

CALCULATION METHODS FOR STEADILY LOADED CENTRAL CIRCUMFERENTIAL GROOVE HYDRODYNAMIC JOURNAL BEARINGS

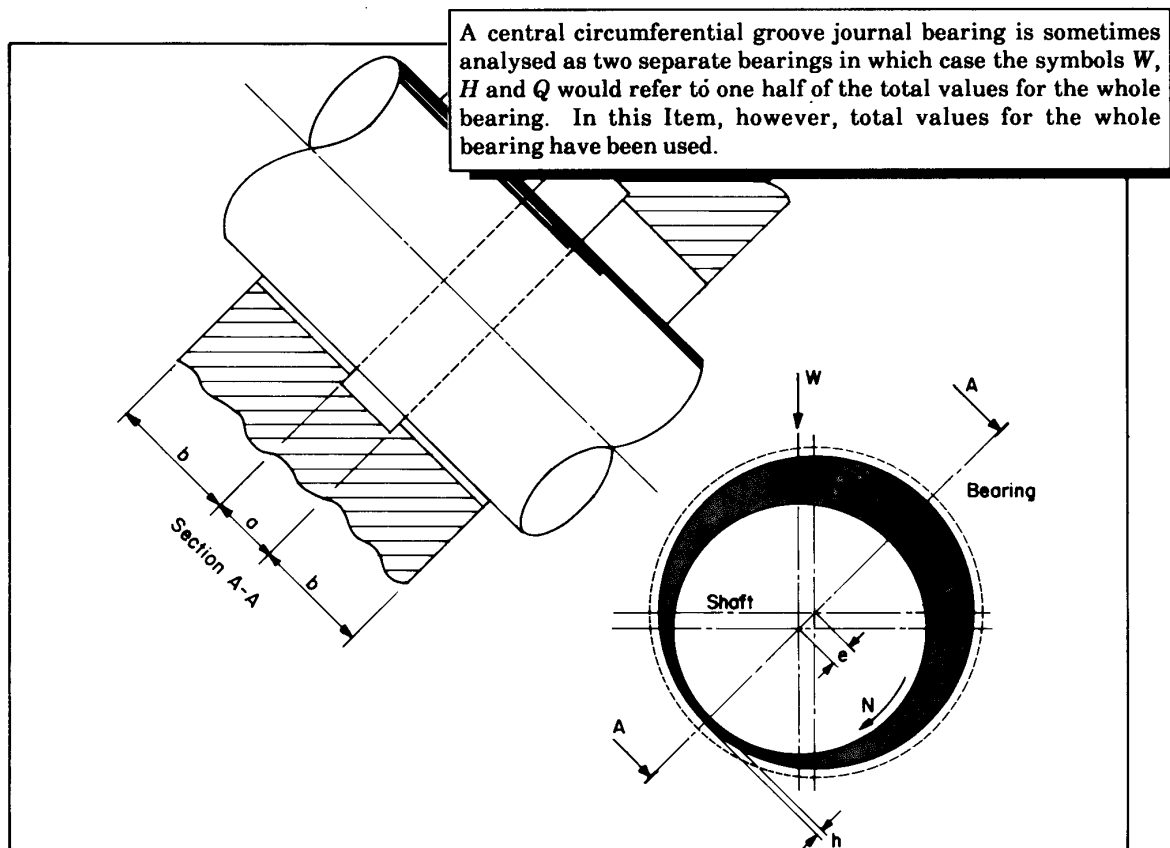
1. NOTATION AND UNITS

The International System of units is given in this Item. Any other coherent system of units may also be used.

a	axial width of lubricant groove	m
b	axial width of one land of bearing	m
C	specific heat capacity of lubricant	J/kg K
C_H	non-dimensional power loss correction	
C_ε	eccentricity ratio correction	
c_d	diametral clearance under running conditions	m
d	diameter of journal	m
e	eccentricity for an aligned bearing	m
g	acceleration due to gravity	m/s ²
H	total power loss of bearing	W
H'	non-dimensional power loss = $\frac{H}{\eta_e N^2 d^2 b} \left(\frac{c_d}{d}\right)$	
H'_0	uncorrected non-dimensional power loss	
h	minimum film thickness	m
h_s	safe allowable minimum film thickness	m
M	rotor mass associated with one bearing	kg
N	frequency of shaft rotation	rev/s
p_f	lubricant feed pressure	N/m ²
p'_f	non-dimensional feed pressure = $\frac{p_f}{\eta_f N} \left(\frac{c_d}{d}\right)^2$	
P_{run}	running specific loading = $W_{run}/(2 bd)$	N/m ²
P_{start}	start-up specific loading = $W_{start}/(2 bd)$	N/m ²
Q	total lubricant flow rate	m ³ /s
Q'	non-dimensional flow rate = $Q \eta_m / (p_f c_d^3)$	
R_a	surface roughness value	m
T_e	effective temperature of lubricant in bearing	°C
T_f	lubricant feed temperature	°C

Issued November 1990
With Amendment A, April 1995

T_{max}	maximum bearing surface temperature	°C
T_{out}	bulk temperature of lubricant at outlet from bearing	°C
W_{run}	total running load	N
W_{start}	total start-up load	N
W'	non-dimensional load = $\frac{W_{run}}{\eta_e N b d} \left(\frac{c_d}{d} \right)^2$	
β	angular misalignment	rad
γ	factor for oil grade	
δ	misalignment over full axial width of bearing	m
ε	eccentricity ratio for an aligned bearing $2e/c_d$	
ε_0	uncorrected eccentricity ratio	
η_e	effective dynamic viscosity of lubricant in bearing	N s/m ²
η_f	dynamic viscosity of lubricant at feed temperature	N s/m ²
η_m	dynamic viscosity for calculation of flow rate	N s/m ²
ρ	density of lubricant	kg/m ³



Sketch 1.1 Illustration of notation

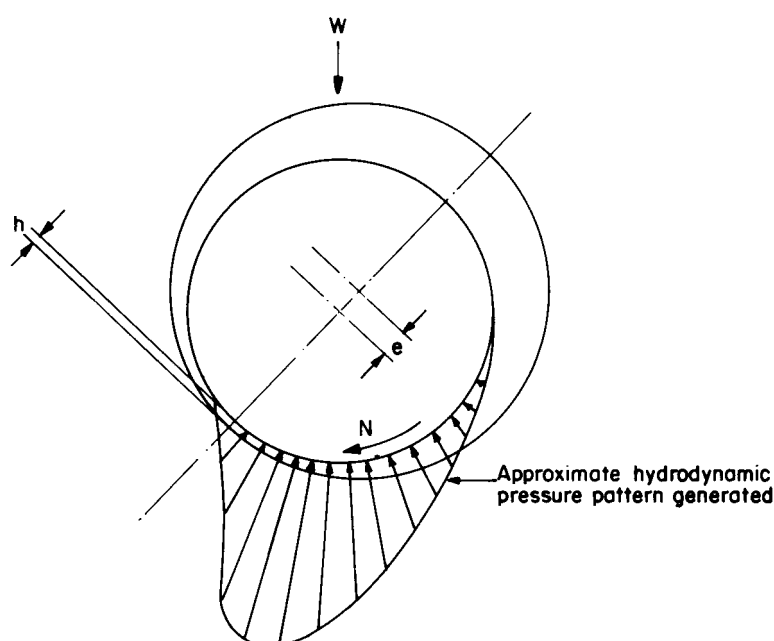
2. INTRODUCTION

This Item provides charts and calculation methods for the design of forced lubricated hydrodynamic journal bearings with a central circumferential groove. The Item may also be used to analyse the performance of existing bearing designs in failure analysis or to investigate the feasibility of uprating their performance. The bearings are considered to be steadily loaded, but bearings with slowly varying loads and speeds may also be analysed.

This Item is divided into three main sections. In the first of these sections a basic specification for the bearing is drawn up. Guidance is given on assigning preliminary values to some of the bearing parameters. In the second of these sections the performance of the bearing is predicted. Finally, limiting criteria are given to allow the adequacy of the bearing design to be assessed.

2.1 Description of Bearing

The operation of a hydrodynamic journal bearing depends upon a viscous lubricant being drawn into a wedge shaped space. The journal takes up an eccentric position within the bearing clearance under the influence of the applied load. The eccentricity adjusts itself until the load is balanced by the pressure generated in the lubricant film as shown in Sketch 2.1.



Sketch 2.1 Pressure distribution in a journal bearing

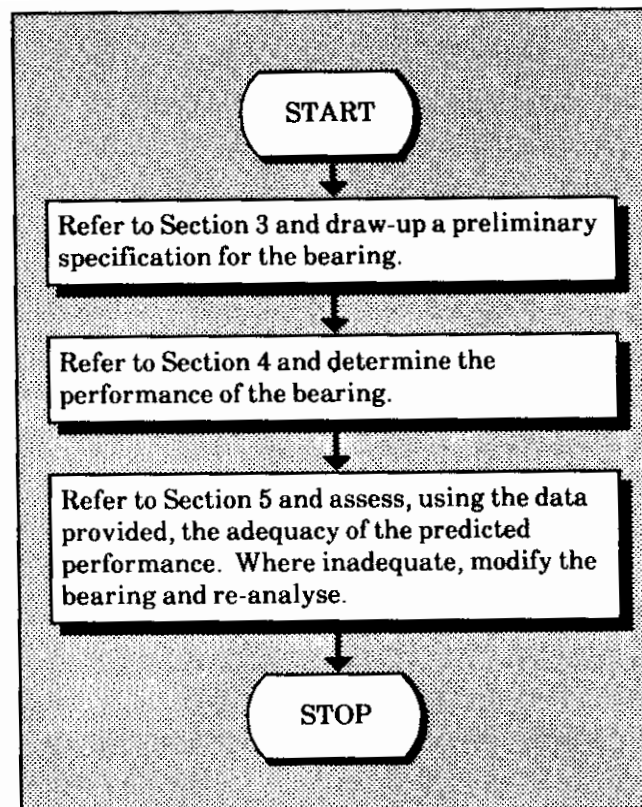
Oil lubricated bearings are usually lined with a material such as whitmetal (see Reference 1) and the journal is commonly manufactured from plain carbon steel. The bearing is often split along a diameter to facilitate assembly. This Item deals with bearings in which the lubricant is supplied via a central circumferential groove. Such a feed arrangement is commonly used where the direction of the load is indeterminate or where it varies by more than 180° during operation. The alternative arrangement of one or two axial grooves is dealt with in Reference 2.

2.2 Guide to the Use of This Item

In order to use this Data Item to design a central circumferential groove hydrodynamic journal bearing you should have available the following information.

- (i) The intended operating speed and loads.
- (ii) The shaft size, which may be preliminary at this stage.
- (iii) The viscosity/temperature characteristics of the lubricant (where the lubricant has been predetermined) and, if known, the feed temperature and pressure.

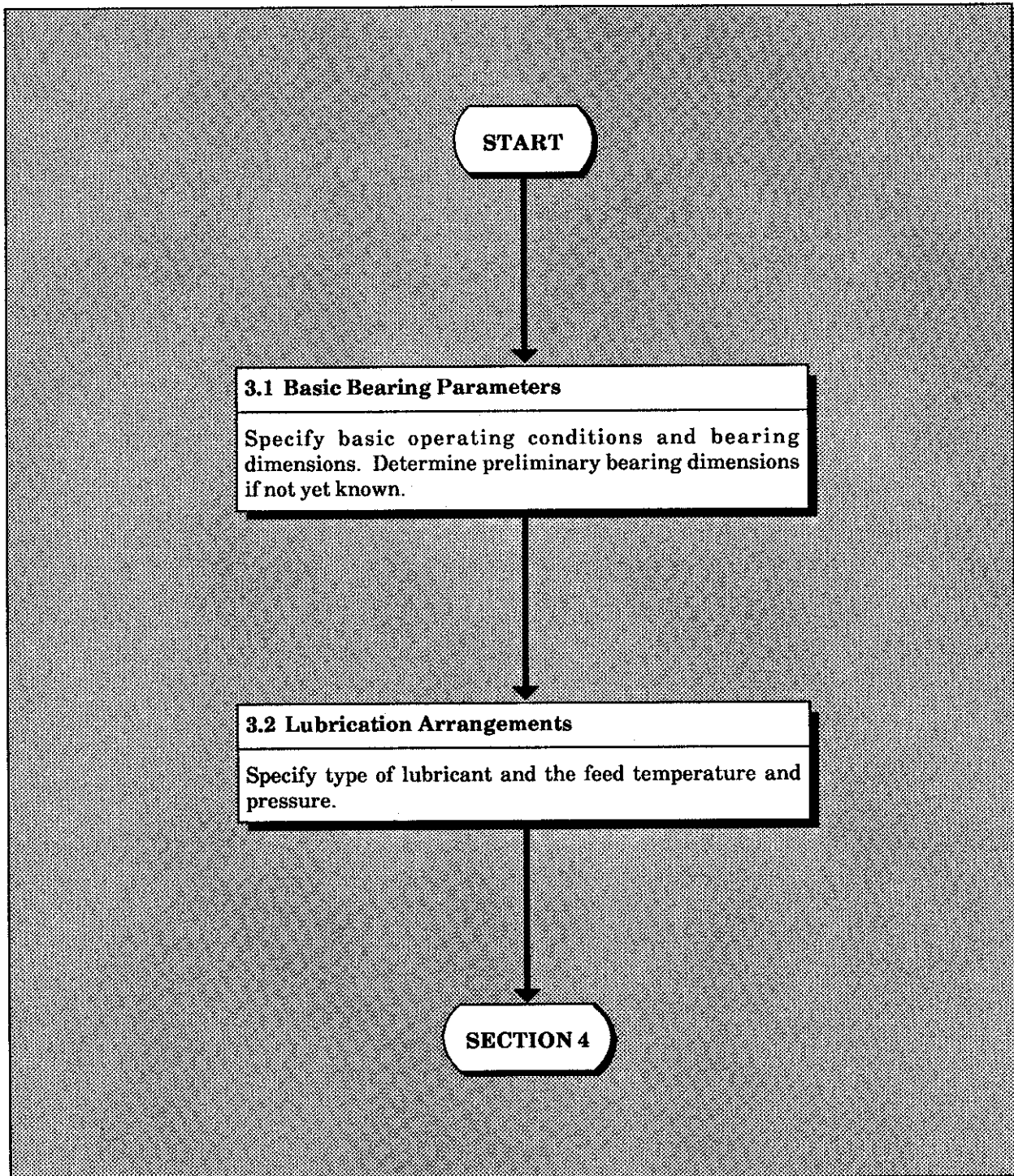
In order to then assess the adequacy of the design some additional information may be required.



A worked example illustrating the use of this Item is presented in Section 6.

3. SPECIFICATION OF BEARING

In this section a specification is drawn up for the bearing that is to be analysed. Where the bearing has not yet been designed, preliminary values are assigned to the basic bearing parameters which may later be revised. The procedure is summarised in Sketch 3.1.



Sketch 3.1 Preliminary values of bearing parameters

3.1 Basic Bearing Parameters

- **Load and Speed**

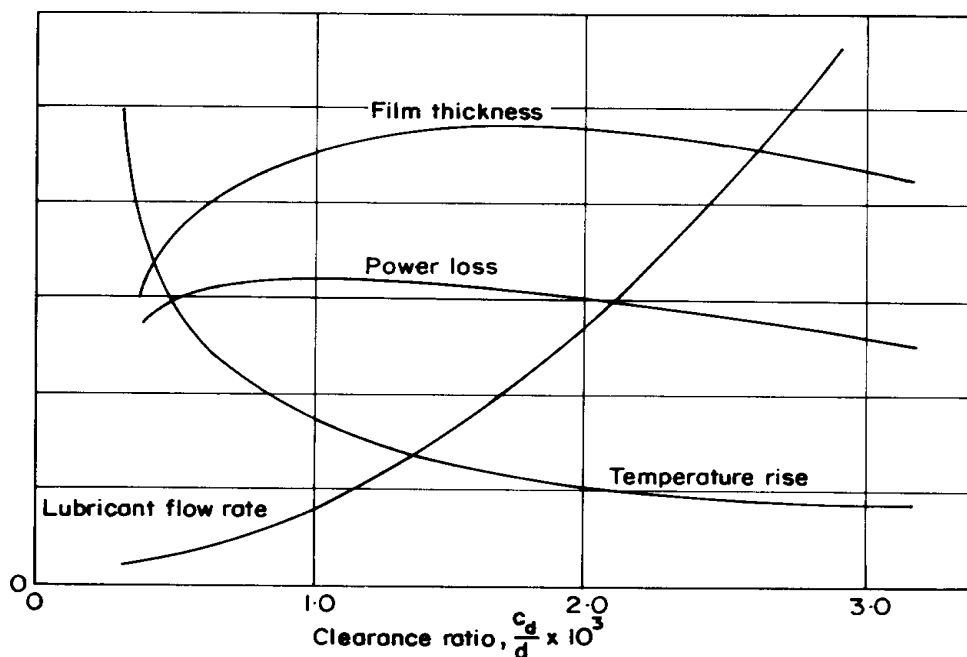
The load, W , and the frequency of rotation, N , must be specified assuming steady loading and a constant rotational speed. In specifying the rotational speed it should be noted that short overspeeds can usually be tolerated. However, sometimes “off design” loads and speeds occur that can give rise to more severe operating conditions. It is, therefore, advisable to investigate the operation of the bearing under all potentially critical conditions. High loads combined with low speeds may be critical with regard to film thickness whilst high loads combined with moderate to high speeds can be critical with regard to temperature. The starting load is also an important factor.

- **Bearing Overall Dimensions**

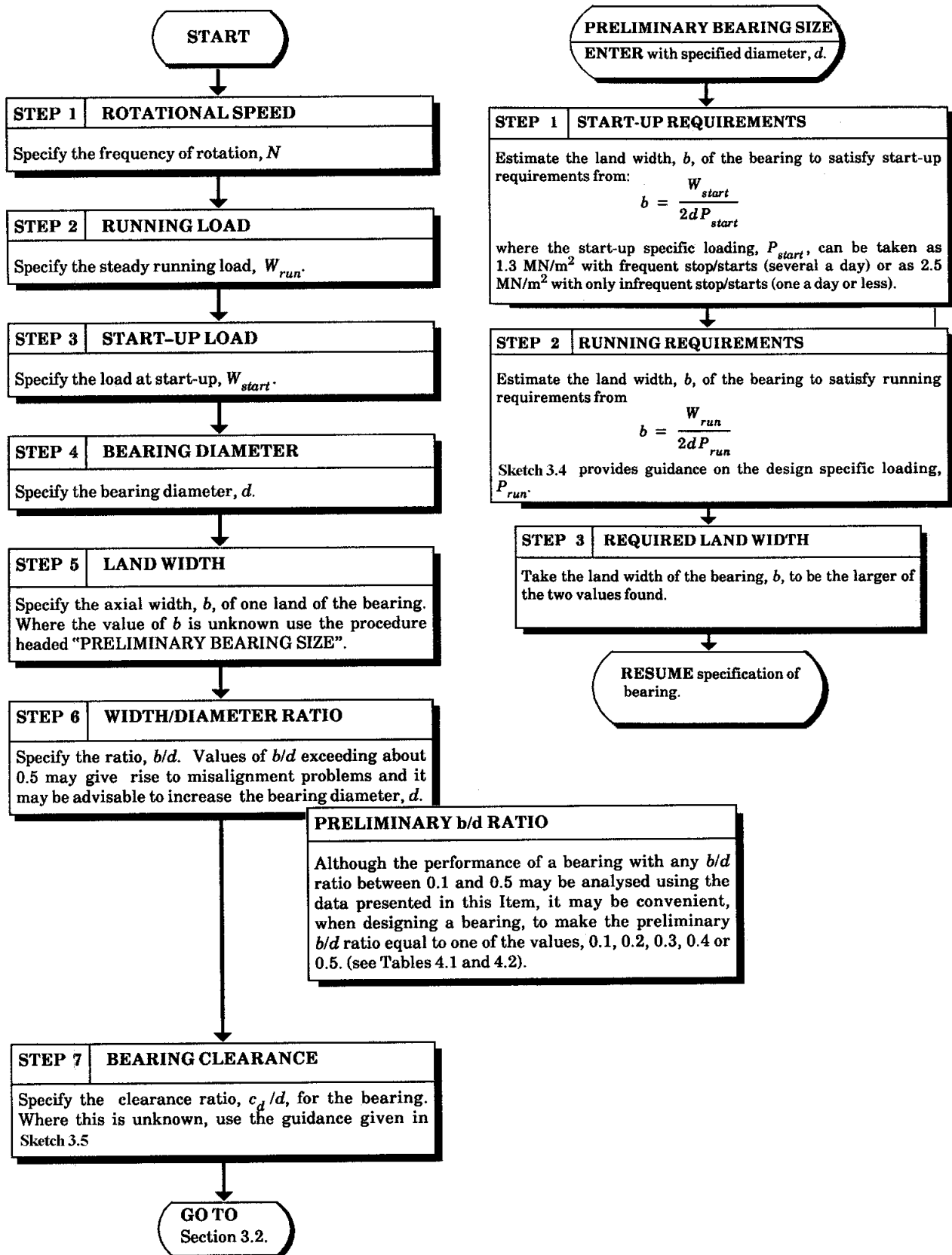
The bearing diameter, d , is usually predetermined by strength or stiffness requirements. However, the overall bearing size must be sufficient to satisfy both start-up and running requirements and so the bearing width will be dictated by either the start-up or the running requirement. A procedure for estimating the required bearing width where this is not known is given in Sketch 3.3.

- **Bearing Clearance**

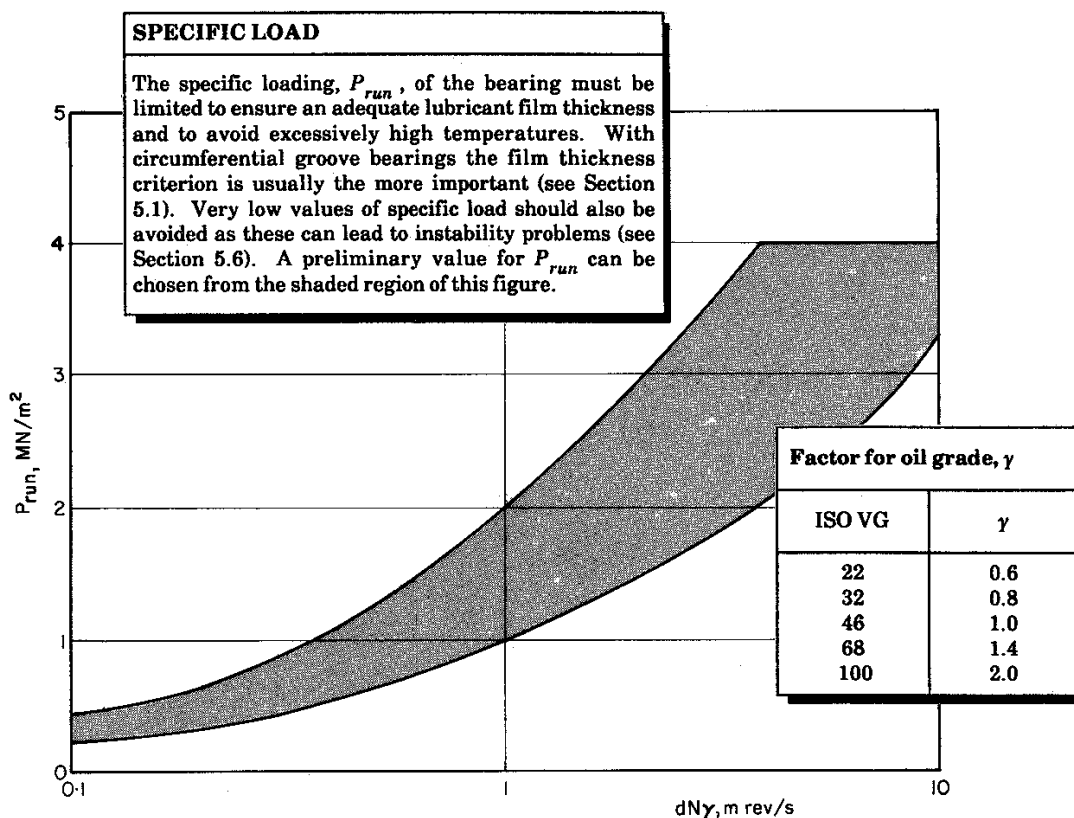
The bearing clearance under running conditions is important for the satisfactory operation of the bearing. Both small and large values of the clearance ratio, c_d/d , can give rise to inadequate lubricant film thicknesses. Small values of c_d/d can give rise to high bearing temperatures while large values can mean excessive lubricant flow rates as shown in Sketch 3.2. The diametral clearance must be held to close tolerance during manufacture.



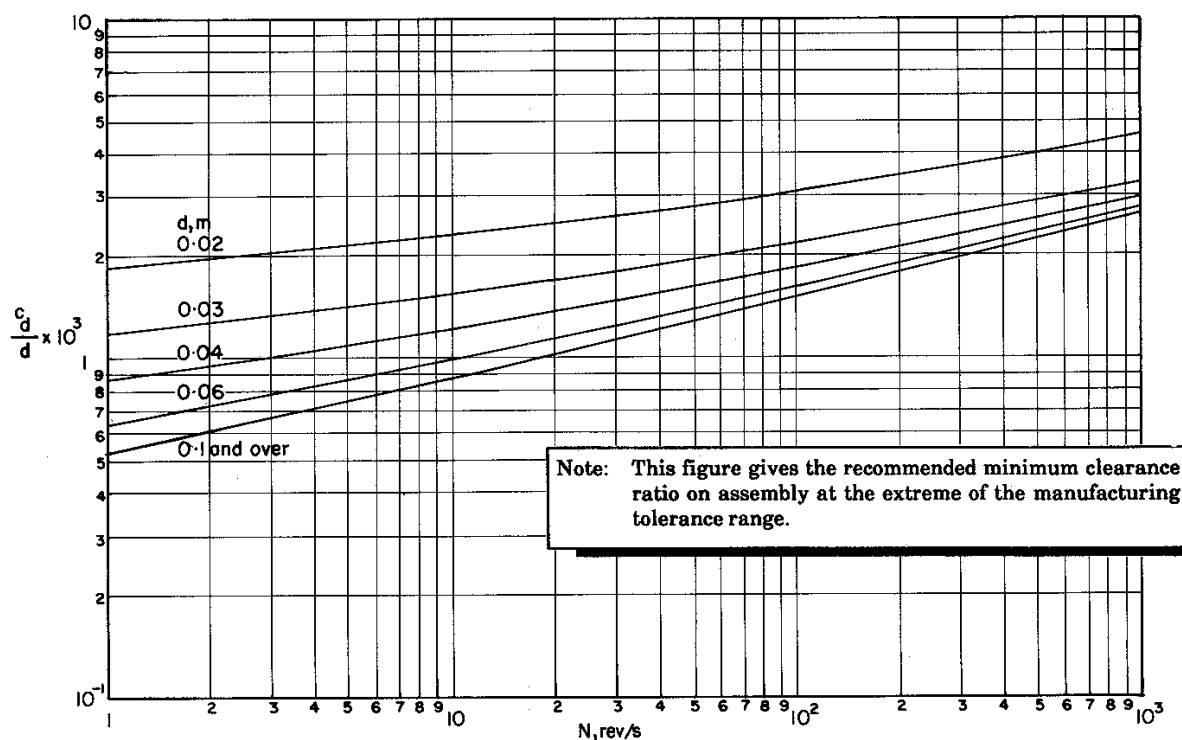
Sketch 3.2 Typical effects of varying the clearance ratio



Sketch 3.3 Procedure for estimation of bearing width



Sketch 3.4 Guide to design running specific load



Sketch 3.5 Recommended minimum clearance ratio

3.2 Lubrication Arrangements

A flowchart describing the lubrication arrangements is given in Sketch 3.6.

- Lubricant grade**

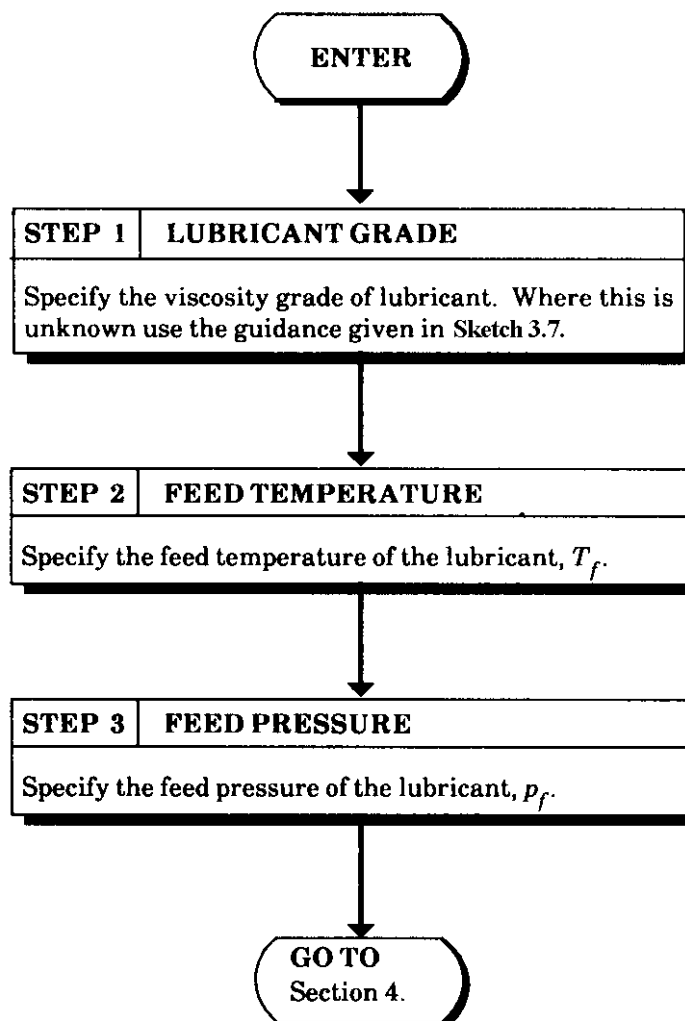
The lubricant is sometimes specified to suit the design requirements of another component. The guidance provide in Sketch 3.7 may be used where the designer has a free choice of lubricant.

- Feed temperature**

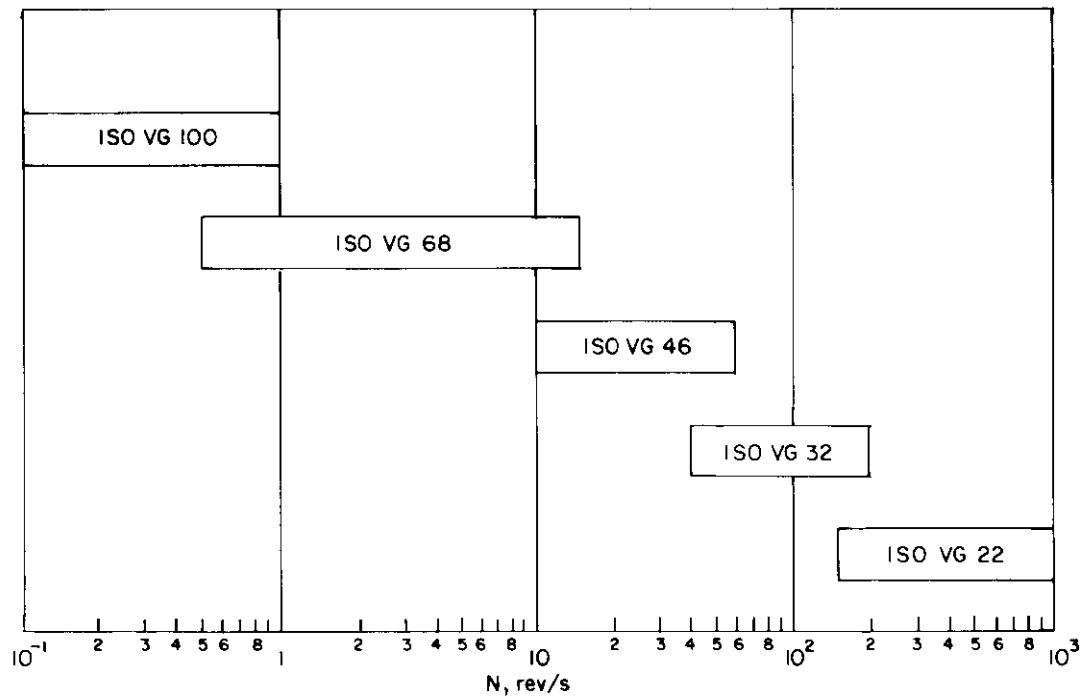
For the purposes of analysing the performance of the bearing it is necessary to assume a value for the temperature of the lubricant supplied to the bearings, T_f , since this is the datum used in calculating temperature rise. Typically, T_f would be between 40°C and 50°C for oil lubricated bearings.

- Feed pressure**

In most circumstances feed pressures between 0.07 MN/m² and 0.35 MN/m² will be found to be adequate.



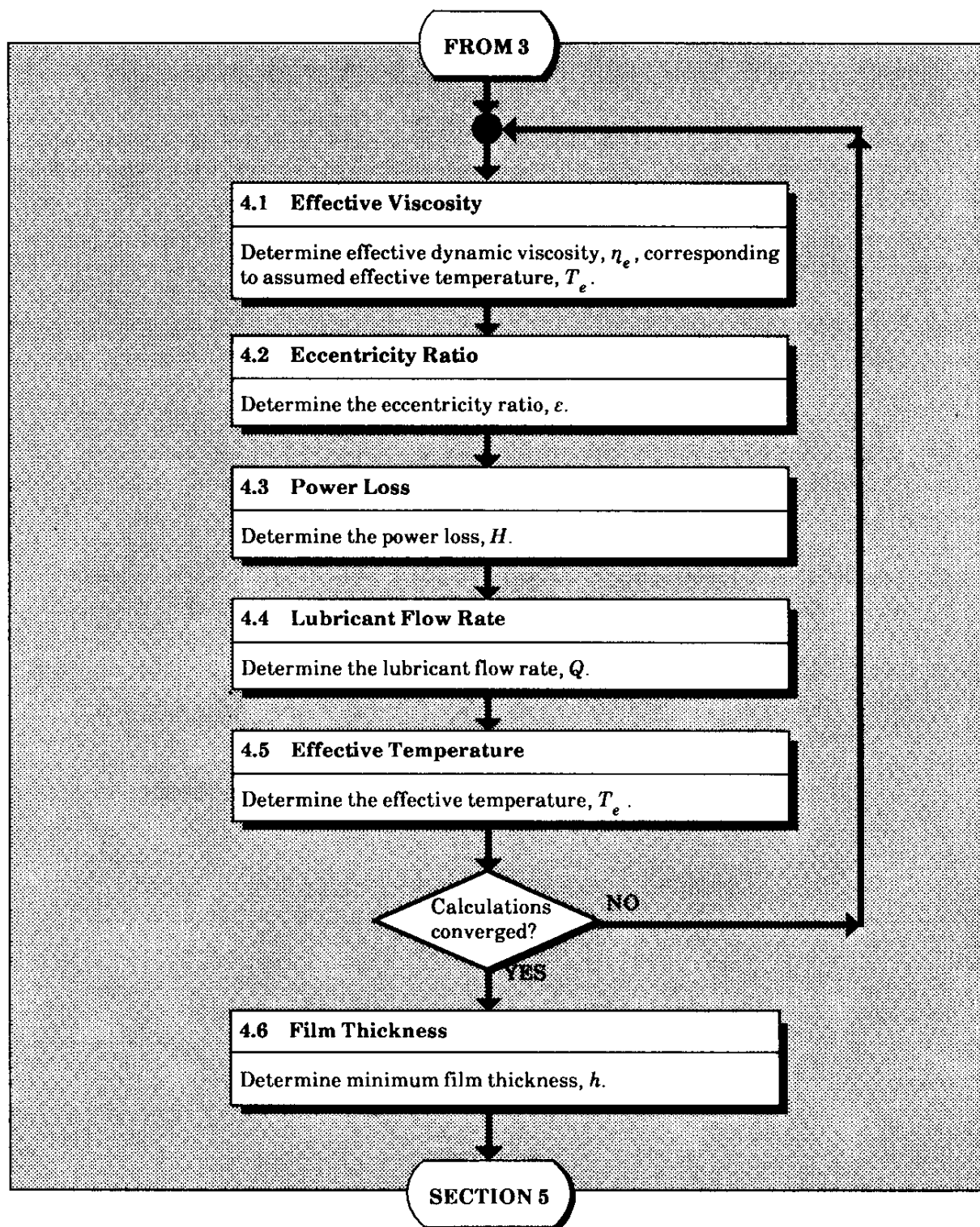
Sketch 3.6 Lubrication arrangements



Sketch 3.7 Guide to choice of oil grade

4. PREDICTION OF PERFORMANCE

This section gives guidance on predicting the performance of a proposed or existing design. The procedure to be followed is summarised in Sketch 4.1.



Sketch 4.1 Performance prediction procedure