5. CURRENT EUROPEAN DEVELOPMENTS IN SOLAR PADDLE DRIVES

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SUMMARY

As part of its Supporting Technology Programme, the European Space Research and Technology Centre (ESTEC) is sponsoring the development of a number of critical spacecraft hardware items. The hardware under development includes two competing Solar Paddle Drives which are being produced to similar specifications by Hawker Siddeley Dynamics (HSD) and by Marconi Space and Defence Systems Ltd. (MSDS) in the U.K. Three mechanisms of each type are being produced and will undergo thermal vacuum testing at the newly commissioned European Space Tribology Laboratory (ESTL) at Risley, England. All mechanisms have lead lubricated bearings according to the process developed by MSDS.

INTRODUCTION

The growing acceptance of the three-axis-stabilized satellite as a vehicle suitable for future applications missions has intensified the need for lightweight solar paddles and a reliable means of orienting them towards the sun. Over the past 3 years ESTEC has been supporting various developments in European Industry in support of this activity and has laid particular emphasis on the provision of a versatile and thoroughly tested solar paddle drive, mainly because of the unavailability of reliability data on existing drives and the status of mechanisms generally as spacecraft single point failures. In September 1971 two competitive contracts were awarded to HSD and MSDS against strong international competition for the "design and build" of prototype solar paddle drives. Concurrently with these programmes the European Space Tribology Laboratory was commissioned, having a two-fold function, basic development on space tribology and vacuum life testing of the various ESTEC-sponsored mechanisms.

The mechanism specifications were intended for the development of an instrumented drive fabricated to a standard approaching that of flight hardware but were unconnected with the physical envelope constraints of a specific spacecraft structure. The designs were consequently derived by a series of trade-offs aimed primarily at mechanism reliability and with only passing deference to structural constraints; both designs were aimed at an "optimised" mechanism and, in fact, although the initial design phase was strictly competitive, both HSD and MSDS arrived at very similar concepts (2 bearing, direct drive). These designs are shown in Figures 1 (HSD) and 2 (MSDS/1).

Following their initial design MSDS were given a redesign contract which adapted their design concept to the specific requirements of a particular spacecraft installation. A controlling requirement was that the length of the mechanism should not exceed 10cm, about 1/3 that of the original design. The result, which is shown in Figure 3 (MSDS/2), was a significant improvement on the previous design and was chosen for further development.

The adoption of this mechanism over the previous "optimised" design is indicative of the stimulus of a challenging design goal.

As a result of the redesign, the characteristics of the two drives differ slightly. A summary is shown in Table 1.

DESCRIPTION OF SOLAR PADDLE DRIVES

During the design studies of both HSD and MSDS two main design considerations dictated the mechanical configuration chosen for the drive units.

The first was the requirement that the contributing loss of overall satellite pointing accuracy due to the motion of the paddle should not exceed 0.02^o. Assuming, for example, a solar paddle inertia of 15 kg-m^2/side and a relevant satellite inertia of 400 kg-m^2 and assuming also, as a worst case, that the satellite control system is unresponsive during the short time scale of the paddle movement dictates that the paddle should not step by more than 0.3^o. This restriction can be met in a simple manner by a stepper motor driving the paddle through a gear reduction. However, this configuration introduces failure modes associated with the gearing and associated bearings and the second design consideration, that mechanism reliability was paramount and that sources of tribological failure were to be minimised, resulted in the adoption of a two bearing, direct drive configuration with a pancake motor mounted directly on the shaft and driving in a pulsed mode.

The exact form of pulsed drive differs in the two mechanisms but in both the result is that, in normal operation, the array steps through about 0.1^o every 20 secs. If the mechanism fails to step, due to friction variation, the increasing error causes a progressively stronger motor pulse until the paddle conti es its rotation. Obviously, a prime objective of the test programme is to ensure the evenness of operation and to assess variability of positioning error.

The choice of motor differs in the two designs; HSD, true to the philosophy of minimising wear surfaces, chose an Aeroflex Brushless DC motor, obtaining commutation and angular position information by means of an optical encoder referred to a read-only memory. MSDS, taking advantage of the limited number of revolutions required and the relative unimportance of wear life and wear debris, used a brushed Inland DC motor and a brushed transformer. The brushes of the motor were replaced with the Ag Cu MoS$_2$ composite used for the slip ring brushes.

Differential thermal expansions caused by overall temperature variations and internal gradients are compensated in part by the use of lead lubricated angular contact bearings and by mounting one of the races on a flexible seating, axially compliant but radially stiff. MSDS, in their first design, (Figure 2) used a bellows mounting but changed to a steel diaphragm in their later design (Figure 3) because of the length restriction. HSD used a titanium diaphragm. Further relief is obtained by HSD in the choice of Beryllium as shaft and housing material. The MSDS design uses a steel shaft and an aluminium housing.

In the case of the HSD drive, launch loads are taken by an off-loading mechanism which presses the shaft against a conical seating. Release is by

TABLE 1: Characteristics of Solar Paddle Drives

CHARACTERISTIC	HSD MECHANISM	MSDS MECHANISM	QUALIFICATION TEST LEVELS		
			TEST	HSD MECHANISM	MSDS MECHANISM
Operational Speed	1 rev/day	1 rev/day	Static Load (all axes)	1750N at shaft flange	900N 0.1m outboard of shaft flange
Acquisition Speed	1 deg/sec	2 deg/sec	Shock (all axes)	deleted	100g of 0.5msec duration with 6 Kg located 8cm outboard of shaft flange
Tracking Accuracy	$\pm$ 2.5 deg	$\pm$ 0.5 deg			
Typical Step Size	0.1 deg	0.1 deg			
Shaft Resolution	0.1 deg	0.1 deg			
Av. Friction Torque (estimated)	0.07 Nm	0.46 Nm	Sinusoidal Vibration (all axes)	5 - 15 Hz 9mm o-p displacement 15 - 200 Hz 8g o-p acceleration at 2 octaves/minute	
Torque Margin	17:1	2.4:1			
Power Required (not including control)	<0.5 W average <10 W peak	<0.5 W average <10 W peak	Random Vibration (all axes)	25 - 100 Hz 3dB/oct increasing to $0.2g^2$/Hz 100 - 2000 Hz $0.2g^2$/Hz	
Slip Rings	12 Signal rings 12 Power rings (500 W) 1 Ground Ring	36 Signal rings 6 Power rings (460 W) 1 Ground Ring	Thermal Vacuum T1=Shaft Flange T2=Housing Flange Mounting and Environment	Duration(hrs)	$T1(^\circ C)$ $T2(^\circ C)$
				8 8 8 8*	20 -25 80 15 80 80 70 45
Envelope	25cm dia x 30cm	18cm dia x 10cm			
Weight	4.2 kg	3 kg	* During this period four 72 minute eclipses will be simulated by decreasing the shaft temperature (T1) to $-30^\circ C$ while maintaining T2 at $45^\circ C$.		

cable cutter. The bearings in the MSDS mechanism are unprotected during launch but are sized to withstand the severe requirements of the shock loading.

Both mechanisms use slip rings as a means for transferring power and signals and utilise a combination of a silver (82.5%) copper (2.5%) MoS_2 (15%) brush in conjunction with a silver ring. The use of these materials is consistent with the common design philosophy of dry lubrication, obviating the need for close tolerance sealing and eliminating the failure modes associated with contamination, lubricant degradation and lubricant loss through creep or evaporation. HSD use a modular design, with the slip ring unit cantilevered off the inboard bearing while the larger number of signal channels required for the MSDS mechanism led to a radial configuration for the signal rings with axially mounted power rings. MSDS limit outgassing contamination with the use of a Lithium Aluminosilicate glass ceramic as an insulating mount for the slip rings.

LUBRICANT

The lead lubricant chosen for both designs has been under development by MSDS for some years sponsored both by the Royal Aircraft Establishment (U.K.) and by ESTEC (References 1 and 2).

This method of lubrication has now reached a level of environmental testing giving a high degree of confidence in its use in long life spacecraft mechanisms. Figure 4 gives a summary of some test experience at speeds of 1/90 rpm, 100 rpm and 3000 rpm. The longest ongoing tests at these speeds have reached operating lives of 2.5 years, 2.8 years and 7 years, respectively.

Most relevant to the solar paddle application are the 1/90 rpm (16 revs/day) tests where the longest running rig has completed over 15000 revolutions without failure in a vacuum of 10^{-8} to 10^{-9} torr. Each rig was subjected to at least 50 hours air running and vibration testing before the life test in simulation of the pre-orbit environment. Unlike the liquid lubricants, where failure can be quite independent of the revolutions achieved, the concept of accelerated testing can, with care, be applied to solid lubricants. For example, Table 2 demonstrates the consistency of test results with accelerated operation. As an indication of torque noise, counts were made of the number of times the torque exceeded a certain level.

TABLE 2: Variation of torque performance with speed of 1/90 rpm test rig bearings (20N preload, average coefficient of friction ≈ 0.002).

Speed (rpm)	Average Torque (Nm)	Frequency of Peak Torque Levels (Counts/sec)				
		0.0011 Nm	0.0014 Nm	0.0017 Nm	0.0021 Nm	0.0003 Nm
1/90	0.00043	2.6	0.6	0.18	0.02	0.02
1/9	0.00040	2.1	0.5	0.10	0.04	0.00
1	0.00045	2.5	0.5	0.10	0.01	0.00

Unlike oil or grease lubricants, lead gives very little torque variation over the range of temperatures experienced by the drive. Figure 5 compares temperature characteristics for grease and lead as obtained in the 100 rpm test rigs.

TEST PROGRAMME

In order to simulate the operational environment foreseen for a flight model, each mechanism will undergo a period of air running which will be used to assess its performance when driving a load of inertia and stiffness representative of a solar paddle. Of the three mechanisms of each type, the first scheduled for delivery is the Accelerated Test Model which will be subjected to Qualification level testing for vibration and thermal vacuum (Table 1) prior to operation at an accelerated rate of 1 rev/hr over a period of about 7 months. During this time, the mechanism's performance under repeated eclipse conditions will be checked periodically at its normal rotational speed. The remaining two mechanisms, designated as Real Time Life Test Models, will undergo Acceptance Level testing and be operated for an extended period at 1 rev/day, again with periodic eclipse simulation. A controlled thermal environment will be maintained at all times.

Many past failures of BAPTA type mechanisms have been as a direct result of variations in thermal environment producing unforeseen degradations and increases in friction torque. The vacuum facilities at ESTL have been designed to simulate the mechanism environment with regard to both vacuum and thermal gradients. Figure 6 shows a typical installation. The chamber is fitted with an ion pump and, empty, can reach 10^{-10} torr. Thermal shrouds control the temperature of the mechanism mounting flange and surrounding environment (T_2). In addition, a thermally controlled pancake independently controls the shaft flange temperatures (T_1) by means of radiative coupling to a dished radiator. In this way, the mechanism can be tested in simulation of the temperature gradients enforced by the differing thermal environments of the paddle and the body of the satellite. A window is provided in the top flange for observation and, in the case of the MSDS mechanism, for control of the mechanism by the use of a mirror mounted on the shaft (see Figure 2).

Each mechanism is instrumented and temperatures, motor power, rotational movement, and slip ring voltage drops will be recorded via a data logger throughout life testing. Preload variation will also be monitored, although this is not expected to vary significantly in operation because of the flexibly mounted bearings.

PROGRAMME STATUS

The Accelerated Test Models of both mechanisms are expected to complete their life tests during the first quarter of 1974, when they will be stripped for examination. The first HSD mechanism began thermal vacuum testing in May 1973. All models are expected at ESTL by September 1973. The competitive nature of the design study is being maintained into the test phase since one of these mechanisms, or a close variant, will ultimately be chosen for use on the Orbital Test Satellite (part of the European Communication Satellite Programme) which is due for launch at the end of 1976.

ACKNOWLEDGEMENTS

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2. P.J. O'Donnell, C.L. Harris & M.G. Warwick: The Lubrication of Bearings and Slip Rings in Vacuum. ESRO CR-91, September 1972.
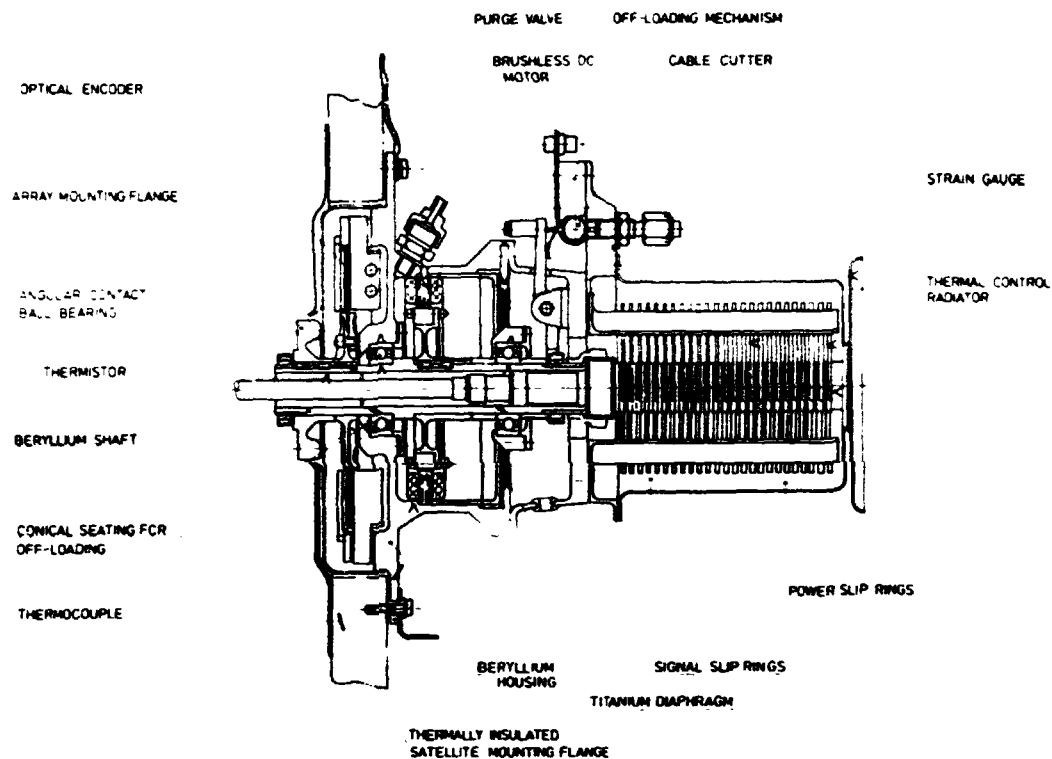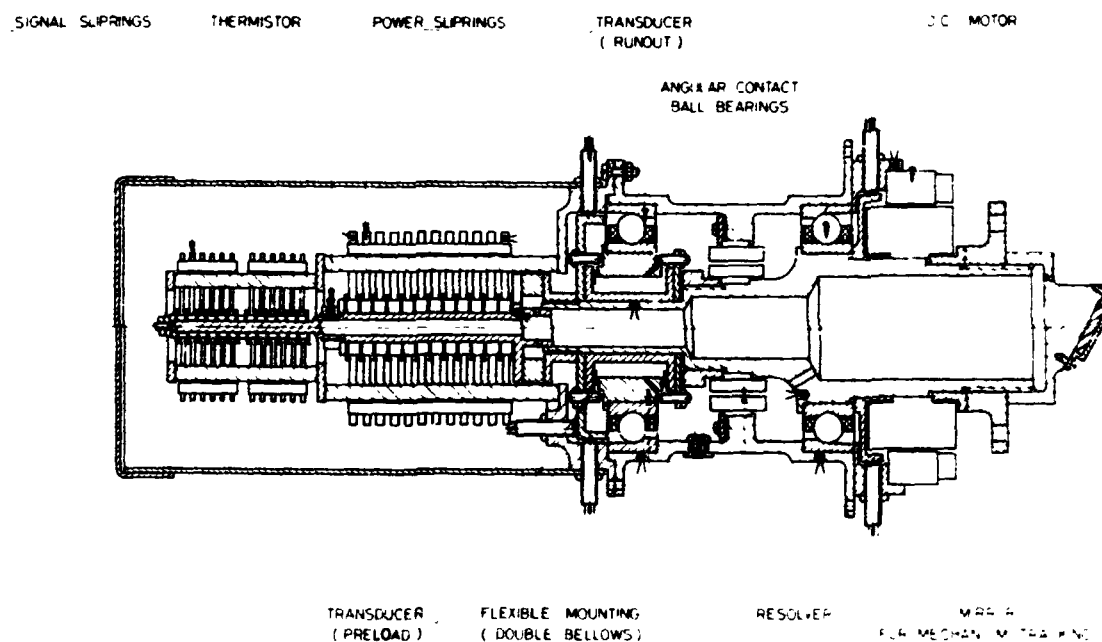
Figure 1.- HSD solar paddle drive.

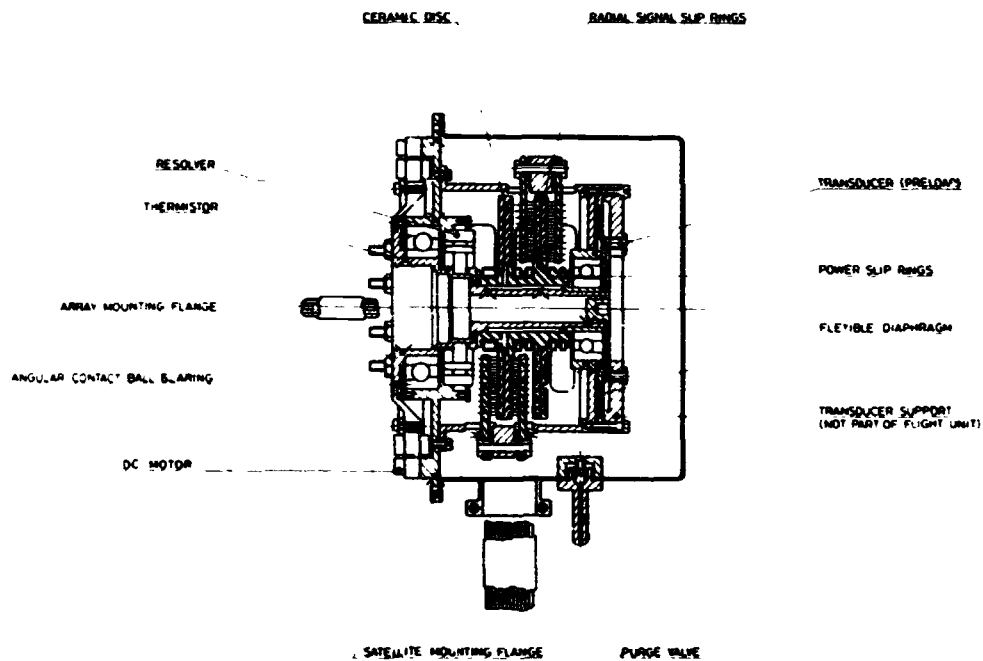**Figure 2.- MSDS solar paddle drive number 1.**

CERAMIC DISC

RADIAL SIGNAL SLIP RINGS

RESOLVER

THERMISTOR

ARRAY MOUNTING FLANGE

ANGULAR CONTACT BALL BEARING

DC MOTOR

TRANSDUCER (PRELOAD)

POWER SLIP RINGS

FLEXIBLE DIAPHRAGM

TRANSDUCER SUPPORT
(NOT PART OF FLIGHT UNIT)

SATELLITE MOUNTING FLANGE

PURGE VALVE

Figure 3.- MSDS solar paddle drive number 2.

RUNNING TIME
IN VACUUM
(YEARS)

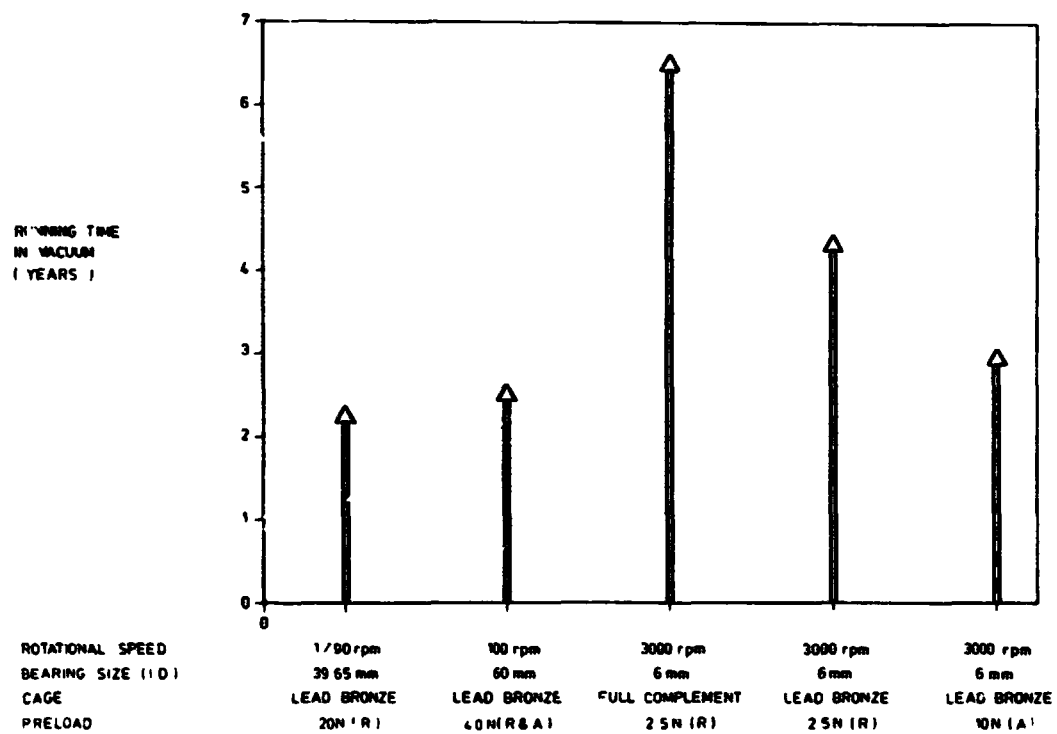| ROTATIONAL SPEED	1 / 90 rpm	100 rpm	3000 rpm	3000 rpm	3000 rpm
BEARING SIZE (I D)	39 65 mm	60 mm	6 mm	6 mm	6 mm
CAGE	LEAD BRONZE	LEAD BRONZE	FULL COMPLEMENT	LEAD BRONZE	LEAD BRONZE
PRELOAD	20N (R)	40N(R & A)	2 5N (R)	2 5N (R)	10N (A)

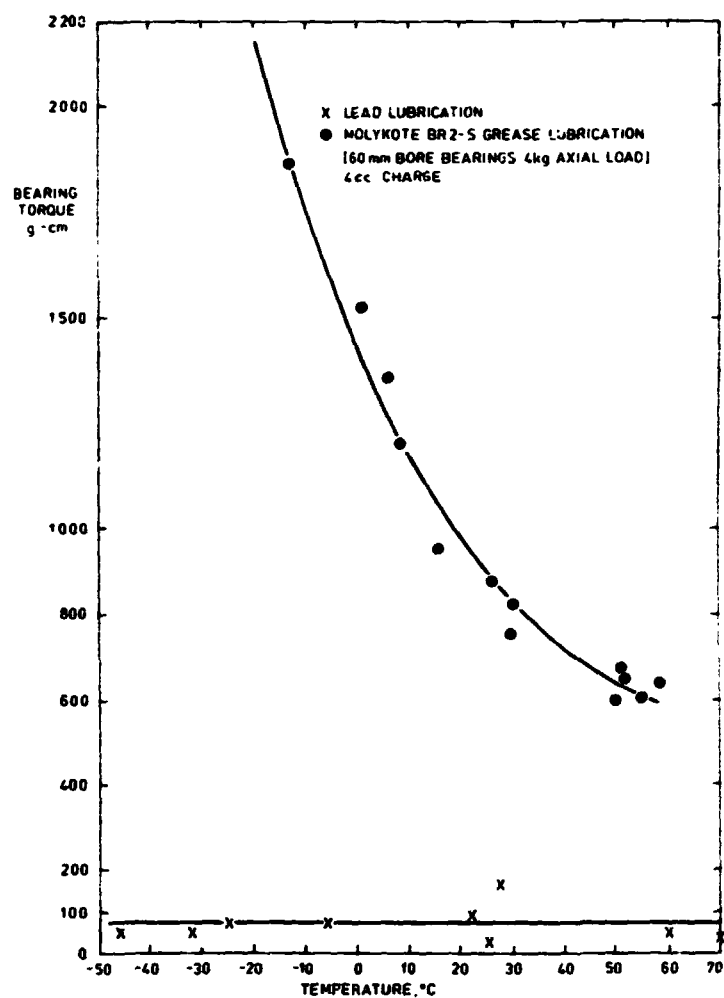Figure 4.- Lifetimes achieved by lead lubricated bearings (all tests continuing)

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Figure 5.- Low-speed torque comparison for dry and liquid
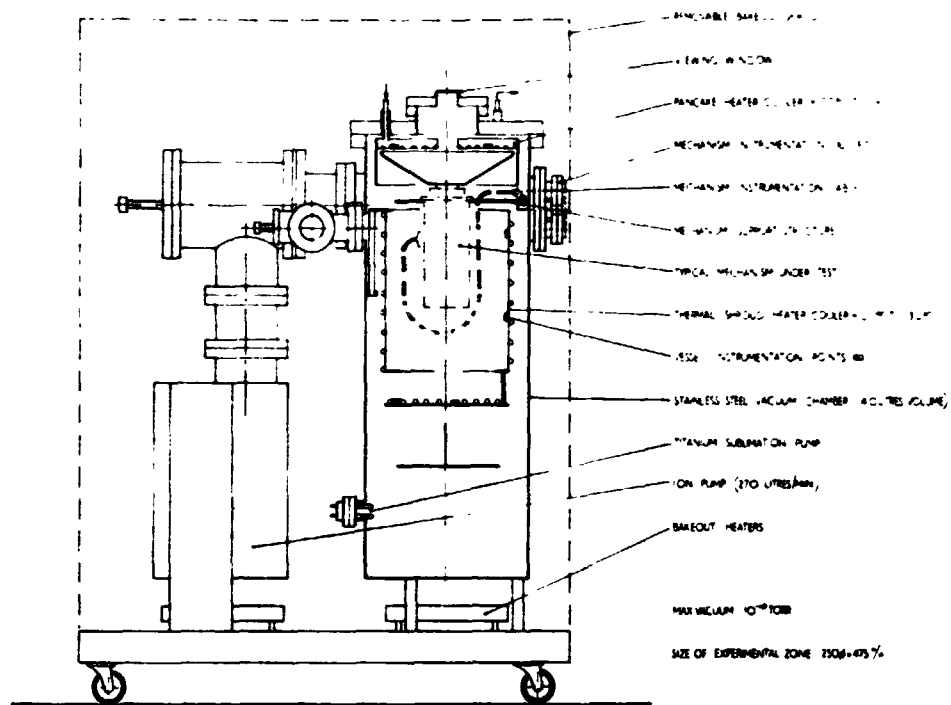lubricated bearings at extremes of temperature.

Figure 6.- Installation of typical mechanism in vacuum test chamber.