TRIBOLOGICAL DESIGN GUIDE

Institution of MECHANICAL ENGINEERS

PART 1: BEARINGS

Tribology Group

The IMechE Tribology Group has produced this guide as Part 1 of a series of guides on Tribological Design which it wishes to make freely available for student use in connection with their studies. There are others on the Wear Analysis Process, Part 2 covers Lubrication, Part 3 discusses Contact Mechanics.

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Institution of MECHANICAL ENGINEERS

PART 1: BEARINGS

CONTENTS

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The Institution of Mechanical Engineers

2nd Edition

FOREWORD

The design of machines elements involves consideration of:

- Kinematic function
- Strength
- Mechanical efficiency
- Required life

Friction and wear directly affect mechanical efficiency and may also undermine kinematic function and strength to the point of premature failure. Wear directly limits life at acceptable performance level.

Tribological considerations in machine element design are no less important than considerations of kinematic function and strength.

Kinematics and strength are comprehensively covered as core subjects in the education and training of mechanical engineers and are commonly addressed in the practice of Engineering Design. The subject of Tribology is much more variably covered and, in consequence, tribological considerations are often overlooked in the subject of Design.

In view of its importance, the Tribology Group of the Institution of Mechanical Engineers is anxious to encourage the inclusion of tribological considerations in the practice of Design in the education of mechanical engineers. To this end, the Tribology Group has prepared a collection of Tribological Design Guides to offer to students of engineering in connection with their design studies. The hope is that, by making such data readily available, awareness in tribological design will be encouraged. The data presented will not, of itself, permit complete tribological design but references are included to more comprehensive sources of data and detailed design procedures.

It is the hope of the Tribology Group that those involved with the education of mechanical engineers will find it useful to reproduce this document for distribution to students or for incorporation into their own in-house produced Design Data Handbooks.

BEARING CONFIGURATIONS

Journal Bearing

Provides radial location for a rotating shaft – carries radial load.

Thrust Bearing

Provides axial location for a rotating shaft – carries axial load.
Radial and axial location capability can be combined in a single bearing.

Slider Pad Bearing

Supports a load perpendicular to a continuous plane surface along which the pad moves (usually in reciprocating motion).

BEARING TYPES (according to operating mechanism)

Rolling Element

The mating surfaces are separated by balls or rollers which transmit the load. The bearing is a bought-in unit comprising a pair of races and a caged (usually) set of rolling elements.

Hydrostatic (externally pressurised)

The mating surfaces are separated by a fluid film. Separation is maintained by supplying fluid at high pressure. Separation is not dependent upon relative motion of the mating surfaces.

Oil Impregnated (porous metal)

The mating surfaces are partially separated by an oil film supplied from a reservoir of oil within the pores of in sintered metal bearing. The bearing is a bought-in unit.

Dry Rubbing

The mating surfaces are in direct contact. The surfaces may be dry (un-lubricated) or a lubricant may be present but unable to maintain complete separation of the surfaces.

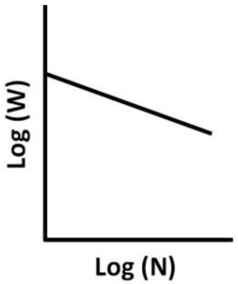
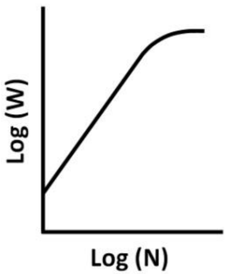
BEARING CHARACTERISTICS

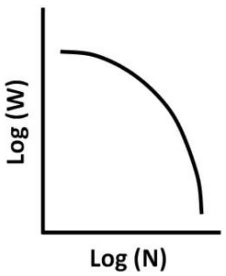
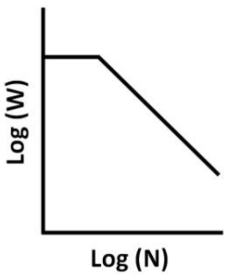
The principle characteristics and relative merits of the different bearing types are listed below and in Table 1.

Load-Speed Characteristics

Rolling element bearings:	Bearing life very sensitive to load. For given life, permissible load decreases with speed.
Hydrodynamic bearings:	Separating film thickness decreases with load but increases with speed. For given film thickness, permissible load increases with speed.
Hydrostatic bearings:	Permissible load unaffected by speed until speed limitation associated with recess cavitation.
Oil impregnated bearings and Dry rubbing bearings:	Frictional heating increases with both load and speed. Temperature limitation means that permissible load decrease with speed. At low speed, load limitation is determined by crushing strength of bearing material.

Table 1: Bearing Characteristics

Bearing type	Load-Speed Characteristic	Limitations	Advantages	Disadvantages
Rolling element		Life limited by fatigue of races and rolling elements.	Excellent availability of standard bearings. Low starting and running friction. Small axial extent. Combined radial and axial load capability configurations available. Simple lubrication system with grease packed bearing.	Limited but predicable life. Vulnerable to shock loading. Requires special stabilized material for operating temperature above 125°C. Large sizes very expensive.
Hydrodynamic – oil		Separating film depends upon sliding motion.	Unlimited life so long as oil film is maintained	Oil circulation system usually required
Hydrodynamic – gas		As with oil.	Ambient fluid (air) may be used. Wide temperature range. Gas viscosity increases with temperature.	Small load capacity due to low viscosity of gases.

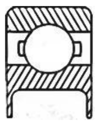
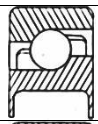
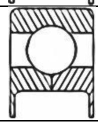
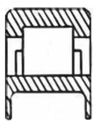
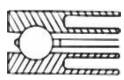
Bearing type	Load-Speed Characteristic	Limitations	Advantages	Disadvantages
Hydrostatic	Externally pressurised film – independent of motion.	Limited overload capacity. Failure of supply leads to immediate failure of film.	Film present under starting and stopping conditions. Zero friction when stationary. Very high stiffness possible.	High pressure supply system required.
Oil impregnated		Operating temperature – oil degrades if too high. No oil flow through bearing to carry away heat.	Simplicity. Good availability of standard bushes.	Low load capacity at high speed because of temperature limitation.
Dry rubbing		Continuous wear. Life limited by permissible wear. Operating temperature limited by softening or thermal breakdown.	Ultimate simplicity. Commercial materials readily available.	High friction. Low load capacity at high speed – frictional heating limitation. Loss of location due to wear.

ROLLING BEARINGS

Introduction

Rolling bearings are available in a range of standard sizes having rolling elements in the form of ball, cylindrical roller, spherical roller or tapered roller and are capable of carrying radial only, axial only or combined radial and axial loading. The relative capabilities of the most common forms of bearing are listed in Table 1. The catalogues of the leading rolling bearing manufacturers give full information concerning the range of bearing geometries, their selection and use for particular requirements and detailed dimensional (tolerance) constraints for full realisation of bearing potential performance.

<u>Table 2: Relative Capabilities</u>

Bearing type	**Radial load capacity**	**Axial load capacity**	**Speed capacity**	**Allowable misalignment (rad)**	**Bearing section**
Ball Deep-groove	Light/ medium	Light/ medium	High	0.001	
Ball Self-aligning	Light/ medium	Light	High	0.04	
Ball Angular contact	Medium (must have axial load)	Medium/ heavy	Medium/ high	0.0005	
Ball Duplex	Light (must have axial load)	Medium	Medium	0.0002	
Cylindrical roller	Heavy	None – very light with lip	Medium/ high	0.001	
Tapered roller	Heavy (must have axial load)	Medium/ heavy	Medium	0.0005	
Spherical roller Double Row	Very heavy	Light/ medium	Low/ medium	0.04	
Ball Thrust	None	Light/ medium	Low/ medium	None	

Design Considerations

Designing with rolling bearings requires the following:

- Selection of bearing to satisfy load, speed and life requirements
- Specification of dimensional tolerances for mating parts

- Provision for appropriate lubrication

Bearing Life

The rolling elements and the races are subject to cyclic stress and bearing life is consequently limited by fatigue.
In common with all fatigue failure the life of bearings shows statistical variation. The load capacity data listed by bearing manufacturers relate to 90% probability of attaining a notional life of one million revolutions. This load capacity is referred to as the basic load rating C. The actual life to be expected for a bearing depends upon the actual load applied to the bearing and may be obtained from the following life equation:

$$L = (C/P)^p$$

where L = basic rating life (in millions of revolutions)
P = *equivalent* bearing load
p = 3 for ball bearings, 10/3 for roller

The 90% probability life so calculated is referred to as the L_{10} life. The average life (i.e. 50% probability) is about 5 times the L_{10} life.

It should be noted that at a speed of 3000 rev/min the one million revolutions represents a running time of less than SIX hours.

Equivalent Bearing Load

The relevant load in bearing selection is that which controls the contact stresses. The effective load for this purpose is not simply the vector sum of the radial and axial loads applied – it is also dependent upon the rolling element/raceway geometry. For radial bearings the equivalent radial load P is given by:

$$P = XP_r + YP_a$$

where P_R = radial load
P_a = axial load
X, Y = factors in manufacturers' catalogues.

Static Load Rating

Under static (non-rotating) conditions, the load limitation is governed not by fatigue considerations but by plastic deformation. The basic static load rating C_o is defined as the static load which would produce a total permanent deformation of rolling element and raceway equal to 0.0001 or rolling element diameter. The resulting indentation will not normally cause unacceptable running conditions except in situations demanding especially smooth and silent running.
For rotating bearings, the equivalent bearing load P may exceed, but should not greatly exceed, C_o.

Limiting Speed

The primary limiting factor for speed is the permissible operating temperature for the lubricant. Manufacturers' catalogues list limiting speeds for both oil and grease lubrication. However, a rough guide to limiting speeds is given by the $d_m.N$ value for the bearing.

d_m = mean of bearing bore and OD (mm)
N = speed (rev/min)

For radial ball and cylindrical roller bearings:

Grease lubrication	$d_m.N = 500{,}000$
Oil bath lubrication	$d_m.N = 650{,}000$

Higher speeds are permissible with oil mist or oil jet lubrication.

Temperature Limitation

Standard bearings can be used at temperatures up to 125°C. Bearings treated to higher temperature stabilisation are available for operation up to about 250°C but at the expense of reduced hardness and load capacity. Bearings made with special materials (e.g. tool steel) are available for operation at temperatures as high as 500°C. In practice permissible operating temperatures are usually limited by the lubricant.

Radial Internal Clearance

Radial bearings are manufactured with initial internal clearance with is reduced when mounted with interference fit on shaft or in housing. The clearance is also influenced by differential thermal expansion of shaft and housing. As a general rule, the internal clearance should approach zero when the bearing is running under load at its operating temperature. Bearings of normal internal clearance are suitable for most situations but bearings with less than normal and greater than normal clearance are also available.

Friction

For design purposes friction in rolling bearings can be estimated on the basis of a coefficient of friction defined by:

$$\mu = 2M/PD$$

where μ = coefficient of friction (Table 3)
M = running frictional torque
P = equivalent bearing load
D = bearing bore (shaft diameter)

Starting torque would be approximately twice running torque. Under very light load ($P < 0.05C$) and/or speeds in excess of 25% of limiting speed, reference should be made to manufacturers' catalogues for more accurate calculation methods.

Table 3: Coefficient of Friction

Bearing type	**μ**
Ball – deep groove	0.0015
Ball – self-aligning	0.0010
Ball – angular contact	0.0020
Ball – duplex	0.0022
Cylindrical roller	0.0011
Tapered roller	0.0018
Spherical roller	0.0018
Ball – thrust	0.0013

For bearings fitted with rubbing seals, the additional friction caused by the seals can be greater than that of the bearing itself.

Mounting of Bearings

The basic principles are as follows:
The rotating component must be axially located in both directions (e.g. by a single deep groove ball bearing, a duplex bearing or a pair of angular contact ball bearings). Where radial loads are large it may be advisable to separate the requirements of axial location and radial load capacity (e.g. a deep groove ball bearing with clearance fit at its outer race mounted in close proximity to a cylindrical roller bearing which carries the radial load).
The bearing arrangement must be able to accommodate axial expansion of shaft relative to bearing housings without imposing excessive axial load on the bearings, or causing loss of radial location.
For radial bearings the race which rotates relative to the load time must have an interference fit with the mating component (i.e. a rotating shaft carrying a non-rotating radial load requires an interference fit with the bearing inner race, a rotating housing carrying a non-rotating load requires an interference fit with the bearing outer race. If the load rotates relative to both inner and outer races, both races must be mounted with interference fits). The necessary fits are given in manufacturers catalogues. They involve very close tolerances and must be achieved if the full potential of rolling bearings is to be realised in practice. An interference fit cannot be relied upon to

provide axial location of races on shaft or in housing. Positive location against a square shoulder is necessary.
The mounting and de-mounting of bearings must be achievable without transmitting force through the rolling elements by local heating or cooling to make or break in interference fit.

<u>Example</u>

A bearing is required to carry a radial load of 3 kN of constant direction and provide axial location for a shaft of 50 mm diameter rotating at 1500 rev/min. An L_{10} life of 10,000 hours is required.

First choice would be a deep groove ball bearing because of axial location capability in both directions.

Total revolutions in a life of 10,000 hours is:

$$10000 \times 60 \times 1500 = 900\ million$$
$$\underline{L_{10} = 900}$$

Since the load on the bearing is purely radial, $P = P_r$ (the factor $X = 1$ here). Thus:

$$C = P \times L^{1/3} = 3000 \times 900^{1/3}$$
$$\underline{C = 29kN}$$

Reference to a (leading) manufacturer's catalogue shows that a suitable bearing would be:

ISO designation	6210
Bore diameter	50 mm
Outer diameter	90 mm
Width	20 mm
C	35.1 kN
Co	19.6 kN
Speed limit	7000 rev/min (grease)
	8500 rev/min (oil)

The required shaft tolerance is k5 (interference fit because the shaft rotates relative to the load direction). The required housing tolerance is H8 (clearance fit because the outer race is stationary relative to the local direction).

HYDRODYNAMIC JOURNAL BEARINGS

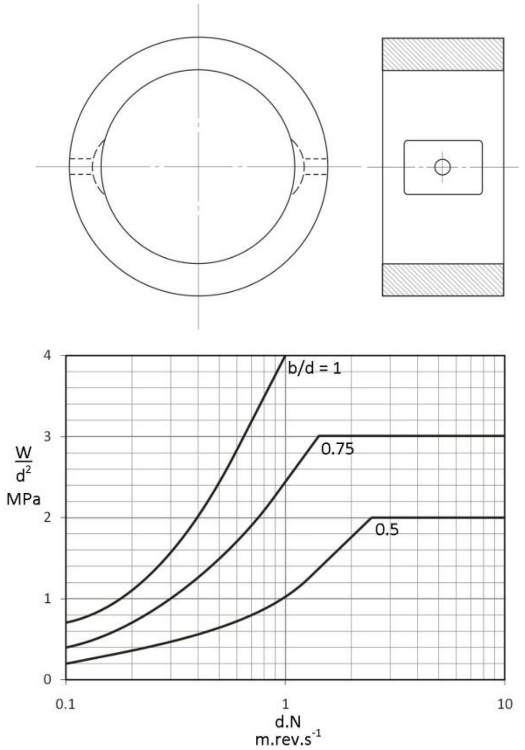
Introduction

The operation of a hydrodynamic journal bearing depends upon the shearing of a film of lubricant in the clearance space between the bearing (bush) and the journal. Load supporting pressure is generated within the film by continuous rotation of the journal. Lubricant must be fed to the bearing continuously to compensate for leakage from the ends of the bearing. The lubricant is normally distributed within the bearing by one or two axial grooves or by a circumferential groove according to the nature of the load which the bearing is required to carry.

Guidance given here applies only to load of constant or slowly varying magnitude. Dynamically loaded bearings (e.g. engine bearings) require consideration beyond the scope of these notes.

Axial groove bearing = if the load direction is constant or varies over an angle not more than 180°.

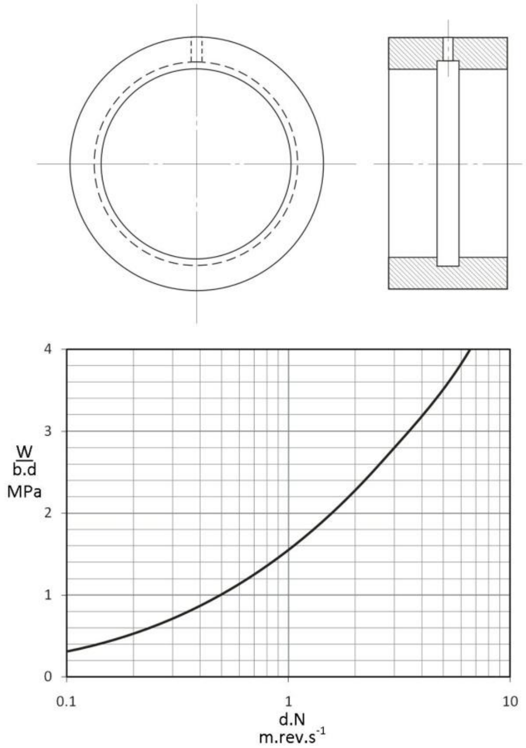
Circumferential groove bearing (effectively two narrow bearings side by side each of axial length b/2) – if the load direction varies over a large angle.



<u>Figure 1a:</u>

<u>Guide to required b/d ratio.</u>

<u>Axial groove bearing</u>



<u>Figure 1b:</u>

<u>Guide to specific load capacity.</u>

<u>Circumferential groove bearing</u>

Design Considerations

The design of hydrodynamic bearings requires:
Selection of bearing dimensions so that the load can be carried with sufficient separation of journal and bush to prevent direct contact of the opposing surfaces.
Selection of appropriate surface hardness and surface finish of journal and appropriate material for the bush or its lining.
Provision of adequate supply of lubricant.

Bearing Dimensions

Diameter, d. Nominal diameter will normally be dictated by strength or stiffness requirements of the shaft journal to be supported.

Bearing length, b. For whitemetal lined bearings, the bearing length should be chosen so that the specific load (W/bd) does not exceed 4 MPa. A lower specific load may be dictated at low speed to ensure adequate oil film thickness in the bearing (see Figure 1a for axial groove bearing or Figure 1b for circumferential groove bearing). For bearings required to start/stop under load, the specific load at start/stop should not exceed 1.3 MPa (twice this value is permissible if start/stop is very infrequent, i.e. about once a day). To avoid misalignment problems, bearing length should not exceed its diameter.

Diametral clearance, C_d. The diameter of the bush must necessarily be larger than that of the journal with which it engages. The different is referred to as the diametral clearance and must be of the order of 1/1000 of journal diameter. Table 1 gives values of C_d likely to prove satisfactory for a range of journal diameters and speeds.

Table 4: Recommended diametral clearance

	Journal speed (rev/sec)		
Journal dia. (mm)	1	10	100
20	0.04	0.05	0.06
30	0.04	0.05	0.07
40	0.04	0.05	0.08
60	0.04	0.06	0.10
100	0.05	0.09	0.15
200	0.10	0.18	0.30

Oil Feed Pockets – Axial Groove Bearing

If the load direction is constant, oil feed pockets (one or two) should be located at 90 deg to the load line.
If the load direction varies by more than 60° use a single pocket located 90° upstream of extreme load line. The load should never be directed into a feed pocket.
Recommended dimensions for feed pockets are 0.8 of bush length and 30° circumferential extent. Pocket depth should be around 3/30.

Circumferential Groove Dimensions

Recommended groove width is a = $d/10$ and groove depth of $d/30$. Overall bearing length will therefore be $(b + a)$.

<u>Example 1</u>

A bearing is required to carry a constant load of 10 kN at a journal speed of 3000 rev/min. The journal diameter is 76 mm. At start-up the load on the bearing would not exceed 1 kN:

Running load $W = 10000$ N
Start-up load $W_s = 1000$ N
Speed $N = 3000/60 = 50$ rev/sec
Diameter $d = 0.076$ m

$$W/d^2 = 10000/0.076^2 = 1.73\text{MPa}$$
$$dN = 0.076 \times 50 = 3.8\text{m rev/sec}$$

From Figure 1a, $b/d = 0.5$ is likely to satisfy requirements. Thus the required bearing length b is 38 mm. At-start-up the specific load would be:

$$\underline{W_s(bd) = 0.35MPa}$$

This is well within the recommended limit at start-up.

From Table 4, for $d = 76$ mm and $N = 50$ rev/sec, a diametral clearance $C_d = 0.1$ mm is indicated. Thus, during running the maximum possible eccentricity of the journal with respect to the bearing would be $C_d/2 = 0.05$ mm. The dimensions so obtained give only a guide for initial design. Refer to ESDU Item 84031 (or its successor) for detailed analysis. Minor design changes may then be necessary.

<u>Example 2</u>

A bearing is required to carry a load of 1.5 kN which rotates at journal speed of 1000 rev/min. The journal diameter is 45 mm. The load under start/stop conditions is negligible:

Load $W = 1500$ N
Speed $N = 1000/60 = 16.7$ rev/sec
Diameter $d = 0.045$ m
The bearing is required to carry a rotating load: hence the choice is for a circumferential groove bearing.

$$dN = 0.045 \times 16.7 = 0.75\text{m rev/sec}$$

From Figure 1b a specific load, $W/(b.d)$ of 1.2 MPa is indicated. Thus b = 28 mm.

Groove width $a = d/10$, say a = 5 mm
Overall length $(b + a)$ = 33 mm
Groove depth $d/30$ = 1.5 mm

From Table 1 C_d of about 0.06 mm is indicated.

Note: Excessive oil feed pressure to a circumferential groove bearing may lead to operational instability (shaft whirl). For this reason oil feed pressure should not exceed $0.1W$. For the present case feed pressure should not exceed 0.1 MPa. However, too low feed pressure would lead to excessive bearing temperatures due to inadequate oil flow. Refer to ESDU Item 90027 (or its successor) for detailed analysis.

HYDROSTATIC JOURNAL BEARINGS

Introduction

The operation of a hydrostatic journal bearing depends upon the maintenance of a supply of liquid lubricant to each of a number of recesses from a common source at suitable pressure. The lubricant flow to each recess is governed partly by the flow impedance downstream of the recess and partly by the impedance of a flow restrictor in the feed line to each recess. An external load applied to the journal causes a displacement of the journal from its concentric position and a readjustment of pressure in the various recesses results in a net hydrostatic force to balance the external load, as shown in Figure 2.

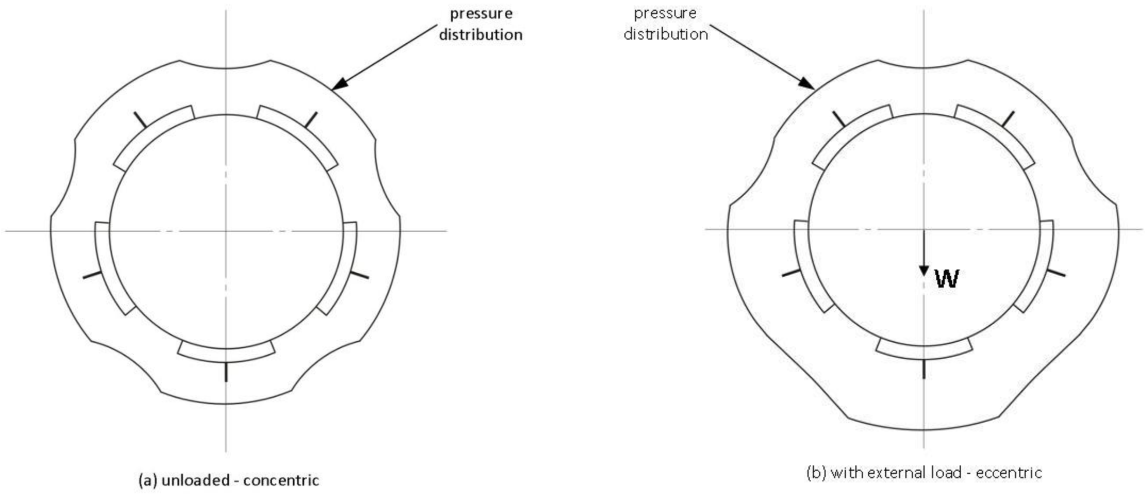


Figure 2: Pressure distribution in hydrostatic journal bearings

Guidance given here applies only to load of constant or slowly varying magnitude.
The data given relate to a bearing having 5 identical equally spaced rectangular recesses (or pockets) fed via identical capillary restrictors from a constant pressure oil supply.
A bearing having 5 (or more) equally spaced recesses has film stiffness which is insensitive to load direction.
A feature of externally pressurised (hydrostatic) bearings is that, unlike hydrodynamic bearings, they do not depend upon journal rotation for the establishment of a separating fluid film. In consequence they provide zero friction support for a non-rotating journal.
A further feature of all externally pressurised bearings is that they have a well-defined limited load capacity at which hydrostatic operation collapses. For this reason, bearing design must have regard for possible overload.

Design Considerations

The design of hydrostatic bearings requires:
Selection of bearing dimensions so that the load can be carried without collapse of the hydrostatic film.
Selection of capillary restrictor dimensions to ensure the bearing has adequate stiffness at the design load.
Provision of adequate supply of lubricant at the appropriate pressure.

Bearing Dimensions

Diameter, d. Nominal diameter will normally be dictated by strength or stiffness requirements of the shaft journal to be supported.
Diameter clearance, C_d. The guidance given in Table 1 for hydrodynamic journal bearings is appropriate for hydrostatic bearings. Tolerances on bearing and journal diameters need to be specified to control diametral clearance to within 10% of nominal value (this therefore requires typically E5/b4 tolerances).
Bearing length, b. The necessary bearing length to support a given load is directly related to the supply pressure.

i.e. $b = 5W/P_s d$

Selection of b by the above formula will ensure an overload capacity of at least 70%. Journal eccentricity will not exceed $0.2C_d$. To reduce potential misalignment problems, it is recommended that the length of a bearing should not exceed its diameter.
Recess/pocket dimensions (Figure 3)
Recommended recess dimensions are 48° circumferential and $b/2$ axial extent (giving side-sills $b/4$) with pocket depth $20C_d$.

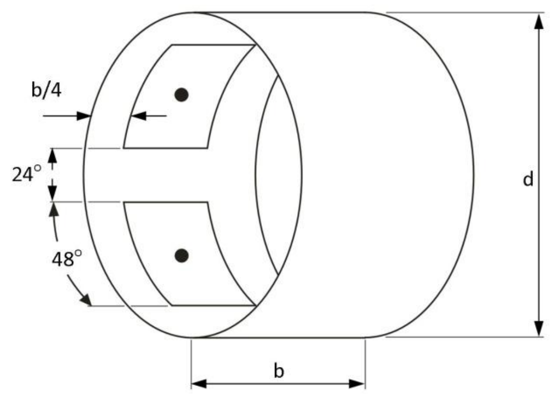


Figure 3: Recommended recess/pocket dimensions

Supply pressure P_s. In the choice of supply pressure, consideration should be given to the pressure capability of different forms of pump.

Typically Gear pump = 30 bar
Vane pump = 100 bar
Piston pump = 200 bar

Capillary restrictors. The capillary restrictors (one for each recess) should drop the pressure when the bearing carries no external load. Hypodermic tubing can be used for the purpose. For each restrictor the required tube length (L) of diameter (d_r) may be estimated from:

$$d_r^4/L = 6d\, C_d^3/b$$

Example

A bearing is required to support a radial load of 2.5 kN. The journal diameter is 0.10 m. The journal has very slow rotational motion within the bearing and, under stationary conditions, the frictional torque is required to be zero. A gear pump capable of a delivery pressure of 25 bar is available.

A hydrostatic bearing is the ideal candidate to satisfy the zero torque requirement.

Load	$W = 2500$ N
Speed	$N = 0$
Diameter	$d = 0.10$ m

$$b = 5W/P_s d$$
$$= 5 \times 2500 / (25 \times 10^5 \times 0.10)$$
$$\underline{b = 0.05\text{m}}$$

From Table 4 (Hydrodynamic Journal Bearings): for d = 100 mm and low speed, a diametral clearance of 0.05 mm is indicated.

For recess dimension, see Figure 3.

For restrictors

$$d_r^4/L = 6d\, C_d^3/\text{b}$$
$$= 6 \times 0.10 \times (0.05 \times 10^{-3})^3 / 0.05$$
$$= 1.50 \times 10^{-12}$$

This could be satisfied by hypodermic tubes 0.5 mm bore and 42 mm length.
The ultimate load capacity of this bearing would be in excess of 70% overload:

$$1.7 \times 2500 = 4250\text{N}$$

Eccentricity of the journal with respect to the bearing will not exceed $0.2C_d$ = 0.01 mm. Calculation of oil flow rate and required pumping power requires knowledge of the lubricant to be used. Refer to ESDU Item 92026 for detailed analysis.

OIL IMPREGNATED (POROUS METAL) BEARINGS

Introduction

Oil impregnated bearings are available in a range of standard sizes (ISO 2795) in the form of cylindrical bushes, flanged bushes, spherical bushes and thrust washers. Lubrication is provided by oil contained within the continuous pores formed by sintering metal powder moulded and partially compacted to the required shape. The resulting degree of porosity is normally around 20%. The oil contained within the bearing will normally provide lubrication for the recommended conditions of load, speed and temperature for long periods but extra oil can be added periodically if an oil-soaked pad and oil hole are incorporated in the bearing mounting.
The most common use of these bearings is for light load conditions but they are capable of carrying heavy load at low speed or if loading is intermittent.

The general purpose metal matrix is bronze (10% tin, 1% graphite) which is suitable with unhardened shafts. A cheaper and stronger alternative is iron (with up to 25% copper) which would be suitable in situations where rusting is unlikely – but the shaft would need to be hardened. Other specifications are available to meet special requirements.

Machining of the bearing surface (e.g. to increase the bore) is not normally undertaken. If it is undertaken at all, it must be done with a burnishing tool and not with any form of cutting tool which would risk closure of the pores by smearing of the metal.

Operational Suitability

Since the bearing is self-contained the permissible load and speed characteristic is dominated by the temperature rise due to friction. Table 5 gives guidance on maximum permissible specific load as a function of shaft speed for bronze matrix bearings.

Table 5: Permissible load as a function of shaft speed for bronze matrix bearings

Shaft speed, V (m/s)	**P_{max} (MPa)**
Static	34
Slow intermittent	21
0.1	15
0.5	3.5
>0.5	1.75/V

Mounting

Cylindrical bearing. For bearings conforming to ISO 2795, a rigid housing should be bored to H7 tolerance with a lead-in chamfer to facilitate pressing in of the bearing on a fitting mandrel. The resulting tolerance on the bearing bore after fitting will then be H7. For less rigid housings a greater degree of interference in the housing bore will be necessary.
Spherical bearing. The self-aligning ability of such bearings requires that the outer spherical surface is free to swivel within the housing. If the bearing carries a rotating load it may be necessary to key the bearing to prevent rotational creep within the housing.

Diametral clearance. For a bearing mounted as recommended to produce a bore tolerance H7 the shaft needs to have diameter to tolerance H7 the shaft needs to have diameter to tolerance 14 for maximum speed 1000 rev/min and e5 for high maximum speed. Ideally, the diametral clearance should range from about 0.001*d* for low speed to about 0.002*d* for high speed.

Shaft surface finish. The shaft should be ground to give a surface finish of 0.4 to 0.8 µm R_a or better.

Bearing friction

Under light load conditions the bearing operates in an essentially fluid film manner. At high load, part of the load is carried by direct contact between the shaft and the bearing. Thus coefficient of friction can vary widely. However, for a bearing with adequate lubrication, coefficient of friction is likely to be within the range 0.01 to 0.05.
For a dried-out bearing the friction will be much higher resulting in noisy whirl of the shaft within the bearing – the whirl direction is opposite to the direction of shaft rotation.

Further information

For more detailed information concerning this form of bearing and recommendations for their mounting see the "Tribology Handbook" (2nd edition.
M J Neale, Butterworth-Heinemann, London, 1995).

DRY RUBBING BEARINGS

Introduction

Bearings which are designed to operate without the use of oil or grease, and in which the journal (or runner) is in direct rubbing contact with the bearing material, are referred to as "dry rubbing bearings". Such bearings are made from materials which exhibit low wear rates over some range of pressure, temperature and sliding velocity.

However, friction is generally significantly higher than in rolling bearings or fluid film bearings.

Application

The advantage of not requiring lubrication leads to a cheap and simple bearing assembly. Thus on economic grounds and within their field of application, dry rubbing bearings are a natural first choice. However, friction and wear characteristics tend to limit their use to applications in which oils or greases are prohibited for reasons such as the presence of:

Oil or grease solvents
Extremes of ambient temperature
High vacuum
High levels of ionising radiation
Circumstances in which contamination of product or surroundings by a lubricant is dangerous or undesirable

The performance of many dry bearing materials is improved by the presence of a lubricant – either a conventional lubricant or a process fluid.

Materials

Commercially available dry bearing materials fall into five basic groups:

Thermoplastic (e.g. polyamide (nylon) and polyacetal).
PTFE
Thermoset
Carbon/ Graphite
Graphite impregnated metal (iron or bronze matrix)

The plastic materials commonly feature a filter (a finely divided solid added to modify mechanical properties, to improve the surface texture, or simply to reduce cost) or reinforcement (long fibres in woven or unwoven form bonded into the plastic to improve its strength, stiffness and impact resistance). The purpose of PTFE, graphite or MoS_2 as a filler is to directly improve friction and wear characteristics. Dry bearing materials commonly exhibit a period of relatively high wear rate during the initial stages of running. After this "bedding in" process the wear proceeds at a much reduced rate.

Form

Bearings may be machined from stock material which is basically homogeneous and which, therefore, has properties which do not vary as wear progresses.

Alternatively, bearings may have a relatively thin wearing layer bonded to a metal backing. Such bearings are brought in as finished components – they are not suitable for machining to size.

Operating limitations

Load limit.	Set by crushing strength of the material.
Speed limit.	Set by temperature rise due to friction (plastic materials have low thermal conductivity).
Temperature limit.	Temperature at which the material softens, degrades or oxidises.
Wear rate.	Within the above limits, wear rate increases with load and speed and, to a first approximation, is proportional to the product PV.

where P = specific load
V = counterface surface speed

Characteristics

The range of commercially available dry bearing materials is very extensive and in critical cases reference should be made to manufacturers. However Table 6 gives approximate data for the five basic material groups.

Table 6: Typical bearing material characteristics

Material group	P_{max} (MPa)	V_{max} (m/s)	T_{max} (°C)	μ
Thermoplastics	10	1	200	0.01 – 0.4
PTFE	3	2	300	0.05 – 02
Thermosets	20	1	200	0.1 – 0.4
Carbon/ Graphite	2	10	350	0.15 – 0.3
Graphite impregnated metal	2	10	600	0.01 – 0.2

NOTE: P_{max} corresponds to very low speed.
V_{max} corresponds to very low load.
P_{max}, V_{max} cannot be used together and not at T_{max}.

NOTE: Bearings comprising a thin polymer layer bonded to a metal backing generally tolerate higher load and higher maximum speed.

Journal bearing running clearance

Initial running clearance must be large enough to allow escape of wear debris and to accommodate thermal expansion of the bearing material. It should be noted that thermal expansion coefficients for polymers are generally much larger, and for graphite much smaller, than for metals. It should also be noted that some polymers swell by up to 5% on continuous immersion in water. Recommended running diametral clearances are:

Plastics 0.005*d* (min 0.1 mm)
Carbon/graphite 0.002*d* (min 0.08 mm)
Thin layer – metal backed 0.001*d* (min 0.03 mm)

Counterface material and surface finish

A counterface material which is hard compared with the bearing material is required to ensure that the replaceable bearing wears in preference to the counterface. In the absence of oil or grease, carbon steel may readily corrode but is otherwise satisfactory. Hard chromium plate on steel is often a good choice. Stainless steel is also good but its low thermal conductivity reduces the permissible value of the PV product for the bearing material for a given temperature rise.

In practical terms, the lower the counterface surface roughness the lower will be the wear rate. For cost reasons a ground surface of 0.2 – 04 μm R_a is normally recommended. Surface corrosion is likely to increase surface roughness leading to increased wear use.

Bearing mounting

It is necessary that the bearing is securely fitted so that it cannot rotate within its housing. Metal-backed, metal-matrix and carbon/graphite bearings should be interference fitted in the housing.

Carbon/graphite has very low thermal expansion coefficient and care must be exercised to ensure the interference is not lost over the range of operating temperature.

Interference fitting of thermoplastic materials is not reliable since such materials creep at quite low stress levels and the bearing can become loose. A suitable adhesive or positive mechanical locating device is necessary.

Further information

HSDU Item 87007 contains detailed information on the very wide range of dry bearing materials available.

LUBRICATION SYSTEMS

Introduction

The type of lubrication system required depends upon both the function and form of the lubricant.

Lubricant function

Primary function: friction and wear control by reduction of adhesion between the relatively moving surfaces.
Secondary function: temperature control, i.e. the lubricant doubles as a convective heat transfer medium.

Lubricant form

Grease: oil with thickener: semi-solid, i.e. not free flowing.
Oil: free flowing, convective heat transport capability.
Process fluid: no separate lubrication system required but fluid must have adequate lubrication qualities.

Further information can be found in The Tribology Handbook.

System Selection

Table 7: Grease systems

System type	Characteristics	Application
Non - replenishable	Sealed for life	Rolling bearings – light duty
Grease gun	Grease nipple at each bearing	Accessible bearings
Centralised	Hand operated or motorised pump with manifold connection to a group of bearings	Large number of bearings. Inaccessible bearings requiring frequent re-lubrication

Table 8: Oil Systems

System Type	Characteristics	Application
Oil Can	Total loss	Light duty accessible bearings
Centralised Non-recirculation	Total loss Hand operated or motorised pump	Large number of bearings. Inaccessible bearings requiring frequent re-lubrication.
Oil mist	Total loss. Aerosol spray. Compressed air supply required Good heat transport capability	Rolling bearings
System Type	**Characteristics**	**Application**
Wick feed **Oil reservoir.**	Total loss.	Light duty plain bearings and mechanisms
Splash **Self contained oil bath**	Oil splashed to bearings by rotating component dipping into oil bath	Enclosed medium-speed horizontal-shaft gearboxes and mechanisms
Rig or disc **Self contained oil bath**	Oil carried to bearing by rotating ring or disc dripping into oil bath. Local recirculation of oil – local heat transport capability	Horizontal shaft plain bearings Low to moderate speed.
Pumped circulation	Pressure feed by pump from internal or external oil reservoir. Good heat transport capability with oil cooler in circuit	All cases requiring large oil flow or involving high energy dissipation

DATA SOURCES

Textbooks:

Tribology Handbook, 2nd edition.
M J Neale, Butterworth-Heinemann, London, 1995

Engineering Sciences Data Unit (ESDU) Items; Tribology Sub-Series:

Bearing Selection:

65007 General guide to the choice of journal bearing type.
67033 General guide to the choice of thrust bearing type.
89044 Friction in bearings.

Rolling Bearings:

81005 Designing with rolling bearings. Part 1: design considerations in rolling bearing selection with particular reference to single row radial and cylindrical roller bearings
81037 Designing with rolling bearings. Part: selection of single row angular contact ball, tapered roller and spherical roller bearings.
82014 Designing with rolling bearings. Part 3: special types.

Journal Bearing Calculations:

84031 Calculation methods for steadily loaded axial groove hydrodynamic journal bearings.
85028 Calculation methods for steadily loaded axial groove hydrodynamic journal bearings.

Superlaminar Operation:

86008 Calculation methods for steadily loaded axial groove hydrodynamic

Journal Bearings:

90027 Calculation methods for steadily loaded central circumferential groove hydrodynamic journal bearings.
92026 Calculation methods for externally pressurised (hydrostatic) journal bearings with capillary restrictor control.

Thrust Bearing Calculations:

82029 Calculation methods for steadily loaded fixed-inclined-pad thrust bearings.
83004 Calculation methods for steadily loaded, off-set pivot, tilting-pad thrust bearings.

Temperatures in Bearings:

78026	Equilibrium temperatures in self-contained bearing assemblies, Part I: outline of method of estimation.
78026	Equilibrium temperatures in self-contained bearing assemblies, Part II: first approximation to temperature rise.
78026	Equilibrium temperatures is self-contained bearing assemblies, Part III; estimation thermal resistance of an assembly.
78026	Equilibrium temperatures in self-contained bearing assemblies, Part IV: heat transfer coefficient and joint conductance.
78026	Equilibrium temperatures in self-contained bearing assemblies, Part V: example of the complete method.

Heavily Loaded Contacts:

78035	Contact phenomena, I: stresses, deflections and contact dimensions for normally-loaded unlubricated elastic components.
84017	Contact phenomena, II: stress fields and failure criteria in concentrated elastic contacts under combined normal and tangential loading.
85007	Contact phenomena, III: calculation of individual stress components in concentrated elastic contacts under combined normal and tangential loading.
85027	Film thicknesses in lubricated Hertzian contacts (EHL). Part I: two-dimensional contacts (line contacts).
89045	Film thicknesses in lubricated Hertzian contacts (EHL). Part 2: point contacts.

Lubrication Systems:

68039	Guide to the design of tanks for forced-circulation oil-lubrication systems.
78032	Grease life estimation in rolling bearings.
83030	Selection of filter rating for lubrication systems.

Material Selection:

84041	Properties of common engineering materials.
86040	Selection of surface treatments and coatings for combating wear of load-bearing surfaces.
87007	Design and material selection for dry rubbing-bearings.
88018	Selection of alloys for hydrodynamic bearings.

Flexible Elements:

67021	The design of crossed flexure-pivots.

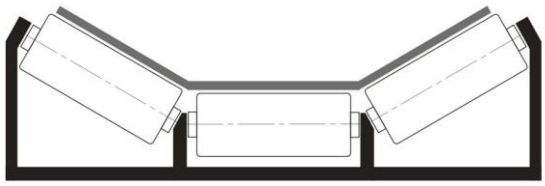
Seal Selection:

80012	Dynamic sealing of fluids. I: guide to selection of rotary seals.
83031	Dynamic sealing of fluids. II: guide to selection of reciprocating seals.

SIMPLE DESIGN EXAMPLES

Conveyor Belt Bearings

A conveyor belt is supported by a number of idler stations spaced at intervals of 1.0 m. Each station consists of three idler rollers which rotate about stationary axles fixed to a support (see figureure). The rollers are of diameter 100 mm and weigh 6 kg. The two side rollers are angled at 30^{O} to the horizontal. A belt, weighing 40 kg/m, passes over the rollers and runs at a speed of 3m/s. The conveyor is designed to transport 3000 tonnes of material per hour. Experience suggests that the control idler supports approximately 60% of this load.



Select bearings for the idler rollers. The conveyor belt is several miles long so that unit cost of the bearings is of utmost importance. Estimate the life of the bearings you have selected. If the conveyor is used in a mining operation what extra precautions should be taken?

<u>Answer:</u> The nature of the installation necessitates low-friction self-contained bearings which will operate for long periods between maintenance shutdowns. Grease lubricated rolling element bearings are likely to best satisfy the requirements.

The load at each idler station is obtained from the idler separation *l*, belt speed *v,* belt mass per unit length *m* and belt loading *L, so:*

$$F = l\left(\frac{L}{v} + m\right)g = 3117N$$

The rotational speed of the rollers is calculated as 573 rev/min. The central idler supports 60% of the load giving an individual radial bearing load of 935 N. Since the axial loads are small, *deep groove ball bearings are selected.* A manufacturer's catalogue is then used to select an appropriate bearing. Larger bearings are more expensive but have higher load ratings and therefore longer lives.

One approach is to decide on the design life and use the equation.

$$L_{10} = \left(\frac{C}{P}\right)^p$$

to determine the required basic load rating, C. For this bearing with a dynamic load of 935 N and a desired life of ten years, the basic load rating required is 13 kN. The cheapest bearing having this load rating could then be used. He bearings in the side idlers support axial as well as radial load. These loads F_r and F_a can be determined by resolving forces. The effective dynamic load on the bearing is now given by:

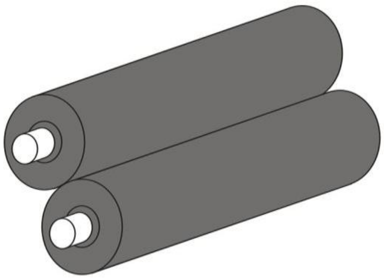
$$P = XF_r + YF_a$$

Where *X* and *Y* are values tabulated in the catalogues. Life is then determined using this value of *P.*

If the conveyor issued in a mining application the environment is likely to be dirty. This contamination would lead to greatly reduced bearing life (through wear or fatigue initiation). It is therefore essential to ensure adequate bearing sealing.

Photocopier Paper Feed Rollers

The feed of a printed page from a photocopier is controlled by two plastic rollers. The rollers are of outer diameter 20 mm pressed onto 10 mm diameter steel shafts. They are sprung together with an estimated force of 20 N. The maximum photocopying rate is 30 pages per minute.



Select bearings to support the two rollers in the body of the copier. The bearings should be cheap, clean and maintenance free.

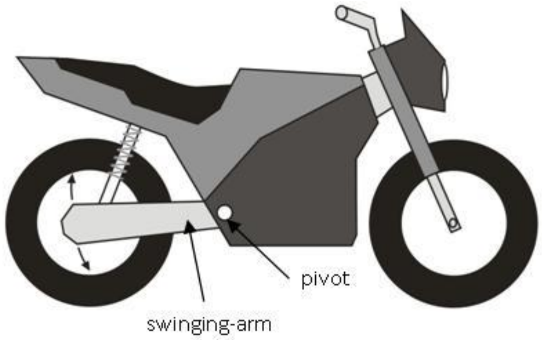
Answer: The rotational speed of the rollers is calculated as 2.4 rev/sec (for an A4 page of length 30 cm) and the spindle speed at the bearing less than 0.1 m/s. The light load and low speed together with the requirement that the bearings should be cheap, clean and maintenance free, suggests that first consideration be given to a *dry rubbing* bearing. The length/diameter ratio should be between 1/2 and 3/2 (to allow free passage of debris from the bearing). So if we choose a bearing length of 5 mm. then the specific load, P is:

$$P = \frac{W}{bd} = 0.4MPa$$

At these low loads and speeds. The cheapest of dry rubbing materials (the thermoplastics) is adequate.

Motor Cycle Suspension

The figure shows part of the suspension linkage of the rear wheel of a motor cycle. The lower member is known as a 'swinging-arm' and pivots at the bike chassis. The pair (one on either side of the chassis) support load from the chain drive (approximately 300 N) and a component from the spring and damper (varying from 300 N to 500 N). As the suspension deflects the swinging-arm undergoes small-scale oscillations. The pin which links the two arms, and acts as the bearing journal has a diameter of 15 mm.

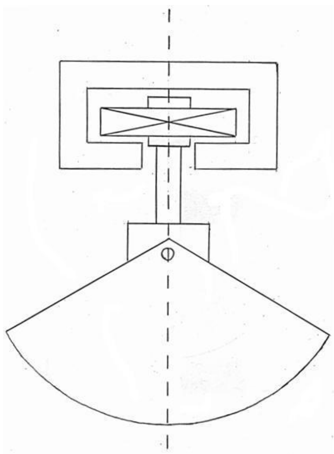


Select a cheap and reliable bearing to act as a pivot between the swinging-arm and the bike chassis.

Answer: Since the motion is oscillatory and of small angular amplitude, a *bonded rubber bush* would provide a simple and cheap solution for a light motor cycle. For a high power machine, an *oil impregnated porous meal bush* would be preferred as this would reduce flexing of the pivot axis. A bronze matrix is recommended for such a bearing since it is located at the underside of the motor cycle and water ingress is possible. For a *b/d* ratio of 1.0 the specific load is about 1.8 MPa which is well within the capacity of bronze matrix bushes.

Crane Grab

A lorry mounted crane grab is used to pick up rubble and builder's waste. It is designed to carry loads up to 30 kN. These loads frequently need to be swivelled before lowering. The grab weighs 30 kg and is supported on a shaft diameter of 25 mm.



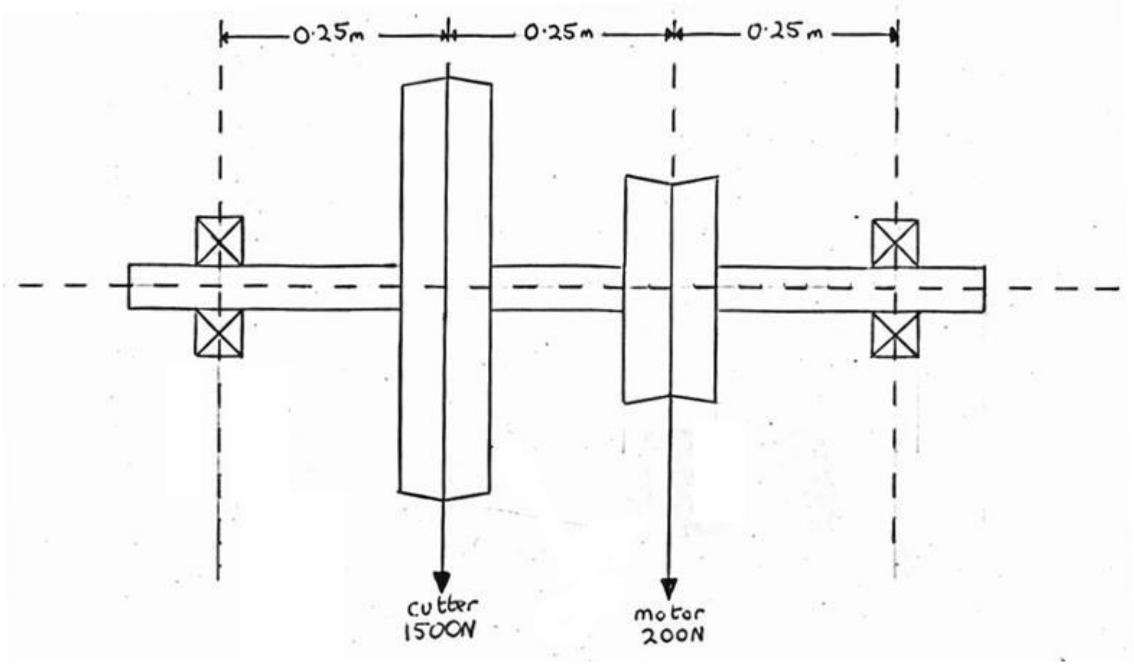
Select a bearing which will support the grab and allow free rotational movement. Estimate the life of this bearing.

Answer: Since the loading is purely in the axial direction, a thrust bearing is required. The free rotational swivel action requirement can be satisfied only if friction is low under static conditions. A *thrust ball bearing* offers the simplest design. The grab is likely to receive rough handling so a double acting bearing is essential to prevent unseating of the bearing when the grab is subjected to an upward force.

A manufacturer's catalogue is used to select the required bearing. The critical factor is the static load rating which for this size of bearing (i.e = 25 mm) is in the range 37-112 kN according to bearing overall dimensions. Bearing life estimation (on the basis of rolling element fatigue) is not applicable under static conditions.

Belt Drive Shaft Bearings

The figure shows the drive shaft from a combine harvester. The smaller pulley is driven by a motor which exerts a force of 200 N vertically downwards. The harvester cutter is driven from the larger pulley which exerts a downwards force of 1500 N. The shaft is 50 mm diameter and designed to rotate at 300 rev/min.



Design *hydrodynamic journal bearings* to support this shaft at the locations shown.

Answer: Resolving forces gives the loads on the bearings as 1067 N and 633 N for the left and right-hand bearings respectively. The direction of the load is constant so an axial groove bearing is suitable. Reference to the section on Hydrodynamic Journal Bearings shows that a ratio of $b/d = 0.5$ (giving a bearing length of 25 mm) and diametral clearance (C_d) of 0.05 mm are likely to satisfy the requirements. The bearings may be required to start up under load but the start-up specific load (W/bd) for the more heavily loaded bearing is only 0.85 MPa which is entirely acceptable for a white metal lined bearing.