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BEARINGS AND GEARS FOR OPERATION in INERT CASES

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by

H. W. FRICKER

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BEARINGS AND GEARS FOR OPERATION IN INERT GASES

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H. W. Fricker

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BEARINGS AND GEARS FOR OPERATION IN INERT GASES

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1. INTRODUCTION

It is known that friction and wear of materials are influenced by the surrounding atmosphere. Often this atmosphere has a beneficial influence, for instance, if metal oxides of low shear strength are formed at the points of contact, leading to a reduction in coefficient of friction and wear.

In the High Temperature Gas Cooled Reactor Dragon, helium of a very high purity is used as a coolant.

The bearings and gears of various mechanisms such as the fuel element charge-discharge machine and the control rod drive mechanism run inside the pressure vessel. Some are exposed to high temperatures and to appreciable levels of nuclear radiation. Inert helium of high purity does not permit appreciable chemical reactions to take place on the rubbing surfaces. With certain rubbing combinations this will lead to high friction and wear.

The main object of our tests was to find bearings and gears which can work satisfactorily under these conditions, rather than to do basic research. For this reason a programme was set up to cover tests on plain bearings, rolling type bearings and gears.

2. PLAIN BEARINGS

In plain or journal bearings the contact stresses are relatively low due to the good conformity of shaft and bush. The parts are easy to make even if unconventional materials are used. Consequently a large choice of materials is principally available making this type of bearing look particularly attractive for applications where temperature or nuclear radiation are high.

The inherent disadvantages of plain bearings are the relatively high friction and wear, which in many cases prohibit their use in precision mechanisms.

Most of the plain bearing tests were done under contract at Napier & Son Ltd., London. A diagrammatic section through their test rig is shown in Fig. 1. A single plain bearing is tested on an overhanging shaft, enclosed by a casing which can be heated from the outside. The shaft is sealed with a face-type seal. The load is applied by means of a weight, attached to the bearing housing by a long rod. This rod is sealed to the casing by a diaphragm. The shaft is supported outside the casing by two roller bearings and driven directly by an electric motor. The whole driving arrangement is supported in frictionless hydrostatic oil bearings which permit accurate measurement of the friction torque of the test bearing and the seal. Helium, or in some cases argon at a pressure slightly higher than atmospheric, is pumped into the casing. This gas has been passed through a heated manganese-oxide bed for oxygen removal and through a molecular sieve bed for adsorption of water vapour and CO_2. After passing through the test rig it is pumped back into the clean-up plant. The small leakage through the shaft seal is replaced from a bottle. Some gas is by-passed through an analyser to measure its water vapour content. Gas can also be exhausted to the atmosphere through a Hersch cell for measuring its oxygen concentration. A diagram of the clean-up system is shown in Fig. 2. The re-circulating system has proved to be much more efficient than the straight through system used in the early tests. The purities achieved are between 10 and 20 vpm (volume parts per million) water vapour and between 1 and 5 vpm oxygen, measured at the outlet of the test rig. These values, measured at atmospheric pressure, correspond to 20 times smaller vpm figures at 20 atmospheres and compare well with the anticipated impurity concentrations of the reactor working gas.

The standard plain bearing dimensions were as follows:

diameter	$1\frac{1}{2}$ in	38.1 mm
length	1 in	25.4 mm
diametral clearance	6.5/1000 in	165 μm
or	0.43% of diameter.	

Since most plain bearing applications in the reactor are for low speed and high temperatures the test conditions were selected accordingly:

speed	100 rpm
temperature	350°C (150°C in some cases).

Tests were started at a load of 0.7 kg/cm^2 on the bearing projected area and continued at 2.1, 4.2 and 7.0 kg/cm^2 until failure occurred. Under each condition the test was run for 18 hrs where possible.

2.1 Metal, Ceramic and Plastic Bearings

In Table 1 a selection of the results from our early tests with metal, ceramic and plastic bearing materials are given (for abbreviations see Appendix I). The specific wear rates are defined as:

$$K_B \text{ or } K_J = \frac{V}{P \, . \, L} \qquad K_B \text{ for bush}$$

$$K_J \text{ for journal}$$

where V = total wear volume

P = applied load

L = total rubbing distance

Tests 1-11 were carried out in somewhat impure helium, containing probably about 100-500 vpm H_2O and 10-20 vpm O_2 (estimated values). The results of these tests are therefore not truly applicable and probably optimistic.

Although the wear rates for some of the hard combinations like Sintox/S80 or Sintox/titanium carbide were rather low the coefficient of friction was always in the order of 0.6 or more which was considered to be excessive for most bearing applications. Furthermore, scoring was evident in all cases. Particularly bad were Sintox/Sintox and Sintox/S110 (austenitic stainless steel).

The unsuccessful results with the graphite and MoS_2 filled sintered iron bushes (Tests 14 and 15) showed clearly that the dry lubricant contained in the pores of the bush is not able to reduce friction and wear to acceptable levels. Deva metal, a hot pressed bronze containing graphite, gave reasonable results at 150°C but at 350°C the result was less satisfactory.

The only plastic material tested was PTFE as used in the Glacier DU bushes where a sintered bronze is impregnated with a mixture of PTFE and lead. Fairly low friction and wear rates at 150°C could be achieved with this material. It was found to be beneficial to reduce the shaft diameter by about 10^{-3} x diameter from the valves given in the makers' handbook.

From the test shown in Table 1 and many more with similar combinations, no bearing with low friction and wear at 350°C was found and a new approach was needed.

2.2 Carbon and Graphite Bearings

It is known that these materials lose their self lubricating properties in environments containing no, or insufficient condensable vapours. [1, 2]. However, having had little success with what seemed to be promising materials, tests with graphite and carbon bushes were started; Table 2.

The first successful tests (Tests 1-3) were taken as an indication that the test head atmosphere was not pure enough. At this stage the re-circulating clean-up system (Fig. 2) was introduced and purities, as quoted, were obtained. This led to the expected high wear rates and high coefficients of friction of graphite and carbon (Tests 4 and 5). No experiments were made to find the critical concentration of water vapour for graphite lubrication. In [2] 4000 vpm of water vapour are quoted but it was evident that a much lower figure, probably around 100 vpm, would be the critical concentration in our tests.

It is understandable that the critical moisture level will depend on the test conditions, because rubbing speed and load affect the wear rate and set a time limit for the adsorption of vapours on the freshly exposed rubbing surface. This adsorption again is temperature dependent and, therefore, again influenced by environment temperature, load and speed.

Apparently MoS_2 does not need condensable vapours to maintain its lubricating properties. This material had already been used successfully on graphite brushes for high altitude aircraft equipment and also on piston rings for compressors pumping dry gases. [3]

TABLE 1

METAL, CERAMIC AND PLASTIC PLAIN BEARINGS

Temperature	350°C
Speed	100 rpm
Environment	Clean helium

* Impure Helium
\+ 150°C
⌿ 3/4" Dia. 150°C

Test No.	Journal	Bush	Pressure kg/cm^2	Duration h	Coeff. of Friction	$K_J \times 10^9$ cm^2/kg	$K_B \times 10^9$ cm^2/kg	Condition
1*	Regalox	Regalox	2.1	2	0.9	13	8.8	Lightly scored
2*	Sintox	Sintox	2.1	0.25	0.35–∞	420	196	Scored
3*	Regalox	S 80	2.1	40	0.7–0.9	1.3	3.3	Scored
4*	Sintox	S 80	4.2	20	0.7	0	0.14	Lightly scored
5*	Sintox	SF 1	2.1	40	0.6–0.9	0.66	1.9	Scored
6*	Sintox	S 110	0.35	6.5	Seizure	–	–	Badly scored
7*	Sintox	S 106	0.7	20	>1	0	1.3	Scored
8*	Sintox	TiC	4.2	20	0.7–1.2	0.33	0.05	Lightly scored
9*	J 34	Sintox	0.7	18	1.2–1.6	–	–	Scored, Build-up
10*	TiC	TiC	0.7	20	0.6	7.9	12	Badly scored
11	S 106	S 106	0.7	20	1.2–∞	–	–	Surface welding
12	J 34	J 34	0.7	18	1.2–1.6	–	–	Badly scored
13	S 106	J 34	0.7	1	–	–	–	Badly scored
14	GA	SFe/MoS_2	0.7	1	1.3–∞	2360	390	Badly scored
15	GA	SFe/Gr	0.7	0.5	0.6–∞	5000	680	Badly scored
16	SF 1	Deva	2.1	18	0.2–0.3	–	1.4	Scored
17+	SF 1	Deva	7.0	3	0.08–0.15	–	–	Lightly scored
18⌿	MS	DU	7.0	60	0.15–0.24	0	Small	Smooth

TABLE 2

CARBON AND GRAPHITE PLAIN BEARINGS

Temperature	$350^{\circ}C$	*	Impure helium
Speed	100 rpm	+	$580^{\circ}C$
Environment	Clean Helium	†	500 rpm
Journal finish	$<8\mu$ in CLA	S	70μ in CLA

Test No.	Journal	Bush	Pressure kg/cm^2	Duration h	Coeff. of Friction	$K_B \times 10^9$ cm^2/kg	Condition
1*	S 80	HX 10	4.2	58	0.27	0.11	Polished
2*	S 106	CY 10	4.2	18	0.3–0.4	0.02	Polished
3*	SF 10	CY 10	4.2	24	0.11–0.15	0.13	Polished
4	MS	CY 10	2.1	3.5	$0.25 \rightarrow \infty$	100	Rough
5	S 106	HX 10	2.1	10	$1.3 \rightarrow \infty$	31	Rough
6	SF 60	CY 10/MoS_2	7.0	22	0.08–0.20	0.06	Polished
7	S 106	CY 10/MoS_2	14.0	18	0.2–0.3	0.02	Polished
8	S 106	CY 10/MoS_2	28.0	18	0.2–0.3	0.03	Polished
9 +	SF 60	CY 10/MoS_2	7.0	78	0.1–0.3	0.06	Polished
10†	SF 60	CY 10/MoS_2	7.0	18	0.03–0.05	<0.01	Polished
11	MS	CY 10/MoS_2	7.0	18	0.2–0.25	0.20	Lightly scored
12[S]	MS	CY 10/MoS_2	7.0	18	0.19–0.25	0.80	Lightly scored
13	S 106	CY 132	14.0	18	0.08–0.16	0.016	Polished
14	S 106	CY 132	28.0	18	0.08–0.48	0.86	Lightly scored
15	S 106	EK 303	7.0	18	0.37–0.43	2.4	Lightly scored
16	S 106	EK 303/MoS_2	7.0	18	0.22–0.37	1.6	Lightly scored
17	S 106	EK 302	7.0	18	0.34–0.40	2.2	Lightly scored
18	S 106	EK 302/MoS_2	7.0	18	0.20–0.24	1.1	Lightly scored

Our carbon bushes were modified by drilling a series of radial holes through them and filling them with cold-pressed pegs made from pure MoS_2 powder, Fig. 3. These pegs have to be positioned so that they are at the leading edge of the contact area between shaft and bush. Wear on the carbon bush will then also produce MoS_2 debris which is smeared over the contact area and acts as a lubricant for the carbon. Tests with these bushes were successful (Tests 6-12). Coefficients of friction were generally below 0.2, when stellite shafts were used, and between 0.2 and 0.3 for nitrided shafts. The surface finish of these journals was better than 8μ in CLA, this figure being necessary for achieving the low wear rates of less than 10^{-10} cm^2/kg. The effect on wear rate by the surface finish was demonstrated by Tests 11 and 12. Hard shafts are also beneficial because they retain their smooth surface better than soft ones. A disadvantage of the plain carbon bush with MoS_2 pegs is its sensitivity to the direction of the applied load; if the shaft touches the bush first in an area without pegs the latter tend to become covered by carbon dust and are rendered partly ineffective. CY 132 carbon, which contains 15% MoS_2 uniformly distributed, does not suffer from this disadvantage. Excellent performance was obtained with this material, (Test 13).

Fig. 4 shows the specific wear rates of the CY 10 bushes with MoS_2 and of the CY 132 bushes as a function of the applied load.

For both materials the wear rate decreases from a high value at low loads to a fairly constant value at medium loads and increases again sharply at a certain load, showing a distinct limit in load carrying capacity. For the CY 132 bushes this limit was at 18 kg/cm^2 and for the CY 10 bushes with MoS_2 pegs it is estimated to be around 25 kg/cm^2, although this could not be verified accurately because of the load limitation in the test rig. The sharp increase in wear rate at the high load seemed to be caused by a breakdown of the material under the locally high contact loads. In a brittle material like carbon an increase in load, leading to large proportions of the contact points being loaded to the ultimate compression strength of the material, may result in a non-linear increase in wear rate because these contact points cannot deform plastically and will be crushed. This high wear rate would also destroy the MoS_2 enriched rubbing surface obtained in the running-in process and thus magnify itself.

The high wear rates at the low loads are connected with the test procedure. With any load the test was run for the same length of time and consequently the proportion of running-in wear is higher at low loads than at high loads. Furthermore, a series of tests with increasing load were normally done with the same bearing starting with the low loads where most of the running-in wear was experienced for this reason. Abrasive wear could also have a relatively large influence at low loads.

The EK bronze impregnated graphites showed a fair behaviour in the unlubricated condition and were, together with the Deva metal, the best materials employing no MoS_2, or PTFE, (Tests 15 and 17). Unfortunately, MoS_2 lubrication was not very effective on them. The bronze seemed to inhibit the action of the MoS_2 by forming a partial film over both journal and MoS_2 inserts (Tests 16 and 18).

3. BALL BEARINGS

3.1 Dry Running Ball Bearings

Early tests on dry running needle roller bearings were completely unsatisfactory. The needles skewed and jammed the bearing after a very short running time. It was decided to eliminate the use of roller type bearings altogether and to concentrate on ball bearings, which do not suffer from misalignment of the rolling parts relative to the tracks.

A first series of tests with deep groove ball bearings was carried out to establish the influence on the performance of the bearing material. The test conditions were:-

Speed	100 rpm
Load	5.7% of static bearing capacity
Temperature	150°C
Environment	Clean helium

The test rig used is shown diagrammatically in Fig. 5. A completely enclosed rotating casing contains the test bearing, which is loaded by hanging an eccentric weight on its outer ring. As the shaft rotates this weight remains stationary but deflects from its normal position by a certain angle to compensate the frictional torque of the bearing. This angle is measured by a differential transformer and recorded. Temperature is measured on a mercury thermometer inside a glass tube extension of the rotating casing. The test rig is evacuated under heat and then filled with helium. This helium is re-circulated until the necessary purity is achieved, the valves are then shut, the pipes disconnected and the rig started.

The results from the tests with different ball bearing materials were as follows:-

EN 31 bearings (1% C, 1% Cr) wear fast but have little tendency to seize, provided they have at least a three dot internal clearance.

Martensitic stainless steel bearings (11% C or 17% Cr) wear slower but show a tendency to transfer material between the rolling members. This leads to early seizure and makes them unsuitable for our conditions.

Tool steel bearings (18% W, 4% Cr, 1% V) have little tendency to pick-up. They are the best bearings for dry running under our conditions.

3.2 Dry Lubricated Ball Bearings

In order to try and find a more practical solution to the ball bearing problem than tool steel bearings, which are difficult to obtain in various sizes, dry lubrication of standard EN 31 bearings was tried. If the high wear rate of these bearings could be reduced significantly by dry lubrication successful bearings would result, because of their freedom from the danger of seizure if the lubricant were lost.

The first tests were made with bearings treated by a resin bonded film of MoS_2. Although this treatment gave some good results its performance was inconsistent and sometimes led to jamming of the bearing. In addition, the organic binder would suffer damage from high doses of nuclear irradiation and cancel the main advantage of dry lubricated ball bearings. Lubrication with pure MoS_2 eliminated the danger of jamming but did not give very good bearing lives. It became evident that pre-treatment was needed in order to increase the adherence of the MoS_2 to the surfaces.

3.2.1 Phosphated Ball Bearings

Phosphating of bearing surfaces is known to give a good keying surface for dry lubricants. Standard bearings were phosphated and then treated in a suspension of pure MoS_2 in acetone. It was difficult to produce a phosphate layer which was strong enough to withstand the high contact stresses of ball bearings without producing too much debris, and at the same time was sufficiently thick to provide an adequate reservoir for the dry lubricant. Another difficulty with phosphating was that, depending on the immersion time, either a reduction or an increase in bearing clearance is possible. Satisfactory results were obtained with a thin layer of a complex iron phosphate of fine structure (10 s immersion in Atram OS), treated with pure MoS_2 microsize powder in acetone. Bearings with exceptionally large internal clearance, phosphated and treated with resin bonded MoS_2 also performed well (Test 4, Table 3).

3.2.2 Nital Etched Bearings

This pre-treatment consists of etching the degreased bearing in a solution of 3% HNO_3 in alcohol for about 1 minute. After cleaning the bearing is run-in by rotating it in a suspension of MoS_2 microsize powder in acetone and dried while rotating. This treatment produces a very thin film of MoS_2 on the running surfaces. Tests showed a more consistent and satisfactory performance than other treatments tried earlier. In Fig. 6 the life of these bearings as a function of load, at 150°C in clean helium, is plotted. The life is defined as the number of revolutions a bearing performs at low coefficients of friction of about 0.01 before measurable wear takes place. Because wear of the dry lubricant film and not fatigue of the rolling parts is the limiting factor, the load-life relationship is different from that for lubricated ball bearings and can be approximated by the formula:

$$L = A \cdot e^{-B \frac{P}{C_o}}$$

where L = life in revolutions

P = radial bearing load

C_o = static bearing capacity

A, B = constants

TABLE 3

DRY LUBRICATED BALL BEARINGS

12 x 32 x 10 mm Deep Groove Ball Bearing, Rings and Balls from EN 31,
Pressed Steel Cage, Temperature – 150°C, Speed – 105 rpm.
Atmosphere – Clean Helium or Vacuum

Test No.	Treatments			Clearance µm	Rel. Load P/Co %	µ	Life 10^6 revs.
	Rings	Balls	Cage				
1	Etched + MoS_2	None	None	250	23	0.01	1.0
2	Etched + MoS_2	None	None	250	46	0.01	0.4
3	Etched + MoS_2	None	None	250	46	–	≈1.7
4	Phosph. + X 106	None	None	250	23	–	≈2.2
5	Vapour BL. + X 15	None	None	250	23	0.008	1.9
6	Vapour BL. + X 15	None	As Rings	250	23	0.01	1.0
7	Vapour BL. + X 15	None	None	40	23	0.01	0.12

For the nital etched ball bearings the constants are:

$$A \approx 5 \times 10^6$$

$$B \approx 4.5$$

These performances are obtained with standard EN 31 bearings having a three dot clearance, (C3 in international standards). This clearance is absolutely necessary in order to provide sufficient space for the thin MoS_2 layer and the small amount of debris formed during operation. For a 12 x 32 x 10 mm ball bearing the optimum amount of deposited MoS_2 is around 20 mg. Less MoS_2 decreases the life, more can lead to jamming.

3.2.3 Special Ball Bearings

A number of bearings with extra large internal clearance of about 0.25 mm have been tested. The results of these tests are given in Table 3. Tests 1-3 were with basically nital etched bearings and gave a similar performance to three dot clearance bearings. Obviously the increase in clearance is not very beneficial if this treatment is used.

On the other hand a large clearance makes the use of bonded lubricant films possible, because the danger of jamming is eliminated. This was demonstrated by the good result in Test 4 of a treatment which was not very successful on three dot clearance bearings. Good results were obtained with bearings pre-treated by vapour blasting and sprayed with X15 (MoS_2 - graphite compound with sodium silicate binder), (Tests 5 and 6.) This binder is temperature and radiation resistant. The disadvantage of these bearings is their large clearance which may prohibit their use in precision mechanisms. Decreasing the internal clearance leads to jamming and premature failure, as demonstrated by Test 7.

3.2.4 Conclusions

From the tests carried out with dry ball bearings the following conclusions can be drawn.

With the exception of tool steel bearings, unlubricated ball bearings perform poorly in helium. Particularly poor are martensitic stainless steel bearings, whereas standard EN 31 bearings wear fast but do not normally seize. Good performance can be obtained with MoS_2 lubricated EN 31 bearings. For bearings with three dot clearance a treatment consisting of etching and running-in with a suspension of MoS_2 in acetone gives good results. A similar treatment on bearings with exceptionally large internal clearance gives equally good results, but in this case bonded dry films can also be used giving the same or slightly better performance. Bonded films on three dot bearings are not recommended because of the danger of jamming.

Lubrication with colloidal graphite instead of MoS_2 is not effective.

Pre-treatment of the bearings by either acid etching, phosphating or vapour blasting is important to get reasonably consistent and long bearing lives.

Cage failures are normally not the primary cause of bearing failures. They only occur after the friction has increased or the clearance has been taken up by the wear debris. This is true for our tests which have all been done at low speeds, but may be different where high speeds produce appreciable forces between balls and cage.

Dry lubricated ball bearings run with low friction (0.004-0.03, depending on load) and negligible wear. A sharp increase in friction defines the end of their life and coincides with the point where appreciable wear starts taking place. However, most bearings could go on running for another period comparable with their life, provided the clearance is large enough to accommodate the wear debris. Too small a clearance will probably result in a cage failure.

3.3 Greased Ball Bearings

The majority of the bearings inside the reactor pressure vessel will run at moderate temperature, mostly below 150°C and will not receive more than 10^9 rads irradiation in their total anticipated lifetime of 20 years. From this point of view grease lubrication looks a feasible proposition.

It was expected that the principal cause of degradation of grease under these conditions would be evaporation of its base oil by the flow of clean helium passed over the machines for cooling purposes. Radiation should not have a major effect since the levels are within the scope of modern radiation resistant greases.

A number of commercial greases, recommended by their manufacturers for high temperature or high radiation resistance, were obtained and subjected to the following tests:

3.3.1 Tests on Greases

3.3.1.1 Accelerated Drying Tests

A small sample of grease was placed into a well of 14 mm diameter and 0.3 mm depth on a microscope slide and inserted into a glass tube. After cold purging with helium the specimen was heated to 250°C for 2 hours while the purge stream was kept at a velocity of 0.6 cm/s. The weight losses of the various greases were measured and compared (Table 4). Two greases had dried out completely and one had degelled and tests on them were discontinued. The remaining greases were used for the next test.

TABLE 4

GREASE DRYING TESTS

Grease	% lost in test 3.3.1.1	condition after 3.3.1.2
APL 700	100, friable deposit	-
APL 701	24, good	possible to rotate
Nucleol G 140	26, good	seized
Nucleol G 121	26, good	possible to rotate
Molytone 265	100, hard deposit	-
Molysil 44	29, degelled	-
Molytherm	23, good	seized

3.3.1.2 Drying of Grease Packed Ball Bearings

A ball bearing was filled with the grease and dried out completely. It was considered to be important that the ball bearing could still be rotated after drying out and was not jammed by the residue of the grease. This test reduced the number of successful greases to 2, namely APL 701 and Nucleol G 121, (Table 4). (It must, however, be stated that our initial selection of greases was by no means complete and it is appreciated that similar products manufactured by different companies may give comparable results. Our present investigations cover a much wider range of greases.)

Both these greases are silica gelled low vapour pressure mineral oils and a number of evaporation tests of static samples in the same set up as described under 3.3.1.1 were conducted with them.

3.3.1.3 Evaporation Rate Versus Time

Fig. 7 shows the results of these tests conducted with samples of APL 701. Similar results were obtained with Nucleol G 121. The evaporation rate at a given temperature can be expressed as:

$$W = k \cdot t^{n}$$

where W = weight loss

t = time

k ; n constants

The weight loss-time relationship followed the logarithmic law extremely well and it seems to be reasonably safe to extrapolate to much longer times than actually run in the experiments.

3.3.1.4 Drying-out Time Versus Temperature

Fig. 8 shows this relationship. It suggests that in order to obtain grease lives of 20 years, as needed for the fuel element charge machine, the temperature of the bearing must not exceed 150°C. An increase from 150°C to 175°C reduces the drying out time by a factor of about 100.

3.3.1.5 Evaporation Versus Gas Velocity

The effect of gas velocity on grease loss was surprisingly small and is proportional to $V^{0.08}$. This is in agreement with mass transport theory which suggest that, at the small Reynold's numbers used (0.5-44), the evaporation rate should be almost independent of the gas velocity.

3.3.1.6 Evaporation Versus Gas Pressure

These experiments were carried out by Castrol Ltd., and show that the grease loss is proportional to $p^{-0.18}$. This is somewhat surprising since the diffusion coefficient for a vapour in a gas is inversely proportional to the gas pressure. It suggests that the evaporation from a static grease sample is controlled by the rate at which the molecules can leave the surface rather than by the rate of diffusion through the gas.

3.3.1.7 Effect of Irradiation

The two greases APL 701 and Nucleol G 121 were both irradiated to 10^9 rads (γ) and their evaporation losses measured again. They showed slightly higher losses, particularly in the initial stages of drying.

3.3.1.8 Conclusion

From the point of view of drying out the two greases APL 701 and Nucleol G 121 promise to remain in usable condition after 20 years exposure in helium, provided the temperature is not more than 150°C and the gas speed is moderate. They will also remain in good condition after irradiation to 10^9 rads.

For the temperature of 150°C the evaporation loss W of the static grease sample described under 3.3.1.1 will be approximately:

non-irradiated grease:

$$\frac{W}{\%} = 1.5 \times \left[\frac{t}{h}\right]^{0.29} \times \left[\frac{V}{cm/s}\right]^{0.08} \times \left[\frac{p}{kg/cm^2}\right]^{-0.18}$$

grease irradiated to 10^9 rads:

$$\frac{W}{\%} = 2.3 \times \left[\frac{t}{h}\right]^{0.25} \times \left[\frac{V}{cm/s}\right]^{0.08} \times \left[\frac{p}{kg/cm^2}\right]^{-0.18}$$

3.3.2 Greased Bearing Tests

For various reasons the final answers about the life of a grease can only be obtained from actual bearing tests. Grease losses in a ball bearing are influenced by more parameters than the static drying tests, namely:

(a) Mechanical working of the grease, which constantly exposes a fresh grease surface

(b) Increased evaporation rate because of locally high temperatures and/or breakdown of long chain oil molecules

(c) Weeping (separation of oil from grease due to gravity or centrifugal forces)

(d) Mechanical extrusion of grease from the bearing

(e) Blowing out of grease by high gas velocities during rapid pressure changes.

Another reason for carrying out bearing tests with greases was to find out whether the absence of oxygen or other chemically active gases in the atmosphere would have a similar detrimental effect on bearing friction and wear as with dry bearings. Some suggestions for such a behaviour were made and the reason for them was finally traced back to some experiments carried out by Shell, where oils had failed to lubricate properly under high vacuum conditions.

Practical bearing tests should also establish the value of bearing seals in the prevention of grease losses.

The bearing tests were carried out on the test rig shown in Fig. 5. The test conditions were:

bearing:	deep groove EN 31 ball bearing 12 x 32 x 10 mm
speed:	105 rpm
load:	16 kg radial or 3% of dynamic catalogue load

30-50% of the free bearing space was filled with grease. This figure is considered to represent a fair compromise between maximum drying out time and minimum risk of jamming the bearing by the dry grease.

At these low loads both APL 701 and Nucleol G 121 lubricated the bearings perfectly for the longest test duration of 1,000 hours continuous running. The condition of bearing and grease after such a test indicated that much longer running times could be achieved. The grease losses were between 1.2 and 2 times those expected from the drying out tests, calculated by assuming the gas speed to be the circumferential speed of the bearing cage. Unfortunately the scatter of the grease loss data is too large to enable a mathematical law to be established. It seems, however, that most of the loss occurs in the early stages of a test, indicating a trend similar to the static drying tests.

Much of the excessive grease loss is caused by weeping. Mechanical extrusion and blowing out were not experienced, partly because of the partial filling of the bearing and the relatively slow changes in ambient pressure.

A number of common ball bearing seals, shown in Fig. 9 have been tested. Rubber seals (a) and felt seals (b) became brittle after irradiation to 10^9 rads and were therefore rated unsuitable. In addition rubber seals tend to be pushed into the bearing by fast pressure increases. Chip seals (c) are satisfactory but if they are used on both sides of the bearing the filling with the correct grease will have to be done by the bearing manufacturer, not always a very practical proposition. Nilos seals (d) are radiation and temperature resistant, stand up to pressure changes and have low friction and negligible wear. They reduce drying and to a limited extent also weeping losses.

It seems reasonable to assume that the effect on grease preservation by seals would become more pronounced under more arduous conditions (higher temperature, longer duration). The use of chip seals or Nilos seals is therefore recommended, particularly because the danger of these seals initiating mechanical bearing failures is remote.

It is doubtful whether a bearing, after a 20 year exposure to 150°C in a helium stream, would still be in working order. However, the bearings in our charge machine will only be exposed to high temperatures while the machine is in operation, which is only for a fraction of its total lifetime and less than the duration of our tests. Most of the time the bearings would be stationary at temperatures well below 100°C where evaporation and weeping losses are small. Under these conditions bearings lubricated with both APL 701 and Nucleol G 121 can be expected to last the full anticipated lifetime of the charge machine.

4. GEARS

4.1 Gear Simulation Tests

Supporting work for finding suitable gear materials has been done on the Napier bearing test rig by loading a flat specimen against the shaft. With this arrangement it was possible to achieve contact pressures and, by reducing the shaft speed to 10 rpm, rubbing speeds also comparable to those in typical gears.

The conditions of these tests were:

shaft diameter	$1\frac{1}{2}$ in	38.1 mm
shaft speed	10 rpm	
load	150 lb/in	27 kg/cm
temperature	150°C	
atmosphere	helium as for the previous tests	

The initial contact pressure for ferrous materials, according to the Hertz formula, was 2,300 kg/cm^2. In all cases a small amount of wear on the flat specimen occurred in the very early stages of testing and due to the finite width of the wear groove formed the contact pressure was decreased to near 200 kg/cm^2 as a typical value.

Only ferrous materials were tested because they are used almost exclusively for medium and high duty spur gears and their manufacturing techniques are well known.

A selection of the tests and their results is given in Table 5. (Specific wear rate K as defined under Section 2.1.) Initial tests with mild steel specimens (Tests 1 and 2) showed that reasonable performance can be obtained for short runs by applying tin plating or phosphating with MoS_2 lubrication. However, after the lubricant film is consumed, catastrophic wear takes place and these combinations cannot be recommended for higher duties.

Excellent results were obtained with mild steel journals, phosphated and impregnated with MoS_2 microsize powder, running against nitrided flats (Test 3). The vast improvement over Test 1 is due mainly to two reasons:

(a) The microsize MoS_2 powder used in this test gives a much better impregnation of the phosphate layer than the relatively coarse MoS_2 used in Test 1.

(b) The hard nitrided flat has good scoring resistance and retains its smooth surface longer than a mild steel flat. This in turn causes less deterioration of the shaft and gives longer life.

Nitrided steel on nitrided steel without lubrication showed a fair wear resistance but high coefficients of friction (Test 4). In order to reduce the friction tin plating was tried and found to be reasonably effective (Test 6), provided the tin layer was about 5 µm thick and was well bonded to the journal by using a vapour blast pre-treatment. The coefficient of friction was reduced significantly by this treatment and normally rose to a constant value of about 0.45 after 3 hours running.

The vapour blast treatment used in connection with MoS_2 lubrication was very successful, (Tests 8 and 9). The coefficient of friction of such a combination is remarkable. Starting from a value around 0.05 it reaches about 0.3 after $\frac{1}{2}$ hour running, but then decreases again to a figure below 0.1 after running for a further hour. The friction values of 0.05 indicate that the MoS_2 lubrication must have been nearly complete. The variation of friction with time can be explained as follows:

In the first stages of running the complete surface is covered with a layer of MoS_2, resulting in low friction, (Fig. 10a). At the surface asperities this layer will soon be used up and occasional metallic contacts occur leading to an increase in coefficient friction (Fig. 10b). Provided the basic metals have good wearing properties these metallic contacts do not result in macroscopic damage to the surface but have the effect of flattening the high spots either by mild wear or by plastic deformation. As this happens the two rubbing surfaces approach and the MoS_2 lying in the pockets of the surface roughness becomes compressed and helps to carry the load with small friction. In addition, some of this MoS_2 is now constantly being transferred from one pocket into another, thus lubricating the flattened high spots. (Fig. 10c.) As a

TABLE 5

GEAR SIMULATION TESTS

Test No.	JOURNAL			Flat Mat.	Dur. h	Coeff. of Friction	Flat Wear K 10^{-9} cm^2/kg	Condition
	Mat.	Pre-treatment	Lubricant					
1	MS	Phosphated	MoS_2	MS	1	0.5–0.9	40	Heavy scoring
2	MS	–	12 μm TIN	MS	2.5	0.06–0.33	0.040	Light scoring
3*	MS	Phosphated	MoS_2 MICR.	S 106	25.5	0.11–0.09	0.0012	Polished
4	S 106	–	–	S 106	7	0.3–0.8	0.18	Moderate scoring
5	S 106	–	5 μm TIN	S 14	6.25	0.05–0.45	0.10	Moderate scoring
6	S 106	Vapour blast	7 μm TIN	S 106	7	0.08–0.45	0.033	Light scoring
7	S 106	Vapour blast	0.7 μm TIN	S 106	1.5	0.5–0.9	0.53	Moderate scoring
8	S 106	Vapour blast	MoS_2 MICR.	S 106	29	0.05–0.3–0.06	0.012	Polished
9+	S 106	Vapour blast	MoS_2 MICR.	S 106	20.75	0.55–0.45	0.015	Light scoring

* 22 h at 10 rpm
2 h at 50 rpm
1.5 h at 100 rpm

+ 100 rpm 400°C

result the coefficient of friction approaches again that of pure MoS_2 and the wear rate becomes very small.

Fig. 11 shows microphotographs of the various stages of the shaft used in Test 8.

11a as ground

11b vapour blasted

11c impregnated with MoS_2 microsize powder

11d after test

In Fig. 11d the dark reservoirs of MoS_2 are clearly visible. The bright areas show hardly any surface damage and must have been efficiently lubricated by the MoS_2 from the reservoirs.

Test 9 was run at 400°C and 100 rpm with a good result. However, the high temperature caused some scoring to take place and the coefficient of friction did not assume the usual low figure.

No life limit has been established for these combinations but judging from the condition of specimens tested for various lengths of time one can conclude that the life is much more than the longest test duration of 29 hours.

4.2 Gear Tests

Tests on gears were carried out on two rigs, both employing a closed gear train with a locked-in torque. Fig. 12 shows the test head of the rig where four identical gears of 4 in diameter, 1 in width, and 40 teeth were tested. On a smaller test rig two gears of 4.5 in diameter, 0.37 in width, having 72 teeth meshing with a common pinion with 18 teeth were tested. The pressure angle for all gears was 20°. Table 6 lists some of the tests performed. An explanation of the symbols used is given in Appendix II. Tests were run at a temperature of 150°C, either in clean helium or in a vacuum of less than 6 m Torr. All gears, except the pinion in the small test rig, were vapour blasted and the tooth flanks rubbed in with MoS_2 microsize powder. In addition, the gears in Test 6 were slightly greased with Nucleol G 121.

In all tests the gears ran satisfactorily. The wear rate was negligible except in Test 2, where the gears had become misaligned due to bearing wear and in Test 3, where the softer VCN 150 gear showed some wear after 300 hours running. For the small gear rig the efficiency of the gears was around 95% at the start and decreased to a minimum of about 85% after some 50 hours of running. Thereafter it increased again to near 90%. This behaviour is in agreement with the variations in coefficient of friction observed in the gear simulation tests. However, the values for the coefficient of friction on the gears were higher, varying between 0.2 and 0.7. This indicates that the MoS_2 lubrication was not equally efficient as in the case of the simulation tests. The grease lubrication

TABLE 6

GEAR TESTS

	TEMPERATURE 150°C			ENVIRONMENT CLEAN HELIUM or VACUUM								
Test No.	Gear Specifications				P kg	P_{max} kg/mm^2	Ke_1 kg/mm^2	S_{cop} psi	n rpm	Duration h	Efficiency %	Remarks
	Size in	Finish	Material	Hardness HV kg/mm^2								
1	4/4	Cut	NITR/NITR.	700	110	42.3	0.27	1,400	52	140	–	Negligible wear
2	4/4	Ground	NITR/NITR.	800	190	58.4	0.46	2,400	52	300	–	Some wear due to bad engagement
3	4/4	Ground	VCN/150/ECN 35 ECN 35/ECN 35	450/800	30	23.2	0.07	380	52	300	–	Slight wear on VCN 150
4	4.5/1.125	Cut	NITR/NITR.	550	21	47.0	0.37	2,000	52/208	94	95–86–94	Good
5	4.5/1.125	Ground	NITR/NITR.	550	21	47.0	0.37	2,000	105/420	190	96–84–88	Good
6	4.5/1.125	Ground	NITR/NITR.	550	21	47.0	0.37	2,000	105/420	218	97–92	Grease dry

in Test 6 was effective in improving the gear efficiency which dropped from 97% at the start to a constant value of 92% after 150 hr running. At this time the grease had dried out and was of little use.

Cut gears gave similar performance as ground gears. However, they ran with considerable noise.

The permissible load values for fully lubricated nitrided gears are

$$K_{e1_{max}} \approx 4 \text{ kg/mm}^2 \qquad (4)$$

$$S_{cop_{max}} \approx 12{,}000 \text{ psi} \qquad (5)$$

The load in Test 2 on the 4 in gears was 12 and 20% of these values respectively and permitted 1 million revolutions without much wear.

For the small gear test rig the corresponding figures were 9 and 16% respectively and 4.8 millions revolutions for the pinion could satisfactorily be made (1.2 millions for the wheel) without measurable wear. Due to some misalignment in Test 3 the true load values were:

$$K_{e1} \approx 0.15 \text{ kg/mm}^2$$

$$S_{cop} \approx 800 \text{ psi}$$

or 9 and 14% respectively of the maximum ratings given in [4] and [5] for a heat treated alloy steel similar to VCN 150. After 1 million revolutions at this load slight wear had taken place, indicating that this material is considerably inferior to nitrided gears, even if the reduction in capacity is taken into account.

4.3 Conclusions

Gears made from nitrided steel, vapour or sand blasted and treated with MoS_2 have successfully carried loads of up to 20% of their capacity under fully lubricated condition. It has been possible to run for at least 1 million revolutions of the wheel without serious wear.

The coefficient of friction on the tooth flanks varied from 0.2 at the start to about 0.7 after $\frac{1}{3}$ of million revolutions and to 0.4 after 1 million revolutions. The corresponding transmission efficiencies for the small test rig were 96%, 85% and 90% and for the large gears 97%, 90% and 94%.

Grease lubrication reduced the friction losses but lasted only 150 hours at 150°C. Heat treated alloy steel gears showed wear at relatively lighter loads than those used on nitrided gears. They can only be used for light to medium duties for dry running in helium.

Ground gears ran much smoother than cut ones, but wear and efficiency were similar.

5. OUTLOOK

To summarize the experiences in our tests one may first say that friction and wear characteristics of most materials are different in clean helium as compared with air or CO_2 as typical oxidising environments. For unlubricated metals and ceramics they are normally much worse and some typical examples are the fast wearing graphite or carbon bushes and the seizing ball bearings made from martensitic steel, which perform very well in CO_2 [6].

The second major peculiarity connected with the inert atmosphere is the overall success of MoS_2, particularly on carbon bushes and ball bearings. This is again in contradiction to [6] and other workers, which have found erratic behaviour of MoS_2 lubricated bearings in air or CO_2. In oxidising environments the MoS_2 converts into molybdenum oxide at temperatures above 400°C, which are easily achieved at the points of contact. Although the present views are that MoO_3 is non-abrasive, it certainly is not as good a lubricant as MoS_2 and its formation could explain the unsuccessful results. In vacuum or inert atmospheres MoS_2 is stable up to about 1100°C and therefore not affected by the temperatures at most points of contact. A second important factor in the use of MoS_2 is the mode of application. Since MoS_2 is a dry lubricant it will not normally get back on a rubbing surface once it has been removed. (Except in the case of CY 132 carbon, where it is part of the load carrying material.) Because of that, surface pre-treatments which ensure a good retention of the lubricant are essential. Suitable pre-treatments were found to be etching for ball bearings, vapour blasting for gears, in special cases also for ball bearings, and phosphating for general use on mild steel. Without pre-treatments no reliable performance could be obtained with MoS_2.

A further remark can be made about the test specimen geometry. Tests with identical material combinations, run both as plain bearings and as gear simulation tests, showed a superior performance by the latter arrangement. This was particularly true for combinations producing some debris, like vapour blasted nitrided steel and phosphated mild steel. This shows that in a plain bearing wear debris can cause an increase in friction and wear, whereas in an arrangement like the gear simulation tests this debris cannot cause much harm. Even without debris there can be differences: the longer the contact distance between the rubbing parts, the more likely it is that local damage increases to serious proportions by the action of the damaged spot moving along the other surface and scoring it for the rest of the contacting distance. However, for a short distance of contact the damage cannot magnify itself so much and has a better chance of being smoothed out before the damaged part of the other surface contacts again. This must be remembered when relating information obtained in pin-disc tests and similar arrangements to plain bearing applications.

The problems of running bearings and gears under the conditions present in the Dragon Reactor have been overcome. The test programme, however, is continuing with the object of obtaining a more complete picture of the factors governing friction and wear in inert atmospheres. This research will no doubt lead to bearings with better performances than the present ones which, in turn, would give the designer more freedom in the layout of another reactor of the Dragon type.

6. REFERENCES

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LIST OF MATERIALS

Sintox, Regalox	Sintered Aluminium Oxide
TiC	Titanium carbide
SFe/MoS_2	Sintered iron, containing MoS_2 or
SFe/Gr	Graphite
DU	Glacier DU bearing
DEVA	Hot pressed bronze containing graphite
SF 1	Stellite, 45% cobalt
SF 60	Stellite, 70% nickel
MS	Mild steel
S 14	Case hardened steel
S 80	Martensitic stainless steel, 18% Cr
S 110	Austenitic stainless steel, 18/8
S 106	Nitrided steel, 3% Cr
J 34	Tool steel, 4% Cr, 6.5% W, 5% Mo, 2% V
ECN 35	Case hardened steel, 3.5% Ni
VCN 150	Heat treated alloy steel
GA	Meehanite GA
HX 10	Plain graphite
CY 10	Plain carbon
CY 132	CY 10 carbon + 15% MoS_2
EK 302	Lead bronze impregnated graphite
EK 303	Tin bronze impregnated graphite
MoS_2/MICR.	Film of pure MoS_2 microsize powder
X 106	Resin bonded MoS_2 film
X 15	MoS_2/graphite film with sodium silicate binder

GEAR TESTS

$$p_{max} = 0.418 \sqrt{\frac{P \cdot E}{b \sin \alpha \cos \alpha} \left(\frac{1}{R_1} + \frac{1}{R_2}\right)}$$

= Hertzian pressure at pitch point.

$$K_{e1} = \frac{P}{2b\ R_1\ Y_{e1}} \qquad (4)$$

$$S_{cop} = \frac{P\ Z\ u^{0.2}}{2.63\ R_1\ b} \qquad (5) \quad \text{(modified symbols)}$$

P = tangential force

R_1 = pitch circle radius, pinion

R_2 = pitch circle radius, wheel

b = gear width

u = circumferential speed

α = pressure angle

E = modulus of elasticity

K_{e1} = surface stress at point of single tooth engagement, pinion [4]

Y_{e1} = factor depending on gear ratio and number of teeth [4]

S_{cop} = basic surface stress, pinion [5]

Z = "zone factor", function of gear ratio and number of teeth [5]

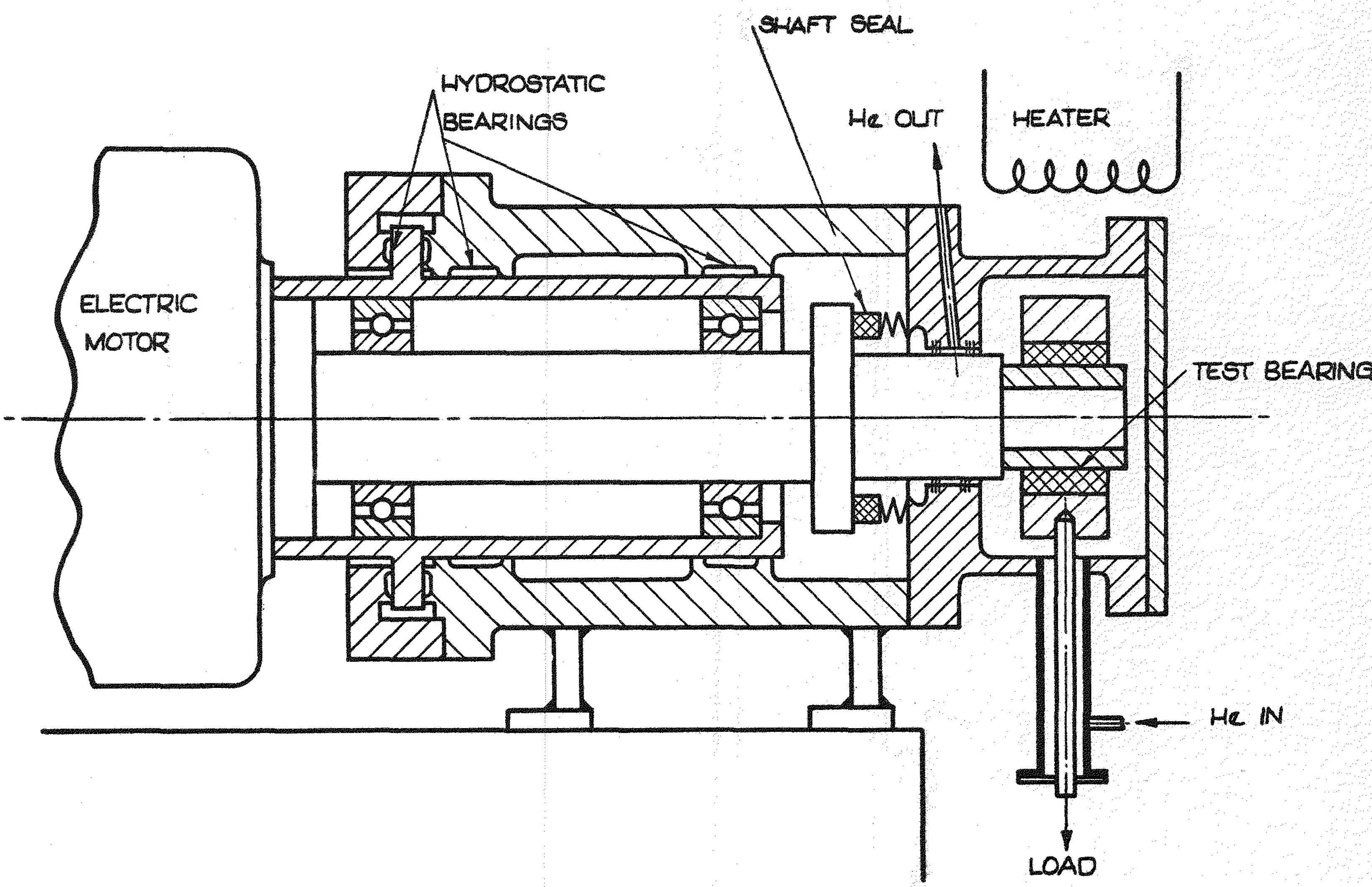


FIG 1. NAPIER TEST RIG.

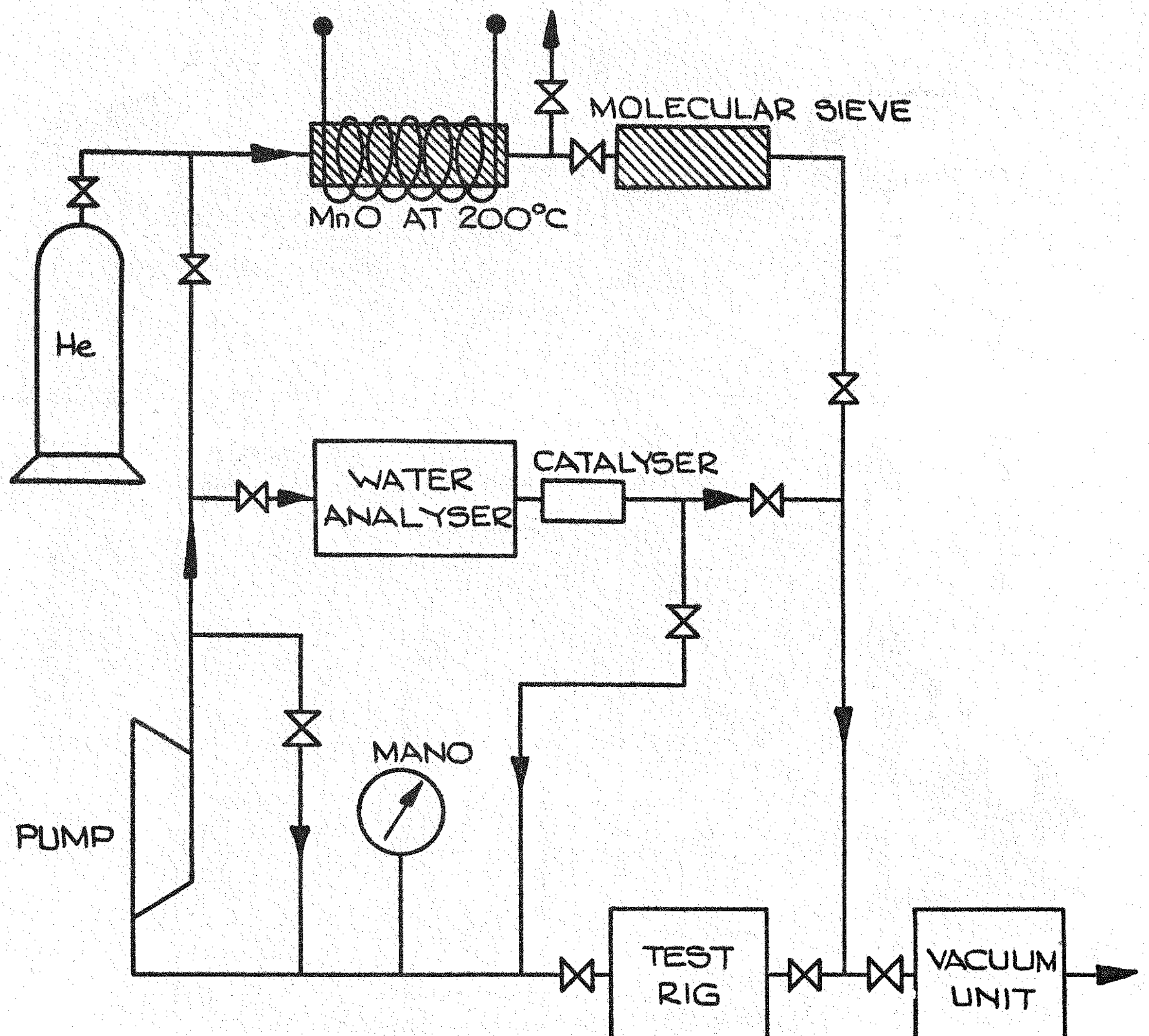


FIG 2. RECIRCULATING HELIUM CLEAN - UP SYSTEM.

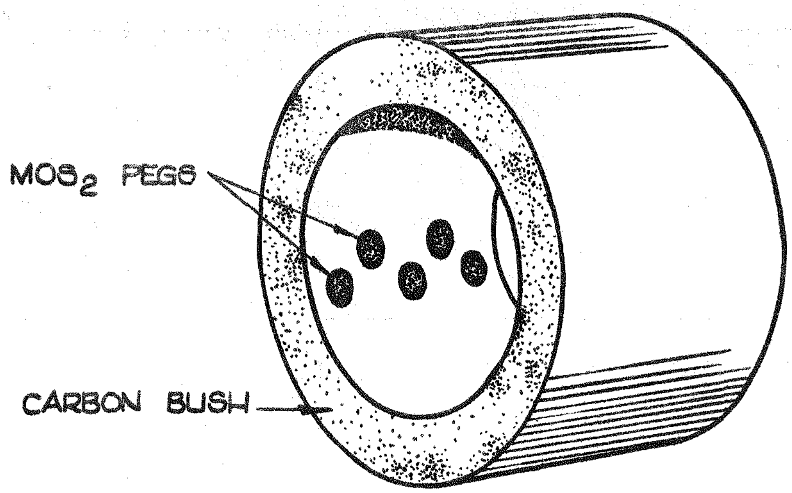


FIG 3. CARBON BUSH WITH MOS_2 PEGS.

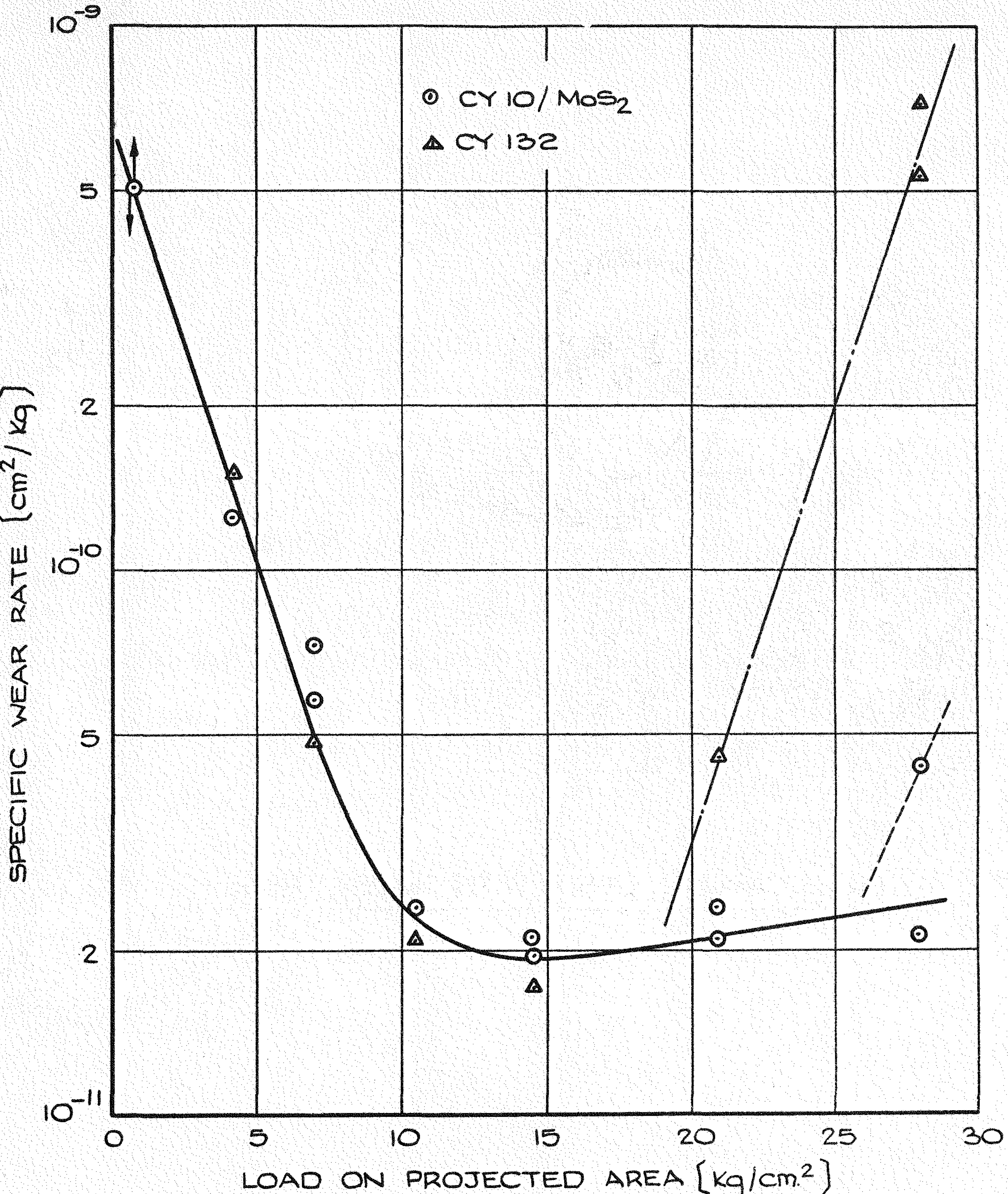


FIG 4. SPECIFIC WEAR RATES OF CARBON / MOS_2 BUSHES.

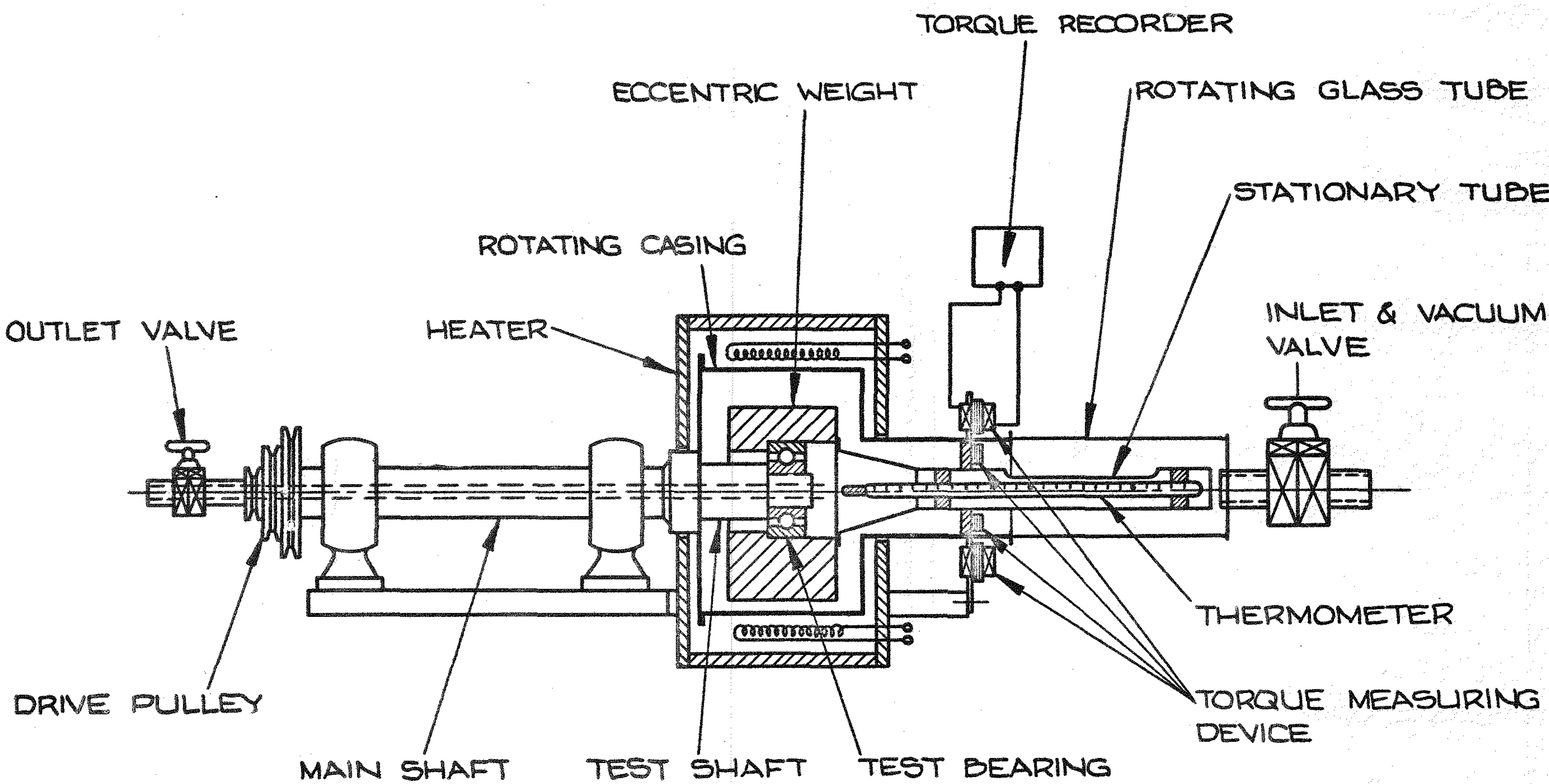


FIG. 5 BALL BEARING TEST RIG.

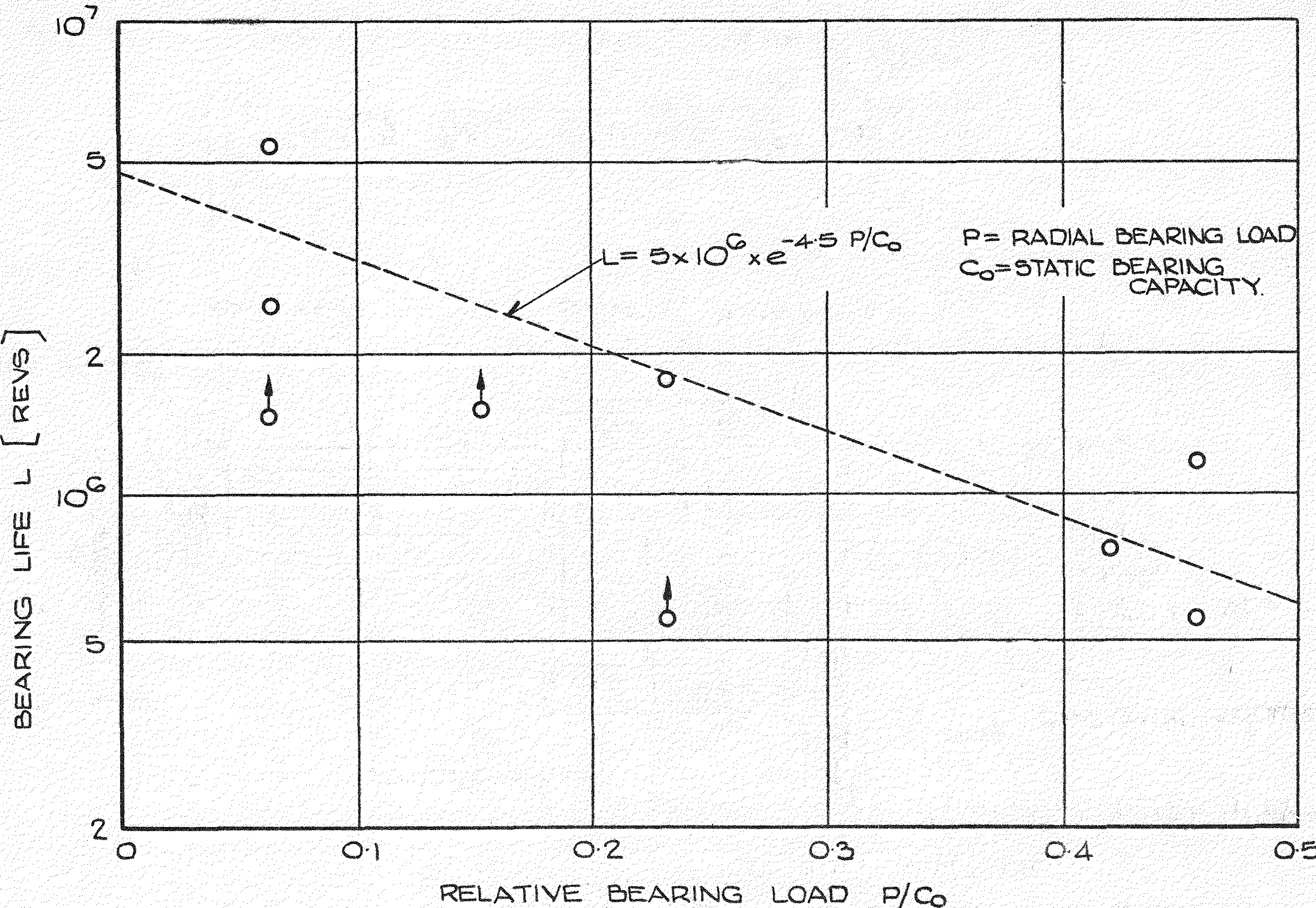


FIG 6. LIFE V/S LOAD FOR ETCHED AND MOS_2 TREATED BALL BEARINGS.

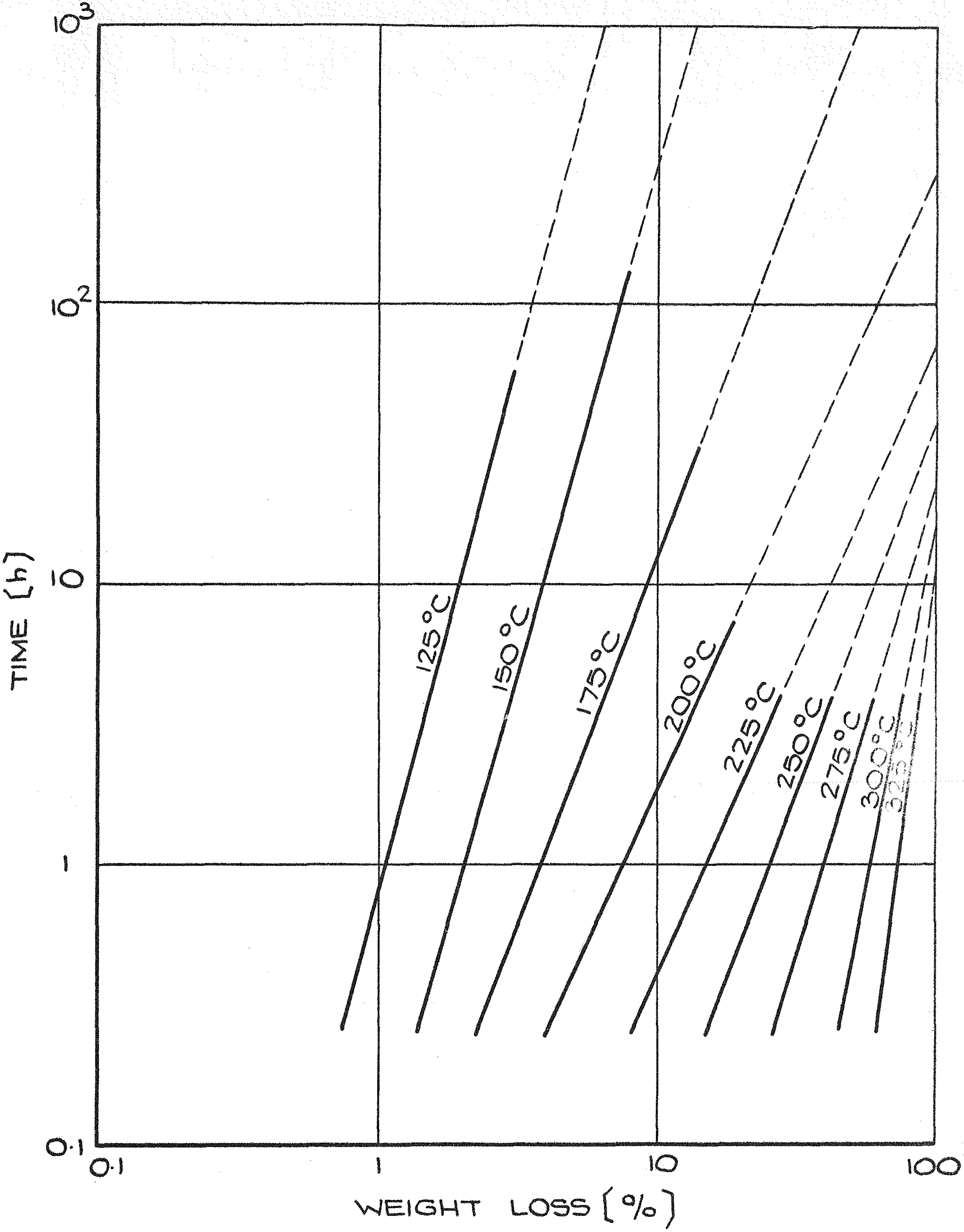


FIG 7. EVAPORATION LOSS OF APL 701 GREASE
V_{HELIUM} = 40 CM/S.

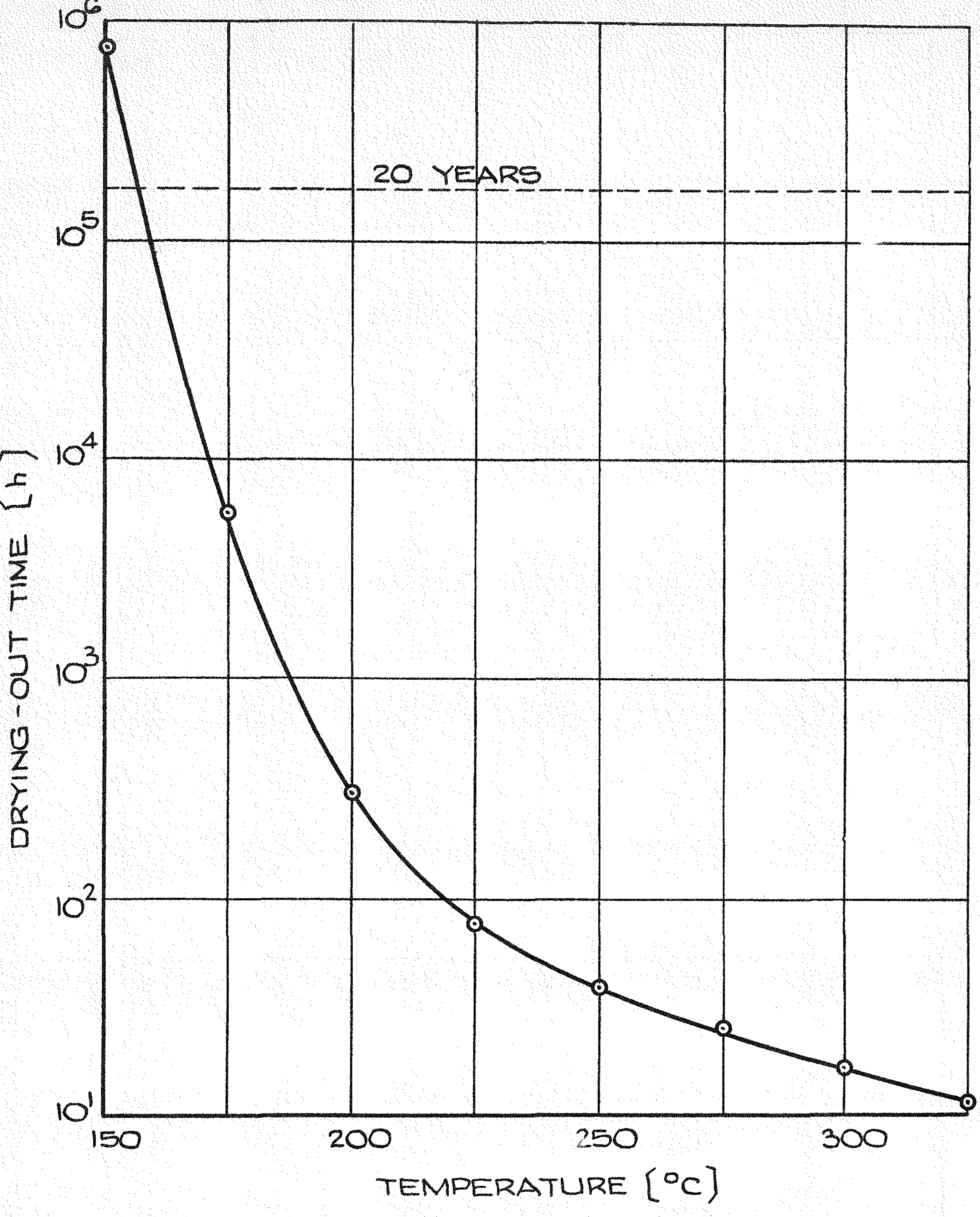
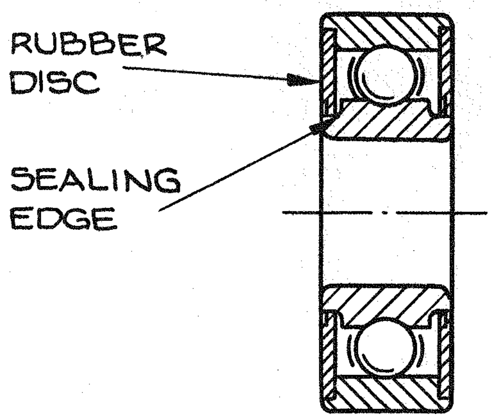
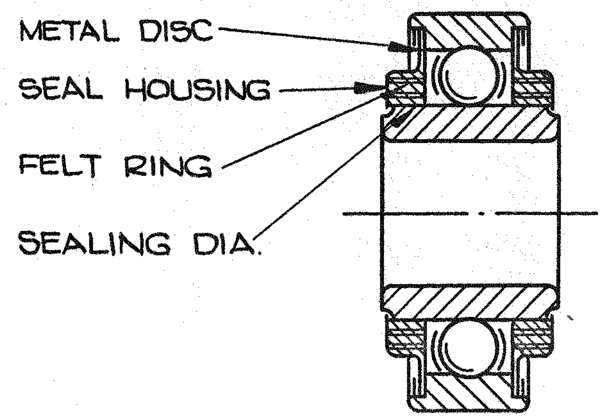


FIG 8. DRYING-OUT TIME FOR APL 701 GREASE EXTRAPOLATED VALUES FOR V_{HELIUM} = 40CM/S.

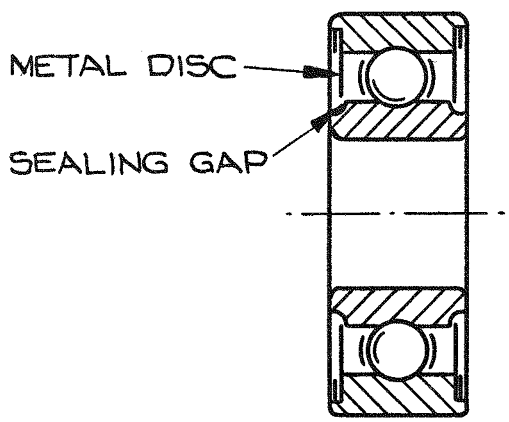
9A. RUBBER SEALS.



9B. FELT RING SEALS



9C. CHIP SEALS



9D. NILOS SEALS

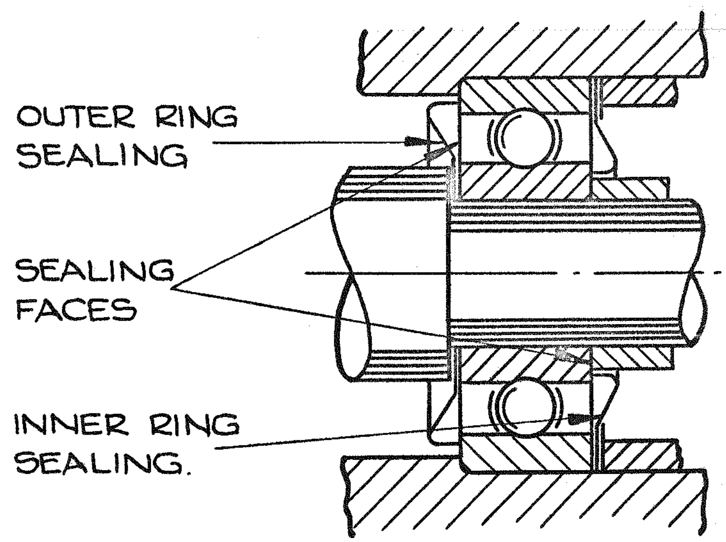


FIG. 9 BALL BEARING SEALS

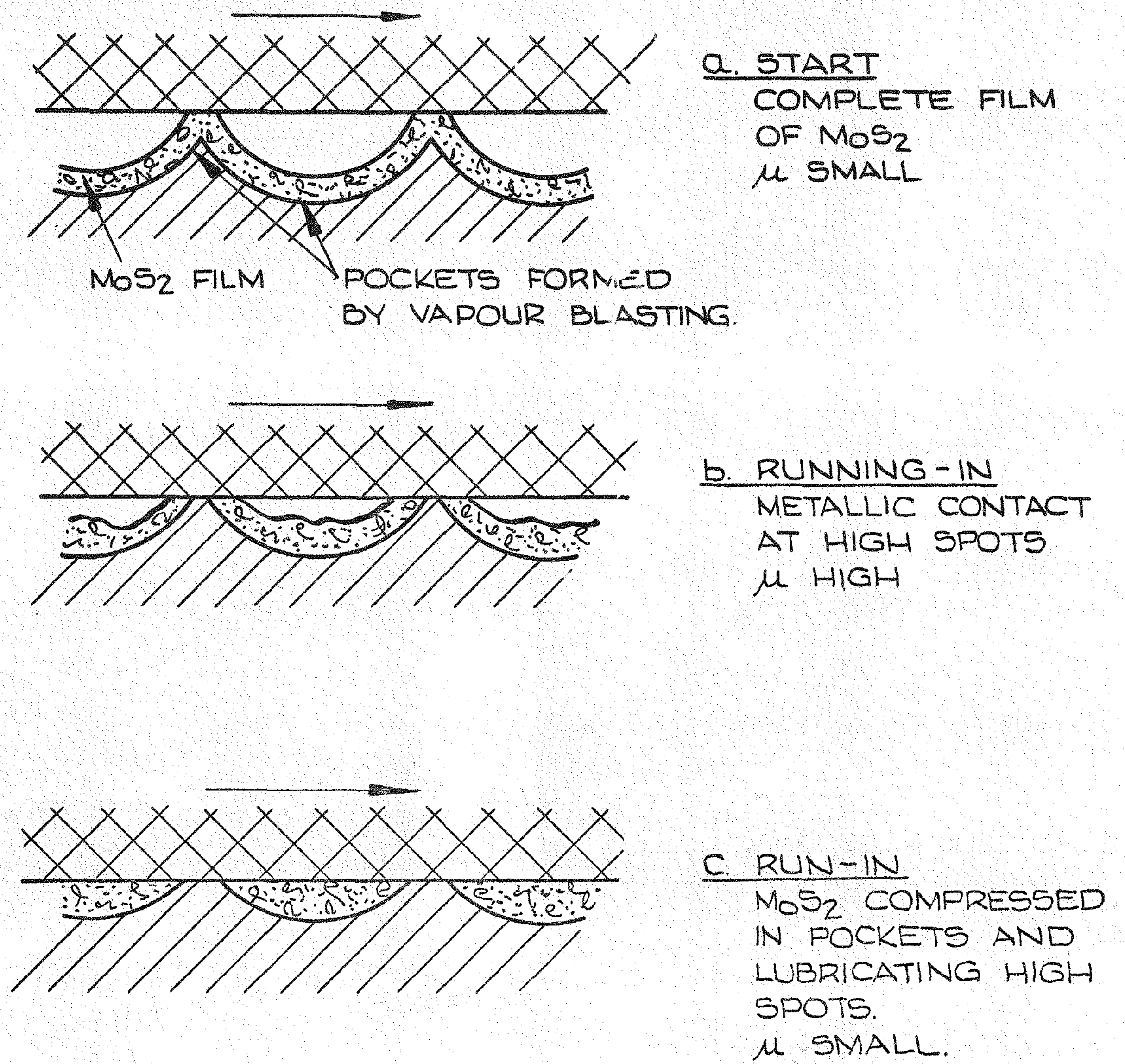
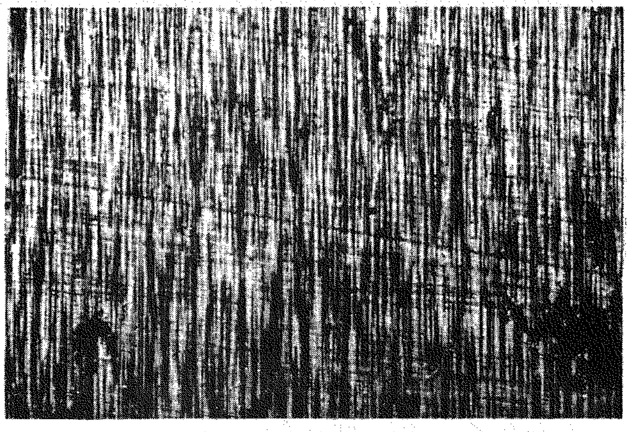
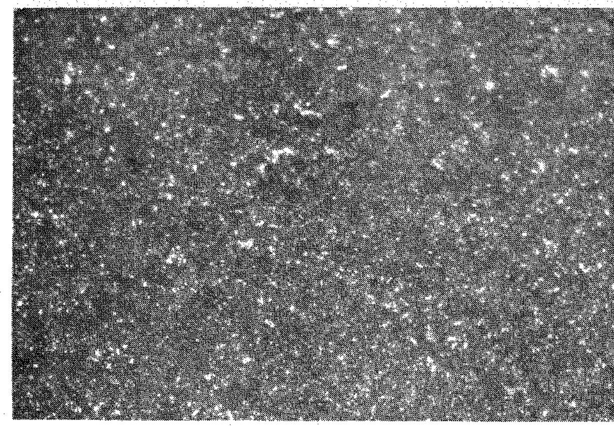


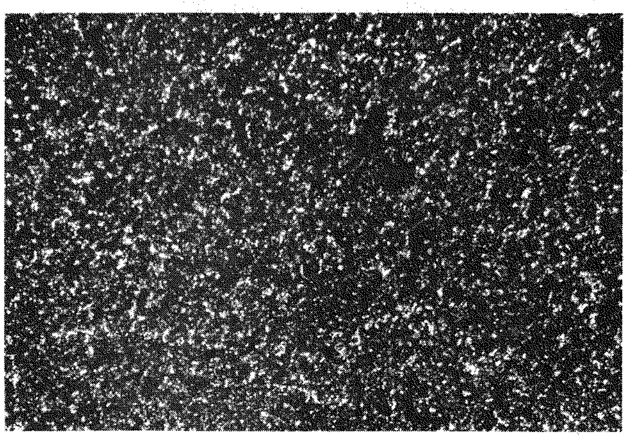
FIG 10. RUNNING-IN OF VAPOUR BLASTED SURFACE.



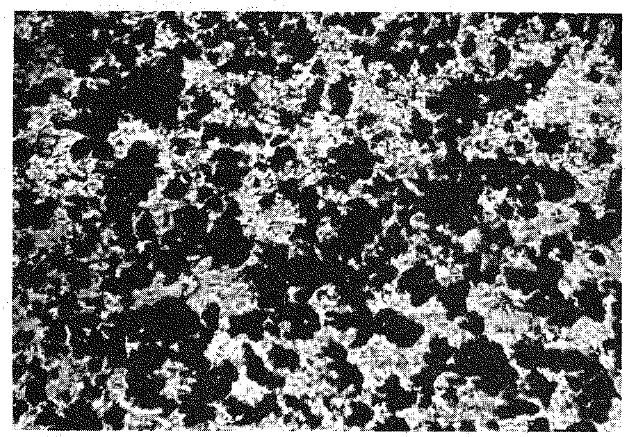
A. AS GROUND



B VAPOUR BLASTED



C. IMPREGNATED WITH MoS_2 MICROSIZE POWDER



D. AFTER GEAR SIMULATION TEST

FIG. 11. MICROPHOTOGRAPHS OF VAPOUR BLASTED JOURNAL (x 100)

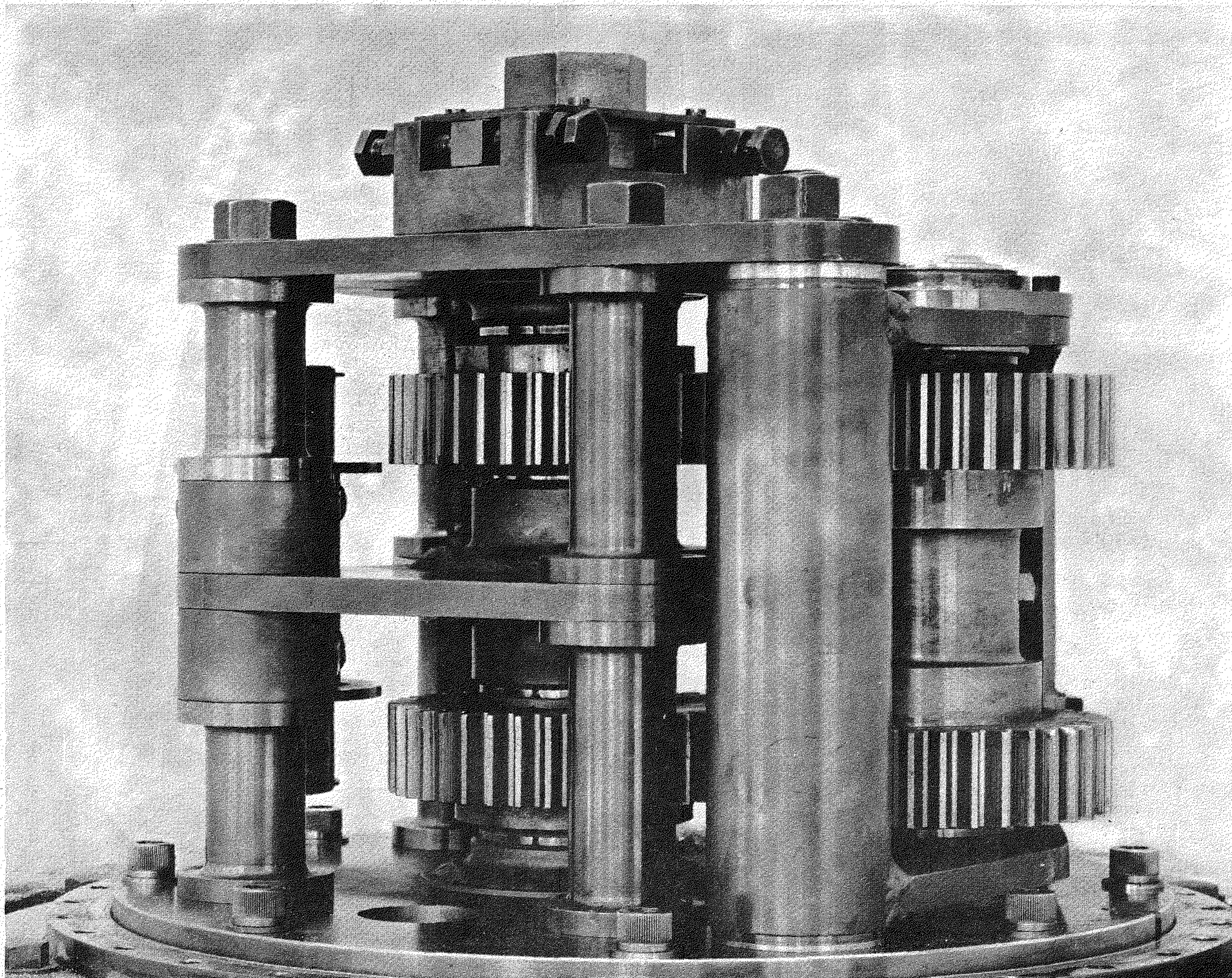


FIG. 12. TEST HEAD OF GEAR RIG