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Advances in oil lubricated plain bearings for high speed applications

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SYNOPSIS Oil lubricated tilting pad plain bearings are widely used in many pumps, fans and compressors. More demanding equipment duties have led in turn to significant advances in the performance capability of tilting pad bearings. This is manifested both in the form of enhanced standard ranges and in special designs which are regularly produced to satisfy ever more demanding customer specifications. This paper presents some of the results of recent work, in experimental research and in engineering design, chosen to illustrate what is now possible. Reference is made to a number of specific designs which include the very large diameter, high speed fan bearings for the European wind tunnel currently being built in Cologne.

1 INTRODUCTION

One of the consistent trends in the development of all plant is the use of more onerous operating conditions as system designers seek to maximise the output from all items of rotating equipment. More demanding equipment duties impose, in turn, new demands on the bearings used to support the shaft and absorb axial and radial loads. Oil lubricated, tilting pad, thrust and journal bearings have been an important design option for the greater part of this century and their use is common in a wide variety of equipment including many pumps, fans and compressors. In recent years, the performance capabilities and cost effectiveness of fluid film bearings have been extended by a combination of enhanced theoretical understanding, experimental work and innovative engineering design.

The purpose of this paper is to present some of the relevant results from current tilting pad bearing research and design development in order to give a greater appreciation of the considerable potential now available. In the sections which follow horizontal and vertical shaft systems are considered separately. Examples of experimental and operating data are given where appropriate and reference made to a number of specific bearings designed to satisfy particular extreme requirements.

2 HORIZONTAL SYSTEMS

2.1 Radial Loads

Horizontal shaft plain bearings are required, like vertical bearings to accommodate radial and axial loads which act perpendicular to and along the shaft respectively. In many instances the principal radial loads are due to gravity. Radial forces are also generated by operation of the system which is being supported. In some cases, such as pelton wheel turbines and certain gearboxes, the radial forces are very considerable and substantially greater than the forces due to gravity. The direction and magnitude of radial loads of this sort are governed by the geometry and nature of the system in question and may vary according to operating conditions. By contrast radial loads due to gravity are constant and act vertically downwards. Additional radial loads can also be caused by out of balance forces which rotate with the turning of the shaft.

There are a number of design options for horizontal shaft radial or journal bearings. The range of choice includes simple circular journal bushes, lemon bore and lobed bearings, bearings constructed with offset halves and tilting pad bearings. The order of these bearing types represents an increase in complexity and cost to provide rotor stability even at very high speeds and light loads.

Tilting journal pad bearings are the most sophisticated in the family of radial bearings and traditionally their main use has been for high speed applications where they can guarantee stability at all speeds and for applications in which the direction of the radial load varies or cannot be predicted with any accuracy. Recent experimental work (1) has extended the range of operation for these bearings while advances in manufacturing methods including the use of computer controlled machine tools, have ensured that they provide a cost effective design option. In the experimental work referred to, a 200 mm diameter, 80 mm axial length, tilting journal pad bearing fitted with centre pivoted pads, figure 1, was operated satisfactorily over a wide range of duties up to a maximum specific load of 4.14 MPa applied at a speed of 10,000 rev/min (equivalent to a sliding speed at the bearing surface of 105 m/s). The experimental programme was designed to isolate some important design parameters and determine their effect on bearing temperatures and energy consumption. It was found that the orientation of the bearing with respect to the direction of the applied radial load has a significant impact on the maximum temperature recorded in the bearing dependent on whether there is a pad directly in line with the load or whether the load line passes between two adjacent pads. Maximum pad temperatures recorded in the latter, load between pads case, were some 15^{o}C to 20^{o}C less than in the former.

Some of the experimental results for the load between pads case are given by figure 2 which shows maximum bearing temperature plotted against rotational speed for a number of applied loads. This figure shows one of the most interesting results from the experimental work which is an inflexion in the relationship between maximum bearing temperature and speed which is thought to be associated with a transition from laminar to turbulent lubricant flow in the bearing and hence to more efficient cooling of the journal pads. Transitions of this sort are common in high speed thrust bearings but are not so well documented in journal bearings.

In these journal pad experiments, lubricant was supplied to the bearing from an external source. The amount of oil delivered was adjusted for each combination of speed and load so that the mean oil temperature rise of the volume passing through the system between supply and drain was 17^{o}C which is a standard figure for temperature rise in many commercial applications. Subsequently the amount of oil delivered to the bearing was reduced and it was found that large reductions in volume were possible, leading to useful energy savings without compromising reliability. Typically, at 10,000 rev/min, a 50% reduction in oil flow from the standard requirement leads to a drop in power absorbed of about 20%.

As indicated earlier, the results referred to above were obtained using centre pivoted journal pads which are appropriate for systems which have an equal requirement for rotation in both directions. Many systems have a single direction of rotation or only operate in reverse under exceptional or run down circumstances. In these cases offset pivot journal pads can be used to provide a significant reduction in bearing temperature. Experimental work, aimed at extending the range of offset pivot pad bearings is continuing and it is planned to publish some of this work in due course.

2.2 Axial Loads

The axial loads generated in horizontal shaft systems may occasionally be very slight and bearing duties confined to providing restraint. In most pump, fan and compressor applications, however, considerable axial forces are generated and must be supported. Tilting pad thrust bearings are a common solution to the problem because of their ability to withstand a wider range of duties for a given bearing size. Figure 3 shows a typical thrust bearing assembly arranged for flooded and low loss lubrication. If large misalignments are a possibility then equalised thrust bearings are a design option. The machine designer can overcome many of the problems associated with flexible shafting systems by combining an equalised thrust pad bearing with a pivoted pad journal bearing in which the ends of the pivots are relieved to provide a self aligning capability in the axial direction.

Since the diameters of thrust bearings are inevitably larger than their associated journal bearings, the sliding speeds at which they operate are consequently higher. Recent experimental work has investigated sliding speeds up to 130 m/s (2). Figure 4 shows maximum pad temperature plotted against sliding speed (measured at the mean pressure diameter) for the following thrust bearing.

Outside diameter	229 mm
Inside diameter	148mm
No. of pads	11

Significant benefits that can be derived by using offset rather than centre pivoted thrust pads. If there is a predominant direction of rotation substantial reductions in pad temperatures as shown on Figure 5 (3) are possible without any increase in power loss (4). Offset pads are not unidirectional in operation as is sometimes believed. Reverse rotation during pump rundown for example can easily be accommodated by offset pads. Figure 5, shows the temperatures of offset and centre pivoted pads at various sliding speeds and specific pad pressures and shows how offset pivoted pads running in reverse generate temperatures not much higher than centre pivoted ones.

If pad temperatures greater than 120^{o}C to 130^{o}C are encountered for either journal or thrust bearings, then traditional whitemetal lined bearings can be supplanted by other alloy linings, for example, copper-lead which allows safe continuous operation up to 140^{o}C. For extreme conditions of load and speed the technique of active cooling of the pads using recirculated oil in conjunction with a backing material such as copper chrome alloy is a practical design solution.

3 COMBINED BEARINGS

In many cases horizontal shaft bearings are required which sustain considerable combined radial and axial loads. This section contains three examples of special bearings designed to accommodate very particular and demanding customer requirements.

3.1 Large Diameters

The thrust and journal bearings so far described are not simply confined to small shaft diameters less than 300 mm. The same principles are applicable to much larger diameters. An impressive example of this is a bearing recently developed for the compressor of the European Transonic windtunnel currently under construction in Cologne. In this application the horizontal shaft rotor of a 50 MW compressor is supported by three tilting pad journal bearings one of which also includes a tilting pad thrust bearing, capable of accepting axial thrust in either direction. The relevant data for the thrust and journal bearing is as follows:

Shaft Diameter	630 mm
Speed	60 to 885 rev/min
Thrust Collar diameter	1020 mm
Journal Load	190 kN
Thrust Load	1170 kN max

The bearing is illustrated, in Figure 6 which shows one of the thrust faces and the journal pads. Incorporated in the design are many of the features such as high pressure jacking, equalised thrust pads self aligning journal pads, oil seals for the containment and control of oil flow which are necessary for the total integrity of the installation.

3.2 Special Self Contained Bearings

In the previous example oil was supplied under pressure from an external source. In many cases it is desirable or essential that self contained bearings are used.

The objective for designers of self contained bearings is to supply a product to a performance specification which is able to provide itself with a continuous supply of cool oil for the bearing surfaces. The bearing has to be able to carry out from within its own engineered resources the functions of circulating and cooling an amount of oil fixed within the confines of the bearing. There are a range of techniques available to assist in fulfilling these functions. Oil circulation is often achieved for example by fixed or loose rings dipping into the sump and conveying oil from these to the top of the bearing. In many cases water is available as a cooling medium and coolers fabricated from specially designed, high heat flux tube, may be introduced into the bearing casing.

The potential for self contained bearings is illustrated by the following examples which show how the development of this type of bearing has enabled two very special duties to be accepted.

3.2.1 Cooling Using Heat Pipes

The emergency feed and cooling water pumps in nuclear power stations are devices which may never need to be called upon during their service life. When they are required, however it is essential that they operate in a completely reliable way with the minimum of outside services. The specification for the thrust bearings in this application called for a normal thrust load of 42 kN to be accommodated at 3000 rev/min. It was further specified that air cooling was the only form permitted and that the oil bath temperature should be no more than 70°C for a maximum ambient temperature of 25°C

The technology to accommodate this load and speed is well known using an IR ring for oil circulation. In an IR ring, oil is collected from the inside of the rim thus counteracting the centrifugal effect of oil being thrown from the outside of a conventional oil ring at high speeds. The oil circulation system is thus able to operate effectively at shaft speeds considerably in excess of those which are practical with a conventional fixed oil ring.

The difficulty facing designers in this case was to find a way of providing sufficient cooling to enable the bearing to operate at a reasonable temperature for a duty level normally appropriate to water cooled bearings. The solution was found in the development of heat pipes specifically for the extraction of heat from the bearing sump. A heat pipe is a closed device containing a working fluid under a reduced pressure. When one end is immersed in the hot environment, the working fluid vaporises and travels to the cool end where it condenses on the walls of the pipe and heat is transferred to the atmosphere as shown in Figure 7. In the case in question, air was blasted over the cool ends using a shaft mounted fan to maximise heat dissipation.

Initial analysis and design work was followed by extensive prototype testing of both heat pipes and bearing. It was found that when operating to the design duty, the cooling bath temperature settled 60°C, some 10°C less than the maximum stipulated in the bearing specification. Figure 7 shows the bearing on test in the manufacturer's works with the cowl removed showing the heat pipes and shaft mounted fan.

This bearing was developed for a special application in this country where water cooling was not a permissable design option. Subsequently to the initial contract further test work was carried out to prove the bearing at very high ambient temperatures up to 55 degrees centigrade appropriate to desert conditions where water is an expensive and not necessarily plentiful commodity.

3.2.2 **Self Pressurised Low-Loss**

The site for this application was an offshore platform near the West Coast of Africa. The problem in this case was an extremely high rotational speed for a self-contained bearing of 7400 r/min.

The heat generated in a bearing caused by collar churning in the oil increases very rapidly with speed so it becomes possible that the power loss due to this is greater than that due to oil shear on the loaded faces. Because of this, it is usual for bearings operating at high speeds to be supplied with oil under pressure from an external system directly on to the thrust collar between the pads as previously shown in Figure 3b. Conventionally, such a bearing has no sump; the oil falls immediately through a large drain in the bottom of the bearing, leaving the outside of the collar free.

The solution in this case was once again found by turning to a version of a bearing incorporating an 'IR' ring. However, hot oil was collected from the inside of the oil ring by a specially developed scoop connected to a pipe extending outside the bearing. Sufficient pressure was developed for up to 3500 l/h of the oil to be passed back into the bearing under pressure in the usual way for a low loss bearing. Figure 8 shows the circuit in schematic form and it can be seen that the oil path is divided just before entrance to the bearing with a proportion of the oil, controlled by a restriction in the line, being allowed to by-pass the thrust chamber to settling tanks mounted alongside the bearing.
These tanks permit entrained air to escapee and thus help to ensure no problems in service due to lubricant foaming.

Once again, considerable prototype test and development work was necessary to find the optimum profile for the oil scoop and to prove the bearing under full operating conditions. Figure 8 shows a photograph of the test arrangement During tests speeds up to 8500 rev/min were achieved without difficulty. Throughout it was found that plenty of oil was delivered for the purposes required and it is possible to envisage circumstances in which several bearings can be supplied with oil from this single source.

4. VERTICAL SYSTEMS

Many machines, pumps, motors and turbines have vertical shafts which require bearings capable of withstanding axial loads and of providing radial restraint. Conventionally tilting pads are used for both thrust and radial bearings which are mounted together in a single combined housing. In a typical design such as figure 9 extracted from a standard range, the natural pumping action of the collar in conjunction with radial grooves between the pads is used as the means of circulating lubricant within the housing.

In this example, oil is constrained to a single path, as shown by Figure 10, from thrust face, to radial bearing, to the reservoir and water cooler, and back to the inner diameter of the thrust face via channels beneath the thrust pads. There is thus no risk of the radial bearing becoming starved under certain operating conditions as was the case in some earlier designs in which there was a division of the oil flow between the thrust and radial bearings.

Recent development work has led to substantial increases in the efficiency of bearings such as those shown in Figures 9 and 10. As indicated by Figure 11, it has been possible to increase the maximum thrust load for normal operation by up to 70% for a given diameter of bearing housing. The result is that for a given application it is now possible to specify a standard bearing which is significantly smaller and therefore less costly than was previously the case.

The detail changes which have led to the advance in load capacity shown by Figure 11 include enlarging the thrust collar diameter within a given casing, increasing the angle subtended by each thrust pad so that the load is carried by a greater proportion of the available area, and allowing an increase in axial specific loads up to 4.1 MPa for normal duties. In previous designs, maximum thrust bearing specific loads of 3.5 MPa were usual. Extensive laboratory testing and experimental work has proved the robustness of the design changes which have been confirmed in a series of trouble free installations.

All bearings generate heat. For vertical shaft, combined thrust and guide bearings, the most usual way for this heat to be conveyed from the bearing is by means of a water cooler immersed in the oil as shown in Figure 9. The advances in load capacity described in this paper have prompted the development of more efficient water cooling systems necessary to dissipate the larger amounts of heat generated in a given size of casing.

Design improvements to the high density, externally wire wound cupro-nickel cooler tubes have led to improved rates of heat transfer. The gains in cooling effectiveness have been such that in addition to increased load capacity, it has been possible to achieve a general increase permissible speeds of operation under maximum load. It is now possible to achieve the two pole speed of 3600 rev/min, supporting a full load axial thrust of 97 kN, with a standard water cooled bearing whose outside casing is no more than 390 mm.

4.1 Specially designed vertical bearing for two pole speeds.

In particular cases it is possible to achieve two pole motor speeds with even larger self contained bearings. One such example is a bearing recently specified for the vertical main line oil pump on an offshore platform. The details of the application are shown in the following table.

Shaft Diameter	76 mm
Rotational speed	3600 rev/min
Operation load	
Normal duty	23.4 kN downwards,
	84.5 kN upwards.
Max Duty	38.4 kN downwards,
	114.7 kN upwards.
Cooling water	3046 l/hr
Casing diameter	680 mm

Figure 12 shows a view of the bearing internals with the top sealing cover removed. The wire wound cooling coils are clearly shown. The efficiency of the cooler in this type of high speed application is highly dependent upon the rate of oil circulation within the confines of the bearing.

5. CONCLUSION

Tilting pad bearings have been the subject of academic investigation and industrial development for many years. The results of this work are that standard ranges of bearings are now capable of absorbing substantially more demanding duties than previously. At the same time a range of possibilities exist, some of which have been described earlier, for meeting special customer needs which are well beyond the limits of standard bearings. Continuing research and development in the field of tilting pad bearings, coupled with modern manufacturing methods, means that these bearings will remain a cost effective design option in many applications for the foreseeable future.

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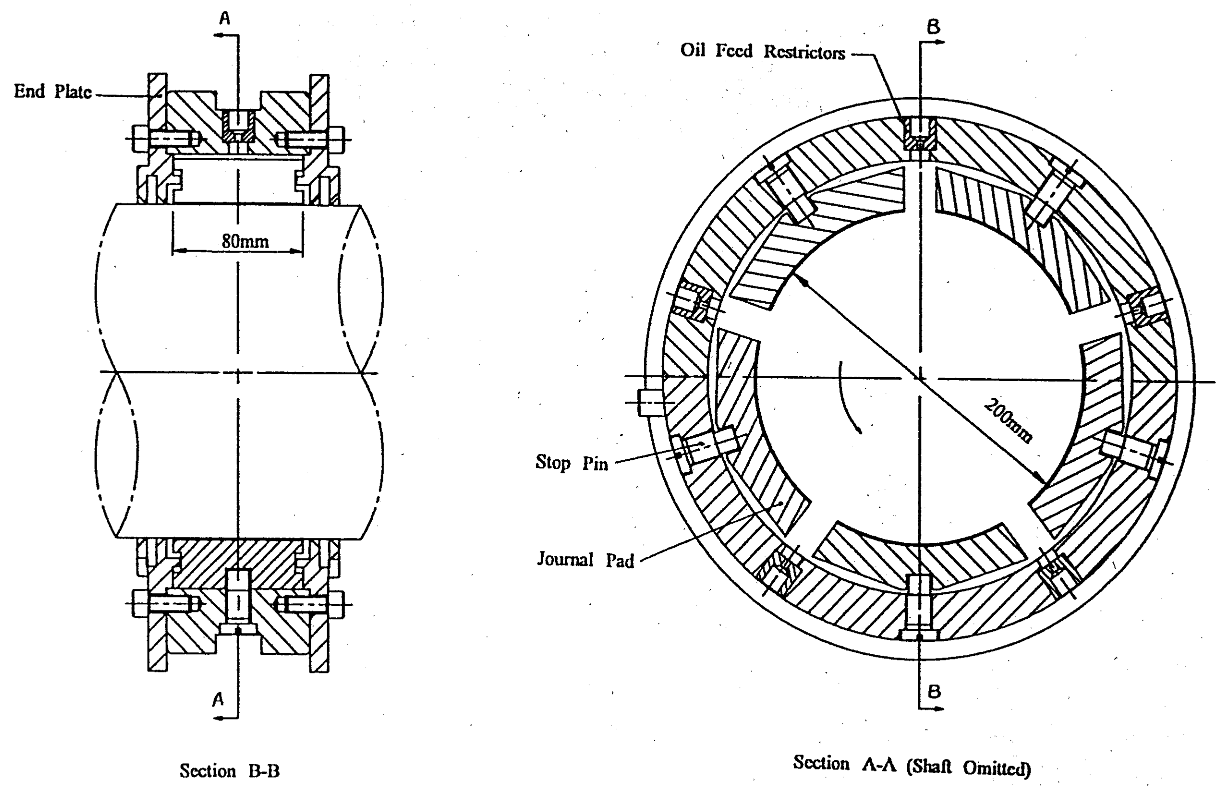


Fig 1 Tilting pad journal bearing used in experimental work

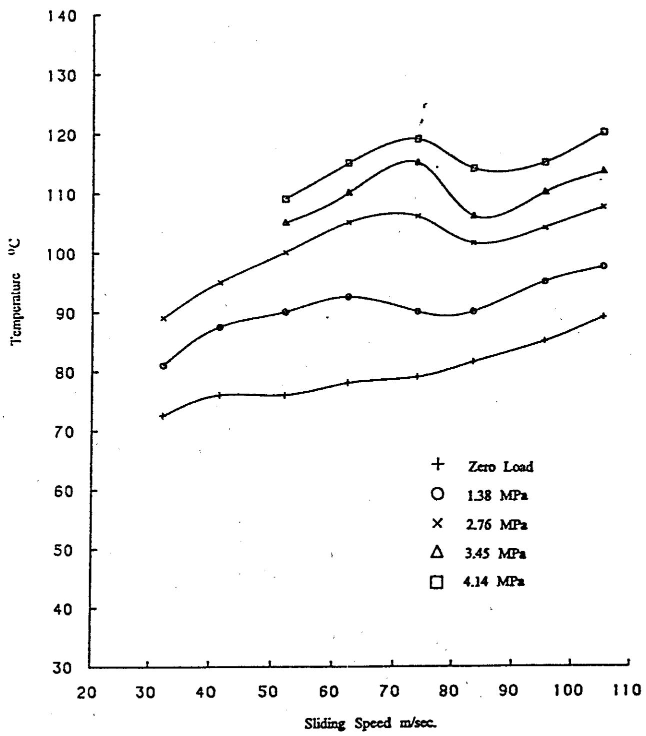


Fig 2 Maximum pad temperature for 200mm journal pad bearing — load between pads

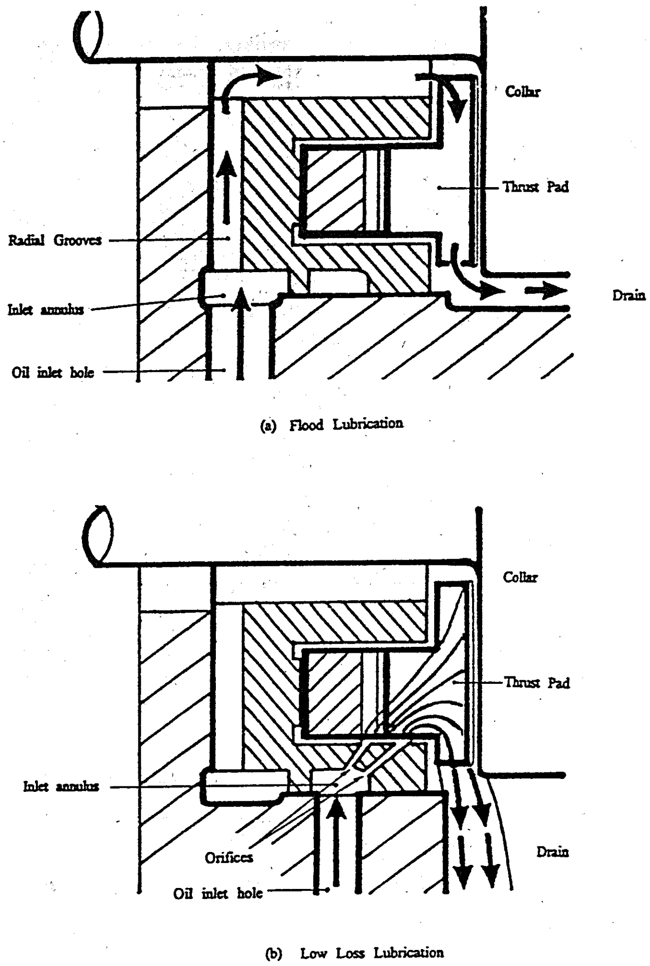


Fig 3 Lubrication methods for thrust assemblies

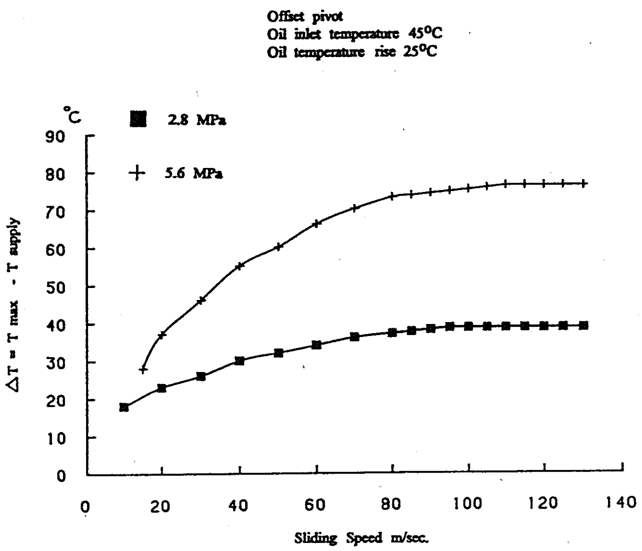


Fig 4 Temperature rise versus sliding speed for tilting pad thrust bearing

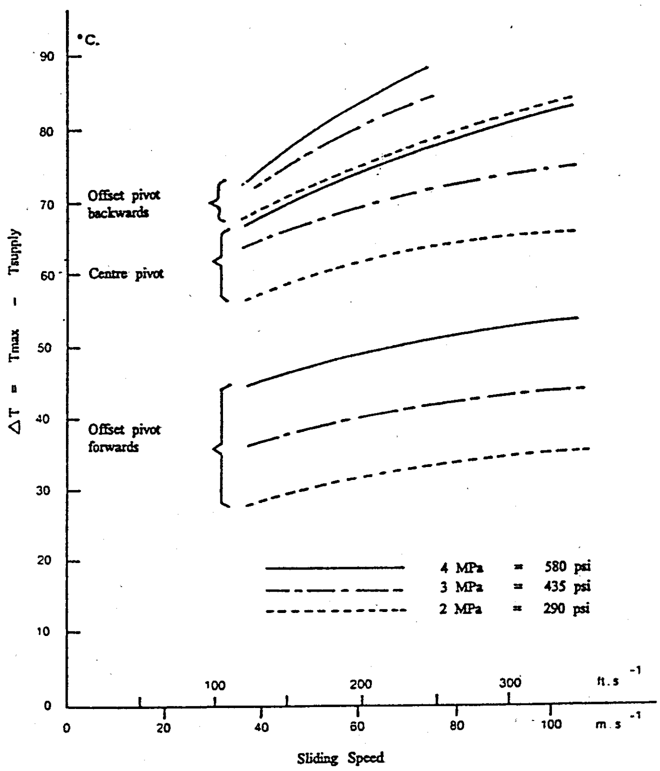


Fig 5 Comparison of offset-pivoted pad reverse rotation with centre and offset-pivoted pads foward rotation, low cost lubrication

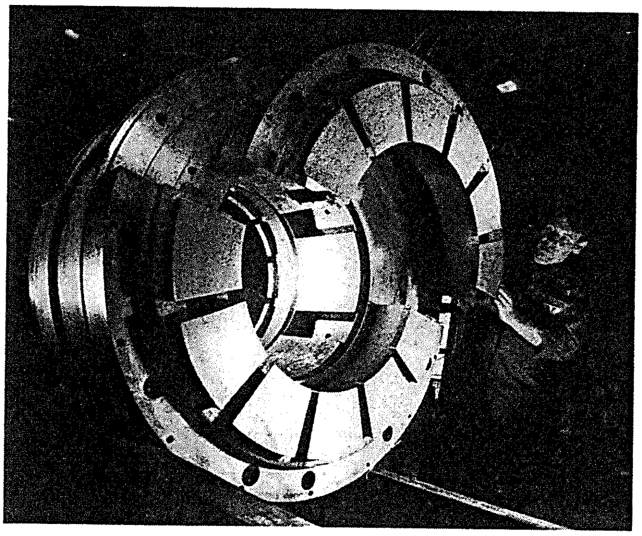
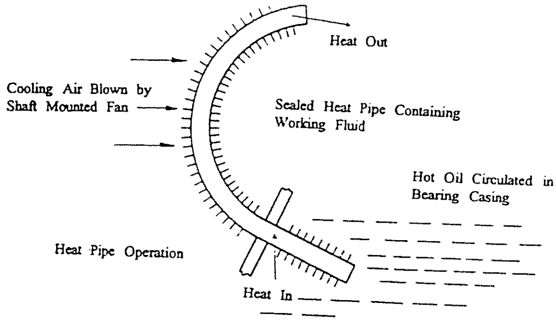
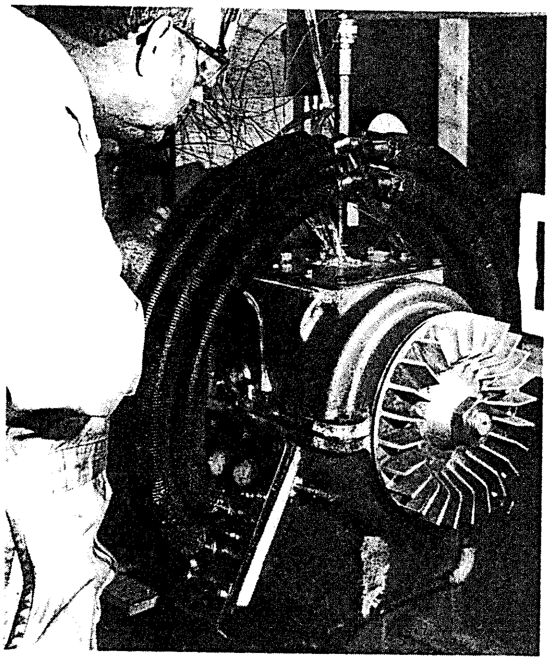


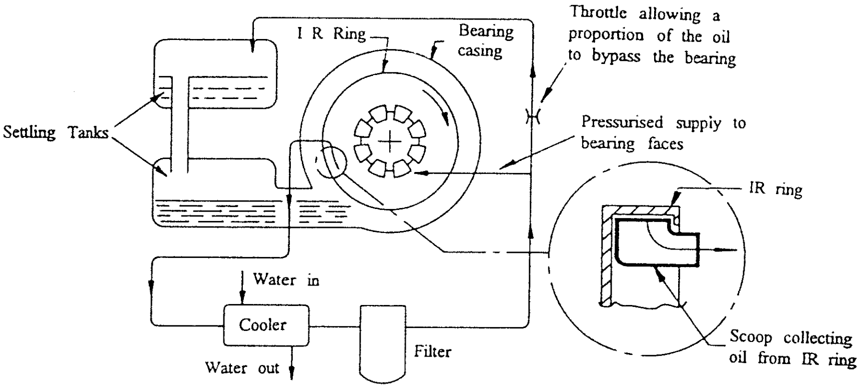
Fig 6 Double thrust and journal bearing for the compressor of the European transonic windtunnel



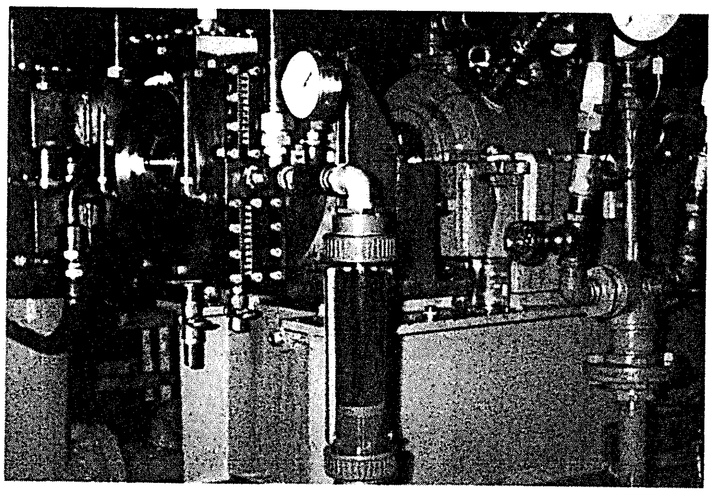


Test Set-up for Heat Pipe Bearing

Fig 7 Cooling with heat pipes



Self pressurised bearing circuit diagram



Test set-up for self pressurised bearing

Fig 8 Self pressurised low loss

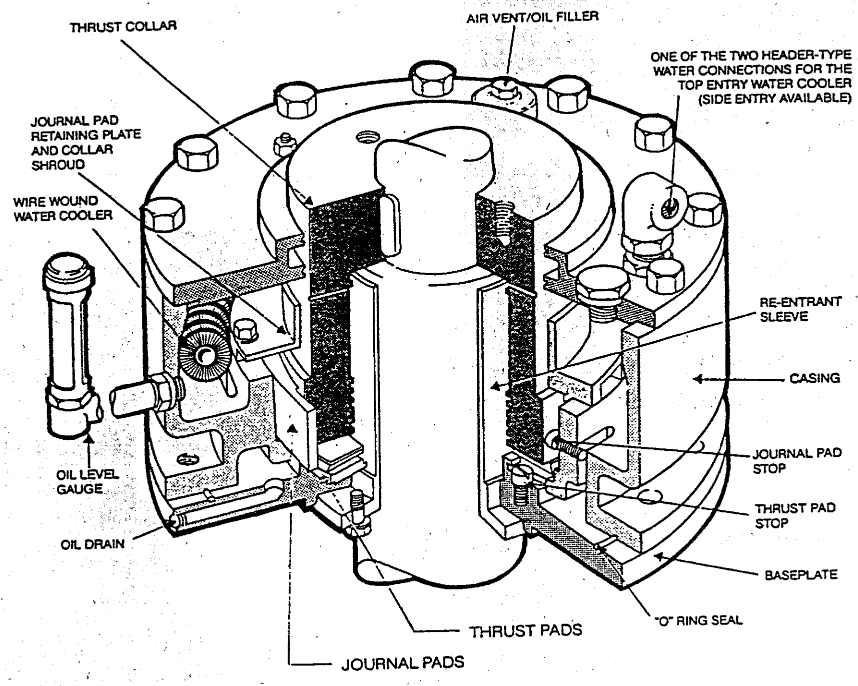


Fig 9 Cut away view of a standard thrust and guide bearing

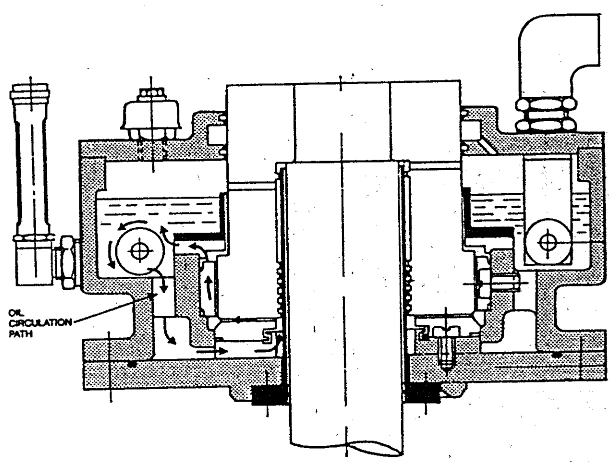


Fig 10 Sectional elevation of a standard thrust and guide bearing

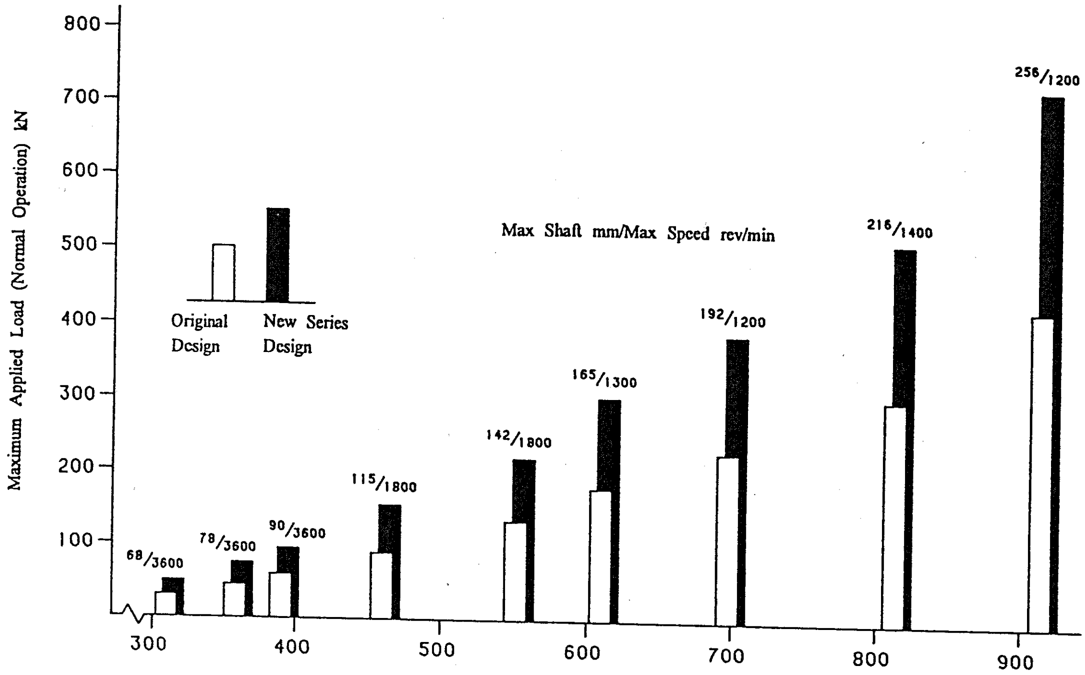


Fig 11

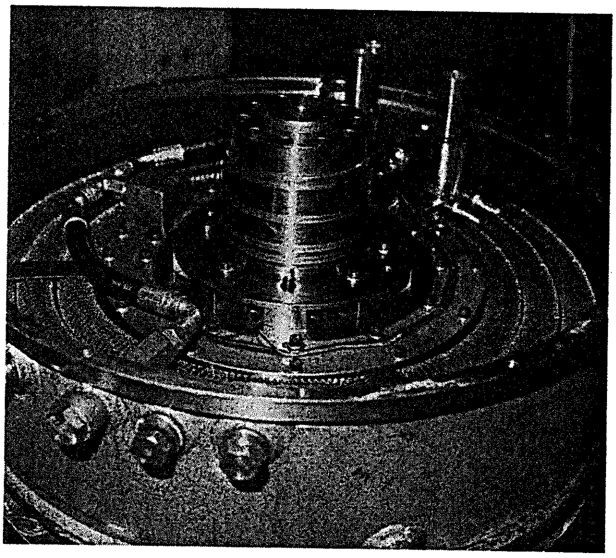


Fig 12 Vertical bearing for main oil line pump: speed — 3600 rev/min