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# THE PROPERTIES OF THIN-SECTION, FOUR-POINT-CONTACT BALL BEARINGS IN SPACE

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## ABSTRACT

Thin-section, four-point-contact ball bearings are increasingly employed in spacecraft mechanisms because of the potential advantages they offer. However little was previously known of their torque, thermal conductance and stiffness properties at conditions anticipated for their use in space. This paper describes an investigation of these properties.

It has been found that frictional (Coulomb) torque, thermal conductance and stiffness all show marked dependence on the bearing preload, the housing design, the bearing external fit (i.e., free fit or interference) and on the thermal gradient across the races. Optimum bearing performance is achieved only if these properties are well understood. This paper sets out to provide the necessary data.

## INTRODUCTION

The ideal of an ideal bearing for any application is usually only a designer's dream. Thin-section, four-point-contact, rolling-element, bearings have been shown to offer a near-ideal solution for some of the novel demands of spacecraft mechanisms. Since one bearing does the work of two, the prime advantages of low weight, small cross section and high stiffness can be utilised to simplify design. With the recognition of these unique properties, thin-section, four-point-contact ball bearings are now employed in many satellite systems and in Europe these include the Olympus and Eurostar series of communication satellites. Brunnen & Bentall (Ref. 1) describe the novel swash plate design of an antenna pointing mechanism (APM) utilizing four-point-contact ball bearings.

To ensure the success of such ball bearings, a knowledge of the bearing properties in space is required. Of major relevance to mechanism designers is bearing stiffness, frictional (Coulomb) torque and thermal conductance. Experimental and theoretical studies have been performed at the European Space Tribology Laboratory (ESTL) to provide this data (Refs. 2-5). A range of parameters has been examined including the influence of load, preload, fit, thermal gradient, lubricant, and cage type. Studies have concentrated on bearings installed with external diametral clearance and held in-situ with clamp rings, since this permits easy assembly and disassembly in experimental work and also proves to be more convenient than interference fitting in practice.

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## BEARING SPECIFICATIONS

Details of the bearings used in these tests are given in Table 1. Bearing size and range of internal diametral preloads (designated A, K or M) were chosen to be similar to those used in the Olympus APM (Ref. 1). Diametral preload is defined as the sum of the interferences in the radial direction between ball and races (i.e., the sum of the normal Hertzian approach between ball and races). In this type of bearing, the groove in each race has two radii whose centres are offset by equal amounts from the plane of the ball centres. This construction gives the bearing a "Gothic Arch" configuration making possible four-point contacts between a ball and the raceways.

Table 1 Bearing details.

| | |
|---|---|
| Kaydon Bearing No. | KA070XP2 |
| External diameter | 190.5mm |
| Internal diameter | 177.8mm |
| Width | 6.35mm |
| Bearing internal fits (nominal) | 0-12.7 m clearance (Brg. A) |
| " " " " | 0-12.7 m preload (Brg. K) |
| " " " " | 12.7-25.4 m preload (Brg. M) |
| No. of balls | 87 |
| Cage | Snap over crown in brass, nylon or spring separators |
| Contact angle | 30 degrees |
| Bearing steel | AISI 52100 |
| Precision | Class 6 (equivalent to ABEC 7 in run-out tolerances) |

## MEASUREMENT OF STIFFNESS AT LOW LOADS

### Stiffness Rig

The ESTL stiffness rig is illustrated in cross-section in Figure 1. The bearing was located against a machined shoulder on both the inner and outer housings and was held in place by clamp rings. The axial clamping force on the bearing could be varied by the torque imparted to the 24 X M3 clamping bolts on each clamp ring. Figure 2 shows, in more detail, the configuration of the clamp rings in relation to the bearing and housings. Bearing deflections were measured by two, diametrically opposite, contacting displacement transducer probes which were rigidly clamped via a support arm to the outer housing. Bending moment was applied using the pulley and weight system attached to the central rod. Axial load was varied by the addition of weights to the inner housing.

### Structural Deflections in the Rig

From the limited information available, very small axial and moment deflections were to be expected. To ensure any additional deflections did not arise from the stiffness rig itself (despite its massive structure), a solid stainless steel ring of nominally "infinite" stiffness was manufactured and substituted for the bearing. Only minor deflections were measured arising from the rig; where necessary the experimental results have been corrected.

## BEARING STIFFNESS RESULTS

### Effect of bearing Internal Fit on Moment Compliance (constant axial clamp)

The influence of bending moment on the moment deflections of the three bearings, with differing internal fits (diametral preload), is shown in Figure 3, at a clamp ring bolt torque of 0.8 Nm. This, arbitrarily chosen, bolt torque is equivalent (assuming dry bolt threads and standard cap head bolts) to an axial clamping force per bearing race of approximately 48 N/mm. The bearing movement is such that a moment load in the direction of probe A (see Figure 1) produces a negative deflection at that probe and a positive deflection at probe B. Moment loading towards B produces the reverse deflections. For clarity the results obtained from probe A have been omitted from Figure 3.

The effect of increasing the bearing internal preload is, as expected, to decrease the moment deflection at a given moment load. By imposing higher race deformation (at the four-point contacts) a larger bending moment is required to produce the same moment deflection. The slopes of the lines of Figure 3, defined as the <u>moment compliance</u> with units of Rad/Nm, have been calculated by linear regression and are presented in Table 2. This table is specific to the axial clamping force imparted by a bolt torque of 0.8 Nm. For comparison the moment compliance derived from information in the bearing manufacturer's catalogue (Ref. 10) is included in Table 2. Although the internal preload is not specified in the catalogue, there is good agreement with the values of moment compliance.

Table 2 Moment compliance as a function of original bearing internal fit (clamp ring bolt torque 0.8 Nm).

| Bearing | Moment Compliance (Rad/Nm) | |
|---|---|---|
| | Experimental | Manufacturer's estimate (internal fit not specified) |
| A (0-12.5 m clearance) | $1.62 \times 10^{-6}$ | |
| K (0-12.5 m preload) | $1.27 \times 10^{-6}$ | $2 \times 10^{-6}$ |
| M (12.5-25 m preload) | $0.85 \times 10^{-6}$ | |

## The Effect of Axial Clamping Force on Moment Compliance

The pronounced effect of the bolt clamping torque (which is proportional to the axial clamping force) on the moment compliance of the three bearings, is illustrated in Figure 4. Increasing the axial clamping force reduces the moment compliance of the bearings. The range of moment compliance values from all the bearings studied can be encompassed using bearing A (0-12.5 m clearance) solely by adjusting the clamp ring bolt torque. At higher clamp ring bolt torques ( 2 Nm), it was not possible to differentiate between the moment compliance of the three bearings, as the deflections were so low ( 0.1 m) that experimental error predominated.

The effect of increasing the axial clamping force is to bend the Gothic-Arched-shaped races and impose additional Hertzian deformation at the four-point contacts. Thus moment compliance will be correspondingly reduced. This result demonstrates the sensitivity of this type of bearing to bending of the rings induced by clamping forces.

A notable feature of Figure 3 is that bearing A (0-12.5 m internal clearance, prior to fitting) responds to moment loads in a similar manner to the internally preloaded bearings; i.e., as the moment load was varied there was no evidence of play or clearance in the resulting moment deflections. Only when the clamp ring bolt torque was reduced to 0.2 Nm was hysteresis and non-linearity, due to ring movement and a highly variable ball complement, exhibited in its moment deflections (Figure 5). The implication of this result is that poor locational accuracy and low stiffness can be avoided by judicious use of axial ring clamping (without the need for interference fitting).

## Effect of Bearing Internal Fit on Axial Compliance (constant axial clamp)

Table 3 summarises the effect of the bearing internal fit on the axial compliance of the three bearings studied (at a clamp ring bolt torque of 0.8 Nm). At axial loads of up to 200 N, there was a linear relationship between load and axial deflection; as expected, increasing the bearing preload reduces the bearing axial compliance. Fair correlation is obtained with the axial compliance deduced from the bearing manufacturer's catalogue.

Table 3 Axial compliance as a function of original bearing internal fit (clamp ring bolt torque 0.8 Nm).

| Bearing | Axial Compliance (m/N) | |
|---|---|---|
| | Experimental | Manufacturer's estimate (internal fit not specified) |
| A (0-12.5 m clearance) | $7.25 \times 10^{-9}$ | $8 \times 10^{-9}$ |
| K (0-12.5 m preload) | $5.81 \times 10^{-9}$ | |
| M (12.5-25 m preload) | $4.04 \times 10^{-9}$ | |

## Effect of Axial Clamping Force on Axial Compliance

The marked influence of axial clamping force on the axial compliance of bearing A (0-12.5 m clearance) is summarised in Table 4, for three bolt torque settings (0.2, 0.8, and 1.86 Nm). At a given axial clamping force there was a linear relationship between axial deflection and load.

Table 4 Axial compliance as a function of axial clamping force (clamp ring bolt torque). Bearing A (0-12.5 m clearance).

| Clamp ring bolt torque | Axial Compliance (m/N) |
|---|---|
| 0.2 Nm | $3.0 \times 10^{-8}$ |
| 0.8 Nm | $7.25 \times 10^{-9}$ |
| 1.86 Nm | $1.56 \times 10^{-9}$ |

# MEASUREMENT OF TORQUE AND RADIAL THERMAL CONDUCTANCE

## Heat Transfer Rig

Figure 6 is a schematic drawing of the rig. Its design was based on an existing bearing heat transfer rig at ESTL (manufactured by the Dutch National Lucht en Ruimtervaart Laboratorium (Ref. 6)). The bearing was mounted between the inner housing and heat flux meter (HFM), locating against shoulders thereon. The bearing housings and seatings were manufactured to ABEC 9 standard. The bearing was held in place by inner and outer clamp rings, whose clamping force could be varied by the torque applied to the M3 clamp ring bolts (24 per clamp ring). The configuration of the clamp rings in relation to the bearing, housing and HFM is as shown in Figure 2. Four radial groups of copper/constantan thermocouples were located at 90-degree intervals around the HFM. Per grouping, two thermocouples measured the inner ($T_I$) and outer ($T_O$) bearing race temperatures and two thermocouples measured the temperature difference across the HFM ($T_I'$ and $T_O'$). The thermocouples measuring $T_I$ and $T_O$ were positioned such that they were in contact with the surfaces of the bearing races. Under isothermal conditions and with a stationary bearing in vacuum, the thermocouples all showed the same reading on a digital voltmeter (DVM) to within 1 V (1/40°C). Radiant heat loss from the bearing and HFM was prevented by multiple superinsulation shields. The inner housing, HFM and clamp rings were manufactured from FV 520B, which has the same coefficient of thermal expansion as AISI 52100, thus ensuring tre isoexpansive nature of the rig with respect to the bearing.

The bearing was rotated using an externally controlled motor via a ferrofluidic vacuum feedthrough. Bearing torque was sensed by a strain gauge torque transducer, the output of which was displayed on a potentiometric pen recorder and on a programmable digital voltmeter.

## Basis of Method

Heat is made to flow radially across the HFM whose conductance, K, is known. The same heat flows across the test bearing in "series" with the HFM. The inner ring of the bearing and outer circumference of the HFM are held at constant (adjustable) temperatures by thermostatically controlled fluids. Thus, from the average temperatures at the four positions, the measured radial temperature difference across the HFM ($T_O'-T_I'$) signifies a known heat flow rate, and a measure of the temperature across the bearing ($T_O-T_I$) yields the bearing conductance, C:

$$C = K \cdot (T_O'-T_I')/(T_O-T_I)$$

## Bearing Test Lubricants

Three space-approved lubricants were used in the torque and conductance tests, two oils and one grease. Their descriptions are given in Table 5.

Table 5 Details of Test Lubricants.

| Lubricant | Manufacturer | Description | Viscosity at 20°C | Thermal Conductivity |
|---|---|---|---|---|
| Fomblin Z25 | Montedison | Fluorinated oil | 240cS | 0.084W/m°C |
| BP110 | British Petroleum | Mineral oil Refined to give low vapour pressure | 350cS | 0.0153W/m°C |
| Braycote 3L-38RP grease | Brayco Oil Company | Based on Z25 with polymer thickening agent(PTFE) | - | 0.084W/m°C |

## BEARING TORQUE AND CONDUCTANCE RESULTS

Unless otherwise indicated, conductance values were measured with the bearing stationary and the torque was measured at a rotational speed of 1 RPM, i.e., slow enough to make any speed-dependent torque effects negligible. Pressure in the vacuum chamber was maintained at $10^{-3}$ torr to ensure no convective heat transfer between bearing and environment.

### Thermal Gradient

Figure 7 illustrates the marked influence of thermal gradient ($T_O$-$T_I$), defined as the temperature dif: ence between the outer and inner bearing race, on the torque and heat t nsfer of bearing A (0-12.5 m clearance). The clamp ring bolts were tightened to 0.8 Nm and this has resulted in preloading the bearing, despite its nominal internal clearance, due to bending of the rings. Conductance is observed to rise linearly with increasing thermal preload (i.e., as $T_I$ is made hotter than $T_O$) whereas the increase in torque is almost a square law relation.

The results shown in Figure 7 imply that, in this particular case, the outer race needs to be about 10°C warmer than the inner race before the induced preload is apparently relieved, i.e., where the torque and conductance reach their minimum values. However, as will be shown later, other factors can influence the precise determination of this point.

### Internal Preload and Devleopment of Internal Clearance

The influence of internal preload on the mean conductance across degreased and unlubricated four-point-contact ball bearings is shown in Figure 8, plotted as a function of thermal gradient. All the bearings have a similar minimum conductance value of 0.08 W/°C; this corresponds to the development of internal clearance in the bearir. The effect of increasing the internal preload is to raise the thermal co uctance at a given thermal preload. It may be noted that at ambient conditions ($T_O$-$T_I$=0°C) all bearings are internally preloaded at a clamp ring bolt torque of 0.8 Nm.

As the bearings were not rotated without the presence of lubricant (because of the risk of damage), torque measurements on the influence of preload were taken with the addition of 1 l of Z25 (Figure 9). Increasing the internal preload causes a corresponding increase in the mean bearing torque. Analysis of Figures 8 and 9 reveals that the frictional torque and "dry" conductance reach a minimum (or steady value) at approximately the same thermal gradient.

### Axial Clamping Force

Figures 10 and 11 summarise the influence of the axial clamping force on the torque and conductance of bearing A (0-12.5 m clearance). Increasing the axial force and imposing additional Hertzian deformation at the four-point contacts would be expected to raise bearing conductance and torque, a result which is demonstrated in Figures 10 and 11. As in the compliance studies, the effect of increasing the internal preload is to reduce the sensitivity of bearing torque and conductance to the axial clamping force (Ref. 4).

## Fomblin Z25 Oil Addition

The influence of Fomblin Z25 lubricant quantity on the torque and conductance of bearing A (0-12.5 m clearance) is illustrated in Figures 12 and 13. There is no effect of lubricant quantity ( 1 l) on torque at 1 RPM (confirming the absence, under these conditions, of a speed-dependent or viscous torque component). At higher rotational speeds ( 25 RPM) the viscous torque component becomes predominant and the quantity of lubricant can then influence the torque. Thermal conductance increases in a nonlinear manner with oil quantity. From Figure 13 it may be concluded that 200 l of oil is lubricant quantity. Smaller angular contact bearings exhibit a similar relationship with lubrication quantity (Ref. 7).

## Comparison of BP110 Oil With Fomblin Z25

As with Fomblin Z25, the addition OF BP110 lubricant had no influence on bearing torque (at 1 RPM) and gave results identical to those for Fomblin Z25. In previous work (Ref. 8) the boundary friction coefficients of BP110 and Fomblin Z25 oils between steel surfaces have been demonstrated to be similar (0.13 and 0.12, respectively).

Table 6 shows, under isothermal conditions, the relative influence of Fomblin Z25 and BP110 lubrication on bearing conductance (BP110 has a thermal conductivity which is approximately twice that of Fomblin Z25). These values have been normalised to indicate the contribution from the oil only (i.e, minus dry component).

Table 6 Influence of oil type on oil conductance component

| Oil Quantity | Normalised conductance (W/°C) | |
|---|---|---|
| | Z25 | BP110 |
| Dry | 0 | 0 |
| 10 l | 0.3 | 0.35 |
| 30 l | 0.37 | 0.51 |
| 500 l | 0.5 | 0.97 |

Inspection of Table 6 reveals that for "flooded" lubrication (500 1) the conductance with BP110 is indeed close to twice that shown with Fomblin Z25. For small additions, however, the extra conductance with BP110 is small. Stevens and Todd (Ref. 7) have shown with smaller angular contact bearings that at low lubricant additions bearing conductance is proportional to the Hertzian contact area, while at large lubricant additions the majority of the heat flows through the oil meniscus at the ball/race contact and thus conductance is related to the oil conductivity. The present results with thin-section bearings are in close agreement.

Grease Lubrication

Figure 14 illustrates the effect of grease lubrication on bearing A (0-12.5 m clearance). The bearing was approximately one-third filled with grease and then rotated for several revolutions to ensure an even distribution. As Braycote 3L-38RP grease is based on Fomblin Z25 oil, bearing torque and conductance may be expected to correspond to the flooded Fomblin Z25 oil values. In fact slightly higher torque and conductance values were obtained for grease lubrication. The small difference in torque is attributable to grease churning losses and the action of the polymer thickening agent, while the increased conductance probably arises from the grease between cage and races, i.e, bridging the conductance path of the ball/race contacts.

Cage Type

The effect of the following cages types was studied:

(i) Standard brass formed ring, "snap-over" type
(ii) Nylon segmented "snap-over" type
(iii) Helical coil springs

Only minor difference in bearing torque were found (Figure 15), where spring separators tended to give higher torques when the bearing was subjected to negative temperature gradients. Whilst the insensitivity of mean torque to cage type implies that, at a rotational speed of 1 RPM no significant cage "hang up" occurs, even with the solid (loose pocketed) brass cage, it should be noted that the precision of the bearing and housings is also crucial. A lower precision of bearing or housing could tend to promote cage "hang-up". Bearing conductance was found to be independent of cage type.

## DISCUSSION

It is apparent from the results that the stiffness, radial conductance and frictional (Coulomb) torque of large-diameter, thin-section, four-point-contact ball bearings involve the interaction of several variables and as such a rigorous theoretical analysis is not feasible. However, the relationships between stiffness, torque and conductance with internal preload and thermal gradient, for preloaded bearings. have been shown to be in broad agreement with those predicted with reasonable assumptions from theory (Ref. 9). The chief uncertainty in the theory is in the number of rolling elements effectively loaded. Clearly for such large bearings with thin, deformable rings the effective ball complement will vary greatly with the load applied and with the probability distribution of the ball to race interference. In the absence of such data, theoretical predictions must be rather imprecise.

An important practical consideration for those bearings is in the manner of the installation employed. Provided that the "unfitted" internal preload is accurately known and that one has a precise knowledge of the amount of interference being applied, then it can be argued that interference fitting is desirable (assuming also that a later dismantling is not contemplated). However, in the case of the test bearings, the wide tolerance bands of internal diametral fit (Ref. 10) mean that there would be considerable uncertainty in the start point and, therefore, in the ball load and dry torque after interference fitting. For example, the effect of a 10- m diametral interference on bearing fitting was found to increase the mean torque by a factor of 4. The characteristics of thin-section bearings also change, the bearing becoming more sensitive to thermal gradients, etc. Axial clamping of four-point-contact bearing rings, which are clearance fits, has a similar effect to fitting with interference, but it is possible then to vary the ball load and torque to match the requirements more exactly.

For most uses, the optimum mounting method would combine the use of external diametral clearance for each of bearing fitting (and removal) with allow axial clamp. Such a value of axial clamp would minimise ring bending, but would be sufficient to ensure that the bearing cannot physically move in its housing. Suitable clamp ring design can reduce the dependence of the bearing on the axial clamping force. It is recommended that the bearing torque or deflection be measured after installation, from which the effective internal preload can be inferred.

## CONCLUSIONS

Measurements of the frictional (Coulomb) torque, radial thermal conductance and stiffness of thin-section, four-point-contact, ball bearings have shown the following:

a. Internal preload, housing design and external axial clamping force on the bearing rings all have a strong influence upon torque, conductance and stiffness.

b. The thermal conductance of a dry or marginally lubricated bearing depends only upon the thermal strain and varies linearly with radial temperature difference between the rings. Additionally, conductance is a function of type and quantity of lubricant.
c. The Coulomb torque exhibits an almost square law relation with thermal strain.

## ACKNOWLEDGEMENTS

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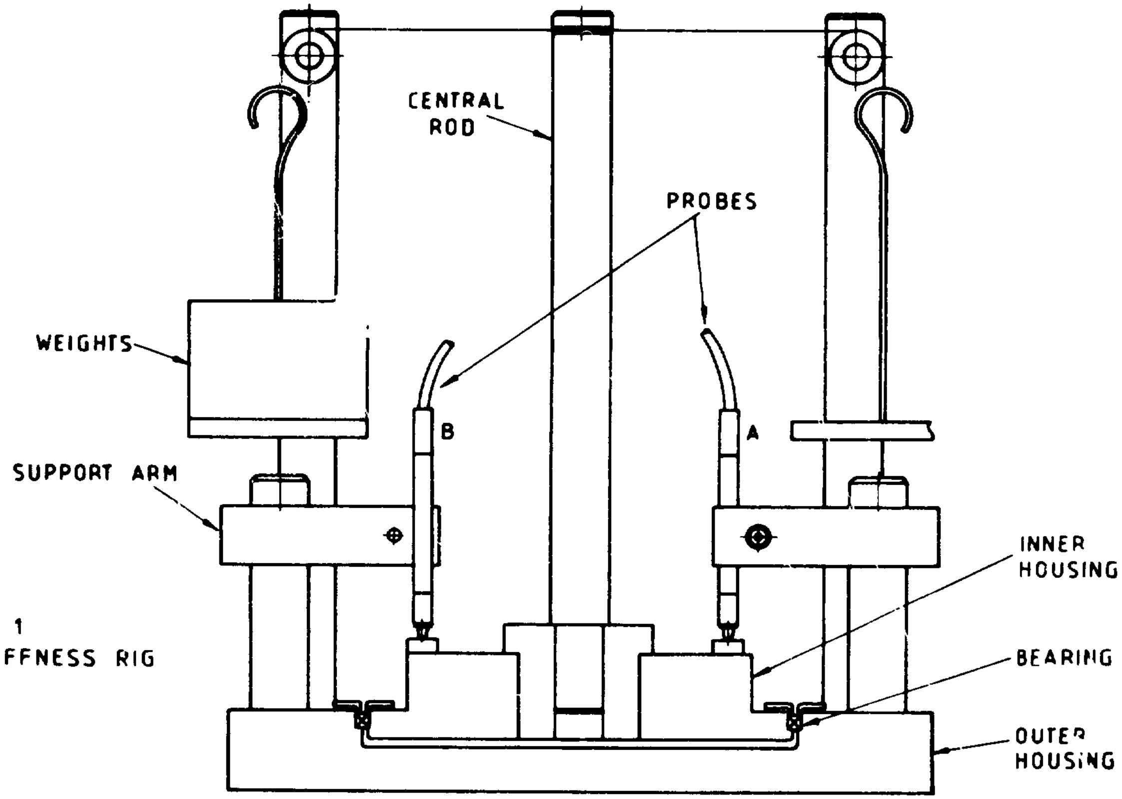


FIG 1
STIFFNESS RIG

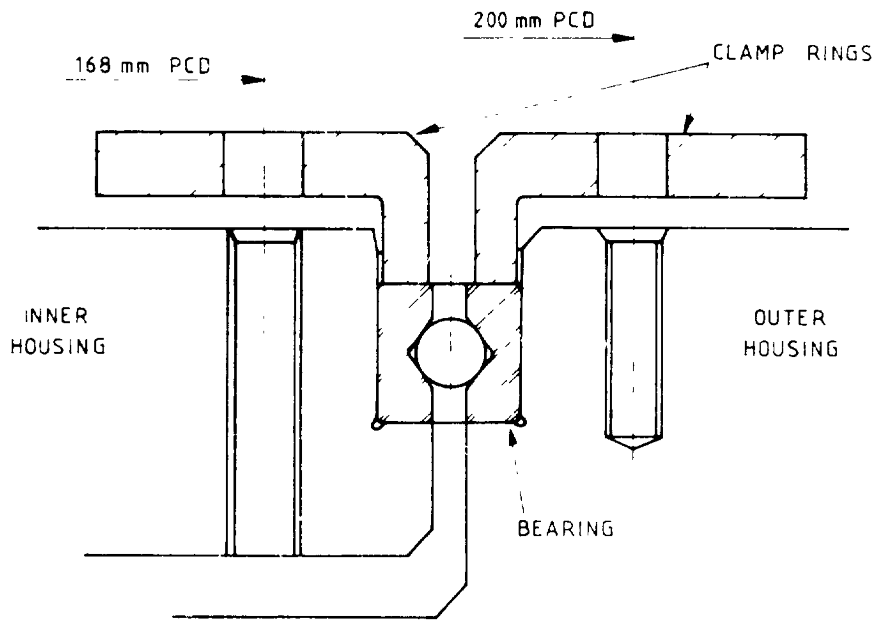


FIG 2 SUPPORT CONFIGURATION

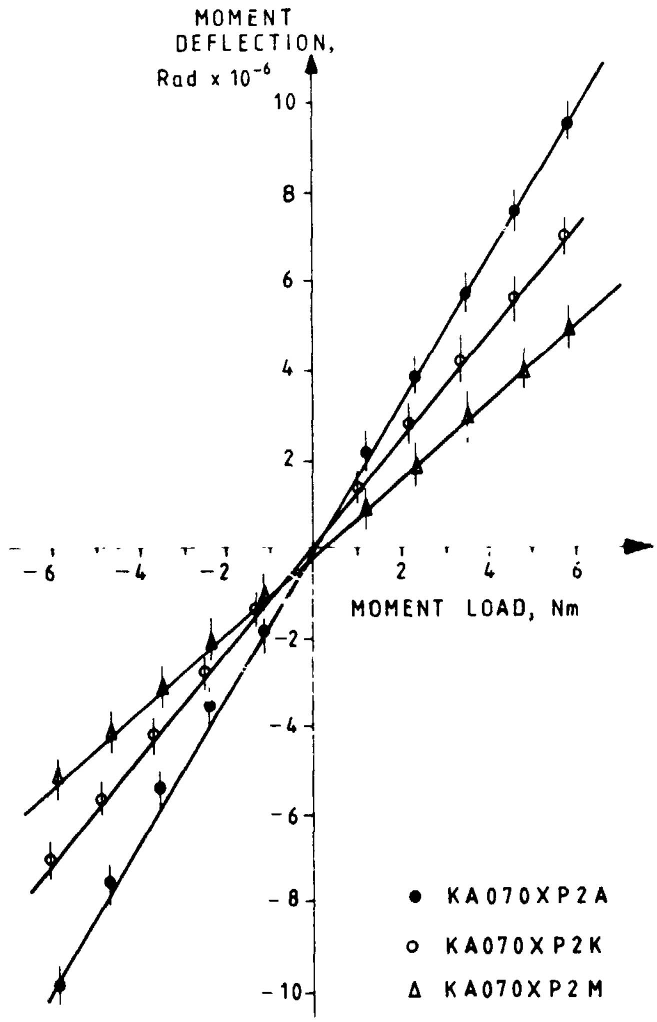


FIG. 3 MOMENT DEFLECTION AS A FUNCTION OF MOMENT LOAD, CLAMP RING BOLT TORQUE 0.8 Nm.

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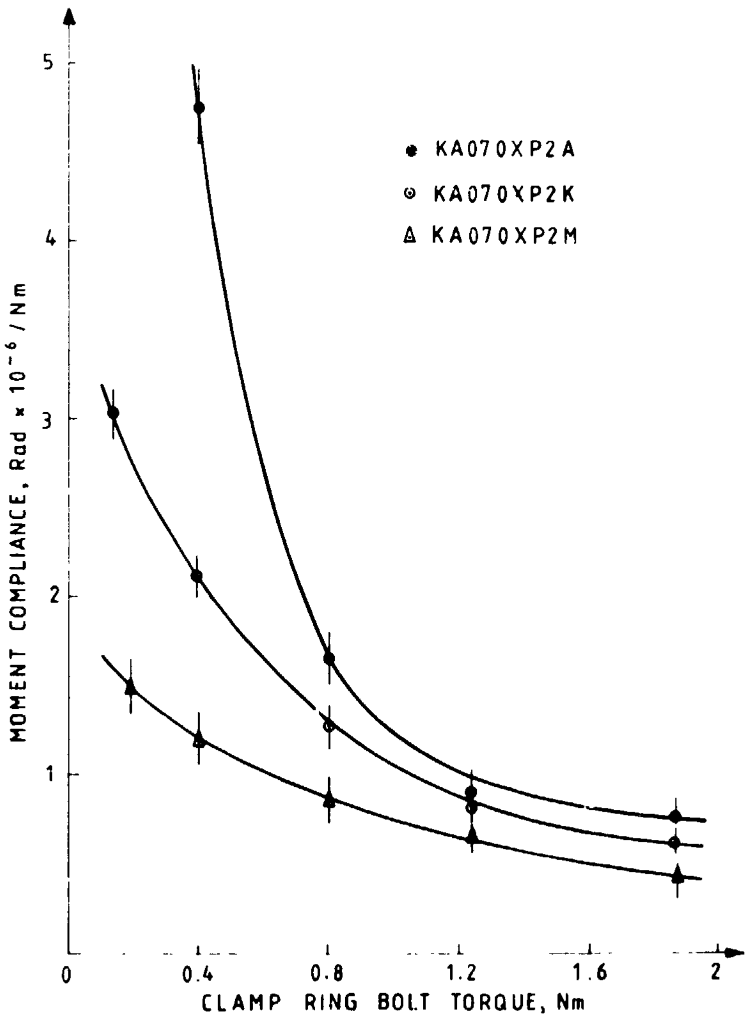


FIG.4 EFFECT OF CLAMP RING BOLT TORQUE ON MOMENT COMPLIANCE OF THREE KAYDON BEARINGS

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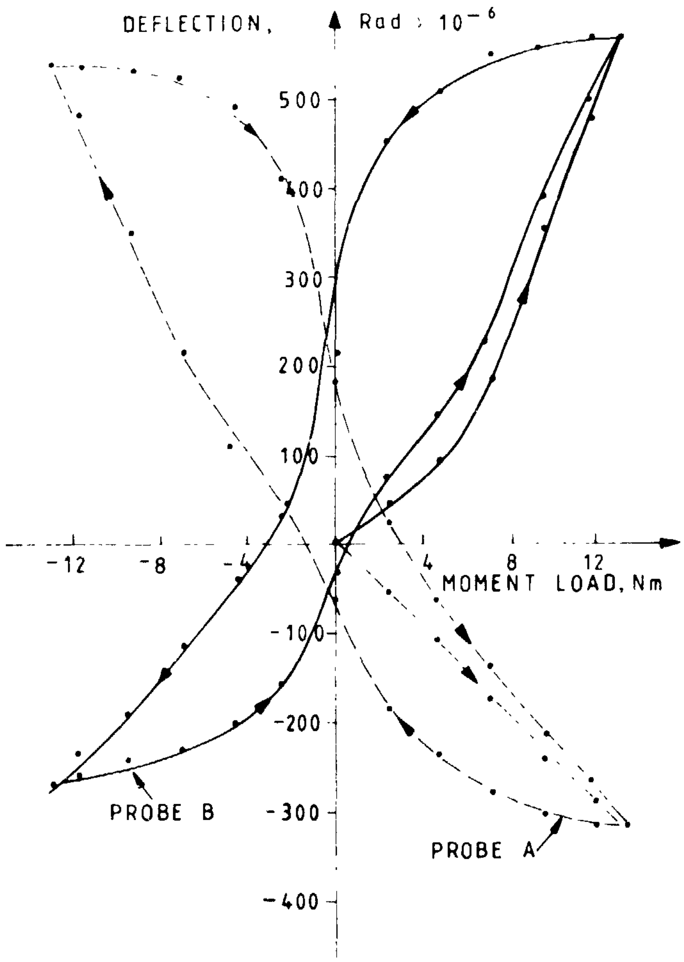


FIG 5 MOMENT DEFLECTION Vs. MOMENT LOAD, BEARING KA070XP2A, 0.2Nm CLAMP RING BOLT TORQUE

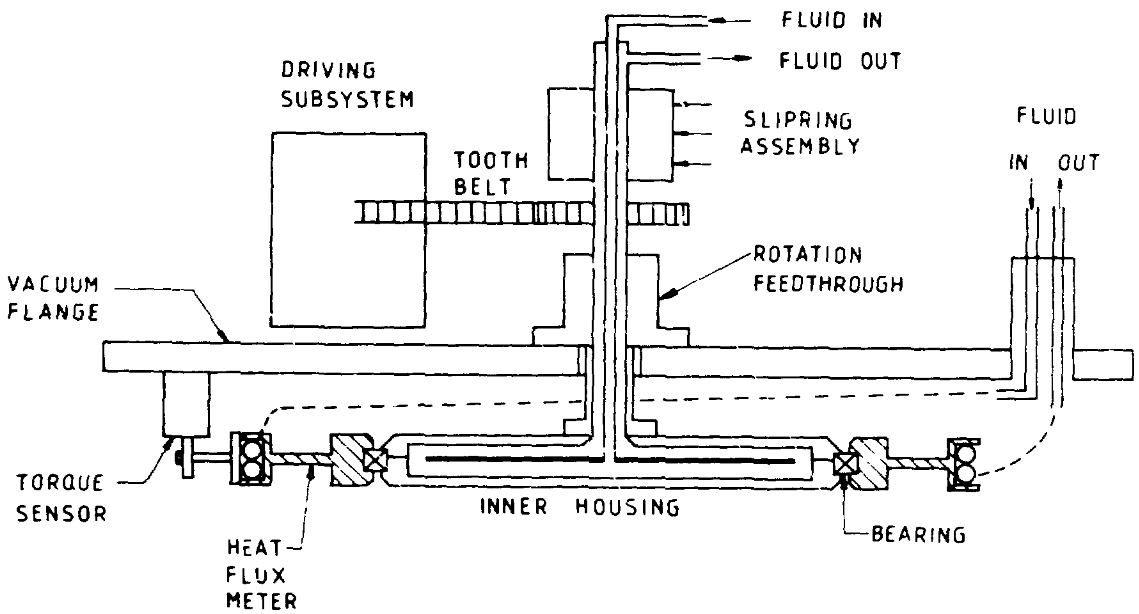


FIG 6 TEST RIG (SCHEMATIC)

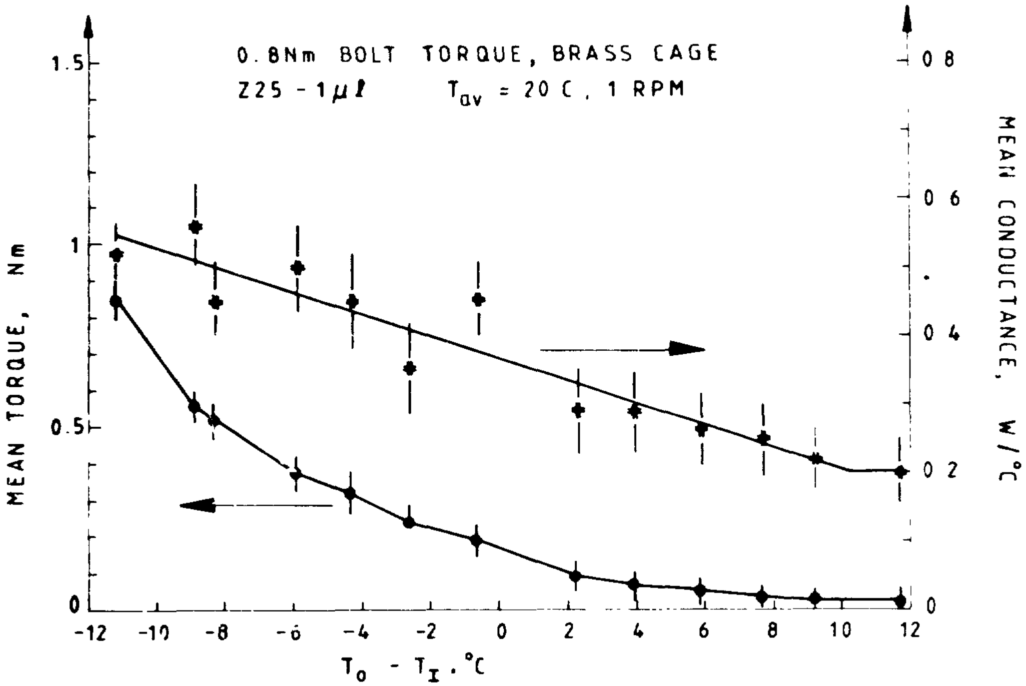


FIG. 7 MEAN BEARING TORQUE & CONDUCTANCE
Vs THERMAL GRADIENT, KA070XP2A

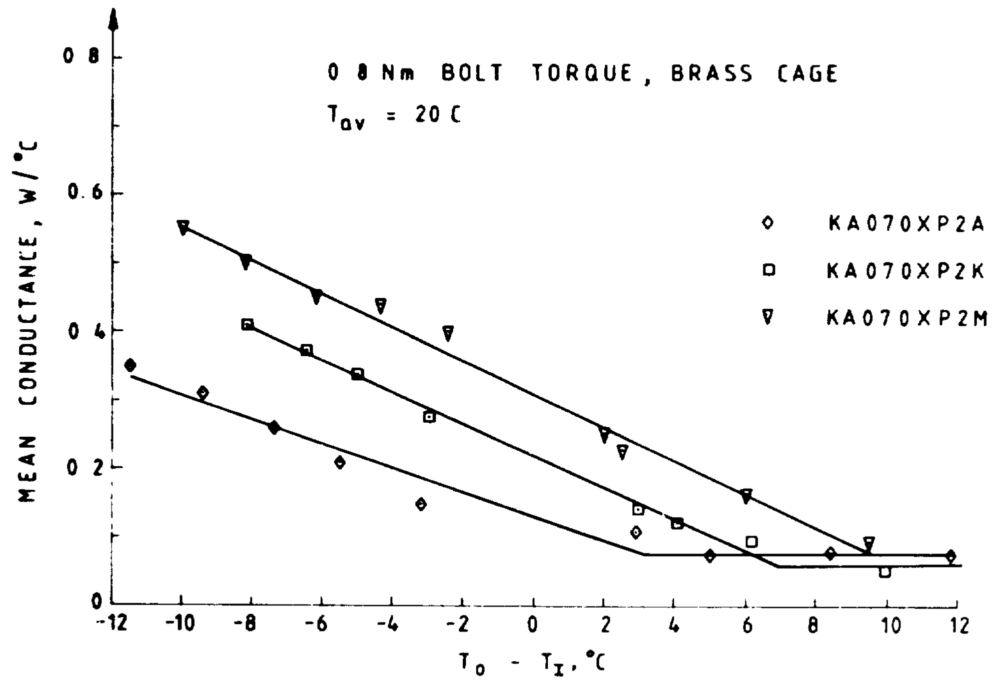


FIG. 8 THERMAL CONDUCTANCE Vs. TEMPERATURE GRADIENT AS A FUNCTION OF BEARING TYPE (UNLUBRICATED)

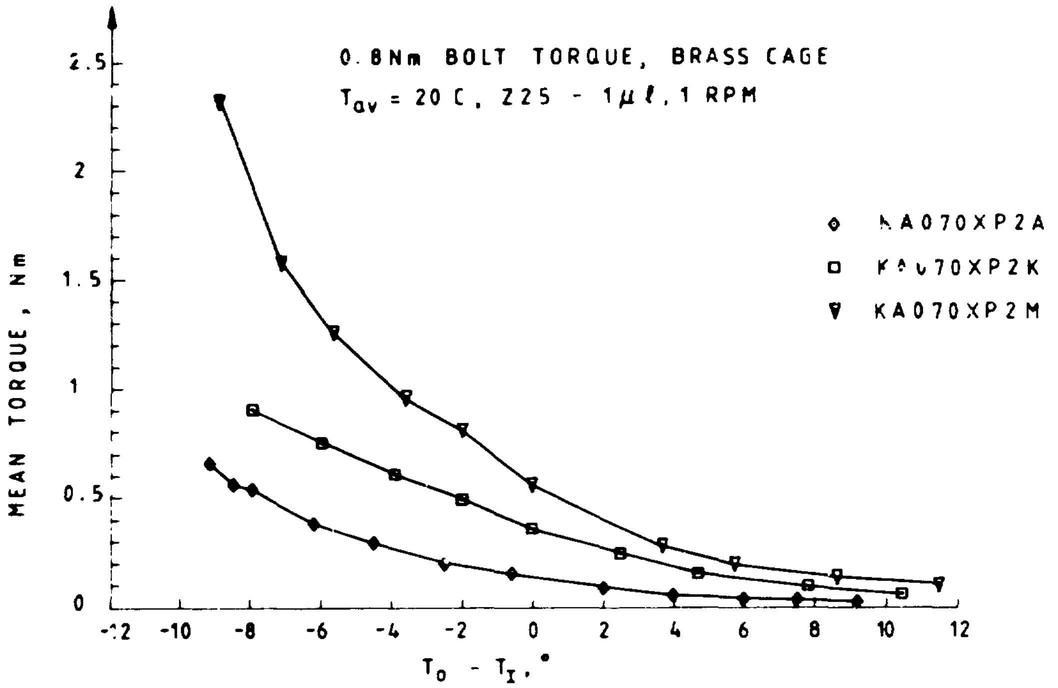


FIG 9 MEAN TORQUE Vs TEMPERATURE GRADIENT AS A FUNCTION OF BEARING TYPE ( LUBRICATED )

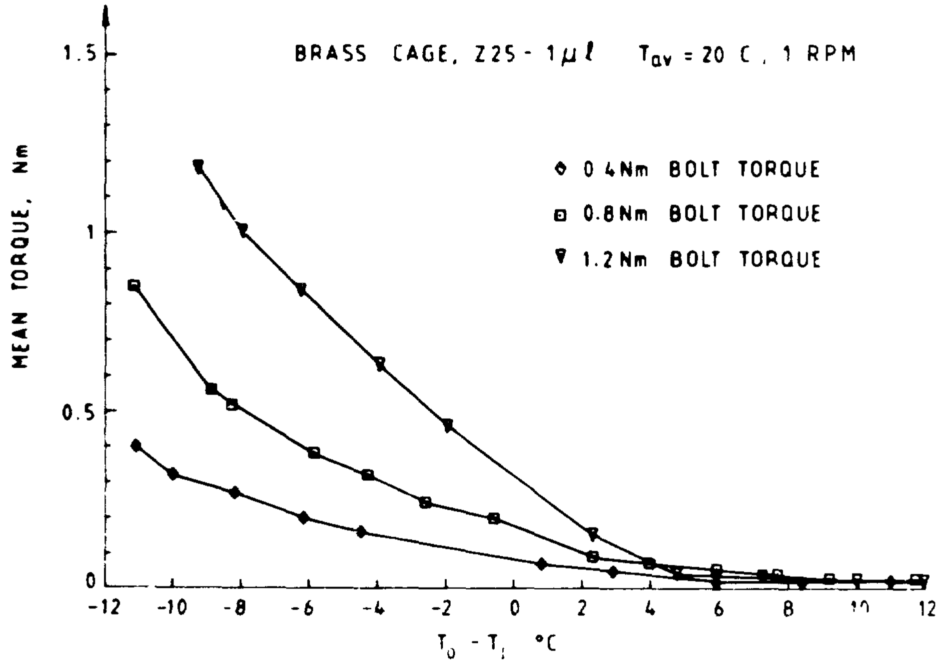


FIG 10 EFFECT OF AXIAL CLAMPING FORCE ON MEAN BEARING TORQUE, KA070XP2A

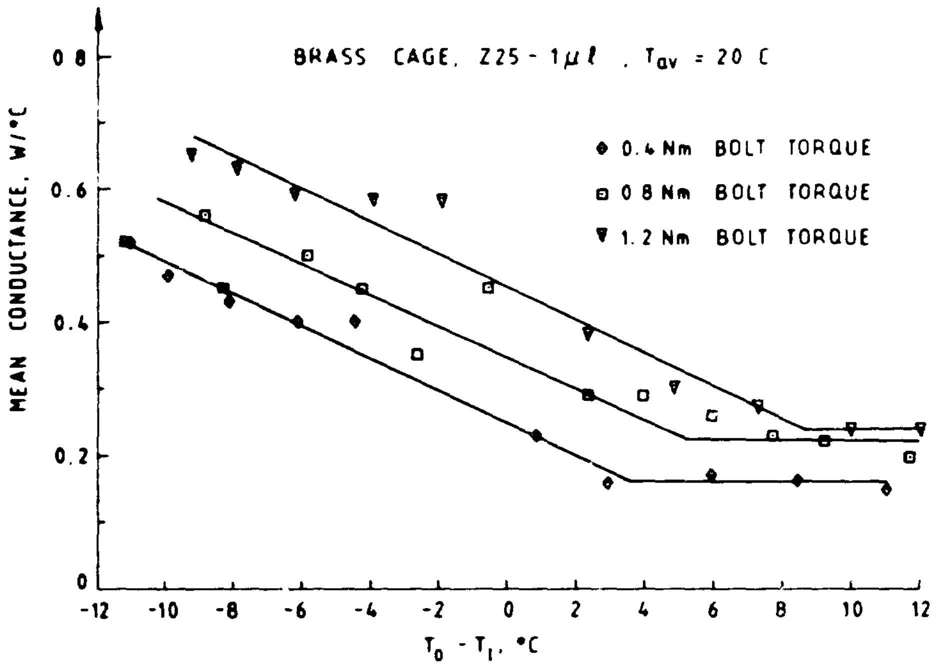


FIG 11 EFFECT OF AXIAL CLAMPING FORCE ON THERMAL CONDUCTANCE, KA070XP2A

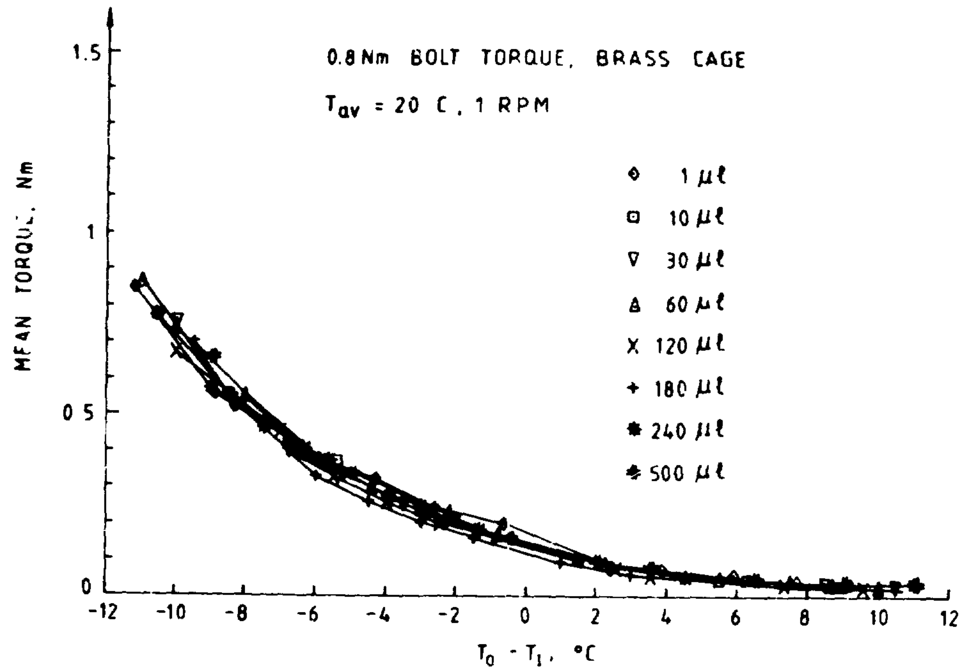


FIG 12 MEAN TORQUE Vs THERMAL GRADIENT AS A FUNCTION OF Z25 OIL QUANTITY, KA070XP2A

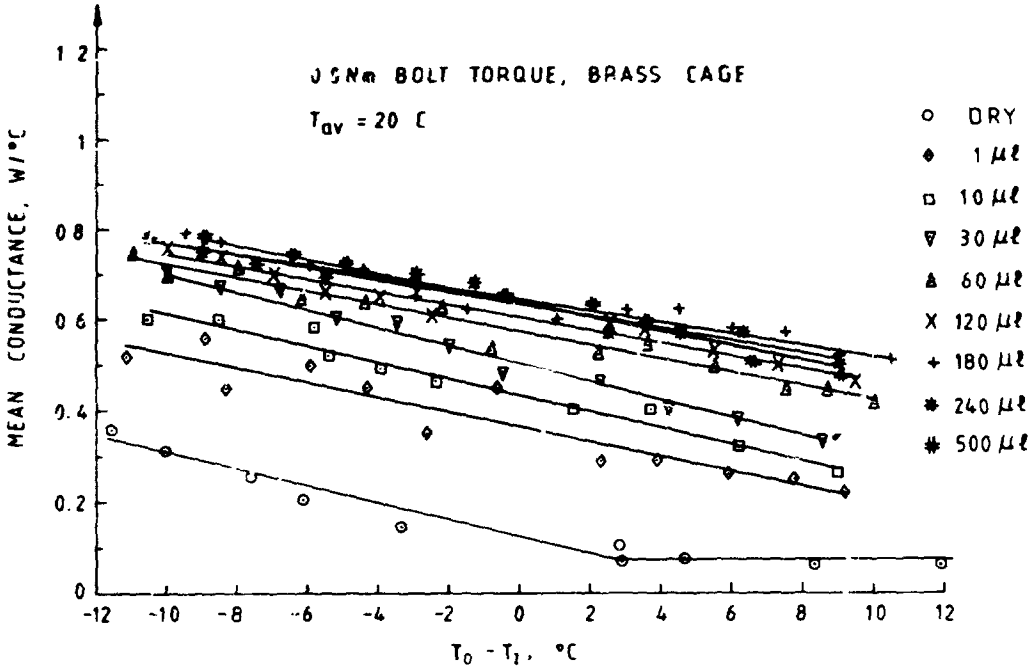


FIG 13 MEAN CONDUCTANCE Vs THERMAL GRADIENT AS A FUNCTION OF Z25 OIL QUANTITY, KA070XP2A

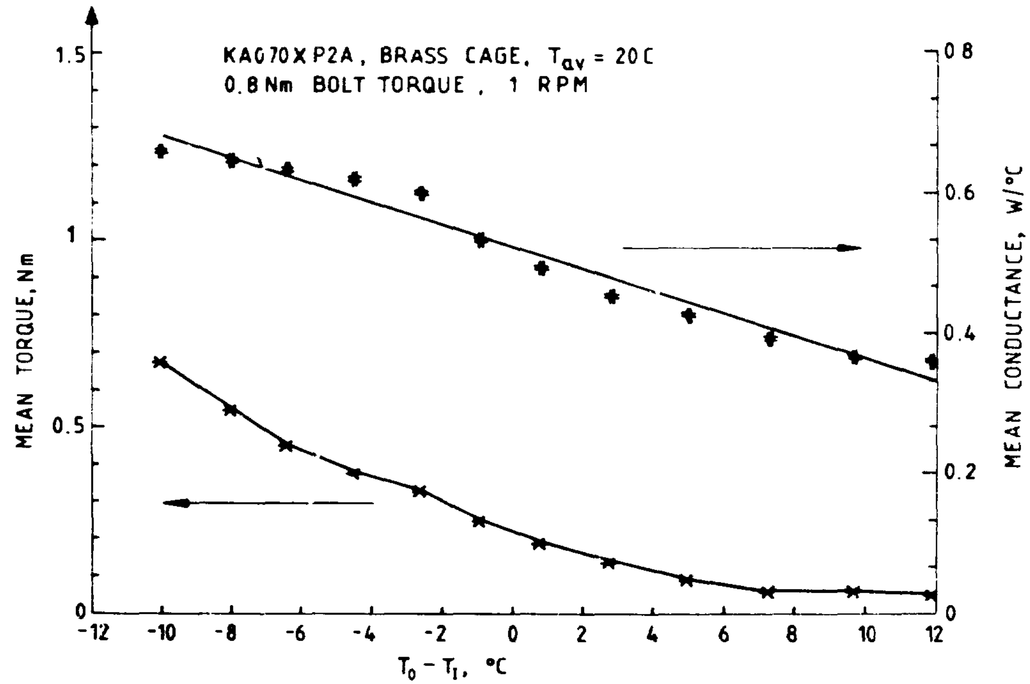


FIG.14 MEAN BEARING TORQUE & CONDUCTANCE Vs. THERMAL GRADIENT, BRAYCOTE 3L-38 RP GREASE

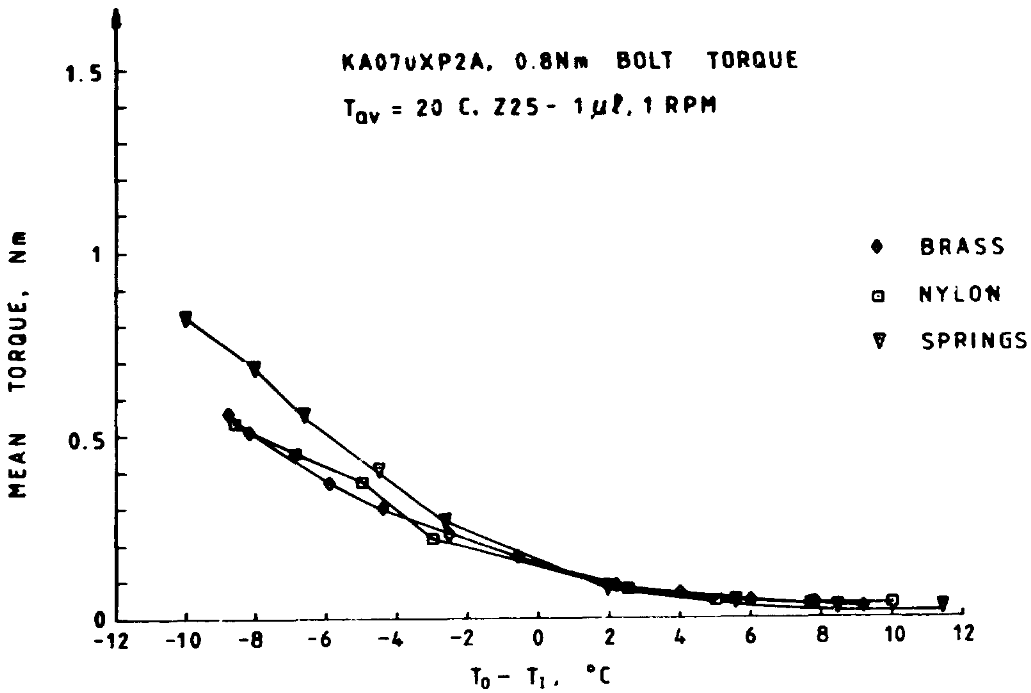


FIG. 15 MEAN BEARING TORQUE Vs. TEMPERATURE GRADIENT AS A FUNCTION OF CAGE TYPE