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DESIGN PRINCIPLES OF A ROTATING  
MEDIUM SPEED MECHANISM

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SUMMARY

Design principles of a medium speed mechanism (MSM) are presented, including discussion on the relative merits of Beryllium and Aluminium as structural materials. Rotating at a speed of 60 rpm, the application envisaged for the MSM was as a despin bearing for the despun platform or despun antenna of a spin stabilised satellite.

The MSM has been built and tested to qualification level and is currently undergoing real time life testing at the European Space Tribology Laboratory.

INTRODUCTION

In 1971/72 the European Space Technology Centre commenced its supporting technology programme for the development of critical applications spacecraft components. At that time the satisfactory performance of the despin configured spacecraft had been demonstrated for satellite missions with medium power requirements. However a critical item of equipment in this configuration was the despin mechanism itself, both from the point of view of its own status as a single point failure and its effect on the stability of the satellite. Accordingly, Dornier System was awarded an ESTEC contract for the design and construction of a "medium speed mechanism" (MSM) capable of operating at despin speeds and suitable for use with either a despun antenna or platform. A central part of the design study was the choice of structural material, considerable thought being given to the use of Beryllium (Ref. 1).

The performance of the MSM and the environmental test requirements are summarised in Tables 1 to 3.

Table 1: MSM PERFORMANCE DATA

DESIGN LIFE:	> 7 years
SIZE:	375mm long x 214mm dia. (housing flange) 35mm bore
MASS:	10.7 kg
NOMINAL SPEED:	60 rpm (30 rpm to 150 rpm range)
POWER CONSUMPTION AT 20°C:	1 Watt (28V DC supply)
WOBBLE ANGLE:	7 arc seconds
RADIAL STIFFNESS:	$1.9 \times 10^5$ Nm/rad
STATIC LOAD CAPACITY:	> 1.5 kN
POWER TRANSFER:	4 x 1.25A channels giving 300 W at 60 V
SLIP RING NOISE:	< 50 mV/A
SIGNAL TRANSFER:	5 channels 2 to 60 kHz at peak signal level of 15 V (see Fig. 5)
CROSS TALK:	-40 dB

Table 2: MECHANICAL TEST ENVIRONMENT (at Dornier System)

The following were applied in all three orthogonal directions.

Static:	+ 20.000 N for 5 minutes
Contant Acceleration:	+ 18 g for 5 minutes
Sinusoidal Vibration:	5 to 15 Hz 9 mm o - p displacement 15 to 200 Hz 8 g o - p acceleration (2 octaves/minute)
Random Vibration:	25 to 100 Hz 3 dB/oct increasing to 0.2g <sup>2</sup> /Hz 100 to 2000 Hz 0.2g <sup>2</sup> /Hz flat (for 5 minutes) notching to zero between 140 and 200 Hz for lateral vibration
Pyrotechnic Test:	all pyrotechnics fired twice
Ambient Integration:	50 hrs. operation with 150 kg and 150 kgm <sup>2</sup> inertia attached to shaft flange.

Table 3: THERMAL VACUUM TEST ENVIRONMENT (at ESTL)

T<sub>1</sub> = Shaft-Flange, T<sub>2</sub> = Housing-Flange mounting and environment

Test No.	Duration (hrs)	T <sub>1</sub> °C	T <sub>2</sub> °C
1	8	+ 60	+ 50
2	8	+ 60	- 10
3	8	- 20	- 10
4	8	- 2	- 10
5 <sup>+</sup>	72 runs and 100 runs	-5 to -17 to -5	+ 20
6 <sup>+</sup>	72 runs and 100 runs	-45 to -2 to +45	+ 20

<sup>+</sup> in simulation of eclipse

#### DESIGN DERIVATION

##### Configuration

A fully modular concept (off-load system, bearings, motor, signals, power slip rings) was rejected because of the inherent penalties of mass and length and the limited available separation of the bearings, which prevented the achievement of a required 20 arc second wobble angle. The bearing separation chosen was 200 mm and the space between was utilized for the location of signal transformers and motor/resolver location. A speed pick-up with a housing-mounted carrier for redundant Gallium Arsenide photo diodes and a shaft-borne slotted disc were arranged outside that space (Fig. 1). This module therefore contained all the elements required for a despun antenna, including a 35 mm bore for waveguide location.

The power slip rings, which would be required for the despun platform configuration, were, however, contained in a separate module, outside the bearing compartment and utilised a smaller diameter shaft for low friction and long life.

The brushless D.C. motor and the contactless signal transformers were chosen to eliminate the possibility of wear debris contaminating the bearings. ABEC 9 angular contact ball bearings were used, made from cold temperature stabilised, consumable electrode vacuum melted 52100 steel.

The bearing off-load was integrated into the MSM shaft and housing flanges in order to maintain a direct load path between the spun and despun structures. Off-loading is achieved by the axial displacement of the shaft flange against a conical seating by the jamming action of a flat headed pin. Release is effected by two pyrotechnics (one being redundant) moving an intervening blocking ring which permits the pins to retract. Note that Fig. 1 is split along the MSM centreline showing the free and off-loaded configuration.

Extreme care in construction was necessary in the region of the central flange due to the tight wobble requirement. The eccentricity of the flange was measured and marked so that on assembly, the most favourable alignment could be attained.

The determination of wobble angle depends upon which part of the MSM (shaft or housing) is despun. References 2,3 and 4 discuss the effects of despin bearings on the stability of despun satellites, affirming that it is preferable that damping processes occur on the despun rather than the spinning structures. It is clear that the shaft, being inherently more compliant than the housing, should be despun for the achievement of maximum stiffness (see also Ref. 5). For this reason also the flexible bearing mounting is favourably placed on the shaft. Calculations gave the maximum shaft wobble as 17 arc seconds (measured later as 7 arc seconds).

### Thermal Design

Bearing preload is maintained and limited by use of a radially stiff, axially compliant bearing mounting. The use of sliding fits to give tolerance to thermal expansion effects is now thoroughly discredited for despin mechanisms, both from the viewpoint of tribological failure and destabilising effects (Refs. 3 and 6).

However, problems of bearing compression due to thermal variations were not completely overcome by the use of flexible mountings especially when choosing between the light metals (Beryllium and Aluminium) necessitated by the requirements of weight optimisation. Steady state thermal model calculations yielded very similar temperature distributions for both materials. Consequently Beryllium appeared to have a slight advantage over the Aluminium due to the close match of its thermal expansion coefficient with steel. However if a temperature differential of 10°C is taken as a design case to cover transient conditions such as those occurring during eclipse the greater stiffness of the Beryllium can distort the bearing (the wall thicknesses are very similar for both Al and Be) since the fits are designed as interference fits for all of the steady state temperature cases required.

For a given bearing dimension the ratio of the contraction pressure to the radial interference ( $p/\Delta r$ ) permits a direct comparison between the materials. Fig. 2 gives  $p/\Delta r$  as a function of housing wall thickness for both materials. The shaded area indicates the practical range of wall thickness, consistent with stiffness requirements. It can be seen that the interference fit of an Aluminium bearing mounting can be a factor 2 or more down on that required

for Beryllium for the same compressive stress.

This advantage of Aluminium is offset by the greater influence of temperature gradients on the Aluminium/Steel fit. The worst temperature condition occurs when the shaft is hotter than the housing, when the outer housing (relatively) compresses the outer ring and the shaft (relatively) extends the inner ring. This thermal state is made less severe by the consequent implied condition that the bearing area is at a medium, near ambient temperature and thus that local thermal expansions are low. Further radial mismatches of this type are compensated by an enforced slight change in contact angle. At the lower bearing Beryllium is superior to Aluminium because the inner race is decoupled from the shaft by the flexible steel bellows mounting. However there the magnitude of the mismatch is less severe, contact angle variation being only about 1/3 of that of the upper bearing.

In order to further reduce the effect of radial thermal mismatch the outer bearing mount was cantilevered and tooth profiled to reduce its stiffness and tendency to distort the bearing.

Axial thermal mismatch is clearly less for Beryllium than Aluminium, but here it is a matter of design choice between reduced preload variation and increased stiffness of the flexible bearing mounting.

### Structural Design

The stability of a despun satellite is critically dependent on the radial stiffness of its bearing (see Refs. 5 and 6). A figure of  $5 \times 10^4$  Nm/rad has been taken as an acceptable figure.

Fig.3 gives the structural and deformation models from which the stiffness is derived. At the lower bearing, the deflections of shaft, housing, and bearing plus bellows are given by  $f_S$ ,  $f_H$  and  $f_C$  respectively. In addition to their radial deflection the bellows are subjected to an angular deflection  $\beta$  so that they exert a resistive moment  $M_B = \phi\beta$ , which results in a deflection  $f_m$ . ( $\phi$  is the angular stiffness of the bellows.)

The total deflection is therefore

$$f = f_S + f_H + f_C - f_M$$

Using the relationship  $\alpha = f/l$ , one can derive the relationship for the radial stiffness:

$$\phi_R = \frac{6EI C_L C_B l^2}{2 l^3 (1+1/K) C_L C_B + 6EI (C_L + C_B) - 3\phi l (C_L + C_B)}$$

where the stiffness of the shaft is EI and that of the housing is K (EI).

This relationship permits the examination of the influences of various parameters on the stiffness. Fig. 4 shows the angular stiffness as a function of bearing distance for different structural materials. The benefit of Beryllium is clearly demonstrated as are the advantages of a hybrid structure (Be-shaft, Al-housing). Alignment dictates indicated a bearing distance of 200 mm where the stiffness of the all Aluminium construction was sufficient.

Although both thermal and structural analyses showed the technical superiority of Beryllium, the design and spacecraft criteria were met adequately

by an Aluminium structure and this was chosen in place of the more costly and fragile alternative.

#### Lubrication System

Liquid lubrication for the bearings was chosen as the most reliable means of long life lubrication. At that time the solid lubricant systems based on metal films and mechanically bonded  $\text{MoS}_2$  systems had not proved their reliability. The choice lay between the lubricants F50, Apiezon C and BP110.

Fig. 6 shows the results of film thickness calculations for these oils, covering the range of temperatures expected. The scatter in film thickness (vertical spread) is caused by the application of two film thickness formulas, based on point load and line load (Ref. 7).

The lower oil film thickness for the F50 led to the rejection of that lubricant and the BP 110 was chosen after consideration of the calculated friction torques, Fig. 7. Bearing friction was calculated from the formula given in Ref. 8. This friction characteristic is not representative at low temperatures since oil starvation results in a reduction of torque, albeit at the expense of oil film thickness.

The lubricant system chosen consisted therefore of 5 % BP2110 grease in the bearing tracks with BP110 oil impregnated in the phenolic cages and nylasint. These lubricants are described in Ref. 9. BP110 is a fine cut mineral oil, and is modified to the grease, BP2110, by the introduction of an oleophilic graphite-lead composite. The cages were boiled in chloroform and outgassed at  $10^{-2}$  Torr for 2 hours at  $90^\circ\text{C}$  to disperse included monomers and impregnated with BP110 at  $70^\circ\text{C}$  and  $10^{-2}$  Torr. This resulted in about 0.25 gms of oil per cage, an effective porosity of about 2.3 %. A similar cleaning and impregnation process on the reservoirs gave about 3.3 gms/reservoir (about 25 %).

With the intention of providing each bearing with its own enclosed lubrication system, the reservoirs were placed close to each bearing, one on the shaft and one on the housing to ensure that at least one reservoir was always at a higher temperature than the bearing. Narrow clearances relative to the moving counterpart for the Aluminium backed reservoirs and carefully applied anticreep barriers (Tillan M2, which is derived from FX 706 of 3M Company) were used to limit the loss of oil from the immediate bearing vicinity. The materials used in the slip ring module were  $\text{AgCuMoS}_2$  compacts against a silver ring, a combination that has been extensively tested (Ref. 10). Brush contact pressure was maintained by a soft spring ensuring uniform contact conditions throughout life.

#### TESTING PROGRAMME

The MSM successfully completed the tests specified in Tables 2 and 3, which were established at qualification level (Ref. 11). The tests were not without incident, however: the MSM slip ring module required refurbishment after its shaft broke during vibration testing and failure occurred during an initial series of thermal vacuum tests. This latter failure was due to a tolerance being disregarded during the manufacture of the upper molecular seal and escaping detection during inspection. This caused jamming of the MSM during setting up of test 3 of Table 3 with the shaft temperature  $32^\circ\text{C}$  and housing temperature  $8^\circ\text{C}$ . This was a disturbing experience in view of the careful attention that had been placed on the thermal design of the MSM and is significant support for thermal vacuum differential temperature

testing at acceptance level (build standard), a test philosophy that could well have been applied with benefits for past despin mechanism programmes.

The MSM is now commencing its real time life test at the European Space Tribology Laboratory. The test will consist of alternating periods of six months with different levels of thermal gradient applied across the mechanism. Simulated eclipse testing will be performed at the end of each six month period.

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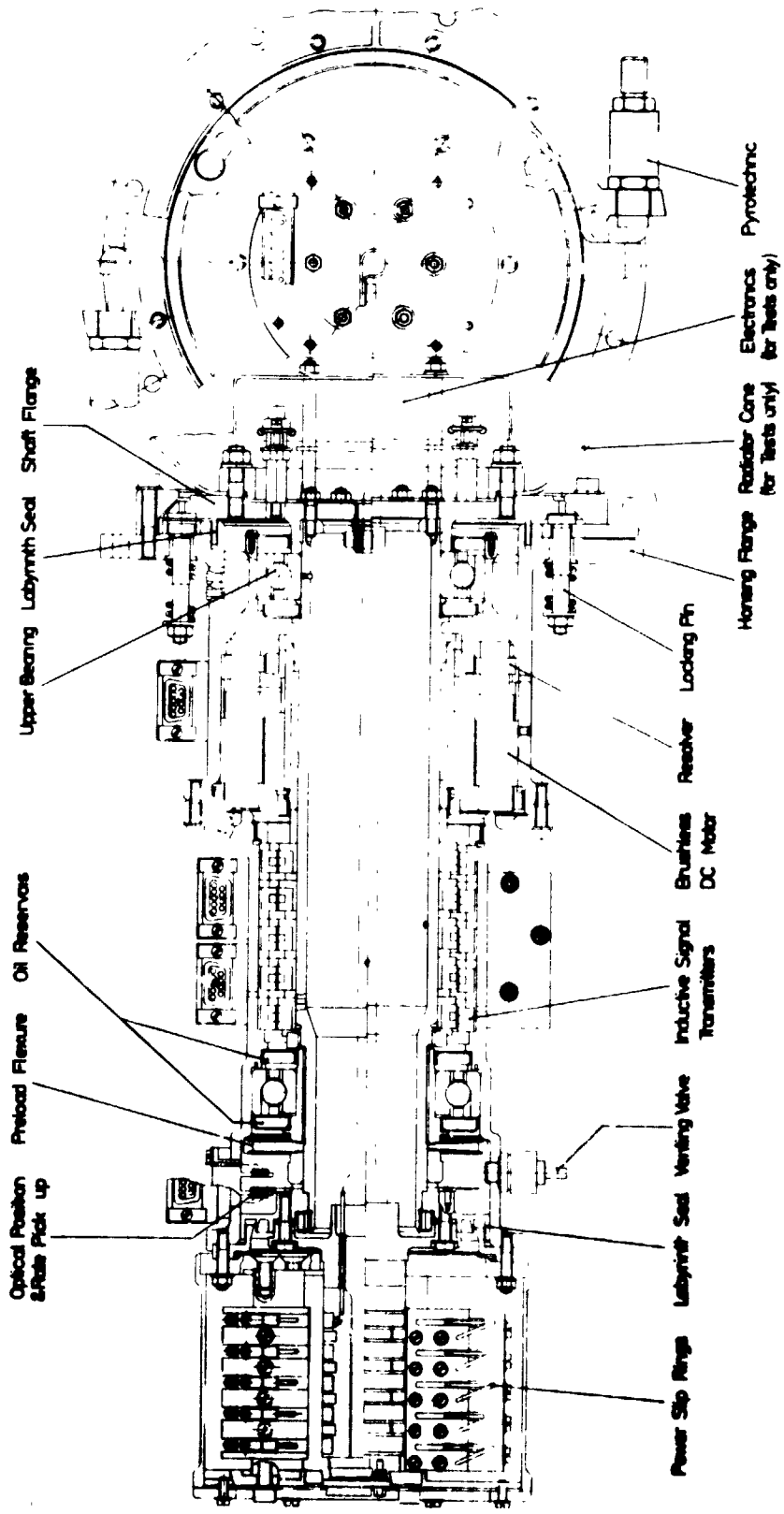


Fig 1 Medium Speed Mechanism

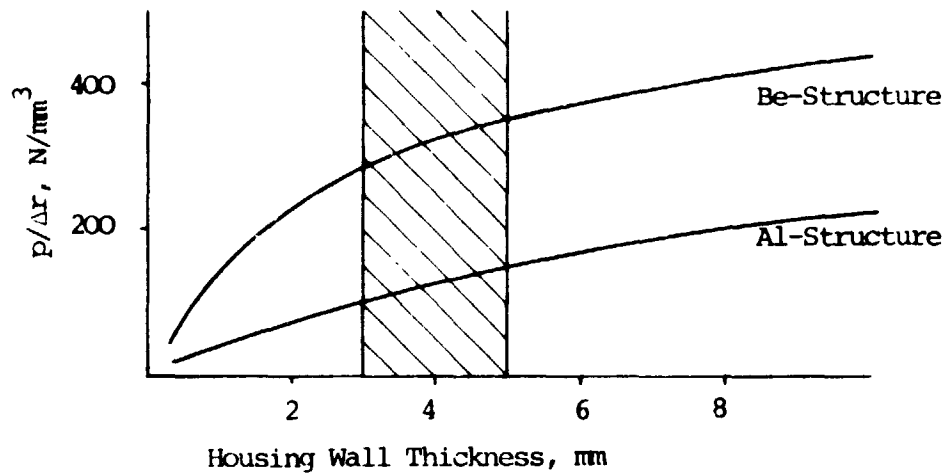


Fig. 2 Ratio of contraction pressure to interference  $p/\Delta r$  vs housing wall thickness

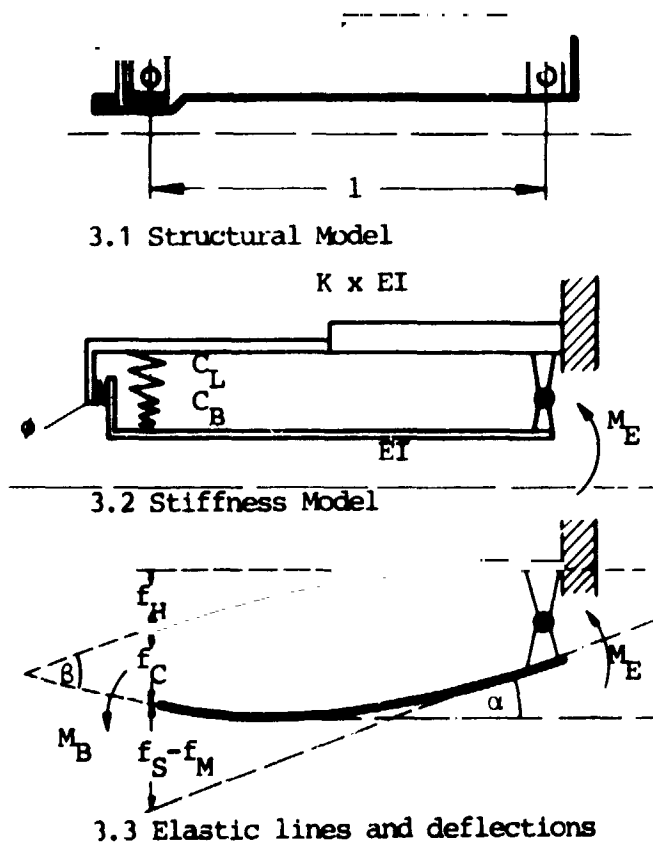


Fig. 3 MSM stiffness model



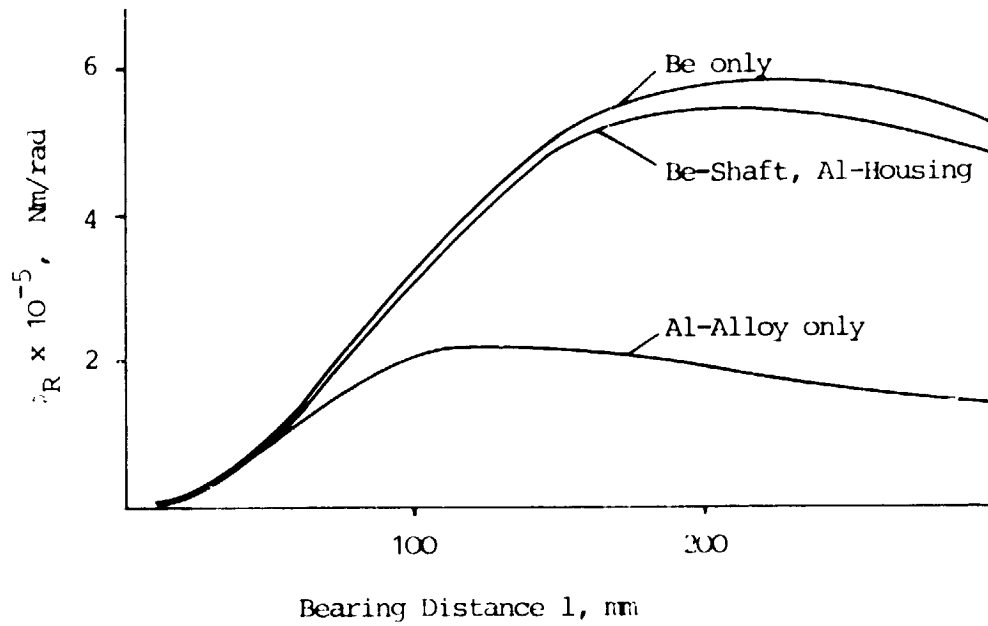


Fig. 4 Angular stiffness  $\phi_R$  vs bearing distance for different structural materials

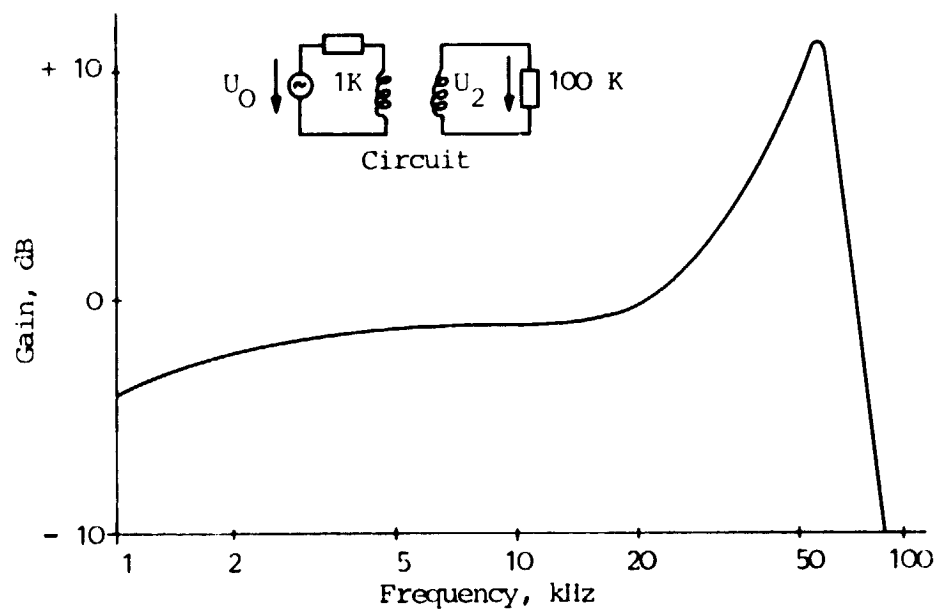


Fig. 5 Gain of inductive signal transmitters vs frequency

Temperature Range: 0 to 50°C  
 Bearing: 6011 GMN 55/90/18 (18 Balls)  
 Combined Surface Roughness: 0,12µm

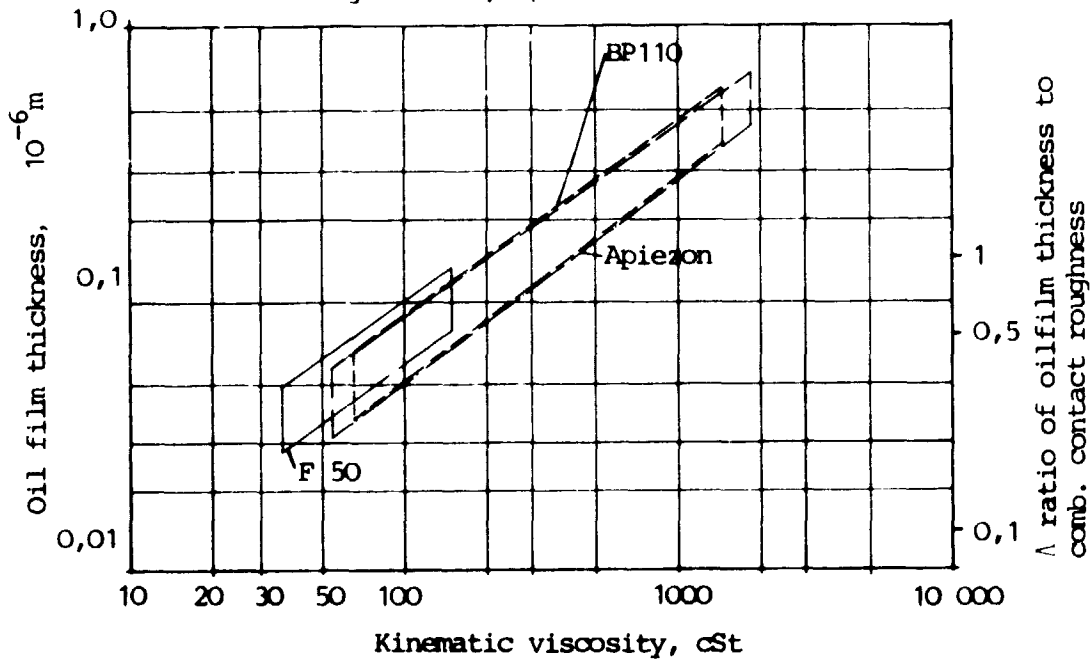


Fig. 6 EHD oil film thickness at 60 rpm

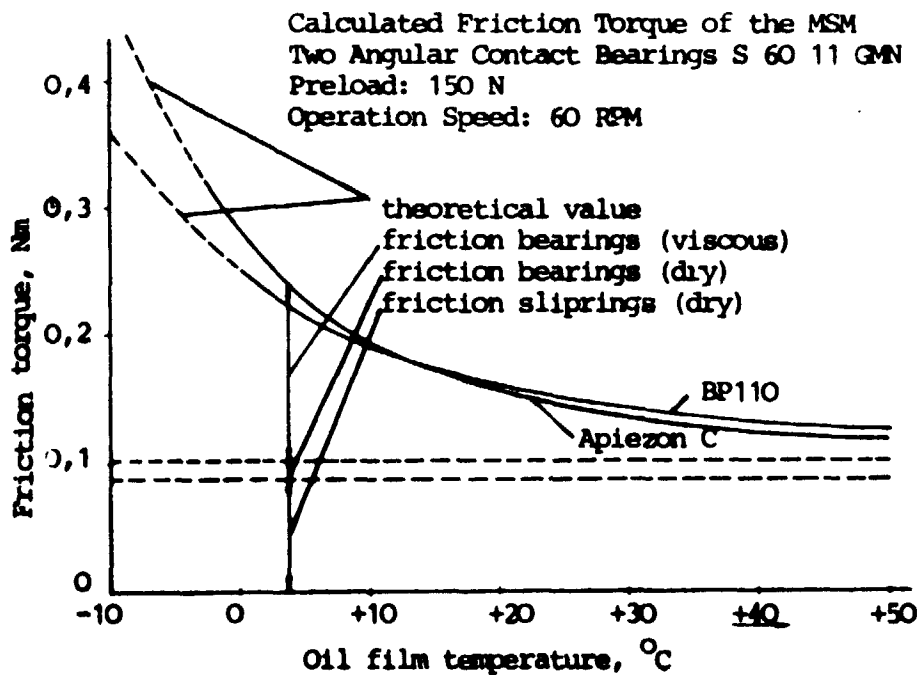


Fig. 7 Calculated friction torque of the MSM at 60 rpm