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Self Contained Bearing Assemblies

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1. INTRODUCTION

Self contained assemblies using plain (hydrodynamic) bearings are characterised as single units each incorporating a journal bearing, sometimes associated with a thrust bearing, a lubricant circulating system and a means of dissipating the heat produced by the shearing and turbulence of the oil in the system.

The commercial attractions of such assemblies to capital equipment manufacturers are those that are common to all the items supplied by specialist manufacturers:

a) cost saving (in not having to design a lubricant pressure feed and cooling system),
b) improvement in reliability (that should result from no dependence upon external services),
c) design by engineers with specialist knowledge outside the capital equipment manufacturers expertise,
d) economies of scale both in the design process and the manufacturing process,
e) inclusion in the design of standard replaceable components for ease of maintenance from a stock source.

In other words, the manufacturer can buy a standard design more cheaply than he can design and manufacture one himself, with the knowledge that the design is proven and tested and spares are easily obtainable.

Historically, self contained bearing assemblies have been available for about 50 years but, with the exception of ring oil lubricated units, the product has had limited acceptance until relatively recently because of the limited operating envelope of early units, a reputation for poor reliability and the previous lack of incentive at a time when bearing design and manufacture was carried out by the capital plant manufacturer himself. However, with the economic climate favouring the move towards purchase of ancillary components and with demands for high reliability and lower cost, there is an obvious advantage in using complete assemblies rather than simple bearing components from specialist manufacturers. There has therefore been a steady increase in demand for such units in recent years, and this trend is likely to continue.

2. BEARING CONFIGURATION

Design for minimum power loss is a major consideration with self contained assemblies since, by their very nature, they have a limited heat dissipating capacity and a smaller oil volume than units with an external pressurised oil supply and cooler system. Consequently, because heat dissipation is limited, it is the amount of power absorbed by the bearing, and dissipated as heat, that controls the operational envelope of the unit.

The bearing configuration will depend upon whether the unit needs to react axial as well as radial loads, whether the shaft axis is vertical or horizontal, whether the load is such as to place a simple cylindrical bearing in an unacceptable part of its operating characteristic and whether the dynamic characteristics of the rotor are such as to demand specific bearing stiffness and damping characteristics in order to limit dynamic perturbations of the shaft.

The most significant factor in bearing power loss is the nominal bearing diameter, over which the bearing designer has little control since it is normally determined by the customer as a result of mechanical design considerations. Thus, the choice of bearing type and axial length are the only variables at the discretion of the bearing designer that will have a significant effect upon the bearing power loss.

However, even though there may be considerable design constraints, a large number of options often remain available to the designer.

2.1 Journal Bearings

Fig. 21.1 shows the range of journal bearing designs available to the engineer, together with their attributes including relative stiffness, damping and power loss. It will be noted that the simple cylindrical bearing has the lowest power loss, and there is approximately a 40% difference in the power absorbed by the various designs at the same speed, load, and oil inlet temperature. Thus, before choosing a bearing which generates a higher loss, the designer should ascertain that the attributes justify the choice.

![Fig. 21.1 Operational attributes of journal bearings with different profiles](image-url)
2.2 Thrust Bearings

With regard to thrust bearings, the choice is normally between a plain thrust face, a taper land thrust bearing or a tilting pad thrust bearing.

In the case of the plain thrust face there is a limited load capacity, in the range from 0.3 MPa to 0.5 MPa, and operation is only possible because the surfaces are not perfectly flat. Thus, the plain thrust face is normally used where the axial load is nominal, or very low, and location only is required. It has the advantage of simplicity, since sufficient oil is usually available from the journal bearing. As the bearing wears (providing the wear is due to fine dirt particles rather than excessive load) there is no change in load capacity.

With the taper land thrust bearing the film thickness is considerably greater, and specific loads up to 2 MPa may be carried before excessively thin oil films and high temperatures lead to unreliability.

The bearing with the highest load capacity is the tilting pad thrust bearing, normally with a specific load limit of approximately 4 MPa which is sometimes restricted to 2 MPa to ensure wiping does not occur at start-up. Tilting pads are less prone to change in load capacity with wear and, where standard units can be used, they may be less expensive to fit than a taper land bearing because of the high cost of machining a taper land face.

Where thrust is a significant part of the total load to be borne, it normally accounts for a disproportionately high percentage of the power loss. This is because power loss increases very rapidly with mean sliding speed, and since the thrust collars extend radially from the shaft, the mean sliding speed of the thrust bearing is always greater than that of the associated journal. Also there is usually a parasitic power loss associated with churning of the oil by the thrust collar, which does not occur to any significant extent in journal bearings.

Fig. 21.2 gives a comparison of power loss in various thrust bearing configurations to suit a 100mm shaft, with a maximum thrust collar diameter of 200mm.

For location and very light loads economic considerations may dictate the use of a plain thrust face, but for higher loads, where conventional taper land faces could be used, there is a case for considering a small number of thrust pads as part of a ring of tilting pads, giving rise to lower power loss and increased reliability (see Fig. 21.2). This may also allow a small collar...
3. LUBRICATION ARRANGEMENTS*

The simplest arrangement is an oil bath surrounding a vertical shaft as in Fig. 21.3.

Vertical bearing assemblies have been used widely in the nuclear industry and substantial thrust loads (up to 200 tonnes) have been carried. There are several different methods of cooling, including water tube, water jacket, fan assisted air, air and circulated oil.

A selection of typical lubrication arrangements used in self contained assemblies for horizontal shafts is shown in Fig. 21.4.

Such systems can be divided into those with and without pressure circulation, which imparts little advantage to the bearing itself provided there is enough oil to meet the hydrodynamic requirements. However, the ability to circulate oil under pressure makes a considerable difference to the choice of cooling system to be designed into the unit and the ability to fit effective filtration.

Ring oil lubricated bearings have been developed into a range of flange mounted and base mounted horizontal bearing assemblies for fitting to electrical plant, fans, blowers, pumps, etc. They offer a simple, self contained bearing for low and moderate speeds; their range of operation can be extended by fitting water coolers, oil circulating pumps or an external pump supply system.

Fig. 21.5 shows the basic arrangement of an end flange mounted ring oil bearing assembly from the Glacier HSR range (suitable for shaft diameters from 80mm to 300mm). Plain thrust faces, taper land or tilting pad thrust bearings may be used in the assemblies to carry axial loads.

Operation with ring oil lubrication is limited by the increase of the bearing oil requirement with speed, whereas the efficiency of oil transfer from the oil reservoir to the bearing inlet gutterway reduces at higher speeds. Consequently, as speed increases a point is reached at which insufficient oil is supplied for the full hydrodynamic oil film to be maintained. A small amount of starvation will not necessarily have any deleterious effect but, as speed increases further, the effect of oil starvation will result in excessive bearing surface temperatures and high power loss, followed ultimately by seizure.

Scoop and disc lubrication are also used to circulate lubricating oil around self contained assemblies. In the latter case the periphery of a disc, attached to the shaft, dips into the oil reservoir and subsequently throws oil by centrifugal force into an upper chamber, from which some drains down to the inlet gutter of the bearing. It is the authors' experience that such systems are very sensitive to speed and detail design. Thus, a disc system may deliver a satisfactory amount of oil at 2000 rev/min but may completely stop delivering oil at 3000 rev/min. This phenomenon is due to the disc windage, which causes a depression of the oil pool and consequently, at a critical speed, failure to wet any part of the disc. It is possible to extend this speed with baffles, but the system is not amenable to simple design procedures.

Fig. 21.4 Schematics of different types of lubricant circulation arrangement for horizontal units

*AE PLC GB Patent Nos. 0026765 and 2079385, USA Patent Nos. 4 398 348 and 4 445 529 and corresponding patents and patent applications in other countries.
and, consequently, the available oil flow can only be established by test. This is not a satisfactory engineering situation, and failure to understand the sensitivity of such systems has led to bearing failure in the past.

Scoop lubricated systems normally consist of a rotating channel which holds a volume of oil in its periphery. The lowest part of the channel is below the oil reservoir level, and is thus maintained full of oil. The oil is removed by a scoop at the top of the channel, directing the oil to the bearing inlet groove. This system requires a relatively large amount of space and is considerably more complicated than the disc lubricated system, but is not as critical.

The ability of the viscosity pump to generate pressure, and thus need external coolers, filters or other bearings, distinguishes it from other types of oil circulator for self contained assemblies. The ability to fit oil filtration is of particular importance, since dirt in bearing systems is the main cause of unreliability, resulting in wear and limiting service life. Even when great care is taken to maintain a clean system, dirt scoring is usually found when continuous filtration is not used.

Circulation by viscosity pump is normally limited to horizontal thrust and journal units, but could be applied to vertical thrust and journal units or journal only units. In this system there is a ring with a dammed groove mounted around the thrust collar periphery. Ports either side of the dam act as oil inlet and outlet areas. The oil is dragged around the groove by the viscous shear force and issues from the exit point under pressure.

Early designs of viscosity pump suffered from rapid wear at the side lands of the pump body, and were known to have priming problems and poor pressure generating characteristics. However, by recognising that the side lands need to be treated as narrow hydrodynamic bearings, they can be designed to give negligible wear.

The pressure characteristics can be considerably improved by fitting a diffuser zone in the exit port. This will convert the kinetic energy of the flow stream into pressure energy, thus producing a simple pump with pressure flow characteristics not dissimilar to those of a centrifugal pump. As can be seen from Fig. 21.6a, considerable pressure can be generated. Of great importance to reliability is the ability of such a pump to give characteristics which are only marginally degraded by significant wear. Fig. 21.6b shows the pressure flow characteristics with up to 0.6mm of wear.

The one major problem with horizontally mounted pumps is that of priming. Fig. 21.7 illustrates the design problem associated with priming small viscosity pumps, due to the need for several centimetres of pump groove to be wetted by oil in the drain down position. It is clear that the allowable difference between maximum oil level (i.e. when oil leaks out of the shaft seal) and minimum oil level (i.e. when the pump fails to prime) is acceptably small if a normal sump condition exists with a considerable tolerance on oil level.

One solution to this problem is to fit a secondary tank to hold the oil at the optimum level for priming. This arrangement is shown in Fig. 21.8. The secondary tank is continuously fed with excess oil by an ejector using the pressure energy of the return oil.
The height in the tank is determined by a weir which allows the excess oil to return to the main oil reservoir. The ejector jet also controls the system pressure. The addition of these passive refinements greatly increases the total system reliability to the extent that it exceeds the external pressure fed conventional bearing system, because of the high reliability of the oil circulation system which is relatively insensitive to wear.

This system may not be the most cost effective for lightly loaded systems or units of low rotational speed. However, where thrust and radial loads are required at moderate industrial speeds, such a system is capable of meeting the full envelope of a pressure fed bearing which would normally require an external oil supply, its envelope often being limited only by the cooler fitted.

4. COOLING

The choice of cooling will be between air cooling (either forced or convection), cooler coils in the sump or external coolers. External cooling can include heat pipes as well as the more conventional oil-to-water heat exchangers.

With air cooled and sump cooled units, the limited ability of the unit to dissipate heat usually reduces the potential total operating envelope of the unit. Consequently, the efficiency of the finning on the system controls the operational envelope of air cooled units, as does the heat transfer of the heat to the water in the case of sump coolers. Generally, water cooled coils in the sump tend to make only small improvements to the operational envelope, because of the poor heat transfer coefficient between near-stagnant oil and the cooler surface. Extended surface tubing (gilled tubing) can even be counter-productive, since it traps cool viscous oil and effectively insulates the cooler from the sump oil. The most effective method must be that in which pressurised flow is available to enable conventional heat exchangers to be used, but this method requires enough pressure and flow to allow the coolers to operate in an efficient part of their working envelope.

5. FUTURE TRENDS

It is expected that the capital plant manufacturer will demand increased reliability, a larger working envelope in terms of speed and load for a given size and, of course, lower costs.

In general it is the limited cooling ability that limits the operational envelope of many designs of assemblies. Thus to increase the envelope it is necessary either to reduce the power absorbed by the bearing or to improve the cooling from the unit. It is therefore worth reviewing the actions possible in each of the areas considered previously.

5.1 Bearing configuration

The shaft diameter is usually fixed by the plant designer as a result of mechanical considerations and, as is shown in Fig. 21.1, only small reductions in power loss may be achieved by changes in bearing design if conventional specific load and normal lubricants are used. If self contained assemblies are to carry higher loads at higher speeds, then bearing operating limits must be extended.

The operation limits will be:

a) the maximum allowable bearing surface temperature
b) the minimum allowable oil film thickness

c) the lubricant temperature which causes an unacceptable rate of degradation.

New bearing surface materials can be used to overcome limitations a) and b) and synthetic lubricants may be used to overcome limitation c).

A change of bearing surface material to one that does not warp, as whitmetal, and can operate at a higher temperature, will extend the working envelope to the temperature limits of the lubricant and the minimum acceptable film thickness. The bulk lubricant temperature limit may be extended from 60°C to, say, 120—150°C by the use of synthetic oils, since they do not degrade at the same rate as mineral oils.

The alternative material must have excellent seizure properties if the mode of damage is not to change from a "soft wipe" to a catastrophic seizure. This largely prohibits metallic bearing lining materials, and suggests the later generation of high temperature engineering polymers that have extremely good anti-seize properties and are able to operate at temperatures as high as 250°C.

5.2 Lubrication arrangements

Clearly, the advantage of the viscosity pump is its ability to supply sufficient pressure to circulate oil through coolers and filters. This must cause them to be more generally acceptable for units that need to withstand significant axial loads, particularly as careful design can make them insensitive to small amounts of wear. The ability to circulate oil at high velocity through conventional heat exchangers prevents such units from being limited to an operational envelope different from that of an externally pressure fed bearing, except at very high speed when viscosity pump port throttling tends to limit the flow.

For moderate speed applications where there is no thrust load, the ring oil lubricated bearing will meet many engineering requirements. The speed envelope will be extended as the ring design is improved to give the required oil quantity at higher speeds.

5.3 Cooling

In order to remove more heat from a system, either the overall heat transfer coefficient or the temperature difference has to be increased. Clearly, if a change in material and lubrication allows operating temperatures of, say, 120—150°C, the thermal dissipation of existing units will be trebled. If this is coupled with a lower heat generation due to optimisation with respect to power loss, then clearly the operational envelope can be greatly extended.

6. CONCLUSIONS

With increased demand for more efficient bearings, in terms both of minimum power loss for a given load and size and of cost, there will be a need for improved designs. There may also need to be an acceptance of revised levels for safe operating conditions, such as specific load or maximum operating temperature, where these can be shown to give equal or better reliability with improved characteristics.

The Assembly Bearings Division of The Glacier Metal Company has the experience and expertise to meet the
challenge of providing both conventional and novel designs of bearing assemblies to meet future demand. It is familiar with the special problems associated with both horizontal and vertical assemblies, whether for the nuclear industry with its demand for high reliability, or for operation in the third world with the ruggedness and simplicity demanded. Modern design techniques enable optimisation of operating characteristics to ensure safe operation and long life.

We therefore welcome enquiries for conventional designs of bearing assemblies, as well as enquiries from progressive plant designers who may have unconventional specifications demanding novel solutions.