Bearings: Searching for a longer life

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PAPER 1 - POROUS METAL BEARINGS AND THEIR APPLICATIONS

by Victor T Morgan, FIM, AIMechE, Development Manager, GKN Bound Brook Ltd

Mr V T Morgan joined his present company in 1944, after a number of war time appointments associated with aircraft production. His many contributions since then to the development of powder metallurgy in general and to porous metals in particular, especially in the field of tribology, have earned him a world-wide reputation. He was Chairman of the Powder Metallurgy Joint Group of the Institute of Metals and the Iron and Steel Institute for 1970 and 1971. He was a member of the Tribology Advisory Committee set by the Department of Trade and Industry, and a contributor to the Tribology Handbook published in 1971. He is a member of the Tribology Steering Committee of the Engineering Sciences Data Unit. In 1981 he was awarded the Tom Colclough Medal by the Metals Society for his contribution to the theory and practice of sintered metal bearings.

This opening paper will present a critical overview of the selection of bearing type, according to the various requirements of the application. It will provide a general backcloth against which can be set the different types of bearing which are to be discussed in detail during the Conference. In particular, it will enable the advantages and limitations of porous metal bearings to be clearly identified in relation to the characteristics of the other bearing types which are available to the design engineer.

PAPER 2 - SPLIT ROLLING BEARINGS - SAVING ON CAPITAL AND LIFE COSTS

by P W R Blake, MA(Cantab), C Eng, MIMechE, Technical Manager and G D Hickman, C Eng, MIMechE, Sales Director, Cooper Roller Bearings Co Ltd

After military service, Peter Blake served an apprenticeship at Metropolitant Vickers. He joined Cooper Roller Bearing Co Ltd in 1953 as Deputy Chief Engineer and subsquently became Chief Designer. He is now Technical Manager.

Mr Hickman has spent some 28 years in the bearing industry having been previously involved in heavy mechanical plant design and operation. He is a keen educationalist with interests in local schools and the University of East Anglia

The introduction of the split roller bearing and its features leads to a discussion of four interesting and unique applications where the split roller bearing has contributed to an easier solution and, in the first case, the only solution to providing a bearing where it is required.

The four cases are:-

(1) The top segment of support rolls on continuous casting machines for casting wide slabs has unique problems of cooling, overall height and support of the very heavy loads. It contributes greatly to the accuracy of the strand if the supporting rolls can be split into two or three shorter lengths with bearings between them.

(2) Vertical applications for large electric motors and heavy rotary heat exchangers present problems in supporting the loads. The split flat thrust bearing combined with the Cooper split radial roller bearing can provide an economic answer.

(3) The main bearing on some large reciprocating pumps is a split full row design in order to increase its load carrying capacity. A discussion of the special problems of the full row roller bearing is included.

(4) The provision of Cooper roller bearings on primary air fans and motors as original equipment and as conversions from plain or solid bearings are discussed with particular reference to the steps taken in the conversion to Cooper bearings, ie, sizes, types and the economic considerations.

EHD lubrication is mentioned in general and for the above applications in particular.

PAPER 3 - THERMOPLASTIC BEARINGS

by F W Pedley, Market Development Manager, Norton Performance Plastics (UK) Ltd

Francis Wm Pedley joined Pampus Fluorplast Ltd in 1973 as Sales Engineer to promote the use of fluoroplastic products and applications in the chemical, automotive and general engineering sectors. He became General Sales Manager in 1978 and with the take-over in 1982 of the Pampus Group by the Norton Company of Worcester Massachusetts USA, was promoted to Marketing Manager for Norton Performance Plastics (UK) Ltd.

This paper will discuss the advantages and design limitations of pure PTFE, the influence of traditional fillers namely glass fibre, carbon, bronze, together with the newer fillers such as PPS and Ekonol (a polyester aromatic polymer) and the load carrying capabilities of filled PTFE/metal/fabric composites. By means of case histories, the selection and use of these fluoroplastic bearing materials will be highlighted in a variety of dry bearing applications where severe loadings and harsh environments cause problems.

PAPER 4 - AIR LUBRICATED BEARINGS

by M C Tempest, Chief Engineer, Westwind Air Bearings Ltd

Following a Degree in Mechanical Engineering the author joined Armstrong Siddeley Motors, a Company largely associated with the production of aero engines. In the late 1950's this Company became involved in research on aerodynamic gas bearings, this work subsequently applied to the manufacture of gas bearings circulators. Having spent some three years working in this field, the author then moved on to Mechanical Research and Development associated with the manufacture of gas turbines. He returned to his first love in 1966, joining Westwind Air Bearings Ltd, and has remained with them up to the present time. He became Chief Engineer in 1977.

After recalling the early history of air bearings and the mechanism of operation, this presentation discusses advantages, disadvantages and applications in a variety of fields. The objective of the paper is to show that air lubrication is now firmly established as a concept and is beginning to play a significant part in the design of machine tools. Not without interest, a significant application is in the manufacture of rolling element bearings.

PAPER 5 - ADVANTAGES OF CLEAN LUBRICATION

by N R Way, Technical Director, and Dr N Moss, SLS Dept, Pall Europe.

Norman Way was trained as a chemist and has been involved with fluid contamination and control for some 32 years. He joined Pall (Europe) Ltd from Vokes Ltd in 1967 and is currently Technical Director.

Dr N Moss is a chemical engineer who took his BSc at Imperial College, London and his Phd at Sheffield University. He has been at Pall (Europe) Ltd for 4½ years in charge of the Fluid Power Laboratory and is now a Staff Scientist.

Technical studies have provided evidence that fine filtration can drastically increase component life in lubrication systems with consequence economic savings.

When the decision is taken to use finer filtration in an existing system or to incorporate fine filtration into a new design, the user should be aware of the necessity for full contamination control in order to achieve the maximum benefit from improved cleanliness.

This presentation introduces a small part of the evidence in favour of fine filtration and outlines actions which should be taken when translating fine filtration from laboratory to service conditions. Cases from field experience will be illustrated.

PAPER 6 - SHOCK PULSE MONITORING

by P J Brown, C Eng, Managing Director, SPM Instrument UK Ltd

Peter Brown, Managing Director of SPM Instrument UK Ltd, a Chartered Engineer, has been involved in condition monitoring activities for the last 12 years. Eight years ago he, in conjunction with SPM Instrument AB co-founded SPM Instrument UK Ltd, as the principal UK operation for the distribution and marketing of the unique Shock Pulse Method for the monitoring of rolling element bearings.

The SPM technique, which was developed approximately 15 years ago by Eivind Sohoel is a revolutionary concept towards the monitoring of rolling element bearings.

Inadequate lubricant film thickness is probably the biggest single cause of premature bearing failure and a new microprocessor unit, in addition to predicting bearing damage, is able to determine the degree of lubrication film thickness.

PAPER 7 - PREDICTION OF BEARING WEAR DUE TO SHAFT VOLTAGES IN ELECTRICAL MACHINES

by M Bradford, BSc, MSc, MIEE, Manager, Machines & Drives Dept, ERA Technology Ltd

Mike Bradford is responsible for all ERA work on various aspects of electrical machines design, construction and application. This work has included a number of studies of bearing damage due to shaft voltages, from site investigations of large motors on offshore platforms to detailed laboratory studies of the characteristics of bearing current.

The importance of bearing reliability in electrical machines will be discussed with respect to the incidence and causes of failure and the significance of bearing currents. The origin of damaging bearing currents and shaft voltages in electrical machines will be reviewed, dealing with faults at the design stage and faults occurring in service. Methods of preventing bearing damage due to circulating currents will be discussed, again separately considering design aspects (eg, insulation) and use aspect (eg, lubrication).

Data on the characteristics of bearing damage will be presented, including the relationship between shaft voltage, bearing current, bearing impedance and bearing wear rate. The possibility of employing alternative methods of measuring or monitoring shaft voltage as a means of increasing machine reliability will be considered.

The paper will draw substantially on ERA work on this topic in recent years and will outline plans for future work.

PAPER 8 - ADVANCES IN MONITORING BEARINGS

by T Henry, MA, C Eng, FIMechE, Director Wolfson Unit.

Joined Metropolitan-Vickers Electrical Company Limited (now GEC Ltd) as a college apprentice in 1955. Took the Mechanical Sciences Tripos at Cambridge and graduated from Trinity College in 1958. Joined AEI Turbine-Generators Ltd at the end of the apprenticeship and worked on the design of large steam turbines. In 1963 took up a post in engineering at the University of Manchester and became Senior Lecturer at the Simon Engineering Laboratories of the University.

He has 25 years experience in the design and maintenance of rotating machinery and specialises in the condition monitoring of machines, particularly the study of vibration for monitoring and diagnosis. Current work includes the detection of crack growth in rotors by monitoring the pedestal vibrations, the development of data handling systems for condition-based maintenance and of techniques of failure prediction from monitored data.

In 1980 he took on the task of setting up the Wolfson Industrial Maintenance Unit in the University of Manchester, to enhance the service provided to industry in maintenance related topics. The Unit is now a thriving concern with contracts in the UK and abroad to provide a wide range of services for plant performance improvement, including courses, system design, maintenance and condition monitoring information and control systems, condition based maintenance survyes, programmes and developments.

The practical techniques available to the engineer for monitoring in-situ the deterioration of rolling element bearings are reviewed. Recent additions to and upgrades of existing techniques are described. The delegate will thus be able to select a technique suited to his requirement.

The potential user of a specific technique is concerned primarily with the reliability of failure prediction and the ease of use. Ways of applying a selected technique in a cost effective manner are presented. A method, employing signal enveloping, to assess the reliability of failure prediction in any specific situation is described.

PAPER 9 - CONVERTING ROTARY MOTION TO LINEAR MOTION - A NEW PRODUCT

by R A E Wood, C Eng, MIMechE, Bearing Engineering Manager, RHP Bearings Ltd

In 1954 the author joined RHP at Chelmsford working on bearing design and application. He was recently appointed to Bearings Engineering Manager at RHP Bearings, Newark. He is Chairman of BSI MEE/30 technical Committee for Rolling Bearings and UK representative on the ISO Technical Committee for Rolling Bearings and its Sub-committees. Prior to joining RHP, Mr Wood was involved in motor body design and in mechanical/electrical engineering design at GEC Research Laboratories.

The paper introduces the ball nut, a new product based on bearing technology that converts rotary motion to linear motion on a threadless shaft. The apparent simplicity of the patented design is described and details of its characteristics for pitch rates, axial load capacity and speed are given. The advantages of the ball nut: direct drive, no backlash, slip on dynamic overload, static security, safety and low cost are highlighted. In conclusion, ways of mounting the ball nut and examples of many applications are provided. The presentation will be illustrated by slides and demonstration models.

CONFERENCE CHAIRMAN

Michael Neale, Wh Sch, DIC, BSc(Eng), FIMechE

Michael Neale and Associates, Consulting Engineers, Farnham, Surrey

CONTENTS

1

Porous metal bearings and their applications

V T MORGAN, FIM, MIMechE
Development Manager, GKN Bound Brook Ltd

Although a "Self-Lubricating Bearing" can be defined as any bearing which lubricates itself, and dry bearings, as described in the previous paper, would readily fall under that definition, this paper deals specifically with oil retaining bearings. These are porous metal bearings which self-lubricate by the oil contained in the porosity.

They are produced by the partial compaction of metal powders in precision tools, followed by the sintering of the powder compact in a reducing atmosphere at a temperature of about 80% of the absolute melting point of the metal. The sintered compact is repressed to restore dimensional accuracy, to produce a high surface finish and, by work hardening, to increase the elasticity. The amount of the porosity depends mainly upon the degree of compaction of the powder, and this porosity is vacuum impregnated with lubricating oil. A selection of the sizes and shapes available is shown in Fig 1.

PROPERTIES

The properties of porous metal bearings depend upon the following factors:

The Alloy Composition

There are two main alloys, bronze and iron. Porous bronze (90:10 copper tin) is the most popular, and it usually has about 1% of added graphite. Porous iron is stronger than porous bronze but has a lower compatibility against a mild steel shaft, and is susceptible to rusting in a humid environment.

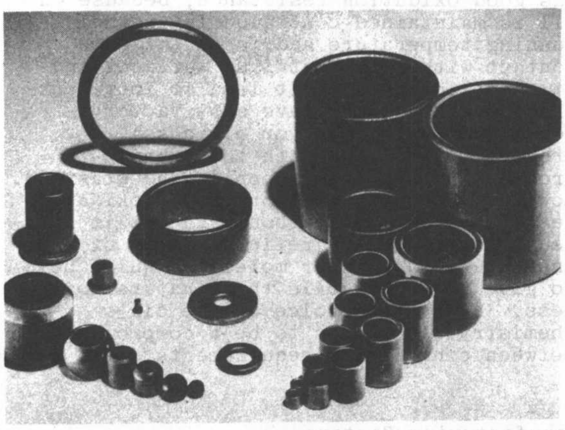


Fig 1

The Quantity of Porosity

The amount of porosity determines both the oil holding capacity and the mechanical properties. For a given alloy, the higher the porosity (lower density) the lower is the tensile and compressive strength, and the thermal conductivity. Hence the choice is a compromise between high oil-holding capacity, and adequate strength and thermal conductivity. Fig 2 shows a microsection of typical high and low porosity porous metal bearings.

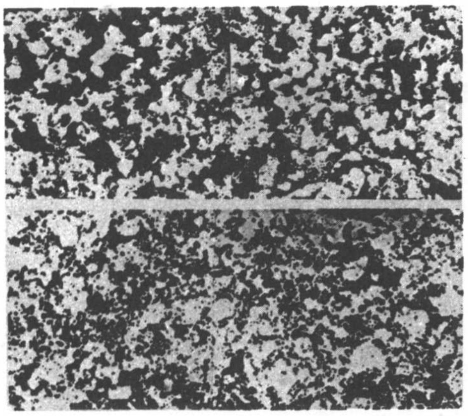


Fig 2

The Quality of the Porosity

For a given porosity it is possible to have either a large number of small pores or a smaller number of larger pores. Smaller pores result in lower permeability and a higher capillarity; that is, a larger resistance to fluid flow and a larger wicking force (or entry capillary pressure) The choice of pore size is again a compromise between conflicting requirements, because smaller pores produce a smaller loss of oil pressure into the pores but are more easily closed by smearing of the surface porosity under boundary conditions.

The Impregnating Oil

It is essential that the impregnating oil has good oxidation resistance, because the oil is maintained continuously at the running temperature and is in intimate contact with metal catalyst surfaces and atmospheric oxygen. It is also improtant that the oil should have satisfactory boundary lubrication (or oiliness) properties, to cope with the running-in process and frequent stopping and starting. However, in order to improve the oxidation resistance of hydrocarbon oils, it is usual to remove, by refining, many of the unsaturated and polar molecules which help to give petroleum oils their natural oiliness. Hence the choice of the oil's chemistry also tends to be a compromise between conflicting requirements.

Manufacturing Factors

Other factors which have an effect on the properties include the degree of sintering, the amount of work hardening induced during repressing, the surface finish, and the cylindricity of the bore (ie, the degree of departure from a perfect cylinder).

Table 1 summarises the properties of some typical porous bearing compositions.

Table 1

Composition	Dry density (g/cc)	Typical porosity (%)	Typical tensile strength (lbf/in^2)
89% Copper, 10% Tin, 1% Graphite	5.8	31	8 000
	6.0	29	9 000
	6.3	25	11 000
Iron(soft) (no graphite)	5.2	34	11 000
	5.5	30	12 500
	5.8	26	14 000
	6.0	23	15 500
95% to 98% Iron, 2% to 5% Copper (no graphite)	5.2	34	12 500
	5.5	30	18 000
	5.8	26	22 500
	6.0	24	25 000
75% to 90% Iron, 10% to 25% Copper (no graphite)	5.8	27	24 500
	6.1	23	31 000
50% Iron 50% Copper	5.6	32	12 000
	5.8	30	14 000

ADVANTAGES

The main advantage of porous metal bearings is their availability in a wide range of standard stock sizes at a low cost. Also, as the whole of the surface is porous, there is no need to drill an oil hole and hence they avoid the necessity of having to be fitted in a particular position with respect of the direction of the load. They are therefore equally suitable for rotating loads. There is no need to provide oil grooves, oil bottles, oil supply pipe lines, grease nipples, etc, and, of course, they overcome the problem of supplying oil where the bearing housing moves or rotates, or is in an inaccessible position. As the oil reservoir is a metallic sponge, the possibility of loose fibres (from a felt or wick entering the clearance space) is avoided. The supply of oil from the porous wall to the clearance space is self-regulating, and this reduces the hazard of oil drip spoiling manufactured articles, as can often happen with a badly adjusted drip feed bearing.

LIMITATIONS

The main limitation of porous metal bearings is the reduced mechanical properties caused by the porosity, as is illustrated in Table 1. Standard porous bronze bearings are unsuitable for loads (including impact loads) greater than about 7 000 lbf/in^2; and porous iron is unsuitable for impact loads greater than 20 000 lbf/in^2, unless the porosity is reduced to a level where the quantity of oil available is probably insufficient for a satisfactory life.

The presence of porosity also reduces the thermal conductivity of the metal and although there is some heat transfer by the circulation of oil within the porosity, this reduction in heat transfer through the porous wall is important in determining the running temperature especially where the conductivity of frictional heat along the shaft is small (or even negative if the shaft is connected to a heat source).

A further limitation of porous metal bearings is the undesirability of machining the surface, and alternative methods of controlling the fitted bore size are given in the paragraph headed "Servicing".

The effect of any metal cutting action, such as reaming, grinding or machining, is to close the surface porosity. If the bearing is to retain its self-lubricating properties, the pores must extend to the working surface. However, many porous metal bearings are used in conjunction with a conventional oil supply or submerged in an oil bath, where the self-lubricating properties are not required, because they are cheaper than a non-porous alternative. In these cases, closure of the surface porosity is unimportant and machining is permissible.

INSTALLATION

At first view it might appear difficult to control the running clearance and align the bearing without machining the bore surface. However, together with the limitation imposed by the porosity on the mechanical properties of the porous metal, these apparent disadvantages become an advantage when the correct methods of installation are employed.

There are two main types of porous metal bearing assemblies to consider.

Self-aligning Bearings

Where a misalignment problem is envisaged (due to a difficulty in maintaining alignment between the two housing bores), a pair of self-aligning porous bearings is recommended. In this case, the outside of the bearing is in the form of a sphere or part of a sphere whose centre lies on the axis of the cylindrical bore. An example is illustrated in Fig 3. It is important that only a small torque is required to turn the spherical bearing in its housing otherwise much of the advantage of this type of assembly will be lost. Where shaft deflection is a problem (as illustrated in Fig 4), a self-aligning bearing also helps to prevent edge loading, but where the load, and hence the deflection is rotating (eg, due to out of balance forces), the torque to turn the spherical bearing in its housing is even more important.

With this type of assembly the room temperature running clearance may be smaller than usual because an increase in temperature tends to cause an increase in running clearance. The variation in clearance between one assembly and another is limited to the sum of the tolerances on the shaft diameter and the bearing bore diameter.

$$\delta = \frac{WbL}{48\,E\,I\,(a+b)} \left[4(a+b)^2 - L^2 - 4b^2 \right] \quad ..(1)$$

where (using any coherent system of units):

(a+b) is the distance between the two bearings A and B
L is the bearing length
E is Young's modulus of the shaft material
I is the moment of inertia of the shaft ($\pi d^4/64$)
is the deflection over half the bearing length of bearing A
W is the load
b is the distance between the load and bearing B

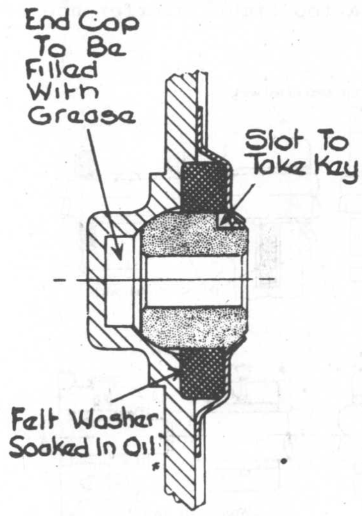


Fig 3

Experience indicates that the shaft deflection over half the bearing length should be less than a quarter of the diametrical running clearance, if problems arising from this source are to be avoided. In some applications the force causing the shaft to deflect arises from an out-of-balance load being imposed by the rotation (ie, shaft whip) and this must then be added to the static deflection calculated from equation (1).

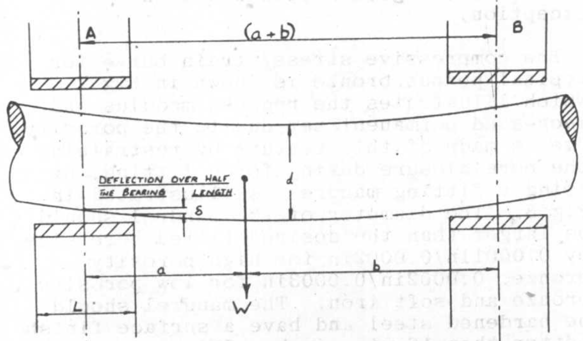


Fig 4 Shaft deflection

Force Fitted Bearings

This is the usual arrangement and several are illustrated in Fig 5. The mean interference between the bearing outside diameter and the housing bore should ideally be about 0.0015 times the square root of the bearing outside diameter in inches, plus 0.001 inches, ie,

$$I = 0.0015 \sqrt{D} + 0.001 \text{ in}$$

(where D and I are in inches)

The interference should not fall to below half or rise to above twice that given by this formula, which applies to rigid housings and applications which are to operate near normal ambient temperatures, and where the force fitting load does not exceed the axial compressive strength of the porous metal. Otherwise the designer is recommended to consult with the manufacturers who will advise accordingly Attention must be given to the stacking of tolerances to avoid the possibility of a too loose or a too tight interference fit.

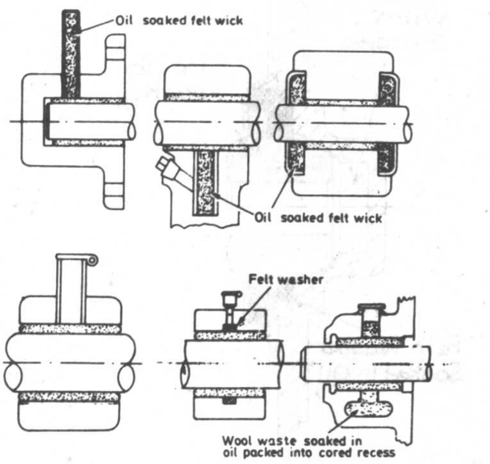


Fig 5

At fitting, the interference produces a reduction in the bore diameter. The ratio of the OD interference to the ID closure depends upon the dimensions of the bearing and the relative rigidity of the housing and bearing. For rigid housings the ratio of interference to closure is given as 'F' in Fig 6 against a function of the bearing dimensions, together with notes on exception.

The compressive stress/strain curve for typical porous bronze is shown in Fig 7 which illustrates the reduced modulus and increased permanent set due to the porosity. Use is made of this feature by restraining the bore closure during force fitting, by using a fitting mandrel as illustrated in Fig 8. The diameter of the mandrel should be larger than the desired fitted bore size by 0.0001in/0.0002in for high porosity bronze, 0.0002in/0.0003in for low porosity bronze and soft iron. The mandrel should be hardened steel and have a surface finish better than 16 microinches CLA.

By adopting this technique a bore diameter controlled to 0.0002in is achieved (irrespective of the stacking of tolerances). This situation is usually better than can be obtained by machining. The variation in running clearance between assemblies is therefore dictated only by the manufacturing tolerance of the shaft.

If it is required to increase the fitted bore diameter, the techniques described in the section marked servicing should be employed.

RUNNING CLEARANCE

The correct choice of running clearance is a compromise between a low running temperature (large clearance), the need for precise radial location of the shaft (eg, to control the air gap of an electrical motor), and the tolerance of noise with an out-of-balance load. In the absence of these or other conflicting features, Fig 9 gives general guidance on the choice of running clearance as a function of shaft diameter and speed. For self-aligning porous bronze bearings the clearances given by Fig 9 can usually be reduced, due to the absence of any misalignment problem and the slight increase in running clearance with rise in temperature.

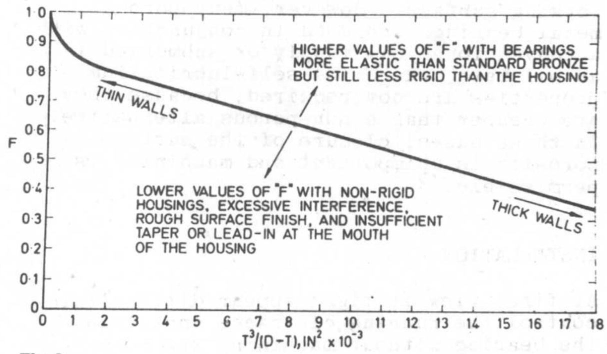


Fig 6

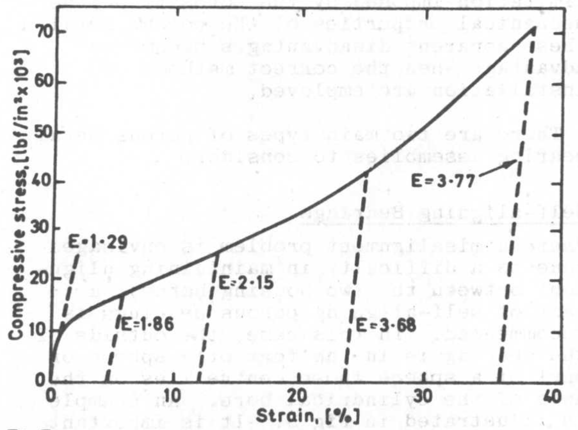


Fig 7

OIL VISCOSITY

General guidance to the choice of oil viscosity is given in Fig 10 (expressed in centipoises at 60°C) as a function of shaft velocity, adjusted for bearing load and operating temperature. It will be noticed that a lower viscosity is suggested for high shaft velocity, and low loads, and a higher viscosity for high operating temperatures, high loads and low shaft speed.

Unless specified otherwise, most porous metal bearings are supplied with an SAE 20 or 30 viscosity oil, which fits the middle of the range.

PERFORMANCE CHARACTERISTICS

There are three principal performance characteristics which the bearing designer looks for: first the life of the bearing; secondly the frequency of servicing; and thirdly the magnitude of the starting and the running torques. With porous metal bearings these factors are interdependent

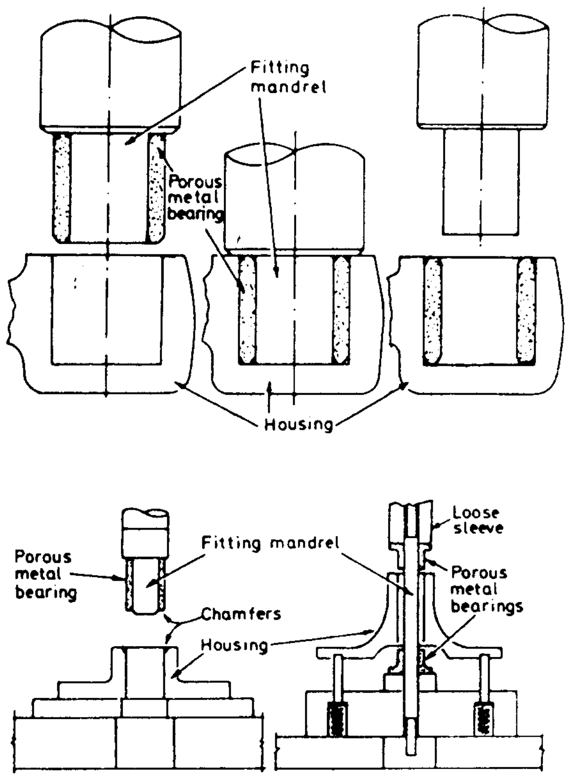


Fig 8 Force fitting of porous metal bearings using a fitting mandrel to control the fitted bore diameter and to achieve alignment of a pair of bearings

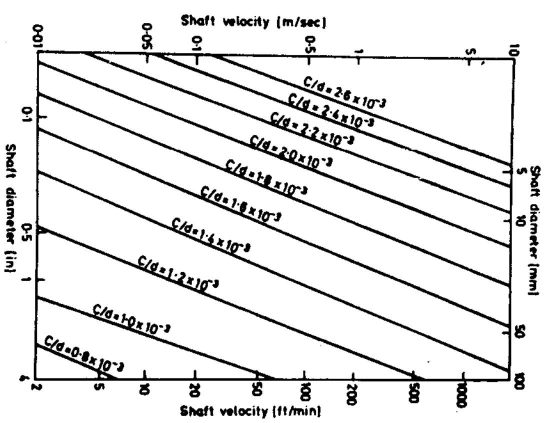


Fig 9

to some extent, and in common with the other oil lubricated bearings, the life of the porous bearing is closely associated with the life of the oil. Porous metal bearings have a similarity with oil drip-feed or felt wick lubricated bearings, in that the oil in the porous metal corresponds to the oil in the bottle of the drip feed, or the oil in the reservoir of the felt wick; and when these are empty, the bearing will commence to wear as soon as the oil in the clearance space has deteriorated. However, with porous metal bearings the situation is different, because oil which would otherwise be lost from the ends of the bearing is re-absorbed by the porous metal, to be filtered in the pores and used again.

The life of a porous metal bearing depends therefore on the life of the oil in terms of both quality and quantity. The life quality of the oil depends upon the running temperature. Fig 11 gives an approximate relationship between oil

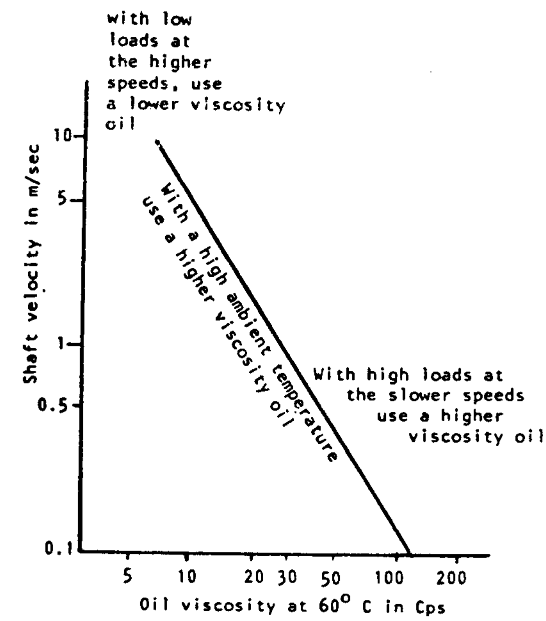


Fig 10 A general guide to the choice of oil viscosity according to the shaft velocity

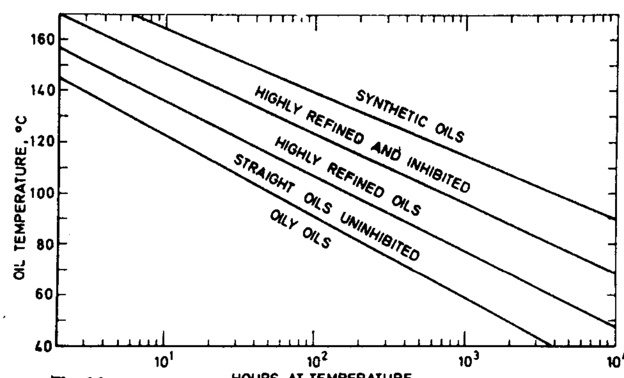


Fig 11

temperature and oil life. Roughly speaking a rise in temperature of 10°C halves the life of a given oil, in terms of oxidation. The temperature of the oil in a porous metal bearing is the sum of "the temperature rise due to friction", plus "the ambient temperature of the surroundings". With very high ambient temperatures, graphite and other dry rubbing bearing materials find wide application.

The life quantity of the oil depends on a more complicated function of the operating variables. When a porous metal is fully charged with oil, its powers of re-absorption are nil, and when it is completely dry it will absorb oil like blotting paper. Hence the oil re-absorption property of porous bearings depends upon how much oil the pores contain.

On the other hand, the forces which tend to draw oil away from the porous bearing depend upon the hydrodynamic oil pressure created by the rotating shaft, and the resulting centrifugal action on the oil film adhering to the shaft at the ends of the bearing. Thus high shaft velocity (rev/min x shaft diameter) increases the rate of oil loss, as does a lower oil viscosity and surface tension, caused by a higher running temperature. When the pores of the bearing are full, the hydrodynamic pressure is a maximum, and some oil is readily lost. As the pores become partially emptied, their power of re-absorption increases, and the hydrodynamic pressure reduces. The quantity of oil in the pores, therefore, tends towards an equilibrium balance which depends upon the operating conditions, and it approaches this equilibrium value at a decreasing rate. Fig 12 shows typical oil loss curves for a porous metal bearing running at different speeds and temperatures. As a safeguard it is recommended that porous bearings should be re-impregnated before the oil content has fallen to about 70 per cent.

As explained there are two criteria for determining the life of a porous metal bearing: oil quantity and oil quality; and before either of these lives are reached, it is receommended that the bearing be serviced by re-impregnation with new oil. However, if the oil life is, say, 4 000 hours, and the design life of the machine is only 2 000/3 000 hours, servicing is not necessary, and provision of re-impregnation need not be made in the design. For example, consider an electric razor which

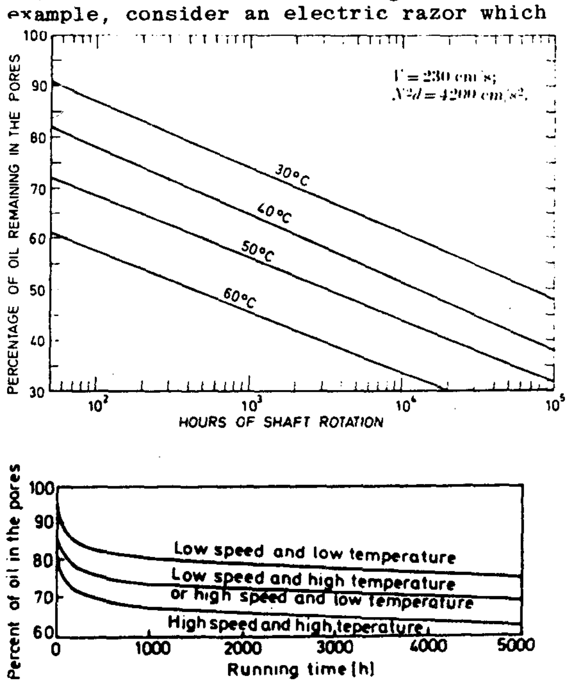


Fig 12 Typical oil loss curves for porous metal bearings

may be used twice a day for a period of 15 minutes, a machine design life of 16 years gives a bearing life of less than 3 000 hours.

As we have seen, the life is very sensitive to the running temperature. The running temperature depends on the load, speed, friction and the ability of the assembly to dissipate the frictional heat. An estimate of the bearing temperature rise can be made by equating the power input to the heat loss, ie:

$$W.V.\mu.\ 0.023 = K\ (\theta_1 - \theta_2)\ \ldots\ldots\ldots\ldots(2)$$

Where

W is the total bearing load in pounds

V is the shaft velocity in feet per minute

μ is the coefficient of friction

θ_1 is the running temperature in degrees Centigrade

θ_2 is the ambient temperature in degrees Centigrade

K is the heat dissipation characteristics in watts per degree Centigrade

A safe temperature rise is about 40°C, but as explained above, this depends upon the ambient temperature, and the required design life.

In still air, K has a value of about 20 watts per square metre of surface (or 0.013 watts per sq.in.). The effect of windage (or forced convection cooling) is to increase K as the square root of the air velocity; such that at about 80 mph, K is about 80 watts per sq.m. of surface.

However, these values are based on the whole of the surface being at temperature θ_1, where the rate controlling factor is the loss of heat by convection and not the thermal conductivity of the housing material. In considering the thermal conductivity of the housing and the heat dissipating surface area, it must be remembered that an interruption in the heat path (due, for example, to flat surfaces in tightly clamped contact) produces a considerably lower heat conducting path than uninterrupted metal. In cases of difficulty it is possible to determine experimentally the value of K for a particular bearing assembly by inserting a thermocouple into the bearing wall, and winding an electric heating element, either in the form of a hollow cylinder to fit

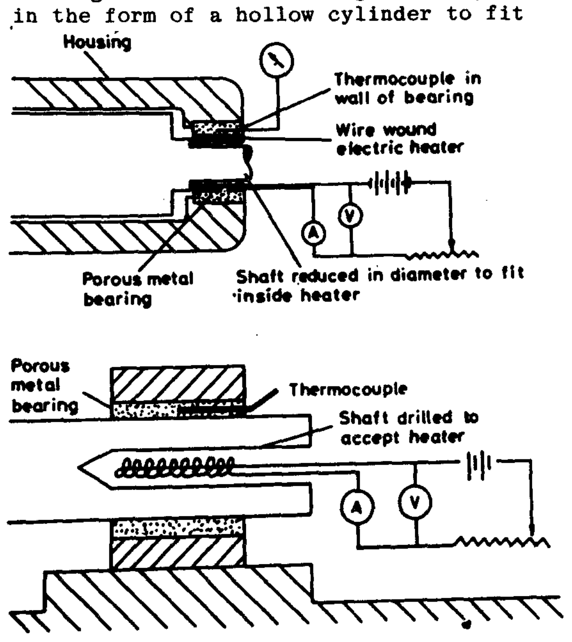


Fig 13

between the bearing bore and an undersize shaft, or by drilling the shaft and inserting a cylindrical heater inside the shaft. By measuring the volts and amps supplied to the heater and the equilibrium temperature of the bearing, the value of K (in watts per degree Centigrade temperature rise) can be calculated. See Fig 13.

A convenient parameter for assessing bearing duty is the pressure velocity product, or PV factor. This arises from a generalisation of equation 2 by making assumptions about the coefficient of friction, an acceptable temperature rise and a proportional relationship between K (the heat dissipation) and the projected area of the bearing (bore x length). The danger of these assumptions is not very large provided the length to diameter ratio is not more than about 1.5, and neither the load nor the speed are strongly predominant. Using this parameter (within its limitation and with a suitable clearance and oil viscosity) one can generalise as follows:

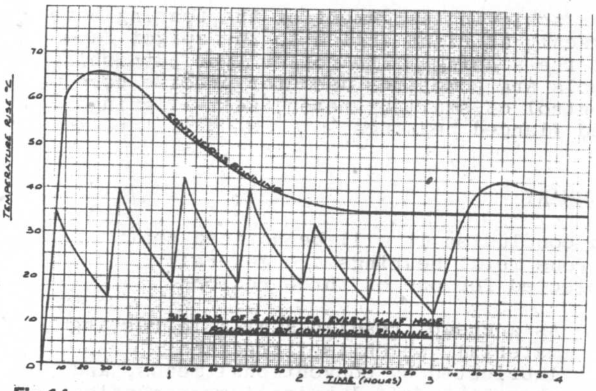
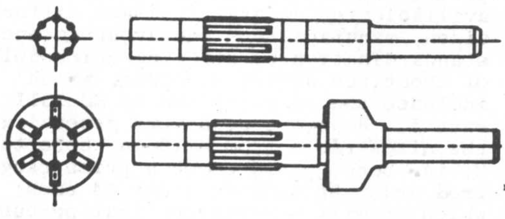
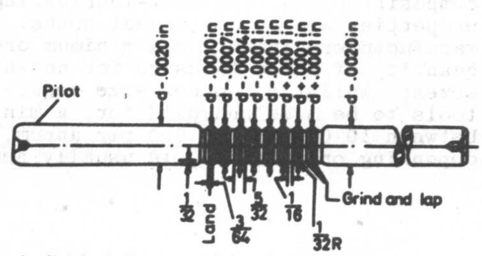


Fig 14

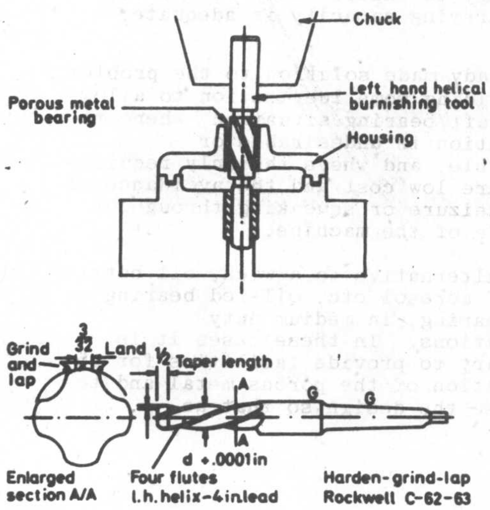


Roller type burnishing tool



d = finished diameter of bearing

Button type drift



Spiral burnishing tool with and without pilot

Fig 15

1. Below 10 000 lb/sq.in. x ft/min. (or 3½ kg/sq.cm. x m/sec.), performance difficulties are unlikely to arise.

2. Between 10 000 and 100 000 lb/sq.in. x ft/min. (or 3½ and 36 kg/sq.cm. x m/sec.), performance difficulties may arise unless the design variables are selected with care.

3. Over 100 000 lb/sq.in. x ft/min. (or 36 kg/sq.cm. x m/sec.), to ensure satisfactory performance, the design variables should be optimised, and checks made to confirm that an adequate life will be obtained.

To an engineer who is not conversant with the principles of bearing design, the above summary of performance prediction may appear to be very complicated; and because it is not possible to cover every aspect and certain exceptions to the general rules in this chapter, it is recommended that the manufacturers should be consulted in all cases of doubt.

SERVICING

It will be noticed from Fig 12 that the highest oil loss rates occur immediately after a fully impregnated bearing commences to run. In practice this is a good thing to happen, because in the first stages of running a new bearing with a new shaft, the running-in process (which removes local irregularities and the peaks of surface roughness) produces a small quantity of debris, which is flushed away by the initial oil loss.

Remembering that the essential design requirement is to minimise the running temperature, it is recommended that the running-in should be carried out under a reduced load. However, in many cases this is not possible, and an alternative method is to run for a number of short periods, none of which are long enough to permit the bearing to reach its equilibrium temperature. Fig 14 illustrates the method and the reduction achieved in the actual bearing temperature. In effect one is using the heat sink of the housing etc to get rid of the additional friction heat which is caused by the partial boundary conditions which exist during running-in.

If, after fitting a new bearing by the methods given in Fig 8, it is required to increase the bore diameter for any reason, one of the methods given in Fig 15 should be used. The plain burnishing tool or button drift is suitable for increasing the bore by about 0.001 in. per inch by drifting on a fly press. The spiral burnishing tools (which may have a pilot) are suitable for use in a lathe, and the piloted arrangement is particularly useful for achieving alignment of the bearing bore exactly at right angles to the spigot face of the housing. The roller type burnishing tools are sometimes used for precise bore control of small diameter, self-aligning and flanged bearings.

SUMMARY

In conclusion it will be noticed that porous metal bearings are generally selected for one of three reasons:

1. As a cheap, readily available bushing material, which is fitted and lubricated by conventional means, and where the self lubricating properties arising from the porosity are not required. In these cases it is only necessary to ensure that the static load carrying capacity is adequate.

2. As a ready made solution to the problem of providing some lubrication to a low duty shaft/bearing situation, where re-lubrication is undesirable or impossible, and where the only requirements are low cost and the avoidance of wear, seizure or squeaking throughout the life of the machine.

3. As an alternative to a wick, oil bottle, grease, aerosol etc, oil-fed bearing or ball bearing, in medium duty applications. In these cases it is necessary to provide facilities for re-lubrication of the porous metal and to optimise the design so that an acceptable life is achieved. Much of the information given above applied to this situation.

It is recommended that the design engineer should, wherever possible, consult the porous metal bearing manufacturer concerning the specific details of the proposed application.

PRICES AND ORDER QUANTITIES

The manufacturers must also be consulted on the question of price. The porous metal bearing industry is strongly orientated towards mass production, and hence the price of a given porous metal bearing reduces rapidly with the quantity in the production batch. Whilst tools are available for several thousand different sizes, manufacturers endeavour to encourage standardisation by offering quick delivery of about 500 different sizes, which includes the sizes listed in BS 1311 : Part 5. However, as a very general guide, the sizes range from 3/16 in. bore to 2½ in. bore, would cover a price range from under 5p each to under £4 each; from which it will be gathered that porous metal bearings are competitive with machined solid metal bearings of the same size and composition, and the self-lubricating properties are an additional bonus. Most manufacturers insist on a minimum order quantity of a few thousand for non-standard sizes; whilst for a new size requiring tools to be made and paid for, a minimum of between 10 000 and 50 000 per annum, depending on the size, is usually necessary.

Split rolling bearings – saving on capital and whole life costs

P W R BLAKE, MA(Cantab), CEng, MIMechE and G D HICKMAN, CEng, MIMechE
Cooper Roller Bearing Co Ltd

The Cooper split roller bearing is the ideal type for radial assembly in difficult positions and for ease of assembly in any position. Only a soft hammer, screw driver and the correct socket head cap screw keys are required for its fitting. It is also universal in its application for both slow and fast moving equipment. Some interesting applications are described below.

As is well known, the Cooper bearing is a split parallel roller bearing with machined joints at an angle to the shaft axis for progressive rolling of the rollers over the joints. The fractured joint has not been used for the standard bearings because of the possibility of starting hair-line cracks at the joints, see Fig 1. The inner race is held down to the shaft with separate low carbon steel clamping rings, using reasonably large socket head cap screws to prevent slip or creep. The clamping rings also locate the split inner ring on the shaft much more accurately and securely than shoulder screws can in machined holes (however accurate), which are subsequently hardened. Also clamping rings which are integral with the split inner ring are not able to pull down the inner ring fully to the shaft - compared with separate clamping rings where the smaller cross sections of the two components permit interference fit, effectively as with solid roller bearings. This is especially relevant should the split inner race be fitted to a shaft below h6 tolerance.

The Cooper split roller bearing is externally aligning which maintains the concentricity of the seals that are fitted in the cartridge and which also holds the outer race halves together.

The parallel roller bearing is the best type that there is for accommodating correct axial positioning. When shaft expansion takes place the rolling elements take up their correct position relative to

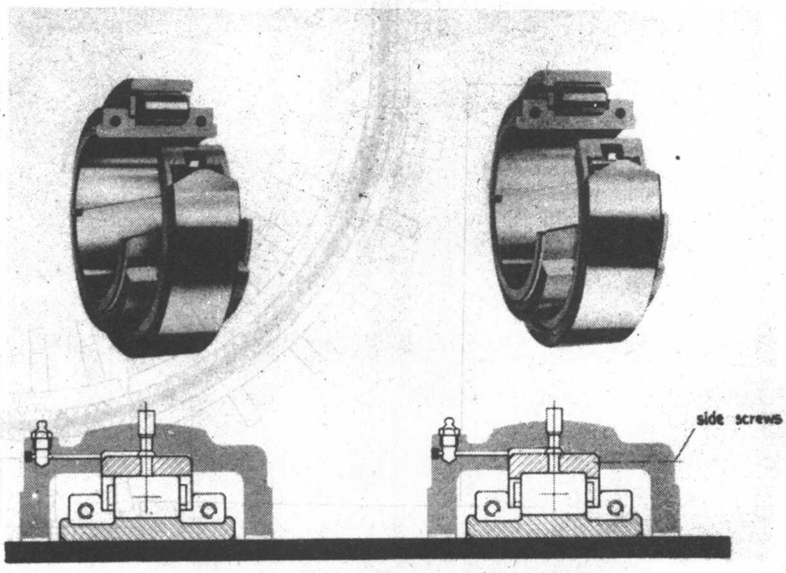


Fig 1 The Cooper split roller bearing in swivel cartridge

the outer race exerting negligible axial thrust. On the other hand the double row spherical bearing can only position itself by the outer race sliding in its housing for which a correct fit is essential; but is difficult to achieve through a range of temperatures and unfavourable, though not uncommon, extremes of limits. For example the coefficient of expansion of hardened steel is greater than that of cast iron. Therefore the higher the operating temperature of the shaft and the greater the induced axial expansion between the fixed and expansion bearing, the greater the tendency for the expansion outer race to stick in its housing and exert thrust. Any resistance to axial sliding causes both fixed and expansion bearings to run on the outer row of rollers, thereby reducing the capacity of the bearing substantially. This tendency must increase in the case of a split outer race joined together with small screws as the inclined faces intrinsic to the geometry produce substantial outward acting radial forces from any thrust applied.

Referring to Fig 1, the fixed GR bearing has a lipped outer race and inner race clamping rings with hardened faces for the rollers to run between and to enable the bearing to take axial loads. The split outer race is also accurately registered axially by side screws in the cartridge as shown.

Lubrication of the fixed and expansion bearing is normally carried out by a grease gun through the grease nipple shown and directly through a hole in the top half of the outer race onto the rolling surfaces. By this method fresh grease is enabled to expel the old grease from the centre of the bearing outwards. The cartridge has a spherical diameter and fits into an outer casting which is bored spherically to enable correct alignment to take place during assembly. The peg shown protruding from the cartridge engages in a recess in the outer casting and prevents the cartridge from turning in the pedestal under heavy radial load whilst allowing alignment of up to 2 to 2.1/2° in any direction.

The first application discussed below illustrates the versatility and value of the split parallel roller bearing. It is true to say that without it the method adopted for splitting the rolls could not have been carried out nearly so economically because of the very heavy loads involved.

Case 1

The continuous casting of steel has been steadily increasing over other methods in the last two decades until now it accounts for most of the steel output of the major industrialised countries. The majority of the steel made by this process is rolled directly into slabs or billets thereby saving a reheating cycle and the "discard" from the cast ingot, which always suffers from "piping" when it solidifies. The slab caster produces rectangualr slabs to very accurate dimensions, but as the size of slab increases so does the load on the driven and hydraulically loaded guide rollers. However, the diameter of these rollers cannot increase with slab size because of the need to support the near molten slab at close centres. There is then a tendency for long rollers to deflect in the centre and for the slab to lose its accuracy. See Fig 2 for overall view of a slab caster. From the tundish which is fed by a BOF steel converter, 2 or more strands pour down through the water cooled mould onto a dummy bar, which starts to guide the strand through the rollers (the dummy bar is subsequently sawn-off and re-used). From the bottom of the curved mould, the first drive and guide rollers start. They are usually water cooled internally and the strand is sprayed externally, so the environment for a bearing on a roller, which may be supported in more than one position is very tough indeed.

Fig 3 shows a typical arrangement of a split parallel roller bearing in a water cooled housing. The bottom half outer race is spherical on the OD for alignment purposes, but the rollers run on the machined surface of the housing in the unloaded zone of the bearing in order to keep the overall height of the housing at least 10mm below the roll surface and the semi-molten slab.

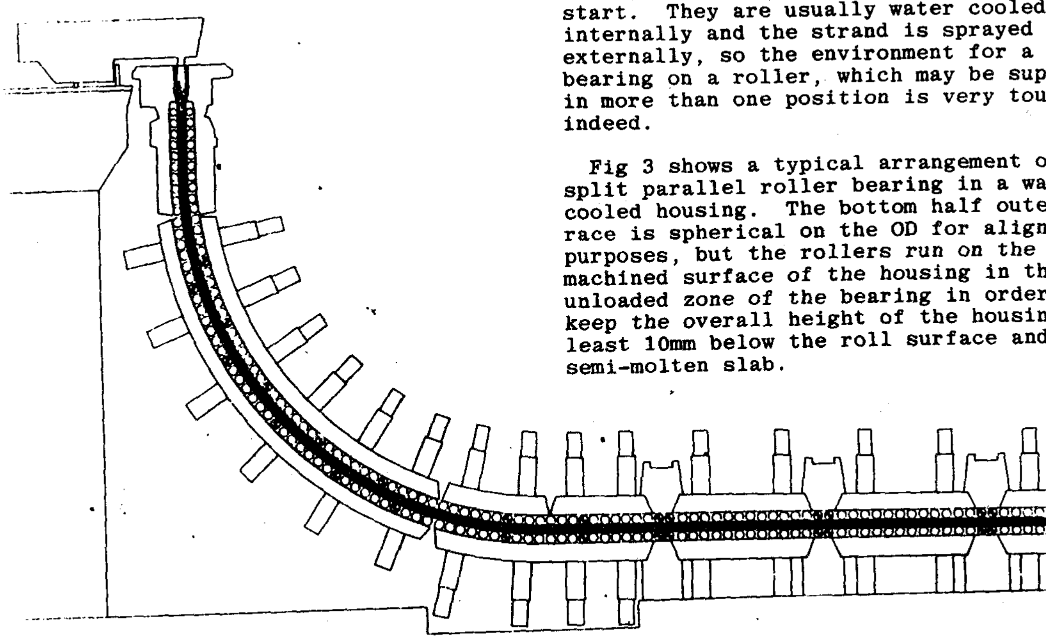


Fig 2 Typical arrangement of slab casting machine