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VISCOSEALS FOR FREE SURFACE SODIUM PUMPS

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Summary

Viscoseals have been designed and tested for use in liquid sodium pumps to prevent the escape of radio-active cover gases along the shaft.

Such shaft seals offer the possibility of interposing an oil film of great thickness between the mating surfaces, thus making for a long life.

In tests described in the paper, these viscoseals did not show oil leakage and were almost gastight in a great range of rotational speeds and other operating conditions.

Oil leakage and gas ingestion are absent thanks to operation at relatively low Reynolds numbers on the ridges as well as in the grooves.

Axial flow and inherent transport of dissolved or ingested gas are prevented by

1. flexibly mounting the seal ring on the shaft
2. providing the ring with grooves in a way that favours the centering action of the oil film.
3. leaving parts of the ring ungrooved.

The speed range in which viscoseals for use in sodium pumps operate satisfactorily, is from 1-100 percent of the nominal speed. At standstill a stationary seal becomes operative.

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Nomenclature

P_o	$\Delta p\, h_o^2 / \eta\, \omega\, r^2$ = dimensionless pressure difference between bearing edges
P_1	$p_{m1}\, h_o^2 / \eta\, \omega\, r^2$ = dimensionless averaged differential pressure in the direction of the eccentricity
P_2	$p_{m2}\, h_o^2 / \eta\, \omega\, r^2$ = dimensionless averaged differential pressure perpendicular to the eccentricity
U	circumferential speed
h_o	radial seal clearance
l	total seal length
l_o	grooved length
p_{m1}	averaged pressure in the direction of the eccentricity
p_{m2}	averaged pressure perpendicular to the eccentricity
Δp	pressure difference between bearing edges
r	seal radius
α	groove angle
γ	ratio of the groove width and the width of a groove plus a ridge
δ	ratio of the film thickness in a groove and the seal clearance ($\varepsilon = 0$)
ε	relative eccentricity
η	dynamic oil viscosity
ν	kinematic oil viscosity
ω	rotational speed

INTRODUCTION

In Fig. 1, a rough sketch of a liquid sodium pump has been given. The pump is placed in a circuit of a nuclear breeder reactor and circulates liquid sodium carrying heat. The free sodium surface has a temperature of about 600°C. The impeller and part of the pump shaft are immersed in the liquid sodium. The hot liquid is covered by an inert gas (argon) in order to prevent oxidation and to provide thermal insulation between the sodium and the upper parts of the pump. Moreover, the inert gas plenum allows for thermal expansion of the sodium.

At the place where the shaft leaves the gas plenum and enters the ambient atmosphere, gas may leak away or ambient air may enter the gas plenum. It is clear that this is the place to provide a seal. Exclusively seals of the mechanical type have been applied in pumps of German, French, British and American manufacture. In the Dutch liquid sodium pumps we were also given a chance to try out an oil lubricated viscoseal and, thus, to compete with mechanical seals.

There are several reasons for doing so:

1. Mechanical seals frequently operate at high oil film temperatures and a minimum film thickness. Viscoseals can be so designed that oil film temperatures are well below 100°C and that film thicknesses are 10 - 100 times greater than with mechanical seals. Thus viscoseals are less sensitive to dirt in the oil.
2. Oil lubrication in mechanical seals is unpredictable. In many cases, conditions may arise provoking contact between the mating surfaces of the seal and, consequently, wear of these surfaces. In viscoseals, contact between surfaces can be prevented during running. This is attributable to designing the seal as a fluid film bearing. Thus, the life of viscoseals can be expected to be longer than that of mechanical seals. In fact, the life of viscoseals can be expected to be infinite in so far as the mating surfaces are concerned.
3. Minor faults due to dirt or incorrect running of mechanical seals may rapidly increase and will lead to leakage of large amounts of oil into the gas plenum. This will interfere with safe running conditions and requires shut-down, inspection and repair which may be quite costly.
4. In most oil lubricated mechanical seals the gas-leakage can probably be expected to be small. However, this leakage is hardly measurable. This is due to the large amount of oil required for cooling purposes. The detection of leakage is blurred by gas solving into or dissolving from the large quantity of cooling oil. Viscoseals can be so designed that gas leakage is small and so that oil leakage out of the clearance space can be prevented completely. Moreover, it is possible to measure the gas leakage thanks to the small oil inventory and to the use of a separable cooling system.

The above potential properties of viscoseals cannot be obtained by simply duplicating viscoseal designs described in the open literature. Indeed, the design viewpoint is not yet represented in literature and most of it is devoted to treating individual phenomena such as:

1. pressure build up and viscous heat generation in the laminar flow regime, Boon and Tal (1) and Hughes (2)
2. pressure build up and viscous heat generation in the turbulent flow regime, Vohr and Chow (3) and Pape and Vrakking (4)
3. optimization of groove patterns, Frössel (5) and others
4. gas bubble ingestion into the oil and leakage of oil out of the clearance space, Fisher and Stair (6) and others.

However, measuring gas leakage has surprisingly enough not yet been considered as such.

Thus, the authors were faced with designing a viscoseal that actually operates at moderate temperatures, with large film thicknesses and a minimum measurable gas leakage.
In order to achieve this, it proved necessary to apply the following design details:

1. the use of a separate cooling system
2. flexibly mounting the grooved seal ring
3. groove patterns favouring the centering action of the seal ring with respect to the bushing
4. carefully centered ungrooved, smooth zones to block axial flow carrying gas in solution or as a foam in the clearance
5. operation at low Reynolds number.

Testing on the prototype seal was particularly devoted to measuring gas leakage in a great range of rotational speeds and other operating conditions. It will be shown that the seal operates satisfactorily in a speed range from 1-100 percent of the nominal speed.

VISCOSEAL DESIGN DETAILS

A common characteristic of oil-filled viscoseals running at higher speeds, greater clearances and lower kinematic viscosities is gas bubble ingestion into the oil in the clearance space and, more often than not, oil leakage out of that space, see Fisher and Stair (6). It is possible that other quantities than speed, clearance and viscosity, such as surface tension, and surface viscosity also exert a certain influence. However, from many experiments described in literature it can be concluded that no oil leakage out of the clearance space and no serious gasbubble ingestion will occur at the Reynolds numbers lower than 100, that is Reynolds numbers based on sliding speed, radial clearance and kinematic viscosity. An example of our experiments is given in Fig. 2. It shows a grooved shaft in a glass bushing. The clearance is 0,1 mm, groove depth 0.2 mm, diameter 50 mm and the rotational speed is 2500 rpm. At the estimated working temperature the viscosity is 10 cS, so that the order of magnitude of the Reynolds number is

$$\frac{Uh_o}{\nu} = 50.$$

It can clearly be seen that no serious gas (light grey) ingestion takes place in the oil (dark grey). However, on closer

inspection the gas liquid boundary appears to be unsteady and gas bubbles tend to enter the oil across the axially directed gas liquid boundary in a groove. Sometimes traces of tiny gas bubbles or of foam can be seen at the trailing edges of the ridges. However, these phenomena do not lead to deterioration of the oil film and the oil leakage is about equal to one volume of the clearance space per week (500 mm^3/week). This leakage can be estimated quite easily by running the seal continuously without fresh oil supply and by waiting to see how long it takes to drain the seal.

In experiments like this we found that vibrating the seal and mounting the shaft in an eccentric position, do not influence the picture of gas ingestion but lead to an increase of oil leakage out of the clearance space. Without much effort we succeeded in draining the seals in one hour. These experiments followed the design of the actual seal and affirmed qualitatively the use of the following three design details:

1. operation at a Reynolds number smaller than 100 by selecting a high viscosity oil and by cooling the seal externally
2. flexibly mounting either the rotating or the stationary part of the seal
3. grooving the seal in a way that favours the centering action.

Thus, a seal, so designed, resembles a flexibly mounted, partly grooved fluid film bearing and has a comparable stiffness of the lubricant film.

This journal bearing type together with two others was first introduced by Hirs (7). Design information is insufficient in that paper and some recently collected pieces of information are presented in Fig. 3 and Table I. The partly grooved bearing in Fig. 3 is so depicted that journal and bearing are concentric and the grooves pump in the direction of the bottom of the pot. No net flow is pumped in the axial direction because it is blocked. This situation resembles that in stationary viscoseals where the pumping action generates no net flow in axial direction and where pumping generates pressure only. The pressure generation is shown in Fig. 3 also. Two pressure profiles have been given. The one on the left is characteristic of a concentric position of journal and bearing. The pressure rises linearly across the grooved zone and is uniform everywhere across the smooth zone. Thanks to the uniform pressure, there are no local flows in axial direction on the smooth zone. The pressure profile on the right is characteristic of an eccentric position of the journal with respect to the bearing. The higher pressures are typical of locations where the film thickness is at its minimum and the lower pressures an characteristic of locations where the film thickness is at its maximum. The pressure differential tends to restore the concentric position of journal and bearing. For such a pressure differential to occur, it is essential that a grooved zone passes over directly into a smooth zone or a zone with grooves of opposite inclination. No deep circumferential grooves must be located between these two zones although it might be tempting to do so.

In Table I theoretical values are given for the component of the averaged pressure differential directed in opposition to the eccentricity vector (P_1) and also the component directed perpendicularly thereto (P_2) at two eccentricities. It is clear that the averaged pressure differential is proportional to the viscosity. Thus, the centering action is increased by increasing the viscosity. It. is also shown in Table I that the pressure difference between the two edges of the bearing decreases with increasing eccentricity:

$$P_o = 0.071 \quad \text{for } \varepsilon = 0.2 \text{ and} \quad P_o = 0.035 \quad \text{for } \varepsilon = 0.8.$$

This is a general property and is not confined to this particular configuration.
A further property of eccentric operation is the occurrence of axial flows, see Fig. 3. Indeed, the net axial flow is zero. However, near the greater film thicknesses, where pressures tend to be lower than with concentric operation, the local axial flow is directed outwards and near the minimum film thickness, where pressures tend to be higher than with concentric operation, the local axial flow is directed inwards. For viscoseals this would mean that the liquid in the seal would be subjected to an overall mixing process of which the intensity increases with increasing eccentricity. Such a mixing process enhances axial transport of gas, either as a solution or as a foam, from one side of the seal to the other and casts doubt on the leak tightness of viscoseals. In any case, it is evident that viscoseals should be so designed that the eccentricity is as small as possible.

Apart from an overall mixing process, there is also a local mixing process generated by the groove pattern. In Fig. 3 it is shown that the total axial flow across a groove ridge pair may be zero. However, locally in grooves and on ridges axial flows are generated. The axial flow on a ridge is a leakage flow. The axial flow in a groove is due to the pumping action and just balances ridge leakage. The authors could get an impression of the speed of the mixing process on a grooved zone by injecting coloured liquid at the gas liquid boundary in Fig. 2. After doing so the grooved zone is uniformly coloured in a matter of seconds (speed 2500 rpm, same groove dimensions and clearance as in Fig. 3). Thus, a grooved zone is a mixer even if it is running in a concentric position and may be expected to be a poor barrier to axial transport of gas in solution or as a foam. The grooved zone may be useful for the pressure generation but hardly so for the sealing action. The sealing action can only be attributed to a zone without inclined grooves, or a smooth zone, provided that it is concentric with the bushing.

What the groove pattern for the seal looked like eventually, is shown in Fig. 4.

bearing dimensions	bearing properties		
asymmetrical, partly grooved	ε	0.2	0.8
$\frac{l}{r} = 0.23$; $\frac{l_o}{r} = 0.15$	$P_o = \frac{\Delta p\, h_o^2}{2\,\eta \omega r}$	0.071	0.035
$\gamma = 0.6$; $\delta = 3$	$P_1 = \frac{p_{m1}\, h_o^2}{2\,\eta \omega r}$	0.019	0.120
$\alpha = 0.39$	$P_2 = \frac{p_{m2}\, h_o^2}{2\,\eta \omega r}$	0.0036	0.026

Table I

The complete pattern for the seal consists of three combinations of a grooved and a smooth zone. The pumping action of two combinations is acting in the same direction and the pumping action of the combination with the longer grooved zone is directed oppositely thereto. Thus, the seal consists of three patterns, each favouring the centering action of the seal, see Fig. 3 and 4. It is felt that more than one pattern is needed for the centering action because skewed as well as parallel displacements of the grooved part with respect to the stationary part should be counteracted. Moreover, the floating gas liquid boundaries in at least two of the three grooved zones may need the assistance of at least one combination of a smooth zone and grooved zone that is completely submerged. Smooth zones are incorporated in the seal because they constitute the actual barriers to axial flow of liquid with gas in solution or as a foam.

FINAL DESIGN AND MANUFACTURING DETAILS

In Fig. 5 the final design of the viscoseal is shown. The rotating, grooved ring A is the same as shown in Fig. 4. The ring is flexibly mounted on the shaft by means of a rubber membrane B. The membrane must be leaktight and transmit the friction torque from the grooved ring to the pump shaft.

Ring A is enveloped by a stationary bushing C with a small clearance between the two parts. The sealing oil is supplied to the circumferential groove D at the inner diameter of the stationary bushing. In that groove a higher pressure than the gas pressure is maintained. When rotating in the right direction the groove pattern on ring A pumps in the direction of the oil supply groove, thus avoiding oil leakage towards the gas plenum and the ambient air.

The heat generated in the oil film is carried away by means of a cooling water flow in chamber E. In a test rig this is a simple cooling method but in a real sodium pump water is not allowed as a cooling medium and a different medium e.g. argon, air or oil can be used.

The axial balance of the rotating ring is achieved by the rolling diaphragm F and so becomes independent of the gas pressure. For the flexible mounting and the proper axial balance of the rotating ring a rubber U-shaped mounting (see Fig. 5) or a metal bellows construction can be applied also. As a matter of fact extreme care must be devoted to the leak tightness of these mountings.

When the shaft does not rotate, the supplied oil leaks continuously out of the clearance space both towards the gas plenum and the ambient. Thus the sealing action of the fluid is still present during shaft standstill. However, the leaking oil must be drained continuously. In order to avoid this the sealing action may be transferred to a separate standing seal, activated by compressed air or oil.

The annular chamber H between the rotating grooved ring A and the stationary ring G serves for accumulating a certain amount of oil leaking out of the clearance space; thus no oil enters the lower parts of the pump. When rotating, the groove pattern pumps the accumulated oil back into the clearance space. It must be understood that the leakage into H can be kept well within 5 percent of the volume of H during normal starts and stops.

The need for a standing seal might be regarded as a drawback of the application of viscoseals. However, in sodium pump practice it must often be possible to shut off the lower parts of the pump by a standing seal. By doing so the upper parts of the pump including the seal and the top bearing can be dismantled and inspected.

The material of the rotating ring is a carbon steel and that of the stationary bushing is a carbon steel with a white metal lining. The grooves can be manufactured by mechanical milling or by etching. The first method gives grooves with a depth variation of plus and minus 20 percent, which is considered to be excessive for this application. Although this method could have been improved, the second method was considered to be more promising. Indeed, with etching the groove depth turns out to be plus and minus 3 percent, which is considered to be quite acceptable. The maximum diameter for the grooved ring which has been etched up till now is 300 mm.

TEST PROCEDURE AND EXPERIMENTAL RESULTS

The flow system for testing the seal is shown in Fig. 6. The purpose of this system is to control the pressure in the circumferential groove D of the stationary bushing C, see Fig. 5. Moreover, in the rare cases where oil is leaking from the clearance space, it can be supplemented from the system.
In Fig. 6 it can be seen that this flow system consists of oil reservoir, pump, cooler, filter and a valve with which the pressure can be controlled. The gas is enclosed in a vessel located underneath the seal, denoted as gas vessel. The gas pressure is controlled by a pressure reducing valve and a blow-off valve in the gas supply. Oil leaking to the gas vessel can be drained and the volume can be measured. The gas supply as wellas the drain can be shut off completely, so that a known volume of gas is enclosed. The gas pressure can be measured with a mercury or water-filled pressure gauge.

The main values measured during the tests are:

- shaft speed
- gas pressure
- atmospheric pressure
- two pressures in the clearance space
- the temperatures at two locations in the stationary bushing 0.5 mm underneath the seal surface
- flow and inlet and outlet temperature of the cooling water
- the oil leakages towards the gas vessel and the ambient

The sealing oil used is Shell Tellus 72.

The gas leakage through the seal is measured in a way which is known as the "pressure drop" method. The gas vessel contains a known volume of gas of a certain pressure and temperature. When gas is leaking through the seal, the pressure and the temperature in the gas vessel will change. By measuring these changes the amount of leaking gas can be estimated.

The heat generated in the oil film is calculated from the temperature rise and the flow of the cooling water. The amount of heat carried away by natural convection occurring at the outer walls of the seal housing is estimated by measuring the temperature difference between the housing and the ambient. The total generated heat turns out to be 6 h.p., at most, at a rotational speed of 1000 r.p.m. The component of the heat carried away by natural convection is negligibly small.

In the present paper special attention will be given to leakage measurements. At a gas pressure of 1.1 - 2.0 kgf/cm^2 abs. and an ambient pressure of about 1 kgf/cm^2 the main results are shown in Table II.

shaft speed (rpm)	1000	550	80	10
oil diff. pressure (kgf/cm^2)	4,1-6,9	2,4-5,5	3,6-6,8	1,05-1,60
oil leakage to the ambient (cc/min)	< 0.6	< 0.8	< 0.04	< 0.30
oil leakage to the gas vessel (cc/min)	zero	zero	zero	zero
gas leakage (ncc/min)	< 0.35	< 0.31	< 0.20	< 0.03
max. seal temperature ($^{\circ}$C)	78	58	18	12

Table II

From this table it is clear that operation of the seal without oil leakage to the gas vessel is possible with an oil differential pressure of about 5 kgf/cm^2 at rotational speeds between 80 and 1000 r.p.m. With 10 r.p.m. and an oil differential pressure below 1.6 kgf/cm^2 no oil leaks to the gas vessel. This lower pressure is due to the reduced pumping action of the groove pattern at lower speeds.

Apparently no reduction in pumping action is occurring at rotational speeds from 80 to 1000 r.p.m. This beneficial effect is mainly attributable to low viscosity at higher speeds and high viscosity at lower speeds. Moreover, the seal length covered with oil can be expected to be greater at the lower speeds than at the higher speeds.

During the test program observations were also made with a shaft misalignment of 1:600 at gas pressures both higher and lower than ambient. The latter condition occurs in connection with the tandem seal construction to be discussed later on. The results with a seal oil differential pressure of 5 kgf/cm^2 and an ambient pressure of about 1 kgf/cm^2 are shown in Table III.

gas pressure (kgf/cm^2 abs.)	approx. 1.1			approx. 0.9	
shaft speed (rpm)	1000	550	80	400	80
gas leakage (ncc/min)	< 0.19	< 0.11	< 0.21	> -0.04	> -0.03
oil leakage to the ambient (cc/min)	zero	zero	< 0.16	zero	zero
oil leakage to the gas vessel (cc/min)	zero	zero	zero	zero	zero
max. seal temperature($^{\circ}$C)	53	37	18	35	19

Table III

The estimated accuracy of the "pressure drop" method used for measuring gas leakage is about $\pm$ 0.05 Ncc/min. During the tests the majority of the measured leakages was of the order of 0.05 Ncc/min. This means that the leakages given in the tables II and III are isolated peak values. It might be possible that such high leakages are due to parasitic leakages via the static seals of the test rig and that leakages via the oil film in the seal are much smaller.

SOME NOTES ON FURTHER WORK

Further work will be concentrated on two main subjects. In the first place, for immediate applications, it seems useful to develop a tandem viscoseal.
The main three reasons for developing a tandem viscoseal are:

1. To mount a space seal, so that one of the seals may break down without severe gas leakage to the ambient.
2. To create a sealing device which has an extremely good gas tightness, $< 10^{-6}$ Ncc/sec. The contaminated argon which may leak through the primary seal can be carried away from the space between the primary and secondary seal by flushing with clean argon.
3. To create between the two seals a space with the help of which the gas leakage through the primary seal can be estimated by measuring the increase of argon gas concentration in the chamber between the two seal stages.

In the second place, for more distant applications, it seems useful to develop a single viscoseal and to measure the gas leakage via the oil film more accurately. The easiest way to do so, is by measuring leakage in the set-up shown in Fig.2. In such a set-up, a small gas space is interposed between two oil filled viscoseals. In the figure light grey is gas and dark grey is liquid. The gas space can be pressurized and the gas leakage through the two adjacent viscoseals can be measured quite accurately. The set-up is also extremely useful for excluding any leakage other than leakage through the two oil films. Indeed, parasitic leakage can be expected to be negligible when some care is given to connections for injecting the leakage gas and for measuring leakage pressure. Measurements of gas leakage, which are expected to be smaller than 10^{-4} Ncc/sec. will be published in due course.

CONCLUSIONS

Practical oil-lubricated viscoseals can be designed provided that:

1. Reynolds numbers based on sliding speed, radial clearance on ridge, density and viscosity are smaller than about one hundred
2. groove patterns are selected that favour the centering action of the rotating part with respect to the stationary part
3. either the rotating part or the stationary part is flexibly mounted
4. smooth zones or zones without inclined grooves are incorporated as barriers for axial flows with gas in solution or as a foam.

A major task remains: the more accurate measurement of gas leakage through seals designed along these lines. An extra advantage of viscoseals over mechanical seals is the fact that the gas leakage is a directly measurable quantity.

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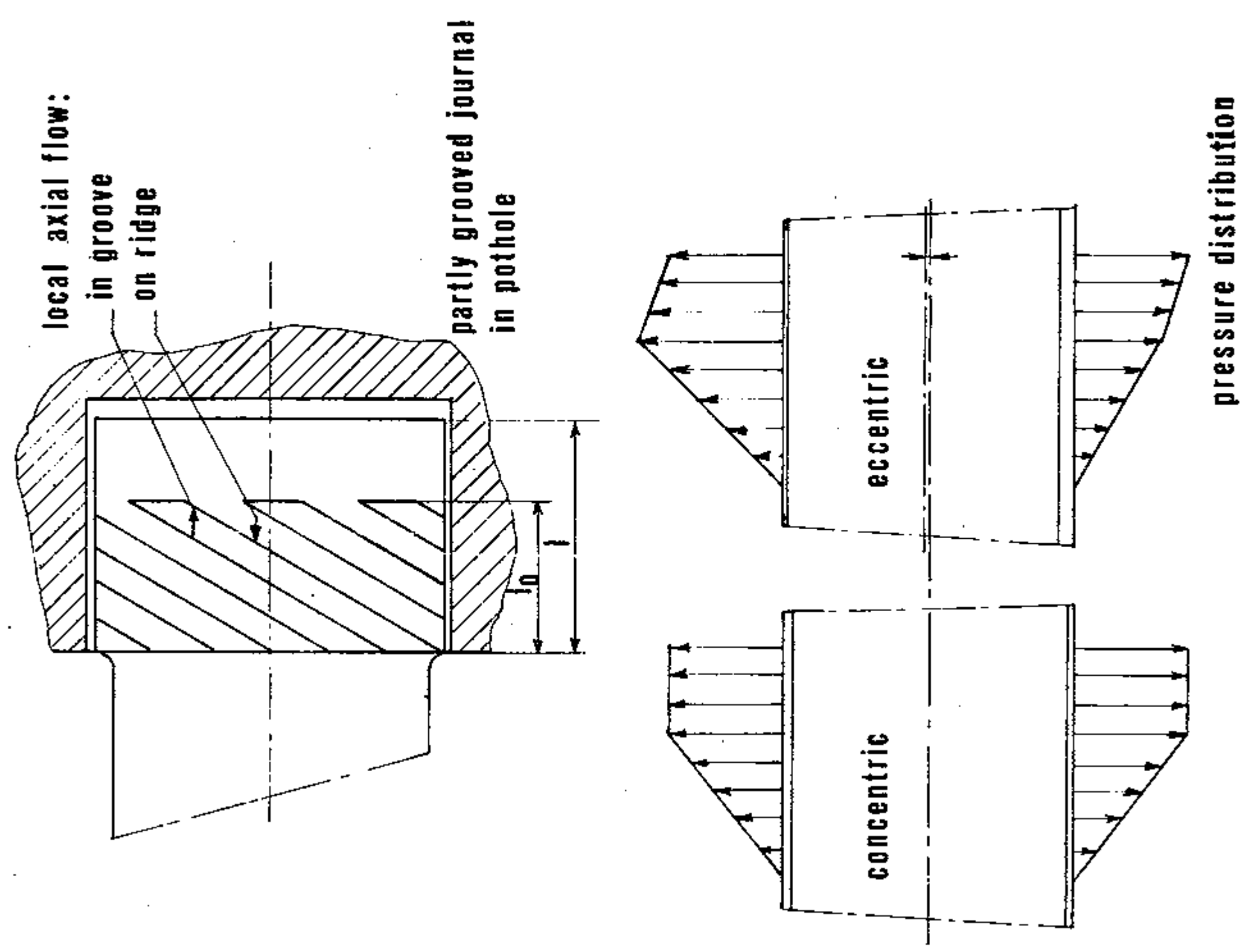


Fig. 3 Partly grooved bearing.

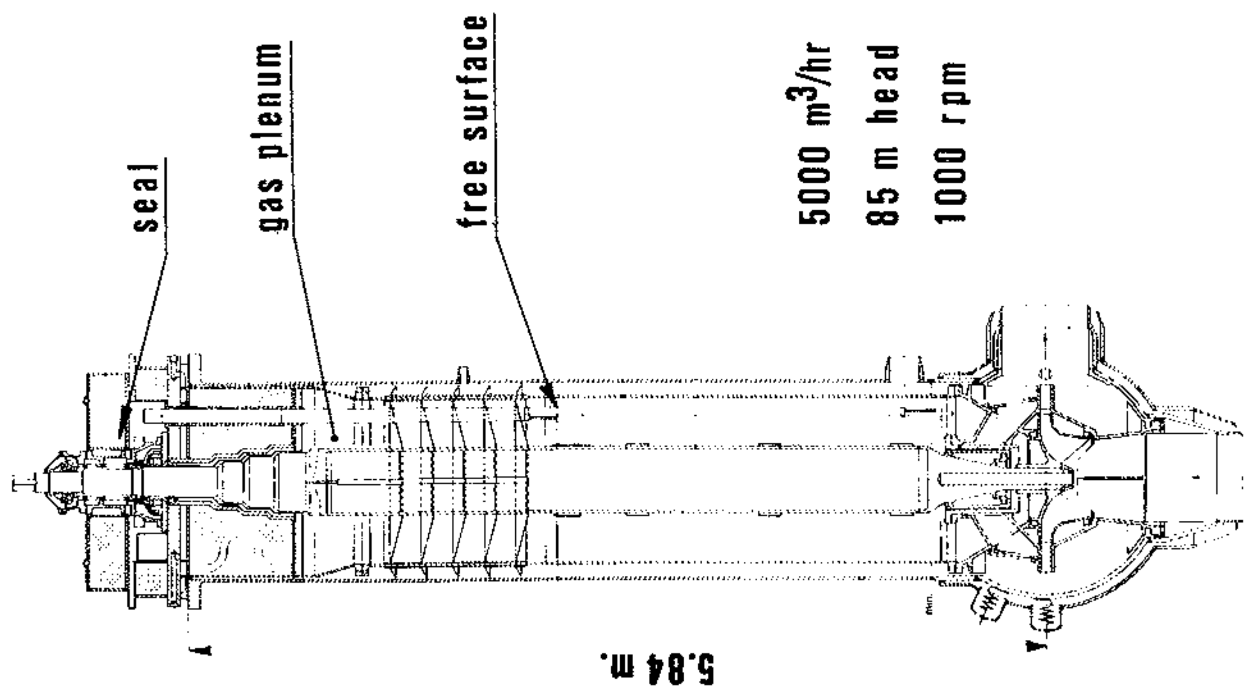


Fig. 1 Stork liquid sodium pump.

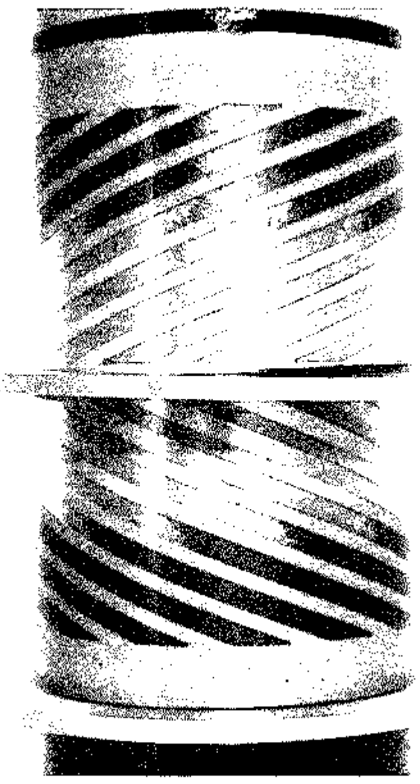


Fig. 2 Viscoseal.

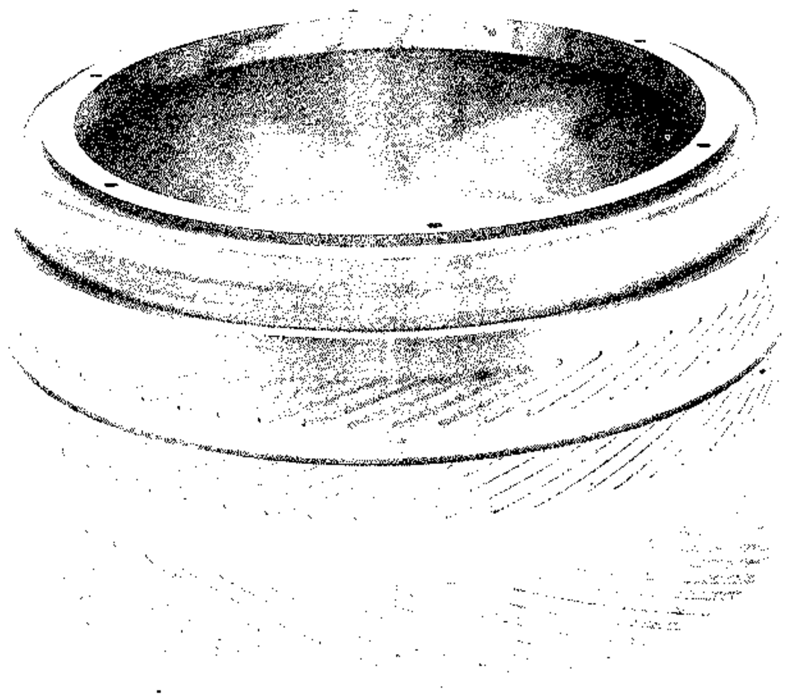
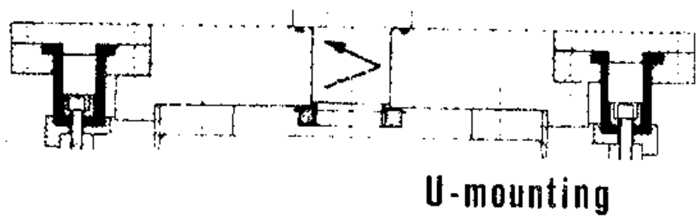


Fig. 4 Grooved ring.



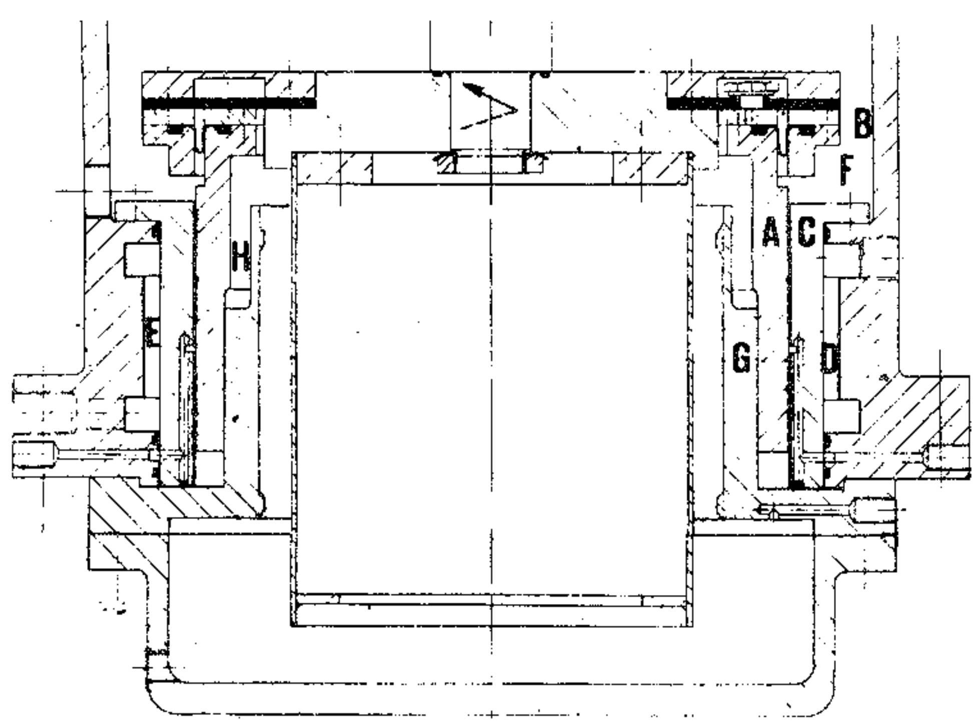


Fig. 5 Final seal design.

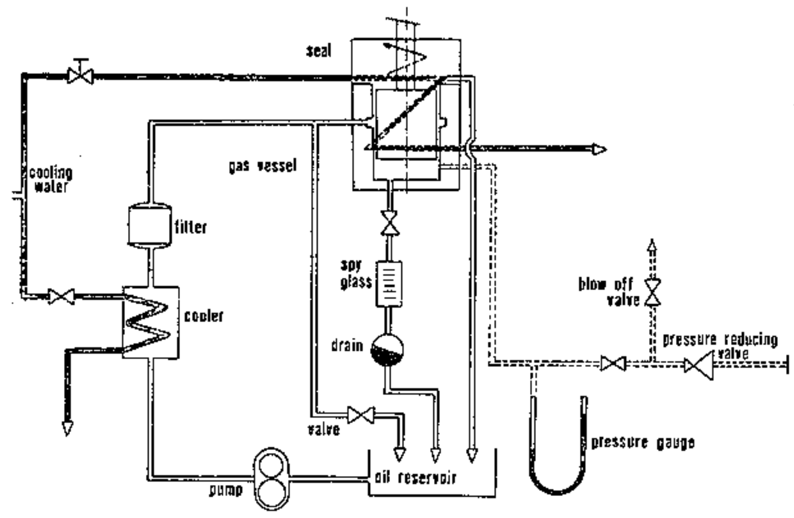


Fig. 6 Flow scheme.

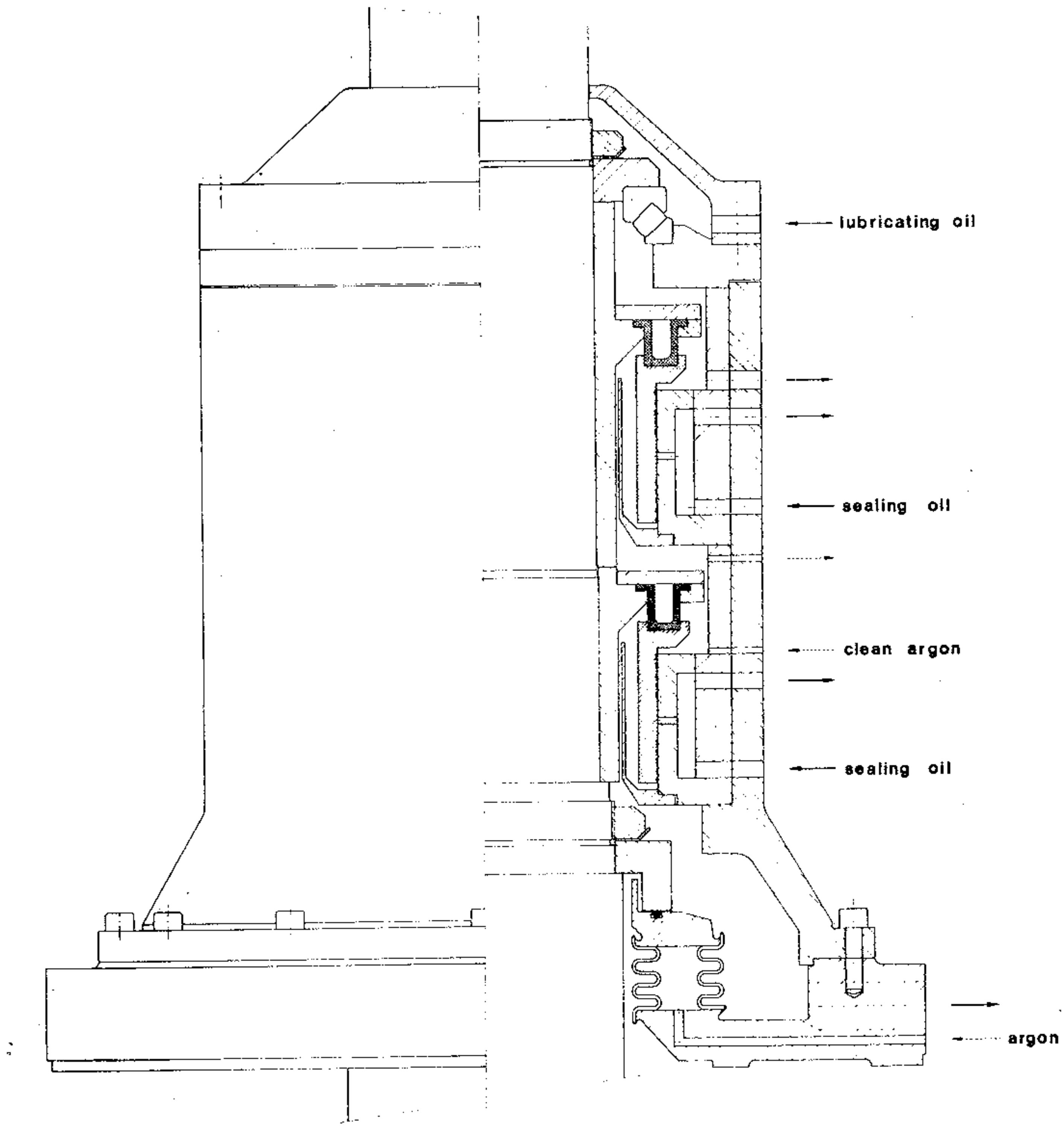


Fig. 7 Double viscoseal.