THE STIFFNESS, TORQUE AND THERMAL CONDUCTANCE OF THIN-SECTION FOUR-POINT CONTACT BALL BEARINGS FOR USE IN SPACECRAFT MECHANISMS

R A Rowntree

European Space Tribology Laboratory, UKAEA, Risley, UK

ABSTRACT

The torque, radial thermal conductance and stiffness of thin-section four-point contact ball bearings have been determined for a number of variables: load, internal preload, axial clamping force, lubricant and cage type.

It was found that the stiffness was principally dependent on the axial clamping force at low values of internal preload and becomes less dependent at higher values.

Bearing conductance and torque have been measured in vacuum over thermal gradients of ±10°C across the races. Both show dependence on internal preload, axial clamping force and lubricant. Conductance is observed to rise linearly with thermal gradient while torque follows an almost square law relationship. The development of internal clearance gives a small constant torque and low conductance. The type of cage was unimportant.

Keywords: Ball Bearings, Torque, Thermal Conductance, Stiffness, Vacuum.

1. INTRODUCTION

Thin-section four-point contact ball bearings offer potential advantages in size and weight reduction for satellite bearing systems.

A bearing is normally considered to have a thin-section when its radial cross-section is less than one quarter of the bore diameter and also less than twice the rolling element diameter. Radial, angular and four-point contact ball bearings are available depending on the duty envisaged. The use of large diameter thin-section bearings allows the employment of hollow shafts further reducing satellite weight. Mechanical and electrical devices can be accommodated (and protected) within such shafts.

Because of these unique properties, thin-section bearings have been selected for use in various satellite mechanisms including the solar array deployment actuators and the antenna pointing mechanisms (APM) for L SAT. Since little was hitherto known about the behaviour of such bearings at conditions anticipated for their use in space, theoretical and experimental studies have been performed at ESTL to investigate their compliance, running torque and radial heat transfer characteristics (Refs. 1-3). Bearing parameters examined include the influence of load (axial and moment), speed, temperature, internal preload, axial ring clamping force, 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 superior to interference fitting in practice.

2. BEARINGS SPECIFICATIONS

Four-point contact ball bearings supplied by the Kaydon Corporation, Muskegon, Michigan were used. A cross-section through a four-point contact ball bearing is shown in Figure 1. The bearing 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 raceways.

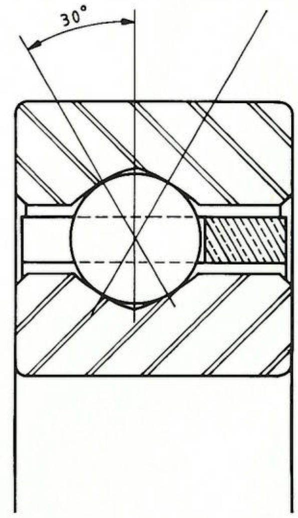

Fig 1 Cross-section through four-point contact ball bearing.

Details of the bearings used in these tests are as follows:

Kaydon Bearing No.	KA070XP2
External diameter	190.5mm
Internal diameter	177.8mm
Width	6.35mm
Bearing internal fits (nominal)	0-12.7μm clearance (Brg. A)
" " " "	0-12.7μm preload (Brg. K)
" " " "	12.7-25.4μm 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)

3. LUBRICANT DETAILS

Three space-approved lubricants were used, two oils and one grease. Their descriptions are given in Table 1.

4. EXPERIMENTAL

4.1 Measurement of stiffness

4.1.1 Bending moment rig. The ESTL bending moment rig is illustrated in cross-section in Figure 2. A dimensional check showed the bearing housings and seatings to be equivalent to ABEC 9 class for conventional bearings. The bearing is mounted on the inner housing and locates against a machined shoulder. The bearing and inner housing are similarly located in the massive outer housing. The bearing is held in place by inner and outer clamp rings, whose clamping force can be varied by the torque imparted to the M3 clamping bolts. There are 24 bolts per clamp ring. Figure 3 shows, in more detail, the geometry of the clamp rings in relation to the bearing and housings.

The bearing deflection is measured by two, diametrically opposite, contacting displacement transducer probes (identified in Figure 2 as A and B) which are rigidly clamped via a support arm to the massive outer housing. Bending moment can be applied in either direction using the pulley and weight system attached to the central rod. Axial load can be varied by the addition of weights to the top of the central rod.

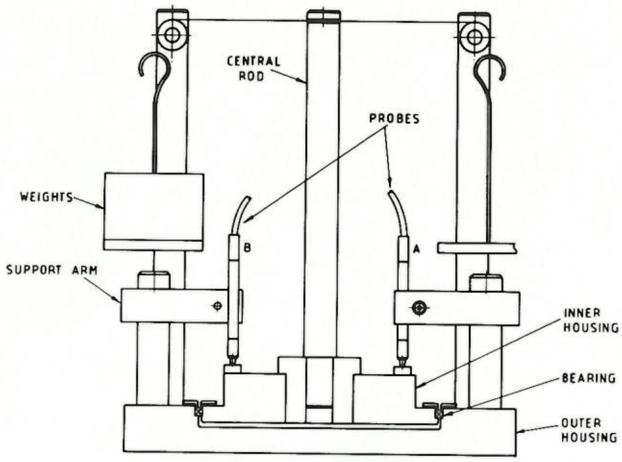

Fig 2 Bending Moment Rig

4.1.2 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 bending moment rig itself (despite its massive structure) a solid stainless steel ring of nominally "infinite" stiffness was manufactured. the dimensions of the solid ring were as follows: 190.35mm (OD), 177.9mm (ID) and 6.35mm (width).

With this ring the structural effects of bending moment, axial load and axial clamping force could be determined. Although some minor structural deflections were measured, the magnitude of these was such that it only became of significance at very small bearing deflections. Where necessary, the experimental results have been corrected.

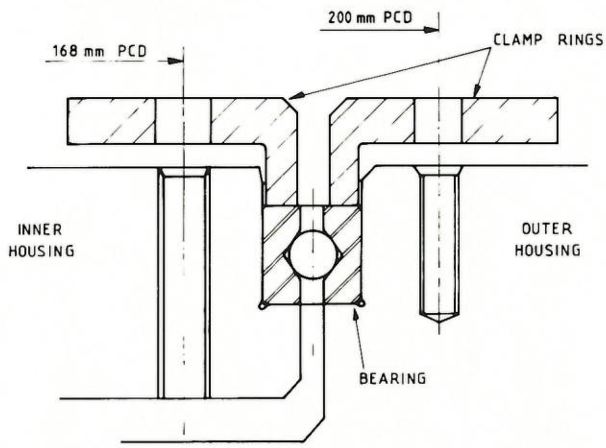

Fig 3 Support Configuration

Table 1 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

4.2 Measurement of torque and radial thermal conductance

4.2.1 Heat transfer rig. The existing radial heat transfer equipment at ESTL (manufactured by the Dutch Nationaal Lucht-en Ruimtevaart Laboratorium under commission by ESA (Ref. 4) was extensively modified to accept the large diameter four-point contact ball bearings.

Figure 4 is a schematic drawing of the rig. The bearing is mounted between the inner housing and heat flux meter (HFM), locating against shoulders thereon. A dimensional check showed the bearing housings and seatings to be equivalent to ABEC 9 class for conventional bearings. The bearing is held in place by inner and outer clamp rings, whose clamping force can be varied by the torque applied to the M3 clamp ring bolts (24 per clamp ring). The geometry of the clamp rings in relation to the bearing, housing and HFM is as shown in Figure 3. Four radial groups of copper/constantan thermocouples are located at 90 degree intervals around the HFM. Per grouping, two thermocouples measure the inner (T_I) and outer (T_O) bearing race temperatures and two thermocouples measure 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 the Solartron digital microprocessor voltmeter (DVM) to within 1μV (1/40°C). Axial radiant heat loss from the bearing/HFM is prevented by multiple super-insulation shields. The inner housing, HFM and clamp rings were all manufactured from FV 520B which has the same coefficient of thermal expansion as AISI 52100, thus ensuring the isoexpansive nature of the rig with respect to the bearing.

Bearing rotation is effected via a Ferrofluidic vacuum feedthrough from an external driving system whose speed can be accurately controlled. Bearing torque is sensed by a calibrated bellows torque transducer, the output of which was displayed on an Oxford pen recorder and on a Solartron DVM.

4.2.2 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:

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

$$\text{thus}\quad C = \frac{K\ (T_O'-T_I')}{(T_O-T_I)} \qquad ..(4.1)$$

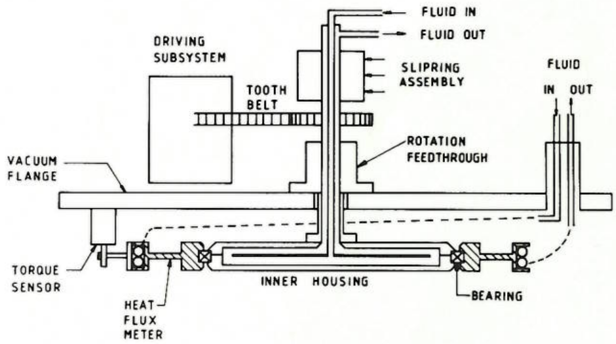

Fig 4 Test Rig (Schematic)

5. BEARING STIFFNESS

5.1 Moment deflections

5.1.1 Effect of bearing internal fit. The influence of bending moment on the moment deflections of the three Kaydon bearings, with differing internal fits (diametral preload) is shown in Figure 5 at a clamp ring bolt torque of 0.8Nm. This, arbitarily chosen, bolt torque is equivalent (assuming dry bolt threads and standard cap head bolts) to an axial clamping force per bearing race of approximately 48N/mm. The bearing movement is such that a moment load in the direction of probe A 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 ommitted from Figure 5.

The effect of increasing the bearing internal preload from A to M in Figure 5 is, as expected, to decrease the moment deflection at a given moment load. By imposing higher race deformation through increasing the internal preload at the four-point contacts, a larger bending moment is required to produce the same moment deflection. The slopes of the lines of Figure 5, defined as the moment compliance with units of Rad/Nm, have been calculated by linear regression and are presented in Table 2. For completeness the moment compliances derived from the measurements at probe A are included. This table is specific to the axial clamping force imparted by a bolt torque of 0.8Nm.

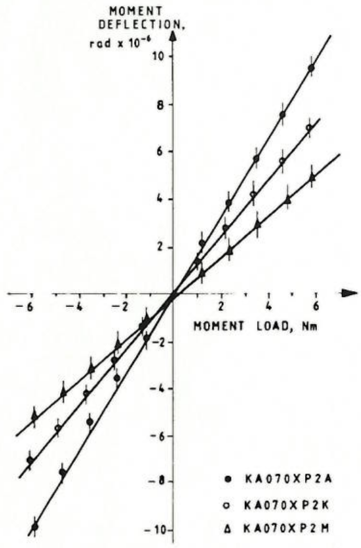

Fig 5 Moment Deflection as a function of Moment Load, Clamp Ring Bolt Torque 0.8Nm.

Table 2 Moment compliance as a function of internal preload

Bearing	Moment Compliance	
	Measured at Probe A Rad/Nm	Measured at Probe B Rad/Nm
KA070XP2A	1.61×10^{-6}	1.62×10^{-6}
KA070XP2K	1.22×10^{-6}	1.29×10^{-6}
KA070XP2M	0.83×10^{-6}	0.86×10^{-6}

5.1.2 The effect of axial clamping force. The pronounced effect of the bolt clamping torque (which is proportional to the axial clamping force) on the moment compliance of the three Kaydon bearings, is illustrated in Figure 6. Increasing the axial clamping force reduces the moment compliance of all bearings. The range of moment compliance values from all the bearings studied can be practically encompassed by the "clearance" bearing, KA070XP2A, alone 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 μm) that experimental error predominated.

The effect of increasing the axial clamping force will be 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 great sensitivity of this type of bearing to bending of the rings induced by clamping forces.

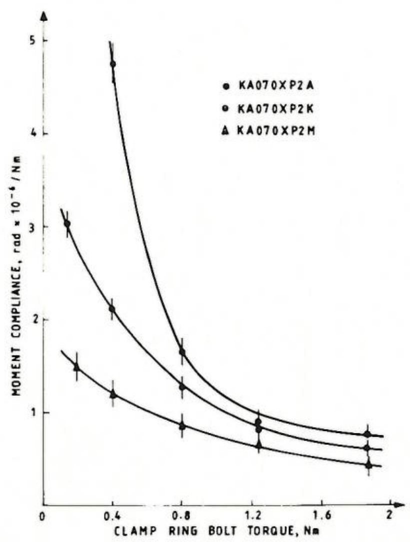

Fig 6 Effect of Clamp Ring Bolt Torque on Moment Compliance of three Kaydon bearings.

A notable feature of Figure 5 is that bearing KA070XP2A (0–12.5μm 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 0.2Nm was hysteresis, due to ball play, exhibited in its moment deflection characteristics (Figure 7). One implication of this result is that the nominal internal fits of this type of bearing, as supplied by the manufacturer, can be significantly altered even by low axial forces.

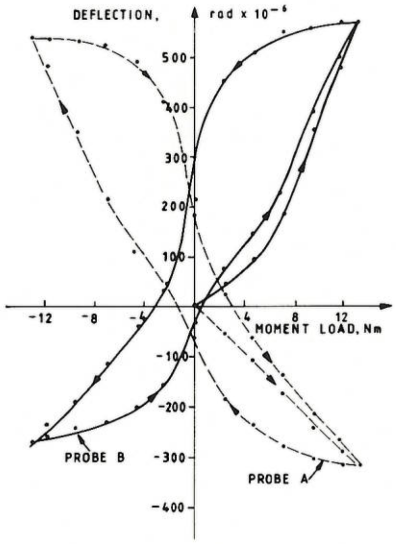

Fig 7. Moment Deflection vs. Moment Load, Bearing KA070XP2A, 0.2Nm Clamp Ring Bolt Torque.

5.2 Axial deflections

5.2.1 Effect of bearing internal fit. The mean axial deflection for the three Kaydon bearings as a function of purely axial load is shown in Figure 8. There is a linear relationship between axial load and deflection; the effect of increasing the bearing preload is to reduce the axial deflection.

5.2.2 Effect of axial clamping force. The marked influence of axial clamping force on the mean axial deflection of bearing KA070XP2A is presented in Figure 9, for three bolt torque settings (0.2, 0.8, and 1.86 Nm). Although no measurements were taken, it is expected that the internally preloaded bearings prior to fitting, K&M) would respond in a similar manner but with lower axial deflections.

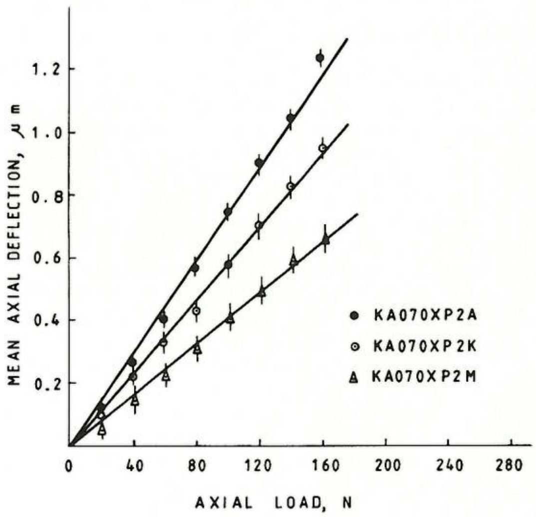

Fig 8. The Mean Axial Deflection of the three Kaydon Bearings as a function of Axial Load. Clamp Ring Bolt Torque 0.8Nm.

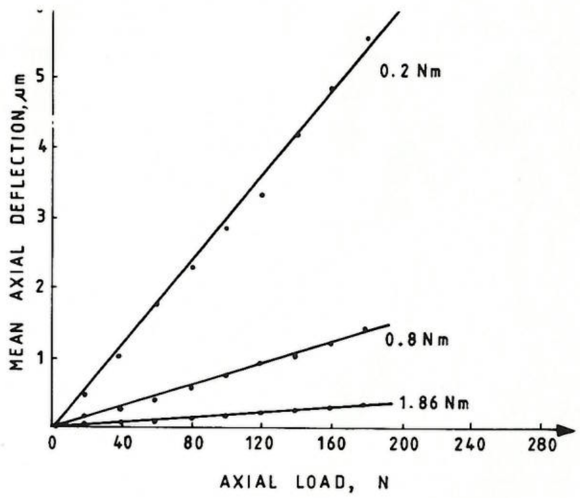

Fig 9 Mean Axial Deflection of Bearing KA070XP2A as a function of Axial Load, for various Clamp Ring Bolt Torque Settings.

6. BEARING TORQUE AND CONDUCTANCE

6.1 General

Due to the number of variables which could directly influence bearing torque and thermal conductance, the effect of various parameters has been, where appropriate, studied against an arbitrarily chosen "standard" set of conditions. That is, a combination of the use of bearing KA070XP2A, the ring clamping bolts tightened to 0.8Nm, a brass cage, lubrication with a trace (1μl) of Z25 and at a mean bearing temperature of 20°C. Unless otherwise indicated, conductance values were measured with the bearing stationary and the torque was measured at a rotational speed of 1RPM, ie slow enough to make the viscous torque effects negligible. Pressure in the vacuum chamber was maintained at $<10^{-3}$ torr to ensure no convective heat transfer between bearing and environment.

6.2 Thermal gradient

Figure 10 illustrates the marked influence of thermal gradient (T_O-T_I), defined as the algebraic temperature difference between the outer and inner bearing race, on the torque and heat transfer of bearing KA070XP2A. The clamp ring bolts were tightened to 0.8Nm 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 (ie 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 10 imply that the outer race needs to be about 10°C warmer than the inner race before the induced preload is apparently relieved, ie 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.

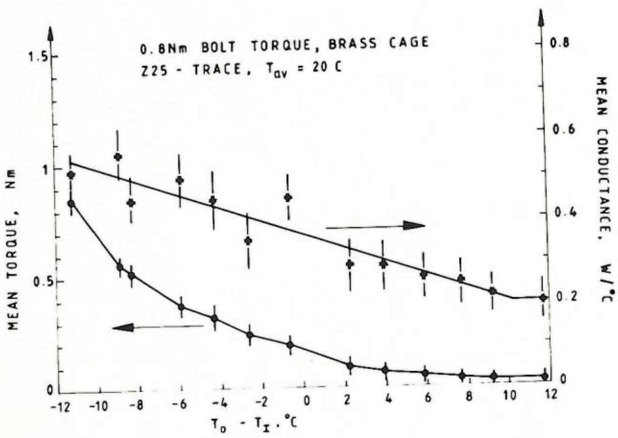

Fig 10. Mean Bearing Torque and Conductance vs. Thermal Gradient, KA070XP2A.

6.3 Internal preload and development of internal clearance

The influence of internal preload on the mean conductance across unlubricated four-point contact ball bearings is given in Figure 11, plotted as a function of thermal gradient. It may be observed that all bearings have a similar minimum conductance value of 0.08W/°C. Raising the internal preload causes the bearing conductance to increase from its base level at a larger value of thermal gradient. This is due to the uptake of any remaining internal diametral clearance as the thermal gradient is reduced from T_O-T_I= 12°C where all three bearings have developed internal clearance, to T_O-T_I=0°C, where all three bearings studied are internally preloaded (under a clamp ring bolt torque of 0.8Nm).

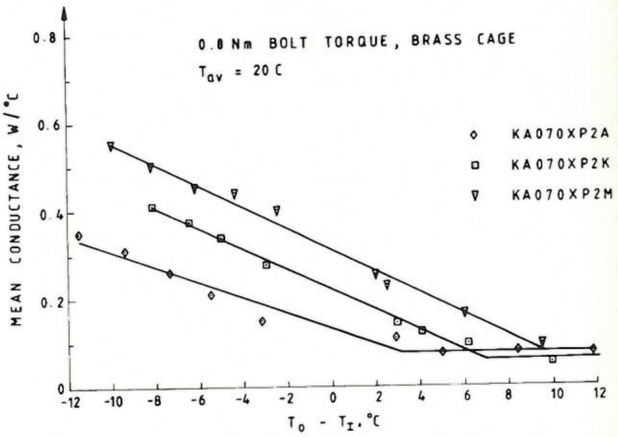

Fig 11. Thermal Conductance vs. Temperature Gradient as a function of Bearing Type (dry).

Bar certain specific tests, the bearings were not rotated without the presence of lubricant (because of the risk of damage) and torque measurements on the influence of preload were taken with addition of 1μl of Z25 (Figure 12). Increasing the internal preload causes a corresponding increase in the mean bearing torque. However, analysis of Figures 11 and 12 reveals that the torque and "dry" conductance do not reach a minimum at the same value of thermal gradient (it is shown later that bearing torque is independent of oil quantity). Table 3 summarises the values of thermal gradient derived from Figures 11 and 12 at which internal clearance, apparently, develops. Clearly, these results are inconsistent with each other since both torque and conductance are directly related to the contact area between ball and race.

Table 3 Values of thermal gradient at which internal clearance develops (caged bearings)

Bearing	Thermal Gradient, T_O-T_I °C	
	Based on minimum conductance (Fig 11)	Based on minimum torque (Fig 12)
KA070XP2A	+3	+9
KA070XP2K	+7	+14 (extrapolation)
KA070XP2M	+10	+17 (extrapolation)

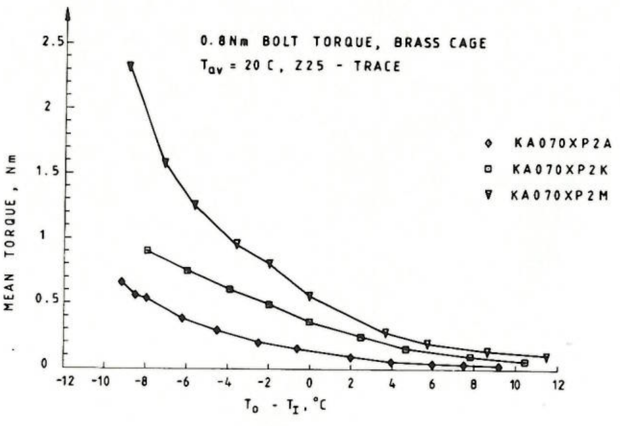

Fig 12. Mean Torque vs Temperature Gradient as a function of Bearing Type (lubricated).

When the conductance has fallen to its base value (ie radiative heat transfer between the races and a small conductive component due to the axial dead load of 20 N) the bearing has "off-loaded" ie developed internal clearance. The residual torque at this point must be due almost entirely to the cage, as calculations of the ball rolling resistance (Coulomb torque) show only a trivial component arising from the axial dead load. The clearance between race and cage is small and, with the races clamped and deformed, there may be some interference. This interference and, therefore, the cage torque, will decrease as $T_O - T_I$ is increased. A test with no cage confirmed that the torque originating from the ball loads does reach a minimum at the same point as the conductance (Figure 13). Additionally, it may be expected that the component of bearing torque, due to the cage, would also increase with thermal preload due to further interference. This may be observed in Figure 13 from the increasing divergence between the torque results for the caged and cageless condition.

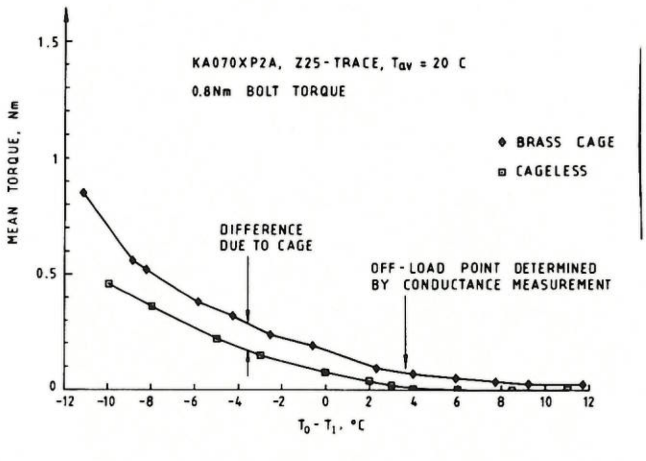

Fig 13. Influence of Cage on Bearing Torque

6.4 Axial clamping force

Figures 14 and 15 summarise the influence of the axial clamping force on the torque and conductance of bearing KA070XP2A. Increasing the axial force and imposing additional Hertzian deformation at the four-point contacts would be expected, therefore, to raise bearing conductance and torque, a result which is clearly demonstrated in Figures 14 and 15. 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 clamp.

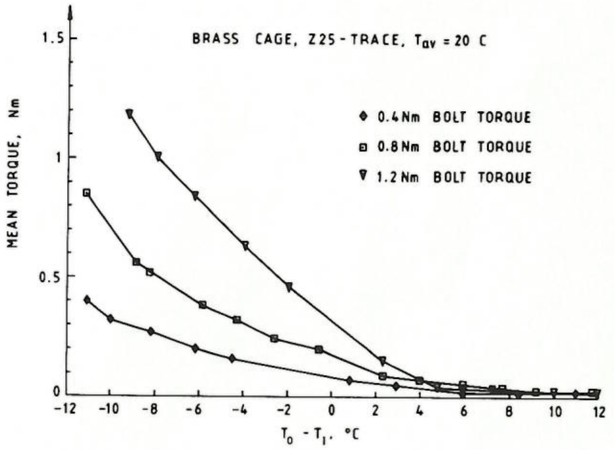

Fig 14. Effect of Axial Clamping Force on Mean Bearing Torque, KA070XP2A.

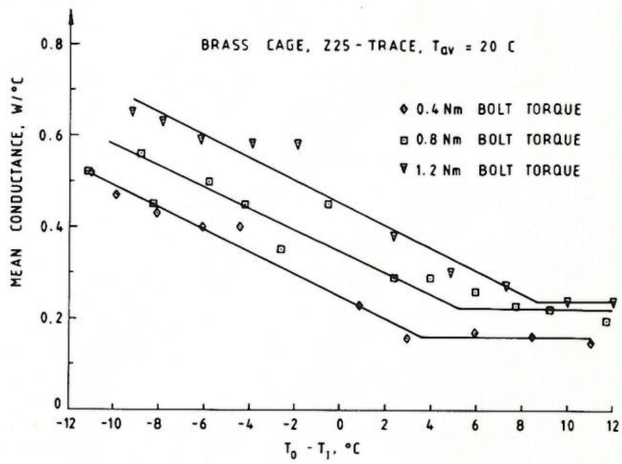

Fig 15. Effect of Axial Clamping Force on Thermal Conductance, KA070XP2A.

6.5 Lubricant type and quantity

6.5.1 Fomblin Z25 oil. The influence of Fomblin Z25 lubricant quantity on the torque and conductance of bearing KA070XP2A is illustrated in Figures 16 and 17. There is no effect of lubricant quantity ($> 1\mu l$) on torque, at 1RPM (confirming the absence, under these conditions, of viscous torque). The consequence of the oil addition is to increase the thermal conductance in a non-linear manner. From Figure 17 it may be concluded that 200μl of oil is required to "flood" the bearing ie when bearing conductance is independent of lubricant quantity. The influence of lubricant quantity is shown more clearly in Figure 18 where mean conductance is plotted as a function of lubricant quantity under isothermal conditions. Smaller angular contact bearings exhibit a similar relationship with lubricant quantity (Ref 5).

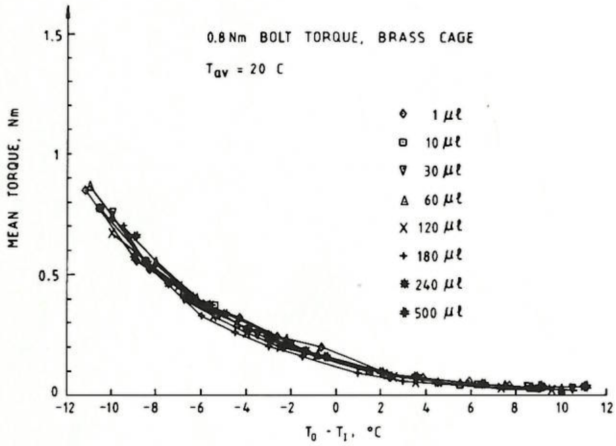

Fig 16. Mean Torque vs. Thermal Gradient as a function of Z25 oil quantity, KA070XP2A.

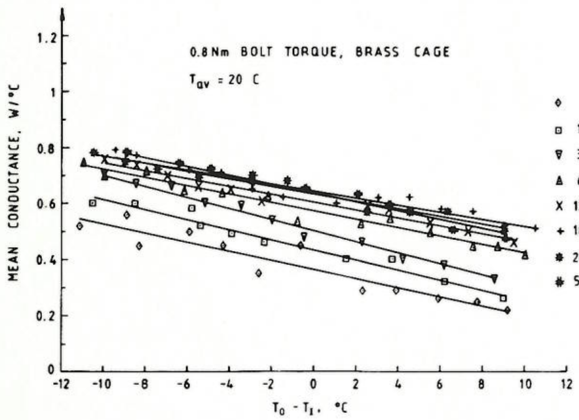

Fig 17. Mean Conductance vs. Thermal Gradient as a function of Z25 oil quantity, KA070XP2A.

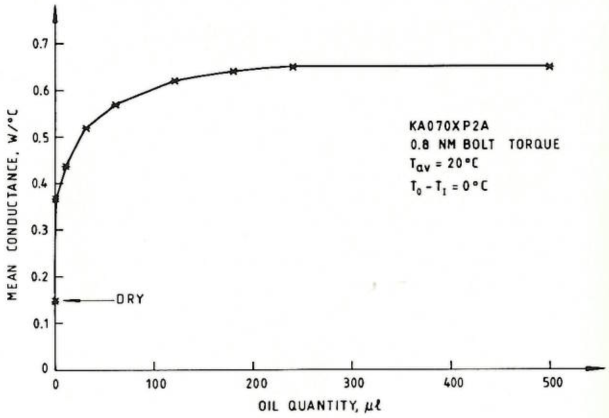

Fig 18. Mean Bearing Conductance vs. Z25 lubricant quantity.

6.5.2 BP110 oil. Like Fomblin Z25, BP110 lubricant addition had no influence on bearing torque and gave identical results to those for Z25, shown in Figure 16. In previous work (Ref. 6) 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).

The effect of BP110 oil addition on the conductance of bearing KA070XP2A is summarised in Figure 19.

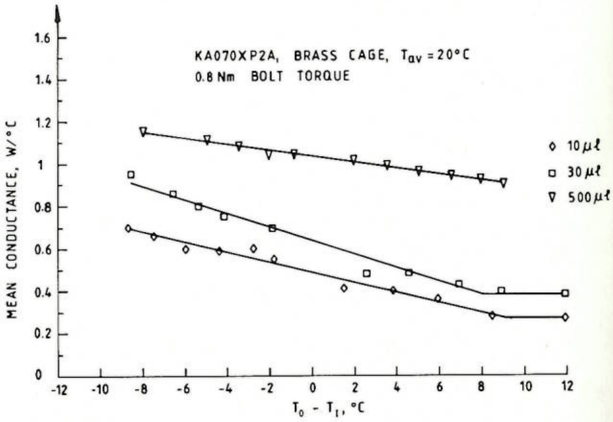

Fig 19. Mean Bearing Conductance vs. Thermal Gradient as a function of BP110 oil quantity.

6.5.3 Comparison of oils. Table 4 (derived from Figures 17 and 19) shows, under isothermal conditions, the relative influence of 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 (ie minus dry component).

Table 4 Influence oil type on oil conductance component

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

Inspection of Table 4 reveals that, for flooded lubrication, 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 5) 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.

6.5.4 Grease lubrication. Figure 20 illustrates the effect of grease lubrication on bearing KA070XP2A. 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 with the addition of a PTFE thickener, 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 ie bridging the conductance path of the ball/race contacts.

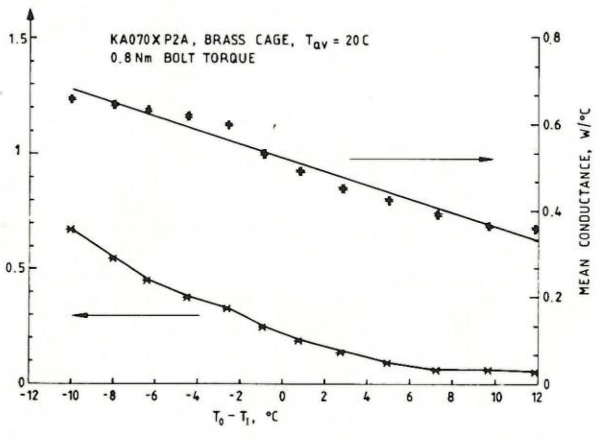

Fig 20. Mean Bearing Torque and Conductance vs. Thermal Gradient, Braycote 3L-38RP Grease.

6.6 Cage type

The influence of differing cage types on the torque and and conductance is shown in Figures 21 and 22. Three cages were studied:

i. Standard brass formed ring, "snap-over" type

ii. Nylon segmented "snap-over" type

iii. Helical coil springs

Figures 21 demonstrates that bearings conductance is independent of cage type.

Only minor differences in bearing torque were found (Figure 22), where spring separators tended to give higher torques when the bearing was subjected to negative temperature gradients. The insensitivity of mean torque to cage type implies that, at a rotational speed of 1RPM no significant cage "hang up" occurs even with the solid (loose pocketed) brass cage. In confirmation of this, longer-term torque variations of four-point contact ball bearings were assessed by rotating a bearing in vacuum over 300 revolutions at 1RPM and continuously monitoring the torque. This test gave a uniform mean torque which was invariant with time. Thus the advantage claimed for the softer nylon cage and for the spring separators, viz, better tolerance to ball speed variation (hang-up), would not be evident under the present conditions.

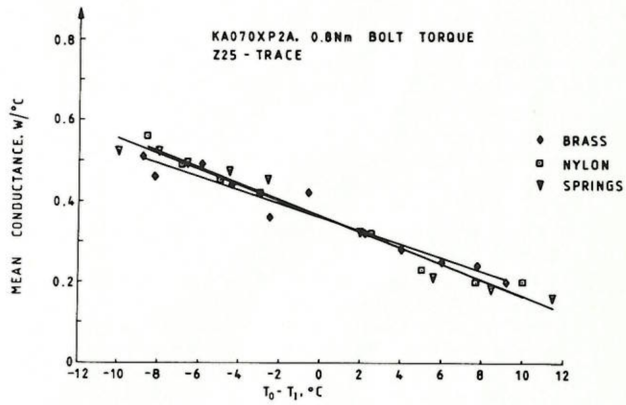

Fig 21. Mean Bearing Conductance vs. Temperature Gradient as a function of Cage Type.

7. GENERAL DISCUSSION

It is apparent from the results that the stiffness, radial conductance and torque of large diameter thin-section four-point contact ball bearings involve the complex

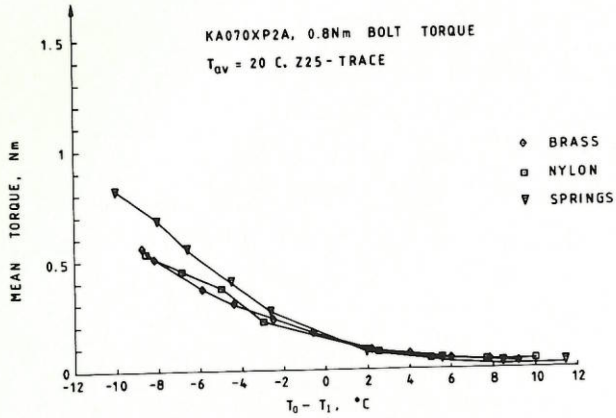

Fig 22. Mean Bearing Torque vs. Temperature Gradient as a function of Cage Type.

interaction of several variables and as such a rigorous theoretical analysis has not been attempted. However the relationships between stiffness, torque and conductance with internal preload and thermal gradient, for preloaded bearings, are in broad agreement with those predicted with reasonable assumptions from theory.

A further consideration for these 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 Kaydon bearings, the wide tolerance bands of internal diametral fit (Ref. 7) mean that there would be considerable uncertainty in the start point and, therefore, in the ball load and dry torque after interference fitting. 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. Also unknown is the effect of interference fitting on torque and conductance; in this work the bearing races have been allowed to flex under thermal load. Constraining the bearing races would tend to give higher bearing torque and conductance values because of the increased load at the ball/race contacts.

The optimum mounting installation would probably combine the use of external diametral clearance for ease of bearing fitting (and removal) with a low axial clamp. Such a value of axial clamp will minimise ring bending but will be sufficient to ensure the bearing cannot physically move in its housing. Measurement of the bearing torque (or deflection) after installation will give the true internal preload of the bearing as mounted.

8. CONCLUSIONS

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

a. Internal preload and external axial clamping force on the rings both 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 low speed torque exhibits an almost square law relation with thermal strain.

d. Unlike a normal ball bearing, the cage makes a small contribution to the torque.

9. ACKNOWLEDGEMENTS

The bearing rigs were designed by J A Duvall, ESTL.

10. REFERENCES

1. Rowntree R A 1982, The axial and moment compliance of thin-section four-point contact ball bearings, ESTL/TM/33.

2. Rowntree R A 1982, The moment compliance of thin-section four-point contact ball bearings at light axial clamping forces, ESTL/TM/35.

3. Rowntree R A & Todd M J 1983, Thermal conductance and torque of thin-section four-point contact ball bearings in vacuum, ESA(ESTL)54.

4. Heemskerk K F et al 1975, A test rig to measure the thermal conductance and frictional torque of bearings in vacuum, Proc First European Space Tribology Symp, Frascati 9-11 April 1975, ESA SP-111, 149-155.

5. Stevens K T & Todd M J 1980, Thermal conductance across ball bearings in vacuum, ESA TRIB 1, 15-40.

6. Todd M J 1980, A method for the calculation of torque in dry lubricated ball bearings including the influence of temperature fit, Proc Second Space Tribology Workshop, Risley 15-17 October 1980, ESA-SP-158, 87-93.

7. Kaydon Corporation, Kaydon bearing division catalogue on "Reali-slim" bearings.