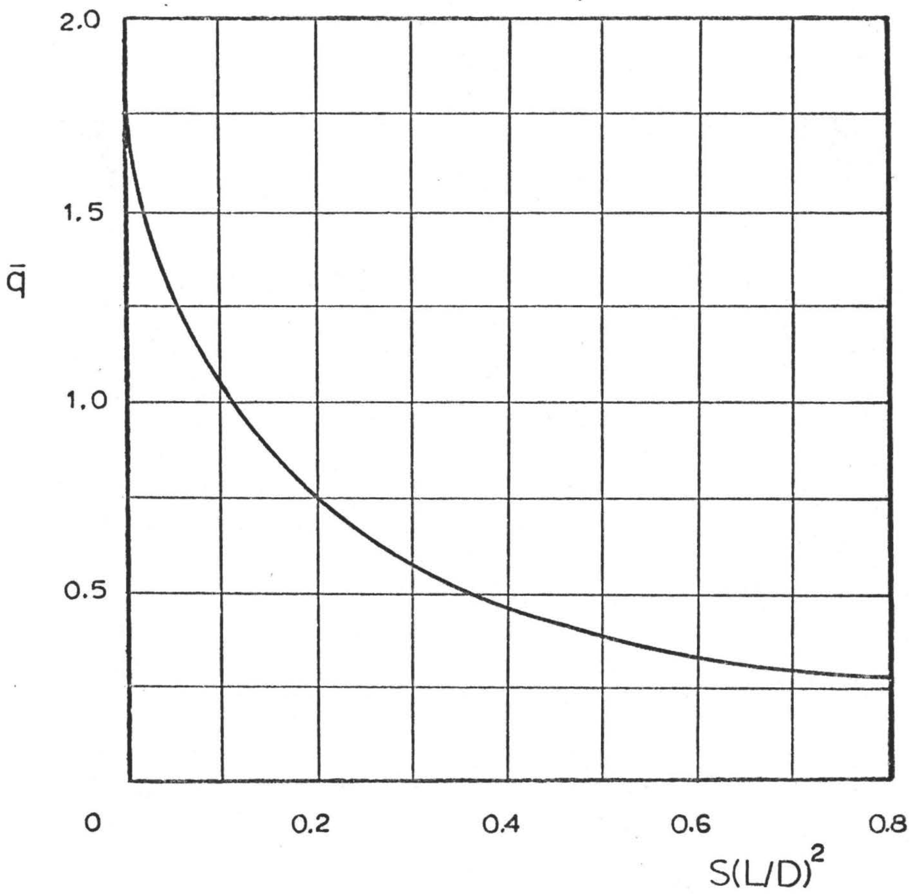


Fig 5_15 Coefficient of oil flow as a function of S(L/D) for all L/D ratios

gives $\bar{q}$ as a function of $S\,(L/D)^2$ and L/D, where

S is the dimensionless Sommerfeld number .

Then the total flow is

$$Q = Q_f + Q_h \quad \text{Gpm.} \qquad (5\text{-}6)$$

5-4.3 Oil Flow Through a Full Cylindrical Bearing with Single Hole, [30]

The oil flow rate through a bearing fed by a single oil hole is expressed by the following equation

$$Q_f = 5.97\times 10^5 \frac{C^3 P_1}{z} \tan^{-1}\left(\frac{2\pi R}{L}\right) (1 + 1.5\,\epsilon^2) \quad \text{Gpm.} \qquad (5\text{-}7)$$

The hydrodynamic oil flow in this case is considered to be equal to the end flow [40] , and whence the total flow will be

$$Q = 2\,Q_f \quad \text{Gpm.} \qquad (5\text{-}8)$$

5-5 Design Procedure

The design procedure consists mainly of two parts. The first part is the determination of the frictional energy loss and the same principles apply for both class of bearings mentioned in sections 5-2 and 5-3. The second part in the design procedure is the thermal equilibrium of the bearing. Here, the problem is different; in self contained bearings the heat balance is based upon the assumption [11], that the heat generated by friction is dissipated entirely from the housing surface to the surrounding air. While in pressure-fed bearings, the heat balance is based upon the assumption that all heat generated in the oil film is carried away by the oil flowing through the bearing.

A- Friction Energy Loss

In section 4-4, we discussed Dennison's design chart figure 4-2 . From that chart is taken the torque number T' for the appropriate L/D ratio.
The actual friction torque is given by equation (4-4) as

$$M = T' \frac{z\ R^3 L\ N}{6.9 \times 10^6\ C} \qquad \text{lb.inch}$$

Having M , then the horsepower loss is

$$F.H.P = \frac{M\ N}{63000.0}$$

and the frictional energy loss will be

$$H_g \quad = \quad F.H.P \times 2545 \qquad Btu/hr \qquad (5\text{-}9)$$

The next step is the heat balance.

B-1 Heat Balance in Self-Contained Bearing

From the preceding chapters, we know that the operating characteristics of a bearing, such as the load-carrying capacity, minimum film thickness, and friction loss depend upon the viscosity of the lubricant. The viscosity of a lubricant, however, is a function of the temperature of the film. Therefore, with the exception of bearings running at low speeds, the heat balance in the bearing is an important part of the analysis of any bearing operating with fluid film lubrication . The operating temperature of bearings is also of interest because the fatigue strength of a bearing material is a function of its temperature. Also, an increase in the bearing shell temperature causes temperature expansion of the elements of the bearing, with a change of bearing clearance, and possible high stresses in the bearing metal due to differential expansion .
Heat generated in a bearing with fluid-film lubrication is due to the viscous friction in the oil film and is given by equation (5-9) in Btu/hr .

The amount of heat carried off by conduction through the shaft or through the pedestal [30], is usually small when compared with the amount of heat generated in the bearing. Therefore, assumption that the entire heat generated in a self-contained bearing is dissipated from the bearing surface to the surrounding air will yield satisfactory results in bearing analysis, hence,

The rate of heat dissipation is

$$H_d = K \, A \, \Delta t_w \qquad \text{Btu/hr} \tag{5-10}$$

where;

- H_d is the rate of heat dissipation, Btu/hr
- K is the heat transfer coefficient, $\text{Btu/hr.ft}^2.^\circ\text{F}$
- A is the hot outside area of the bearing housing, ft^2
- Δt_w is the temperature rise of this outside surface above ambient air temperature, $^\circ\text{F}$

The dissipation of heat from the outer surface of the bearing housing depends upon the velocity of the ambient air moving across the surface .

Karelitz [20], estimated these values, these are:

$$K = 2.1 \qquad \text{Btu/hr.ft}^2.^\circ\text{F}$$

for quiet air, and

$$K = 5.9 \qquad \text{Btu/hr.ft}^2.^\circ\text{F}$$

for air with a velocity of 500 fpm.

The surface area of the bearing housings to be considered effective as a heat-dissipating area may be estimated in average as, 15 (L.D), where L is the total length of the bearing in inches and D is the journal diameter in inches.

A chart has been made by Fuller [11], based on measurements of temperature gradients in typical bearings, figure 5-16. Three types of bearings are included, they are; waste packed, oil ring, and oil bath. A range is indicated for each type between the limits of still ambient air and moving ambient air surrounding the housing . For the wick-fed bearings, it was assumed that the temperature gradients will behave as in the waste-pack bearings .

From the heat balance, by equating equation (5-9) and equation (5-10)

$$H_g = H_d$$

i.e

$$\text{Btu/hr} = K\ A\ \Delta t_w \tag{5-11}$$

The temperature rise Δt_w, of the outside surface of the bearing above ambient air temperature, can be obtained. Knowing Δt_w, the temperature rise of the oil film above housing wall (Δt_o) is determined from the chart of figure 5-16, whence the oil operating temperature will be

$$T_o = T_1 + \Delta t_w + \Delta t_o$$

where T_1 is the ambient temperature , ^oF.

B-2 Heat Balance in Bearings with Forced-Fed Lubrication

In a bearing lubricated under pressure, heat generated in the oil film is partly dissipated by the surface of the beating housing and partly removed by the oil circulating through the bearing. The temperature of the oil returning from the bearing into the sump is therefore higher than the temperature of oil supplied to the bearing. The difference in temperature of the oil flowing in and flowing out of the bearing is denoted by Δt . The lubricant in the sump must be cooled before it is pumped again to the bearing.

In a bearing with oil supplied under pressure, the temperature of the body of the bearing is usually relatively low, and the amount of heat dissipated by the surface of the bearing housing is relatively small compared with the amount of heat generated in the bearing. For this reason it may be assumed that the entire heat generated in the bearing is removed by the flow of the lubricant through the bearing. Hence, the rate of heat dissipation is

$$\text{Btu/hr} = Q.C_p.\ \Delta t \qquad (5-12)$$

where;

Q = total amount of oil flowing through the bearing, Gpm.

C_p = heat capacity of lubricant in Btu/gal. $^{\circ}F$

Δt = oil temperature rise, $^{\circ}F$.

then, the oil operating temperature, in this case, will be

$$T_o = T_{oil} + \Delta t$$

where T_{oil} is the oil inlet temperature, $^{\circ}F$.

In order to get Δt from the thermal equilibrium of equation (5-12) the quantity of oil flow through the bearing

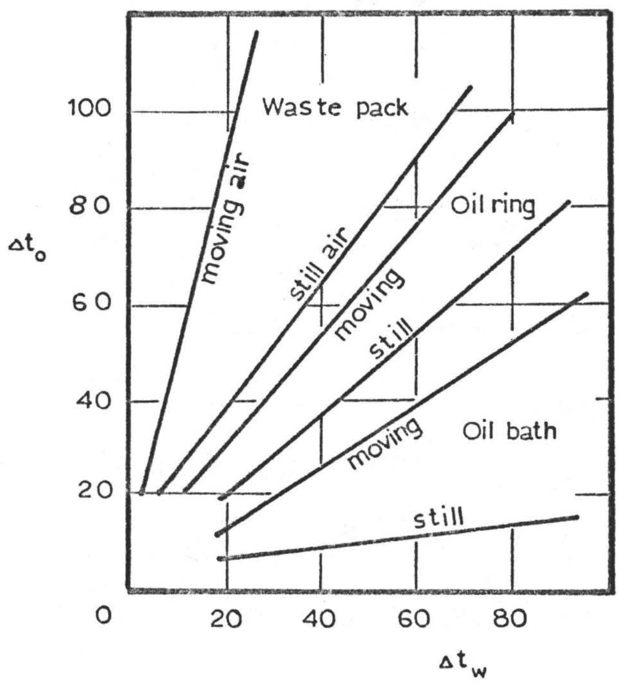


Fig 5_16 Temperature gradients Δt_w vs Δt_o

(Q), must be known. Section 5-4 , gives this quantity for each type of bearing in this class .

VI PRACTICAL CONSIDERATIONS IN BEARING DESIGN

6-1 Introduction

In the derivation of the hydrodynamic theory, in previous chapters, many assumptions were made. Actually, these assumptions are not quite true and, accordingly, no succesful design based upon these theoretical assumptions can be guaranteed under all conditions, and the conclusions will be only true within certain limits and under certain conditions .

In the next sections, some practical considerations concerning bearing design will be explained . Most of these considerations are based upon practical engineering experience while some of them are supported theoretically. References [9] , [10] , [30] , and [40] discuss in detail these considerations .

Practical considerations, however, have helped us in the selection of the limiting conditions used in the design of bearing.

6-2 Length- to -Diameter Ratio L/D

Actually, there is no general rule govern the best length to diameter ratio, and in selecting this ratio many considerations are important. Both large and small length to diameter ratios have their own advantages and disadvantages, however, a ratio close to one appears to be the optimum. The ratio could be increased when the shaft is rigid and no deflection problem exists, good alignment conditions are available, and cooling the bearing is not a problem. Otherwise, it should be decreased.

Limits of L/D ratio have been set according to the application such as, generators, machine tools,.. etc. Four kinds of applications have been selected and they are available for the designer.
Table 6-1 , shows the common values of L/D ratios for these applications in accordance with current practice .

TABLE 6-1

L/D ratios, Current practice

Machine	L/D
Turbo-Generators	0.8 - 1.8
Steam turbins	0.8 - 2.0
Generators & Motors	1.0 - 2.5
Machine tools	1.5 - 4.0

Due to limitations in design data, the maximum limit in table 6-1 has been considered not more than two .

6-3 Clearance in Journal Bearings C

One of the important factors in bearing performance is the radial clearance. Under certain operating conditions there is a range of clearances that provides the optimum performance of the bearing. Load carrying capacity and minimum film thickness, represented by equations (3-19) and (3-20) respectively, are affected directly by the value of C and consequently so are the eccentricity ratio ϵ , frictional loss, oil flow, and the operating temperature. Experience in bearing design provides the designer with average clearance values that help in a preliminary selection . Table 6-2, from reference [34] , gives average values for the clearance as a function of bearing metal. While Fuller [11] recommends "medium fits" for journals running at speeds under 600 rpm, while above 600 rpm "free fits" may be used. Kingsbury has suggested a radial clearance for bearings equal to $0.001 + 0.001\,R$, where R is the journal radius. For high speed journals the radial clearance is usually set equal to $0.001\,R$ to decrease the frictional losses and to increase the flow of oil through the bearing.

TABLE 6-2

Radial clearance, Current practice

Bearing alloys	C/R
Tin - base babbitt	0.0005
Cadmium silver	0.0008
Copper - lead	0.0010
Silver - lead - indium	0.0010
Aluminum	0.0010

From Shaw and Macks [34]

6-4 Minimum Film Thickness

The load carrying capacity of a bearing as shown from equation (3-19) depends upon the eccentricity ratio which, in turn, depends upon the minimum film thickness as given by equation (3-20) . According to the theoretical formula of equation (3-19), the load carrying capacity of a bearing increases with an increase of the eccentricity ratio ϵ , that is, with a decrease of the minimum film thickness h_0. Consequently, the load carrying capacity will increase infinitely if the film thickness is reduced to zero.
Thus the minimum film thickness is one of the factors governing the load capacity of a hydrodynamically lubricated bearings .
The following is a list of recommended minimum allowable film

thickness, $(h_o)_{all}$, given by various authors and based upon their experiments and experience in the bearing design field.

Karelitz & Kenyon [19] :	It is of the order of 0.00005"
McKee [23] :	It is of the order of 0.00005"
Fuller [11] :	The lower limit for commercial type bearings is approximately equal to 0.0001" .
Slaymaker [33] :	Suggested a value always greater than 0.0001" .
Black & Paul [5] :	It is of the order of 0.0002" per inch of diameter of the journal .
Karelitz [20] :	In actual bearings it is of the order of 0.0001" for finely bored small bronze bearings . It is at least 0.00075" for ordinary operating conditions of babbitted bearings running at high speeds; and not less than one-half this value in all other cases .
Vallance & Doughtie [39] :	It should be; 0.0001" for fine-finished bronze bearings. 0.00075" for ordinary babbitted bearings .

0.003 - 0.005" for bearings with large steel shafts.

Wilcock & Booser [40] : It should be not less than 0.0001 - 0.00015" .

In general, afull flow filtration of the lubricant is recommended specially for highly loaded bearings fed under pressure with a small minimum film thickness .

6-5 Bearing Pressure

Experience shows that the allowable bearing pressure should be selected for a given bearing on the basis of a complete analysis concerning the values of the minimum film thickness, temperature, viscosity, speed, clearance, and bearing metal characteristics. No direct relation has been found between the bearing unit load and its fatigue life, because the latter is influenced by factors other than the unit load .

Etchells and Underwood [10] give maximum values of unit load for various bearings metals for engine bearings. These values are shown in Table 6-3, where the pressures are based on a fatigue life of 500 hours at a bearing temperature of 300 ^{o}F .

TABLE 6-3

Maximum values of load per projected bearing area

Bearing Metal	Maximum unit load, psi
Lead - base babbitt	600 - 800
Tin - base babbitt	800 - 1000
Cadmium - base bearing metal.	1200 - 1500
Cadmium - base with 0.003 to 0.004" overlay of babbitt	2000 - 4000
Copper - lead (Pb,45% ; Cu,55%)	2000 - 3000
Copper - lead (Pb,25% ; Sn,3% ; Cu,72%)	3000 - 4000
Silver (lead - indium overlay)	5000 and up
Bronzes	10000

From Radzimovsky [30]

6-6 Sommerfeld Number and zN/P

The relationship between Sommerfeld number, obtained theoretically and experimentally, and the coefficient of friction quantity (R/C)f is shown in figure 6-1 . While figure 6-2 shows atypical experimental relationship between the coefficient of friction and the zN/P .

It has been found in practice that the bearing operates with fluid-film lubrication if the Sommerfeld number and the symbol group zN/P are of such values in the region to the right of point B and the friction, therefore, will depend

upon the viscosity of the lubricant. Whereas between B and C, figures 6-1 and 6-2, the film becones so thin that the surface high points begin to touch and boundary operating conditions will exist.
Therefore, the portion of the curve between B and C represents the transition from thick to thin film lubrication. Leftward from point C, any decrease in Sommerfeld number or the symbol group zN/P, will cause a very rapid increase in the coefficient of friction f. The friction then depends not only on the viscosity, but also upon the journal and bearing materials used as well as upon the degree of surface roughness and rigidity of the rubbing surfaces.

Shaw and Macks [34], recommend minimum values of Sommerfeld number and zN/P as given in Table 6-4.

TABLE 6-4

Recommended minimum values of S and zN/P

Bearing Metal	zN/P	S
Tin - base babbitt	20.00	0.050
Lead - base babbitt	10.00	0.025
Cadmium - base alloy	3.75	0.009
Copper - lead	3.75	0.009
Silver - lead - indium	2.00	0.005

From Shaw & Macks [34]

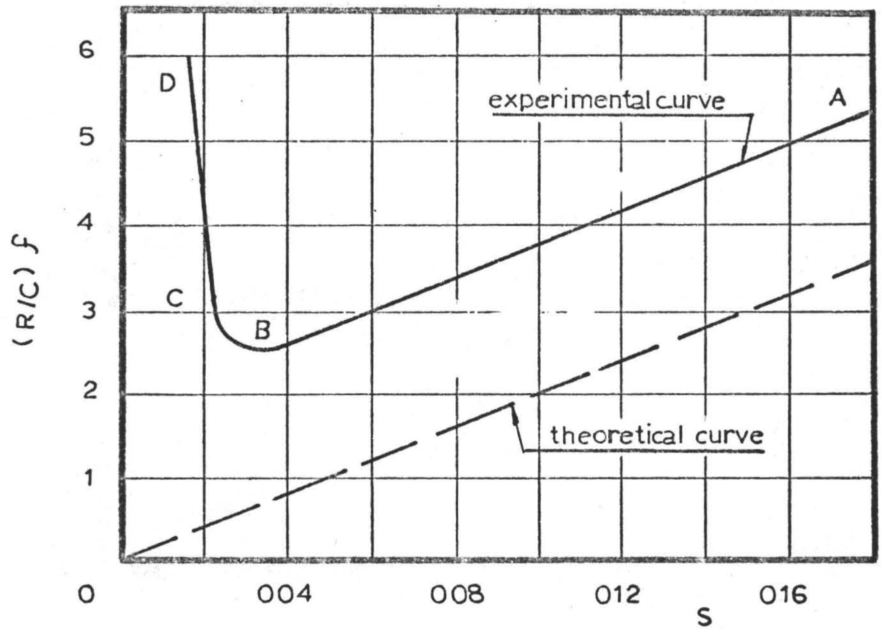


Fig 6_1 Sommerfeld number against the value (R/C) f

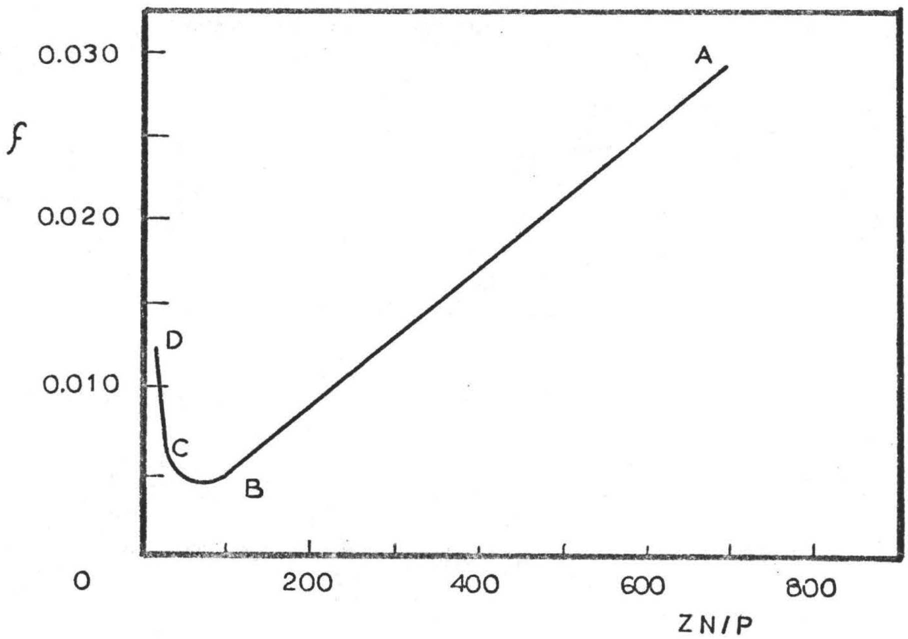


Fig 6_2 Value of f against ZN/P

6-7 Turbulence

turbulence in the lubricant film is a phenomenon that occurs at high journal surface velocities.
The theoretical analysis of the bearing performance of this thesis is based on the assumption that the flow of lubricant in the clearance space is laminar or streamline , in which , the lubricant flows in parallel or concentric layers having a relative velocities to each other, but no mixing between them. When the velocity of the flow reaches a certain upper limit, turbulent flow may occur in the oil film. This causes increase in the power loss, increase in the operating temperature, and decrease in the rate of flow.

The critical flow velocity at which turbulence is initiated, in a cylindrical tube, can be determined by the dimensionless "Reynolds number", that equals

$$\frac{V \cdot d}{\nu}$$

where; V is the linear flow velocity,

d is the tube inside diameter, and

ν is the kinematic viscosity .

In tubes and pipes, turbulence generally begins when Reynolds number is about 2000. In journal bearings the following expression for Reynolds number, suggested by Wilcock [42], is applied

$$\frac{\pi D N' C}{\nu} = 41.1 \sqrt{\frac{R}{C}} \quad \ldots\ldots\ldots\ldots (6\text{-}1)$$

From equation (6-1), the critical speed at which turbulence in the bearing start to occur is given by

$$N_{cr} = 392.4 \frac{\nu}{R.C} \sqrt{\frac{R}{C}} \quad \text{rpm.} \qquad ......(6\text{-}2)$$

where;

N_{cr} = critical shaft speed, rpm.

ν = kinematic viscosity, in^2/sec.

C = radial clearance, in.

R = radius of journal, in.

VII OPTIMUM DESIGN OF HYDRODYNAMIC JOURNAL BEARINGS

7-1 Introduction

The overall purpose of this chapter is to present the optimum design problem, its objective function, geometrical parameters, and its constraints.

The bearing design procedure presented here, is based upon Dennison's design chart, explained in chapter IV, in determining the frictional torque. While the optimum design objective function and the independent geometrical parameters are found by one of the optimization subroutines, adapted from OPTISEP [31] .

The optimum design of a hydrodynamic journal bearing will be considered on the basis of minimizing either the frictional torque, or the oil temperature rise, or both .

7-2 Definition of the Problem

As mentioned earlier, a computer aided design package is to be designed for determining an optimum solution for hydrodynamic journal bearings.

The following classes of bearings have been considered.

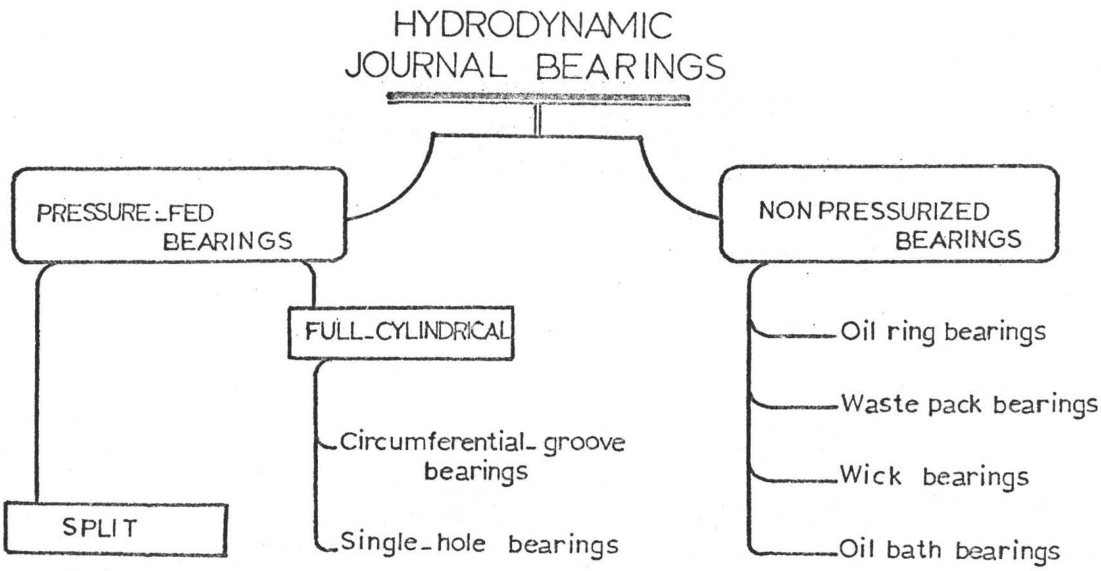
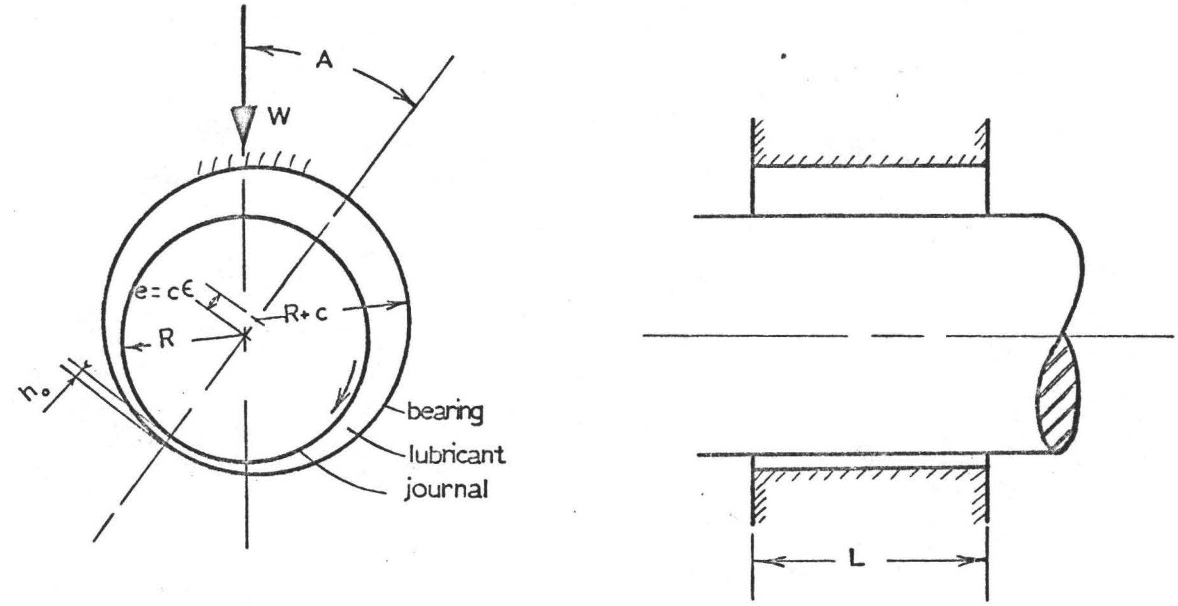


Figure 7-1 shows the general configuration of a bearing, without provision of oil supply. The different geometrical variables are indicated .

7-3 Optimum Design

The optimum design of a mechanical element in engineering work is the selection of the material and determination of the optimum values of the independent geometrical parameters in order to either minimize an undesirable objective or maximize a functional requirement.
The design procedure is conducted so as to satisfy other functional requirements and to keep other undesirable effects within tolerable limits .

Optisep [31], is a package of subroutines for performing such an optimum solution .



L	=	Bearing length, inch
R	=	Journal radius, inch
C	=	Radial clearance, inch
e	=	Eccentricity, inch
ϵ	=	Eccentricity ratio
h_o	=	Minimum film thickness, inch
A	=	Attitude angle, degrees
m	=	Eccentricity modulus $= \frac{1}{1-\epsilon} = \frac{c}{h_o}$

Fig 7_1 Typical hydrodynamic journal bearing in steady state position

7-4 Optimization Criterion

The optimum design for a hydrodynamic journal bearing is conducted on the basis of minimizing either of the following significant quantities .

A- Frictional torque loss.

B- Oil-temperature rise.

C- Both criteria, A and B, if the input torque in lb.inch is known to the designer. This case will be explained in section 7-5.4 when setting up the optimization function.

7-5 Formulation for Optimization

7-5.1 Input

In general, the designer usually knows the bearing load (the reaction at the bearing), the rotational speed of the shaft, and the ambient temperature. He might decide upon the shaft size or oil inlet pressure. However, the designer is not restricted to specifying either the shaft size or oil inlet pressure, and is allowed to set up either or both as a design variable.

For all kinds of bearings given in section 7-2, the designer is essentially required to give the following input data .

1- Load at bearing in pounds

-If the load is not stationary, the theory of bearings has proved experimentaly [33], that such a bearing with a rotating load at the same shaft speed has the same load capacity as a bearing with a constant directional load. A reciprocating load can probably be approximated by an average load.

2- Shaft rotational speed in revs/min.

3- Design operating ambient temperature, $^{\circ}F$.

4- Maximum expected ambient temperature, $^{\circ}F$.

In addition to the abovementioned input data, a set of code names (see user's manual, Appendix 1) are used to define user's options.

7-5.2 Design Variables

The maximum number of design variables that can be set, including geometrical and non geometrical variables, is as follows :

1- L = length of bearing, inch

2- C = radial clearance, inch

3- z = oil operating viscosity, centipoise

4- R = radius of journal, inch

5- P = oil inlet pressure, psi

6- T = oil inlet temperature, $^{\circ}F$.

7- l = axial length of chamfer, inch

8- a = chamfer dimension, inch

The last two variables belong to the split type of journal bearing. In addition to those independent design variables, there are also the following dependent variables.

9- oil operating temperature, ^{o}F.

10- minimum film thickness, inch

11- eccentricity ratio

12- attitude angle, degrees

13- frictional torque, lb.inch

14- amount of oil to be supplied, gpm.

15- grade of lubricating oil, SAE number

16- Bearing metal

7-5.3 Optimization Function

The primary design criterion may be one of the following :

A- Minimum frictional torque loss.

In this case the frictional torque is given by equation (4-4), as a function of the following variables

$$T_f = f(L,R,C,z,W,N)$$

hence, the primary design equation is

$$U = T_f = \text{minimum}$$

B- Minimum oil temperature rise.

Here, we have to differentiate between the two classes of bearings, that is, self - contained and forced feed bearings.

The primary design equation, is

$$U = \Delta t = \text{minimum}$$

where;

Δt = temperature rise of oil above ambient

$= \Delta t_W + \Delta t_O$ for self contained.

$= \Delta t$ in equation (5-4) for forced feed bearings.

C- Minimum friction torque loss and minimum oil temperature rise .

In the last case the target is different, and what is called a multifactor optimization technique is required. In such a case the utility contribution of both factors constitutes the objective function that is to be maximized rather than minimizing the specific value of the design characteristics, as in case A or B .

Anumber of multifactor optimization techniques for developing the utility contributions is explained by Siddall [32], The inverted utility technique suggests minimizing the undesirability rather than maximizing combined desirability of both parameters (frictional torque and oil temperature rise, in this case) . To carry out the inverted utility concept [36], a functional relationship is sketched for each parameter as shown in figure 7-2 where the vertical axis represents the undesirability (optional) values and the horizontal axis represents the parameter values . If the undesirability functional values are U_1 for the frictional torque and U_2 for the oil temperature rise, then the total optimization objective function to be minimized is given by :

$$U = U_1 + U_2$$

7-5.4 Problem Constraints

In addition to all preceding pertinent design equations, there are several significant limit equations that should be written for expressing permissible ranges of values for certain significant parameters. They are discussed in chapter VI.

The primary constraint, here, is that a thick film lubrication should occur and that the bearing will work hydrodynamically .

The following is a group of constraints according to practice and current experience in bearing design.

1- Constraint for minimum film thickness

$$h_o \geqslant (h_o)_{all}$$

where $(h_o)_{all}$ is the allowable lower limit, taken as a function of bearing material, See section 6-4.

2- Length -to- diameter ratio

$$L/D \geqslant (L/D)_{min}$$

$$(L/D)_{max} \geqslant L/D$$

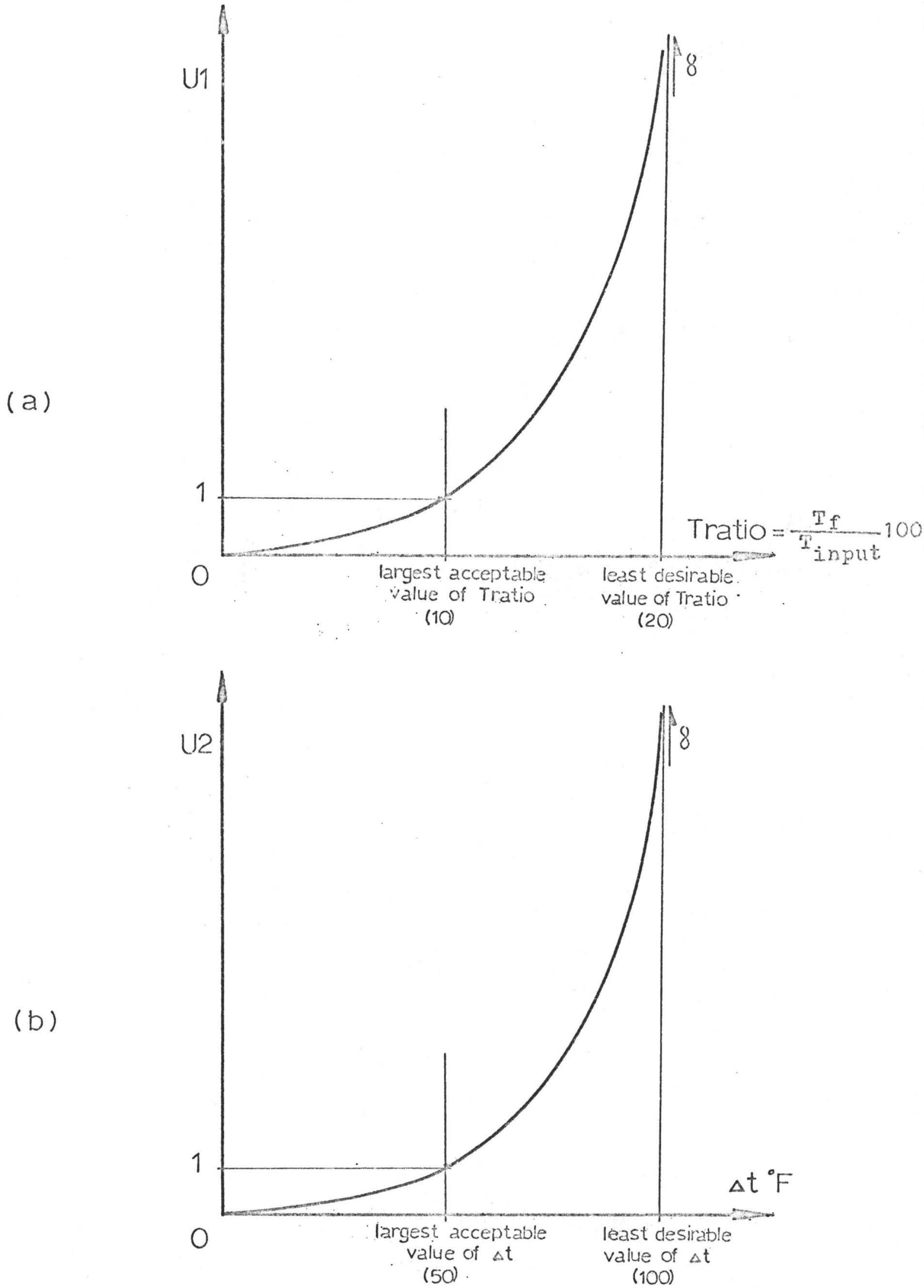


Fig 7_2 Undesirability curves for journal bearing

where the maximum and minimum L/D ratio depend upon the application. See table 6-1 .

3- Constraint for Sommerfeld number S.

$$S \geqslant S_{min}$$

where the minimum value of Sommerfeld number depends upon the bearing metal as given by table 6-4.

4- Constraint for zN/P.

$$zN/P \geqslant (zN/P)_{min}$$

where the minimum value is a function of bearing metal as given in table 6-4.

5- Constraint for oil film turbulence.

$$N_{cr} \geqslant N$$

where N_{cr} is the critical shaft speed at which turbulance in the oil film start to occur. See section 6-7 for shaft critical speed .

6- Oil operating temperature constraint.

$$180\ ^{o}F \geqslant T_o \qquad \text{(self contained)}$$

$$250\ ^{o}F \geqslant T_o \qquad \text{(pressure feed)}$$

Whereas 180^{o} is the highest recommended operating oil temperature for the first class of bearings [11], [31], 250^{o} is assumed for the other class as a maximum reasonable practical temperature.

7- Oil inlet temperature for forced feed bearings.

$$T_{oil} \geqslant T_1 + 25\ ^{o}F$$

where T_{oil} is the oil inlet temperature, ^{o}F and T_1 is the ambient temperature, ^{o}F .

8- There are limits on the grade of lubricating oil, that could be used.

$$\text{SAE } 70 \geqslant G \geqslant \text{SAE } 10$$

9- Constraint on journal radius.

$$R_{max} \geqslant R \geqslant R_{min}$$

where R_{max} and R_{min} should be specified by the designer if the radius is to be considered as a design variable .

lo- A group of other constraints to satisfy the non - negative requirements of the variables are :

bearing length	L	$\geqslant$	0.0
radial clearance	C	$\geqslant$	0.0
viscosity	z	$\geqslant$	0.0
chamfer length	l	$\geqslant$	0.0
chamfer dimension	a	$\geqslant$	0.0
oil inlet pressure	P	$\geqslant$	0.0

7-5.5 <u>Computer Output</u>

The program gives the designer the following main items :

1- Value of criterion function

2- Value of design variables

3- Oil operating temperature, ^{o}F.

4- Frictional torque, in.lb.

5- Amount of oil that must be supplied, gpm.

6- Oil grade, SAE number

7- Bearing metal advice

8- Surface finish advice

9- Shaft hardness advice

and for forced - feed bearings,

10- Oil inlet temperature, ^{o}F.

11- Oil inlet pressure, psi.

7-6 User - Oriented Journal Bearing Program

A complete design program covering all design aspects discussed in the preceding chapters has been created with a view to making the user input requirements as simple as possible. The user is required to supply only four main values mentioned before in section 7-5.1 ; load that the bearing is to carry, rotational speed, constant operating ambient temperature, and the upper expected limit of the ambient temperature. Plus up to 11 options including radius of journal, input oil pressure, and input shaft torque.

The design program consists of ten FORTRAN subroutines in addition to 13 others for optimization purpose adapted from OPTISEP [31] . The whole program needs approximately 70,000 storage locations in the central memory of a CDC 6400 computer . Figure 7-3 shows an outline of the main subroutines of the program. How the user sets up the calling program is included in the user's manual, together with a complete listing, in Appendix 1 .

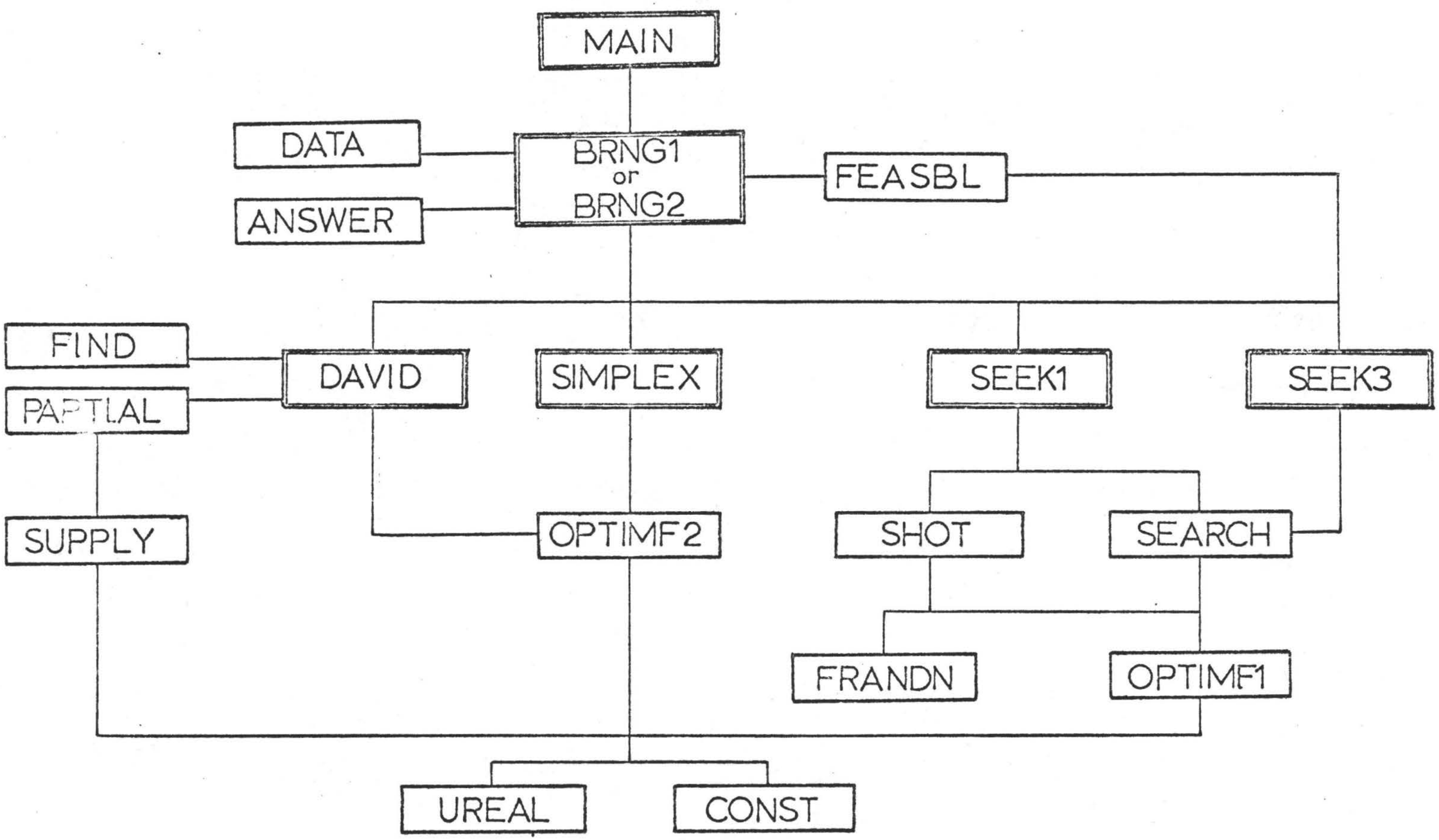


Fig 7_3 Program flow chart

7-7 Sample Problems

A- Pressure-fed journal bearings.

Problem 1

The following specifications are given for a journal bearing lubricated under pressure, and operating under steady load. [30] .

Applied load	5460	lb.
Journal speed	1600	rpm
Journal radius	2.0	inch
Bearing length	4.2	inch
Radial-clearance	0.003	inch
Oil operating temperature	210	^{o}F.
Minimum film thickness	0.00069	inch
Oil inlet temperature	110	^{o}F.
Oil inlet pressure	33.5	psi

Solution

The problem is solved by specifying only the load, journal speed, and then also the journal radius and oil inlet pressure using DAVID optimization method (gradient method) .

The following is also specified :

1- The bearing is to be used for turbo-generators.

2- The optimization criterion is minimum frictional torque loss.

3- The bearing is to be a split cylindrical type.

4- The design ambient temperature is 75 $^{\circ}$F .

5- The maximum expected ambient temperature is assumed 150 $^{\circ}$F .

Accordingly, the calling program is set up as shown in figure 7-4 where, in this case, the oil inlet pressure is taken as a variable. See Appendix 1 for the definition of the code names and how to set up the calling program. The input informations as given by the calling program is printed out in the computer output for reference. This is shown in figure 7-4a . The optimum solution found by DAVID is shown in figure 7-4b together with the time consumed.

The same problem is solved again considering that both the journal radius and the oil inlet pressure are unknowns. The optimum solution found by DAVID is shown in figure 7-5b. Figure 7-5a shows the input data print out.

Observe that the optimum solution found in the first case is restricted to the design ambient temperature (T1). While in the second case the optimum solution found is feasible up to 124 $^{\circ}$F. ambient temperature .

```
      COMMON /111/ RPM,W,RADIUS,T1,KOPTIM,KBRG,MTHD,RADU,RADL,NMBR,KOILS
     1UP,KAPLIC,PINLET,KEY,KAIR,TINPUT,TMAX
      W=5460.
      RPM=1600.
      T1=75.
      TMAX=150.
      RADIUS=2.
      RADU=RADIUS
      RADL=RADIUS
      NMBR=7
      KOPTIM=1
      KAPLIC=1
      MTHD=7
      TINPUT=0.
      KBRG=1
      PINLET=0.0
      KEY=1
      CALL BRNG2
      STOP
      END
```

Fig 7_4

- OPTIMUM HYDRODYNAMIC BEARING DESIGN -

INPUT DATA

LOAD AT BEARING, LBS.	W	=	5.4600000E+03
JOURNAL SPEED, REVS/MIN.	RPM	=	1.6000000E+03
AMBIENT TEMPERATURE, DEG.F.	T1	=	7.5000000E+01
MAXIMUM EXPECTED AMBIENT TEMPERATURE	TMAX	=	1.5000000E+02
JOURNAL RADIUS, INCH	RADIUS	=	2.0000000E+00
ESTIMATED UPPER LIMIT OF RADIUS	RADU	=	2.0000000E+00
ESTIMATED LOWER LIMIT OF RADIUS	RADL	=	2.0000000E+00
OPTIMIZATION CRITERION	KOPTIM	=	1
TYPE OF APPLICATION	KAPLIC	=	1
NUMBER OF DESIGN VARI BLES	NMBR	=	7
OPTIMIZATION METHOD USED	MTHD	=	7
INPUT TORQUE	TINPUT	=	0.
TYPE OF JOURNAL BEARING USED.......................	KBRG	=	1
FLAG NUMBER	KEY	=	1
OIL INLET PRESSURE	PINLET	=	0.

Fig 7_4 a

```
                    OPTIMUM SOLUTION FOUND
                    ----------------------

    VALUE OF CRITERION FUNCTION. . . . . .U =     40.27793162



    VALUE OF DESIGN VARIABLES
    -------------------------

 1- BEARING LENGTH . . . . . . . . . . . X(1) =     5.49100007     INCH
 2- RADIAL CLEARANCE . . . . . . . . . . X(2) =      .00200000     INCH
 3- OIL OPERATING VISCOSITY. . . . . . . X(3) =     8.81457261     CENTIPOISE
 4- OIL INLET PRESSURE . . . . . . . . . X(4) =    10.00000000     LB/SQ.INCH
 5- AXIAL LENGTH OF CHAMFER. . . . . . . X(5) =      .10000000     INCH
 6- CHAMFER DIMENSION. . . . . . . . . . X(6) =      .05000000     INCH
 7- OIL INLET TEMPERATURE. . . . . . . . X(7) =   163.88103870     DEG. FAHRENHEIT

 8- MINIMUM OIL FILM THICKNESS . . . . . . .  =      .00079357     INCH
 9- ECCENTRICITY RATIO . . . . . . . . . . .  =      .60321620
10- ATTITUDE ANGLE . . . . . . . . . . . . .  =    49.99210247     DEGREES
11- OIL OPERATING TEMPERATURE. . . . . . . .  =   200.66143967     DEG. FAHRENHEIT
12- FRICTIONAL TORQUE. . . . . . . . . . . .  =    40.27793162     LB. INCH
13- AMOUNT OF OIL THAT MUST
    BE CONTINUOUSLY SUPPLIED . . . . . . . .  =      .29948609     GALLON/MIN.
14- OIL GRADE. . . . . . . . . . . . . . . .  =   (SAE   20)
15- BEARING METAL IS NO.     ( 1)    - SEE USERS MANUAL, MATERIAL LIST -

    -RECOMMENDATIONS-
    -----------------

 1- MAXIMUM SURFACE FINISH ROUGHNESS SHOULD BE LESS OR EQUAL TO (20-MICRON INCH)
 2- SHAFT BRINELL HARDNESS MUST BE GREATER OR EQUAL TO (300)

    INEQUALITY CONSTRAINTS
    ----------------------

                    PHI( 1) =   4.67335605E+01
                    PHI( 2) =   4.35676005E+02
                    PHI( 3) =   1.12037586E+03
                    PHI( 4) =   2.43689268E+06
                    PHI( 5) =   6.00000000E+02
                    PHI( 6) =   4.93385603E+01
                    PHI( 7) =   5.00000000E+10
                    PHI( 8) =   1.00000000E+01
                    PHI( 9) =   1.73033336E+03
                    PHI(10) =   5.00000000E+02
                    PHI(11) =   9.00000000E+01

                    -IMPORTANT-
                    -----------
    THE OPTIMUM SOLUTION FOUND IS ONLY FEASIBLE FOR THE DESIGN OPERATING
     TEMPERATURE - 75.00   DEG. FAHRENHEIT .
    OR REOPTIMIZE WITH A HIGHER DESIGN OPERATING TEMPERATURE .
```

Fig 7_4 b

Computer time:3.459

- OPTIMUM HYDRODYNAMIC BEARING DESIGN -

INPUT DATA

LOAD AT BEARING, LBS.	W	=	5.4600000E+03
JOURNAL SPEED, REVS/MIN.	RPM	=	1.6000000E+03
AMBIENT TEMPERATURE, DEG.F.	T1	=	7.5000000E+01
MAXIMUM EXPECTED AMBIENT TEMPERATURE	TMAX	=	1.5000000E+02
JOURNAL RADIUS, INCH	RADIUS	=	0.
ESTIMATED UPPER LIMIT OF RADIUS	RADU	=	2.2500000E+00
ESTIMATED LOWER LIMIT OF RADIUS	RADL	=	1.7500000E+00
OPTIMIZATION CRITERION	KOPTIM	=	1
TYPE OF APPLICATION	KAPLIC	=	1
NUMBER OF DESIGN VARI BLES	NMER	=	8
OPTIMIZATION METHOD USED	MTHD	=	7
INPUT TORQUE	TINPUT	=	0.
TYPE OF JOURNAL BEARING USED	KBRG	=	1
FLAG NUMBER	KEY	=	1
OIL INLET PRESSURE	PINLET	=	0.

Fig 7_5 a

OPTIMUM SOLUTION FOUND

VALUE OF CRITERION FUNCTION. . . . . .U = 48.29975104

VALUE OF DESIGN VARIABLES

1-	BEARING LENGTH . . . . X(1) =	5.59462132	INCH
2-	RADIAL CLEARANCE . . . . X(2) =	.00204000	INCH
3-	OIL OPERATING VISCOSITY. . . . . X(3) =	8.81457261	CENTIPOISE
4-	JOURNAL RADIUS . . . . X(4) =	2.16341091	INCH
5-	OIL INLET PRESSURE . . . . X(5) =	10.00000000	LB/SQ.INCH
6-	AXIAL LENGTH OF CHAMFER. . . . . X(6) =	.10000000	INCH
7-	CHAMFER DIMENSION. . . . . X(7) =	.05000000	INCH
8-	OIL INLET TEMPERATURE. . . . . X(8) =	163.88103870	DEG. FAHRENHEIT
9-	MINIMUM OIL FILM THICKNESS . . . . =	.00091623	INCH
10-	ECCENTRICITY RATIO . . . . =	.55086775	
11-	ATTITUDE ANGLE . . . . =	53.20461076	DEGREES
12-	OIL OPERATING TEMPERATURE. . . . . =	200.66143967	DEG. FAHRENHEIT
13-	FRICTIONAL TORQUE. . . . . =	48.29975104	LB. INCH
14-	AMOUNT OF OIL THAT MUST BE CONTINUOUSLY SUPPLIED . . . . =	.36092293	GALLON/MIN.
15-	OIL GRADE. . . . . =	(SAE 20)	

16- BEARING METAL IS NO. (1) - SEE USERS MANUAL, MATERIAL LIST -

-RECOMMENDATIONS-

1- MAXIMUM SURFACE FINISH ROUGHNESS SHOULD BE LESS OR EQUAL TO (20-MICRON INCH)
2- SHAFT BRINELL HARDNESS MUST BE GREATER OR EQUAL TO (300)

INEQUALITY CONSTRAINTS

PHI(1) = 5.25271028E+01
PHI(2) = 1.66229794E+03
PHI(3) = 1.44857864E+03
PHI(4) = 2.27437082E+06
PHI(5) = 6.61364362E+02
PHI(6) = 4.93385603E+01
PHI(7) = 5.00000000E+10
PHI(8) = 1.00000000E+01
PHI(9) = 1.76487377E+03
PHI(10) = 5.00000000E+02
PHI(11) = 9.00000000E+01

Fig 7_5 b

Computer time: 3.548

-IMPORTANT-

THE OPTIMUM SOLUTION FOUND IS NOT FEASIBLE ON THE MAXIMUM EXPECTED AMBIENT TEMPERATURE.
MAXIMUM EXPECTED TEMPERATURE MUST BE REDUCED TO 124.00 DEG.FAHRENHEIT
OR REOPTIMIZE WITH A HIGHER DESIGN OPERATING TEMPERATURE.

Problem 2

A full cylindrical bearing with a circumferential oil groove is designed with the following data, [30] .

Applied load	3000	lb.
Journal speed	1800	rpm
Journal radius	1.25	inch
Bearing effective length	3.9	inch
Radial clearance	0.0015	inch
Oil operating temperature	225	^{O}F
Oil inlet temperature	105	^{O}F
Oil inlet pressure	40.	psi
Lubricating oil used	SAE 20	

Solution

The problem is solved for optimum with the same conditions stated in problem 1, however the maximum expected temperature is assumed 90 ^{O}F. and the bearing is full cylindrical with a circumferential oil groove . The calling program is set up in the same way, as shown in figure 7-4. Optimum solutions have been found first for known or given journal radius and oil inlet pressure and then for these as unknowns (design variables).

For known radius and oil inlet pressure, the calling program and the input data print out are shown in figures 7-6 and 7-6a respectively. The optimum solution found is shown in figure 7-6b. For journal radius and oil inlet pressure as design variables, the optimum solution is shown in figure 7-6c. Observe that in both cases the optimum solution found by SIMPLEX is restricted to the design ambient temperature .

```
      CCMMCN /111/ RPM,W,RADIUS,T1,KOPTIM,KBRG,MTHD,RADU,RADL,NMBR,KOILS
     1UF,KAPLIC,PINLET,KEY,KAIR,TINPUT,TMAX
      W=3000.
      RPM=1800.
      T1=75.
      TMAX=90.
      RADIUS=1.25
      RADU=RADIUS
      RADL=RADIUS
      PINLET=40.
      NMBR=4
      KOPTIM=1
      KAPLIC=1
      TINPUT=0.0
      KBRG=2
      KEY=2
      MTHD=5
      CALL BRNG2
      STOP
      END
```

Fig 7_6

- OPTIMUM HYDRODYNAMIC BEARING DESIGN -

INPUT DATA

LOAD AT BEARING, LBS.	W	=	3.0000000E+03
JOURNAL SPEED, REVS/MIN.	RPM	=	1.8000000E+03
AMBIENT TEMPERATURE, DEG.F.	T1	=	7.5000000E+01
MAXIMUM EXPECTED AMBIENT TEMPERATURE	TMAX	=	9.0000000E+01
JOURNAL RADIUS, INCH	RADIUS	=	1.2500000E+00
ESTIMATED UPPER LIMIT OF RADIUS	RADU	=	1.2500000E+00
ESTIMATED LOWER LIMIT OF RADIUS	RADL	=	1.2500000E+00
OPTIMIZATION CRITERION	KOPTIM	=	1
TYPE OF APPLICATION	KAPLIC	=	1
NUMBER OF DESIGN VARIABLES	NMBR	=	4
OPTIMIZATION METHOD USED	MTHD	=	5
INPUT TORQUE	TINPUT	=	0.
TYPE OF JOURNAL BEARING USED	KBRG	=	2
FLAG NUMBER	KEY	=	2
OIL INLET PRESSURE	PINLET	=	4.0000000E+01

Fig 7_6a

```
                OPTIMUM SOLUTION FOUND
                ----------------------

    VALUE OF CRITERION FUNCTION. . . . . .U =     10.09229809


    VALUE OF DESIGN VARIABLES
    -------------------------

 1- BEARING LENGTH . . . . . . . . . . . X(1) =     3.13453570     INCH
 2- RADIAL CLEARANCE . . . . . . . . . . X(2) =      .00153798     INCH
 3- OIL OPERATING VISCOSITY. . . . . . . X(3) =     8.49055191     CENTIPOISE
 4- OIL INLET TEMPERATURE. . . . . . . . X(4) =   164.80457184     DEG. FAHRENHEIT

 5- MINIMUM OIL FILM THICKNESS . . . . . . .  =      .00037501     INCH
 6- ECCENTRICITY RATIO . . . . . . . . . . .  =      .75616642
 7- ATTITUDE ANGLE . . . . . . . . . . . . .  =    39.62972508     DEGREES
 8- OIL OPERATING TEMPERATURE. . . . . . . .  =   170.97426155     DEG. FAHRENHEIT
 9- FRICTIONAL TORQUE. . . . . . . . . . . .  =    10.09229809     LB. INCH
10- AMOUNT OF OIL THAT MUST
    BE CONTINUOUSLY SUPPLIED . . . . . . . .  =      .06874738     GALLON/MIN.
11- OIL GRADE. . . . . . . . . . . . . . . .  =   (SAE  10)
12- BEARING METAL IS NO.    ( 3)    - SEE USERS MANUAL, MATERIAL LIST -

    -RECOMMENDATIONS-
     ---------------

 1- MAXIMUM SURFACE FINISH ROUGHNESS SHOULD BE LESS OR EQUAL TO (20-MICRON INCH)
 2- SHAFT BRINELL HARDNESS MUST BE GREATER OR EQUAL TO (300)

    INEQUALITY CONSTRAINTS
    ----------------------

                PHI( 1) =  3.59209071E+01
                PHI( 2) =  1.23246452E-01
                PHI( 3) =  5.46965685E+02
                PHI( 4) =  4.34933445E+06
                PHI( 5) =  3.46201527E+02
                PHI( 6) =  7.90257385E+01
                PHI( 7) =  6.00000000E+10
                PHI( 8) =  0.
```

Fig 7_6 b

Computer time:10.123 sec.

```
                -IMPORTANT-
                 ---------
    THE OPTIMUM SOLUTION FOUND IS ONLY FEASIBLE FOR THE DESIGN OPERATING
     TEMPERATURE - 75.00   DEG. FAHRENHEIT .
    OR REOPTIMIZE WITH A HIGHER DESIGN OPERATING TEMPERATURE .
```

OPTIMUM SOLUTION FOUND

```
VALUE OF CRITERION FUNCTION. . . . . .U =     23.15003037


VALUE OF DESIGN VARIABLES
-------------------------

 1- BEARING LENGTH . . . . . . . . . . . . X(1) =      5.02162861     INCH
 2- RADIAL CLEARANCE . . . . . . . . . . . X(2) =       .00147761     INCH
 3- OIL OPERATING VISCOSITY. . . . . . . . X(3) =      8.28905174     CENTIPCISE
 4- JOURNAL RADIUS . . . . . . . . . . . . X(4) =      1.57218611     INCH
 5- OIL INLET PRESSURE . . . . . . . . . . X(5) =     12.40842287     LB/SQ.INCH
 6- OIL INLET TEMPERATURE. . . . . . . . . X(6) =    105.40599352     DEG. FAHRENHEIT

 7- MINIMUM OIL FILM THICKNESS . . . . . . . . =       .00075000     INCH
 8- ECCENTRICITY RATIO . . . . . . . . . . . . =       .49242143
 9- ATTITUDE ANGLE . . . . . . . . . . . . . . =     56.76691366     DEGREES
10- OIL OPERATING TEMPERATURE. . . . . . . . . =    172.43440768     DEG. FAHRENHEIT
11- FRICTIONAL TORQUE. . . . . . . . . . . . . =     23.15003037     LB. INCH
12- AMOUNT OF OIL THAT MUST
    BE CONTINUOUSLY SUPPLIED . . . . . . . . . =       .20602495     GALLON/MIN.
13- OIL GRADE. . . . . . . . . . . . . . . . . =  (SAE   10)
14- BEARING METAL IS NO.    ( 1)    - SEE USERS MANUAL, MATERIAL LIST -

-RECOMMENDATIONS-
-----------------

 1- MAXIMUM SURFACE FINISH ROUGHNESS SHOULD BE LESS OR EQUAL TO (20-MICRON INCH)
 2- SHAFT BRINELL HARDNESS MUST BE GREATER OR EQUAL TO (300)

INEQUALITY CONSTRAINTS
----------------------

                    PHI( 1) =  6.85298272E+01
                    PHI( 2) =  9.88799542E-03
                    PHI( 3) =  1.89746004E+03
                    PHI( 4) =  4.02286905E+06
                    PHI( 5) =  4.81113874E+02
                    PHI( 6) =  7.75655923E+01
                    PHI( 7) =  6.00000000E+10
                    PHI( 8) =  0.
                    PHI( 9) =  8.75915771E+01

                    -IMPORTANT-
                    -----------
THE OPTIMUM SOLUTION FOUND IS ONLY FEASIBLE FOR THE DESIGN OPERATING
 TEMPERATURE - 75.00   DEG. FAHRENHEIT .
OR REOPTIMIZE WITH A HIGHER DESIGN OPERATING TEMPERATURE .
```

Fig 7_6 c

Computer time:60.166 sec.

B- Self-contained journal bearings.

Problem 3

A waste-packed bearing is to be designed for a turbo generator with the following specifications.

Applied load	2800	lb.
Journal speed	300	rpm
Journal radius	2.	inch

Solution

The problem is solved for a combined criterion of minimum friction loss and minimum temperature rise. The journal radius is considered first as known (as given = 2") and then as unknown.

In solving the problem by SIMPLEX the following is assumed :

1. The ambient is a moving air.
2. The design ambient temperature is 75 ^{o}F.
3. The maximum expected ambient temperature is 90 ^{o}F.
4. The input shaft torque is 2000 lb.inch .

According to these assumptions, the calling program is set up as shown in figure 7-7. Figure 7-7b shows the optimum solution found considering the journal radius as known. Figure 7-7a gives the intermediate output for iterations of the optimization process in this case. The optimum solution found considering the radius as unknown is

```
      COMMON /111/ RPM,W,RADIUS,T1,KOPTIM,KBRG,MTHD,RADU,RADL,NMBR,KOILS
     1UP,KAPLIC,PINLET,KEY,KAIR,TINPUT,TMAX
      W=2800.
      RPM=300.
      T1=75.
      TMAX=90.
      RADIUS=2.
      RADU=RADIUS
      RADL=RADIUS
      NMBR=3
      KOPTIM=3
      KAPLIC=1
      KAIR=2
      KOILSUP=3
      TINPUT=2000.
      MTHD=5
      CALL BRNG1
      STOP
      END
```

Fig 7_7

INTERMEDIATE OUTPUT FOR SIMPLEX

UART IS THE VALUE OF ARTIFICIAL UNCONSTRAINED OBJECTIVE FUNCTION AT THE CENTROID

STEP NO.	U	UART	VARIABLES X(I) AT THE CENTROID OF THE SIMPLEX		
2	8.26202021E-01	9.00000152E+30	4.65777778E+00	1.98222222E-03	9.00000000E+00
4	2.20209625E+00	2.24660738E+00	4.61333333E+00	1.82666667E-03	1.00000000E+01
6	2.20209625E+00	2.24660738E+00	4.61333333E+00	1.82666667E-03	1.00000000E+01
8	2.74951852E+00	2.79548349E+00	4.70083333E+00	1.73708333E-03	1.00156250E+01
10	2.19318963E+00	2.23941677E+00	4.80245370E+00	1.70641204E-03	9.98350694E+00
12	1.77937253E+00	1.82952573E+00	4.75082047E+00	1.70240612E-03	9.95635610E+00
14	1.70372830E+00	1.75533623E+00	4.77876293E+00	1.69716153E-03	9.94970234E+00
16	1.65958249E+00	1.71227310E+00	4.79171618E+00	1.69359364E-03	9.94558447E+00
18	1.55218734E+00	1.60906442E+00	4.79919345E+00	1.68233911E-03	9.93484786E+00
20	1.48413388E+00	1.54298195E+00	4.79962253E+00	1.72527148E-03	9.93336089E+00
22	1.43651609E+00	1.50312997E+00	4.78531010E+00	1.72378053E-03	9.92847237E+00
24	1.43108062E+00	1.49592700E+00	4.78577220E+00	1.73929558E-03	9.92961240E+00
26	1.36764037E+00	1.43480816E+00	4.78810665E+00	1.83764424E-03	9.93282733E+00
28	1.33236219E+00	1.39890957E+00	4.79580826E+00	1.93655201E-03	9.93787546E+00
30	1.31214910E+00	1.37532336E+00	4.81144423E+00	2.06898814E-03	9.94643949E+00
32	1.29867194E+00	1.36281105E+00	4.82336714E+00	2.18933715E-03	9.95363778E+00
34	1.29541337E+00	1.36098750E+00	4.82603082E+00	2.21552408E-03	9.95500055E+00
36	1.09375764E+00	5.00000201E+30	4.99107727E+00	2.40895299E-03	8.95157269E+00
38	1.53116563E+00	1.18007187E+24	4.94663282E+00	2.25339744E-03	9.95157269E+00
40	1.27848239E+00	4.26587007E+22	4.87996616E+00	2.22006410E-03	9.95157269E+00
42	2.[illegible]702947E+00	2.33705118E+00	4.77441060E+00	2.22284188E-03	1.00557394E+01
44	1.[illegible]8129605E+00	1.58131911E+00	4.69107727E+00	2.22561966E-03	9.99323935E+00
46	1.45750839E+00	1.45753261E+00	4.70272850E+00	2.21477861E-03	9.97790370E+00
48	1.38537529E+00	1.38540058E+00	4.70799042E+00	2.20953920E-03	9.96821845E+00
50	1.34208407E+00	1.34211041E+00	4.71048569E+00	2.20686451E-03	9.96209808E+00
52	1.31560119E+00	1.31562864E+00	4.71173480E+00	2.20542538E-03	9.95822901E+00
54	1.29920135E+00	1.29922997E+00	4.71239479E+00	2.20461494E-03	9.95578256E+00
56	1.28896628E+00	1.28899608E+00	4.71276083E+00	2.20414177E-03	9.95423542E+00
58	1.28254703E+00	1.28257794E+00	4.71297204E+00	2.20385808E-03	9.95325692E+00
60	1.26746528E+00	1.26750317E+00	4.67995458E+00	2.20892230E-03	9.95125471E+00
62	1.26334172E+00	1.26338340E+00	4.69120575E+00	2.20687230E-03	9.95052533E+00
64	1.25658837E+00	1.25665152E+00	4.70064548E+00	2.20504827E-03	9.94938058E+00
66	1.20368991E+00	5.00000212E+30	5.15813516E+00	2.29161516E-03	8.94858961E+00
68	1.23351340E+00	1.21643499E+23	4.84702405E+00	2.26939294E-03	9.94858961E+00
70	1.24021215E+00	4.68829422E+22	4.78035738E+00	2.23605960E-03	9.94858961E+00
72	2.36015739E+00	2.36015740E+00	4.78591294E+00	2.18328182E-03	1.00527563E+01
74	1.62685203E+00	1.62685204E+00	4.79146850E+00	2.14161516E-03	9.99025627E+00
76	1.48083859E+00	1.48083860E+00	4.76978640E+00	2.14744077E-03	9.97492062E+00
78	1.39980004E+00	1.39980005E+00	4.75930757E+00	2.15007173E-03	9.96523538E+00
80	1.35241455E+00	1.35241456E+00	4.75395820E+00	2.15131937E-03	9.95911500E+00
82	1.32384861E+00	1.32384863E+00	4.75107993E+00	2.15194392E-03	9.95524593E+00
84	1.30630938E+00	1.30630940E+00	4.74945906E+00	2.15227392E-03	9.95279948E+00
86	1.29541907E+00	1.29541909E+00	4.74851273E+00	2.15245694E-03	9.95125234E+00
88	1.28861009E+00	1.28861010E+00	4.74794533E+00	2.15256254E-03	9.95027384E+00
90	1.26631772E+00	1.26631774E+00	4.73585156E+00	2.16938714E-03	9.94827164E+00
92	1.26197618E+00	1.26197621E+00	4.73434768E+00	2.17128520E-03	9.94775888E+00
94	1.25921422E+00	1.25921425E+00	4.73293583E+00	2.17321280E-03	9.94748385E+00

Fig 7_7a

OPTIMUM SOLUTION FOUND

VALUE OF CRITERION FUNCTION. . . . . .U = 1.25247017

VALUE OF DESIGN VARIABLES

```
 1- BEARING LENGTH . . . . . . . . . . . . X(1) =      7.19999322    INCH
 2- RADIAL CLEARANCE . . . . . . . . . . . X(2) =       .00220273    INCH
 3- OIL OPERATING VISCOSITY. . . . . . . . X(3) =     14.01032491    CENTIPOISE

 4- MINIMUM OIL FILM THICKNESS . . . . . . . . =       .00075025    INCH
 5- ECCENTRICITY RATIO . . . . . . . . . . . . =       .65940122
 6- ATTITUDE ANGLE . . . . . . . . . . . . . . =     47.05646574    DEGREES
 7- OIL OPERATING TEMPERATURE. . . . . . . . . =    140.28367603    DEG. FAHRENHEIT
 8- FRICTIONAL TORQUE. . . . . . . . . . . . . =     15.33299450    LB. INCH
 9- AMOUNT OF OIL THAT MUST
    BE CONTINUOUSLY SUPPLIED . . . . . . . . . =       .06555345    GALLON/MIN.
10- OIL GRADE. . . . . . . . . . . . . . . . . =  (SAE   10)
11- BEARING METAL IS NO.    ( 1)     - SEE USERS MANUAL, MATERIAL LIST -
```

-RECOMMENDATIONS-

```
1- MAXIMUM SURFACE FINISH ROUGHNESS SHOULD BE LESS OR EQUAL TO (20-MICRON INCH)
2- SHAFT BRINELL HARDNESS MUST BE GREATER OR EQUAL TO (300)
```

INEQUALITY CONSTRAINTS

```
PHI( 1) =  3.32318190E+01
PHI( 2) =  2.45886906E+00
PHI( 3) =  6.10878968E+02
PHI( 4) =  3.26932852E+06
PHI( 5) =  5.79727373E+02
PHI( 6) =  3.97163240E+08
PHI( 7) =  6.00000000E+01
PHI( 8) =  0.
```

Fig 7_7 b

-IMPORTANT-

THE OPTIMUM SOLUTION FOUND IS ONLY FEASIBLE FOR THE DESIGN OPERATING
TEMPERATURE - 75.00 DEG. FAHRENHEIT .
OR REOPTIMIZE WITH A HIGHER DESIGN OPERATING TEMPERATURE .

Computer time:8.457 sec.

OPTIMUM SOLUTION FOUND

VALUE OF CRITERION FUNCTION. . . . . . .U = 1.10826499

VALUE OF DESIGN VARIABLES

```
 1- BEARING LENGTH . . . . . . . . . . . . X(1) =     7.66697679     INCH
 2- RADIAL CLEARANCE . . . . . . . . . . . X(2) =      .00347462     INCH
 3- OIL OPERATING VISCOSITY. . . . . . . . X(3) =    14.41777002     CENTIPOISE
 4- JOURNAL RADIUS . . . . . . . . . . . . X(4) =     2.13146412     INCH

 5- MINIMUM OIL FILM THICKNESS . . . . . . . . =      .00075018     INCH
 6- ECCENTRICITY RATIO . . . . . . . . . . . . =      .78409800
 7- ATTITUDE ANGLE . . . . . . . . . . . . . . =    38.42121362     DEGREES
 8- OIL OPERATING TEMPERATURE. . . . . . . . . =   138.89600066     DEG. FAHRENHEIT
 9- FRICTIONAL TORQUE. . . . . . . . . . . . . =    16.44715923     LB. INCH
10- AMOUNT OF OIL THAT MUST
    BE CONTINUOUSLY SUPPLIED . . . . . . . . . =      .09135013     GALLON/MIN.
11- OIL GRADE. . . . . . . . . . . . . . . . . =  (SAE  10)
12- BEARING METAL IS NO.    ( 1)   - SEE USERS MANUAL, MATERIAL LIST -
```

-RECOMMENDATIONS-

1- MAXIMUM SURFACE FINISH ROUGHNESS SHOULD BE LESS OR EQUAL TO (20-MICRON INCH)
2- SHAFT BRINELL HARDNESS MUST BE GREATER OR EQUAL TO (300)

INEQUALITY CONSTRAINTS

```
PHI( 1) =  4.04886185E+01
PHI( 2) =  1.76976786E+00
PHI( 3) =  2.08917980E+02
PHI( 4) =  1.64390323E+06
PHI( 5) =  5.05123850E+02
PHI( 6) =  4.11039993E+08
PHI( 7) =  6.00000000E+01
PHI( 8) =  0.
```

Fig 7.7c

Computer time:9.081 sec.

-IMPORTANT-

THE OPTIMUM SOLUTION FOUND IS ONLY FEASIBLE FOR THE DESIGN OPERATING
TEMPERATURE - 75.00 DEG. FAHRENHEIT .
OR REOPTIMIZE WITH A HIGHER DESIGN OPERATING TEMPERATURE .

shown in figure 7-7c .

Problem 4

A bearing for a motor-generator set is to be designed with the following specifications.

Applied load	1200	lbs.
Journal radius	2.75	inch
Journal speed	450	rpm

The bearing is to be designed for a combined minimum friction loss and minimum temperature rise. The shaft torque is given 500 lb.inch .

Solution

The problem is solved by SIMPLEX , assuming quiet ambient with temperature 75 ^{o}F. and the maximum expected ambient temperature is 100 ^{o}F .

The optimum solution found is shown in figure 7-8a and, as shown, is feasible up to 88 ^{o}F ambient temperature. Figure 7-8 shows the calling program.

The same problem is solved for minimum temperature rise considering the journal radius first as known and then as unknown. Solutions found are given in figure 7-8b and 7-8c respectively .

```
      COMMON /111/ RPM,W,RADIUS,T1,KOPTIM,KBRG,MTHD,RADU,RADL,NMBR,KOILS
     1UP,KAPLIC,PINLET,KEY,KAIR,TINPUT,TMAX
      W=1200.
      RPM=450.
      T1=75.
      TMAX=100.
      RADIUS=2.75
      RADU=RADIUS
      RADL=RADIUS
      NMBR=3
      KOPTIM=3
      KAPLIC=3
      KOILSUP=1
      TINPUT=500.
      KAIR=1
      MTHD=5
      CALL BRNG1
      STOP
      END
```

Fig 7_8

OPTIMUM SOLUTION FOUND

VALUE OF CRITERION FUNCTION. . . . . .U = 4.04405235

VALUE OF DESIGN VARIABLES

```
 1- BEARING LENGTH . . . . . . . . . . . . X(1) =     9.56053105      INCH
 2- RADIAL CLEARANCE . . . . . . . . . . . X(2) =      .00350956      INCH
 3- OIL OPERATING VISCOSITY. . . . . . . . X(3) =     9.13236633      CENTIPOISE

 4- MINIMUM OIL FILM THICKNESS . . . . . . . . =      .00197157      INCH
 5- ECCENTRICITY RATIO . . . . . . . . . . . . =      .43822848
 6- ATTITUDE ANGLE . . . . . . . . . . . . . . =    60.06997438      DEGREES
 7- OIL OPERATING TEMPERATURE. . . . . . . . . =   166.32343237      DEG. FAHRENHEIT
 8- FRICTIONAL TORQUE. . . . . . . . . . . . . =    26.34366507      LB. INCH
 9- AMOUNT OF OIL THAT MUST
    BE CONTINUOUSLY SUPPLIED . . . . . . . . . =      .43341209      GALLON/MIN.
10- OIL GRADE. . . . . . . . . . . . . . . . . =  (SAE   10)
11- BEARING METAL IS NO.     ( 1)     - SEE USERS MANUAL, MATERIAL LIST -
```

-RECOMMENDATIONS-

1- MAXIMUM SURFACE FINISH ROUGHNESS SHOULD BE LESS OR EQUAL TO (20-MICRON INCH)
2- SHAFT BRINELL HARDNESS MUST BE GREATER OR EQUAL TO (300)

INEQUALITY CONSTRAINTS

```
PHI( 1) =  1.70077436E+02
PHI( 2) =  1.22157339E+04
PHI( 3) =  2.42065562E+03
PHI( 4) =  9.13120615E+05
PHI( 5) =  7.49043551E+02
PHI( 6) =  1.36765676E+08
PHI( 7) =  6.00000000E+01
PHI( 8) =  0.
```

Fig 7_8a

Computer time:8.203 sec.

-IMPORTANT-

THE OPTIMUM SOLUTION FOUND IS NOT FEASIBLE ON THE MAXIMUM EXPECTED AMBIENT TEMPERATURE.
MAXIMUM EXPECTED TEMPERATURE MUST BE REDUCED TO 88.00 DEG.FAHRENHEIT
OR REOPTIMIZE WITH A HIGHER DESIGN OPERATING TEMPERATURE.

OPTIMUM SOLUTION FOUND

VALUE OF CRITERION FUNCTION. . . . . .U = 81.36432422

VALUE OF DESIGN VARIABLES

```
 1- BEARING LENGTH . . . . . . . . . . . X(1) =     9.62841465     INCH
 2- RADIAL CLEARANCE . . . . . . . . . . X(2) =      .00334531     INCH
 3- OIL OPERATING VISCOSITY. . . . . . . X(3) =     8.95475985     CENTIPOISE

 4- MINIMUM OIL FILM THICKNESS . . . . . . . =      .00194751     INCH
 5- ECCENTRICITY RATIO . . . . . . . . . . . =      .41783741
 6- ATTITUDE ANGLE . . . . . . . . . . . . . =    61.31280970     DEGREES
 7- OIL OPERATING TEMPERATURE. . . . . . . . =   167.61043590     DEG. FAHRENHEIT
 8- FRICTIONAL TORQUE. . . . . . . . . . . . =    26.94917257     LB. INCH
 9- AMOUNT OF OIL THAT MUST
    BE CONTINUOUSLY SUPPLIED . . . . . . . . =      .42545637     GALLON/MIN.
10- OIL GRADE. . . . . . . . . . . . . . . . =  (SAE  10)
11- BEARING METAL IS NO.    ( 1)    - SEE USERS MANUAL, MATERIAL LIST -
```

-RECOMMENDATIONS-

1- MAXIMUM SURFACE FINISH ROUGHNESS SHOULD BE LESS OR EQUAL TO (20-MICRON INCH)
2- SHAFT BRINELL HARDNESS MUST BE GREATER OR EQUAL TO (300)

INEQUALITY CONSTRAINTS

```
PHI( 1) =  1.67829041E+02
PHI( 2) =  1.19751267E+04
PHI( 3) =  2.65265708E+03
PHI( 4) =  9.62651702E+05
PHI( 5) =  7.65469284E+02
PHI( 6) =  1.23895641E+08
PHI( 7) =  6.00000000E+01
PHI( 8) =  0.
```

Fig 7_8b

-IMPORTANT-

THE OPTIMUM SOLUTION FOUND IS NOT FEASIBLE ON THE MAXIMUM EXPECTED AMBIENT TEMPERATURE.
MAXIMUM EXPECTED TEMPERATURE MUST BE REDUCED TO 86.00 DEG.FAHRENHEIT
OR REOPTIMIZE WITH A HIGHER DESIGN OPERATING TEMPERATURE.

Computer time:11.005 sec.

```
                 OPTIMUM SOLUTION FOUND
                 ----------------------

VALUE OF CRITERION FUNCTION. . . . . .U =     79.58552317


VALUE OF DESIGN VARIABLES
-------------------------

 1- BEARING LENGTH . . . . . . . . . . . X(1) =      9.72615563     INCH
 2- RADIAL CLEARANCE . . . . . . . . . . X(2) =       .00262610     INCH
 3- OIL OPERATING VISCOSITY. . . . . . . X(3) =      9.19981344     CENTIPOISE
 4- JOURNAL RADIUS . . . . . . . . . . . X(4) =      2.43309699     INCH

 5- MINIMUM OIL FILM THICKNESS . . . . . . .  =       .00175979     INCH
 6- ECCENTRICITY RATIO . . . . . . . . . . .  =       .32988398
 7- ATTITUDE ANGLE . . . . . . . . . . . . .  =     65.82580107     DEGREES
 8- OIL OPERATING TEMPERATURE. . . . . . . .  =    165.83468519     DEG. FAHRENHEIT
 9- FRICTIONAL TORQUE. . . . . . . . . . . .  =     23.53993382     LB. INCH
10- AMOUNT OF OIL THAT MUST
    BE CONTINUOUSLY SUPPLIED . . . . . . . .  =       .31438642     GALLON/MIN.
11- OIL GRADE. . . . . . . . . . . . . . . .  =  (SAE  10)
12- BEARING METAL IS NO.    ( 1)    - SEE USERS MANUAL, MATERIAL LIST -

-RECOMMENDATIONS-
 ---------------

 1- MAXIMUM SURFACE FINISH ROUGHNESS SHOULD BE LESS OR EQUAL TO (20-MICRON INCH)
 2- SHAFT BRINELL HARDNESS MUST BE GREATER OR EQUAL TO (300)

INEQUALITY CONSTRAINTS
----------------------

                  PHI( 1) =  1.53282981E+02
                  PHI( 2) =  1.00979042E+04
                  PHI( 3) =  3.13561467E+03
                  PHI( 4) =  1.51084487E+06
                  PHI( 5) =  7.10628985E+02
                  PHI( 6) =  1.41653148E+08
                  PHI( 7) =  6.00000000E+01
                  PHI( 8) =  0.
```

Fig 7_8c

```
                  -IMPORTANT-
                   ---------
THE OPTIMUM SOLUTION FOUND IS NOT FEASIBLE ON THE MAXIMUM EXPECTED
 AMBIENT TEMPERATURE.
MAXIMUM EXPECTED TEMPERATURE MUST BE REDUCED TO  88.00   DEG.FAHRENHEIT
OR REOPTIMIZE WITH A HIGHER DESIGN OPERATING TEMPERATURE.
```


Computer time:12.982 sec.

CONCLUSIONS

A design package has been developed that makes the problem of finding an optimum design for a hydrodynamic journal bearing, within the classes of bearings mentioned in section 7-2, much easier and available within seconds of computer time without going into a long series of trials in bearing design calculations .

This thesis illustrates how a complex design problem, such as hydrodynamic bearings, which relies heavily on the designer's judgement, can be successfully computerized so as to give an optimum design. User input is very short and simple and yet the user's judgement can still be fully exercised .

The package does not require any knowledge of computer programming, FORTRAN, or optimization theory. However a knowledge of the fundamentals of journal bearing operation and lubrication is essential. The user can directly specify his options from the calling program which has been developed in a very simple form. He may specify a value for the journal radius or leave it as a design variable, giving the upper and lower possible limits. He may also specify the oil inlet pressure, for forced-feed bearings, or considers it as one of the design variables .

The class and type of the bearing are left to the user judgement and background. Moreover, the bearing design criterion may be either minimum frictional torque, or minimum oil temperature rise, or both in a combined criterion.

A set of design failure comments will be printed out in the computer output, when an optimum solution can not be found. Those comments are based upon the results of several worked examples solved by the package.

Most of the bearing manufacturers rely largely on the limits of good practice resulting from past experience. The group of constraints that were incorporated in the design procedure seem to work satisfactorily and the optimum solutions remain within the allowable practical limits. This can be seen clearly from the results of the sample problems given in chapter VII .

Although the work discussed in this thesis does not cover all classes and types of bearings, it can be extended easily to cover all classes by some additions and minor modifications .

Appendix 1

(USER'S MANUAL)

USER'S MANUAL

FOR

OPTIMUM COMPUTER DESIGN
OF
HYDRODYNAMIC JOURNAL BEARINGS

BY. KHATTAB MOHAMED

. 1971 .

CONTENTS

Page

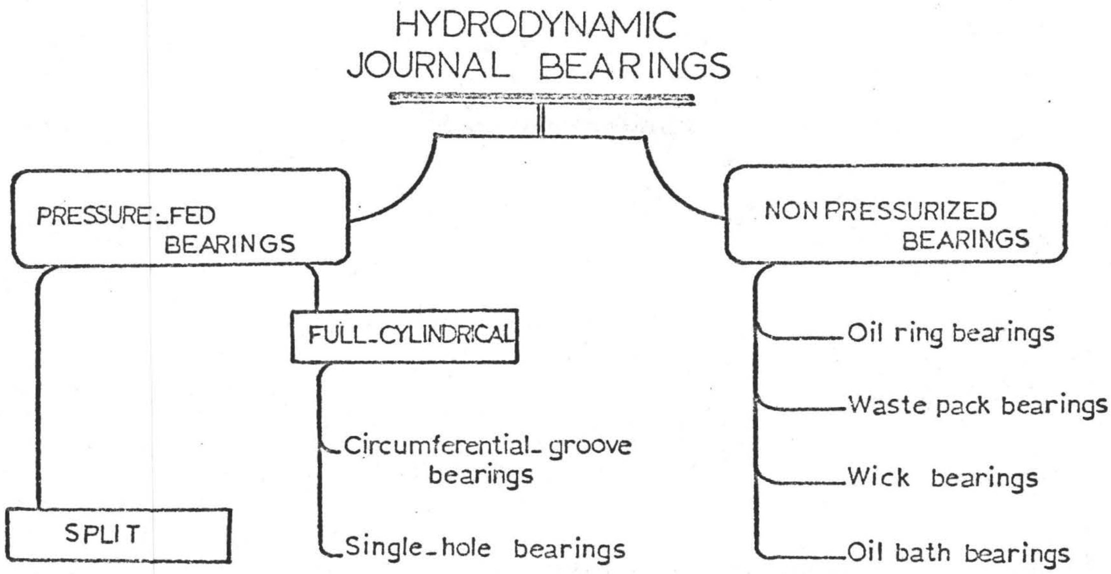
LIST OF ILLUSTRATIONS

Page

I USER'S GUIDE

1-1 Introduction

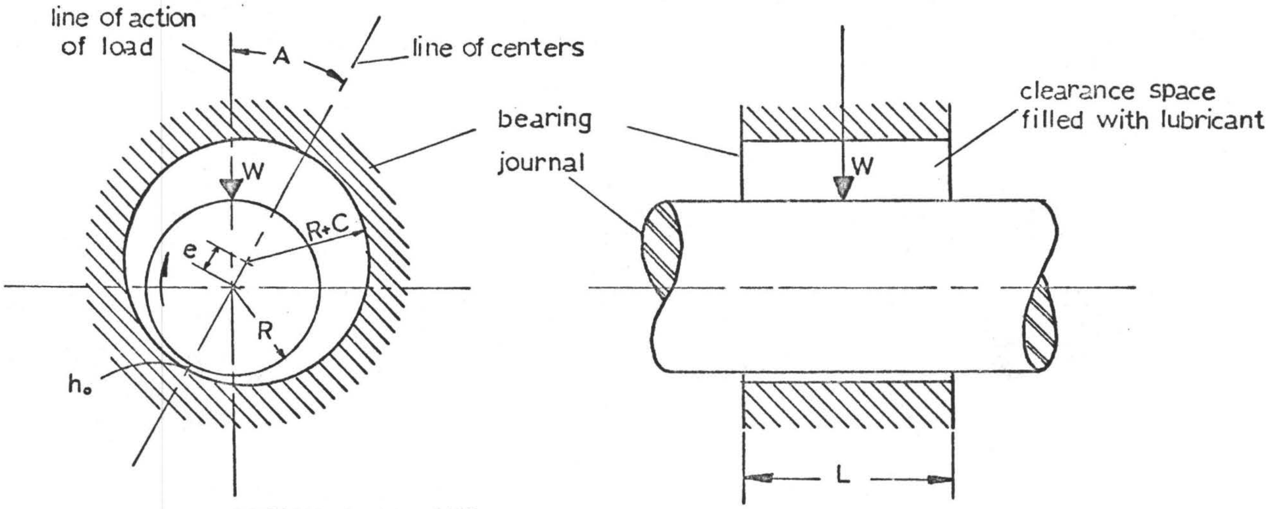
This package is one of a series for automatic optimum design of engineering components or devices. Its purpose is the optimum design of hydrodynamic journal bearings with the following options :



The package is highly user oriented and requires a minimum of knowledge of the theory of bearings, and no knowledge of computer programming, FORTRAN, or optimization. However, judgement cannot be completely removed from any design, nor can a successful design be guaranteed by any theoretical analysis if the design assumptions do not match the real life situation .
The user exercises his judgement by options available in the input coding .

1-2 Configuration

The following is a group of schematic sketchs shows the different type of bearings available in this manual.



- DESIGN VARIABLES -

L	=	Bearing length , inch
R	=	Journal radius , inch
C	=	Radial clearance , inch
e	=	Eccentricity , inch
ϵ	=	Eccentricity ratio $= \frac{e}{C}$
h_o	=	Minimum film thickness , inch
	=	$C(1-\epsilon)$ inch
A	=	Attitude angle , degrees

- Oil operating temperature , degree fahrenheit
- Oil operating viscosity , centipoise
- Frictional torque , lbs.inch
- Required amount of oil , gallon/min.
- Oil grade
- Bearing metal
- Oil inlet pressure , lb/square inch
- Oil inlet temperature , degree fahrenheit
- Axial length of chamfer , inch
- Chamfer dimension,inch

Fig 1_1 General schematic sketch showing the position of the journal in bearing & design variables

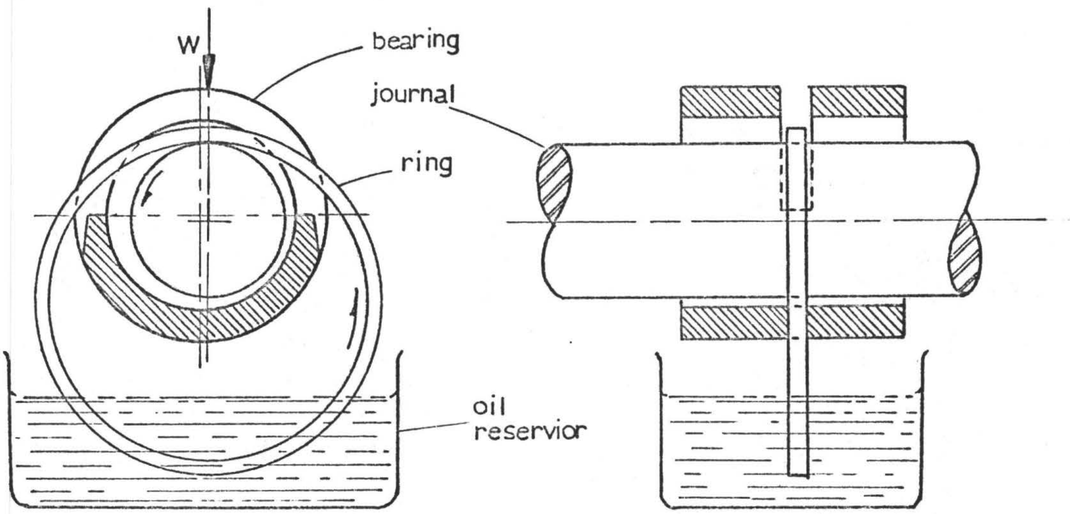


Fig1_2 Oil_ring bearing representation

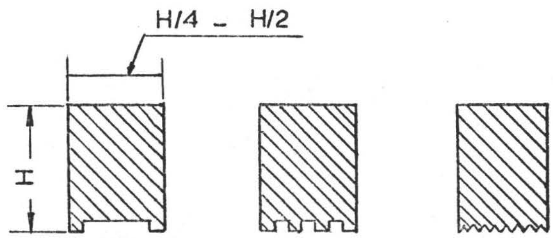


Fig1_3 Some oil_ring cross sections

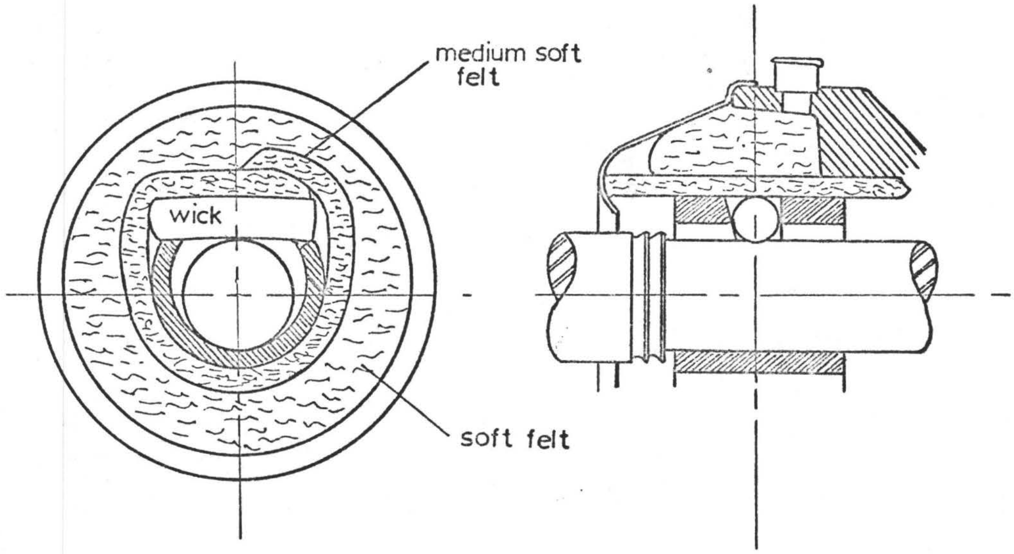


Fig1_4 Sketch showing a wick_oiled bearing for good oil_storage and retention

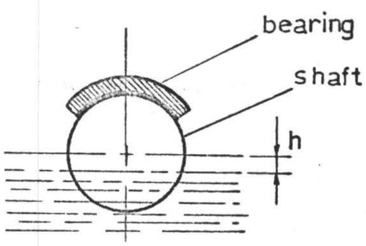


Fig1_5 Oil bath bearing representation

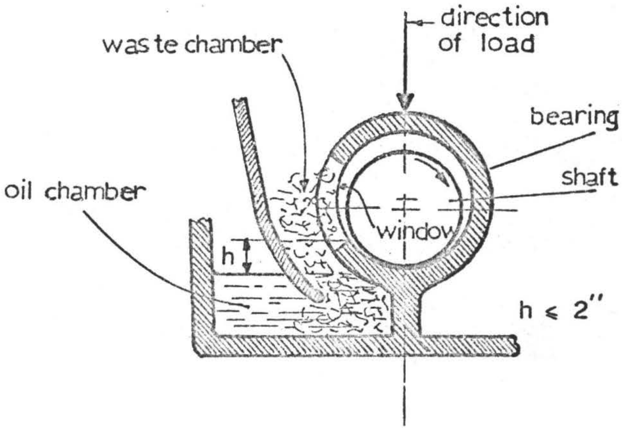


Fig1_6 Waste packed bearing representation

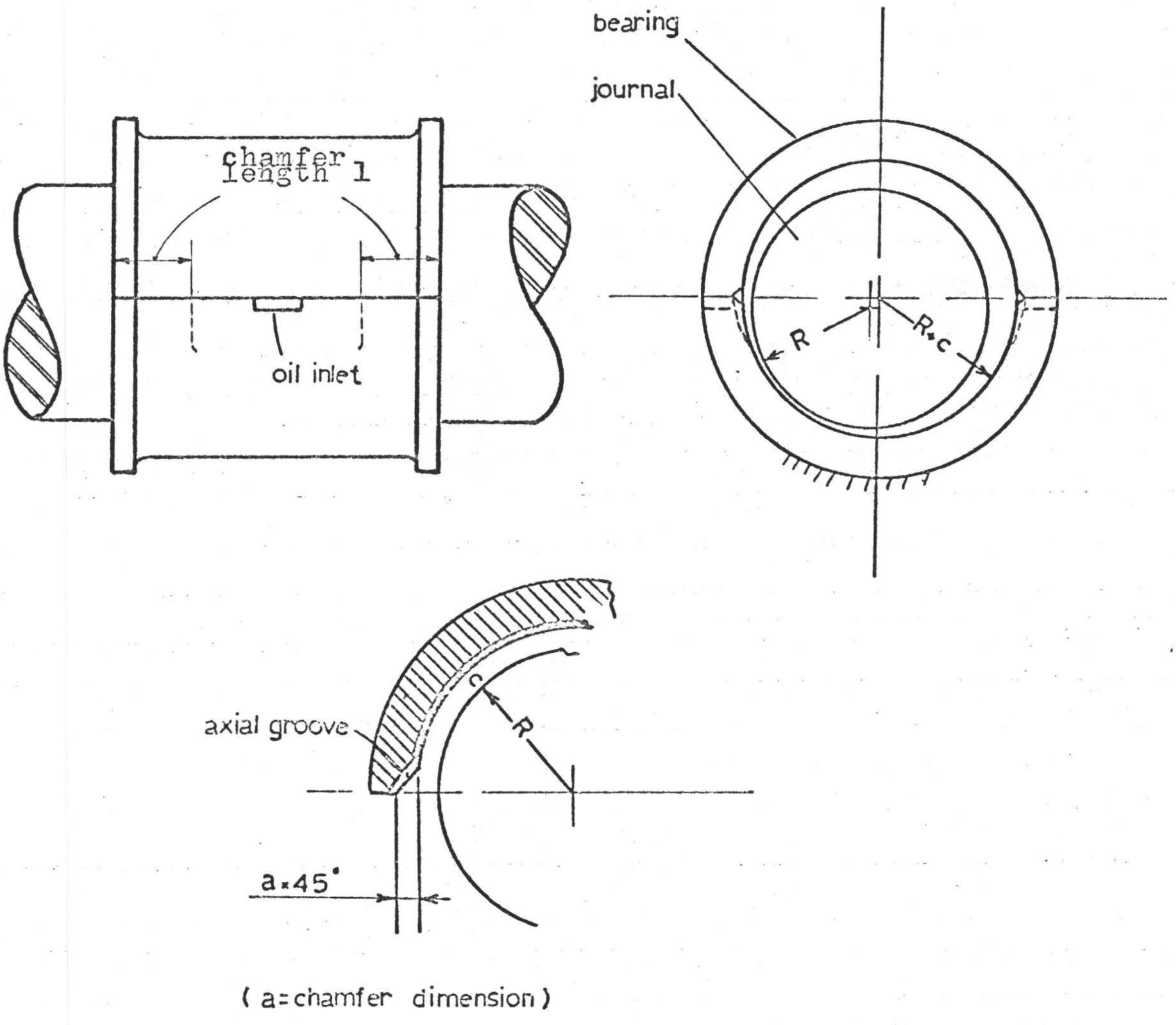


Fig 1_7 Cross section at chamfer opening for split cylindrical bearing

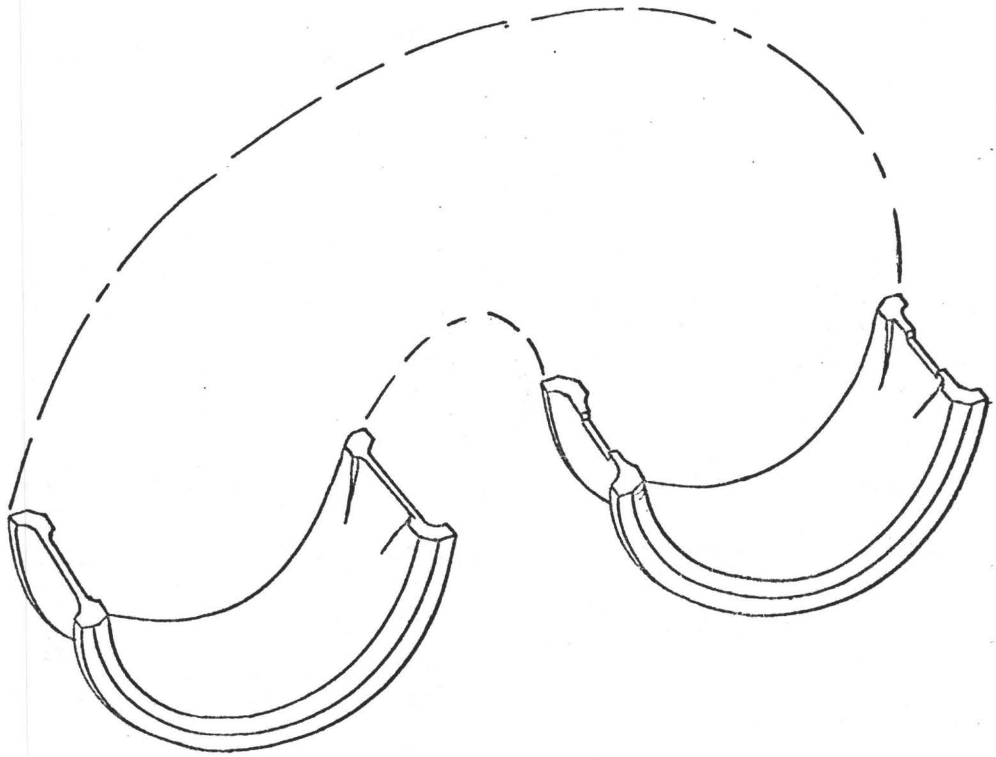


Fig 1_8 Lower and upper half of split bearing with axial oil grooves

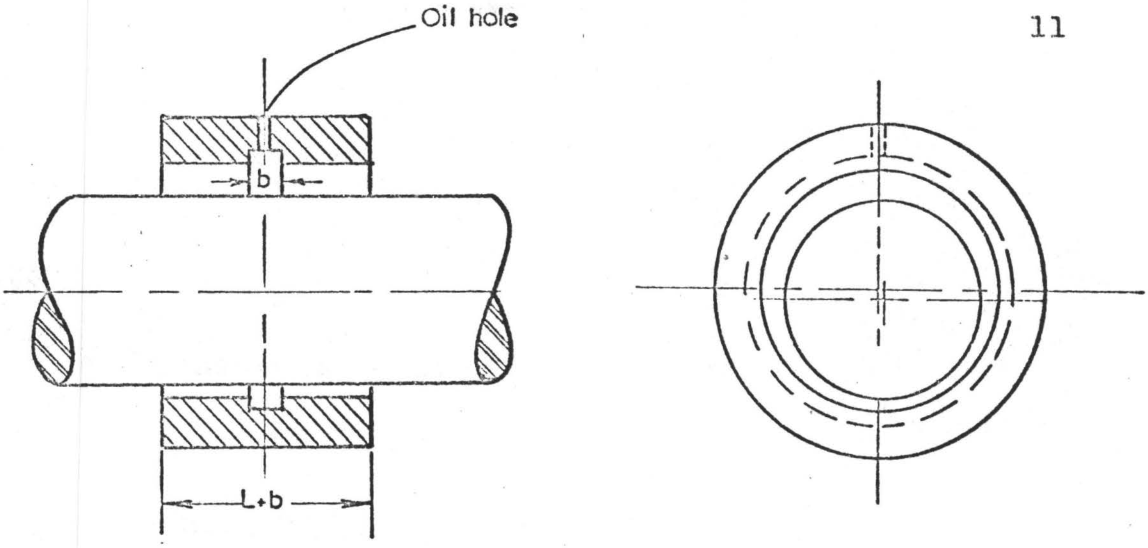


Fig 1_9 Full_cylindrical journal bearing with a circumferential oil groove

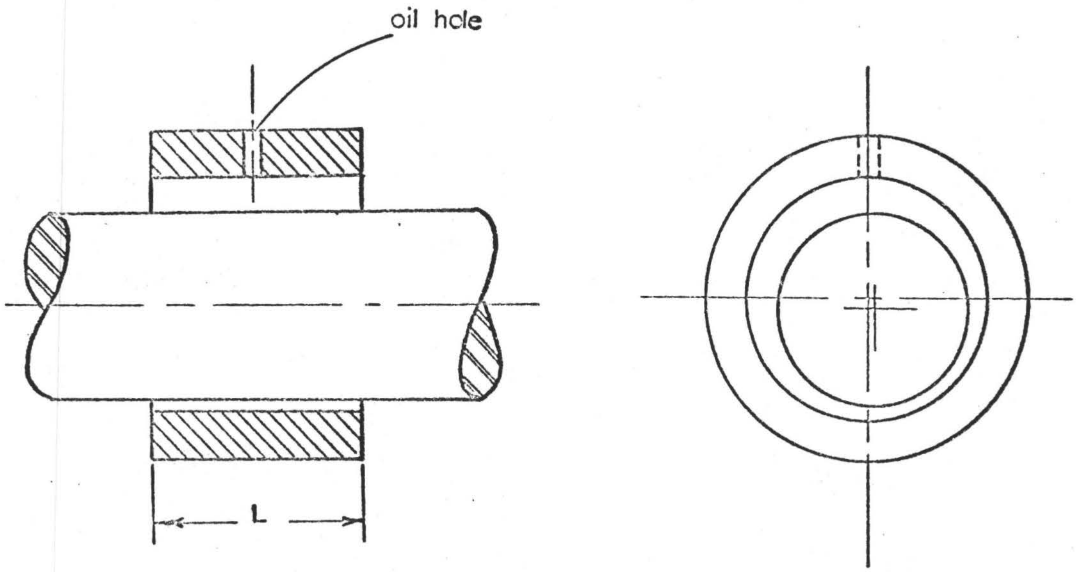


Fig 1_10 Full_cylindrical journal bearing with a single oil hole

1-3 Optimization Criterion

The user can select either of the following as the optimization criterion :

A- Minimum frictional torque loss.

B- Minimum oil - temperature rise.

C- Both criteria A and B can be used in combined criterion if the shaft input torque in lb.inch is known to the user.

In case C, since more than a single factor is to be optimized, a multifactor optimization technique is used such that the utility contribution of both factors constitutes the objective function that is to be maximized, rather than minimizing the specific value of the individuals, as in case A or case B.

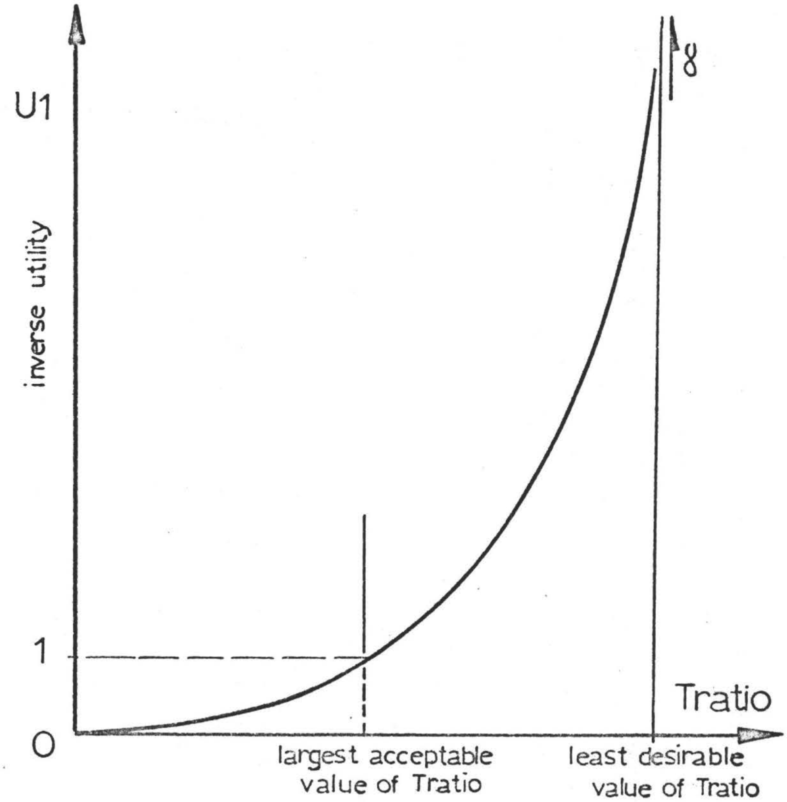
The concept of inverted utility is used here, where instead of maximizing the utility contribution of both factors, we minimize the reciprocal of the utility contribution, that is, we minimize the undesirability .

If the undesirability functional values are; U1 for the frictional torque ratio (Tratio), and U2 for the temperature rise (Δt) , see figure 1-11, then the total undesirability is given by $U1 + U2$

and the optimization objective function to be minimized is

$$U = U1 + U2$$

(a)



(b)

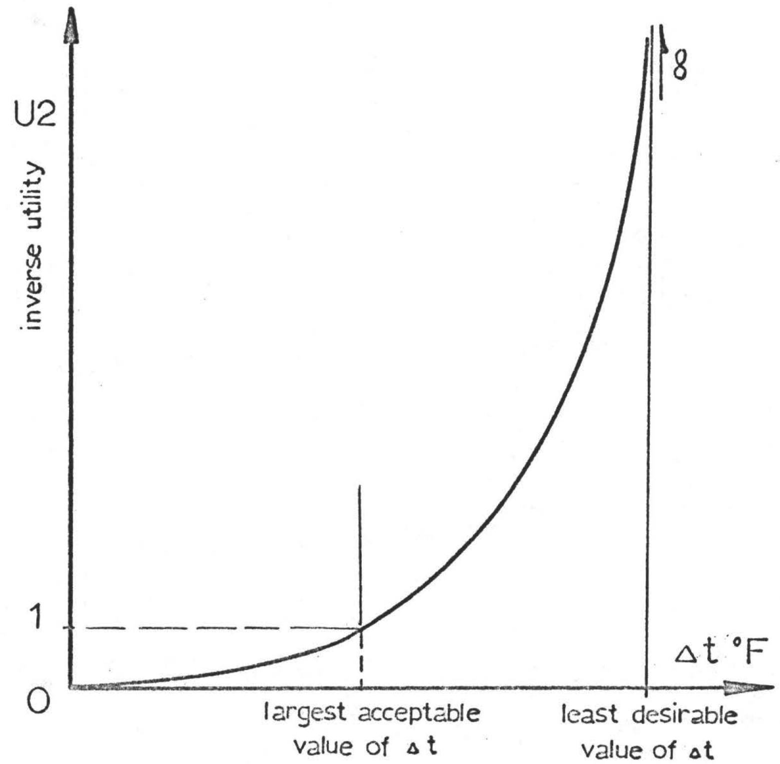


Fig 1_11 Undesirability curves for journal bearing

1-4 Input Data

The user must define quantities for all of the following input variables for both non-pressurized and pressure-fed bearings .

W	Load at journal, lbs.
RPM	Continious operating speed, revs/min.
T1	Design operating temperature, ^{O}F.
	(this is the average operating temperature. However if the optimum design does not work at the maximum expected ambient temperature, TMAX, it may be necessary to arbitrarily increase this design temperature) .
TMAX	Maximum expected ambient temperature, ^{O}F.
RADIUS	Journal radius, inch
	= 0.0 if it is one of the design variables.
	= its fixed value, if it is not one of the design variables .
RADU	Upper limit of the shaft radius, inch
	(specify the upper limit if RADIUS = 0.0)
RADL	Lower limit of the shaft radius, inch
	(specify the lower limit if RADIUS = 0.0), otherwise set RADL = RADU = RADIUS.
KOPTIM	= 1 , criterion is minimum frictional loss.
	= 2 , criterion is minimum oil-temperature rise.
	= 3 , combined criterion for minimizing both frictional torque loss and oil-temperature rise .

KAPLIC	= 1 , application for Turbo-generators
	= 2 , application for Steam turbines
	= 3 , application for Generators & motors
	= 4 , application for Machine tools
	(if user's application is none of these. Use Table 2-1 to select the type of application that has the limits of L/D ratio close to your application).
MTHD	= 7 , optimization method is DAVID
	= 5 , optimization method is SIMPLEX
	= 1 , optimization method is SEEK1
	Try SIMPLEX first, i.e, set MTHD = 5
TINPUT	Shaft torque, lb.inch
	(specify the shaft torque if KOPTIM = 3, otherwise set it up = 0.0).

* Additional input variables for Non-Pressurized bearings:

NMBR	Number of design variables,
	= 3 , if RADIUS $\neq$ 0.0 (not variable)
	= 4 , if RADIUS = 0.0 (variable)
KOILSUP	= 1 , for oil-ring bearings
	= 2 , for wick bearings
	= 3 , for waste-pack bearings
	= 4 , for oil-bath bearings
KAIR	= 1 , for quiet air ambient
	= 2 , for moving air ambient with velocity of about 500 ft/min.

* Additional input variables for Pressure-Fed bearings:

* Additional input variables for <u>Pressure-Fed</u> bearings :

PINLET Oil inlet pressure, lbs/square inch

= 0.0 if it is one of design variables

= its fixed value, if it is not variable

KBRG = 1 , for Split Bearings

= 2 , for Full Cylindrical Bearings with Circumferential - Groove

= 3 , for Full Cylindrical Bearings with Single Hole

NMBR Number of design variables, to be set up as follows:

KBRG = 1	KBRG = 2 or 3	
6	4	if RADIUS & PINLET are given i.e they are not variables
7	5	if either RADIUS or PINLET is given
8	6	if both RADIUS & PINLET are not given(both are variables)

KEY = any number, if NMBR = 4 or 6 or 8

= 1 or 2 , if NMBR = 5 or 7

where;

KEY = 1 if RADIUS is given (not variable)
KEY = 2 if PINLET is given (not variable)

The way that the input variables are entered into the user's program is illustrated in the following section .

1-5 How to Set Up the Calling Program

The calling program must have exactly the form of the following examples. Decimal points must be added where indicated, (note that no decimal point is required when the code name starts with N, M, or K) . Control cards will be required in accordance with the computer being used.

A- Calling Program for SELF - CONTAINED Class of Bearings.

```
 7                                                                  72
 COMMON/111/RPM,W,RADIUS,T1,KOPTIM,KBRG,MTHD,RADU,RADL,
1NMBR,KOILSUP,KAPLIC,PINLET,KEY,KAIR,TINPUT,TMAX

 W          1200.
 RPM         450.
 T1          75.
 TMAX        90.
 RADIUS       0.
 RADU         3.
 RADL         2.25
 NMBR         4
 TINPUT       500.
 KOPTIM       3
 KOILSUP      1
 KAPLIC       3
 KAIR         1
 MTHD         7
 CALL BRNG1
 STOP
 END
```

B- Calling Program for PRESSURE-FED Class of Bearings.

```
 7                                                                 72
 COMMON/111/RPM,W,RADIUS,T1,KOPTIM,KBRG,MTHD,RADU,RADL,
1NMBR,KOILSUP,KAPLIC,PINLET,KEY,KAIR,TINPUT,TMAX

 W          20000.
 RPM         1800.
 T1            75.
 TMAX         100.
 RADIUS         0.
 RADU           3.5
 RADL           2.5
 PINLET         0.
 NMBR           8
 TINPUT      1000.
 KOPTIM         3
 KAPLIC         2
 KBRG           1
 KEY            1
 MTHD           5
 CALL BRNG2
 STOP
 END
```

1-6 Output Information

The input data is printed for referance in the computer output. The optimum value of the criterion characteristic is printed with the corresponding values of the design variables and characteristics as follows :

1. Value of criterion function.... U
2. Values of design independent variables.... X(I)'s
3. Values of design characteristics:
   - a- Minimum oil film thickness
   - b- Eccentricity ratio
   - c- Attitude angle
   - d- Oil operating temperature
   - e- Frictional torque
   - f- Amount of oil to be supplied
   - g- Oil grade
   - h- Bearing metal
4. Recommendations
5. Values of the inequality constraint functions, are also provided for the user familiar with optimization theory. Messages and results each two iterations, related to the optimization process, may be also printed out.
6. A set of design failure comments will be printed out if an optimum solution can not be found, or if the optimum design does not work at the maximum ambient temperature .

II THEORETICAL BASIS

2-1 Introduction

The following information is not usually essential for use of the program. It is provided to assist in the event of difficulties with the package, and as a reference for further development.

A complete list of references is also provided.

2-2 Formulation for Optimization

2-2.1 Optimization Function

As mentioned in section 1-3, alternate criteria are available. Either minimum frictional torque loss, minimum oil temperature rise, or both is a combined criterion.

A. Frictional torque is obtained from Dennison's design chart [9], figure 2-1, as a function of the following variables

$$TRQ = f\,(L,R,C,W,RPM,z)$$

hence, the objective function for minimum frictional torque is,

$$U = TRQ = \text{minimum}$$

B. The oil temperature rise will depend on friction heat generated and heat taken away by the oil.

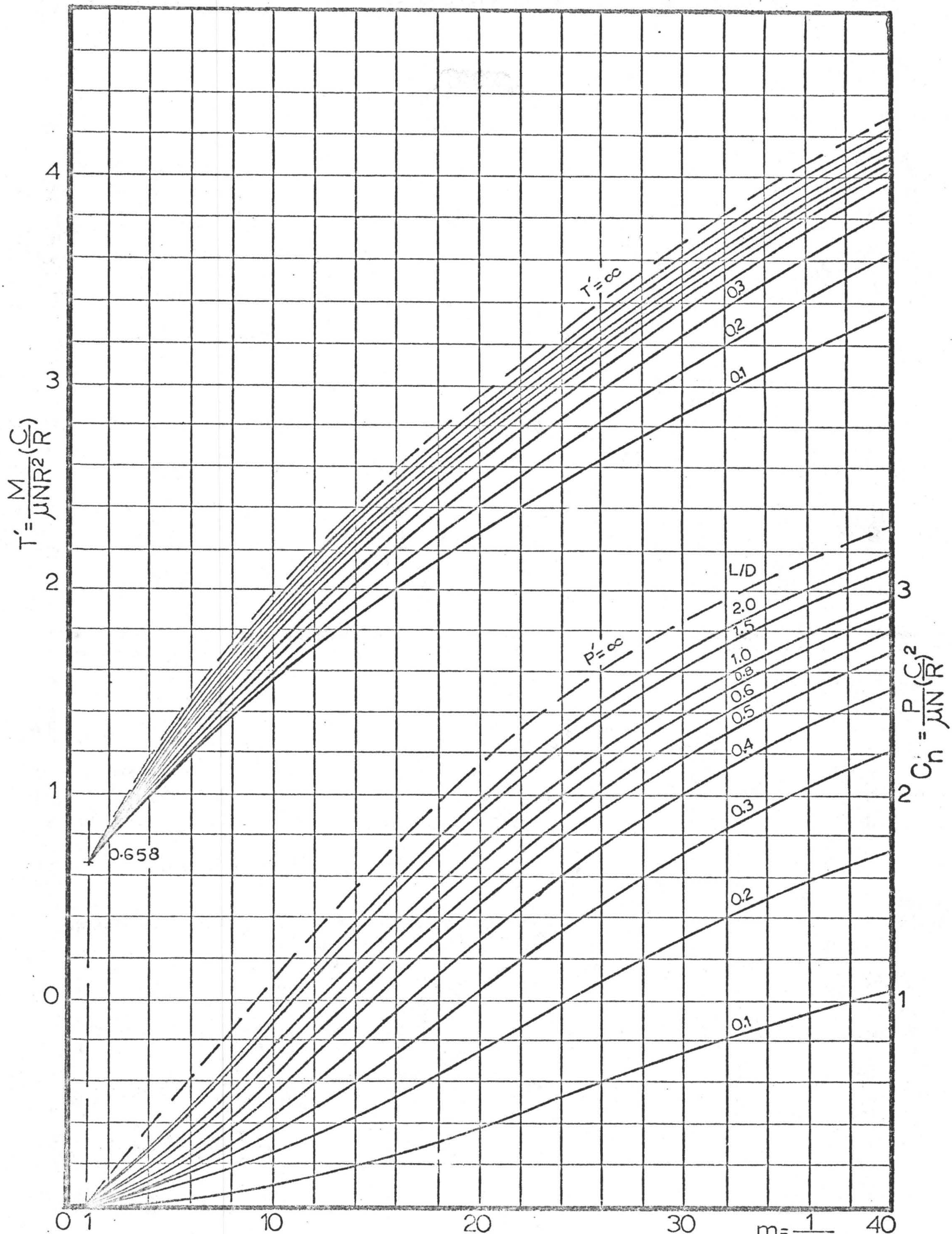


Fig 2-1 DENNISONS' Design Chart, [9]

Frictional horsepower is given by the following

$$FHP = \frac{2\ \pi . TRQ.RPM}{12 \times 33000}$$

and therefore the energy loss in friction will be

$$BTU/hr = FHP \times 2545$$

The thermal equilibrium for self-contained bearings is different than that for pressurized journal bearings.

Heat balance for self-contained journal bearings:

According to Fuller [11], the heat balance in self-contained bearings is based upon the assumption that the heat generated by friction is entirely dissipated by the housing surface to the surrounding air, hence the rate of dissipation is

$$BTU/hr = K.A. \Delta T_W \qquad (2\text{-}1)$$

where;

K is a dissipation heat factor, $Btu/hr.ft^2.^oF$.

= 2.1 for quiet air ambient.

= 5.9 for moving air ambient.

A is the hot outside area of the bearing housing

$\approx 15\ (2\ RL)$ $inch^2$

ΔT_W is the temperature rise of the outside surface above ambient air temperature.

Knowing ΔT_W from the heat balance equation (2-1), the temperature increment of the oil above the housing surface temperature - ΔT_o - can be determined from Fuller's chart, figure 2-2, where it is given as a function of ΔT_w and type of oil supply such as; oil-ring, wick, ...etc. The oil temperature rise is, then

$$\Delta T = \Delta T_w + \Delta T_o \qquad {}^oF.$$

and the oil operating temperature is

$$T_o = Tl + \Delta T \qquad {}^oF.$$

<u>Heat balance for pressure-fed journal bearings:</u>

In this case, the heat balance is based upon the assumption [11], [30], [40], that the heat generated in the oil film is carried away by the oil flowing through the bearing clearance, therefore

$$BTU/hr = Q.C_p.\Delta T \qquad (2-2)$$

where;

Q is the total oil flow through the bearing, Gal/hr

C_p is the heat capacity of lubricant, Btu/gal.^{o}F.

ΔT is the oil temperature rise, ^{o}F.

If T_{oil} is the inlet oil temperature, then the oil operating temperature will be

$$T_o = T_{oil} + \Delta T \qquad {}^oF.$$

For the criterion of minimum oil temperature rise, the optimization function is

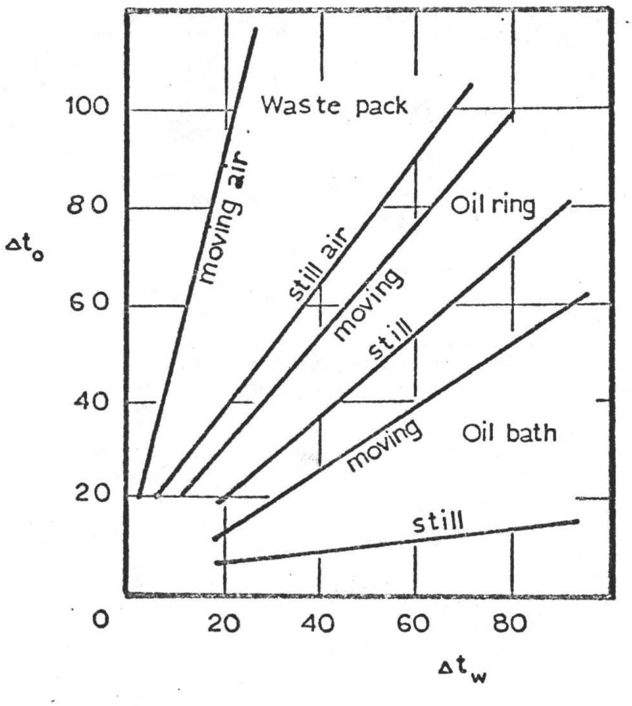


Fig 2 - 2 Temperature gradients Δt_w vs Δt_o

$$U = \Delta T = \text{minimum}$$

C. The combined criterion is explained in section 1-3.

2-2.2 Constraints

Practical limitations in bearing design that are inherently unavoidable in real problems form the major part of the following constraints which express the permissible ranges of values for different design parameters. References [10], [11], [30], [40] discuss in details those current practical limitations.

1. Length- to -diameter ratio L/D

$$(L/D)_{max} \geqslant L/D \geqslant (L/D)_{min}$$

where the maximum and minimum limits depend on the application as shown in Table 2-1 .

2. Minimum film thickness h_o

$$h_o \geqslant h_{oall}$$

where h_{oall} is the allowable lower limit. Table 2-2 gives recommended minimum values for h_o in accordance with the bearing metal used.
The actual minimum film thickness is one of the factors governing the load capacity of a hydrodynamically lubricated bearing. Equation (2-4) shows how the load carrying capacity

TABLE 2-1

Common values for L/D ratio

Machine	L/D
Turbo - generators	0.8 - 1.8
Steam turbines	0.8 - 2.0
Generators and motors	1.0 - 2.5
Machine tools	1.5 - 4.0

TABLE 2-2

Recommended minimum film thickness

No.	Bearing Metal	h_{oall}
1	Lead-base babbitt	0.00075
2	Tin-base babbitt	0.00075
3	Cadmium-base metal	0.000375
4	Copper-lead (Pb 45% ; Cu 55%)	0.000375
5	Copper-lead (Pb 25% ; Sn 3% ; Cu 72%)	0.000375
6	Silver (Lead-indium overlay)	0.000375
7	Bronzes	0.00010

depends upon the eccentricity ratio ϵ which, in turn, depends upon the minimum film thickness as given by equation (2-3) .

$$h_o = C(1 - \epsilon) \qquad (2\text{-}3)$$

$$C_n = \frac{\pi^2 . \epsilon}{5(2 + \epsilon^2)\sqrt{1 - \epsilon^2}} \qquad (2\text{-}4)$$

References [5, 11, 19, 20, 23, 33, 39, 40] discuss the minimum allowable oil film thickness.

3. Sommerfeld number S, and zN/P value

$$S \geqslant S_{min}$$

$$zN/P \geqslant (zN/P)_{min}$$

Both of these numbers is to indicate the occurrence of stable thick film lubrication. Table 2-3 gives recommended minimum values for both S and zN/P .

4. Critical shaft speed

$$N_{cr} \geqslant RPM$$

where N_{cr} is the critical shaft speed at which turbulance in the oil film start to occur. N_{cr} should be beyond the maximum speed of the shaft, and is given by [42] ,

$$N_{cr} = 392.4 \frac{\nu}{R.C} (R/C)^{\frac{1}{2}} \quad \text{rpm}$$

where;

ν is the kinematic viscosity $= \frac{z}{\rho}$ in^2/sec

TABLE 2-3

Minimum recommended values of S and zN/P

Bearing Metal	zN/P	S
Tin - base babbitt	20.00	0.050
Lead - base babbitt	10.00	0.250
Cadmium - base alloy	3.75	0.009
Copper - lead	3.75	0.009
Silver - lead - indium	2.00	0.005

from Shaw and Macks [34]

ρ is the mass density of the oil, $lb.sec^2/in^4$.

5. Oil operating temperature

$$180\ ^{o}F \geq T_o$$ for self-contained bearings [11] .

$$250\ ^{o}F \geq T_o$$ for pressure-fed bearings.

The maximum limit of $250\ ^{o}F$ for pressure-fed bearings is considered to be the most reasonable practical value for T_o.

6. Oil inlet temperature for forced-fed bearings

$$T_{oil} \geq T1 + 25 \quad ^{o}F.$$

7. Oil grade constraint

$$\text{SAE } 70 \geq \text{Grade} \geq \text{SAE } 10$$

SAE oil grades curves are shown in figure 2-4 .

8. Journal radius

$$R_{max} \geq R \geq R_{min}$$

where R_{max} and R_{min} are specified by the user if the radius is to be considered as a design variable .

9. Non - negative requirements of the variables :

bearing length	L	≥ 0.0
radial clearance	C	≥ 0.0
viscosity	z	≥ 0.0
chamfer length	l	≥ 0.0

chamfer dimension	a	$\geqslant 0.0$
oil inlet pressure	P1	$\geqslant 0.0$

2-3 Program Information

The program consists of ten FORTRAN subroutines, in addition to 13 others for optimization purposes adapted from OPTISEP [31] .

The package requires approximately 70000 storage locations in the central memory of the CDC 6400 computer .

An optimum solution can usually be found on the CDC 6400 in about eight seconds using DAVID, and in about 13 seconds using SIMPLEX or SEEK1 . In general, solutions found by SIMPLEX are better than solutions given by any other method, if SIMPLEX works .

Subroutine BRNG1 is used for self-contained bearings while subroutine BRNG2 is used for pressure-fed journal bearings. The function of these two subroutines is to select the optimization method required according to the value of the code name MTHD, as defined in section 1-4, read in design data, check the feasibility of the bearing for the maximum expected ambient temperature, and finally to output the results by calling upon subroutine ANSWER .
Figure 2-3 gives an outline of the main subroutines used.

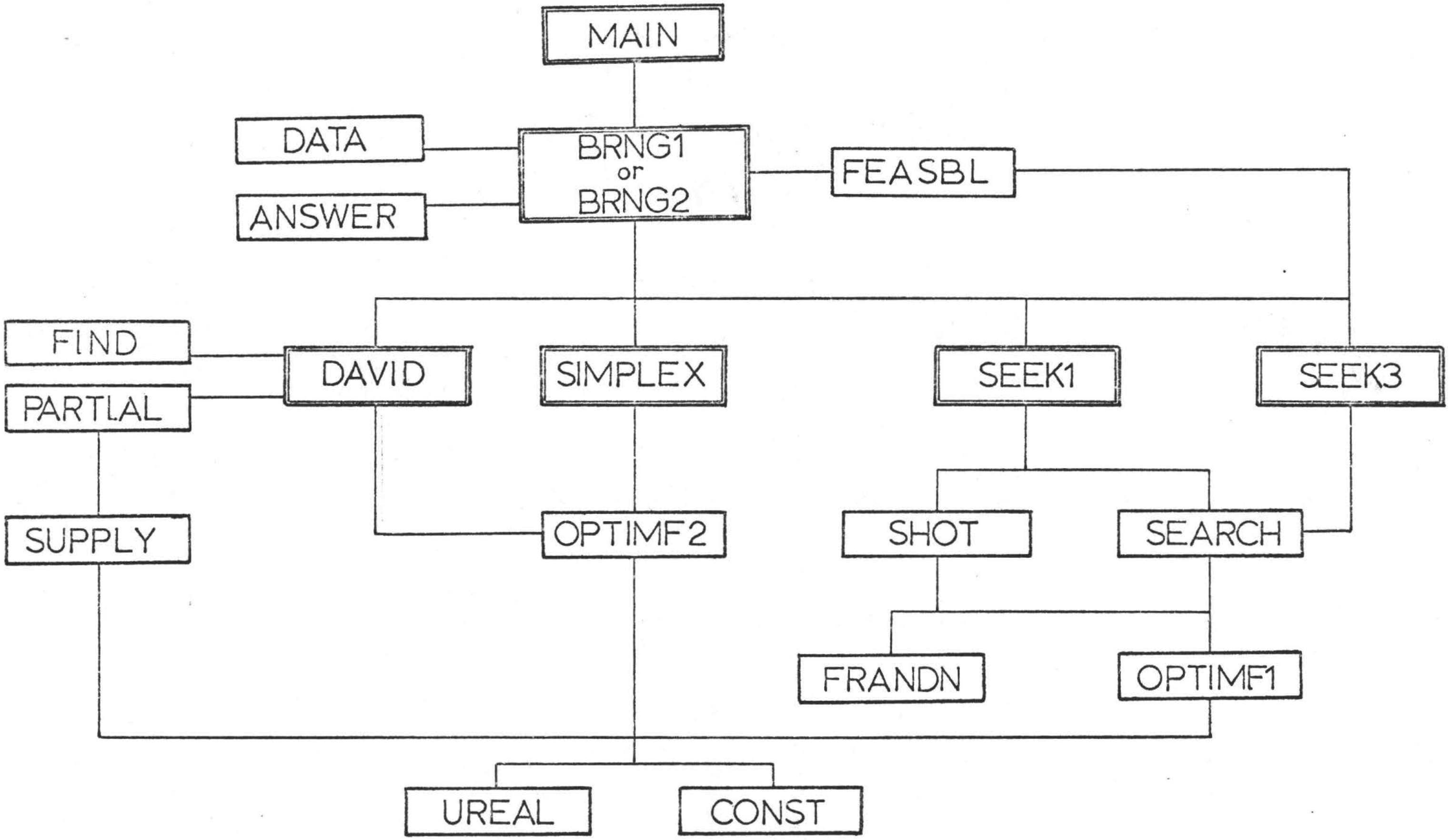
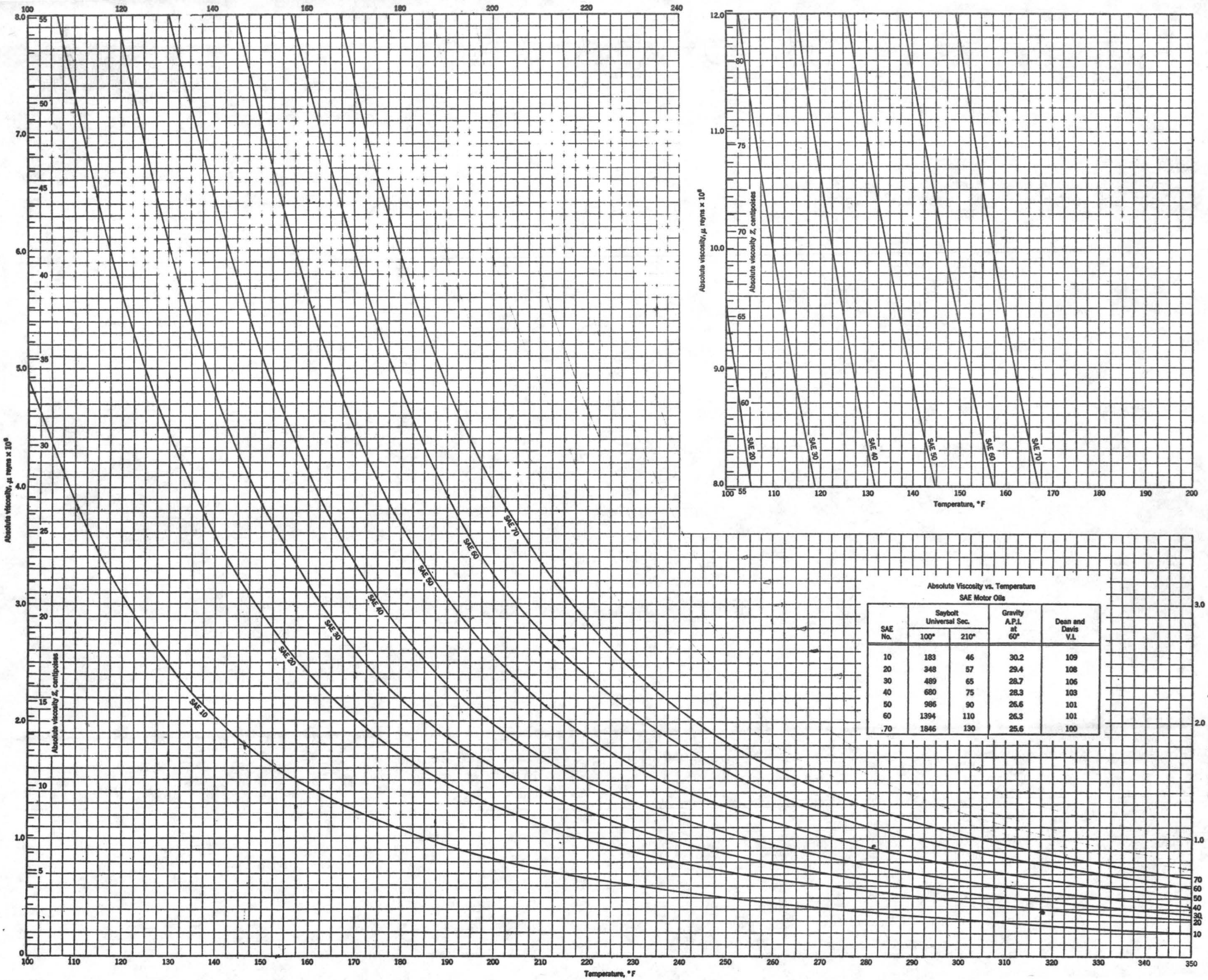


Fig 2_3 Program flow chart



Absolute Viscosity vs. Temperature

SAE Motor Oils

SAE No.	Saybolt Universal Sec. 100°	Saybolt Universal Sec. 210°	Gravity A.P.I. at 60°	Dean and Davis V.I.
10	183	46	30.2	109
20	348	57	29.4	108
30	489	65	28.7	106
40	680	75	28.3	103
50	986	90	26.6	101
60	1394	110	26.3	101
.70	1846	130	25.6	100

Fig 2-4

III SOME DESIGN CONSIDERATIONS

3-1 Ring-oiled bearings

The ring must be made of a heavy material. Brass is recommended. For small rings sizes, die-cast zinc finds a considerable use .

The diameter of the ring is about 1.5 times the journal diameter. The cross section is generally rectangular, where the width of the ring frequently from one half to one fourth the radial height.
Grooves on the inner surface are usually about 1/16" wide and from 1/32" to 1/16" depth. They should be spaced from 1/16" to 1/32" apart.

3-2 Waste-packed bearings

Wool-waste is recommended as packing material. Lamp wick or cotton thread should be used when the mechanical properties are of secondry importance, as in auxiliary oil drippers, syphon feeds, ... etc .

The oil saturated waste packing should cover the whole window, see figure 1-6. A wick long enough to reach the bottom of the oil well can be placed across the window and extended so as to fill the waste. The pack must be tamped very tight to force the wick against the journal and to prevent any loosening of the waste.

3-3 Wick bearings

Figure 1-4 indicates a good way of providing good oil storage capacity and oil retention feature.
The felt wick shown has a cylindrical shape. The packing is a soft felt or mass of fibers more deformable than the wick felt. It should fill the cavities around the wick to provide good oil storage and to decrease leakage and oxidation .

Wick and packing materials are best made from high quality wool fibers. These should not contain any alkaline or acidic materials in order to decrease the rate of oil oxidation.
A packing material should have a good oil storage and absorption capacity.

An X-shaped grooves is used on the bearing surface.

3-4 Bearing material

Bearing materials are selected according to the maximum pressure in the oil film, rather than the unit load carried by the bearing.

Experimently, Needs [27] has determined the ratios between the maximum pressure in the oil film and the unit load (load/inch square of bearing projected area) as a function of the eccentricity ratio ϵ and the dimensions of the bearing.
Figure 3-1 shows the chart developed by Needs for central partial bearings having an angular length of 120° arc, however, the chart may be used also for determining approximately the maximum pressure for all types of journal bearing [27] .

Many references [11, 30, 33, 40] and others explaining the bearing material requirements for a successfully working bearing .
Tables 3-2, 3-3, and 3-4 give the recommended unit load values and other properties of various bearing metals.
Table 3-1 gives the material list that has been considered, according to the information given by the abovementioned tables, the bases of selecting the bearing meatal .

TABLE 3-1

Bearing materials list

No.	Bearing Metal	Maximum Unit Load, psi
1	Lead-base babbitt	up to - 800
2	Tin-base babbitt	800 - 1200
3	Cadmium-base	1200 - 1500
4	Copper-lead (Pb 45% ; Cu 55%)	1500 - 3000
5	Copper-lead (Pb 25% ; Sn 3% ; Cu 72%)	3000 - 4000
6	Silver (overplated)	4000 - up
7	Bronzes	10000

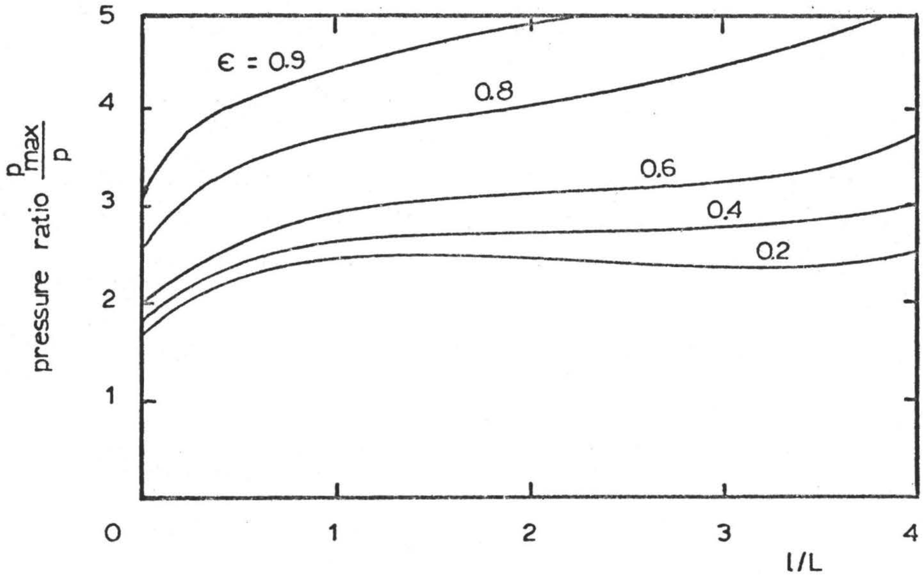


Fig 3–1 Pressure ratio against l/L for various eccentricity ratios, [27]

TABLE 3-3

Maximum values of unit load

Bearing Metal	Maximum unit load, psi
Lead-base babbitt	600 - 800
Tin-base babbitt	800 - 1000
Cadmium-base	1200 - 1500
Cadmium-base with 0.003-0.004" overlay of babbitt	2000 - 4000
Copper-lead (Pb 45% ; Cu 55%)	2000 - 3000
Copper-lead (Pb 25% ; Sn 3% ; Cu 72%)	3000 - 4000
Silver (lead-indium overlay)	5000 - up
Bronzes	10000

From Etchells and Underwood [10]

TABLE 3-4

Fatigue strength, maximum load,

and deformability of bearing metals

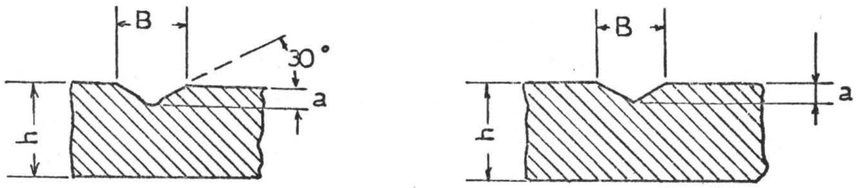
Bearing Alloy	Order of Fatigue Strength	Approximate Maximum Load (psi)	Order of Deformability
Bronzes	1	10000	7
Copper-lead with tin or silver	2	3000 - 4000	6
Thin-babbitt overlays(0.003" or less)	3	2000 - 4000	5
Aluminum alloys	4	2000 - 3000	4
Copper-lead alloys	5	1500 - 2500	3
Cadmium alloys	6	1200 - 1500	2
Lead-and tin-base babbitt	7	800 - 1500	1

From Fuller [11]

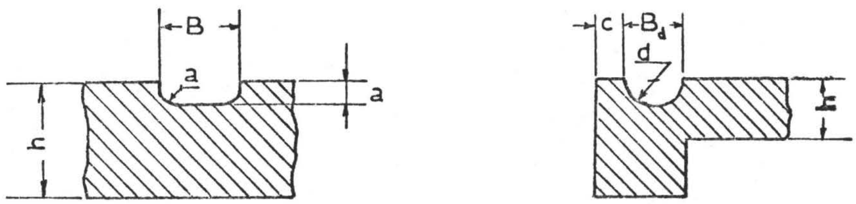
3-5 Grooving

There are two main kinds of lubrication grooves; axial grooves for lubricant distribution along the axis of the bearing, and circumferential grooves for lubricant distribution around the journal .
Drainage grooves may be placed at the ends of the bearing to collect the lubricant that is ordimarily forced out of the bearing ends. Figure 3-2 shows cross sections through some types of lubrication and drainage grooves. Grooves must be carefully chamfered or rounded to avoid disrupting the continuity of the oil film or scraping the oil from the journal .

In horizontal forced-feed bearings, it is recommended that the lubricant supply hole be placed at the middle of the bearing. For vertical bearings it should be nearer the middle than the top to avoid top leakage. The supply hole in either case should be anywhere within the no-load sector of the bearing (approximately at the attitude angle -A- as shown in figure 1-1. Figure 3-3 gives the attitude angle as a function of the eccentricity ratio and L/D ratio. If the load is rotating the lubricant may be supplied from the rotating shaft through a radial groove.



(a) Lubrication Grooves



(b) Drainage Grooves

Fig 3-2 Lubrication & drainage grooves

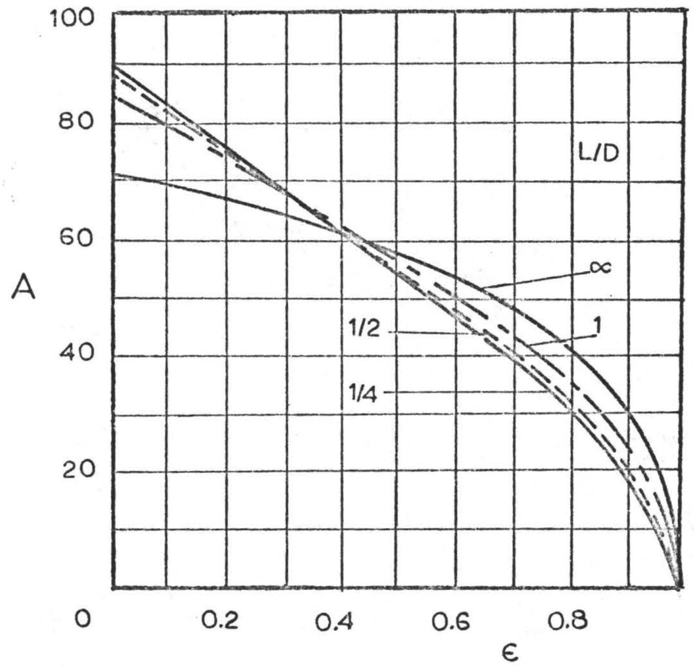


Fig 3-3 Attitude angle A vs eccentricity ratio ϵ for different L/D ratio

IV SAMPLE PROBLEMS

Problem 1

A force-fed lubricated journal bearing is to be designed for a steam turbine with the following data, [38].

Applied load	10000	lbs.
Journal speed	1800	rpm.
Journal radius	3	in.

Solution

We assume the following :

a. The bearing is full-cylindrical with a circumferential oil groove .

b. The journal radius is a design variable.

c. The oil inlet pressure is a design variable.

d. The design ambient temperature is 75 ^{o}F .

e. The maximum ambient temperature expected is 95 ^{o}F .

Under these conditions the problem is solved using two methods; SIMPLEX, and DAVID for a combined criterion of minimum friction loss and minimum temperature rise. The shaft torque is considered 1000 lb.inch .

The calling program, using SIMPLEX, is set up as shown in figure 4-1. The input information as given by the calling program is printed out for reference. This is shown in figure 4-1a. The optimum solution found is shown in figure 4-1b . The solution found using DAVID is given in figure 4-2b. Figures 4-2 and 4-2a show the calling program, in this case, and the intermediate output each two iterations in the optimization process respectivily .

Observe that SIMPLEX gives a solution better than that given by DAVID .

```
      COMMON /111/ RPM,W,RADIUS,T1,KOPTIM,KBRG,MTHD,RADU,RADL,NMBR,KOILS
     1UP,KAPLIC,PINLET,KEY,KAIR,TINPUT,TMAX
      W=10000.
      RPM=1800.
      T1=75.
      TMAX=95.
      RADIUS=0.0
      RADU=3.5
      RADL=2.5
      KOPTIM=3
      KAPLIC=2
      PINLET=0.0
      TINPUT=1000.
      NMBR=6
      KBRG=2
      KEY=2
      MTHD=5
      CALL BRNG2
      STOP
      END
```

Fig 4_1

- OPTIMUM HYDRODYNAMIC BEARING DESIGN -

INPUT DATA

LOAD AT BEARING, LBS.................................	W	=	1.0000000E+04
JOURNAL SPEED, REVS/MIN.	RPM	=	1.8000000E+03
AMBIENT TEMPERATURE, DEG.F.	T1	=	7.5000000E+01
MAXIMUM EXPECTED AMBIENT TEMPERATURE	TMAX	=	9.5000000E+01
JOURNAL RADIUS, INCH	RADIUS	=	0.
ESTIMATED UPPER LIMIT OF RADIUS	RADU	=	3.5000000E+00
ESTIMATED LOWER LIMIT OF RADIUS	RADL	=	2.5000000E+00
OPTIMIZATION CRITERION	KOPTIM	=	3
TYPE OF APPLICATION	KAPLIC	=	2
NUMBER OF DESIGN VARIABLES	NMBR	=	6
OPTIMIZATION METHOD USED	MTHD	=	5
INPUT TORQUE	TINPUT	=	1.0000000E+03
TYPE OF JOURNAL BEARING USED......................	KBRG	=	2
FLAG NUMBER	KEY	=	2
OIL INLET PRESSURE	PINLET	=	0.

Fig 4_1a

OPTIMUM SOLUTION FOUND

VALUE OF CRITERION FUNCTION. . . . . .U = .06431820

VALUE OF DESIGN VARIABLES

```
 1- BEARING LENGTH . . . . . . . . . . . . . X(1) =      5.18245489     INCH
 2- RADIAL CLEARANCE . . . . . . . . . . . . X(2) =       .00338653     INCH
 3- OIL OPERATING VISCOSITY. . . . . . . . . X(3) =      3.89042817     CENTIPOISE
 4- JOURNAL RADIUS . . . . . . . . . . . . . X(4) =      2.50000021     INCH
 5- OIL INLET PRESSURE . . . . . . . . . . . X(5) =     10.80892366     LB/SQ.INCH
 6- OIL INLET TEMPERATURE. . . . . . . . . . X(6) =    220.44570233     DEG. FAHRENHEIT

 7- MINIMUM OIL FILM THICKNESS . . . . . . . .      =       .00037722     INCH
 8- ECCENTRICITY RATIO . . . . . . . . . . . .      =       .88861129
 9- ATTITUDE ANGLE . . . . . . . . . . . . . .      =     20.04996743     DEGREES
10- OIL OPERATING TEMPERATURE. . . . . . . . .      =    237.68608887     DEG. FAHRENHEIT
11- FRICTIONAL TORQUE. . . . . . . . . . . . .      =     42.27836810     LB. INCH
12- AMOUNT OF OIL THAT MUST
    BE CONTINUOUSLY SUPPLIED . . . . . . . . .      =       .39795153     GALLON/MIN.
13- OIL GRADE. . . . . . . . . . . . . . . . .      =  (SAE  10)
14- BEARING METAL IS NO.      ( 4)      - SEE USERS MANUAL, MATERIAL LIST -
```

-RECOMMENDATIONS-

```
1- MAXIMUM SURFACE FINISH ROUGHNESS SHOULD BE LESS OR EQUAL TO (20-MICRON INCH)
2- SHAFT BRINELL HARDNESS MUST BE GREATER OR EQUAL TO (300)
```

INEQUALITY CONSTRAINTS

```
PHI( 1) =  1.41457732E+01
PHI( 2) =  2.22157732E+01
PHI( 3) =  1.48860592E+02
PHI( 4) =  4.42057813E+05
PHI( 5) =  6.61346745E+02
PHI( 6) =  1.23139111E+01
PHI( 7) =  6.00000000E+10
PHI( 8) =  0.
PHI( 9) =  8.91910763E+01
```

Fig 4_1b

Computer time: 12.339 sec

```
      COMMON /111/ RPM,W,RADIUS,T1,KOPTIM,KBRG,MTHD,RADU,RADL,NMBR,KOILS
     1UF,KAPLIC,PINLET,KEY,KAIR,TINPUT,TMAX
      W=10000.
      RPM=1800.
      T1=75.
      TMAX=95.
      RADIUS=0.0
      RADU=3.5
      RADL=2.5
      KOPTIM=3
      KAPLIC=2
      PINLET=0.0
      TINPUT=1000.
      NMBR=6
      KBRG=2
      KEY=2
      MTHD=7
      CALL BRNG2
      STOP
      END
```

Fig 4_2

INTERMEDIATE OUTPUT FOR DAVIDON FLETCHER AND POWELL

UAPT IS THE ARTIFICIAL UNCONSTRAINED OPTIMIZATION FUNCTION

STEP NO.	U	UART	INDEPENDENT VARIABLES X(I)			
2	8.84919962E-01	9.69776216E-01	6.12000925E+00	8.50026612E-03	1.50000462E+01	3.00000798E+00
			1.00000000E+01	1.50000000E+02		
4	8.75792996E-01	8.80586074E-01	6.12004952E+00	9.29949101E-03	1.50002490E+01	3.00004658E+00
			1.00000000E+01	1.50000000E+02		
6	4.99376637E-01	5.07700741E-01	6.15951483E+00	7.09395179E-03	1.51963794E+01	3.03789863E+00
			9.99993327E+00	1.50000000E+02		
8	4.98782708E-01	5.07066721E-01	6.15953934E+00	7.02459199E-03	1.51966168E+01	3.03792104E+00
			9.99993320E+00	1.50000000E+02		
10	2.59082473E-01	2.63449535E-01	6.18575251E+00	5.46535358E-03	1.53165525E+01	3.06225234E+00
			9.99988015E+00	1.50000000E+02		
12	2.58536333E-01	2.62811560E-01	6.18576404E+00	5.39271880E-03	1.53167422E+01	3.06226249E+00
			9.99988010E+00	1.50000000E+02		
14	2.22399883E-01	2.29842783E-01	6.18739130E+00	5.15319372E-03	1.53346998E+01	3.06370521E+00
			9.99987482E+00	1.50000000E+02		
16	2.22003852E-01	2.26997621E-01	6.18740597E+00	5.02916552E-03	1.53349580E+01	3.06371811E+00
			9.99987475E+00	1.50000000E+02		
18	2.21740085E-01	2.22105364E-01	6.18740965E+00	5.13957041E-03	1.53350232E+01	3.06372135E+00
			9.99987474E+00	1.50000000E+02		
20	2.18701006E-01	2.18962420E-01	6.18749990E+00	5.00052837E-03	1.53366156E+01	3.06380069E+00
			9.99987433E+00	1.50000000E+02		
22	2.18405182E-01	2.18775768E-01	6.18750353E+00	5.07999997E-03	1.53366801E+01	3.06380388E+00
			9.99987432E+00	1.50000000E+02		
24	2.17063181E-01	2.18002651E-01	6.18754117E+00	5.08581339E-03	1.53373479E+01	3.06383697E+00
			9.99987415E+00	1.50000000E+02		
26	2.15467214E-01	2.15837109E-01	6.18758709E+00	5.02530239E-03	1.53381641E+01	3.06387733E+00
			9.99987394E+00	1.50000000E+02		

Fig 4_2a

OPTIMUM SOLUTION FOUND

VALUE OF CRITERION FUNCTION. . . . . .U = .21546721

VALUE OF DESIGN VARIABLES

1-	BEARING LENGTH	X(1) =	4.06444311	INCH
2-	RADIAL CLEARANCE	X(2) =	.00502530	INCH
3-	OIL OPERATING VISCOSITY.	X(3) =	7.31529791	CENTIPCISE
4-	JOURNAL RADIUS	X(4) =	2.50602752	INCH
5-	OIL INLET PRESSURE	X(5) =	9.99987394	LB/SQ.INCH
6-	OIL INLET TEMPERATURE.	X(6) =	163.88103870	DEG. FAHRENHEIT
7-	MINIMUM OIL FILM THICKNESS	=	.00037660	INCH
8-	ECCENTRICITY RATIO	=	.92505887	
9-	ATTITUDE ANGLE	=	13.03975681	DEGREES
10-	OIL OPERATING TEMPERATURE.	=	181.23824356	DEG. FAHRENHEIT
11-	FRICTIONAL TORQUE.	=	52.81865977	LB. INCH
12-	AMOUNT OF OIL THAT MUST BE CONTINUOUSLY SUPPLIED	=	.43520713	GALLON/MIN.
13-	OIL GRADE.	=	(SAE 10)	

14- BEARING METAL IS NO. (4) - SEE USERS MANUAL, MATERIAL LIST -

-RECOMMENDATIONS-

1- MAXIMUM SURFACE FINISH ROUGHNESS SHOULD BE LESS OR EQUAL TO (20-MICRON INCH)
2- SHAFT BRINELL HARDNESS MUST BE GREATER OR EQUAL TO (300)

INEQUALITY CONSTRAINTS

PHI(1) = 2.28238680E+01
PHI(2) = 1.60184524E+01
PHI(3) = 7.11270053E+01
PHI(4) = 4.48410175E+05
PHI(5) = 4.99880770E+02
PHI(6) = 6.87617564E+01
PHI(7) = 6.00000000E+10
PHI(8) = 0.
PHI(9) = 9.00001261E+01

Fig 4_2b

Computer time: 12.496 sec

-IMPORTANT-

THE OPTIMUM SOLUTION FOUND IS ONLY FEASIBLE FOR THE DESIGN OPERATING
TEMPERATURE - 75.00 DEG. FAHRENHEIT .
OR REOPTIMIZE WITH A HIGHER DESIGN OPERATING TEMPERATURE .

Problem 2

A waste-packed bearing is to be designed for a turbo-generator with the following specifications.

Applied load	1120	lbs.
Journal speed	350	rpm.
Journal radius	4.5	in.

Solution

The problem is solved for a criterion of minimum temperature rise using DAVID. The journal radius is considered a design variable.
Figure 4-3 and figure 4-3a show the calling program and the print out of the input informations respectivily. The optimum solution found is shown in figure 4-3b .

```
      COMMON /111/ RPM,W,RADIUS,T1,KOPTIM,KBRG,MTHD,RADU,RADL,NMBR,KOILS
     1UP,KAPLIC,PINLET,KEY,KAIR,TINPUT,TMAX
      W=1120.
      RPM=350.
      T1=75.
      TMAX=100.
      RADIUS=0.0
      RADU=5.
      RADL=4.
      NMBR=4
      KOPTIM=2
      KAPLIC=1
      KOILSUP=3
      TINPUT=0.0
      KAIR=1
      MTHD=7
      CALL BRNG1
      STOP
      END
```

Fig 4_3

- OPTIMUM HYDRODYNAMIC BEARING DESIGN -

INPUT DATA

LOAD AT BEARING, LBS.................................. .	W	=	1.1200000E+03
JOURNAL SPEED, REVS/MIN.	RPM	=	3.5000000E+02
AMBIENT TEMPERATURE, DEG.F.	T1	=	7.5000000E+01
MAXIMUM EXPECTED AMBIENT TEMPERATURE	TMAX	=	1.0000000E+02
JOURNAL RADIUS, INCH	RADIUS	=	0.
ESTIMATED UPPER LIMIT OF RADIUS	RADU	=	5.0000000E+00
ESTIMATED LOWER LIMIT OF RADIUS	RADL	=	4.0000000E+00
OPTIMIZATION CRITERION	KOPTIM	=	2
TYPE OF APPLICATION	KAPLIC	=	1
NUMBER OF DESIGN VARI BLES	NMBR	=	4
OPTIMIZATION METHOD USED	MTHD	=	7
INPUT TORQUE	TINPUT	=	0.
KIND OF OIL SUPPLY	KOILSUP	=	3
AMBIENT AIR CONDITION	KAIR	=	1

Fig 4_3a

OPTIMUM SOLUTION FOUND

VALUE OF CRITERION FUNCTION. . . . . . .U = 89.86828558

VALUE OF DESIGN VARIABLES

1-	BEARING LENGTH	X(1) =	14.71266187	INCH
2-	RADIAL CLEARANCE	X(2) =	.00600178	INCH
3-	OIL OPERATING VISCOSITY.	X(3) =	8.00414687	CENTIPOISE
4-	JOURNAL RADIUS	X(4) =	4.62177058	INCH
5-	MINIMUM OIL FILM THICKNESS	=	.00420419	INCH
6-	ECCENTRICITY RATIO	=	.29950915	
7-	ATTITUDE ANGLE	=	67.92699675	DEGREES
8-	OIL OPERATING TEMPERATURE.	=	174.99866968	DEG. FAHRENHEIT
9-	FRICTIONAL TORQUE.	=	71.26409397	LB. INCH
10-	AMOUNT OF OIL THAT MUST BE CONTINUOUSLY SUPPLIED	=	1.71611989	GALLON/MIN.
11-	OIL GRADE.	=	(SAE 10)	

12- BEARING METAL IS NO. (1) - SEE USERS MANUAL, MATERIAL LIST -

-RECOMMENDATIONS-

1- MAXIMUM SURFACE FINISH ROUGHNESS SHOULD BE LESS OR EQUAL TO (20-MICRON INCH)
2- SHAFT BRINELL HARDNESS MUST BE GREATER OR EQUAL TO (300)

INEQUALITY CONSTRAINTS

PHI(1) = 3.30168977E+02
PHI(2) = 3.45419126E+04
PHI(3) = 4.62249200E+03
PHI(4) = 2.76836892E+05
PHI(5) = 1.24853033E+03
PHI(6) = 5.00133032E+07
PHI(7) = 6.00000000E+01
PHI(8) = 0.

Fig 4_3b

Computer time: 3.897 sec

-IMPORTANT-

THE OPTIMUM SOLUTION FOUND IS NOT FEASIBLE ON THE MAXIMUM EXPECTED AMBIENT TEMPERATURE.
MAXIMUM EXPECTED TEMPERATURE MUST BE REDUCED TO 80.00 DEG.FAHRENHEIT OR REOPTIMIZE WITH A HIGHER DESIGN OPERATING TEMPERATURE.

V . PROGRAM LISTING

CONTENTS

```
      SUBROUTINE BRNG1
C     ***************
C
C
C        THIS SUBROUTINE IS USED MAINLY TO SELECT THE OPTIMIZATION
C       SUBROUTINE  ,  TO OUTPUT THE FINAL SOLUTION BY CALLING UPON
C       SUBROUTINE ANSWER, AND TO CHECK THE FEASIBILITY OF THE BEARING
C       FOR THE MAXIMUM EXPECTED AMBIENT TEMPERATURE................
C
      DIMENSION X(4), PHI(8), PSI(1), RMAX(4), RMIN(4), XSTRT(4), WORK1(
     14), WORK2(4), WORK3(4), WORK4(4), XA(4,5), XJ(4), XH(4), XS(4), XL
     2(4), XO(4), XR(4), XE(4), XC(4), STEP(4), FUN(5), STEPP(4), H(4,4)
     3, GS(4), D(4), GN(4), GA(4), Y(4), DT(4,4), C(4,4), YT(4,4), PHX(4
     4,8), PSX(4,1), PART(8), PAST(1), CH(4), UX(4)
      COMMON /111/ RPM,W,RADIUS,T1,KOPTIM,KBRG,MTHD,RADU,RADL,NMBR,KOILS
     1UP,KAPLIC,PINLET,KEY,KAIR,TINPUT,TMAX
      COMMON /KKHH/ HO,TO,JJJ,MTRL,TRQ,EPS,OILFLOW,P1,TOIL1,CHMFL,CHMFW,
     1ATITUD,X1,X2,X3,RAD
      COMMON /OPTI/ KO,NNDEX
      COMMON /BLOCK/ KLASS
      COMMON /BLOCK1/ KCHECK
      KLASS=1
      KCHECK=0
      N=NMBR
C
C        READ IN DESIGN DATA ........................................
C
      CALL DATA (N,NCONS,KB,KE)
C
C        DEFINE INPUT VARIABLES FOR OPTIMIZATION SUBROUTINE...........
C
      NN=N+1
      IPRINT=2
      IDATA=1
      NEQUS=0
      NSHOT=4
      NTEST=100
      INDEX=1
      ALPHA=1.
      BETA=0.5
      GAMA=2.
      RMAX(1)=2.*(RADU+RADL)
      RMAX(2)=0.001*(RADU+RADL)
      RMAX(3)=30.
      RMAX(4)=RADU
      RMIN(1)=0.0
      RMIN(2)=0.0
      RMIN(3)=0.0
      RMIN(4)=0.0
      XSTRT(1)=RADU+RADL
      XSTRT(2)=0.0005*(RADU+RADL)
      XSTRT(3)=10.
      XSTRT(4)=(RADU+RADL)/2.
      F=0.01
      G=0.01
      MAXM=300
      CALL FEASBL (N,RMAX,RMIN,NCONS,NEQUS,XSTRT,F,G,MAXM,IPRINT,IDATA,U
     1,X,PHI,PSI,STEPP,WORK1,WORK2,WORK3,WORK4)
      DO 1 I=1,N
      XSTRT(I)=X(I)
    1 CONTINUE
      IF (MTHD.EQ.7) GO TO 6
      IF (MTHD.EQ.1.OR.MTHD.EQ.3) GO TO 2
      F=0.1
      G=0.01
      MAXM=4000
      GO TO 3
    2 F=0.01
      G=0.01
      MAXM=4000
    3 R=1.
      IF (MTHD.EQ.5) GO TO 4
      IF (MTHD.EQ.3) GO TO 5
      CALL SEEK1 (N,RMAX,RMIN,NCONS,NEQUS,F,G,XSTRT,NSHOT,NTEST,MAXM,IPR
     1INT,IDATA,X,U,PHI,PSI,WORK1,WORK2,WORK3,WORK4)
      GO TO 7
```

```
4     REDUCE=0.0005
      CALL SIMPLEX (N,RMAX,RMIN,NCONS,NEQUS,XSTRT,NN,ALPHA,BETA,GAMA,RED
     1UCE,R,F,G,MAXM,IPRINT,IDATA,U,X,PHI,PSI,XA,XJ,FUN,XH,XS,XL,XO,XR,X
     2E,XC,STEP)
      GO TO 7
5     REDUCE=0.0004
      CALL SEEK3 (N,RMAX,RMIN,NCONS,NEQUS,XSTRT,F,G,R,REDUCE,MAXM,INDEX,
     1IPRINT,IDATA,U,X,PHI,PSI,NVIOL,WORK1,WORK2,WORK3,WORK4)
      GO TO 7
6     R=1.
      REDUCE=0.05
      F=1.E-06
      G=1.E-06
      MAXM=500
      CALL DAVID (N,RMAX,RMIN,NCONS,NEQUS,XSTRT,G,F,MAXM,IPRINT,IDATA,R,
     1REDUCE,U,X,PHI,PSI,H,GS,D,GN,GA,Y,DT,C,YT,PHX,PSX,PART,PAST,CH,UX)
7     IF (KO.NE.0). GO TO 8
      CALL ANSWER (U,X,PHI,PSI,N,NCONS,NEQUS)
C
C        CHECK BEARING FEASIBILITY FOR MAXIMUM AMBIENT TEMPERATURE....
C
      GO TO 20
8     WRITE (6,25)
C
C        CHECK THE VIOLATED CONSTRAINTS AND PRINT OUT DESIGN
C        FAILURE COMMENTS AND ADVICE ..................................
C
      I=0
      DO 17 J=1,NCONS
      IF (PHI(J).LT.0.0) GO TO 9
      GO TO 17
9     I=I+1
      IF (J.GT.8) GO TO 19
      GO TO (10,11,12,13,17,15,14,16), J
10    WRITE (6,26) I
      GO TO 17
11    WRITE (6,27) I
      GO TO 17
12    WRITE (6,34) I
      GO TO 17
13    WRITE (6,28) I
      GO TO 17
14    WRITE (6,29) I
      GO TO 17
15    WRITE (6,30) I
      GO TO 17
16    WRITE (6,31) I
17    CONTINUE
      IF (I.EQ.0) GO TO 18
      GO TO 19
18    WRITE (6,32)
      GO TO 24
19    I=I+1
      WRITE (6,33) I
      GO TO 24
20    IF (TO .LT. 70.)  GO TO 38
      KCHECK=1
      TF=TO
      ISTEP=IFIX((TMAX-T1)/2.)+1
      TMAX=TMAX+2.
      DO 22 NM=1,ISTEP
      TMAX=TMAX-2.
      TO=TF+(TMAX-T1)
      CALL UREAL (X,U)
      CALL CONST (X,NCONS,PHI)
      DO 21 M=1,NCONS
      IF (PHI(M).LT.0.0) GO TO 22
21    CONTINUE
      GO TO 23
22    CONTINUE
      WRITE (6,35) T1
      GO TO 24
23    IF (NM.EQ.1) GO TO 24
      WRITE (6,36) TMAX
      GO TO 24
```

```
38    WRITE(6,37)
24    RETURN
C
C
C
25    FORMAT (//5X,*NO OPTIMUM SOLUTION HAS BEEN FOUND FOR YOUR DESIGN V
     1ARIABLES,  NOTE THE FOLLOWING.*,/)
26    FORMAT (2X,I2,*- THE BEARING MIGHT WORK UNDER THIN FILM LUBRICATIO
     1N ,IF THE SHAFT RADIUS IS NOT ONE OF THE VARIABLES ,*,/6X,* TRY AG
     2AIN WITH THE MAXIMUM POSSIBLE RADIUS , OR DECREASE THE APPLIED LOA
     3D- IF POSSIBLE.*)
27    FORMAT (2X,I2,*- THE MINIMUM FILM THICKNESS IS LESS THAN IT IS REQ
     1UIRED FOR THIS APPLICATION,TRY CHANGING SOME VARIABLES.*)
28    FORMAT (2X,I2,*- THE NEGATIVE SIGN OF PHI(4) MEANS THAT THE RPM OF
     1 THE SHAFT IS LESS THAN THE CRITICAL SPEED AT WHICH THE*,/7X,*TURB
     2ULANCE IN THE OIL FILM MIGHT START TO OCCUR. TRY TO DECREASE THE S
     3HAFT RADIUS IF IT IS NOT ONE OF THE*,/7X,*VARIABLES OR DECREASE TH
     4E RPM AT LEAST BY THE ABSOLUTE VALUE OF PHI(4).*)
29    FORMAT (2X,I2,*- NO OIL GRADE HAS BEEN FOUND SUITABLE FOR THIS SOL
     1UTION .   IF IT IS POSSIBLE , SLIGHTLY INCREASING THE*,/7X,*SHAFT
     2REVS/MIN. OR RADIUS, IF IT IS NOT VARIABLE, OR BOTH MIGHT GIVE SOM
     3E IMPROVEMENT.*)
30    FORMAT (2X,I2,*- THE OIL OPERATING TEMPERATURE IS HIGHER THAN THE
     1ALLOWABLE FOR THIS CLASS OF BEARINGS*,/6X,*USE MOVING AIR AMBIENT,
     2 OR FORCED FEED LUBRICATED BEARING.*)
31    FORMAT (2X,I2,*- NO OIL GRADE HAS BEEN FOUND SUITABLE FOR THIS SOL
     1UTION .   IF IT IS POSSIBLE, SLIGHTLY DECREASING THE*,/7X,*SHAFT R
     2EVS/MIN. OR REDIUS, IF IT IS NOT VARIABLE, OR BOTH MIGHT GIVE SOME
     3 IMPROVEMENT.*)
32    FORMAT (//7X,*THE ABOVE SOLUTION IS NOT THE OPTIMUM ONE, HOWEVER,
     1IT IS FEASIBLE AND*,/7X,*THE USER  MAY STILL USE IT FOR HIS DESIGN
     2.*)
33    FORMAT (2X,I2,*- TRY , IN GENERAL, TO USE ANOTHER TYPE OF BEARING.
     1*)
34    FORMAT (2X,I2,*- THE NEGATIVE SIGN OF PHI(3) MEANS THAT THE BEARIN
     1G MIGHT NOT WORK HYDRODYNAMICALLY .*,/6X,*TRY CHANGING ,IF POSSIBL
     2E, SOME VARIABLES.*)
35    FORMAT (//23X,*-IMPORTANT-*,/24X,*-----------*,/5X,*THE OPTIMUM SOLU
     1TION FOUND IS ONLY FEASIBLE FOR THE DESIGN OPERATING*,/6X,*TEMPERA
     2TURE -*,F6.2,3X,*DEG. FAHRENHEIT .*,/5X,*OR REOPTIMIZE WITH A HIGH
     3ER DESIGN OPERATING TEMPERATURE .*)
36    FORMAT (//23X,*-IMPORTANT-*,/24X,*-----------*,/5X,*THE OPTIMUM SOLU
     1TION FOUND IS NOT FEASIBLE ON THE MAXIMUM EXPECTED*,/6X,*AMBIENT T
     2EMPERATURE.*,/5X,*MAXIMUM EXPECTED TEMPERATURE MUST BE REDUCED TO
     3*,F6.2,3X,*DEG.FAHRENHEIT*,/5X,*OR REOPTIMIZE WITH A HIGHER DESIGN
     4 OPERATING TEMPERATURE.*)
37    FORMAT (2X,*NOTE-*,1X,*OIL OPERATING TEMPERATURE IS BELOW 70  DEG.
     1F. WHICH LIES*,/9X,*OUTSIDE VALID  INTERNAL DATA RELATING OIL VISC
     2OSITY TO TEMPERATURE*)
      END
```

```
      SUBROUTINE BRNG2
C     **************
C
C
C        THIS SUBROUTINE IS USED MAINLY TO SELECT THE OPTIMIZATION
C        SUBROUTINE  ,  TO OUTPUT THE FINAL SOLUTION BY CALLING UPON
C        SUBROUTINE ANSWER, AND TO CHECK THE FEASIBILITY OF THE BEARING
C        FOR THE MAXIMUM EXPECTED AMBIENT TEMPERATURE.................
C
      DIMENSION X(8), PHI(11), PSI(1), RMAX(8), RMIN(8), XSTRT(8), WORK1
     1(8), WORK2(8), WORK3(8), WORK4(8), XA(8,9), XJ(8), XH(8), XS(8), X
     2L(8), XO(8), XR(8), XE(8), XC(8), STEP(8), FUN(9), H(8,8), GS(8),
     3D(8), GN(8), GA(8), Y(8), DT(8,8), C(8,8), YT(8,8), PHX(8,11), PSX
     4(8,1), PART(11), PAST(1), CH(8), UX(8), STEPP(8)
      COMMON /111/ RPM,W,RADIUS,T1,KOPTIM,KBRG,MTHD,RADU,RADL,NMBR,KOILS
     1UP,KAPLIC,PINLET,KEY,KAIR,TINPUT,TMAX
      COMMON /KKHH/ HO,TO,JJJ,MTRL,TRQ,EPS,OILFLOW,P1,TOIL1,CHMFL,CHMFW,
     1ATITUD,X1,X2,X3,RAD
      COMMON /OPTI/ KO,NNDEX
      COMMON /BLOCK/ KLASS
      COMMON /BLOCK1/ KCHECK
      KLASS=2
      KCHECK=0
      N=NMBR
      KB=KBRG
      KE=KEY
C
C        READ IN DESIGN DATA ......................................
C
      CALL DATA (N,NCONS,KB,KE)
C
C        DEFINE INPUT VARIABLES FOR OPTIMIZATION SUBROUTINE..........
C
      NN=N+1
      IPRINT=2
      IDATA=1
      NEQUS=0
      NSHOT=4
      NTEST=100
      INDEX=1
      ALPHA=1.
      BETA=0.5
      GAMA=2.
      RMAX(1)=2.*(RADU+RADL)
      RMAX(2)=0.001*(RADU+RADL)
      RMAX(3)=30.
      RMIN(1)=0.0
      RMIN(2)=0.0
      RMIN(3)=0.0
      XSTRT(1)=RADU+RADL
      XSTRT(2)=0.0005*(RADU+RADL)
      XSTRT(3)=15.
      IF (KBRG.EQ.2.OR.KBRG.EQ.3) GO TO 5
      IF (N.EQ.7) GO TO 1
      IF (N.EQ.8) GO TO 2
      RMAX(4)=0.05*RMAX(1)
      RMAX(5)=0.1
      RMAX(6)=5.*T1+2.
      RMIN(4)=0.0
      RMIN(5)=0.0
      RMIN(6)=T1+1.
      XSTRT(4)=0.025*XSTRT(1)
      XSTRT(5)=0.05
      XSTRT(6)=2.*T1+1.
      GO TO 10
1     GO TO (3,4), KEY
2     RMAX(4)=RADU
      RMAX(5)=60.0
      RMAX(6)=0.05*RMAX(1)
      RMAX(7)=0.1
      RMAX(8)=5.*T1+2.
      RMIN(4)=RADL
```

```
      RMIN(5)=0.0
      RMIN(6)=0.0
      RMIN(7)=0.0
      RMIN(8)=T1+1.
      XSTRT(4)=(RADU+RADL)/2.
      XSTRT(5)=10.
      XSTRT(6)=0.025*XSTRT(1)
      XSTRT(7)=0.05
      XSTRT(8)=2.*T1+1.
      GO TO 10
3     RMAX(4)=60.0
      RMAX(5)=0.05*RMAX(1)
      RMAX(6)=0.1
      RMAX(7)=5.*T1+2.
      RMIN(4)=0.0
      RMIN(5)=0.0
      RMIN(6)=0.0
      RMIN(7)=T1+1.
      XSTRT(4)=10.
      XSTRT(5)=0.025*XSTRT(1)
      XSTRT(6)=0.05
      XSTRT(7)=2.*T1+1.
      GO TO 10
4     RMAX(4)=RADU
      RMAX(5)=0.05*RMAX(1)
      RMAX(6)=0.1
      RMAX(7)=5.*T1+2.
      RMIN(4)=RADL
      RMIN(5)=0.0
      RMIN(6)=0.0
      RMIN(7)=T1+1.
      XSTRT(4)=(RADU+RADL)/2.
      XSTRT(5)=0.025*XSTRT(1)
      XSTRT(6)=0.05
      XSTRT(7)=2.*T1+1.
      GO TO 10
5     IF (N.EQ.5) GO TO 6
      IF (N.EQ.6) GO TO 7
      RMAX(4)=5.*T1+2.
      RMIN(4)=T1+1.
      XSTRT(4)=2.*T1+1.
      GO TO 10
6     GO TO (8,9), KEY
7     RMAX(4)=RADU
      RMAX(5)=60.
      RMAX(6)=5.*T1+2.
      RMIN(4)=RADL
      RMIN(5)=0.0
      RMIN(6)=T1+1.
      XSTRT(4)=(RADU+RADL)/2.
      XSTRT(5)=10.
      XSTRT(6)=2.*T1+1.
      GO TO 10
8     RMAX(4)=60.
      RMAX(5)=5.*T1+2.
      RMIN(4)=0.0
      RMIN(5)=T1+1.
      XSTRT(4)=10.
      XSTRT(5)=2.*T1 +1.
      GO TO 10
9     RMAX(4)=RADU
      RMAX(5)=5.*T1+2.
      RMIN(4)=RADL
      RMIN(5)=T1+1.
      XSTRT(4)=(RADU+RADL)/2.
      XSTRT(5)=2.*T1 +1.
10    F=0.01
      G=0.01
      MAXM=300
      CALL FEASBL (N,RMAX,RMIN,NCONS,NEQUS,XSTRT,F,G,MAXM,IPRINT,IDATA,U
     1,X,PHI,PSI,STEPP,WORK1,WORK2,WORK3,WORK4)
      DO 11 I=1,N
      XSTRT(I)=X(I)
11    CONTINUE
```

```
      IF (MTHD.EQ.7) GO TO 16
      IF (MTHD.EQ.1.OR.MTHD.EQ.3) GO TO 12
      F=0.1
      G=0.01
      MAXM=4000
      GO TO 13
12    F=0.01
      G=0.01
      MAXM=4000
13    R=1.
      IF (MTHD.EQ.5) GO TO 14
      IF (MTHD.EQ.3) GO TO 15
      CALL SEEK1 (N,RMAX,RMIN,NCONS,NEQUS,F,G,XSTRT,NSHOT,NTEST,MAXM,IPR
     1INT,IDATA,X,U,PHI,PSI,WORK1,WORK2,WORK3,WORK4)
      GO TO 17
14    REDUCE=0.0005
      CALL SIMPLEX (N,RMAX,RMIN,NCONS,NEQUS,XSTRT,NN,ALPHA,BETA,GAMA,RED
     1UCE,R,F,G,MAXM,IPRINT,IDATA,U,X,PHI,PSI,XA,XJ,FUN,XH,XS,XL,XO,XR,X
     2E,XC,STEP)
      GO TO 17
15    REDUCE=0.0004
      CALL SEEK3 (N,RMAX,RMIN,NCONS,NEQUS,XSTRT,F,G,R,REDUCE,MAXM,INDEX,
     1IPRINT,IDATA,U,X,PHI,PSI,NVIOL,WORK1,WORK2,WORK3,WORK4)
      GO TO 17
16    R=1.
      REDUCE=0.05
      F=1.E-06
      G=1.E-06
      MAXM=500
      CALL DAVID (N,RMAX,RMIN,NCONS,NEQUS,XSTRT,G,F,MAXM,IPRINT,IDATA,R,
     1REDUCE,U,X,PHI,PSI,H,GS,D,GN,GA,Y,DT,C,YT,PHX,PSX,PART,PAST,CH,UX)
17    IF (KO.NE.0) GO TO 18
      CALL ANSWER (U,X,PHI,PSI,N,NCONS,NEQUS)
C
C        CHECK BEARING FEASIBILITY FOR MAXIMUM AMBIENT TEMPERATURE....
C
      GO TO 29
18    WRITE (6,34)
C
C        CHECK THE VIOLATED CONSTRAINTS AND PRINT OUT DESIGN
C        FAILURE COMMENTS AND ADVICE ..................................
C
      I=0
      DO 26 J=1,NCONS
      IF (PHI(J).LT.0.0) GO TO 19
      GO TO 26
19    I=I+1
      IF (J.GT.8) GO TO 28
      GO TO (20,21,22,23,26,26,24,25), J
20    WRITE (6,35) I
      GO TO 26
21    WRITE (6,36) I
      GO TO 26
22    WRITE (6,42) I
      GO TO 26
23    WRITE (6,37) I
      GO TO 26
24    WRITE (6,38) I
      GO TO 26
25    WRITE (6,39) I
26    CONTINUE
      IF (I.EQ.0) GO TO 27
      GO TO 28
27    WRITE (6,40)
      GO TO 33
28    I=I+1
      WRITE (6,41) I
      GO TO 33
29    IF (TO .LT. 70.) GO TO 46
      KCHECK=1
      TF=TO
      ISTEP=IFIX((TMAX-T1)/2.)+1
      TMAX=TMAX+2.
      DO 31 NM=1,ISTEP
```

```
      TMAX=TMAX-2.
      TO=TF+(TMAX-T1)
      CALL UREAL (X,U)
      CALL CONST (X,NCONS,PHI)
      DO 30 M=1,NCONS
      IF (PHI(M).LT.0.0) GO TO 31
30    CONTINUE
      GO TO 32
31    CONTINUE
      WRITE (6,43) T1
      GO TO 33
32    IF (NM.EQ.1) GO TO 33
      WRITE (6,44) TMAX
      GO TO 33
46    WRITE(6,45)
33    RETURN
C
C
C
34    FORMAT (//5X,*NO OPTIMUM SOLUTION HAS BEEN FOUND FOR YOUR DESIGN V
     1ARIABLES,  NOTE THE FOLLOWING.*,/)
35    FORMAT (2X,I2,*- THE BEARING MIGHT WORK UNDER THIN FILM LUBRICATIO
     1N ,IF THE SHAFT RADIUS IS NOT ONE OF THE VARIABLES ,*,/6X,* TRY AG
     2AIN WITH THE MAXIMUM POSSIBLE RADIUS , OR DECREASE THE APPLIED LOA
     3D- IF POSSIBLE.*)
36    FORMAT (2X,I2,*- THE MINIMUM FILM THICKNESS IS LESS THAN IT IS REQ
     1UIRED FOR THIS APPLICATION,TRY CHANGING SOME VARIABLES.*)
37    FORMAT (2X,I2,*- THE NEGATIVE SIGN OF PHI(4) MEANS THAT THE RPM OF
     1 THE SHAFT IS LESS THAN THE CRITICAL SPEED AT WHICH THE*,/7X,*TURB
     2ULANCE IN THE OIL FILM MIGHT START TO OCCUR. TRY TO DECREASE THE S
     3HAFT RADIUS IF IT IS NOT ONE OF THE*,/7X,*VARIABLES OR DECREASE TH
     4E RPM AT LEAST BY THE ABSOLUTE VALUE OF PHI(4).*)
38    FORMAT (2X,I2,*- NO OIL GRADE HAS BEEN FOUND SUITABLE FOR THIS SOL
     1UTION .   IF IT IS POSSIBLE , SLIGHTLY INCREASING THE*,/7X,*SHAFT
     2REVS/MIN. OR RADIUS, IF IT IS NOT VARIABLE, OR BOTH MIGHT GIVE SOM
     3E IMPROVEMENT.*)
39    FORMAT (2X,I2,*- NO OIL GRADE HAS BEEN FOUND SUITABLE FOR THIS SOL
     1UTION .   IF IT IS POSSIBLE, SLIGHTLY DECREASING THE*,/7X,*SHAFT R
     2EVS/MIN. OR REDIUS, IF IT IS NOT VARIABLE, OR BOTH MIGHT GIVE SOME
     3 IMPROVEMENT.*)
40    FORMAT (//7X,*THE ABOVE SOLUTION IS NOT THE OPTIMUM ONE, HOWEVER,
     1IT IS FEASIBLE AND*,/7X,*THE USER  MAY STILL USE IT FOR HIS DESIGN
     2.*)
41    FORMAT (2X,I2,*- TRY , IN GENERAL, TO USE ANOTHER TYPE OF BEARING.
     1*)
42    FORMAT (2X,I2,*- THE NEGATIVE SIGN OF PHI(3) MEANS THAT THE BEARIN
     1G MIGHT NOT WORK HYDRODYNAMICALLY .*,/6X,*TRY CHANGING ,IF POSSIBL
     2E, SOME VARIABLES.*)
43    FORMAT (//23X,*-IMPORTANT-*,/24X,*---------*,/5X,*THE OPTIMUM SOLU
     1TION FOUND IS ONLY FEASIBLE FOR THE DESIGN OPERATING*,/6X,*TEMPERA
     2TURE -*,F6.2,3X,*DEG. FAHRENHEIT .*,/5X,*OR REOPTIMIZE WITH A HIGH
     3ER DESIGN OPERATING TEMPERATURE .*)
44    FORMAT (//23X,*-IMPORTANT-*,/24X,*---------*,/5X,*THE OPTIMUM SOLU
     1TION FOUND IS NOT FEASIBLE ON THE MAXIMUM EXPECTED*,/6X,*AMBIENT T
     2EMPERATURE.*,/5X,*MAXIMUM EXPECTED TEMPERATURE MUST BE REDUCED TO
     3*,F6.2,3X,*DEG.FAHRENHEIT*,/5X,*OR REOPTIMIZE WITH A HIGHER DESIGN
     4 OPERATING TEMPERATURE.*)
45    FORMAT (2X,*NOTE-*,1X,*OIL OPERATING TEMPERATURE IS BELOW 70  DEG.
     1F. WHICH LIES*,/9X,*OUTSIDE VALID  INTERNAL DATA RELATING OIL VISC
     2OSITY TO TEMPERATURE*)
      END
```

```
      SUBROUTINE CONST (X,NCONS,PHI)
C     ****************************
C
      DIMENSION X(1), PHI(1)
      COMMON /KH/ CNST(11)
      DO 1 J=1,NCONS
      PHI(J)=CNST(J)
1     CONTINUE
      RETURN
      END
```

```
C     SUBROUTINE UREAL (X,U)
C     ********************
C
C        THE MAIN BEARING DESIGN PROCEDURE, ACCORDING TO THE CLASS OF
C       BEARING REQUIRED, IS INCORPORATED IN SUBROUTINE UREAL .......
C
C               ** DEFINITIONS **
C               *****************
C
C     ***********************************************************************
C     * FF         UNIT LOAD/SQ.IN. OF PROJECTED AREA                        *
C     * PP         CAPACITY NUMBER, MIN./SEC.                                *
C     * SNO        SOMMERFELD NUMBER.                                        *
C     * LRATIO     MODIFIED LENGTH TO DIAMETER RATIO                         *
C     * ECM        ECCENTRICITY MODULUS .                                    *
C     * EM         ECCENTRICITY MODULUS ARRAY.                               *
C     * CN         CAPACITY NUMBER ARRAY.                                    *
C     * CNST(I)    INEQUALITY CONSTRAINT NO. I                               *
C     * EPS        ECCENTRICITY RATIO .                                      *
C     * FACTOR     OIL VOLUME FACTOR.                                        *
C     * ECCR       ECCENTRICITY RATIO ARRAY .                                *
C     * FKE        OIL VOLUME FACTOR ARRAY .                                 *
C     * HO         MINIMUM FILM THICKNESS, INCH.                             *
C     * TT         TORQUE NUMBER,MIN./SEC.                                   *
C     * TRQ        FRICTIONAL TORQUE, LB.INCH .                              *
C     * BTU        ENERGY LOSS, BTU/HR. )                                    *
C     * QF         OIL-FLOW THROUGH THE BEARING OIL-FILM, GPM.               *
C     * PINLET     OIL INLET PRESSURE, PSI.                                  *
C     * QCO        UNCORRECTED CHAMFER FLOW, GPM.                            *
C     * XI         KINETIC-ENERGY CORRECTION FACTOR FOR CHAMFER FLOW.        *
C     * QC         CHAMFER FLOW, GPM.                                        *
C     * QH         HYDRODYNAMIC PRESSURES FLOW, GPM.                         *
C     * CHMFL      AXIAL LENGTH OF CHAMFER, INCH.                            *
C     * CHMFW      CHAMFER DIMENSION, INCH.                                  *
C     * TOTALQ     TOTAL OIL FLOW THROUGH THE BEARING, GPM.                  *
C     * DELTAT     OIL TEMPRATURE RISE, DEG. F.                              *
C     * TOIL1      OIL INLET TEMPERATURE, DEG. F.                            *
C     * Q1         OIL-FLOW COEFFICIENT, FUNCTION OF EPS.                    *
C     * EPSLON     ARRAY FOR ECCENTRICITY RATIO.                             *
C     * QU         ARRAY FOR THE HYDRODYNAMIC OIL-FLOW COEFFICIENT.          *
C     * CNO        A CAPACITY NUMBER.                                        *
C     * Q2         OIL-FLOW COEFFICIENT, FUNCTION OF CNO.                    *
C     * TINPUT     INPUT TORQUE.                                             *
C     * DISSF      HEAT DISSIPATION COEFFICIENT, BTU/(HR.*SQ.FT.*DEG.F.)     *
C     * AREA       OUTSIDE AREA OF THE BEARING HOUSING,SQ.FT.   .            *
C     * DELW       TEMPERATURE RISE OF HOUSING SURFACE ABOVE AMBIENT         *
C     *            AIR TEMPERATURE, DEG.F. .                                 *
C     * DELO       TEMPERATURE RISE OF HOUSING SURFACE ABOVE AMBIENT         *
C     *            AIR TEMPERATURE, DEG.F.   .                               *
C     * TO         OPERATING OIL TEMPERATURE, DEG.F.   .                     *
C     * Z          OIL OPERATING VISCOSITY ARRAY (VARIABLE).                 *
C     * TEMP       OIL OPERATING TEMPERATURE ARRAY (FUNCTION).               *
C     * TTEMP      OIL OPERATING TEMPERATURE ARRAY (VARIABLE).               *
C     * ZZ         OIL OPERATING VISCOSITY ARRAY (FUNCTION).                 *
C     * RO         SPECIFIC GRAVITY .                                        *
C     * DELTA      MASS DENSITY AT TEMPERATURE TO, SLUG/CU.INCH.   .         *
C     * VISC       KINEMATIC VISCOSITY, SQ.INCH/SEC.   .                     *
C     * CRPM       CRITICAL SPEED, REVS/MIN.   .                             *
C     * OILFLOW    ENTRANCE OIL VOLUME, CU.INCH/SEC.   .                     *
C     * RATIO      RATIO BETWEEN THE DEVELOPED BEARING LENGTH IN THE         *
C     *            DIRECTION OF MOTION TO THE AXIAL WIDTH.                   *
C     * POP1       INTERPOLATED VALUE OF THE RATIO-PMAX/F-FOR CURVE NO. 1*
C     * POP2       INTERPOLATED VALUE OF THE RATIO-PMAX/F-FOR CURVE NO. 2*
C     * ELOL       RATIO ARRAY .                                             *
C     * POP        ARRAY FOR PMAX/FF .                                       *
C     * PRATIO     INTERPOLATED VALUE OF-PMAX/F-CORRESPONDS TO RATIO         *
C     *            AT THE VALUE OF  EPS.                                     *
C     * C          ARRAY FOR ECCENTRICITY RATIO.                             *
C     * EPSI       ARRAY FOR ATTITUDE ANGLE .                                *
C     * ATITUD     INTERPOLATED VALUE OF ATTITUDE ANGLE.                     *
C     * PMAX       MAX. PRESSURE IN OIL FILM, LB./SQ.INCH   .                *
C     * KCHECK     FLAG NAME, =1, WHEN UREAL IS CALLED FOR FEASIBILITY       *
C     *            CHECK..OTHERWISE IT IS EQUAL TO  ZERO.                    *
C     * U          CRITERION FUNCTION  .                                     *
C     ***********************************************************************
C
```

```
C
      DIMENSION X(1), CN(20,13), EM(20,13), TORQ(20,20), EMM(20,20), Z(1
     14,23), TEMP(14,23), T(14), ECCR(20,7), FKE(20,7), ELOL(11,9), POP(
     211,9), EPSLON(20,9), QU(20,9), C(20,6), EPSI(20,6), TTEMP(7,16), Z
     3Z(7,16)
      COMMON /AAA/ CN,EM,EMM,TORQ,Z,TEMP,ECCR,FKE,ELOL,POP,EPSLON,QU,C,E
     1PSI,TTEMP,ZZ
      COMMON /KKHH/ HO,TO,JJJ,MTRL,TRQ,EPS,OILFLOW,P1,TOIL1,CHMFL,CHMFW,
     1ATITUD,X1,X2,X3,RAD
      COMMON /KH/ CNST(11)
      COMMON /111/ RPM,W,RADIUS,T1,KOPTIM,KBRG,MTHD,RADU,RADL,NMBR,KOILS
     1UP,KAPLIC,PINLET,KEY,KAIR,TINPUT,TMAX
      COMMON /BLOCK1/ KCHECK
      COMMON /BLOCK/ KLASS
C
C
      IF (KCHECK.EQ.1) GO TO 76
      GO TO (13,1), KLASS
1     IF (KBRG.EQ.2.OR.KBRG.EQ.3) GO TO 7
C
C        SETTING THE DESIGN VARIABLES ..................................
C
      IF (NMBR.EQ.7) GO TO 2
      IF (NMBR.EQ.8) GO TO 3
      RAD=RADIUS
      P1=PINLET
      CHMFL=ABS(X(4))
      CHMFW=ABS(X(5))
      TOIL1=(T1+25.)+(2.*T1-25.)*(SIN(X(6)))**2
      GO TO 15
2     GO TO (4,5), KEY
3     RAD=RADL+(RADU-RADL)*(SIN(X(4)))**2
      P1=ABS(X(5))
      CHMFL=ABS(X(6))
      CHMFW=ABS(X(7))
      TOIL1=(T1+25.)+(2.*T1-25.)*(SIN(X(8)))**2
      GO TO 15
4     RAD=RADIUS
      P1=ABS(X(4))
      GO TO 6
5     P1=PINLET
      RAD=RADL+(RADU-RADL)*(SIN(X(4)))**2
6     CHMFL=ABS(X(5))
      CHMFW=ABS(X(6))
      TOIL1=(T1+25.)+(2.*T1-25.)*(SIN(X(7)))**2
      GO TO 15
7     IF (NMBR.EQ.5) GO TO 8
      IF (NMBR.EQ.6) GO TO 9
      RAD=RADIUS
      P1=PINLET
      TOIL1=(T1+25.)+(2.*T1-25.)*(SIN(X(4)))**2
      GO TO 15
8     GO TO (10,11), KEY
9     RAD=RADL+(RADU-RADL)*(SIN(X(4)))**2
      P1=ABS(X(5))
      TOIL1=(T1+25.)+(2.*T1-25.)*(SIN(X(6)))**2
      GO TO 15
10    RAD=RADIUS
      P1=ABS(X(4))
      GO TO 12
11    P1=PINLET
      RAD=RADL+(RADU-RADL)*(SIN(X(4)))**2
12    TOIL1=(T1+25.)+(2.*T1-25.)*(SIN(X(5)))**2
      GO TO 15
13    IF (NMBR.EQ.4) GO TO 14
      RAD=RADIUS
      GO TO 15
14    RAD=RADL+(RADU-RADL)*(SIN(X(4)))**2
15    CONTINUE
      GO TO (16,17,18,19), KAPLIC
16    X1=1.6*RAD+2.*RAD*(SIN(X(1)))**2
      GO TO 20
17    X1=1.6*RAD+2.4*RAD*(SIN(X(1)))**2
      GO TO 20
```

```
18    X1=2.*RAD+2.*RAD*(SIN(X(1)))**2
      GO TO 20
19    X1=3.*RAD+RAD*(SIN(X(1)))**2
20    X2=ABS(X(2))
      X3=56.*(SIN(X(3)))**2
      FF=W/(X1*2.*RAD)
      PP=6.9E+06*FF*(X2/RAD)**2/(X3*RPM)
      SNO=1./(PP*60.)
C
      LRATIO=X1/(2.*RAD)*1000.
      II=0
C
C          * SELECT THE DESIGN CURVE FOR THE FRICTIONAL TORQUE LOSS *
C
      DO 21 I=100,280,20
      II=II+1
      IF (LRATIO.GE.I.AND.LRATIO.LE.(I+20)) GO TO 22
21    CONTINUE
      GO TO 24
22    CONTINUE
      IF ((LRATIO-I).LE.(I+20-LRATIO)) GO TO 23
      LL=II+1
      ECM=FTABLE1(CN,EM,PP,13,LL)
      GO TO 42
23    ECM=FTABLE1(CN,EM,PP,13,II)
      LL=II
      GO TO 42
24    DO 25 I=300,350,50
      II=II+1
      IF (LRATIO.GE.I.AND.LRATIO.LE.(I+50)) GO TO 26
25    CONTINUE
      GO TO 28
26    CONTINUE
      IF ((LRATIO-I).LE.(I+50-LRATIO)) GO TO 27
      LL=II+1
      ECM=FTABLE1(CN,EM,PP,13,LL)
      GO TO 42
27    ECM=FTABLE1(CN,EM,PP,13,II)
      LL=II
      GO TO 42
28    DO 29 I=400,500,100
      II=II+1
      IF (LRATIO.GE.I.AND.LRATIO.LE.(I+100)) GO TO 30
29    CONTINUE
      GO TO 32
30    CONTINUE
      IF ((LRATIO-I).LE.(I+100-LRATIO)) GO TO 31
      LL=II+1
      ECM=FTABLE1(CN,EM,PP,13,LL)
      GO TO 42
31    ECM=FTABLE1(CN,EM,PP,13,II)
      LL=II
      GO TO 42
32    DO 33 I=600,800,200
      II=II+1
      IF (LRATIO.GE.I.AND.LRATIO.LE.(I+200)) GO TO 34
33    CONTINUE
      GO TO 36
34    CONTINUE
      IF ((LRATIO-I).LE.(I+200-LRATIO)) GO TO 35
      LL=II+1
      ECM=FTABLE1(CN,EM,PP,13,LL)
      GO TO 42
35    ECM=FTABLE1(CN,EM,PP,13,II)
      LL=II
      GO TO 42
36    DO 37 I=1000,1500,500
      II=II+1
      IF (LRATIO.GE.I.AND.LRATIO.LE.(I+500)) GO TO 38
37    CONTINUE
      GO TO 40
38    CONTINUE
      IF ((LRATIO-I).LE.(I+500-LRATIO)) GO TO 39
      LL=II+1
```

```
      ECM=FTABLE1(CN,EM,PP,13,LL)
      GO TO 42
39    ECM=FTABLE1(CN,EM,PP,13,II)
      LL=II
      GO TO 42
40    II=II+1
      IF ((LRATIO-2000).LE.(4000-LRATIO)) GO TO 41
      LL=II+1
      ECM=FTABLE1(CN,EM,PP,13,LL)
      GO TO 42
41    ECM=FTABLE1(CN,EM,PP,13,II)
      LL=II
42    EPS=1.0-1.0/ECM
      FACTOR=FTABLE1(ECCR,FKE,EPS,7,LL)
      ATITUD=FTABLE1(C,EPSI,EPS,6,LL)
C
C                 * TORQUE DETERMINATION AND HEAT BALANCE *
C        THE HEAT BALANCE IN THIS CALCULATIONS IS BASED UPON THE ASSUMP-
C        TION THAT ALL HEAT GENERATED IN THE OIL-FILM IS CARRIED AWAY BY
C        THE OIL FLOWING THROUGH THE BEARING (TOTALQ). ................
C
      HO=X2*(1.-EPS)
      TT=FTABLE1(EMM,TORQ,ECM,20,LL)
      TRQ=TT*X3*RAD**3*X1*RPM/(6.9E+06*X2)
      BTU=0.040380497*TRQ*RPM
C
      GO TO (50,43), KLASS
C
C        ESTIMATING OIL-FLOW THROUGH THE BEARING ACCORDING TO ITS TYPE.
C
43    GO TO (44,47,48), KBRG
C
C
C                       * PRESSURIZED JOURNAL BEARINGS *
C
C                  * FLOW THROUGH THE BEARING OIL-FILM (QF) *
C
44    QF=4.6E+06*X2**3*P1/X3
      COND=(CHMFW+X2)**4
      QCO=47200.*COND*P1/(CHMFL*X3)
      IF (QCO.LE.0.0) GO TO 45
      A=QCO*40./(CHMFL*X3)
      XI=(-1.+SQRT(1.+4.*A*CHMFL))/(2.*A)
C
C                       * FLOW THROUGH CHAMFERS (QC) *
C
      QC=XI*QCO
      GO TO 46
45    QC=0.0
46    Q1=FTABLE1(EPSLON,QU,EPS,9,LL)
C
C                 * FLOW DUE TO HYDRODYNAMIC PRESSURES (QH) *
C
      QH=RPM*RAD*X1*X2*Q1/73.5
C
C                            * TOTAL OIL-FLOW *
C
      TOTALQ=QF+4.*QC+QH
      GO TO 49
47    QF=3.76E+06*RAD*(X2)**3*P1*(1.+1.5*EPS**2)/(X1*X3)
      CNO=SNO*(X1/(2.*RAD))**2
      Q2=2.-10.924999*CNO+30.604166*CNO**2-39.999999*CNO**3+19.270833*CN
     1O**4
      QH=RPM*RAD*X1*X2*Q2/73.5
      TOTALQ=QF+QH
      GO TO 49
48    QF=5.97E+05*(X2)**3*P1/X3*(1.+1.5*EPS**2)*ATAN(2.*3.1415927*RAD/X1
     1)
      TOTALQ=2.*QF
49    DELTAT=BTU/(3.47*TOTALQ*60.)
      TO=DELTAT+TOIL1
      GO TO 61
C
C
```

```
C                * NON PRESSURIZED JOURNAL BEARINGS *
C
50    GO TO (51,52), KAIR
51    DISSF=2.1
      GO TO 53
52    DISSF=5.9
53    AREA=15.*(X1*2.*RAD)/144.
      DELW=BTU/(DISSF*AREA)
C
C          DELO IS THE TEMP. RISE OF FILM ABOVE WALL, IT IS A FUNCTION OF
C          DELW, FOR EACH KOILSUP AND KAIR  .............................
C
      GO TO (54,56,56,58), KOILSUP
54    IF (KAIR.EQ.1) GO TO 55
      DELO=20.+78.*(DELW-11.)/69.
      GO TO 60
55    DELO=20.+6.*(DELW-20.)/7.
      GO TO 60
56    IF (KAIR.EQ.1) GO TO 57
      DELO=45.+4.*(DELW-10.)
      GO TO 60
57    DELO=25.+1.3*(DELW-10.)
      GO TO 60
58    IF (KAIR.EQ.1) GO TO 59
      DELO=20.+2.*(DELW-30.)/3.
      GO TO 60
59    DELO=10.+0.1*(DELW-30.)
60    DELTAT=DELW+DELO
      TO=DELTAT+T1
C
C                          * SELECTING OIL GRADE *
C
61    J=1
      M=1
62    MM=M+1
      DO 63 L=M,MM
      T(L)=FTABLE2(Z,TEMP,X3,23,L)
63    CONTINUE
      IF (TO.LT.T(1)) GO TO 71
64    IF (TO.GE.T(J).AND.TO.LE.T(J+1)) GO TO 65
      M=MM+1
      J=J+1
      IF (L.LT.14) GO TO 62
      IF (J.GT.13) GO TO 72
      GO TO 64
65    CONTINUE
      IF ((TO-T(J)).LT.(T(J+1)-TO)) GO TO 66
      TO=T(J+1)
      J=J+1
      GO TO 67
66    TO=T(J)
67    IF (J.GE.3) GO TO 70
      GO TO (68,69), J
68    JJJ=1
      GO TO 73
69    JJJ=5
      GO TO 73
70    JJJ=(J-2)*10
      GO TO 73
71    JJJ=0
      GO TO 73
72    JJJ=130
C
73    RO=0.875-0.00035*(TO-60.)
      DELTA=9.37E-05*RO
      VISC=6.9E-06*X3/DELTA
      CRPM=392.4*VISC*SQRT(RAD/X2)/(RAD*X2)
      OILFLOW=0.0272*FACTOR*RAD*RPM*X1*X2
C
C
C                     * SELECTING BEARING MATERIAL *
C         THE SELECTION OF THE MATERIAL IS BUILT UPON THE MAX. PRESSURE
C         IN THE FILM (PMAX), WHICH IS EQUAL TO ..FF*(PMAX/FF). THE RATI
C         PMAX/FF IS A FUNCTION OF (ELOL), WHICH IS THE RATIO OF THE
```

```
C           DEVELOPED BEARING LENGTH IN THE DIRECTION OF MOTION TO THE
C           AXIAL WIDTH FOR EACH LRATIO ..................................
C
C
      RATIO=2.094395*RAD/X1
      EC=0.0
      DO 74 KK=1,10
      IF (EPS.GT.EC.AND.EPS.LE.(EC+0.1)) GO TO 75
      EC=EC+0.1
74    CONTINUE
75    POP1=FTABLE3(ELOL,POP,RATIO,9,KK)
      POP2=FTABLE3(ELOL,POP,RATIO,9,(KK+1))
      PRATIO=10.*(EPS-EC)*(POP2-POP1)+POP1
      PMAX=PRATIO*FF
C
      IF (PMAX.LE.800.) MTRL=1
      IF (PMAX.GT.800..AND.PMAX.LE.1200.) MTRL=2
      IF (PMAX.GT.1200..AND.PMAX.LE.1500.) MTRL=3
      IF (PMAX.GT.1500..AND.PMAX.LE.3000.) MTRL=4
      IF (PMAX.GT.3000..AND.PMAX.LE.4000.) MTRL=5
      IF (PMAX.GT.4000..AND.PMAX.LT.10000.) MTRL=6
      IF (PMAX.GE.10000.) MTRL=7
      GO TO 77
C
C
C           THE FOLLOWING IS A CHECK OF THE BEARING FEASIBILITY FOR
C           MAXIMUM EXPECTED AMBIENT TEMPERATURE....................
C
76    X3=FTABLE4(TTEMP,ZZ,TO,16,J)
      FF=W/(2.*X1*RAD)
      PP=6.9E+06*FF*(X2/RAD)**2/(X3*RPM)
      SNO=1./(PP*60.)
      ECM=FTABLE1(CN,EM,PP,13,LL)
      EPS=1.-1./ECM
      HO=X2*(1.-EPS)
      RO=0.875-0.00035*(TO-60.)
      DELTA=9.37E-05*RO
      VISC=6.9E-06*X3/DELTA
      CRPM=392.4*VISC*SQRT(RAD/X2)/(RAD*X2)
C
C
C                       * INEQUALITY CONSTRAINTS *
C
C
77    GO TO (78,79,80,80,80,81,82), MTRL
78    CNST(1)=X3*RPM/FF-10.
      CNST(2)=10.E+06*(HO-0.00075)
      CNST(3)=10.E+03*(SNO-0.025)
      GO TO 83
79    CNST(1)=X3*RPM/FF-20.
      CNST(2)=10.E+06*(HO-0.00075)
      CNST(3)=10.E+03*(SNO-0.050)
      GO TO 83
80    CNST(1)=X3*RPM/FF-4.
      CNST(2)=10.E+06*(HO-0.000375)
      CNST(3)=10.E+03*(SNO-0.009)
      GO TO 83
81    CNST(1)=X3*RPM/FF-2.
      CNST(2)=10.E+06*(HO-0.000375)
      CNST(3)=10.E+03*(SNO-0.005)
      GO TO 83
82    CNST(1)=X3*RPM/FF-2.
      CNST(2)=10.E+06*(HO-0.0001)
      CNST(3)=10.E+03*(SNO-0.005)
83    CNST(4)=CRPM-RPM
      CNST(5)=10.E+04*(0.004*RAD-X2)
      GO TO (91,84), KLASS
84    CNST(6)=250.-TO
      CNST(7)=10.E+08*FLOAT(70-JJJ)
      CNST(8)=FLOAT(JJJ-10)
      IF (KBRG.EQ.2.OR.KBRG.EQ.3) GO TO 89
      IF (NMBR.EQ.6) GO TO 85
      IF (NMBR.EQ.8) GO TO 86
      GO TO (87,88), KEY
85    CNST(9)=10.E+02*(X1/3.-CHMFL)
```

```
      CNST(10)=10.E+03*(0.1-CHMFW)
      GO TO 92
86    CNST(9)=10.E+02*(X1/3.-CHMFL)
      CNST(10)=10.E+03*(0.1-CHMFW)
      CNST(11)=100.-P1
      GO TO 92
87    CNST(9)=10.E+02*(X1/3.-CHMFL)
      CNST(10)=10.E+03*(0.1-CHMFW)
      CNST(11)=100.-P1
      GO TO 92
88    CNST(9)=10.E+02*(X1/3.-CHMFL)
      CNST(10)=10.E+03*(0.1-CHMFW)
      GO TO 92
89    IF (NMBR.EQ.4) GO TO 92
      IF (NMBR.EQ.6) GO TO 90
      GO TO (90,92), KEY
90    CNST(9)=100.-P1
      GO TO 92
91    CNST(6)=10.E+06*(180.-TO)
      CNST(7)=FLOAT(70-JJJ)
      CNST(8)=10.E+08*FLOAT(JJJ-10)
C
      IF (KCHECK.EQ.1) RETURN
C
C                           * OPTIMIZATION CRITERION *
C
C
92    GO TO (93,94,95), KOPTIM
93    U=TRQ
      GO TO 96
94    U=DELTAT
      GO TO 96
95    TRATIO=TRQ/TINPUT*100.
      U1=-0.15333*TRATIO+0.06867*TRATIO**2-0.006867*TRATIO**3+.000253*TR
     1ATIO**4
      IF (TRATIO.GT.20.) U1=10.E+30
      U2=-0.025333*DELTAT+0.0023733*DELTAT**2-.000047467*DELTAT**3+3.63E
     1-07*DELTAT**4
      IF (DELTAT.GT.100.) U2=10.E+30
      U=U1+U2
96    RETURN
      END
```

```
      SUBROUTINE DATA (N,NCONS,KB,KE)
C     ******************************
C
C
C         THIS SUBROUTINE IS USED TO READ IN THE DESIGN DATA AND TO
C        DEFINE THE NUMBER OF INEQUALITY CONSTRAINTS ACCORDING
C        TO THE TYPE OF BEARING SELECTED ..........................
C
      DIMENSION CN(20,13), EM(20,13), EMM(20,20), TORQ(20,20), Z(14,23),
     1 TEMP(14,23), ECCR(20,7), FKE(20,7), ELOL(11,9), POP(11,9), EPSLON
     2(20,9), QU(20,9), C(20,6), EPSI(20,6), TTEMP(7,16), ZZ(7,16)
      COMMON /AAA/ CN,EM,EMM,TORQ,Z,TEMP,ECCR,FKE,ELOL,POP,EPSLON,QU,C,E
     1PSI,TTEMP,ZZ
      COMMON /BLOCK/ KLASS
      READ (5,14) ((CN(I,J),J=1,13),I=1,20)
      READ (5,15) ((EM(I,J),J=1,13),I=1,20)
      READ (5,16) (EMM(1,J),J=1,20)
      READ (5,17) ((TORQ(I,J),J=1,20),I=1,20)
      READ (5,18) (Z(1,J),J=1,23)
      READ (5,19) ((TEMP(I,J),J=1,23),I=1,14)
      READ (5,20) (ECCR(1,J),J=1,7)
      READ (5,21) ((FKE(I,J),J=1,7),I=1,20)
      READ (5,22) (ELOL(1,J),J=1,9)
      READ (5,23) ((POP(I,J),J=1,9),I=1,11)
      READ (5,26) (C(1,J),J=1,6)
      READ (5,27) ((EPSI(I,J),J=1,6),I=1,20)
      READ (5,24) (EPSLON(1,J),J=1,9)
      READ (5,25) ((QU(I,J),J=1,9),I=1,20)
      READ (5,28) (TTEMP(1,J),J=1,16)
      READ (5,29) ((ZZ(I,J),J=1,16),I=1,7)
      DO 1 I=1,20
      DO 1 J=2,20
    1 EMM(J,I)=EMM(J-1,I)
      DO 2 I=1,23
      DO 2 J=2,14
    2 Z(J,I)=Z(J-1,I)
      DO 3 I=1,7
      DO 3 J=2,20
    3 ECCR(J,I)=ECCR(J-1,I)
      DO 4 I=1,9
      DO 4 J=2,11
    4 ELOL(J,I)=ELOL(J-1,I)
      DO 5 I=1,9
      DO 5 J=2,20
    5 EPSLON(J,I)=EPSLON(J-1,I)
      DO 6 I=1,6
      DO 6 J=2,20
    6 C(J,I)=C(J-1,I)
      DO 7 I=1,16
      DO 7 J=2,7
    7 TTEMP(J,I)=TTEMP(J-1,I)
      IF (KLASS.EQ.1) GO TO 12
      IF (KB.EQ.2.OR.KB.EQ.3) GO TO 10
      IF (N.EQ.6) GO TO 9
      IF (N.EQ.8) GO TO 8
      GO TO (8,9), KE
    8 NCONS=11
      GO TO 13
    9 NCONS=10
      GO TO 13
   10 IF (N.EQ.4) GO TO 12
      IF (N.EQ.6) GO TO 11
      GO TO (11,12), KE
   11 NCONS=9
      GO TO 13
   12 NCONS=8
   13 RETURN
C
C
C
   14 FORMAT (13F5.0)
   15 FORMAT (13F5.0)
   16 FORMAT (20F4.0)
   17 FORMAT (10F6.0)
   18 FORMAT (1F6.0/10F6.0/12F6.0)
   19 FORMAT (1F6.0/10F6.0/12F6.0)
```

```
20    FORMAT (7F4.2)
21    FORMAT (7F7.5)
22    FORMAT (9F4.2)
23    FORMAT (9F5.3)
24    FORMAT (9F4.2)
25    FORMAT (9F5.4)
26    FORMAT (6F4.2)
27    FORMAT (6F6.3)
28    FORMAT (16F5.1)
29    FORMAT (7F7.3/9F7.3)
      END
```

```
      SUBROUTINE ANSWER (U,X,PHI,PSI,N,NCONS,NEQUS)
C     ******************************************
C
      COMMON /OPTI/ KO,NNDEX
      COMMON /111/ RPM,W,RADIUS,T1,KOPTIM,KBRG,MTHD,RADU,RADL,NMBR,KOILS
     1UP,KAPLIC,PINLET,KEY,KAIR,TINPUT,TMAX
      COMMON /KKHH/ HO,TO,JJJ,MTRL,TRQ,EPS,OILFLOW,P1,TOIL1,CHMFL,CHMFW,
     1ATITUD,X1,X2,X3,RAD
      COMMON /BLOCK/ KLASS
      DIMENSION X(1), PHI(1), PSI(1)
C     THIS SUBROUTINE IS USED MERELY TO OUTPUT THE FINAL SOLUTION IN A
C     STANDARD FORM. IF AN OPTIMUM IS NOT REACHED(KO=1)THEN THE RESULTS
C     AT THE LAST ITERATION MAY BE PRINTED OUT.
      CALL UREAL (X,U)
      IF (KO.EQ.0) GO TO 1
      WRITE (6,14)
      WRITE (6,15) U
      GO TO 2
 1    WRITE (6,16)
      WRITE (6,17) U
 2    WRITE (6,20) X1,X2,X3
      IF (KLASS.EQ.1) GO TO 12
      IF (KBRG.EQ.2.OR.KBRG.EQ.3) GO TO 7
      IF (N.EQ.7) GO TO 3
      IF (N.EQ.8) GO TO 6
      WRITE (6,21) CHMFL,CHMFW,TOIL1
      GO TO 13
 3    GO TO (4,5), KEY
 4    WRITE (6,22) P1,CHMFL,CHMFW,TOIL1
      GO TO 13
 5    WRITE (6,23) RAD,CHMFL,CHMFW,TOIL1
      GO TO 13
 6    WRITE (6,24) RAD,P1,CHMFL,CHMFW,TOIL1
      GO TO 13
 7    IF (N.EQ.5) GO TO 8
      IF (N.EQ.6) GO TO 11
      WRITE (6,25) TOIL1
      GO TO 13
 8    GO TO (9,10), KEY
 9    WRITE (6,26) P1,TOIL1
      GO TO 13
 10   WRITE (6,27) RAD,TOIL1
      GO TO 13
 11   WRITE (6,28) RAD,P1,TOIL1
      GO TO 13
 12   IF (N.EQ.4) WRITE (6,29) RAD
 13   I=N+1
      WRITE (6,30) I,HO
      I=I+1
      WRITE (6,31) I,EPS
      I=I+1
      WRITE (6,32) I,ATITUD
      I=I+1
      WRITE (6,33) I,TO
      I=I+1
      WRITE (6,34) I,TRQ
      I=I+1
      WRITE (6,35) I,OILFLOW
      I=I+1
      WRITE (6,36) I,JJJ
      I=I+1
      WRITE (6,37) I,MTRL
      WRITE (6,38)
      CALL CONST (X,NCONS,PHI)
      WRITE (6,18)
      WRITE (6,19) (I,PHI(I),I=1,NCONS)
      RETURN
C
C
 14   FORMAT (1H-,16X,25HRESULTS AT LAST ITERATION)
 15   FORMAT (/5X,*VALUE OF CRITERION FUNCTION. . . . . .U =*,F15.8//)
 16   FORMAT (1H1,21X,22HOPTIMUM SOLUTION FOUND,/21X,*  ----------------
     1-----*,/)
 17   FORMAT (/5X,*VALUE OF CRITERION FUNCTION. . . . . .U =*,F15.8//)
 18   FORMAT (/5X,22HINEQUALITY CONSTRAINTS,/5X,*----------------------*
     1,/)
```

```
19     FORMAT (23X,4HPHI(,I2,3H) =,E16.8)
20     FORMAT (///5X,*VALUE OF DESIGN VARIABLES*,/5X,*-------------------
      1------*,//2X,3H1- ,*BEARING LENGTH . . . . . . . . . . . X(1) =*,F15
      2.8,5X,*INCH*,/2X,3H2- ,*RADIAL CLEARANCE . . . . . . . . . . X(2) =*
      3,F15.8,5X,*INCH*,/2X,3H3- ,*OIL OPERATING VISCOSITY. . . . . . X(3
      4) =*,F15.8,5X,*CENTIPOISE*)
21     FORMAT (2X,3H4- ,*AXIAL LENGTH OF CHAMFER. . . . . . . X(4) =*,F15.8
      1,5X,*INCH*,/2X,3H5- ,*CHAMFER DIMENSION. . . . . . . . . . X(5) =*,F
      215.8,5X,*INCH*,/2X,3H6- ,*OIL INLET TEMPERATURE. . . . . . . X(6)
      3=*,F15.8,5X,*DEG. FAHRENHEIT*)
22     FORMAT (2X,3H4- ,*OIL INLET PRESSURE . . . . . . . . . X(4) =*,F15.8
      1,5X,*LB/SQ.INCH*,/2X,3H5- ,*AXIAL LENGTH OF CHAMFER. . . . . . X(5
      2) =*,F15.8,5X,*INCH*,/2X,3H6- ,*CHAMFER DIMENSION. . . . . . . . .
      3 X(6) =*,F15.8,5X,*INCH*,/2X,3H7- ,*OIL INLET TEMPERATURE. . . . .
      4 . . X(7) =*,F15.8,5X,*DEG. FAHRENHEIT*)
23     FORMAT (2X,3H4- ,*JOURNAL RADIUS . . . . . . . . . . . X(4) =*,F15.8
      1,5X,*INCH*,/2X,3H5- ,*AXIAL LENGTH OF CHAMFER. . . . . . . X(5) =*,F
      215.8,5X,*INCH*,/2X,3H6- ,*CHAMFER DIMENSION. . . . . . . . . . X(6)
      3=*,F15.8,5X,*INCH*,/2X,3H7- ,*OIL INLET TEMPERATURE. . . . . . . . X
      4(7) =*,F15.8,5X,*DEG. FAHRENHEIT*)
24     FORMAT (2X,3H4- ,*JOURNAL RADIUS . . . . . . . . . . . X(4) =*,F15.8
      1,5X,*INCH*,/2X,3H5- ,*OIL INLET PRESSURE . . . . . . . . X(5) =*,F
      215.8,5X,*LB/SQ.INCH*,/2X,3H6- ,*AXIAL LENGTH OF CHAMFER. . . . . .
      3 X(6) =*,F15.8,5X,*INCH*,/2X,3H7- ,*CHAMFER DIMENSION. . . . . . .
      4 . . X(7) =*,F15.8,5X,*INCH*,/2X,3H8- ,*OIL INLET TEMPERATURE. . .
      5 . . . . X(8) =*,F15.8,5X,*DEG. FAHRENHEIT*)
25     FORMAT (2X,3H4- ,*OIL INLET TEMPERATURE. . . . . . . . X(4) =*,F15.8
      1,5X,*DEG. FAHRENHEIT*)
26     FORMAT (2X,3H4- ,*OIL INLET PRESSURE . . . . . . . . . X(4) =*,F15.8
      1,5X,*LB/SQ.INCH*,/2X,3H5- ,*OIL INLET TEMPERATURE. . . . . . . X(5
      2) =*,F15.8,5X,*DEG. FAHRENHEIT*)
27     FORMAT (2X,3H4- ,*JOURNAL RADIUS . . . . . . . . . . . X(4) =*,F15.8
      1,5X,*INCH*,/2X,3H5- ,*OIL INLET TEMPERATURE. . . . . . . X(5) =*,F
      215.8,5X,*DEG. FAHRENHEIT*)
28     FORMAT (2X,3H4- ,*JOURNAL RADIUS . . . . . . . . . . . X(4) =*,F15.8
      1,5X,*INCH*,/2X,3H5- ,*OIL INLET PRESSURE. . . . . . . . . X(5) =*,F
      215.8,5X,*LB/SQ.INCH*,/2X,3H6- ,*OIL INLET TEMPERATURE. . . . . . .
      3 X(6) =*,F15.8,5X,*DEG. FAHRENHEIT*)
29     FORMAT (2X,3H4- ,*JOURNAL RADIUS . . . . . . . . . . . X(4) =*,F15.8
      1,5X,*INCH*)
30     FORMAT (/1X,I2,*- MINIMUM OIL FILM THICKNESS . . . . . . . =*,F15.8
      1,5X,*INCH*)
31     FORMAT (1X,I2,*- ECCENTRICITY RATIO . . . . . . . . . . . =*,F15.8)
32     FORMAT (1X,I2,*- ATTITUDE ANGLE . . . . . . . . . . . . . =*,F15.8,
      15X,*DEGREES*)
33     FORMAT (1X,I2,*- OIL OPERATING TEMPERATURE. . . . . . . . =*,F15.8,
      15X,*DEG. FAHRENHEIT*)
34     FORMAT (1X,I2,*- FRICTIONAL TORQUE. . . . . . . . . . . . =*,F15.8,
      15X,*LB. INCH*)
35     FORMAT (1X,I2,*- AMOUNT OF OIL THAT MUST   *,/5X,*BE CONTINUOUSLY
      1SUPPLIED . . . . . . . . =*,F15.8,5X,*GALLON/MIN.*)
36     FORMAT (1X,I2,*- OIL GRADE. . . . . . . . . . . . . . . . =  (SAE*,
      1I4,1H))
37     FORMAT (1X,I2,*- BEARING METAL IS NO.    (*,I2,1H),*    - SEE USER
      1S MANUAL, MATERIAL LIST -*)
38     FORMAT (//5X,*-RECOMMENDATIONS-*,/6X,*---------------*,//2X,3H1- ,
      1*MAXIMUM SURFACE FINISH ROUGHNESS SHOULD BE LESS OR EQUAL TO (20-M
      2ICRON INCH)*,/2X,3H2- ,*SHAFT BRINELL HARDNESS MUST BE GREATER OR
      3EQUAL TO (300)*)
       END
```

```
      FUNCTION FTABLE1 (VAR,FUNC,XX,M,L)                                E
C     *********************************                                 E
C                                                                       E
      DIMENSION VAR(20,1), FUNC(20,1)                                   E
      NEND=M-1                                                          E
      DO 1 I=1,NEND                                                     E
      INT=I                                                             E
      IF (XX.GE.VAR(L,I).AND.XX.LE.VAR(L,I+1)) GO TO 2                  E
1     CONTINUE                                                          E
2     FTABLE1=FUNC(L,INT)+(FUNC(L,INT+1)-FUNC(L,INT))*(XX-VAR(L,INT))/(V E
     1AR(L,INT+1)-VAR(L,INT))                                           E
      RETURN                                                            E
      END                                                               E


      FUNCTION FTABLE2 (VAR,FUNC,XX,M,L)                                F
C     *********************************                                 F
C                                                                       F
      DIMENSION VAR(14,1), FUNC(14,1)                                   F
      NEND=M-1                                                          F
      DO 1 I=1,NEND                                                     F
      INT=I                                                             F
      IF (XX.GE.VAR(L,I).AND.XX.LE.VAR(L,I+1)) GO TO 2                  F
1     CONTINUE                                                          F
2     FTABLE2=FUNC(L,INT)+(FUNC(L,INT+1)-FUNC(L,INT))*(XX-VAR(L,INT))/(V F
     1AR(L,INT+1)-VAR(L,INT))                                           F
      RETURN                                                            F
      END                                                               F


      FUNCTION FTABLE3 (VAR,FUNC,XX,M,L)                                G
C     *********************************                                 G
C                                                                       G
      DIMENSION VAR(11,1), FUNC(11,1)                                   G
      NEND=M-1                                                          G
      DO 1 I=1,NEND                                                     G
      INT=I                                                             G
      IF (XX.GE.VAR(L,I).AND.XX.LE.VAR(L,I+1)) GO TO 2                  G
1     CONTINUE                                                          G
2     FTABLE3=FUNC(L,INT)+(FUNC(L,INT+1)-FUNC(L,INT))*(XX-VAR(L,INT))/(V G
     1AR(L,INT+1)-VAR(L,INT))                                           G
      RETURN                                                            G
      END                                                               G


      FUNCTION FTABLE4 (VAR,FUNC,XX,M,L)                                H
C     *********************************                                 H
C                                                                       H
      DIMENSION VAR(7,1), FUNC(7,1)                                     H
      NEND=M-1                                                          H
      DO 1 I=1,NEND                                                     H
      INT=I                                                             H
      IF (XX.GE.VAR(L,I).AND.XX.LE.VAR(L,I+1)) GO TO 2                  H
1     CONTINUE                                                          H
2     FTABLE4=FUNC(L,INT)+(FUNC(L,INT+1)-FUNC(L,INT))*(XX-VAR(L,INT))/(V H
     1AR(L,INT+1)-VAR(L,INT))                                           H
      RETURN                                                            H
      END                                                               H
```

```
      SUBROUTINE SEEK1(N,RMAX,RMIN,NCONS,NEQUS,F,G,XSTRT,NSHOT,NTEST,MAX
     1M,IPRINT,IDATA,X,U,PHI,PSI,WORK1,WORK2,WORK3,WORK4)
      DIMENSION RMAX(1),RMIN(1),XSTRT(1),X(1),PHI(1),PSI(1),WORK1(1),WOR
     1K2(1),WORK3(1),WORK4(1)
      COMMON/OPTI/KO,NNDEX
      COMMON/111/RPM,W,RADIUS,T1,KOPTIM, KBRG  ,MTHD,RADU,RADL,NMBR
     1,KOILSUP,KAPLIC,PINLET,KEY,KAIR,TINPUT,TMAX
      COMMON/BLOCK/KLASS
      IF(IDATA.NE.1)GO TO 116
C
      WRITE(6,713)
      WRITE(6,300) W
      WRITE(6,301) RPM
      WRITE(6,302) T1
      WRITE(6,717) TMAX
      WRITE(6,303) RADIUS
      WRITE(6,304) RADU
      WRITE(6,305) RADL
      WRITE(6,306) KOPTIM
      WRITE(6,307) KAPLIC
      WRITE(6,712) NMBR
      WRITE(6,310)MTHD
      WRITE(6,716) TINPUT
      IF (KLASS.EQ.1) GO TO 117
      WRITE(6,308) KBRG
      WRITE(6,309) KEY
      WRITE(6,311) PINLET
      GO TO 116
 117  WRITE(6,714) KOILSUP
      WRITE(6,715) KAIR
  116 WRITE(6,19)
C     ZERO WORKING ARRAYS
      DO 100 I=1,N
      X(I)=0.0
      WORK1(I)=0.0
      WORK2(I)=0.0
      WORK3(I)=0.0
  100 WORK4(I)=0.0
C
C     SEARCH IS USED BY BOTH SEEK1 AND SEEK3
C
      KO=0
      NNDEX=1
      INDEX=1
C     INDEX=0 INDICATES TO SEARCH THAT IT IS BEING USED BY FEASBL
      KOUNT=0
      R=1.
    2 CALL SEARCH(X,U,N,XSTRT,RMAX,RMIN,PHI,PSI,NCONS,NEQUS,MAXM,NVIOL,F
     1,G,IPRINT,INDEX,R,WORK1,WORK2,WORK3,WORK4)
      CALL SHOT(U,X,N,KK,PHI,PSI,NCONS,NEQUS,RMAX,RMIN,F,NTEST,NSHOT,WOR
     1K1,WORK2,WORK3)
C     CHECK TO SEE WHETHER SUBR.SHOT HAS FOUND AN IMPROVED POINT
      IF(KK.EQ.1) GO TO 4
      IF(KO.EQ.0)RETURN
C     KO  CANNOT BE RESET IN SUBR.SHOT, THEREFORE IF KO=1 AT THIS STAGE
C     THEN SUBR.SEARCH FAILED AND SHOT FOUND NO IMPROVEMENT
      WRITE(6,5)
      GOTO16
    4 IF(IPRINT.GT.0)WRITE(6,25)U,(X(I),I=1,N)
      KOUNT=KOUNT+1
      IF(KOUNT.LE.NSHOT)GOTO13
      WRITE(6,17)NSHOT
      KO=1
      GOTO16
C     REDEFINE STARTING POINT FOR SEARCH
   13 DO 14 I=1,N
   14 XSTRT(I)=X(I)
      GOTO 2
C     PRINT OUT LAST ITERATION RESULTS(KO=1)
   16 CALL ANSWER(U,X,PHI,PSI,N,NCONS,NEQUS)
    5 FORMAT(1H-,71HDIRECT SEARCH HAS HUNG UP AND SHOTGUN SEARCH CANNOT
     1FIND A BETTER POINT/41HTRY A DIFFERENT STARTING POINT AT LEVEL=1/)
    7 FORMAT(1H-,15X,1HU,25X,23HINDEPENDENT VARIABLES X//)
   19 FORMAT(1H1,10X,38HDIRECT SEARCH OPTIMIZATION USING SEEK1//)
   17 FORMAT(1H-,48HSHOTGUN SEARCH FOUND AN IMPROVEMENT BUT  NSHOT =,I6,
     118H HAS BEEN EXCEEDED)
```

```
  25 FORMAT(1H-,7H.SHOT. ,5E16.8/(24X,4E16.8))
300  FORMAT(61H0LOAD AT BEARING, LBS..................................... .
    1 W   =,E15.7)
301  FORMAT(61H0JOURNAL SPEED, REVS/MIN. ...............................R
    1PM   =,E15.7)
302  FORMAT(61H0AMBIENT TEMPERATURE, DEG.F. ..............................
    1T1   =,E15.7)
303  FORMAT(61H0JOURNAL RADIUS, INCH ........................... RADI
    1US   =,E15.7)
304  FORMAT(61H0ESTIMATED UPPER LIMIT OF RADIUS .................... RA
    1DU   =,E15.7)
305  FORMAT(61H0ESTIMATED LOWER LIMIT OF RADIUS .................... RA
    1DL   =,E15.7)
306  FORMAT(61H0OPTIMIZATION CRITERION ............................ KOPT
    1IM   =,I6)
307  FORMAT(61H0TYPE OF APPLICATION ............................... KAPL
    1IC   =,I6)
308  FORMAT(61H0TYPE OF JOURNAL BEARING USED.......................... KB
    1RG   =,I6)
309  FORMAT(61H0FLAG NUMBER ........................................... K
    1EY   =,I6)
310  FORMAT(61H0OPTIMIZATION METHOD USED ............................ MT
    1HD   =,I6)
311  FORMAT(61H0OIL INLET PRESSURE ............................... PINL
    1ET   =,E15.7)
712  FORMAT(61H0NUMBER OF DESIGN VARI BLES ............................ NM
    1BR   =,I6)
713  FORMAT(1H1,15X,39H- OPTIMUM HYDRODYNAMIC BEARING DESIGN -,/17X,*
    1-------------------------------*,//29X,10HINPUT DATA,/29X,*-----
    2-----*,//)
714  FORMAT(61H0KIND OF OIL SUPPLY .............................. KOILS
    1UP   =,I6)
715  FORMAT(61H0AMBIENT AIR CONDITION ............................. KA
    1IR   =,I6)
716  FORMAT(61H0INPUT TORQUE .................................... TINP
    1UT   =,E15.7)
717  FORMAT(61H0MAXIMUM EXPECTED AMBIENT TEMPERATURE .............. TM
    1AX   =,E15.7)
     RETURN
     END
```

```
      SUBROUTINE SIMPLEX(N,RMAX,RMIN,NCONS,NEQUS,XSTRT,NN,ALPHA,BETA,
     1 GAMA,REDUCE,R,F,G,MAXM,IPRINT,IDATA,U,X,PHI,PSI,XA,XJ,FUN,XH,XS,
     2 XL,XO,XR,XE,XC,STEP)
      DIMENSION X(1),XSTRT(1),RMAX(1),RMIN(1),PHI(1),PSI(1),XA(N,NN),
     1 XJ(1),FUN(1),XH(1),XS(1),XL(1),XO(1),XR(1),XE(1),XC(1),STEP(1)
      COMMON/111/RPM,W,RADIUS,T1,KOPTIM, KBRG  ,MTHD,RADU,RADL,NMBR
     1,KOILSUP,KAPLIC,PINLET,KEY,KAIR,TINPUT,TMAX
      COMMON/OPTI/KO,NNDEX
      COMMON/BLOCK/KLASS
C       CLEARING ALL THE ARRAYS BEFORE USE
      DO 1 J=1,NN
      DO 1 I=1,N
      XA(I,J)=0.0
      XH(I)=0.0
      XJ(I)=0.0
      FUN(I)=0.0
      XS(I)=0.0
      XL(I)=0.0
      XO(I)=0.0
      XR(I)=0.0
      XE(I)=0.0
    1 XC(I)=0.0
      WRITE(6,301)
  301 FORMAT(15X,*OPTIMIZATION BY SIMPLEX METHOD*,/)
      KOUNT=0
      DO 2 I=1,N
      STEP(I)=F*(ABS(RMAX(I)-RMIN(I)))
    2 XA(I,1)=XSTRT(I)
      LL=0
      KKK=0
      IF(IDATA.NE.1)GO TO 299
      WRITE(6,713)
      WRITE(6,700) W
      WRITE(6,701) RPM
      WRITE(6,702) T1
      WRITE(6,717) TMAX
      WRITE(6,703) RADIUS
      WRITE(6,704) RADU
      WRITE(6,705) RADL
      WRITE(6,706) KOPTIM
      WRITE(6,707) KAPLIC
      WRITE(6,712) NMBR
      WRITE(6,710)MTHD
      WRITE(6,716) TINPUT
      IF (KLASS.EQ.1) GO TO 117
      WRITE(6,708) KBRG
      WRITE(6,709) KEY
      WRITE(6,711) PINLET
      GO TO 299
 117  WRITE(6,714) KOILSUP
      WRITE(6,715) KAIR
C       GENERATING N+1 POINTS TO FORM A SIMPLEX
  299 DO 5 J=2,NN
      DO 5 I=1,N
      IF(I.EQ.(J-1))GO TO 4
      XA(I,J)=XA(I,1)
      GO TO 5
    4 XA(I,J)=XA(I,1)+STEP(I)
    5 CONTINUE
   80 CONTINUE
      LL=LL+1
      KOUNT=KOUNT+1
C     NOW WE COMPUTE ARTIFICIAL OBJECTIVE FUNCTION AT VARIOUS POINTS
      DO 10 J=1,NN
      DO 15 I=1,N
   15 XJ(I)=XA(I,J)
      CALL OPTIMF2(XJ,UART,PHI,PSI,NCONS,NEQUS,NVIOL,R)
   10 FUN(J)=UART
C     NOW WE ARRANGE FUNCTION VALUES IN ASCENDING ORDER TO SELECT HIGHEST
C     LOWEST,AND NEXT TO THE HIGHEST VALUES
      DO 20 K=1,N
      KK=K+1
      DO 20 J=KK,NN
      IF(FUN(K).LE.FUN(J)) GO TO 20
      TEMP=FUN(K)
      FUN(K)=FUN(J)
```

```
      FUN(J)=TEMP
      DO 45 I=1,N
      TEMP=XA(I,K)
      XA(I,K)=XA(I,J)
   45 XA(I,J)=TEMP
   20 CONTINUE
C      SELECTING VECTORS XH,XL,XO,XS ETC.
C      XH IS THE VECTOR GIVING MAXIMUM VALUE OF OBJECTIVE FUNCTION
C      XL IS THE VECTOR GIVING LOWEST VALUE OF THE OBJECTIVE FUNCTION
C      XO IS THE AVERAGE OF ALL POINTS OTHER THAN HIGHEST POINT XH
C      XS IS THE VECTOR WITH SECOND HIGHEST VALUE OF OBJECTIVE FUNCTION
      DO 30 I=1,N
      XH(I)=XA(I,N+1)
      XS(I)=XA(I,N)
      XL(I)=XA(I,1)
   30 XO(I)=0.0
      DO 35 I=1,N
      DO 35 J=1,N
   35 XO(I)=XO(I)+(1./FLOAT(N))*XA(I,J)
      UH=FUN(N+1)
      US=FUN(N)
      UL=FUN(1)
      CALL OPTIMF2(XO,UO,PHI,PSI,NCONS,NEQUS,NVIOL,R)
      IF(IPRINT.LE.0)GO TO 901
      IF(LL.GT.1)GO TO 804
      WRITE(6,801)
  801 FORMAT(1H1)
      WRITE(6,802)
  802 FORMAT(1HO,*INTERMEDIATE OUTPUT FOR SIMPLEX*,/)
      WRITE(6,799)
  799 FORMAT(1HO,*UART IS THE VALUE OF ARTIFICIAL UNCONSTRAINED OBJECTIV
     1E FUNCTION AT THE CENTROID*,/)
      WRITE(6,803)
  803 FORMAT(1HO,*STEP NO.        U                UART                VA
     1RIABLES X(I) AT THE CENTROID OF THE SIMPLEX*,/)
  804 IF(IPRINT.NE.KOUNT)GO TO 901
      KOUNT=0
      CALL UREAL(XO,U)
      WRITE(6,800)LL,U,UO,(XO(I),I=1,N)
  800 FORMAT(I5,3X,6E16.8,95(/,40X,4E16.8))
C     CRITERION FOR OPTIMUM
  901 USUM=0.0
      DO 300 I=1,NN
      UDIF=(FUN(I)-UO)
      UDIFSQ=UDIF*UDIF
  300 USUM=USUM+UDIFSQ
      CRIT=SQRT(USUM/FLOAT(N))
      IF(CRIT.LT.G)GO TO 400
      IF(LL.GE.MAXM)GO TO 350
C      WE TRY REFLECTION NOW
      DO 40 I=1,N
   40 XR(I)=XO(I)+ALPHA*(XO(I)-XH(I))
      CALL OPTIMF2(XR,UR,PHI,PSI,NCONS,NEQUS,NVIOL,R)
C      IF US IS GREATER THAN UR AND UR GREATER THAN UL, WE REREPLACE XH
C      BY XR AND RESTART FROM NEWLY FORMED SIMPLEX
      IF(US.GE.UR.AND.UR.GE.UL)GO TO 50
C      IF ABOVE CONDITION NOT MET WE TAKE NEXT STEP TO SEE IF UR IS LT UL
      GO TO 60
   50 DO 70 I=1,N
   70 XA(I,NN)= XR(I)
      GO TO 80
C      IF UR.LT.UL WE TRY EXPANSION HOPING THAT FURTH5R IMPROVEMENT IS
C      POSSIBLE
   60 IF(UR.LT.UL) GO TO 90
      GO TO 100
   90 DO 110 I=1,N
  110 XE(I)=XO(I)+GAMA*(XR(I)-XO(I))
      CALL OPTIMF2(XE,UE,PHI,PSI,NCONS,NEQUS,NVIOL,R)
C      IF EXPANSION IS SUCCESSFUL WE REPLA3E XH 2Y XE OTHERWISE BY XR
      IF(UE.LT.UL) GO TO 120
      GO TO 50
  120 DO 130 I=1,N
  130 XA(I,NN)=XE(I)
      GO TO 80
```

```
C         IF UR IS GT. UH WE DONT REPLACE XH BY XR OTHERWISE WE DO
  100  IF(UR.GT.UH)GO TO 150
       IF(UH.GT.UR.AND.UR.GT.US)GO TO 155
       GO TO 225
C         CHANGE XH BY XR
  155  DO 160 I=1,N
  160  XH(I)=XR(I)
       CALL OPTIMF2(XH,UH,PHI,PSI,NCONS,NEQUS,NVIOL,R)
C         WE NOW MAKE CONTRACTION MOVE
  150  DO 180 I=1,N
  180  XC(I)=XO(I)+BETA*(XH(I)-XO(I))
       CALL OPTIMF2(XC,UC,PHI,PSI,NCONS,NEQUS,NVIOL,R)
C         WE CHECK IF CONTRACTION HAS BEEN SUCCESSFUL
       IF (UH.GT.UC)GO TO 200
C         IF ABOVE MOVE IS NOT SUCCESSFUL WE REPLACE ALL POINTS OF SIMPLEX
C         AND RESTART AGAIN FROM THIS CHANGED SIMPLEX
  225  DO 220 J=1,NN
       DO 220 I=1,N
  220  XA(I,J)=0.5*(XA(I,J)+XL(I))
       GO TO 80
  200  DO 210 I=1,N
       XH(I)=XC(I)
  210  XA(I,N+1)=XH(I)
       GO TO 80
C         WE CHANGE THE OPTIMUM POINT IN AN ARRAY X
  350  KO=1
       DO 500 I=1,N
  500  X(I)=XL(I)
       WRITE(6,600)MAXM
  600  FORMAT(1H0,*SIMPLEX HAS HUNG UP AFTER*,I4,*ITERATIONS*,/)
       CALL ANSWER(U,X,PHI,PSI,N,NCONS,NEQUS)
       RETURN
  400  CALL UREAL(XL,UNEW)
       IF(NCONS.EQ.0.AND.NEQUS.EQ.0)GO TO 402
       KKK=KKK+1
       IF(KKK.EQ.1)GO TO 401
       IF(ABS(UOLD-UNEW).LT.G)GO TO 402
  401  DO 403 I=1,N
       XA(I,1)=XL(I)
  403  CONTINUE
       R=R*REDUCE
       UOLD=UNEW
       GO TO 299
402    CALL CONST (X,NCONS,PHI)
       DO 777 J=1,NCONS
       IF (PHI(J).LT.0.0) GO TO 888
 777   CONTINUE
       GO TO 999
 888   KO=1
       CALL ANSWER (U,X,PHI,PSI,N,NCONS,NEQUS)
       RETURN
 999   KO=0
       U=UNEW
       DO 501 I=1,N
  501  X(I)=XL(I)
 700   FORMAT(61H0LOAD AT BEARING, LBS................................ .
      1 W   =,E15.7)
 701   FORMAT(61H0JOURNAL SPEED, REVS/MIN. ...............................R
      1PM   =,E15.7)
 702   FORMAT(61H0AMBIENT TEMPERATURE, DEG.F. ............................
      1T1   =,E15.7)
 703   FORMAT(61H0JOURNAL RADIUS, INCH ........................... RADI
      1US   =,E15.7)
 704   FORMAT(61H0ESTIMATED UPPER LIMIT OF RADIUS ................. RA
      1DU   =,E15.7)
 705   FORMAT(61H0ESTIMATED LOWER LIMIT OF RADIUS ................. RA
      1DL   =,E15.7)
 706   FORMAT(61H0OPTIMIZATION CRITERION ........................ KOPT
      1IM   =,I6)
 707   FORMAT(61H0TYPE OF APPLICATION ............................ KAPL
      1IC   =,I6)
 708   FORMAT(61H0TYPE OF JOURNAL BEARING USED.................... KB
      1RG   =,I6)
 709   FORMAT(61H0FLAG NUMBER .................................... K
```

```
     1FY   =,I6)
710   FORMAT(61H0OPTIMIZATION METHOD USED ............................. MT
     1HD   =,I6)
711   FORMAT(61H0OIL INLET PRESSURE .................................. PINL
     1ET   =,E15.7)
712   FORMAT(61H0NUMBER OF DESIGN VARIABLES ........................... NM
     1BR   =,I6)
713   FORMAT(1H1,15X,39H- OPTIMUM HYDRODYNAMIC BEARING DESIGN -,/17X,*
     1-----------------------------------*,//29X,10HINPUT DATA,/29X,*-----
     2------*,//)
714   FORMAT(61H0KIND OF OIL SUPPLY .................................. KOILS
     1UP   =,I6)
715   FORMAT(61H0AMBIENT AIR CONDITION ............................... KA
     1IR   =,I6)
716   FORMAT(61H0INPUT TORQUE ........................................ TINP
     1UT   =,E15.7)
717   FORMAT(61H0MAXIMUM EXPECTED AMBIENT TEMPERATURE ................ TM
     1AX   =,E15.7)
      RETURN
      END
```

```
      SUBROUTINE DAVID(N,RMAX,RMIN,NCONS,NEQUS,XSTRT,G,F,MAXM,IPRINT,IDA
     1TA,R,REDUCE,U,X,PHI,PSI,H,GS,D,GN,GA,Y,DT,C,YT,PHX,PSX,PART,PAST,
     2 CH,UX)
      COMMON/111/RPM,W,RADIUS,T1,KOPTIM, KBRG   ,MTHD,RADU,RADL,NMBR
     1,KOILSUP,KAPLIC,PINLET,KEY,KAIR,TINPUT,TMAX
      COMMON/BLOCK/KLASS
C       DAVIDON FLETCHER AND POWELL METHOD OF OPTIMIZATION
      DIMENSION X(1),RMAX(1),RMIN(1),XSTRT(1),H(N,N),GS(1),D(1),GN(1),
     1 GA(1),Y(1),DT(N,N),YT(N,N),C(N,N),PHI(1),PSI(1),PHX(N,1),PSX(N,1)
     2,PART(1),PAST(1),CH(1),UX(1)
      COMMON/OPTI/KO,NNDEX
C       CLEARING ALL THE ARRAYS BEFORE USE
      DO 31 I=1,N
      GS(I)=0.0
      D(I)=0.0
      GN(I)=0.0
      GA(I)=0.0
      Y(I)=0.0
      PART(I)=0.0
      PAST(I)=0.0
      CH(I)=0.0
      UX(I)=0.0
      DO 31 J=1,N
      DT(I,J)=0.0
      YT(I,J)=0.0
      C(I,J)=0.0
   31 H(I,J)=0.0
      DO 50 I=1,N
      CH(I)=F*(ABS(RMAX(I)-RMIN(I)))
   50 X(I)=XSTRT(I)
      LK=1
      L=0
      WRITE(6,301)
  301 FORMAT(1H0,*OPTIMIZATION BY DAVIDON FLETCHER AND POWELL METHOD*,/)
      KOUNT=0
      IF(IDATA.NE.1)GO TO 299
      WRITE(6,713)
      WRITE(6,700) W
      WRITE(6,701) RPM
      WRITE(6,702) T1
      WRITE(6,717) TMAX
      WRITE(6,703) RADIUS
      WRITE(6,704) RADU
      WRITE(6,705) RADL
      WRITE(6,706) KOPTIM
      WRITE(6,707) KAPLIC
      WRITE(6,712) NMBR
      WRITE(6,710)MTHD
      WRITE(6,716) TINPUT
      IF (KLASS.EQ.1) GO TO 117
      WRITE(6,708) KBRG
      WRITE(6,709) KEY
      WRITE(6,711) PINLET
      GO TO 299
 117  WRITE(6,714) KOILSUP
      WRITE(6,715) KAIR
  299 CALL OPTIMF2(X,FUN1,PHI,PSI,NCONS,NEQUS,NVIOL,R)
C     SUBROUTINE PARTIAL RETURNS THE GRADIENTS REQUIRED FOR COMPUTATION
C     TO START WITH MATRIX H IS CHOSEN AS A UNIT MATRIX
          CALL    PARTIAL(X,N,NCONS,NEQUS,PHI,PSI,GS, R,    CH,UX,PSX,PHX,
     1PART,PAST)
      JJ=0
   52 DO 1 I=1,N
      DO 1 J=1,N
    1 H(I,J)=0.0
      DO 2 I=1,N
      KK=I
    2 H(I,KK)=1.0
      JJ=JJ+1
  100 DO 3 I=1,N
    3 D(I)=0.0
      DO 4 I=1,N
      DO 5 J=1,N
    5 D(I)=(H(I,J)*GS(J))+D(I)
      IF(D(I).EQ.0)D(I)=1.E-50
    4 D(I)=-D(I)
C     IF D(I) DOES NOT ENSURE THAT FUNCTION WILL DECREASE THEN RESET
```

```
C     H MATRIX AS A UNIT MATRIX
      IF(JJ.GT.1)GO TO 300
      DO 53 I=1,N
      IF((GS(I)/D(I)).GT.0.)GO TO 52
   53 CONTINUE
      JJ=0
      L=L+1
      FUNC=FUN1
C     SUBROUTINE FIND RETURNS ALMDA, WHICH GIVES OPTIMUM STEP LENGTH
      CALL FIND(X,ALMDA,D,N,PHI,PSI,NCONS,NEQUS,FUNC,R)
      DO 6 I=1,N
    6 X(I)=X(I)+ALMDA*D(I)
      CALL OPTIMF2(X,FUN2,PHI,PSI,NCONS,NEQUS,NVIOL,R)
C      IF FUNCTION STARTS INCREASING  PROGRAM IS RESTARTED WITH NEW R
      KOUNT=KOUNT+1
      IF(IPRINT.LE.0)GO TO 326
      IF(L.GT.1)GO TO 804
      WRITE(6,801)
  801 FORMAT(1H1)
      WRITE(6,802)
  802 FORMAT(1H0,* INTERMEDIATE OUTPUT FOR DAVIDON FLETCHER AND POWELL*,
     1 /)
      WRITE(6,799)
  799 FORMAT(1H0,* UART IS THE ARTIFICIAL UNCONSTRAINED OPTIMIZATION FUN
     1CTION*,/)
      WRITE(6,803)
  803 FORMAT(1H0,*STEP NO.*,,6X,*U*,12X,*UART*,30X,*INDEPENDENT VARIABLE
     1S X(I)*,/)
  804 IF(IPRINT.NE.KOUNT)GO TO 326
      KOUNT=0
      CALL UREAL(X,U)
      WRITE(6,327)L,U,FUN2,(X(I),I=1,N)
  327 FORMAT(I5,3X,6E16.8,/(40X,4E16.8))
C     CRITERION FOR OPTIMUM
  326 IF(ABS(FUN1-FUN2).LE.G)GO TO 89
      IF(L.GE.MAXM)GO TO 300
      IF(FUN2.LE.FUN1)GO TO 250
      DO 249 I=1,N
  249 X(I)=X(I)-ALMDA*D(I)
      FUN2=FUN1
      GO TO 89
  250 CONTINUE
         CALL    PARTIAL(X,N,NCONS,NEQUS,PHI,PSI,GN, R,    CH,UX,PSX,PHX,
     1 PART,PAST)
C      ******************************************************************
C     THIS SECTION COMPUTES MATRIX H TO BE USED IN THE NEXT ITERATION
      DO 7 I=1,N
    7 Y(I)=GN(I)-GS(I)
      DO 8 I=1,N
    8 GA(I)=0.0
      DO 9 I=1,N
      DO 9 K=1,N
    9 GA(I)=GA(I)+(H(I,K)*GS(K))
      PROD1=0.0
      DO 10 I=1,N
   10 PROD1=PROD1+GA(I)*GS(I)
      DO 11 I=1,N
   11 GA(I)=0.0
      DO 12 I=1,N
      DO 12 K=1,N
   12 GA(I)=GA(I)+(H(I,K)*Y(K))
      PROD2=0.0
      DO 13 I=1,N
   13 PROD2=PROD2+(GA(I)*Y(I))
      DO 14 I=1,N
      DO 14 J=1,N
   14 DT(I,J)=D(I)*D(J)
      DO 15 I=1,N
      DO 15 J=1,N
   15 YT(I,J)=Y(I)*Y(J)
      DO 16 I=1,N
      DO 16 J=1,N
   16 C(I,J)=0.0
      DO 17 I=1,N
      DO 17 J=1,N
```

```
      DO 17 K=1,N
   17 C(I,J)=(H(I,K)*YT(K,J))+C(I,J)
      DO 19 I=1,N
      DO 19 J=1,N
      SUM=0.0
      DO 20 K=1,N
   20 SUM=SUM+(C(I,K)*H(K,J))
   19 C(I,J)=SUM
      IF(ABS(PROD1).LT.1.E-30)PROD1=1.E-30
      IF(ABS(PROD2).LT.1.E-30)PROD2=1.E-30
      QUO1=ALMDA/PROD1
      QUO2=1./PROD2
      DO 21 I=1,N
      DO 21 J=1,N
      DT(I,J)=DT(I,J)*QUO1
   21 C(I,J)=C(I,J)*QUO2
      DO 22 I=1,N
      DO 22 J=1,N
   22 H(I,J)=H(I,J)+DT(I,J)-C(I,J)
C     ****************************************************************************
      DO 23 I=1,N
   23 GS(I)=GN(I)
      FUN1=FUN2
      GO TO 100
  300 KO=1
      WRITE(6,325)L
  325 FORMAT(1HO,*DAVIDON HAS HUNG UP AFTER *,I4,*ITERATIONS*,/)
      CALL ANSWER(U,X,PHI,PSI,N,NCONS,NEQUS)
      RETURN
   89 R=R*REDUCE
      CALL UREAL(X,UNEW)
      IF(LK.EQ.1)GO TO 88
      IF(ABS(UOLD-UNEW).LE.G)GO TO 200
   88 UOLD=UNEW
      LK=LK+1
      GO TO 299
  200 CALL CONST (X,NCONS,PHI)
      DO 777 J=1,NCONS
      IF (PHI(J).LT.0.0) GO TO 888
  777 CONTINUE
      GO TO 999
  888 KO=1
      CALL ANSWER (U,X,PHI,PSI,N,NCONS,NEQUS)
      RETURN
  999 KO=0
      U=UNEW
  302 FORMAT(61H0INTERMEDIATE OUTPUT EVERY IPRINT(TH) CYCLE. . . .  IPRI
     1NT  =,I6)
  700 FORMAT(61H0LOAD AT BEARING, LBS................................ .
     1 W  =,E15.7)
  701 FORMAT(61H0JOURNAL SPEED, REVS/MIN. ...........................R
     1PM  =,E15.7)
  702 FORMAT(61H0AMBIENT TEMPERATURE, DEG.F. .........................
     1T1  =,E15.7)
  703 FORMAT(61H0JOURNAL RADIUS, INCH ........................... RADI
     1US  =,E15.7)
  704 FORMAT(61H0ESTIMATED UPPER LIMIT OF RADIUS ................. RA
     1DU  =,E15.7)
  705 FORMAT(61H0ESTIMATED LOWER LIMIT OF RADIUS ................. RA
     1DL  =,E15.7)
  706 FORMAT(61H0OPTIMIZATION CRITERION .......................... KOPT
     1IM  =,I6)
  707 FORMAT(61H0TYPE OF APPLICATION ............................. KAPL
     1IC  =,I6)
  708 FORMAT(61H0TYPE OF JOURNAL BEARING USED.................... KB
     1RG  =,I6)
  709 FORMAT(61H0FLAG NUMBER ..................................... K
     1EY  =,I6)
  710 FORMAT(61H0OPTIMIZATION METHOD USED ........................ MT
     1HD  =,I6)
  711 FORMAT(61H0OIL INLET PRESSURE .............................. PINL
     1ET  =,E15.7)
  712 FORMAT(61H0NUMBER OF DESIGN VARI BLES ...................... NM
     1BR  =,I6)
```

```
713  FORMAT(1H1,15X,39H- OPTIMUM HYDRODYNAMIC BEARING DESIGN -,/17X,*
    1-------------------------------------*,//29X,10HINPUT DATA,/29X,*-----
    2-----*,//)
714  FORMAT(61H0KIND OF OIL SUPPLY ............................... KOILS
    1UP   =,I6)
715  FORMAT(61H0AMBIENT AIR CONDITION ................................ KA
    1IR   =,I6)
716  FORMAT(61H0INPUT TORQUE ..................................... TINP
    1UT   =,E15.7)
717  FORMAT(61H0MAXIMUM EXPECTED AMBIENT TEMPERATURE ............... TM
    1AX   =,E15.7)
     RETURN
     END
```

```
      SUBROUTINE SEEK3(N,RMAX,RMIN,NCONS,NEQUS,XSTRT,F,G,R,REDUCE,MAXM,I
     1NDEX,IPRINT,IDATA,U,X,PHI,PSI,NVIOL,WORK1,WORK2,WORK3,WORK4)
      DIMENSION RMAX(1),RMIN(1),XSTRT(1),X(1),PHI(1),PSI(1),WORK1(1),WOR
     1K2(1),WORK3(1),WORK4(1)
      COMMON/111/RPM,W,RADIUS,T1,KOPTIM, KBRG  ,MTHD,RADU,RADL,NMBR
     1,KOILSUP,KAPLIC,PINLET,KEY,KAIR,TINPUT,TMAX
      COMMON/OPTI/KO,NNDEX
      COMMON/BLOCK/KLASS
      IF(INDEX.EQ.0) GO TO 116
      IF(IDATA.NE.1)GO TO 116
      WRITE(6,713)
      WRITE(6,300) W
      WRITE(6,301) RPM
      WRITE(6,302) T1
      WRITE(6,717) TMAX
      WRITE(6,303) RADIUS
      WRITE(6,304) RADU
      WRITE(6,305) RADL
      WRITE(6,306) KOPTIM
      WRITE(6,307) KAPLIC
      WRITE(6,712) NMBR
      WRITE(6,310)MTHD
      WRITE(6,716) TINPUT
      IF (KLASS.EQ.1) GO TO 117
      WRITE(6,308) KBRG
      WRITE(6,309) KEY
      WRITE(6,311) PINLET
      GO TO 116
 117  WRITE(6,714) KOILSUP
      WRITE(6,715) KAIR
  116 IF(INDEX.NE.0)WRITE(6,9)
C     ZERO WORKING ARRAYS
      DO 100 I=1,N
      X(I)=0.0
      WORK1(I)=0.0
      WORK2(I)=0.0
      WORK3(I)=0.0
  100 WORK4(I)=0.0
      KO=0
C     IF INDEX=0 SEEK3 HAS BEEN CALLED BY FEASBL
      ULAST=10.0E+40
      KOUNT=0
C     DEFINE NNDEX=2 SO THAT SEARCH WILL FUNCTION CORRECTLY
      NNDEX=2
    1 CALL SEARCH(X,U,N,XSTRT,RMAX,RMIN,PHI,PSI,NCONS,NEQUS,MAXM,NVIOL,F
     1,G,IPRINT,INDEX,R,WORK1,WORK2,WORK3,WORK4)
      IF(INDEX.EQ.0)RETURN
      IF(KO.NE.1)GOTO5
      WRITE(6,14)
      GOTO6
    5 KOUNT=KOUNT+1
      IF(IPRINT.EQ.0)GOTO2
      IF(KOUNT.EQ.IPRINT)WRITE(6,10)
      IF((KOUNT/IPRINT)*IPRINT.NE.KOUNT)GOTO2
      WRITE(6,4)R
      WRITE(6,11)U,(X(I),I=1,N)
    2 IF(ABS(U-ULAST).GT.1.E-07*ABS(ULAST))GOTO7
C     OPTIMUM HAS BEEN REACHED
      RETURN
    6 CALL ANSWER(U,X,PHI,PSI,N,NCONS,NEQUS)
      RETURN
    7 IF(R.GT.1.0E-20)GOTO8
      WRITE(6,12)R
      KO=1
      GOTO6
    8 ULAST=U
      R=R*REDUCE
      DO 3 I=1,N
    3 XSTRT(I)=X(I)
      GOTO1
    4 FORMAT(1H0,3HR =,F16.8)
    9 FORMAT(1H1,45HOPTIMIZATION USING DIRECT SEARCH METHOD SEEK3,//)
   10 FORMAT(1H0,38X,27HINDEPENDENT VARIABLES  X(I)//)
   11 FORMAT(1X,3HU =,E16.8,1X,4E16.8/(21X,4E16.8))
   12 FORMAT(1H0,23HNO CONVERGENCE WITH R =,E16.8)
   14 FORMAT(66H1SEEK3 UNABLE TO FIND A FEASIBLE STARTING POINT(ALL PHI(
```

```
     1I).GE.0.0)/)
300   FORMAT(61HOLOAD AT BEARING, LBS.............................. .
     1 W    =,E15.7)
301   FORMAT(61HOJOURNAL SPEED, REVS/MIN. ..........................R
     1PM    =,E15.7)
302   FORMAT(61HOAMBIENT TEMPERATURE, DEG.F. .........................
     1T1    =,E15.7)
303   FORMAT(61HOJOURNAL RADIUS, INCH ........................... RADI
     1US    =,E15.7)
304   FORMAT(61HOESTIMATED UPPER LIMIT OF RADIUS .................. RA
     1DU    =,E15.7)
305   FORMAT(61HOESTIMATED LOWER LIMIT OF RADIUS .................. RA
     1DL    =,E15.7)
306   FORMAT(61HOOPTIMIZATION CRITERION ......................... KOPT
     1IM    =,I6)
307   FORMAT(61HOTYPE OF APPLICATION ............................ KAPL
     1IC    =,I6)
308   FORMAT(61HOTYPE OF JOURNAL BEARING USED...................... KB
     1RG    =,I6)
309   FORMAT(61HOFLAG NUMBER ........................................ K
     1EY    =,I6)
310   FORMAT(61HOOPTIMIZATION METHOD USED ......................... MT
     1HD    =,I6)
311   FORMAT(61HOOIL INLET PRESSURE ............................. PINL
     1ET    =,E15.7)
712   FORMAT(61HONUMBER OF DESIGN VARI BLES ....................... NM
     1BR    =,I6)
713   FORMAT(1H1,15X,39H- OPTIMUM HYDRODYNAMIC BEARING DESIGN -,/17X,*
     1-------------------------------------*,//29X,10HINPUT DATA,/29X,*-----
     2-----*,//)
714   FORMAT(61HOKIND OF OIL SUPPLY ............................ KOILS
     1UP    =,I6)
715   FORMAT(61HOAMBIENT AIR CONDITION ............................ KA
     1IR    =,I6)
716   FORMAT(61HOINPUT TORQUE ................................... TINP
     1UT    =,E15.7)
717   FORMAT(61HOMAXIMUM EXPECTED AMBIENT TEMPERATURE ............... TM
     1AX    =,E15.7)
      END
```

```
      SUBROUTINE FEASBL(N,RMAX,RMIN,NCONS,NEQUS,XSTRT,F,G,MAXM,IPRINT,ID
     1ATA,U,X,PHI,PSI,STEPP,WORK1,WORK2,WORK3,WORK4)
      DIMENSION RMAX(1),RMIN(1),XSTRT(1),X(1),PHI(1),PSI(1),STEPP(1)
      DIMENSION WORK1(1),WORK2(1),WORK3(1),WORK4(1)
      COMMON/OPTI/KO,NNDEX
C     THIS SUBROUTINE USES  SEEK3  TO DRIVE ALL PHI(I) FEASIBLE AND THEN
C     REDUCES THE PSI(I)S BY MINIMIZING  SIGMA(PSI(I)) SUBJECT TO THE
C     CONDITION THAT ALL PHI(I) REMAIN FEASIBLE(.GE.0.)
      DO 91 I=1,N
      X(I)=0.0
   91 STEPP(I)=0.0
      KO=0
      R=1.
      REDUCE=.04
      INDEX=0
      KUT=0
      DO 9 I=1,N
    9 X(I)=XSTRT(I)
      IF(NCONS.EQ.0)GOTO13
      CALL UREAL(X,U)
      CALL CONST(X,NCONS,PHI)
      DO 10 I=1,NCONS
      IF(PHI(I).LT.0.0)GOTO11
   10 CONTINUE
      GOTO13
C     IF ANY PHI(I).LT.0. CALL SEEK3 TO DRIVE THEM FEASIBLE
   11 CALL SEEK3(N,RMAX,RMIN,NCONS,NEQUS,XSTRT,F,G,R,REDUCE,MAXM,INDEX,I
     1PRINT,IDATA,U,X,PHI,PSI,NVIOL,WORK1,WORK2,WORK3,WORK4)
      IF(NVIOL.EQ.0)GOTO13
C     IF SEEK3 COULD NOT  GET ALL PHI(I).GE.0. THEN SUBR.FEASBL CANNOT
C     OBTAIN A FEASIBLE POINT
      KO=1
      GO TO 32
   13 IF(NEQUS.EQ.0) RETURN
C     MINIMIZE SIGMA(PSI(I)) KEEPING ALL PHI(I).GE.0.
C     NOTE...THE FRACTION OF THE RANGE USED AS STEP SIZE SHOULD NOT
C     EXCEED 5 PERCENT. IF THE USER IS INTERESTED IN A VERY FEASIBLE
C     POINT(IE.ALL PSI(I)S VERY SMALL)HE CAN GIVE (F) A VERY SMALL VALUE
      PERCNT=0.05
      IF(ABS(F).LT.0.05)PERCNT=F
      DO 14 I=1,N
   14 STEPP(I)=PERCNT*(RMAX(I)-RMIN(I))
C     INITIALIZE  THE SUM OF THE PSI(I)S
      CALL SUMPSI(X,PSI,NEQUS,SUMO)
   15 NFAIL=0
      DO 25 I=1,N
      X(I)=X(I)+STEPP(I)
      CALL UREAL (X,U)
      CALL CONST(X,NCONS,PHI)
      DO 17 J=1,NCONS
C     IGNORE A MOVE WHICH MAKES ANY PHI(I).LT.0.0
      IF(PHI(J).LT.0.0)GOTO19
   17 CONTINUE
      CALL SUMPSI(X,PSI,NEQUS,SUM1)
      IF(SUM1.GE.SUMO)GOTO19
      SUMO=SUM1
      GOTO25
   19 X(I)=X(I)-2.0*STEPP(I)
      CALL UREAL (X,U)
      CALL CONST(X,NCONS,PHI)
      DO 21 L=1,NCONS
      IF(PHI(L).LT.0.0)GOTO23
   21 CONTINUE
      CALL SUMPSI(X,PSI,NEQUS,SUM2)
      IF(SUM2.GE.SUMO)GOTO23
      SUMO=SUM2
      GOTO25
   23 X(I)=X(I)+STEPP(I)
      NFAIL=NFAIL+1
   25 CONTINUE
      IF(NFAIL.EQ.N)GOTO27
      GOTO15
C     REDUCE STEPP(I) BY A FACTOR OF 4.0 UP TO 4 TIMES. THIS MEANS STEPP
C     REDUCES TO  LESS THAN .0002*(RMAX(I)-RMIN(I)), OR IF  F.LT.0.05
C     THEN MINIMUM STEPP(I)=(F/256)*(RMAX(I)-RMIN(I)). THEREFORE THE
C     USER MAY DRIVE THE PSI(I) VALUES AS SMALL AS HE WISHES BY ENTERING
```

```
C     A VERY SMALL VALUE OF  F  AT LEVEL=1
   27 KUT=KUT+1
      IF(KUT.GT.4)GOTO31
      DO 29 I=1,N
   29 STEPP(I)=STEPP(I)/4.0
      GOTO15
   31 CALL UREAL(X,U)
      RETURN
   32 WRITE(6,41)
   41 FORMAT(1H0,37HFEASBL COULD NOT FIND FEASIBLE REGION,/)
      RETURN
      END
```

```
      SUBROUTINE FRANDN(A,N,M)
      DIMENSION A(1)
      B=262147.0
      X=M
      X=X/0.8719467
   20 IF(X.NE.0.0)Y=AMOD(ABS(X),3.18967)
      DO 10 K=1,N
      DO 11 J=1,2
   11 Y=AMOD(B*Y,1.0)
      A(K)=Y
   10 IF(Y.EQ.0.0.OR.Y.EQ.1.0)Y=0.182818285
      RETURN
      END
```

```
      SUBROUTINE OPTIMF1(X,UART,PHI,PSI,NCONS,NEQUS,NVIOL)
      DIMENSION X(1),PHI(1),PSI(1)
C     VERY MINOR VIOLATIONS OF INEQUALITY CONSTRAINTS SHOULD NOT MAKE
C     THE ENTIRE SOLUTION INFEASIBLE. THEREFORE TEST FOR PHI(I).GE.ZERO
C     WHERE ZERO=-1.0E-10
      ZERO=-1.0E-10
      NVIOL=0
      SUM1=0.0
      SUM2=0.0
      CALL UREAL(X,U)
C
C     SEEK1  PENALTY  FUNCTIONS -
C
C     A  ROUTINE TO CALCULATE A VALUE FOR AN ARTIFICIAL OBJECTIVE
C     FUNCTION OF THE FORM
C        UART=UREAL+SUM(ABS(PHI(I)))*10.E20+SUM(ABS(PSI(I)))*10.E20
C     WHERE
C     PSI(I) AND PHI(I) IN THE ABOVE EXPRESSION ARE THE VALUES OF THE
C     CORESPONDING EQUALITY AND INEQUALITY CONSTRAINTS THAT HAVE BEEN
C     VIOLATED
      IF(NCONS.EQ.0)GOTO2
      CALL CONST(X,NCONS,PHI)
      DO 1 I=1,NCONS
      IF(PHI(I).GE.ZERO)GOTO1
      SUM1=SUM1 + ABS(PHI(I))*10.0E+20
      NVIOL=NVIOL + 1
    1 CONTINUE
    2 IF(NEQUS.EQ.0)GOTO115
      CALL EQUAL(X,PSI,NEQUS)
      DO 3 I=1,NEQUS
    3 SUM2=SUM2 + ABS(PSI(I))*10.0E+20
  115 UART=U+SUM1+SUM2
      RETURN
      END
```

```
      SUBROUTINE SHOT(U,X,N,KK,PHI,PSI,NCONS,NEQUS,RMAX,RMIN,F,NTEST,NSH
     1OT,RR,XX,RF)
      DIMENSION PHI(1),PSI(1),RMAX(1),RMIN(1),X(1),RR(1),XX(1),RF(1)
      COMMON/OPTI/KO,NNDEX
C     U=OPTIMUM DETERMINED BY DIRECT SEARCH.  IT IS CHANGED TO IMPROVED
C     VALUE IF SUCH A VALUE IS OBTAINED
C     XX= TRIAL VALUES OF X(I) FROM SHOTGUN SEARCH
C     RF= FRACTION OF RANGE USED IN SHOTGUN SEARCH
C     KK= INDICATOR TO SHOW IF U RETURNED IS AN IMPROVEMENT
C     INITIALIZE RANDOM NUMBER GENERATOR
      CALL FRANDN(RR,N,1)
      UMIN=U
      KK=0
C     THIS SHOTGUN SEARCH IS INTENDED TO GET THE SOLUTION OFF A FENCE
C     RATHER THAN TO INCH IT TOWARDS THE OPTIMUM. THEREFORE LARGE STEPS,
C     EQUAL 10. TIMES THE INITIAL STEP SIZE IN SEARCH ARE TRIED.
      DO 1 I=1,N
    1 RF(I)=10.*F*ABS(RMAX(I)-RMIN(I))
      DO 4 J=1,NTEST
      CALL FRANDN(RR,N,0)
      DO 2 I=1,N
    2 XX(I)=(X(I)-RF(I))+RR(I)*2.0*RF(I)
      CALL OPTIMF1(XX,UTEST,PHI,PSI,NCONS,NEQUS,NVIOL)
      IF(NVIOL.NE.0)GOTO4
      IF(UTEST.GE.UMIN)GOTO4
      UMIN=UTEST
      U=UTEST
      DO 3 I=1,N
    3 X(I)=XX(I)
    4 CONTINUE
      KK=1
      RETURN
      END
```

```
      SUBROUTINE SEARCH(X,U,N,XSTRT,RMAX,RMIN,PHI,PSI,NCONS,NEQUS,MAXM,N
     1VIOL,F,G,IPRINT,INDEX,R,XO,XB,DXXX,TXXX)
      DIMENSION X(1),XSTRT(1),RMAX(1),RMIN(1),PHI(1),PSI(1),XO(1),XB(1),
     1DXXX(1),TXXX(1)
      COMMON/OPTI/KO,NNDEX
C
C     DIRECT  SEARCH  PORTION  OF  SEEK1  AND  SEEK3
C
C     THIS IS THE DIRECT SEARCH ALGORITHM OF HOOKE AND JEEVES
C     SEARCH IS USED BE SEEK1 AND SEEK3
C     NNDEX=1  MEANS SEARCH HAS BEEN CALLED BY SEEK1
C     NNDEX=2  MEANS SEARCH HAS BEEN CALLED BY SEEK3
      NVIOL1=1
      KKK=0
      M1 = 0
   20 K1=1
      K2=N
   30 DO 40    I=K1,K2
      DXXX(I)=0.
      TXXX(I)=0.
      XO(I)=0.
   40 XB(I)=0.
      DO 60  I=K1,K2
   60 X(I) = XSTRT(I)
C  SET  FIRST  BASE  POINT
      DO 70 I=K1,K2
   70 XO(I) =X(I)
C     GENERATE  DELX(I)  AND  TEST(I)
      DO 80 I=K1,K2
      DXXX(I) = F*(RMAX(I)-RMIN(I))
   80 TXXX(I)=DXXX(I)*G
      NCALL=1
  100 CONTINUE
      GO TO (101,102)NNDEX
  101 CALL OPTIMF1(X,UART,PHI,PSI,NCONS,NEQUS,NVIOL)
      GO TO 110
  102 CALL OPTIMF2(X,UART,PHI,PSI,NCONS,NEQUS,NVIOL,R)
  110 IF(NCALL.NE.1)GOTO    120
      UARTO = UART
  120 CONTINUE
      IF(NVIOL.EQ.0)NVIOL1=0
      IF(NNDEX.EQ.1) GO TO 130
C     INDEX=0 INDICATES TO SEARCH THAT IT IS BEING USED BY FEASBL
      IF(INDEX.EQ.1) GO TO 130
C     IF SEARCH IS BEING USED MERELY TO OBTAIN A FEASIBLE STARTING POINT
C     THEN RETURN AS SOON AS SOLUTION GOES FEASIBLE
      IF(NVIOL1.EQ.0)GO TO    385
  130 GO TO (170, 200,   210,   355) NCALL
  170 CONTINUE
C  MAKE  SEARCH
  180 NFAIL=0
      DO 240 I=K1,K2
      X(I)=X(I)+DXXX(I)
      NCALL=2
      GO TO 100
  200 CONTINUE
      IF(UART.LT.UARTO)  GOTO  230
      X(I)=X(I) - 2.0*DXXX(I)
      NCALL=3
      GO TO 100
  210 CONTINUE
      IF(UART.LT.UARTO)  GOTO  230
      NFAIL = NFAIL + 1
      X(I)=X(I)+DXXX(I)
      GOTO  240
  230 UARTO = UART
  240 CONTINUE
  250 IF(NFAIL.EQ.N)GOTO 260
      GOTO  315
  260 DO 280 I=K1,K2
      IF(DXXX(I).GT.TXXX(I)) GO TO  290
  280 CONTINUE
      GO TO 385
  290 DO 310  I=K1,K2
  310 DXXX(I)=DXXX(I)/2.
      GOTO 180
```

```
C     ESTABLISH  NEW  BASE  POINT
  315 DO 320  I=K1,K2
  320 XB(I) = X(I)
      M1 = M1 + 1
      IF (NNDEX.EQ.1) GO TO 330
      GO TO  340
  330 KKK=KKK+1
      IF(KKK.NE.IPRINT) GO TO 340
      CALL UREAL(X,ULOW)
      WRITE (6,2)  M1,ULOW , (X(I), I=1,N)
      KKK=0
  340 CONTINUE
      IF(M1.GT.MAXM) GO TO 385
C     MAKE A PATTERN MOVE
      DO 350  I=K1,K2
  350 X(I) = X(I) + (X(I) - XO(I))
      NCALL=4
      GO TO 100
  355 CONTINUE
      IF(UART.LT.UARTO)  GOTO 370
      DO 360 I=K1,K2
      XO(I) = XB(I)
  360 X(I) = XB(I)
      GOTO 180
  370 DO 380  I=K1,K2
  380 XO(I) = XB(I)
      UARTO = UART
      GOTO 180
  385 CALL UREAL(X,U)
      GO TO(103,104)NNDEX
  103 CALL OPTIMF1(X,UART,PHI,PSI,NCONS,NEQUS,NVIOL)
      GO TO 105
  104 CALL OPTIMF2(X,UART,PHI,PSI,NCONS,NEQUS,NVIOL,R)
  105 IF(NVIOL.EQ.0)GOTO387
      IF(M1.GT.MAXM)WRITE(6,4)MAXM
      KO=1
  387 RETURN
    2 FORMAT(1H0,I4,3X,5E16.8/(24X,4E16.8))
    4 FORMAT(1H0,60HNO FEASIBLE SOLUTION AFTER ALLOWABLE NUMBER OF MOVES
     1, MAXM =,I6/)
      END
```

```
      SUBROUTINE PARTIAL(X,N,NCONS,NEQUS,PHI,PSI,G , R,    CH,UX,PSX,PHX,
     1 PART,PAST)
      COMMON LLST(50),NS(100),FN(100),SI(4,30),SO(4,30),SN(17,30)         HYD
      COMMON IS,NE,JJ,LOOP,NIN,NOUT,MSN,MODE,NPLNT,ISP                    HYD
      COMMON KPRNT(10),NCALC,NOCOMP,NSR                                   HYD
      COMMON EEN(600),NPOINT(25,2),NCOUNT                                 HYD
      COMMON XYZ(50), NC, NHEX
       COMMON M21,N21,M2P,N2P,NSNQ,NS1,NS2,NS3,NCOMP1,NCOMP2,NCOMP3
      DIMENSIONX(1),G(1),PHI(1),PSI(1),UX(1),PHX(N,1),PSX(N,1),CH(1),
     1 PART(1),PAST(1)
      DIV=SQRT(R)
      ZERO=-1.0E-10
      CALL SUPPLY(X,CH,PHI,PSI,PSX,PHX,UX,N,NCONS,NEQUS,PART,PAST)
       WRITE(6,49) ((J,X(J)),J=1,N)
      WRITE(6,51)  ((J,CH(J)),J=1,N)
      WRITE(6,52)  ((J,PHI(J)),J=1,NCONS)
      WRITE(6,53)  ((I,J,PHX(I,J),J=1,NCONS),I=1,N)
   49 FORMAT(2X,*  X(*,I2,*) =*,E15.7)
   51 FORMAT(2X,* CH(*,I2,*) =*,E15.7)
   52 FORMAT(2X,*PHI(*,I2,*) =*,E15.7)
   53 FORMAT(2X,*PHX(*,I2,*,*,I2,*) =*,E15.7)
      DO 10 I=1,N
   10 G(I)=UX(I)
      IF(NCONS.EQ.0)GO TO 1
      CALL UREAL (X,U)
      CALL CONST(X,NCONS,PHI)
      DO 20 I=1,N
      DO 20 J=1,NCONS
      IF(PHI(J).GT.ZERO)GO TO 21
      G(I)=G(I)+(10.E+20)*ABS(PHX(I,J))
      GO TO 20
   21 IF(PHX(I,J).LT.-ZERO)GO TO 20
      G(I)=G(I)+R/ABS(PHX(I,J))
   20 CONTINUE
    1 CONTINUE
      IF(NEQUS.EQ.0)GO TO 2
      DO 40 I=1,N
      DO 50 J=1,NEQUS
   50 G(I)=G(I)+ABS(PSI(J) **2)/DIV
   40 CONTINUE
    2 RETURN
      END
```

```
      SUBROUTINE SUPPLY(X,CH,PHI,PSI,PSX,PHX,UX,          N,NCONS,NEQUS,
     1 PART,PAST)
      COMMON LLST(50),NS(100),EN(100),SI(4,30),SO(4,30),SN(17,30)         HYD
      COMMON IS,NE,JJ,LOOP,NIN,NOUT,MSN,MODE,NPLNT,ISP                     HYL
      COMMON KPRNT(10),NCALC,NOCOMP,NSR                                    HYL
      COMMON EEN(600),NPOINT(25,2),NCOUNT                                  HYD
      COMMON XYZ(50), NC, NHEX
       COMMON M21,N21,M2P,N2P,NSNQ,NS1,NS2,NS3,NCOMP1,NCOMP2,NCOMP3
      DIMENSION X(1),UX(1),PSX(N ,1 ),PHX(N ,1 ),PHI(1),PSI(1),
     1 PART(1),PAST(1),CH(1)
      CALL UREAL(X,UO)
      DO 10 I=1,N
      X(I)=X(I)+CH(I)
      CALL UREAL(X,U)
      X(I)=X(I)-CH(I)
   10 UX(I)=(U-UO)/CH(I)
      IF(NCONS.EQ.0)GO TO 20
      CALL UREAL (X,U)
      CALL CONST(X,NCONS,PART)
      DO 30 I=1,N
      X(I)=X(I)+CH(I)
      CALL UREAL (X,U)
      CALL CONST(X,NCONS,PHI)
      X(I)=X(I)-CH(I)
      DO 30 J=1,NCONS
   30 PHX(I,J)=(PHI(J)-PART(J))/CH(I)
   20 IF(NEQUS.EQ.0)GO TO 40
      CALL EQUAL(X,PAST,NEQUS)
      DO 60 I=1,N
      X(I)=X(I)+CH(I)
      CALL EQUAL(X,PSI,NEQUS)
      X(I)=X(I)-CH(I)
      DO 60 J=1,NEQUS
   60 PSX(I,J)=(PSI(J)-PAST(J))/CH(I)
   40 RETURN
      END
```

```
      SUBROUTINE OPTIMF2(X,UART,PHI,PSI,NCONS,NEQUS,NVIOL,R)
      COMMON LLST(50),NS(100),EN(100),SI(4,30),SO(4,30),SN(17,30)          HYD
      COMMON IS,NE,JJ,LOOP,NIN,NOUT,MSN,MODE,NPLNT,ISP                     HYD
      COMMON KPRNT(10),NCALC,NOCOMP,NSR                                     HYD
      COMMON EEN(600),NPOINT(25,2),NCOUNT                                   HYD
      COMMON XYZ(50), NC, NHEX
       COMMON M21,N21,M2P,N2P,NSNQ,NS1,NS2,NS3,NCOMP1,NCOMP2,NCOMP3
      COMMON/OPTI/KO,NNDEX
      DIMENSION X(1),PHI(1),PSI(1)
C     VERY MINOR VIOLATIONS OF INEQUALITY CONSTRAINTS SHOULD NOT MAKE
C     THE ENTIRE SOLUTION INFEASIBLE, THEREFORE TEST FOR PHI(I).GE.ZERO
C     WHERE ZERO=-1.0E-10
      ZERO=-1.0E-10
      NVIOL=0
      SUM1=0.0
      SUM2=0.0
      CALL UREAL(X,U)
C     SEEK3  PENALTY  FUNCTIONS -
C
C     THE ARTIFICIAL OBJECTIVE FUNCTION IS OF THE FORM
C        UART=UREAL + R*SUM(1./PHI(I)) + SUM((PSI(J)**2)/SQRT(R))
C
  110 DIV=SQRT(R)
      IF(NCONS.LE.0)GOTO113
      CALL CONST(X,NCONS,PHI)
      DO 112 I=1,NCONS
      IF(PHI(I).GE.ZERO)GOTO111
      NVIOL=NVIOL+1
C     ADD A SEVERE PENALTY TO ANY PHI(I) WHICH IS VIOLATED
      SUM1=SUM1+ABS(PHI(I))*10.0E+20
      GOTO112
C     AVOID DIVIDING BY APPROXIMATELY ZERO, THERE IS NO POINT PENALIZING
C     A VERY SMALL PHI(I) ANYWAY
  111 IF(ABS(PHI(I)).LT.-ZERO)GOTO112
      SUM1=SUM1+R/ABS(PHI(I))
  112 CONTINUE
  113 IF(NEQUS.LE.0)GOTO115
      CALL EQUAL(X,PSI,NEQUS)
      DO 114 J=1,NEQUS
  114 SUM2=SUM2+(ABS(PSI(J))**2)/DIV                                        18/
  115 UART=U+SUM1+SUM2
      RETURN
      END
```

```
      SUBROUTINE FIND(X,ALMDA,D,N,PHI,PSI,NCONS,NEQUS,FUN1,R)
      COMMON LLST(50),NS(100),EN(100),SI(4,30),SO(4,30),SN(17,30)       HYD
      COMMON IS,NE,JJ,LOOP,NIN,NOUT,MSN,MODE,NPLNT,ISP                   HYD
      COMMON KPRNT(10),NCALC,NOCOMP,NSR                                  HYD
      COMMON EEN(600),NPOINT(25,2),NCOUNT                                HYD
      COMMON XYZ(50), NC, NHEX
       COMMON M21,N21,M2P,N2P,NSNQ,NS1,NS2,NS3,NCOMP1,NCOMP2,NCOMP3
      COMMON/OPTI/KO,NNDEX
      DIMENSION X(1),D(1),PHI(1),PSI(1)
      L=0
      A1=1.E-7
      K=1
      S=2.0
C      ***************************************************************
C     THIS SECTION FINDS BOUNDS ON THE VALUE OF ALMDA
   50 AL=A1*(((S**K)-1.)/(S-1.))
      DO 1 I=1,N
    1 X(I)=X(I)+AL*D(I)
      CALL OPTIMF2(X,FUN2,PHI,PSI,NCONS,NEQUS,NVIOL,R)
      DO 2 I=1,N
    2 X(I)=X(I)-AL*D(I)
      IF(FUN2.GT.FUN1)GO TO 10
      K=K+1
      FUN1=FUN2
      IF(K.GT.75)GO TO 40
      GO TO 50
   10 IF(K.NE.1)GO TO 9
      ALMDA=0.5*1.E-7
      GO TO 49
    9 IF(K.EQ.2)GO TO 11
      GO TO 12
   11 A=0.0
      B=1.E-6
      C=AL
      GO TO 13
   12 A=A1*((S**(K-2))-1.)/(S-1.)
      B=A1*((S**(K-1))-1.)/(S-1.)
      C=AL
   13 CONTINUE
C      ***************************************************************
C     THIS SECTION FINDS THE EXACT VALUE OF ALMDA BY QUADRATIC POLYNOMIAL S
C     BEST VALUE OF ALMDA IS BRACKETED WITHIN A AND C
      DO 3 I=1,N
    3 X(I)=X(I)+A*D(I)
      CALL OPTIMF2(X,FA   ,PHI,PSI,NCONS,NEQUS,NVIOL,R)
      DO 4 I=1,N
    4 X(I)=X(I)-A*D(I)
      DO 5 I=1,N
    5 X(I)=X(I)+B*D(I)
      CALL OPTIMF2(X,FB   ,PHI,PSI,NCONS,NEQUS,NVIOL,R)
      DO 6 I=1,N
    6 X(I)=X(I)-B*D(I)
      DO 7 I=1,N
    7 X(I)=X(I)+C*D(I)
      CALL OPTIMF2(X,FC   ,PHI,PSI,NCONS,NEQUS,NVIOL,R)
      DO 8 I=1,N
    8 X(I)=X(I)-C*D(I)
   19 AD1=(((B*B)-(C*C))*FA)+(((C*C)-(A*A))*FB)+(((A*A)-(B*B))*FC)
      AD2=2.*(((B-C)*FA)+((C-A)*FB)+((A-B)*FC))
      AD=AD1/AD2
      IF(ABS(A-C).LT.1.E-4)GO TO 20
      L=L+1
      IF(L.GT.20)GO TO 21
C     AD IS THE MINIMUM OF THE APPROXIMATE POLYNOMIAL PASSING THROUGH A B AN
      DO 51 I=1,N
   51 X(I)=X(I)+AD*D(I)
      CALL OPTIMF2(X,FD   ,PHI,P   ,NCONS,NEQUS,NVIOL,R)
      DO 52 I=1,N
   52 X(I)=X(I)-AD*D(I)
      IF(B.GT.AD)GO TO 15
      IF(FB.GT.FD)GO TO 16
      C=AD
      FC=FD
      GO TO 19
   16 A=B
      FA=FB
```

```
      B=AD
      FB=FD
      GO TO 19
   15 IF(FB.GT.FD)GO TO 17
      A=AD
      FA=FD
      GO TO 19
   17 C=B
      FC=FB
      B=AD
      FB=FD
      GO TO 19
C      ***********************************************************************
   21 ALMDA=B
      GO TO 49
   20 ALMDA=(A+C)/2.0
      GO TO 49
   40 WRITE(6,200)
  200 FORMAT(1HO,*BOUNDS COULD NOT BE ESTABLISHED*)
   49 RETURN
      END
```

Program Internal Data

CN

0.00	0.10	0.20	0.30	0.40	0.50	0.60	0.70	0.80	0.90	1.00	1.03	1.05
0.00	0.10	0.20	0.30	0.40	0.50	0.60	0.70	0.80	0.90	1.00	1.10	1.20
0.00	0.10	0.15	0.20	0.30	0.40	0.60	0.70	0.80	1.00	1.10	1.20	1.35
0.00	0.10	0.20	0.40	0.50	0.60	0.70	0.80	0.90	1.00	1.20	1.40	1.50
0.00	0.10	0.20	0.40	0.60	0.80	1.00	1.10	1.20	1.30	1.40	1.50	1.60
0.00	0.10	0.20	0.40	0.60	0.80	1.00	1.10	1.20	1.30	1.40	1.60	1.70
0.00	0.10	0.20	0.30	0.40	0.60	0.80	1.00	1.20	1.40	1.60	1.80	1.85
0.00	0.10	0.20	0.40	0.60	0.70	0.80	1.00	1.20	1.40	1.60	1.80	1.95
0.00	0.10	0.20	0.40	0.60	0.80	1.00	1.20	1.40	1.60	1.80	2.00	2.04
0.00	0.10	0.20	0.40	0.60	0.80	1.00	1.20	1.40	1.60	1.80	2.00	2.10
0.00	0.10	0.20	0.40	0.60	0.80	1.00	1.20	1.40	1.60	1.80	2.00	2.20
0.00	0.10	0.20	0.40	0.60	0.80	1.00	1.20	1.40	1.80	2.00	2.20	2.40
0.00	0.10	0.20	0.40	0.60	0.80	1.20	1.40	1.60	2.00	2.20	2.40	2.50
0.00	0.10	0.20	0.40	0.80	1.20	1.40	1.60	1.80	2.00	2.20	2.60	2.70
0.00	0.10	0.20	0.40	0.60	1.00	1.40	1.60	1.80	2.00	2.20	2.40	2.80
0.00	0.10	0.20	0.40	0.60	1.00	1.40	1.60	1.80	2.00	2.40	2.80	2.90
0.00	0.20	0.40	0.60	1.00	1.40	1.60	1.80	2.00	2.40	2.60	2.80	3.00
0.00	0.20	0.60	0.80	1.20	1.40	1.60	1.80	2.00	2.20	2.60	2.80	3.10
0.00	0.40	0.80	1.00	1.20	1.40	1.80	2.00	2.20	2.40	2.60	2.80	3.20
0.00	0.38	0.60	0.85	1.10	1.55	1.95	2.15	2.30	2.60	2.85	3.10	3.30

EM

01.0	10.0	14.0	17.0	20.0	23.0	26.0	29.0	32.0	35.0	38.0	39.0	40.0
01.0	09.0	12.8	15.9	19.0	21.25	23.5	26.25	29.0	31.5	34.0	37.0	40.0
01.0	09.0	10.5	12.0	14.5	17.0	21.3	23.8	26.3	31.0	33.5	36.0	40.0
01.0	08.0	10.5	16.0	18.0	20.0	22.0	24.0	26.0	28.0	33.0	37.0	40.0
01.0	07.5	10.0	14.4	19.0	22.4	26.0	28.0	30.0	32.0	34.0	37.0	40.0
01.0	07.0	09.5	13.5	17.5	21.0	24.0	26.0	28.0	30.0	32.0	36.5	40.0
01.0	06.3	09.0	11.0	13.0	16.8	20.0	23.2	27.0	30.5	34.5	39.0	40.0
01.0	06.4	08.7	12.7	16.0	17.5	19.0	22.2	25.5	28.5	32.5	37.0	40.0
01.0	06.0	08.3	12.0	15.3	18.4	21.0	24.0	29.0	30.5	34.5	39.0	40.0
01.0	05.5	08.0	11.5	14.5	17.5	20.0	23.0	26.0	29.0	33.0	37.0	40.0
01.0	05.5	07.5	11.0	14.0	17.0	19.5	22.0	25.0	28.0	31.5	35.0	40.0
01.0	05.0	07.0	10.5	13.0	15.5	18.0	20.5	23.0	29.0	33.0	36.0	40.0
01.0	04.5	06.5	09.5	12.0	14.0	19.0	21.4	24.0	30.0	33.5	37.5	40.0
01.0	04.0	05.5	08.0	13.0	17.0	19.0	21.5	24.0	27.0	30.0	38.0	40.0
01.0	03.5	05.0	07.5	10.0	14.0	18.0	20.5	23.0	25.5	28.5	32.0	40.0
01.0	03.0	04.0	06.5	09.0	13.0	17.0	19.0	21.5	24.0	30.0	38.0	40.0
01.0	04.0	06.0	08.0	12.0	16.0	18.5	20.5	23.0	29.0	32.5	36.0	40.0
01.0	03.5	07.4	09.0	13.0	15.0	16.7	19.0	21.0	23.5	29.5	33.5	40.0
01.0	05.0	09.0	10.5	12.5	14.0	18.0	20.0	22.5	25.0	28.5	32.0	40.0
01.0	04.0	06.0	08.0	10.0	14.0	18.0	20.0	22.0	26.0	30.0	34.0	40.0

EMM

1.	4.	6.	8.	10.	12.	14.	16.	18.	20.	22.	24.	26.	28.	30.	32.	34.	36.	38.	40.

TORQ

0.658	1.000	1.200	1.400	1.570	1.720	1.870	2.000	2.150	2.280
2.400	2.500	2.650	2.750	2.850	2.950	3.070	3.160	3.250	3.350
0.658	1.000	1.200	1.405	1.576	1.732	1.886	2.020	2.174	2.304
2.430	2.540	2.684	2.790	2.896	2.996	3.116	3.208	3.308	3.406
0.658	1.000	1.200	1.410	1.582	1.744	1.902	2.040	2.198	2.328
2.460	2.580	2.718	2.830	2.942	3.042	3.162	3.256	3.366	3.462
0.658	1.000	1.200	1.415	1.588	1.756	1.918	2.060	2.222	2.352
2.490	2.620	2.752	2.870	2.988	3.088	3.208	3.304	3.424	3.518
0.658	1.000	1.200	1.420	1.594	1.768	1.934	2.080	2.246	2.376
2.520	2.660	2.786	2.910	3.034	3.134	3.254	3.352	3.482	3.574
0.658	1.000	1.200	1.425	1.600	1.780	1.950	2.100	2.270	2.400
2.550	2.700	2.820	2.950	3.080	3.180	3.300	3.400	3.540	3.630
0.658	1.004	1.206	1.432	1.610	1.794	1.968	2.120	2.292	2.426
2.578	2.728	2.849	2.980	3.114	3.216	3.340	3.440	3.580	3.672
0.658	1.008	1.212	1.439	1.620	1.808	1.986	2.140	2.314	2.452
2.606	2.756	2.878	3.010	3.148	3.252	3.380	3.480	3.620	3.714
0.658	1.012	1.218	1.446	1.630	1.822	2.004	2.160	2.336	2.478
2.634	2.784	2.907	3.040	3.182	3.288	3.420	3.520	3.660	3.756
0.658	1.016	1.224	1.453	1.640	1.836	2.022	2.180	2.358	2.504
2.662	2.812	2.935	3.070	3.216	3.324	3.460	3.560	3.700	3.798
0.658	1.020	1.230	1.460	1.650	1.860	2.040	2.200	2.380	2.530
2.690	2.840	2.965	3.100	3.250	3.360	3.500	3.600	3.740	3.840
0.658	1.025	1.240	1.480	1.675	1.878	2.070	2.240	2.425	2.575
2.735	2.890	3.023	3.160	3.310	3.425	3.565	3.670	3.810	3.910
0.658	1.030	1.250	1.500	1.700	1.906	2.100	2.280	2.470	2.620
2.780	2.940	3.081	3.220	3.370	3.490	3.630	3.740	3.880	3.980
0.658	1.030	1.280	1.500	1.730	1.950	2.150	2.330	2.500	2.650
2.830	3.000	3.150	3.300	3.400	3.550	3.680	3.800	3.920	4.030
0.658	1.050	1.300	1.540	1.780	2.000	2.[illegible]00	2.400	2.550	2.700
2.830	3.030	3.180	3.300	3.450	3.600	[illegible].700	3.830	3.950	4.080
0.658	1.080	1.340	1.580	1.820	2.040	2.240	2.440	2.590	2.740
2.870	3.070	3.220	3.340	3.490	3.640	3.740	3.870	3.990	4.110
0.658	1.100	1.370	1.620	1.870	2.100	2.280	2.480	2.630	2.780
2.910	3.110	3.260	3.380	3.530	3.680	3.780	3.910	4.030	4.150
0.658	1.130	1.420	1.680	1.910	2.140	2.320	2.520	2.670	2.820

```
3.000 3.150 3.300 3.450 3.570 3.710 3.850 3.950 4.080 4.200
0.658 1.130 1.440 1.700 1.950 2.150 2.350 2.540 2.700 2.900
3.050 3.200 3.350 3.500 3.620 3.750 3.900 4.000 4.100 4.250
0.658 1.180 1.490 1.750 2.000 2.200 2.400 2.600 2.770 2.950
3.100 3.250 3.400 3.550 3.700 3.810 3.950 4.070 4.180 4.300
00.00
02.76 04.14 05.52 06.90 08.28 09.66 11.04 12.42 13.80 15.18
16.56 17.94 19.32 20.70 22.08 23.46 24.84 26.22 27.60 34.50 41.40 56.20      Z
464.0
196.0 170.0 149.5 136.3 127.5 120.0 114.5 109.0 105.0 101.0
097.0 092.5 090.0 087.0 082.5 080.0 077.0 075.0 070.0 065.0 055.0 052.0
464.0
237.5 201.0 176.0 161.0 150.0 140.0 133.0 127.0 122.0 117.5
113.0 109.0 105.0 102.0 098.7 095.0 092.5 090.0 087.5 080.0 070.0 062.0
464.0
272.5 230.0 202.5 185.0 172.5 162.5 153.8 146.3 141.0 136.3
132.5 127.5 125.0 121.3 118.0 115.0 113.0 111.0 109.0 100.0 092.0 072.5
464.0
317.5 272.5 240.0 220.0 205.0 193.8 185.0 177.5 171.0 165.0                  TEMP
160.0 156.3 152.5 148.8 145.0 142.5 140.0 137.5 135.0 125.0 117.5 106.5
464.0
237.5 290.0 257.5 237.5 222.5 210.0 200.0 192.5 185.0 180.0
175.0 170.0 166.3 162.5 160.0 156.5 153.8 151.0 149.0 138.0 130.5 119.0
464.0
355.0 337.5 277.5 255.0 240.0 225.0 215.0 207.0 199.0 192.5
187.5 183.0 180.0 175.0 172.5 169.0 166.3 163.8 161.0 151.3 143.0 130.0
464.0
370.0 330.0 295.0 273.0 255.0 242.5 231.0 222.5 215.0 209.0
203.8 199.0 195.0 191.0 187.5 183.5 181.0 179.0 176.0 165.0 157.5 145.0
464.0
390.0 347.5 315.0 290.0 272.5 260.0 250.0 240.0 232.5 225.0
220.0 213.5 209.0 205.0 201.0 197.5 194.0 192.0 189.0 178.2 170.0 156.5
464.0
400.0 357.5 327.5 305.0 286.3 272.5 260.0 251.3 243.5 237.5
231.0 226.0 221.0 217.0 212.5 209.0 206.0 203.0 200.0 188.5 180.0 167.0
480.0
405.0 375.0 340.0 315.0 295.0 280.5 270.0 206.5 253.0 247.0
241.3 236.0 231.3 227.5 223.7 220.0 217.0 213.7 211.0 198.8 190.0 177.5
490.
410.0 380.0 350.0 322.5 302.5 288.8 277.5 270.0 262.0 255.0
249.0 244.6 240.0 235.5 232.0 228.6 225.0 222.5 220.0 208.5 199.5 187.5
500.
415.0 385.0 360.0 330.0 310.0 295.0 285.0 277.0 270.0 263.0
257.5 252.5 247.5 243.0 239.0 236.3 233.0 230.0 228.0 217.0 208.0 196.0
510.
420.0 390.0 370.0 340.0 317.5 302.0 290.0 282.5 275.0 270.0
264.5 259.0 255.0 250.0 247.0 243.0 240.8 238.7 236.0 225.0 217.0 205.0
520.
425.0 395.0 380.0 350.0 325.0 308.7 297.0 287.5 280.0 275.0
270.0 265.0 261.2 257.0 253.0 250.0 247.5 245.0 243.0 232.5 225.0 214.0
0.0 0.2 0.4 0.6 0.8 0.9 1.0
0.5300 0.5450 0.5530 0.5450 0.5380 0.4440 0.1460                             ECCR
0.5300 0.5430 0.5486 0.5390 0.5316 0.4368 0.1432
0.5300 0.5410 0.5442 0.5330 0.5252 0.4296 0.1404
0.5300 0.5390 0.5398 0.5270 0.5188 0.4224 0.1376
0.5300 0.5370 0.5354 0.5210 0.5124 0.4152 0.1348
0.5300 0.5350 0.5310 0.5150 0.5060 0.4080 0.1320
0.5300 0.5330 0.5266 0.5090 0.4996 0.4008 0.1292
0.5300 0.5310 0.5222 0.5030 0.4932 0.3936 0.1264
0.5300 0.5293 0.5183 0.4974 0.4868 0.3864 0.1232                             FKE
0.5300 0.5279 0.5149 0.4923 0.4804 0.3792 0.1196
0.5300 0.5266 0.5116 0.4872 0.4740 0.3720 0.1160
0.5300 0.5232 0.5032 0.4744 0.4580 0.3540 0.1070
0.5300 0.5198 0.4948 0.4616 0.4420 0.3360 0.0980
0.5300 0.5130 0.4780 0.4360 0.4100 0.3000 0.0800
0.5300 0.5078 0.4668 0.4196 0.3780 0.2680 0.0740
0.5300 0.4988 0.4460 0.3880 0.3170 0.2110 0.0620
0.5300 0.4940 0.4300 0.3600 0.2650 0.1750 0.0500
0.5300 0.4830 0.4000 0.3150 0.2100 0.1350 0.0200
0.5300 0.4750 0.3800 0.2800 0.1900 0.1130 0.0100
0.5300 0.4380 0.3100 0.2350 0.1220 0.0630 0.0050
0.0 0.4 0.8 1.2 1.6 2.4 3.2 3.6 4.0
1.60 2.10 2.25 2.27 2.26 2.20 2.00 2.00 2.10                                 ELOL
1.65 2.15 2.32 2.39 2.38 2.33 2.19 2.20 2.30
```

POP

```
1.70 2.20 2.40 2.50 2.50 2.45 2.38 2.39 2.50
1.74 2.25 2.50 2.60 2.62 2.61 2.62 2.65 2.78
1.78 2.30 2.60 2.70 2.74 2.76 2.85 2.90 3.05
1.89 2.40 2.70 2.85 2.97 3.01 3.10 3.20 3.40
2.00 2.50 2.80 3.00 3.20 3.25 3.35 3.50 3.75
2.33 2.93 3.23 3.40 3.58 3.75 3.98 4.13 4.43
2.65 3.35 3.65 3.80 3.95 4.25 4.60 4.76 5.10
3.20 3.90 4.00 4.55 4.75 5.10 5.60 5.90 6.25
3.75 4.55 5.00 5.30 5.65 6.25 6.75 7.10 7.50
```

C

```
0.0 0.2 0.4 0.6 0.8 1.0
```

EPSI

```
90.10 75.00 60.40 46.40 29.80
90.02 75.00 60.48 46.48 29.96
89.94 75.00 60.56 46.56 30.12
89.86 75.00 60.64 46.64 30.28
89.78 75.00 60.72 46.72 30.44
89.70 75.00 60.80 46.80 30.60
89.62 75.00 60.88 46.88 30.76
89.54 75.00 60.96 46.96 30.92
89.46 75.00 61.04 47.04 31.08
89.38 75.00 61.12 47.12 31.24
89.30 75.00 61.20 47.20 31.40
89.10 75.00 61.40 47.40 31.80
88.90 75.00 61.60 47.60 32.20
88.50 75.00 62.00 48.00 33.00
87.80 74.90 62.10 48.25 33.60
86.40 74.70 62.30 48.75 34.80
85.00 74.50 62.50 49.25 36.00
82.85 73.40 62.40 50.21 36.66
80.68 72.32 62.32 51.17 37.32
72.00 68.00 62.00 55.00 40.00
```

EPSLON

```
0.0 0.2 0.3 0.4 0.5 0.6 0.7 0.8 1.0
```

QU

```
.0000.3880.5700.7440.9100.9440.9700.9700.9360
.0000.3856.5640.7328.8920.9328.9640.9640.9232
.0000.3832.5580.7216.8740.9216.9580.9580.9104
.0000.3808.5520.7104.8560.9104.9520.9520.9876
.0000.3784.5460.6992.8380.8992.9460.9460.8848
.0000.3760.5400.6880.8200.8880.9400.9400.8720
.0000.3736.5340.6768.8020.8768.9340.9340.8592
.0000.3712.5280.6656.7840.8656.9280.9280.8464
.0000.3672.5216.6560.7704.8556.9198.9192.8294
.0000.3616.5148.6480.7612.8468.9094.9076.8082
.0000.3560.5080.6400.7520.8380.8990.8960.7870
.0000.3420.4910.6200.7290.8160.8730.8670.7340
.0000.3280.4740.6000.7060.7940.8470.8380.6810
.0000.3000.4400.5600.6600.7500.7950.7800.5750
.0000.2920.4200.5300.6200.7000.7390.7200.5300
.0000.2760.3800.4700.5400.6000.6270.6000.4400
.0000.2600.3400.4100.4600.5000.5150.4800.3500
.0000.1900.2600.3100.3500.3550.3500.3250.2350
.0000.1400.1900.2250.2350.2250.2050.1750.0800
```

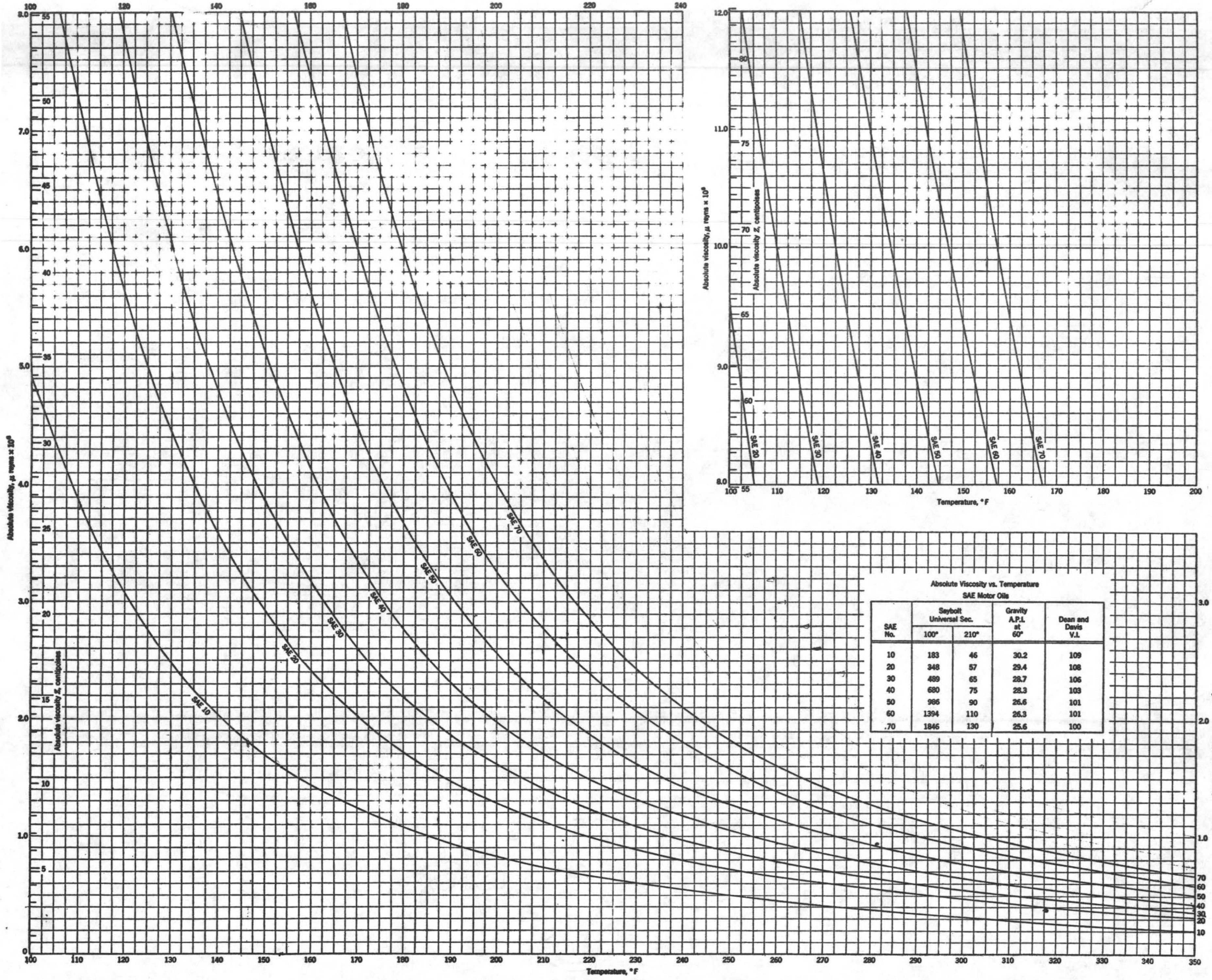
TTEMP

```
070. 100. 110. 120. 130. 140. 150. 170. 190. 210. 230. 260. 300. 350. 464.0 999.
```

ZZ

```
063.00 034.00 027.00 021.30 017.20 014.00 011.73
008.63 006.42 005.00 004.14 003.00 002.00 001.38
100.00 066.00 050.40 038.60 037.00 024.80 020.15
014.00 010.00 007.80 006.00 004.62 003.31 002.14
120.00 085.00 069.00 053.80 042.00 033.00 026.70
018.00 012.90 009.66 007.52 005.45 003.79 002.48
140.00 105.00 089.00 073.20 057.20 044.50 035.20
023.00 015.88 011.73 009.10 006.56 004.48 002.90
160.00 125.00 109.00 093.00 075.50 061.00 049.00
030.85 021.00 015.00 011.18 007.94 005.32 003.45
180.00 145.00 129.00 113.00 095.50 079.40 064.50
041.40 027.00 019.00 014.28 009.66 006.35 004.13
200.00 165.00 149.00 133.00 115.50 099.40 081.80
051.20 033.48 023.00 016.83 011.03 007.25 004.62
```

(See subroutine UREAL, page 62, for definitions).



Absolute Viscosity vs. Temperature

SAE Motor Oils

SAE No.	Saybolt Universal Sec. 100°	Saybolt Universal Sec. 210°	Gravity A.P.I. at 60°	Dean and Davis V.I.
10	183	46	30.2	109
20	348	57	29.4	108
30	489	65	28.7	106
40	680	75	28.3	103
50	986	90	26.6	101
60	1394	110	26.3	101
.70	1846	130	25.6	100

Appendix 2

Appendix 3

Data of Fig. 4-2

	m	1	5	10	15	20	25	30	35	40
	L/D \ ϵ	0	0.800	0.900	0.933	0.950	0.960	0.967	0.971	0.975
C_n	0.1	0	0.023	0.114	0.230	0.400	0.560	0.730	0.910	0.104
	0.2	0	0.053	0.228	0.455	0.076	1.030	1.290	1.540	1.72
	0.3	0	0.086	0.338	0.660	1.050	1.400	1.710	1.990	2.21
	0.4	0	0.121	0.450	0.850	1.300	1.680	2.020	2.280	2.52
	0.5	0	0.161	0.551	1.000	1.470	1.860	2.210	2.460	2.71
	0.6	0	0.195	0.625	1.100	1.560	1.960	2.290	2.560	2.80
	0.8	0	0.251	0.712	1.200	1.670	2.070	2.400	2.650	2.89
	1.0	0	0.293	0.778	1.270	1.760	2.140	2.480	2.730	2.96
	1.2	0	0.320	0.825	1.330	1.820	2.220	2.540	2.800	3.03
	1.5	0	0.349	0.892	1.400	1.900	2.300	2.630	2.880	3.11
	2.0	0	0.380	0.940	1.470	1.980	2.380	2.700	2.950	3.19
	∞	0	0.484	1.084	1.638	2.154	2.540	2.844	3.096	3.32
T'	0.1	0.658	1.110	1.562	1.935	2.270	2.570	2.860	3.110	3.35
	0.2	0.658	1.125	1.613	2.030	2.410	2.750	3.080	3.360	3.63
	0.3	0.658	1.142	1.662	2.110	2.530	2.890	3.240	3.540	3.83
	0.4	0.658	1.160	1.710	2.190	2.620	2.990	3.350	3.670	3.97
	0.5	0.658	1.181	1.754	2.240	2.680	3.060	3.420	3.730	4.04
	0.6	0.658	1.203	1.786	2.280	2.720	3.100	3.460	3.770	4.08
	0.8	0.658	1.232	1.829	2.330	2.770	3.140	3.500	3.820	4.11
	1.0	0.658	1.252	1.858	2.360	2.800	3.180	3.530	3.850	4.14
	1.2	0.658	1.267	1.881	2.380	2.820	3.200	3.550	3.870	4.16
	1.5	0.658	1.284	1.104	2.410	2.850	3.230	3.580	3.900	4.19
	2.0	0.658	1.301	1.931	2.440	2.880	3.270	3.620	3.930	4.23
	∞	0.658	1.355	1.998	2.511	2.948	3.328	3.673	3.990	4.285

From Dennison [9]

Appendix 4

Oil requirements factor K_e*

	0.9	0.8	0.6	0.4	0.2
0.25	0.390	0.490	0.500	0.520	0.530
0.50	0.300	0.410	0.436	0.478	0.513
0.75	0.220	0.330	0.395	0.450	0.500
1.00	0.175	0.265	0.360	0.430	0.494
1.25	0.150	0.230	0.335	0.415	0.488
1.50	0.135	0.210	0.315	0.400	0.483
2.00	0.113	0.190	0.280	0.380	0.475
2.50	0.100	0.175	0.260	0.368	0.465
3.00	0.088	0.158	0.250	0.340	0.455
4.00	0.063	0.122	0.235	0.310	0.438

From Fuller [11]

* This is the oil flow coefficient required to determine the quantity of oil that has to be continuously supplied to the bearing which is determined by [11]:

$$Q_e = 0.0272\ K_e.R.N.L.C \qquad \text{gal/min.}$$

where;

R is the journal radius, in.

N is the shaft revs/min.

L is the bearing length, in.

C is the radial clearance, in.

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