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BEARINGS FOR PUMPS

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BEARINGS FOR PUMPS

Introduction

For many years most pumps have used either rolling element bearings, or whitemetal faced plain or tilting pad thrust and journal bearings operating on the hydrodynamic principle (Fig 1); for a limited number of special applications product lubrication has been used with special bearing materials. In this paper we intend to look at some developments both in conventional whitemetal bearings and in bearings using other materials and concepts. The emphasis will be on the application of these new ideas to pumps though in many cases they are also applicable to other types of rotating plant machinery.

The specific subjects which we intend to cover are:-

1 Horizontal self-contained assemblies for high thrust loads using conventional whitemetal lined bearings.

2 High temperature polymer linings for tilting pad bearings.

3 Ceramic bearings for process fluid lubrication.

4 Magnetic bearings.

Horizontal Self-contained Assemblies for High Thrust Loads

From the point of view of the user it is always preferable to have a self-contained bearing assembly, i e a bearing which does not use any external lubrication or cooling system. In practice there are limitations to the use of such assemblies due to the necessity of dissipating heat generated at higher speeds and also the desirability in all except the most conservative designs, of providing a pressure lubricant supply to the bearing. In horizontal shaft assemblies there is an additional problem of ensuring that the bearings are sufficiently lubricated at start up as at standstill the static oil level is below most of the bearing surfaces.

In order to extend the limits of the horizontal self-contained type of assembly Glacier has developed the HSV range of units based on the use of a viscosity pump to ensure a positive flow of oil to the bearings. The general arrangement of an assembly of this type is shown in Fig 2. It will be seen that a double thrust bearing of the tilting pad type is incorporated, together with a journal bearing of the sleeve type: all of these bearings would normally be of the conventional whitemetal faced design.

Oil circulation in the HSV unit is achieved by means of a viscosity pump (Fig 3) situated around the thrust collar periphery which circulates oil direct through the bearings and then through the cooler and filter; the oil us returned to the inner reservoir through an injector which also draws oil from the outer reservoir as required. The pumping characteristic of the viscosity pump is such that for most applications it is possible to have a supply pressure in the region of 1 to 2 bar with ample flow to ensure correct cooling of the bearings. The pump also has sufficient capacity to supply at least one other journal bearing at a remote location with the oil from this being returned to the main reservoir as shown in Fig 2.

The inner reservoir ensures that the viscosity pump can prime immediately on starting of the unit as the viscosity pump suction is at all times fully flooded, even with a low outer reservoir levels.(Fig 4) Thus if in the course of prolonged running there is some minor leakage from the system it will not affect the priming ability of the unit until the outer reservoir capacity is almost exhausted.

The availability of ample oil at a positive pressure enables a large thrust bearing to be correctly lubricated and the heat generated in it to be removed efficiently by using a forced flow cooler. Thus a compact self-contained unit is achieved capable of taking high thrust loads at relatively high shaft speeds in a reliable way. The only external supply required for the unit is water for those applications which have a water cooler; air coolers can also be used either with a shaft driven or electrically driven fan. This in itself enables a rather more compact pump assembly to be installed than with conventional externally cooled and lubricated bearings but in addition there is a big saving in that the external lubrication package with its associated pipework and controls is completely eliminated.

Particularly in oil rig applications the saving in space requirements and weight results in a total pump unit which has significant advantages over the conventional pump. (Fig 5)

HSV type units have also been used for emergency standby duty pumps where they normally remain unused for prolonged periods. In tests it has been shown that even if all the parts in the HSV units are completely dry above the static oil level there is no problem in starting or subsequent damage to the bearings. (Fig 6)

High Temperature Polymer Materials for Tilting Pad Bearings

While conventional whitemetals are very good materials for the majority of applications, there are an increasing number of cases where they cannot be used satisfactorily. This can either be because of high temperatures, low viscosity lubricants or corrosive elements in the lubricant. Various alternatives to whitemetal have been used in the past but all of these have some disadvantages as well as advantages. Recently polymer materials have been used as bearing surface materials and one of these is Glacier Hi-eX. This is a PEEK based polymer which is bonded to steel through a sintered interlayer (see Fig 7).

The advantages of Hi-eX are as follows:-

A High temperature ability - it can run at temperatures in excess of 200 deg C without deterioration or weakening.

B Good boundary properties - Hi-eX has good boundary properties when running against steel which enables it to operate satisfactorily with the small film thicknesses often associated with product lubricants. It has also been successfully used with oil in applications where full lubrication may be temporarily absent such as at start-up of a machine.

C Dimensional and chemical stability - Hi-eX is highly resistant to many chemicals though in some cases care has to be taken when the environment is other than neutral. When operating in water there is no problem of dimensional stability.

D Good insulator - Hi-eX is a good electrical insulator with a resistance in excess of 1×10^9 ohm - cm.

E Tolerance of dirt - Unlike many other high temperature bearing materials, Hi-eX has similar hardness and hence embeddability properties to whitemetal; similarly it does not require a hardened bearing surface to run on.

In principle Hi-eX can be applied to a steel backing to form either plain thrust or journal bearings or tilting pads to be used either in thrust bearings or journal bearings. In case of the latter the pads are dimensionally identical to the equivalent whitemetal faced pads and therefore can easily be retrofitted in an existing standard bearing.

The main areas of application of Hi-eX to rotating plant machinery are seen as follows:-

1 Bearings for operation at high ambient temperatures - there are an increasing number of applications where the bearing for various reasons has to operate in high temperature surroundings. A typical example of this is a geothermal pump which might have to operate in a well at temperatures in the region of 200 deg C. Hi-eX tilting pad thrust bearings have been successfully used in a number of applications of this type.

2 Heat soak applications - for this type of application the normal operating environment is kept cool by a supply of lubricating oil. At shutdown conventional practice would ensure that a supply of this oil is kept in operation until the machine has cooled to an acceptable level. However savings can be made if the bearing material is designed to withstand heat soak through the shaft etc at shut down without the need for a cooling oil flow. A number of applications of this type are now in successful operation.

3 Product lubrication - Hi-eX can be successfully used using low viscosity fluids, such as water, for lubrication due to its good boundary properties under marginal film conditions. Using lubricants of this type it is essential that some form of alignment device is provided and a typical system is shown in Fig 9. This shows a tilting pad thrust bearing with Hi-eX lined pads where the carrier ring is mounted on a nitrile cushion ring for alignment purposes. Bearings of this type have seen successful service in various types of pump.

4 Extension of operating range for self-contained units - as mentioned in the section above on self-contained horizontal units, the limitation in many cases is the ability to dissipate heat. By fitting a self-contained assembly with Hi-eX lined pads and using a synthetic lubricant the maximum allowable temperature of the oil system can be raised which in turn allows the maximum load and/or speed of the unit to be raised. While this has been proven in tests to be a feasible option it has yet to be adopted in a production machine.

Ceramic Bearings for Process Fluid Lubrication

Ceramic bearings are seen as an extension of existing bearing materials, enabling the designer to achieve the full benefits of product lubrication. The concept of product lubrication has by far the greatest implications on the pump industry where the pumped medium can be used to lubricate the bearings.

Of the wide range of ceramic materials available, the most widely used for bearings of the type being considered are silicon nitride and silicon carbide. The former has the grater strength and the latter the better heat conductivity as well as greater hardness. Typical values are shown in Fig 10.

For most applications in rotating plant the mechanical strength of silicon carbide, with careful design, is adequate and the good heat dissipation is of great benefit; therefore silicon carbide is the preferred material for the applications described in this paper.

The main advantages of silicon carbide bearings can be summarised as follows:-

1 Chemically inert.

2 High hardness.

3 High load capacity.

4 Wide temperature range capability.

Each of these advantages could have implications on pump designs, for instance:-

1 Chemical inertness enables the bearings to be lubricated by most fluids, which in turn enables lubrication systems and sealing systems to be simplified. In the case of canned motor pumps sealing systems become completely unnecessary. Cost savings can be made from the removal of oil systems, as well as savings in overall pump size (shaft diameter and length). See Fig 11.

2 High hardness enables dirt and contamination in the lubricant to be tolerated, it also means that a filter system is not normally required. For example Fig 12 shows sand in suspension in a product lubricant before and after it has passed through the bearing. It will be noted that the sand particles have been crushed - no change occurred to the bearing surfaces.

3 High load capacity, particularly of tilting pad thrust bearings, leads to compact bearing designs. These bearings have been tested up to 30 MPa specific load although for design purposes the maximum specific load is usually restricted to between one third and one quarter of this value. The high load capacity can mean reduction or removal of balance pistons thus improving power loss and noise. Fig 13 is an example of test results from a thrust bearing showing temperature rise and torque losses for loads up to 30 MPa.

4 The wide temperature range capability enables the designer to use a standard bearing over a wider range of applications.

Magnetic Bearings

Forces of attraction between magnetised bodies can be very large but cannot be used by themselves except in simple applications such as load hoists. This is because when there is a gap between magnetised bodies such configurations are not stable. The force-distance characteristics of such configurations are shown in Fig 14. To convert these characteristics and to produce stable configurations it is necessary to use electromagnets whose strength can be altered by varying the current in them. This is also shown in Fig 14 where the object to be held is a steel ball against the force of gravity. The position of the steel ball relative to the electromagnet is sensed by an optical transducer and the current in the electromagnet varied by the electronic power amplifier in proportion to the strength of the position signal received, with appropriate compensation, from the optical transducer. This in essence is the principle of the technology of magnetic suspension systems using control dc electromagnets.

Magnetic bearings: when this technology is to be applied to bearings the configurations naturally become more involved than in the simple example given above. The single electromagnet acting against the force of gravity is now replaced by a pair of electromagnets acting on a shaft on two axes at right angles to each other and against each other, giving the configuration the capability to withstand much larger disturbance forces. This configuration of four electromagnets constitutes one radial bearing. A single radial bearing on its own cannot counteract tilting of the shaft. It is, therefore, necessary to provide at least two radial bearings. Further, in most cases displacement of the shaft in axial directions has to be countered by a thrust bearing. This is achieved by providing circular electromagnets acting on a steel disc attached to the shaft. A complete magnetic bearing system, therefore, has two radial and one thrust bearing.

By and large optical transducers of the kind shown in Fig 14 are not likely to be suitable for application to magnetic bearings and special purpose inductive, magnetic and capacitance sensors have been developed to meet specific requirements. The technology of magnetic suspension is now fully proven in the context of bearing applications.

Application of magnetic bearings to canned pumps: the exploded diagram (Fig 15) shows the construction of a pump with magnetic bearings in which the rotor and the stationary members are protected from, say, corrosive liquids by a pair of stainless steel screens. The impeller is also made of stainless steel. The insertion of these screens makes a great deal of difference to the design of the bearings themselves, the transducers and the motor which in this case is located between the two radial bearings. However, it can be seen immediately that the need for any seals is completely obviated. Ideally the magnetic bearings should have as small an air gap as possible between the shaft and the face of the magnets, of the order of 0.5mm. The stainless steel covering on the rotor and the screen adjacent to the stator are 1mm thick. The effective air gap in the present case is 3mm for the bearings, the motor and the transducers. The least affected by this increased air gap is the design of the electromagnets. Care is needed to ensure that as much of the magnetic flux generated by currents flowing in the stator windings as possible is contributing to the force of attraction. In spite of careful design an increase in the mmf will be necessary and techniques have been evolved which do not adversely affect the heat dissipation and/or efficiency. The motor in the pump illustrated in Fig 15 is an induction motor operating normally at 0.3mm air gap. Increased air gap leading to increased leakage flux and leakage reactance can have deleterious effect on the performance of an induction motor. Steps, therefore, had to be taken to reduce this and the derating of the motor has been from 4 kW to 2 kW. With a new design it is expected that this derating could be halved. The most serious challenge, however, is presented by the position transducers. These have to be capable of looking at the rotor through the stainless steel barriers which are conducting. Special transducers had to be developed for this purpose. The increased air gap presents further problems of linearity of operation, resolution and effects of temperature drift. All these have been the subject of intensive study and investigation and have been satisfactorily resolved.

Design specifications and operating experience: the pump with magnetic bearings has been developed as a prototype and has been in operation on the customer's premises since February 1987. apart from an initial one or two teething problems, there have been no problems in satisfying the original specifications.

The specifications of the bearings and pump are as follows:-

Front Bearing

Diameter	90 mm
Length	100 mm
Max static load (integrator control)	1000N (flux intensity 1T)'
Stiffness	500 N/mm up to 50 Hz
Power dissipation	90 W for max load

Rear Bearing

Diameter	90 mm
Length	50 mm
Max static load (integrator control)	500 N
Stiffness	500 N/mm up to 50 Hz
Power dissipation	60 W for max load

Thrust Bearing

Disc diameter	210 mm
Max static load	3000 N
Stiffness	500 N/mm up to 50 Hz
Power dissipation	130 W
Pump pressure	varying from 2.2 bar at 0 litres/min to 0.2 bar at 100 litres/min

Motor

Induction

3000 rpm

2 kW output (4 kW at original gap)

Conclusions: magnetic bearings offer unique opportunity for eliminating problems of seals and protection of external members from corrosive fluids. Many of the problems perceived in the early days of development of the pump described have been successfully overcome, proving that pumps incorporating magnetic bearings are an alternative way of designing such pieces of equipment.

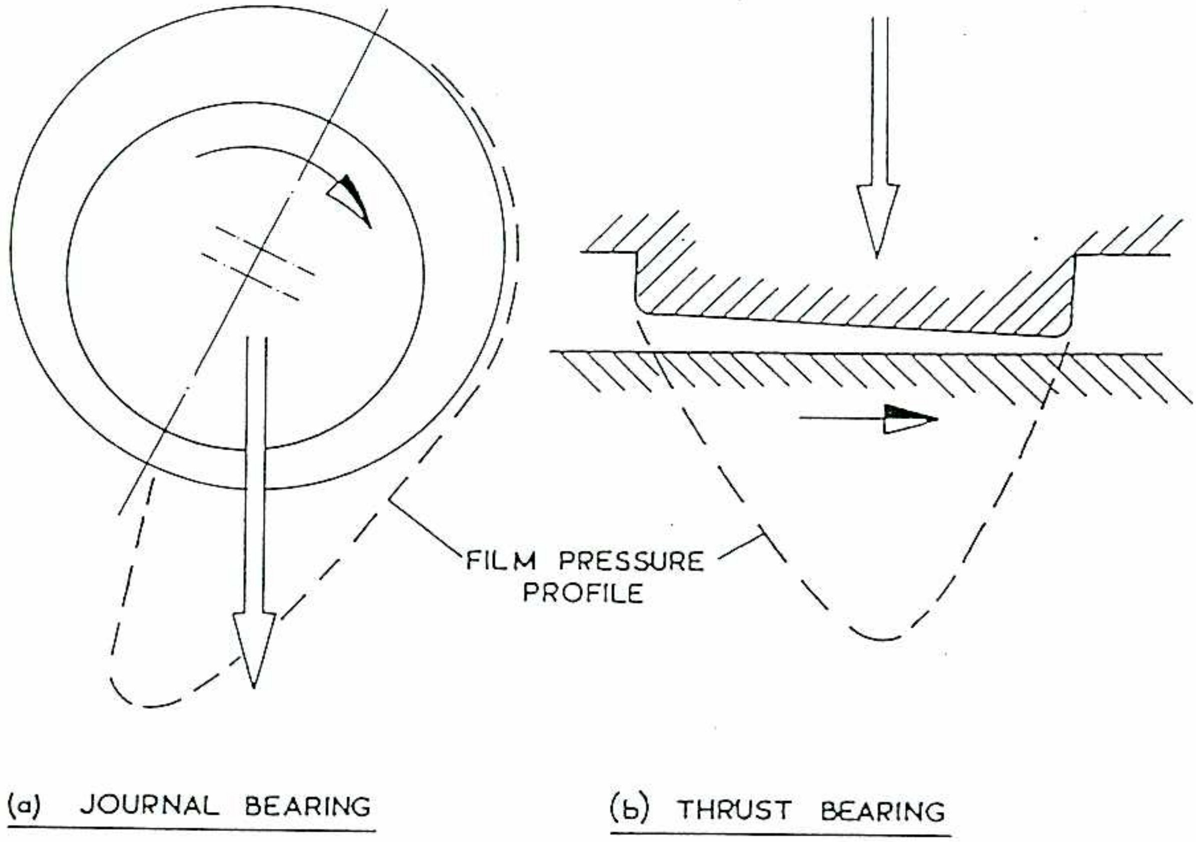


Fig 1 Hydrodynamic converging wedge principle on journal and thrust bearing

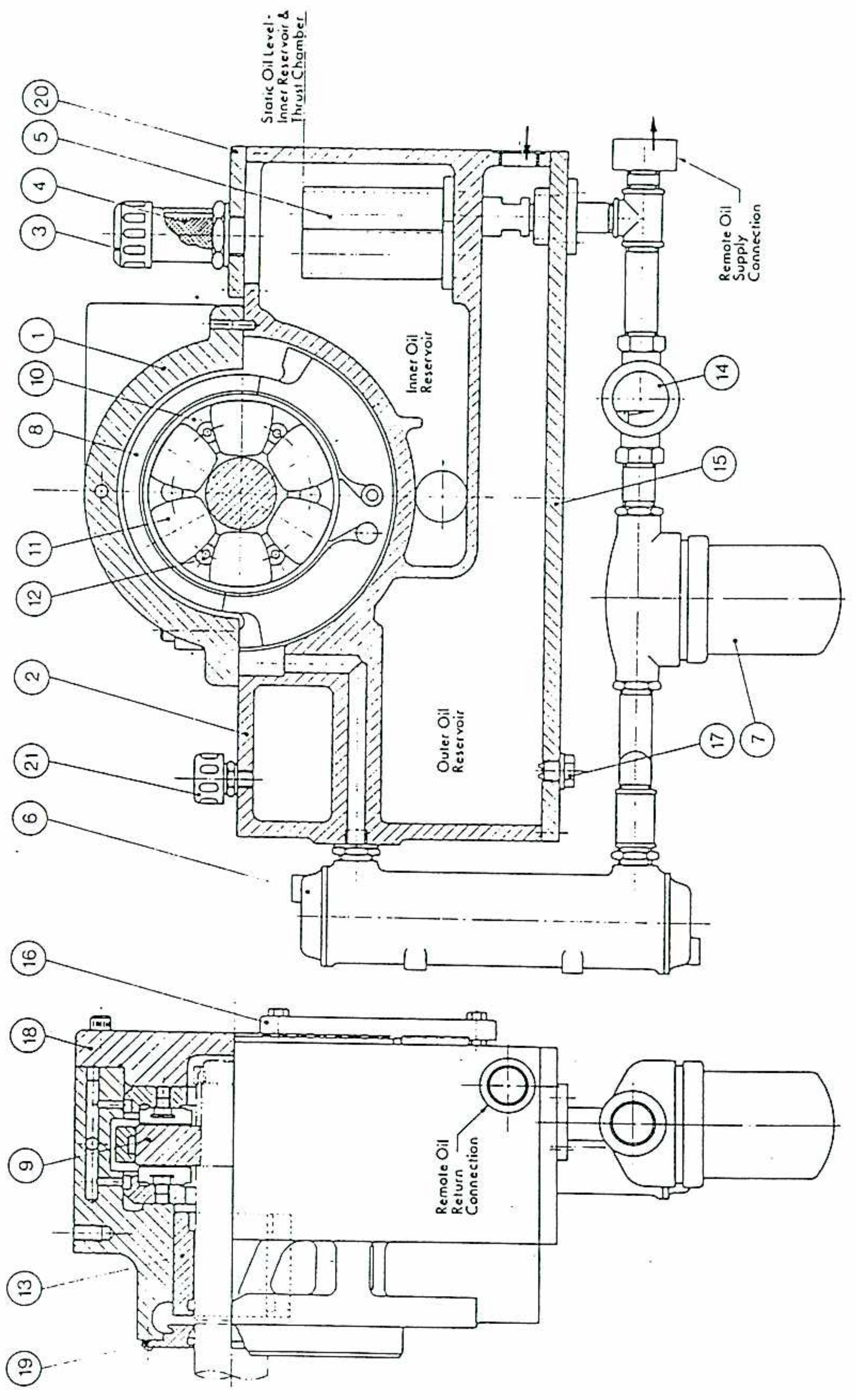


Fig 2 General arrangement of HSV bearing assembly

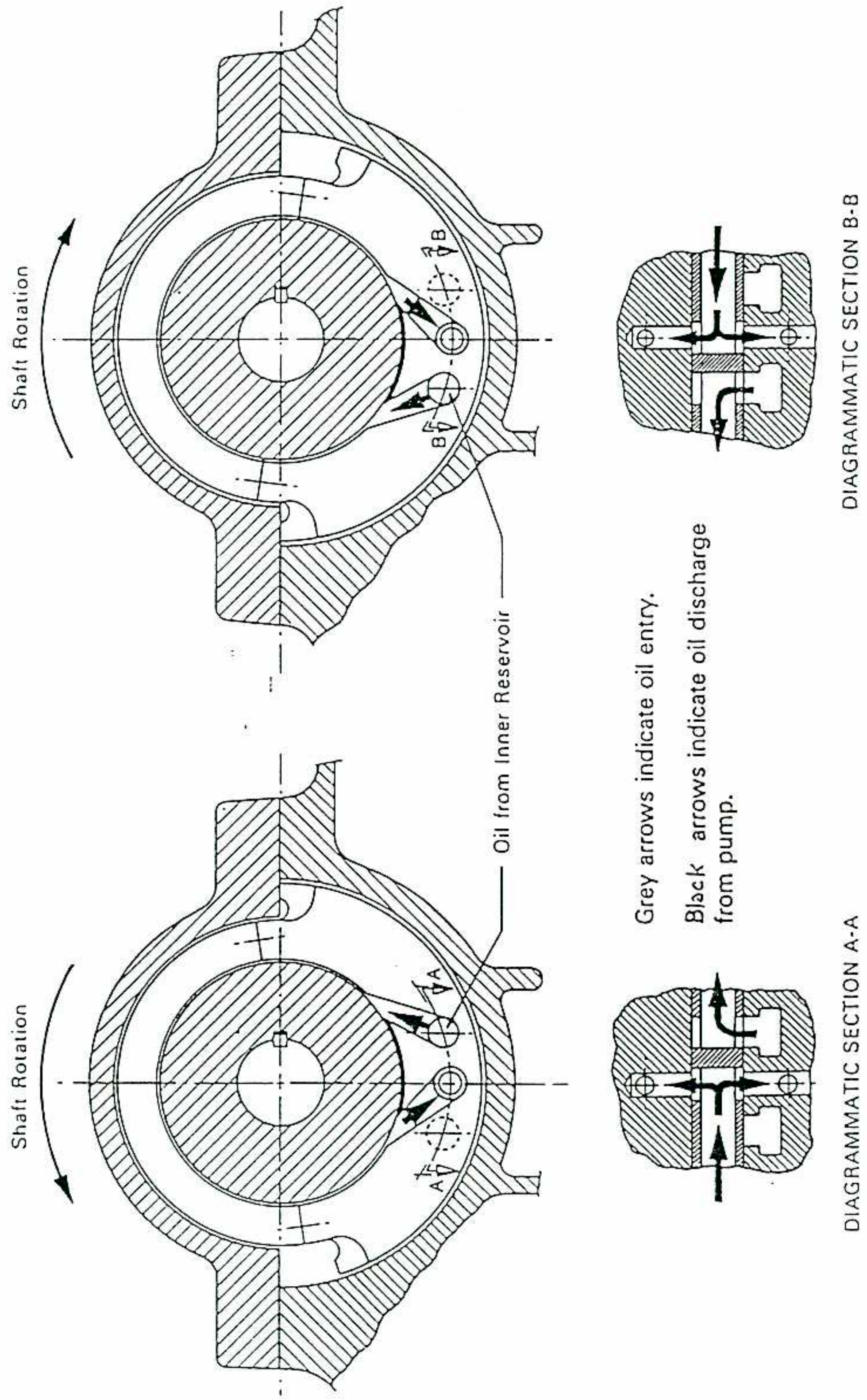


Fig 3 Operating priciple of reversible viscosity pump

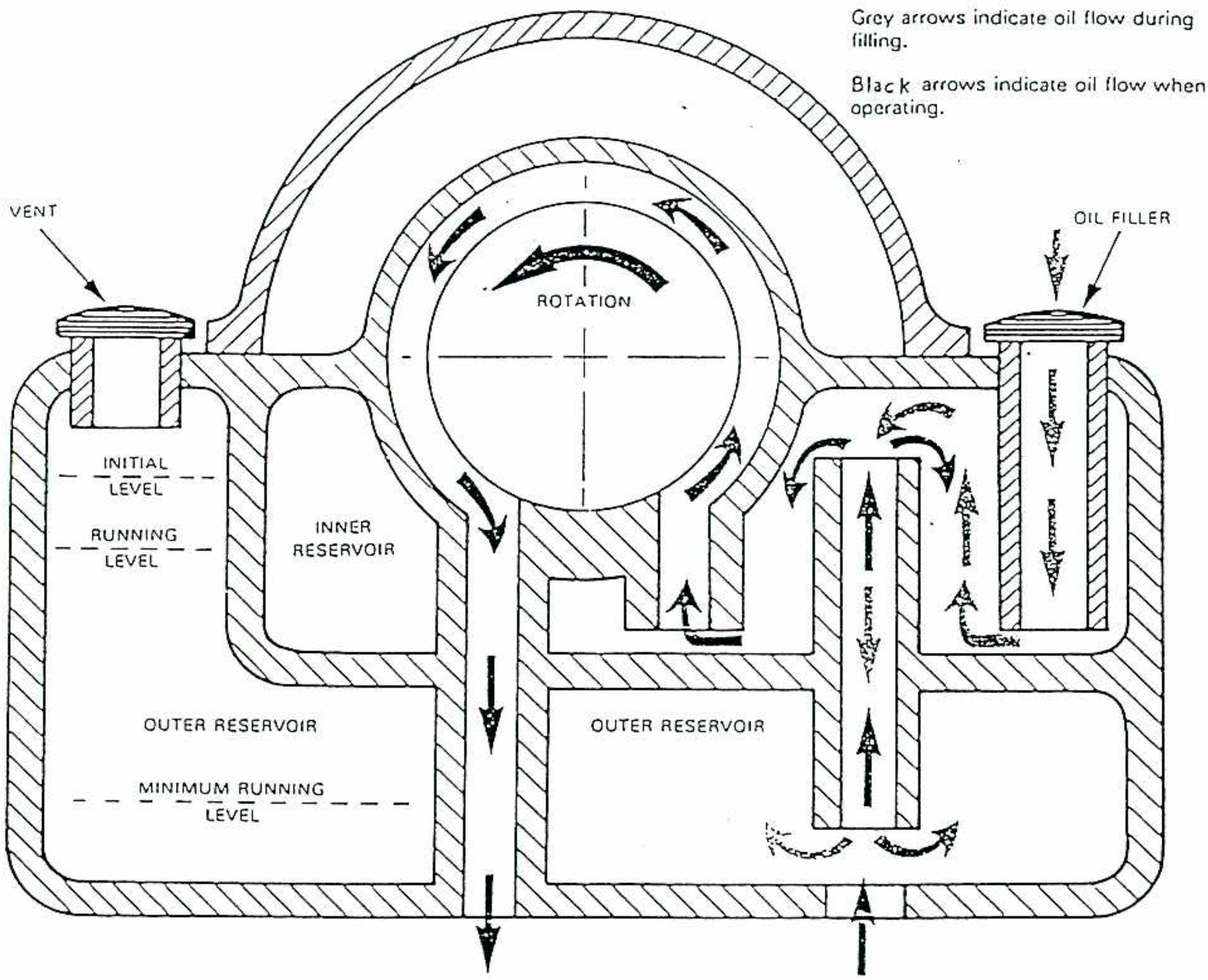
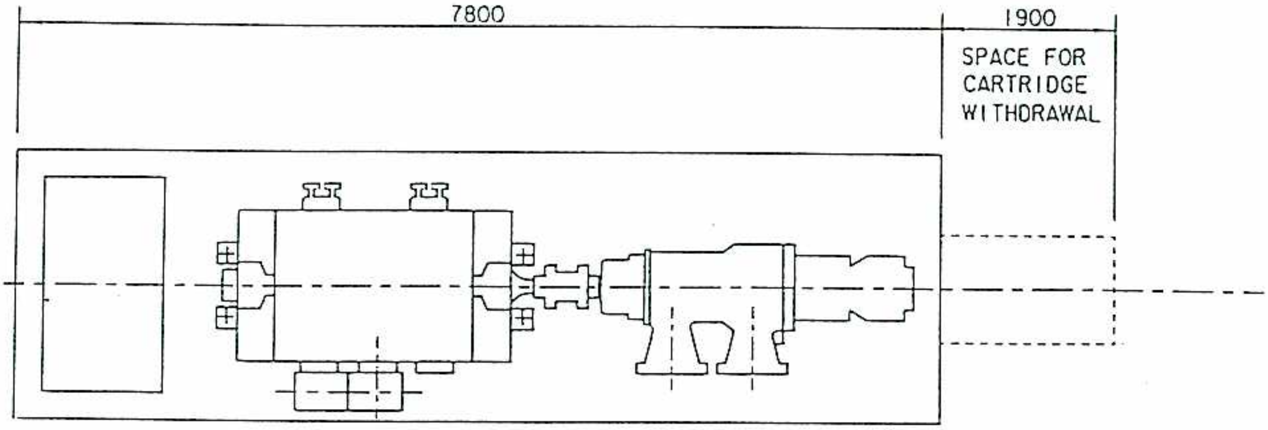
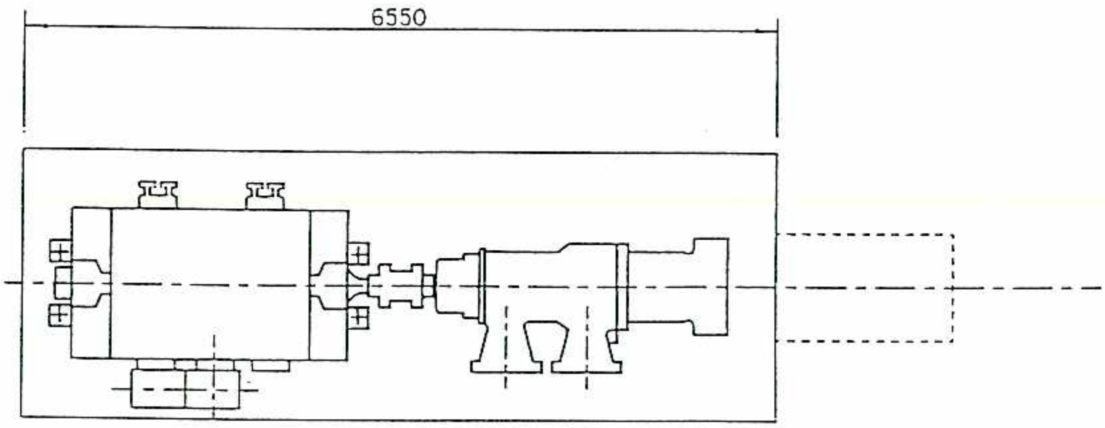


Fig 4 HSV bearing assembly flow system



STANDARD OIL SYSTEM

WEIGHT = 21500 kg



GLACIER HSV OIL SYSTEM

WEIGHT = 18000 kg

Fig 5 Comparison of Standard & HSV Lubrication Space Requirements
(Courtesy of Sulzer Pumps Ltd)

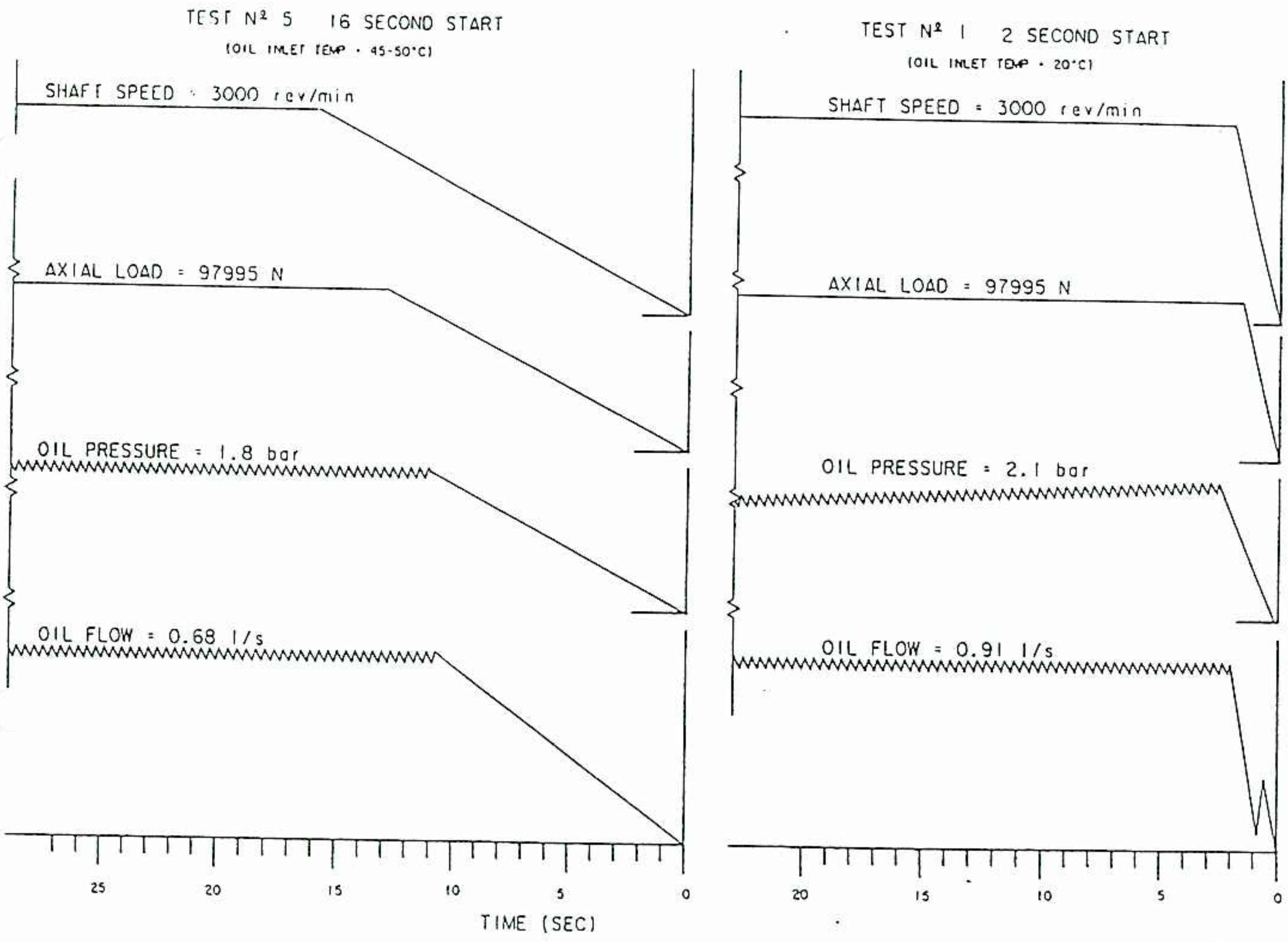


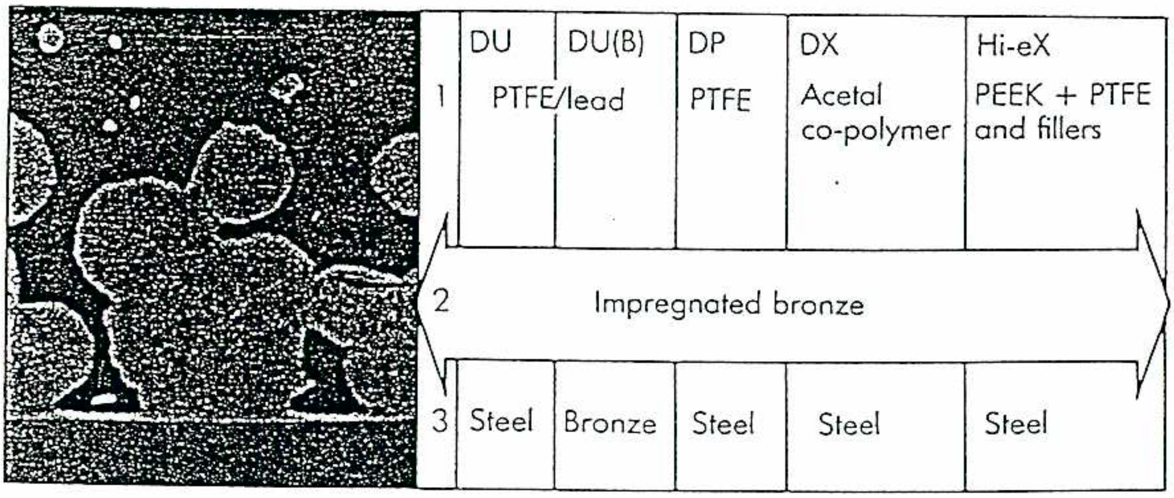
Fig 6 Viscosity Pump Response time at Start-up

MATERIAL TYPE	SAE EQUIV.	NOMINAL COMPOSITION						NOMINAL HARDNESS (HV)	MAX. DESIGN SURFACE TEMP °C
		Al	Cu	Pb	Sb	Sn	Ni		
TIN BASE WHITEMETAL GM130	SAE12	---	3.5	---	7.5	89	---	27	130
LEAD BASE WHITEMETAL GM155	SAE13	---	0.5	83.5	10	6	---	16	130
ALUMINIUM BASE ALLOY AS45	---	60	---	---	---	40	---	27	155
ALUMINIUM BASE ALLOY A11	SAE770	92	1	---	---	6	1	45	155
COPPER BASE ALLOY SL	SAE48	---	70	30	---	---	---	30-45	170
COPPER BASE ALLOY SP	---	---	72	26	---	2	---	44-50	170
HIGH TEMP. POLYMER Hi-eX	---	---	---	---	---	---	---	20	250
CERAMIC SILICON CARBIDE	---	---	---	---	---	---	---	3000	380

Glacier ROTATING PLANT BEARINGS

T&N

Fig 7 Bearing materials for rotating plant applications



1 Overlay

2 Interlayer

3 Metal backing
(full depth not shown)

Fig 8 Section through polymer bearing material

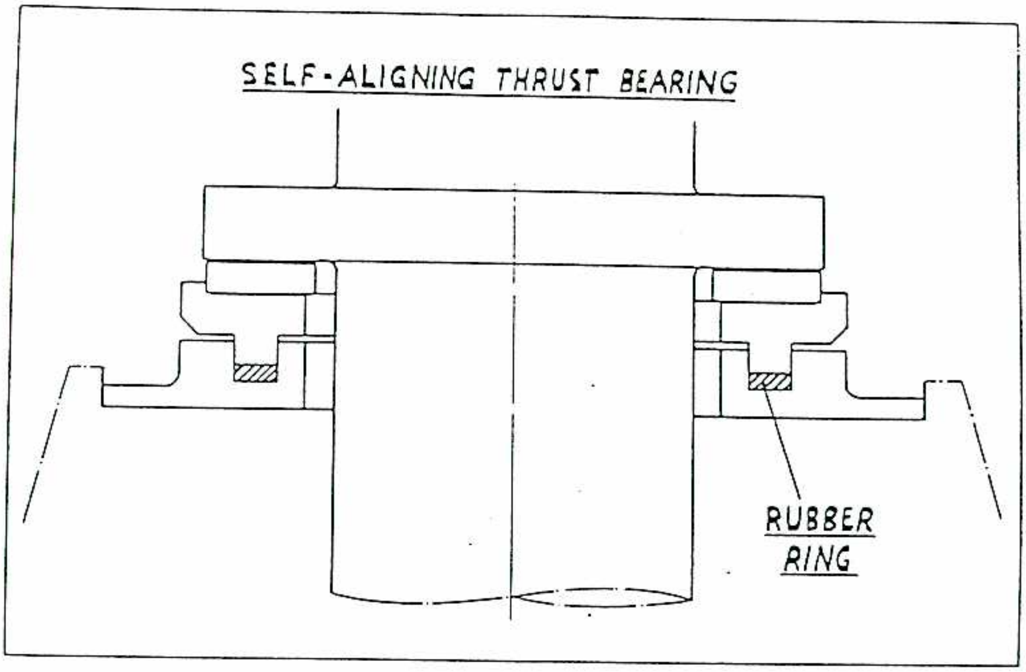


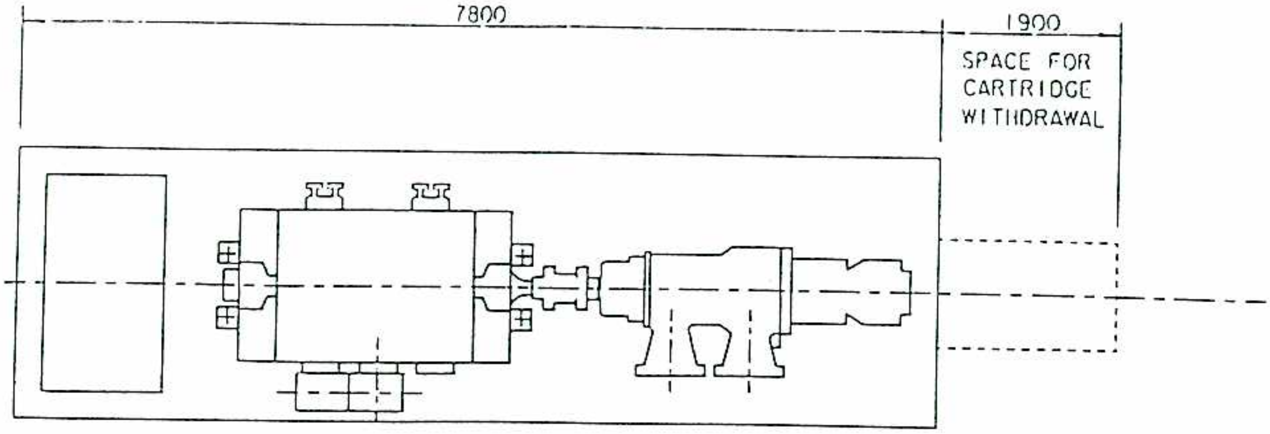
Fig 9 Nitrile Cushion Ring Alignment Device

PROPERTIES OF ENGINEERING CERAMICS COMPARED WITH OTHER MATERIALS

UNITS	MATERIAL	Silicon carbide		Silicon nitride		Zirconia			Alumina	Alumina	Cemented tungsten carbide	Cast iron	Steel
		RBSC	SSC	RBSN	SSN	PSZ	CSZ	HTZ					EN32
	COMPOSITION	SiSiC	SiC	Si_3N_4	Si_3N_4 + additives	ZrO_2 5% Y_2O_3	ZrO_2 10% Y_2O_3	ZrO_2 CeO	99% Al_2O_3	95% Al_2O_3	6% Co		
	TENMAT TRADE NAME	REFEL	REFEL	NITRASIL	NITRASIL	CERAFINE	CERAFINE	CERAFINE					
	GRADE	F	B	R	S	PSZ	CSZ	HTZ					
UNITS	PROPERTY												
	Physical Properties												
$g\,cm^{-3}$	Density	3 10	3 16	2 4	3 25	6 05	6 0	6 [illegible]	3 [illegible]	3 8	[illegible]5	7 2	7 37
%	Open porosity	0	0	25	0	0	0	0	[illegible]	0	0	0	0
μm	Grain size (average)	10	1-2		1-2	0 5	0 5	1-2	[illegible]	10-40	10-40		
	Poissons ratio	0 24	0 17	0 27	0 24	0 3	0 25	0 3	0 27	0 27	0 26		0 27
	Mechanical Properties												
	Flexural strength (3 point bend)												
MPa (psi × 10^3)	at 20°C	400 (58)	410 (59)	190 (30)	650 (94)	1000 (145)	350 (51)	325-700 (47-102)	400 (5[illegible])	300 (43 5)	1400 (203)	500 (72)	
MPa (psi × 10^3)	at 1000°C	400 (58)	410 (59)	190 (30)	450 (65)						Degrades	Degrades	Degrades
GPa (psi × 10^6)	Tensile modulus 20°C	390 (56)	450 (65)	170 (25)	290 (42)	200 (29)	160 (23)	215 (31)	360 (52)	300 (44)	600 (88)	117 (17)	206 (30)
MPa (psi × 10^6)	Compressive strength 20°C	2000 (290)	2000 (290)	550 (80)	2000 (290)	2000 (290)	1500 (217)	1500 (217)	2700 (391)	2100 (305)	7000 (1015)		
$MPa\,m^{1/2}$	Fracture toughness K_{IC} 20°C	4	4	3	8	9	1	20-15	4	4 5	6		
Kg/mm^2	Hardness Hv (50g load)	3000	2800	1100	1500	1400	1500	900	1600	1500	1500	250HB-600HB	450HB-650HB
	Thermal Properties												
$Wm^{-1}K^{-1}$	Thermal conductivity at 20°C	150	100	16	25	1.9	2.0	1.9	26	21	60	54	50
J/kg°C	Specific heat	1100	1000	800	800	540	500		390	200	480	485	
10^{-6}/°C	Thermal expansion coeff 20°–800°C	4.3	3.8	3.0	3.0	8	12	8	8.3	8.0	4.8	12	13
°C	Maximum temperature of use (continuous)	1350	1400	1150	1150	1000	1800	1000					
	(short term, no load)	1600	1600	1400	1400		2000						
°C	Thermal shock resistance (ΔT)	400	380	600	650	250	110			350		300	
	Electrical Properties												
ohm m	Electrical resistivity at 20°C	10		10^{10}	10^{11}	10^{10}			10^{13}	10^{12}			

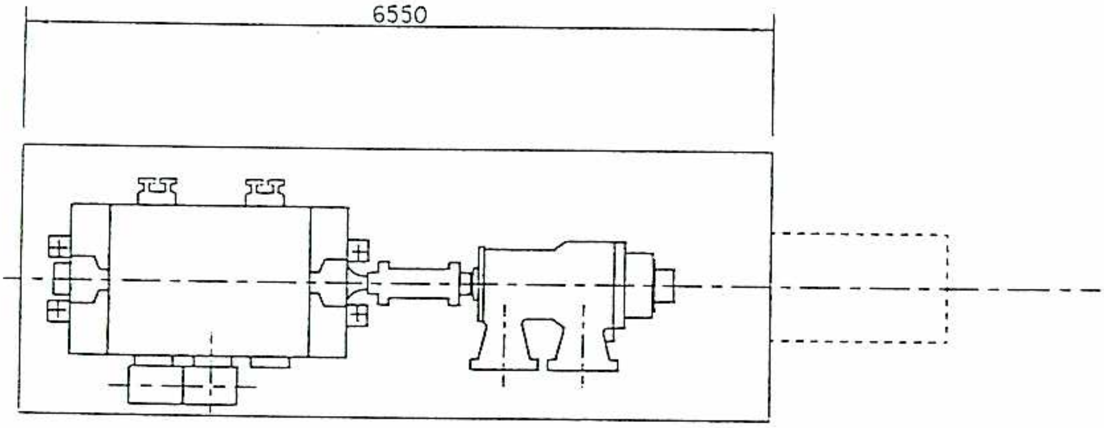
[illegible] 1000°C Zirconia ceramics become electrically conductive
[illegible] values are typical of those obtained from test pieces and are offered for guidance in the design of engineering components

Fig 10 Typical Ceramic Properties (Courtesy of Tenmat Ltd)



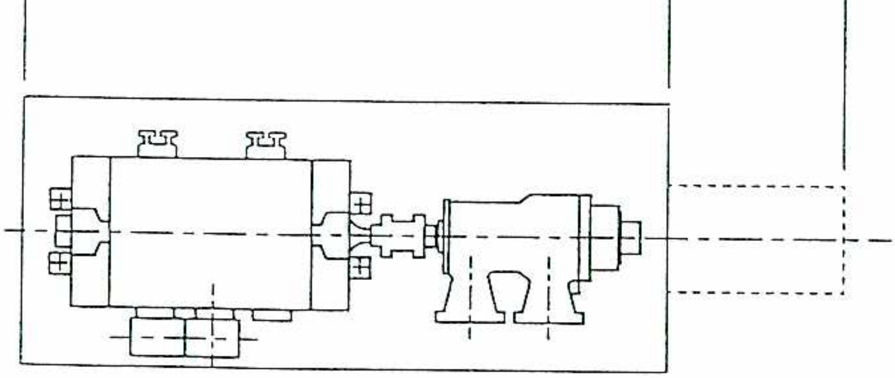
STANDARD OIL SYSTEM

WEIGHT = 21500 kg



PRODUCT LUB, RETRO-FIT

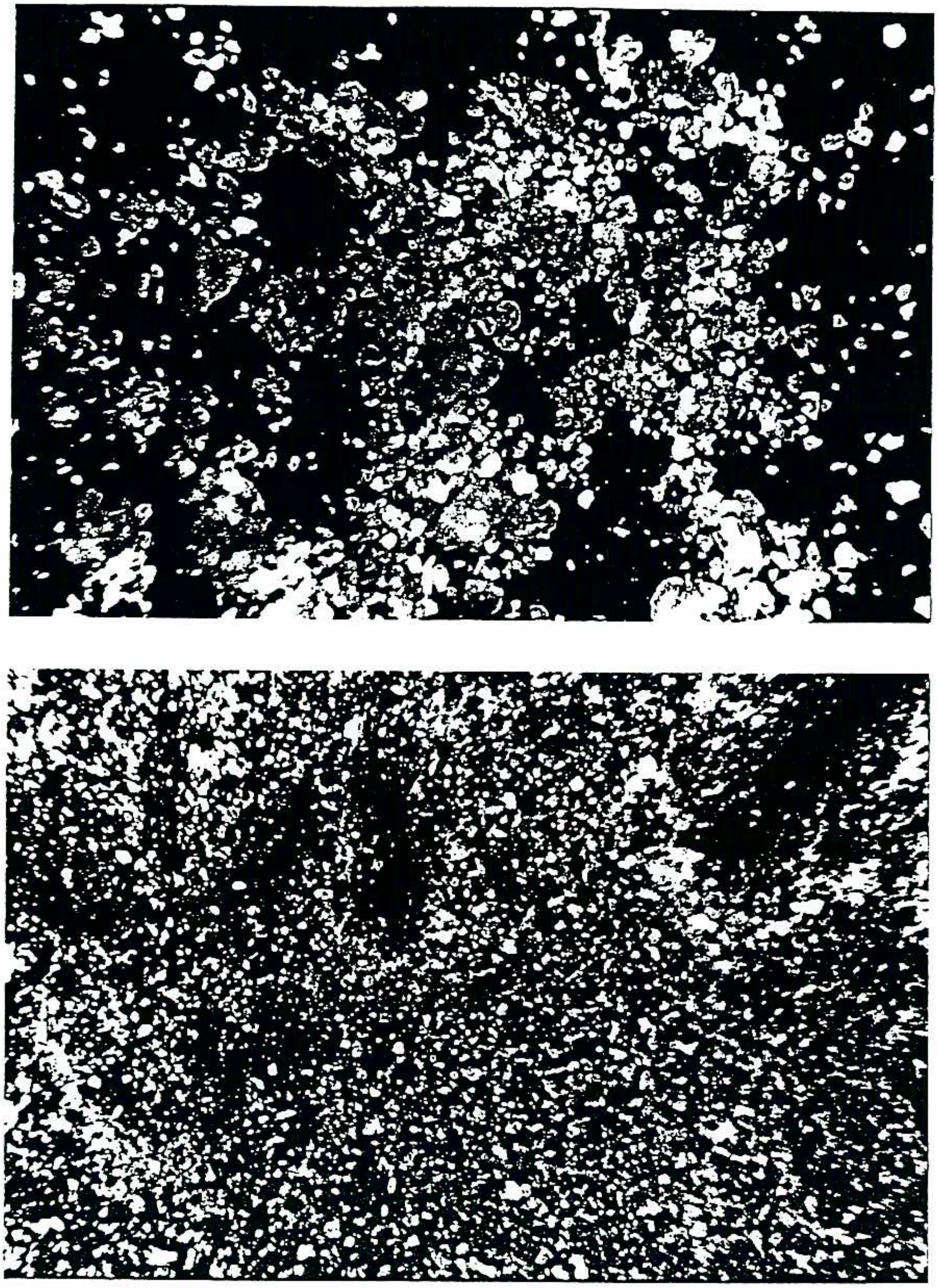
WEIGHT = 17000kg



PRODUCT LUB, SYSTEM

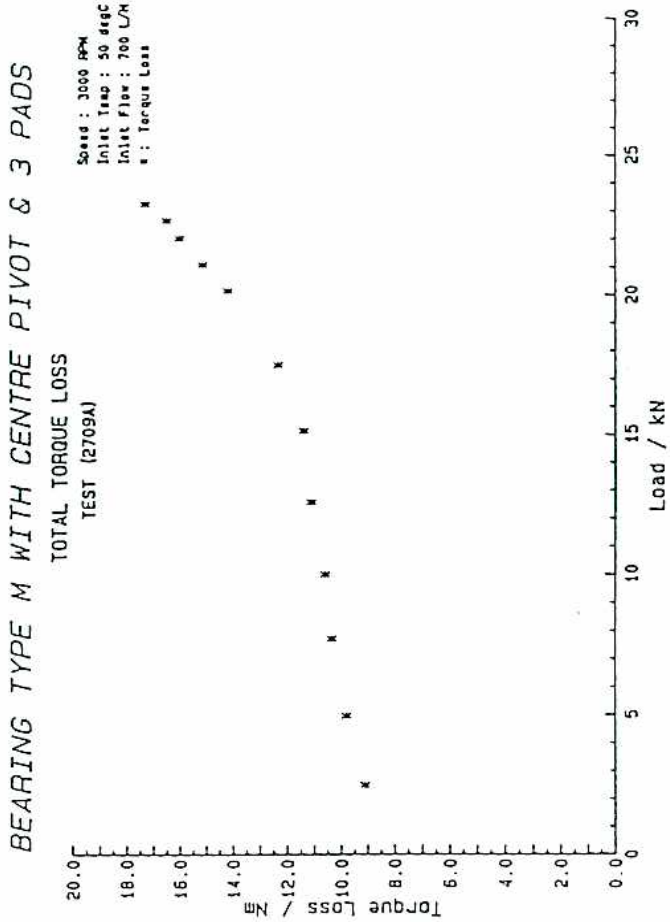
WEIGHT = 16500 kg

Fig 11 Comparison of Standard & Product Lubrication Space requirements.



Scale: 0 0.5 1.0 mm.

Fig 12 Sand before and after passing through a ceramic bearing



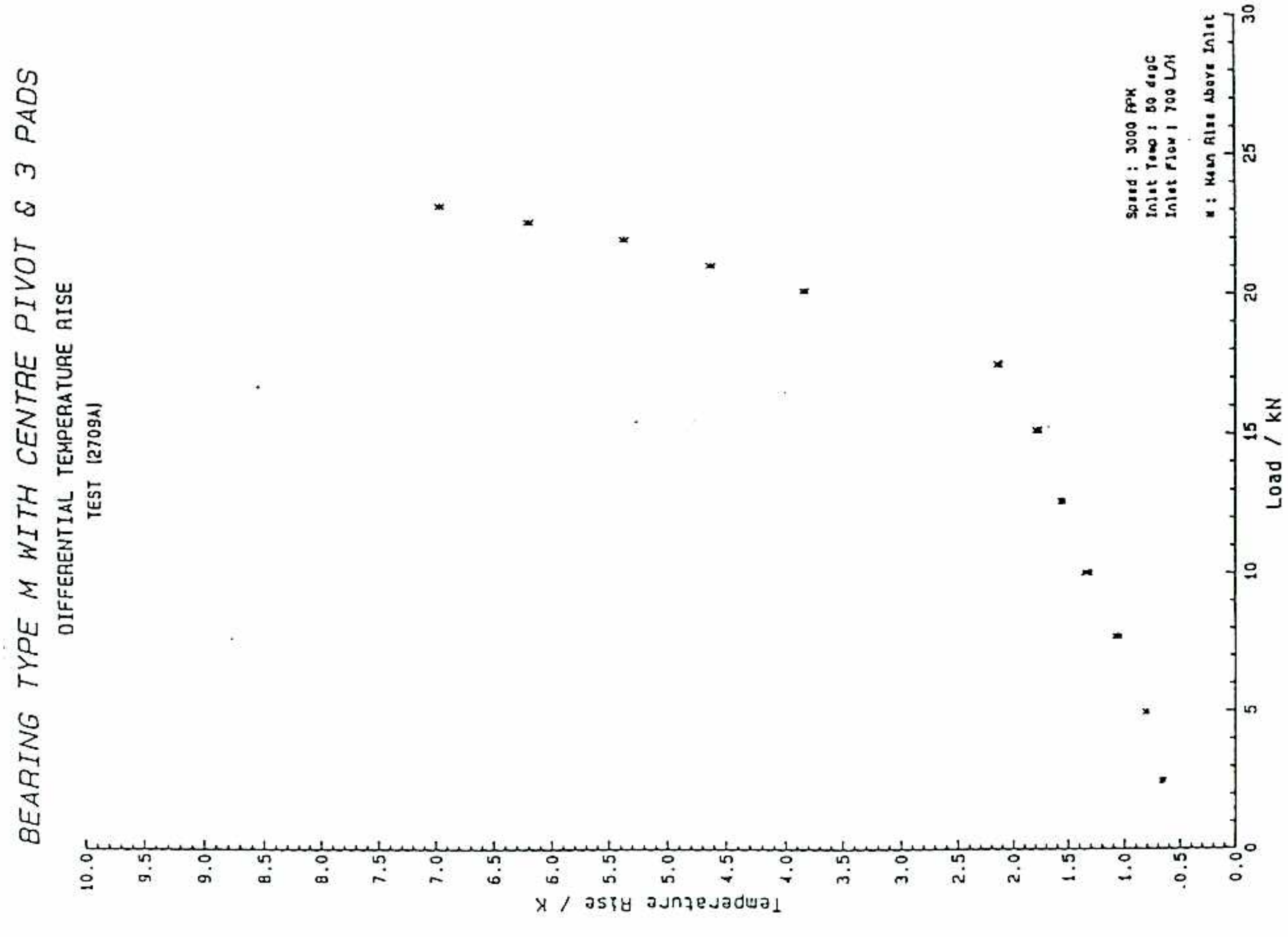
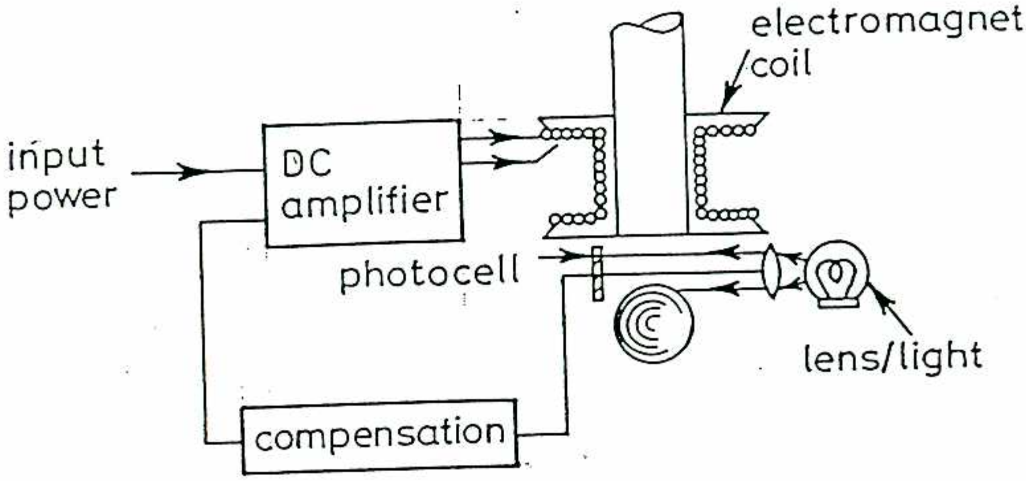
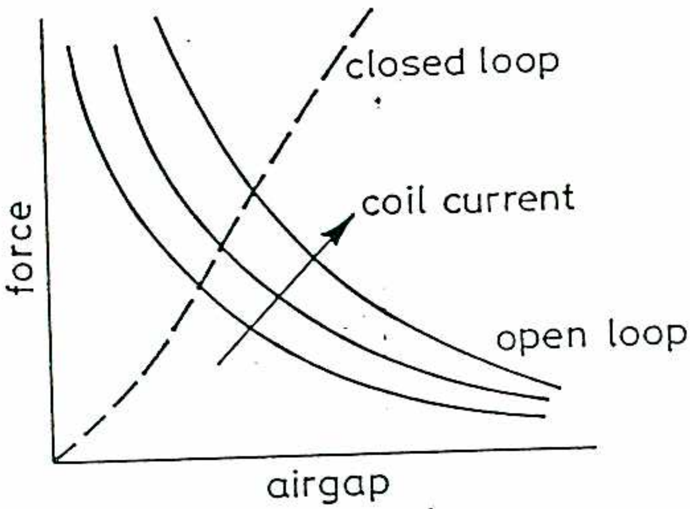


Fig 13 Temperature rise and power losses for thrust bearing loaded up to 30 MPa. (Joint Glacier, GEC. MOD(N) tests)



(a)



(b)

Fig. 14 Principle of magnetic suspension
using optical transducers

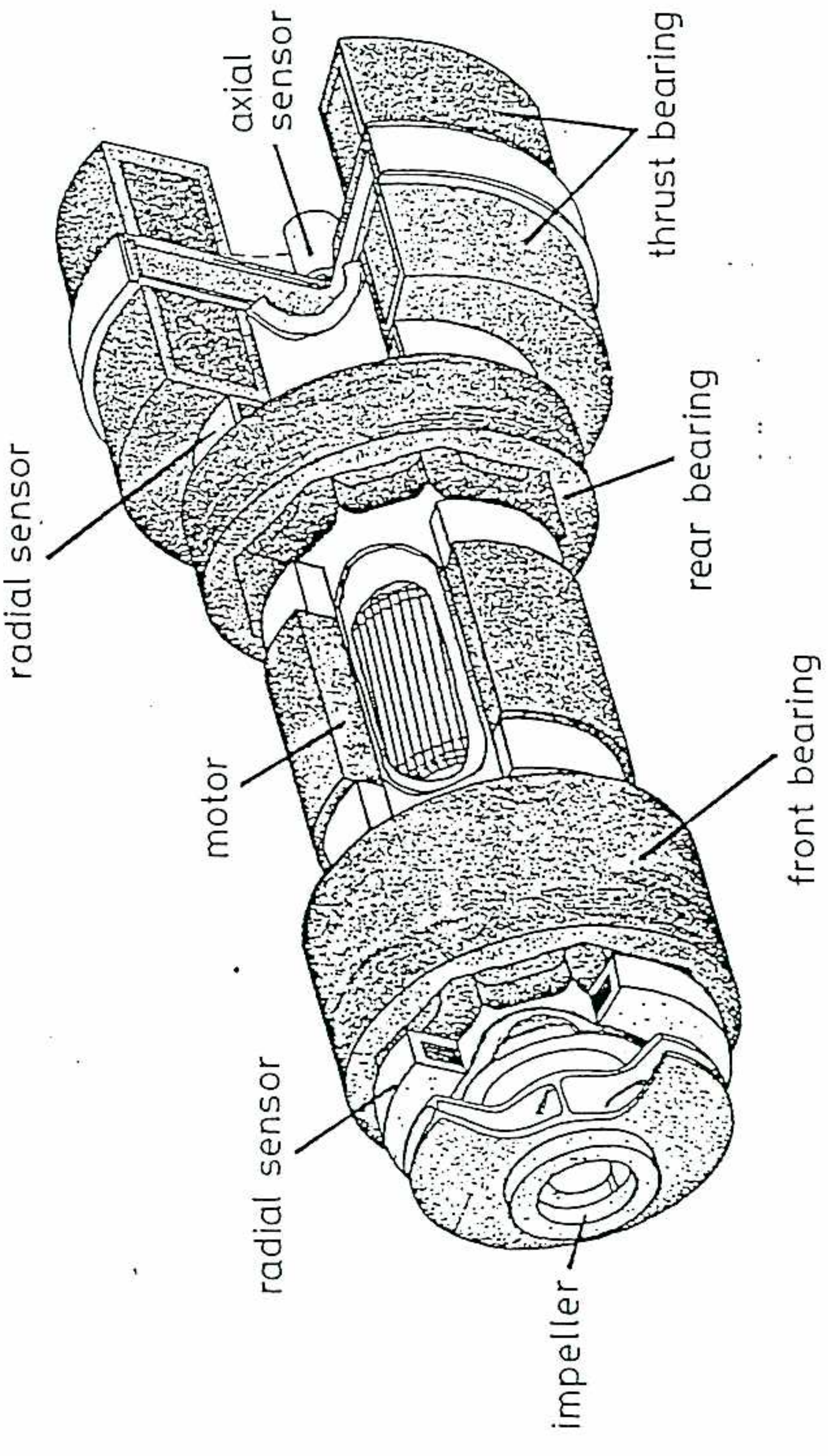


Fig 15 Pump for corrosive liquids with magnetic bearings