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DEVELOPMENT OF LONG-LIFE, LOW-NOISE LINEAR BEARINGS FOR ATMOSPHERIC INTERFEROMETRY

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ABSTRACT

This paper describes the development of dry-lubricated linear bearings for use on the Michelson Interferometer for Passive Atmospheric Sounding (MIPAS). Two candidate bearing systems were developed and tested. In the first, use was made of linear roller (needle) bearings equipped with a pulley-and-cable arrangement to prevent cage drift and to minimise roller slip. The second design was of a roller-guided bearing system in which guidance was provided by ball bearings rolling along guide rods.

The paper focuses on the development of these linear bearing systems and describes the approach taken in terms of bearing design, lubrication methods, screening programmes and thermal-vacuum testing. Development difficulties are highlighted and the solutions ultimately adopted are described.

INTRODUCTION

The Michelson Interferometer for Passive Atmospheric Sounding (MIPAS) is an ESA-developed instrument for use on the first European Polar Platform, ENVISAT-1, which is planned for launch in 1998. The design calls for very high precision linear bearings for the two interferometer slides. These slides carry corner cube reflectors which describe a back-and-forth motion, this motion being in countermovement so as to cancel disturbing forces. The bearings should be capable of maintaining a low-noise performance whilst operating continuously at low temperature (-70 deg.C) over four years.

The requirement to operate at low temperature and the need for zero contamination of the optical components, precludes oil lubrication. However, the requirements of long duty and low frictional noise combine so to push the capability of solid lubrication systems to their limits. The work reported here is principally concerned with assessing the ability of MoS_2-based solid lubricants to meet these system requirements.

Accelerated life tests were undertaken on two candidate bearing systems. The first comprised a pair of linear roller (needle) bearings lubricated with sputter deposited molybdenum disulphide. The second test was carried out on a roller guide system which utilised conventional rotary ball bearings loaded against two parallel rods. This system was also lubricated with sputtered MoS_2.

Additionally, supplementary tests were carried out - simultaneously with the life tests - with the aim of assessing alternative lubricants and material combinations.

MECHANISM REQUIREMENTS

The task of the bearings is to carry and guide a corner cube slide of mass 1.7kg over a stroke of 110mm. The nominal cycling motion requires a trapezoidal speed versus time profile. The absolute speed of the corner cubes is 25 mm/sec and must be controlled to achieve a relative velocity (w.r.t the speed of the second slide) error of < 1.2% (3σ).

The main requirements, crucial to the successful performance of the instrument, are:

- low and stable friction, so as to maintain a drive force of <1N
- linear motion over 110mm with a velocity of 25mm/sec
- long lifetime: four years life on-orbit under continuous operation (9 secs per cycle)
- operation at -70 deg.C
- low vibration and play (< 10 μm)
- no release of contamination

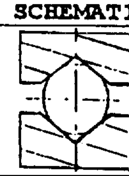
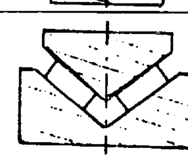
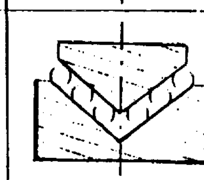
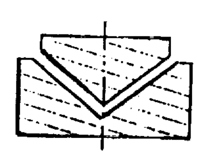
DEVELOPMENT OF THE TWO CANDIDATE BEARING SYSTEMS

a)Design 1: linear roller bearing guided slide

Initial screening tests on candidate linear bearings

In order to select the most suitable type of linear bearing for this design, screening tests were undertaken at the start of the programme on four types of linear bearing. The bearing types and their specifications are given in Table 1. The types examined were a slide bearing; a roller bearing; and two types of ball bearing. In each case, lubrication was provided by applying a 1-micron thick coating of magnetron sputtered MoS_2 (according to ESTL procedure ESTL/QP/073) to the races. Bearings were tested in pairs and operated until completion of 370,000 cycles at a stroke length of 90 mm. The tests were undertaken under high vacuum at a temperature of -50 deg.C, the drive force being monitored throughout the test period.

Table 1 Types of linear bearing assessed in screening tests

MAKE	TYPE	SPECIFICATION	COATINGS	CAGE	SCHEMATIC
Schneeberger	Ball	R9200 R9150 AK 9 x 6 Balls: 6mm diam	Sputtered MoS2 on balls & races	MoS2-coated steel	
Hydrel	Roller	ML 5020/15Lx200 M 4020/x200 V 4020/15x150 MW 15 x 83.5	Sputtered MoS2 on raceways	PTFE-coated	
Hydrel	Ball	ML 5020/15Lx200 M 4020/ x200 V 4020/15 x150 MBW 2X15X83.5 Balls: 2mm diam	Sputtered MoS2 on raceways. Ball: TiC-coated	PTFE-coated	
Hydrel	Slide	ML 5020/15Lx200 M 4020/ x200 V 4020/15 x 100	Sputtered MoS2 on raceways: Precoated with Me-CH	No cage	

A major finding of these screening tests was that those bearings which utilised rolling elements (ie. all the bearings except the slide bearing) were adversely affected by creep of the rolling elements and cages. In all cases this led to high force spikes at one or both ends of travel. These high forces were generated as the cages were driven into contact with the bearing end-stops at which point any further movement of the non-stationary races resulted in sliding motion between races and rolling elements. In addition to causing higher friction forces this effect also resulted in more rapid wear of the MoS_2 film.

The best overall performance was given by the roller bearing which lasted the planned test duration, maintained its low friction and exhibited the squarest drive force profile. Furthermore, theoretical analysis indicated that, under identical operating conditions, frictional losses would be lower in the roller bearing than in the linear ball bearings and the slide bearing. For these reasons it was decided to select the linear roller bearing for accelerated life testing.

Design details

The linear roller bearing guided slide (Fig.1) consisted of 2 sets of Hydrel V- and M-shaped raceways (Fig.2a). The bearings were preloaded by a compliant suspension (achieved using flat springs) of one stationary raceway. This compliant suspension provided a constant preload, insensitive to thermal changes, wear-out and residual misalignment. Preloading was adjustable using four compression springs, housed in special set screws (Fig.2b).

Each bearing (Fig.2a) was fitted with a PTFE-coated (ALTEF coating 40-50μm) aluminium cage of length 165mm, the length of the races being 230mm. Each bearing contained 12 steel rollers (of length 4.5mm and diameter 2mm), with six rollers arranged symmetrically at each end of the cage. The roller groups at each end were separated at a distance greater than the travel of the rollers so as to prevent overlapping of the wear tracks. The bearing races and rollers were lubricated with sputtered MoS_2 (thickness 1 μm).

In order to prevent roller and cage creep, and thus eradicate high end forces, a pulley-guide system was devised which ensured that the cages were driven at half the speed of the linear carriage. This was achieved as follows. Each end of the cage was fitted with a Vespel SP1 pulley which ran on an MoS_2-coated steel axle. A thin stranded steel cable looped around each pulley and was solidly clamped to the end of the carriage mounted raceway, whilst being flexibly loaded via a spring to the end of the static raceway. A schematic diagram illustrating the principle of the pulley guide system is shown in Fig.2c.

b) Design 2: ball bearing roller guide

A second design of bearing system was devised in which guidance was provided by ball bearings rolling against guide rods. This design was chosen as it was expected to yield inherently low friction and, since no conventional linear bearings were employed, problems associated with cage wandering and its control were eliminated.

The roller guide system is depicted schematically in Figs.3a and 3b. The carriage is supported by radial ball bearings which run on a pair of parallel guide rods (precision ground shafts). The guide rods were manufactured from hardened steel and coated with thin dense chrome (TDC, an Armoloy Technology Coating) prior to being sputter coated with molybdenum disulphide (to ESTL process ESTL/QP/073). The bearings used throughout were of standard 440C material fitted with TiC-coated balls and Duroid (PTFE/MoS_2/glass fibre) cage. The raceways were coated with sputtered MoS_2.

The slide utilised a total of eight ball bearing pairs. Of these, five pairs were used for guiding purposes and the remaining three pairs were used to preload the system. Two sets of triple bearing pairs (spaced 120 degrees apart on the circumference of the guide rod) ran on the upper guide rod. This bearing arrangement is tolerant of misalignment and thermal- or load-induced deflections. Zero play was achieved by means of springs which provided radial preloading of each roller. Each ball bearing pair was axially soft preloaded (by means of wavy washers) - again with the aim of achieving high running precision. The nominal radial preload of the triple bearing set was 3 to 5N. The nominal preload of the lower ball bearing pair was 9 to 15N. This difference in bearing loads was chosen, following calculation of frictional losses, so as to achieve equal friction forces on both the upper and lower guide rod, thus minimising torque disturbances on the slide.

TEST CONDITIONS

Tests on candidate bearings systems

Each bearing system was subjected to oscillatory motion over stroke lengths of up to 110mm. All tests were undertaken in high vacuum (< 10^{-6} torr) at a temperature of -70 deg.C. The tests were accelerated by running the bearings at speeds which were three times higher than their design speed.

In the case of the Hydrel needle bearings, further acceleration of the test was achieved by increasing the bearing preload above its design value. The aim was to accelerate the life by a factor five through increases in load. This was achieved in the following manner. First, the variation of contact stress (per roller) as a function of bearing load was calculated (Fig.4). The nominal design preload for the Hydrel bearing is 10N. This corresponds to a mean contact stress of about 50MPa (Fig.4). Secondly, it is known from empirical data for (angular contact) bearings lubricated with sputtered MoS_2 that the low-torque life is inversely proportional to (contact stress)$^{3.8}$. Using this relationship as a guide we calculated how the MoS_2 life on linear roller bearings would be reduced for values of contact stress in excess of 50MPa. This reduction, which we term the acceleration factor, is plotted in Fig.5. as a function of contact stress and load. It follows from these plots that in order to reduce film life at a bearing load of 10N by factor five, the load should be increased to 23N.

Acceleration of the life test of the ball bearing roller guide system was limited to a threefold increase in slide velocity. Accelerating the life test by other means (such as increasing the radial load between bearings and guide rod) was rejected since it was difficult to predict with confidence the relationship between lifetime and load. However, by accelerating the test using a higher speed (x3) the desired number of cycles could not be achieved on the programme timescale. Nevertheless, this approach gave the option of continuing testing if this was considered appropriate at a later stage.

Initially, tests on the roller guide system were made with a stroke of 100mm. Following completion of 2 x 10^6 cycles it was decided to introduce short stroke cycling into the test so at to (a) increase the total number of cycles that would otherwise be achieved and (b) more accurately simulate the operation of the

MIPAS bearings which, in practice, undergo short strokes during calibration periods. Thus following the first 2 x 10^6 cycles, testing comprised cycling alternately over short and long strokes. The shortened stroke had a length of 20mm and in fact was too short to allow a period of constant velocity at the values of acceleration used (250 mm/sec^2). All force measurements were made over the longer stroke length of 100mm.

Supplementary tests on alternative material combinations

The lubricants and materials used in both the linear roller bearing and ball-bearing roller guides were chosen following a survey and trade-off of promising candidates. It was, however, considered worthwhile to undertake supplementary tests in which alternative methods of lubrication for the critical design areas could be assessed. These critical areas were deemed to be the bearing/guide rod interface, the ball bearings and the pulley wheel/axle interface.

To this end a simple rig was designed in which ball bearings could be rolled under load against a guide rod and pulley wheels could be made to rotate under the action of a loaded cable. In this way the material combinations shown in Tables 2 and 3 were tested and compared. Conditions of testing (ie loads, speeds, vacuum environment etc.) were representative of those occurring within the candidate bearing systems under life test. The supplementary tests were continued until completion of 2.5x 10^6 cycles.

Table 2 Material Combinations in Pulley/Cable Tests

Pulley Material	Cable Coating	Axle Lubrication
Vespel SP1	MoS2	Sputtered MoS2
Vespel SP3	Nylon	Sputtered MoS2
Vespel SP3	MoS2	Sputtered MoS2
Lead Bronze	MoS2	Ion-plated Lead

Table 3 Material combinations in bearing/rod tests

Race	Balls	Cage	Outer Race (outer surface)	Guide Rod
MoS_2	TiC	Duroid	-	TDC/MoS_2
Lead	440C	Lead bronze	Lead	TDC
Lead	440C	Lead bronze	-	TDC/MoS_2

THERMAL-VACUUM TEST RIG

The test rig employed for the accelerated life tests is depicted schematically in Fig.6. Each bearing system was mounted in a housing which was itself attached to a heat exchanger which controlled the specimen temperature (assisted by an enclosing thermal shroud). The heat exchanger support was bolted to an annular support plate which was carried by three piezoelectric force transducers which monitored the bearing drive force. The transducer bodies were supported in a further annular support plate attached to the vacuum chamber lid by three support pillars.

Drive was applied from a crosshead to the linear carriage through a link arm, which comprised a pair of spherical rod-end bearings. The crosshead was connected to a Roh'lix linear drive mechanism. The Roh'lix is a proprietary component which incorporates six ball bearings mounted in two sets of three on a block around a central shaft. The bearings are angled relative to the drive shaft such that shaft rotation induces linear motion of the Roh'lix block. The drive shaft was supported at both ends by support bearings and was rotated, via a rotary vacuum feedthrough, by a high-resolution microstepping motor (25,000 steps per revolution).

A linear position encoder (Sony Magnescale) was fitted to provide feedback on the position of the crosshead/test bearings. The required motion profile was programmed via a computer terminal. The motion profile was then generated by a microprocessor indexer card whilst monitoring the feedback from the encoder and thus ensuring that the bearings underwent consistent reciprocating motion over the same length of stroke.

The Roh'lix bearings, shaft support bearings and linear encoder were all lubricated with Braycote 601 grease. It was found necessary to refurbish these components on a regular basis (every 1.5 million cycles) due to severe degradation of the grease.

TEST RESULTS

Linear roller bearing guided slide (Design 1)

Prior to the vacuum life test, measurements were made (in dry nitrogen gas) of drive force versus bearing load. These measurements were undertaken so as to gain a measure of the contribution of the pulley-guide system to the overall drive force. Fig.7 shows the resulting plot of drive force versus preload. Clearly there is a residual drive force at zero preload which is attributable to frictional losses within the pulley system (and, to a degree, to friction at the cage/roller interfaces). Thus for the intended operational preload of 10N (and indeed the test preload of 23N), this residual component represents a significant contribution to the overall drive force.

Following the above tests, the preload springs were adjusted to give a bearing preload of 23N and the test rig mounted in the vacuum chamber. Prior to evacuation of the chamber, the bearings were run over a few cycles in nitrogen to confirm satisfactory operation.

The chamber was then evacuated to a pressure of better than 10^{-6} torr and the temperature of the bearings reduced (by passing refrigerated alcohol through the heat exchanger) until the outer bearing temperature reached -75 deg.C (the inner races attaining a temperature of -58 deg.C). These temperatures were then maintained for the duration of the accelerated life test which proceeded until completion of 3.5×10^6 cycles.

Fig. 8 shows the variation in mean drive force and peak drive force as a function of cycles over this period.

The behaviour of the bearing pair can be summarised as follows. During the first 10^5 cycles there occurs a sharp decrease in drive force (this, we believe, is principally attributable to the running-in of the pulley/cable system). Thereafter there is a more steady decrease in force until approximately 1.2 x 10^6 cycles are completed, after which the force does not change greatly with number of cycles. In this region, the mean force has a value of 0.3N.

Examination of the curve of peak force shows that there is a sharp decrease initially corresponding to the decrease observed in the mean force. Between 10^5 and 2.5 x 10^6 cycles the peak force does not show any great variation and lies in the range 0.6N to 1.0N. Cycles undertaken thereafter, however, show a distinct trend - the peak force increasing almost monotonically, reaching values of 1.7N at the end of test.

Ball bearing roller guide (Design 2)

Fig.9 shows the variation in mean force and peak force as a function of number of cycles. In general, the mean force has remained in the range 0.05N to 0.1N with no evidence of degradation. Likewise, no distinct trend is observed with the peak force, this lying in the range 0.15N to 0.24N.

Supplementary tests on alternative lubricants/material combinations

The results of the supplementary tests may be summarised as follows:

- wear of the pulley wheels (at axle interface) was least for the leaded bronze combination, followed by SP3 and SP1

- the highest and most variable drive forces were observed for the combination of lead-bronze pulley and lead-coated axle

- consistently low drive forces were observed for the combinations of Vespel pulleys (both SP1 and SP3) in conjunction with MoS_2-lubricated axles

- wear of ball bearing/guide rod interface was less with lead lubrication than with MoS_2 lubrication.

- lead lubrication of the ball bearings yielded torques which were approximately twice that generated by the MoS_2-lubricated bearings and thus gave a higher drive force.

DISCUSSION

Linear roller bearing guided slide

The use of a pulley/cable arrangement was successful in controlling the stroke and speed of cages within the Hydrel bearing, but its presence made a significant contribution to the drive force, and it is believed that wear debris from the pulley wheel generated additional frictional noise.

The decrease in mean force seen in the early stages of testing is, we believe, attributable to the running-in of the pulley cable system. Preliminary testing had demonstrated that, prior to life testing, approximately 80% of the drive force was attributable to frictional losses within the pulley-cable system (Fig.7). Thus any decrease in these losses would result in a significant decrease in drive force. That such decreases in frictional losses did occur is supported by evidence from the supplementary tests. In these, the mean force needed to rotate an SP1 pulley wheel decreased by a factor four during the first 10^5 cycles.

Whilst the mean drive force of the Hydrel bearings showed little change after the running-in phase, the peak force exhibited larger variations. Up to 2.5×10^6 cycles, and after running-in, the peak force remained within the range 0.5 - 1 N. However, as the cycles increased beyond this point, the peak became larger, its value at the end of testing being 1.7N, the highest force observed. These peak forces tended to occur near the end of the stroke. Examination of the race wear tracks formed by the rollers indicate that MoS_2 lubricant is still present in these regions, thus precluding lubricant loss as the reason for the high forces. However, there was a second much narrower wear track observed running parallel to some of the wear tracks. These additional tracks which extended beyond the ends of the roller tracks are lined by wear debris. From their position, it is clear that these tracks were caused either by rubbing of the cage on the raceway or by abrasive action of debris entrapped between cage and race. The latter effect is the more likely since there was little sign of heavy wear on the cages themselves. The most likely sources of debris are wear particles (Vespel SP1) from the pulley wheel whose path to the raceways would be via the relief holes in the pulley slots. It seems plausible therefore that the higher forces observed near the end of strokes are due to this entrapped debris rubbing against the raceways. Effects such as these could be reduced by having larger clearances in the bearings eg by having larger diameter rollers or thinner cages.

Another method of minimising this effect would be to manufacture the pulley wheel from Vespel SP3 since measurements of the wear of the different pulley wheel materials indicated that SP3 yielded lower wear whilst still providing low friction.

The use of sputtered MoS_2 on the raceways can be considered successful in that the coatings withstood 3.5 x 10^6 cycles under enhanced load. Our calculations show that this number of cycles is equivalent to 17.5 x 10^6 cycles at the operational load of 10N. This number is similar to that required in the lifetime of MIPAS.

Ball bearing roller guide

The roller guide system completed a total of 5 x 10^6 cycles under operational bearing loads. For most of the test period the mean drive force remained in the range 0.05 to 0.1N, the overall trend being one of a slow increase in mean force. The peak force varied between 0.15N and 0.25N but no trend was discernible. Force peaks were uniformly distributed across the force profiles with no particularly strong peaks occurring at the end of stroke. At completion of the test period there was no indication of bearing distress or that degradation was imminent.

Since the test bearings were not disassembled it was not possible to examine the component parts in detail. There were well defined wear tracks on the guide rods but the amount of MoS_2 coating remaining could not be assessed. However, our supplementary tests clearly show that lead coatings on the outer surfaces of the ball bearings are more effective than the sputtered MoS_2 in reducing wear at the interface between the ball bearing and the TDC-coated guide rod interface. It should be noted however that a depletion of lubricant on the guide rods would not necessarily lead to a higher drive force since the major contribution to frictional losses in the roller guide occurs within the ball bearings.

CONCLUSIONS

The use of sputtered MoS_2 on the raceways of the Hydrel bearings maintained effective lubrication between rollers and races over the equivalent of 17.5 million cycles (consistent with life requirements). The pulley/cable arrangement proved successful in controlling the stroke and speed of cages and the extremely high end-forces observed with uncontrolled cages were not seen. However, the pulley/cable system made a significant contribution to the drive force, and wear debris generated from the pulley wheel gave rise to additional frictional noise.

The ball bearing roller guide system generated low consistent friction forces throughout the test duration. The use of MoS_2 lubrication within the bearings was demonstrated to be the best choice, but supplementary tests indicated that thin lead films were more effective (than MoS_2) in preventing wear of the guide rods.

ACKNOWLEDGEMENTS

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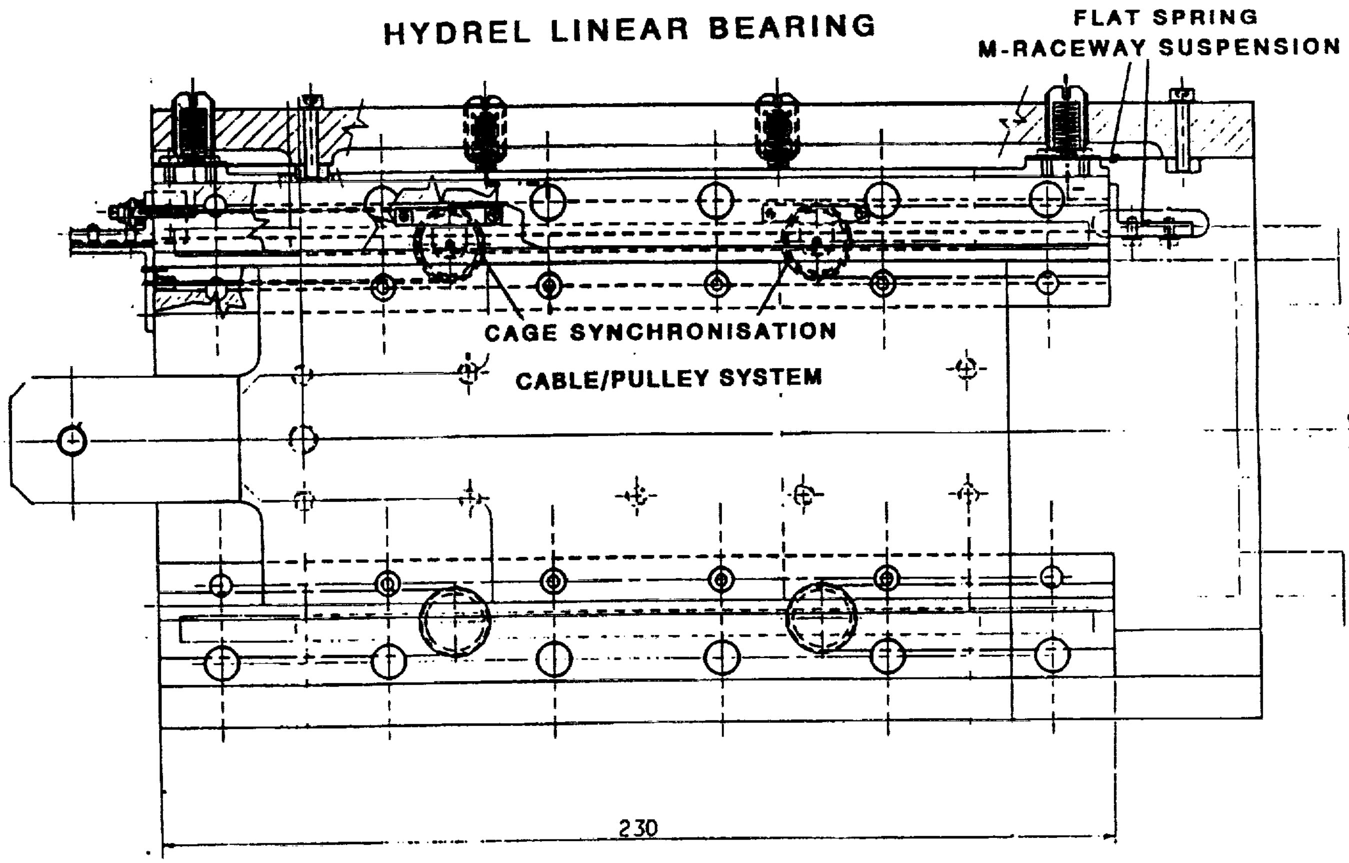
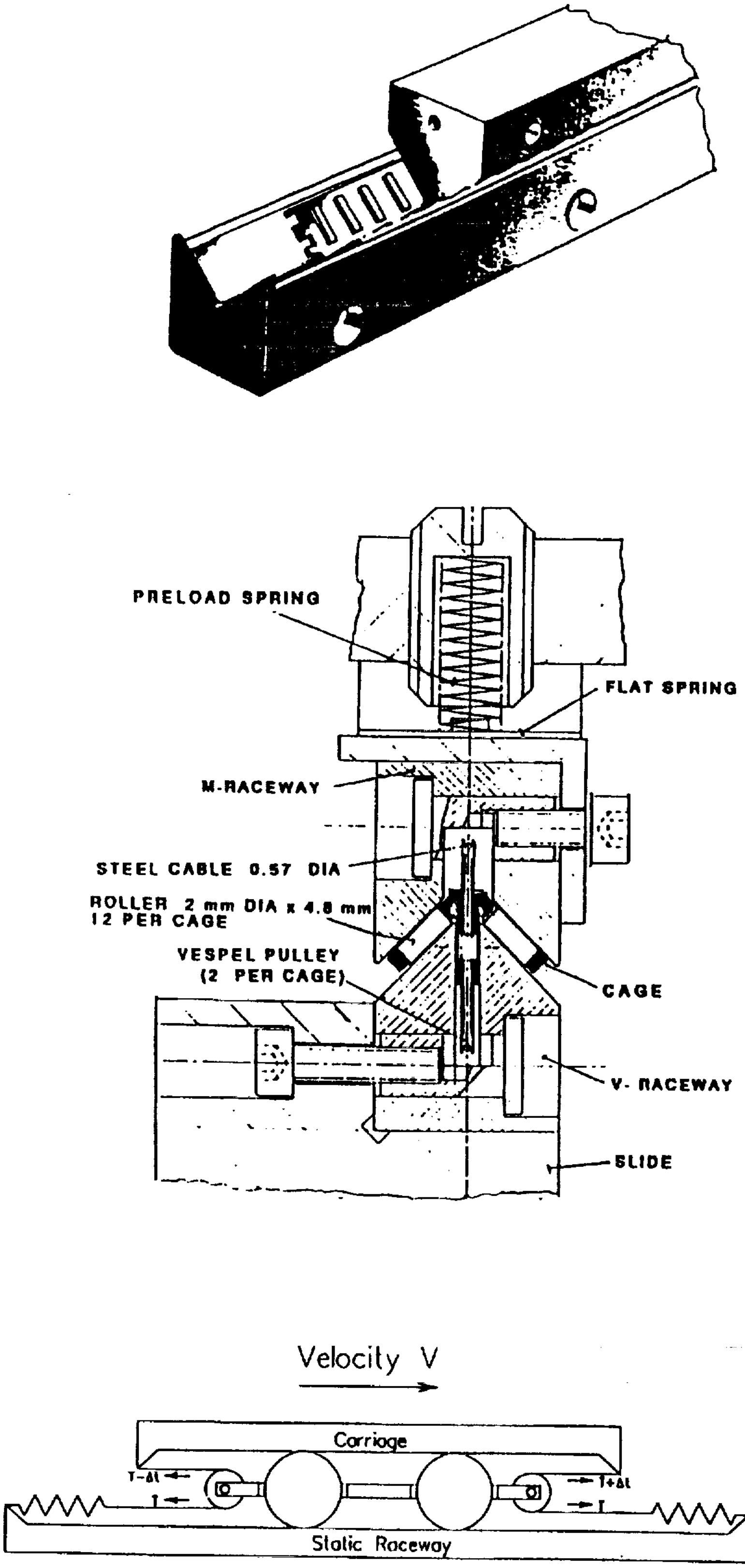


FIG.1 THE LINEAR ROLLER-BEARING GUIDED SLIDE

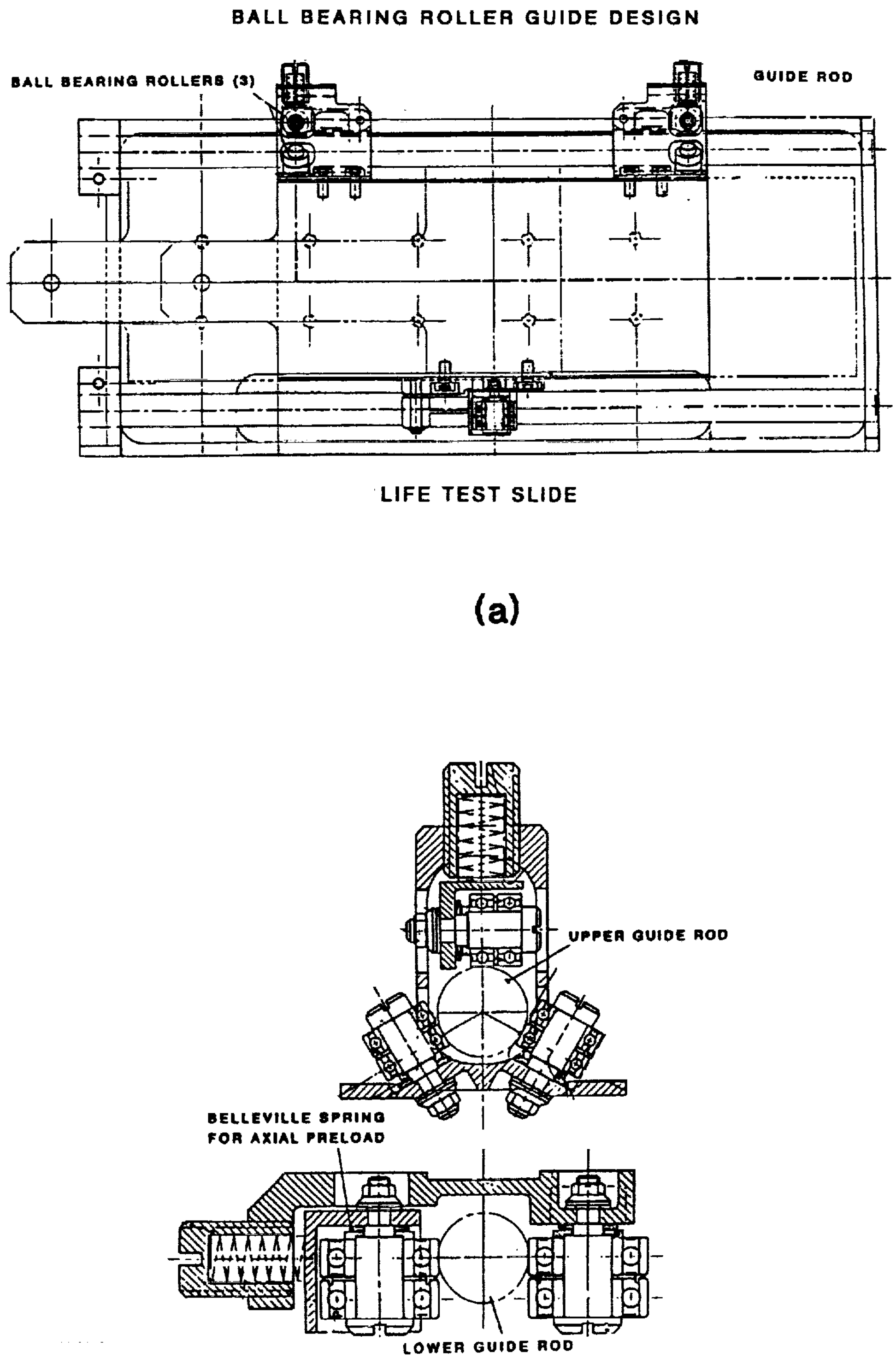


(a) HYDREL ROLLER BEARING

(b) X-SECTION OF HYDREL ROLLER BEARING SHOWING PRELOADING SYSTEM

(c) SHOWING PRINCIPLE OF PULLEY/CABLE SYSTEM FOR CONTROLLING CAGE SPEED

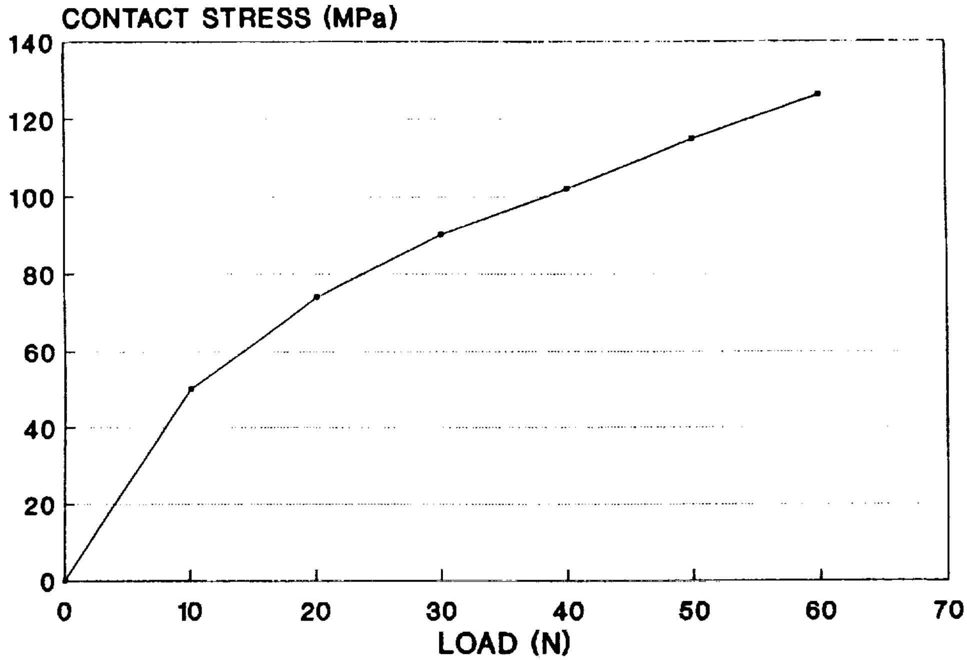
FIG.2



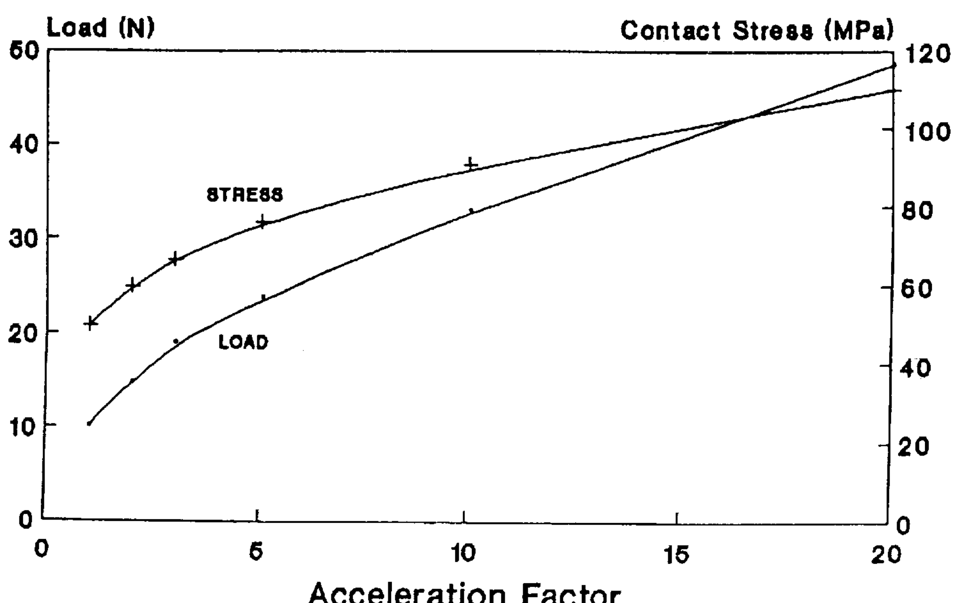
(b)

FIG.3 TWO VIEWS OF THE BALL-BEARING ROLLER GUIDE SYSTEM

FIG.4 HYDREL ROLLER BEARING
COMPUTED MEAN CONTACT STRESS PER ROLLER



Film Life Acceleration Factor
Hydrel Roller Bearing



(relative to design load of 10N)

FIG.5

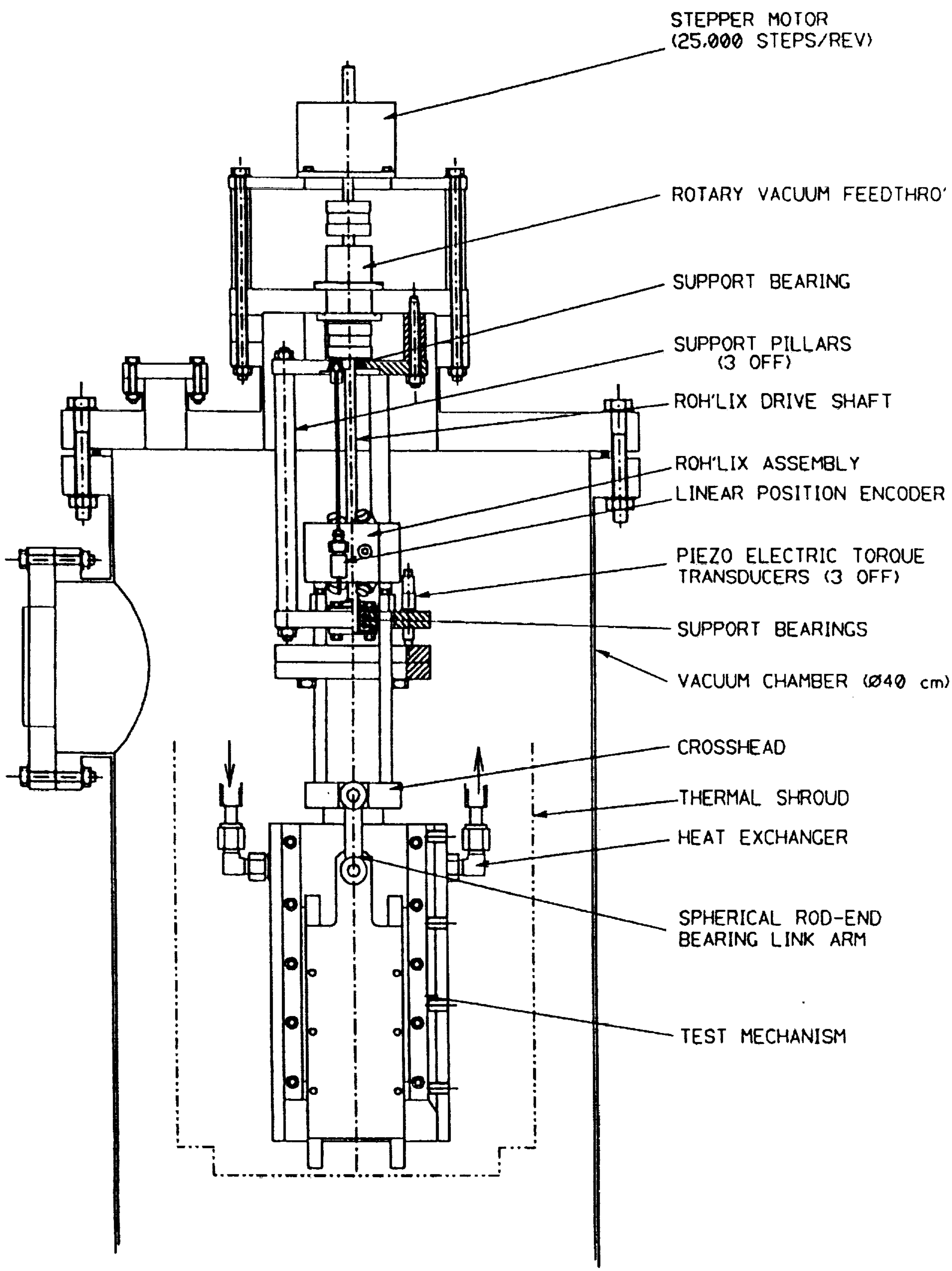


FIG.6 RIG FOR TESTING HYDREL BEARINGS UNDER THERMAL VACUUM CONDITIONS

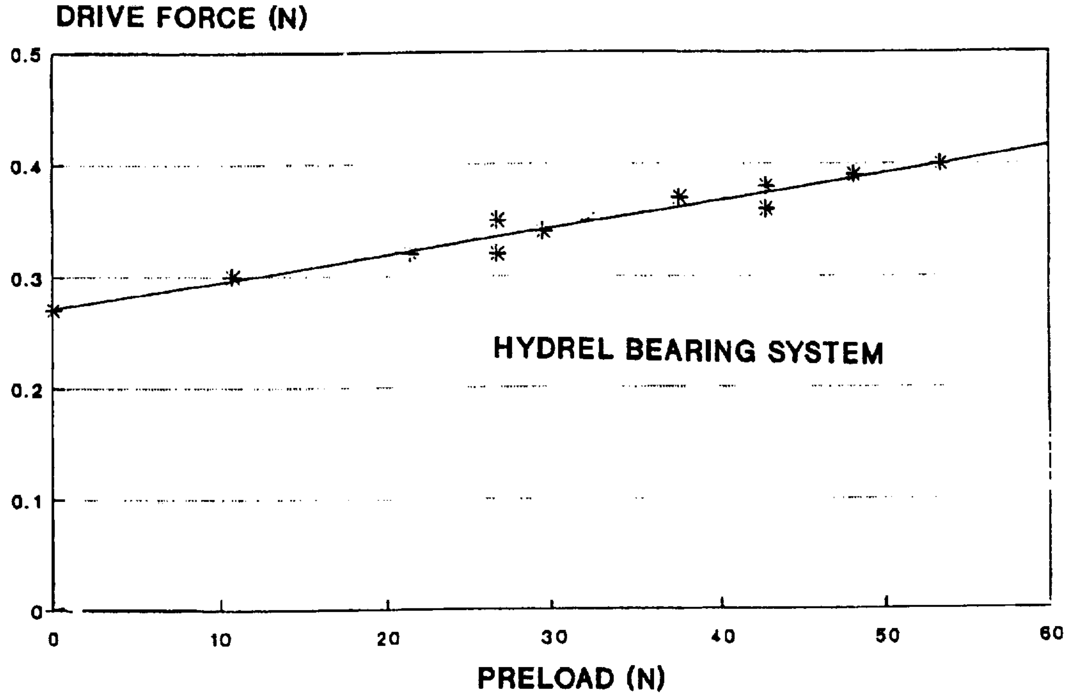


FIG.7 MEAN DRIVE FORCE VS PRELOAD (IN NITROGEN)

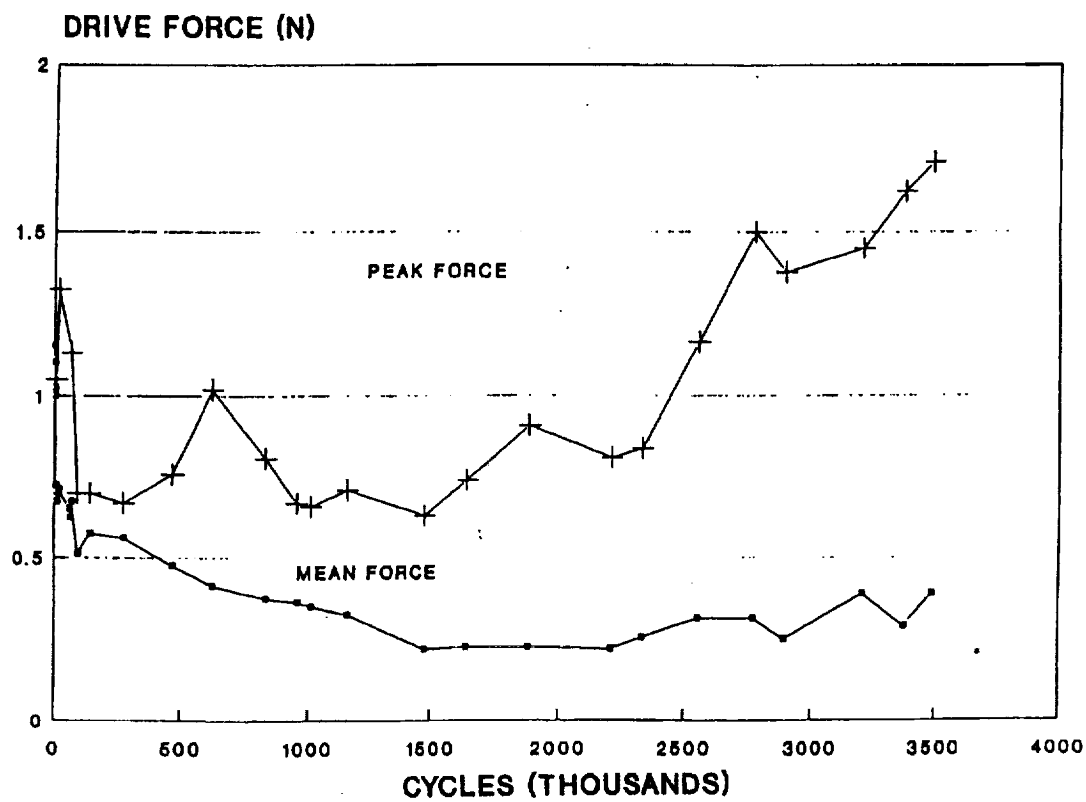


FIG.8 DRIVE FORCE VS. CYCLES OF HYDREL BEARING SYSTEM

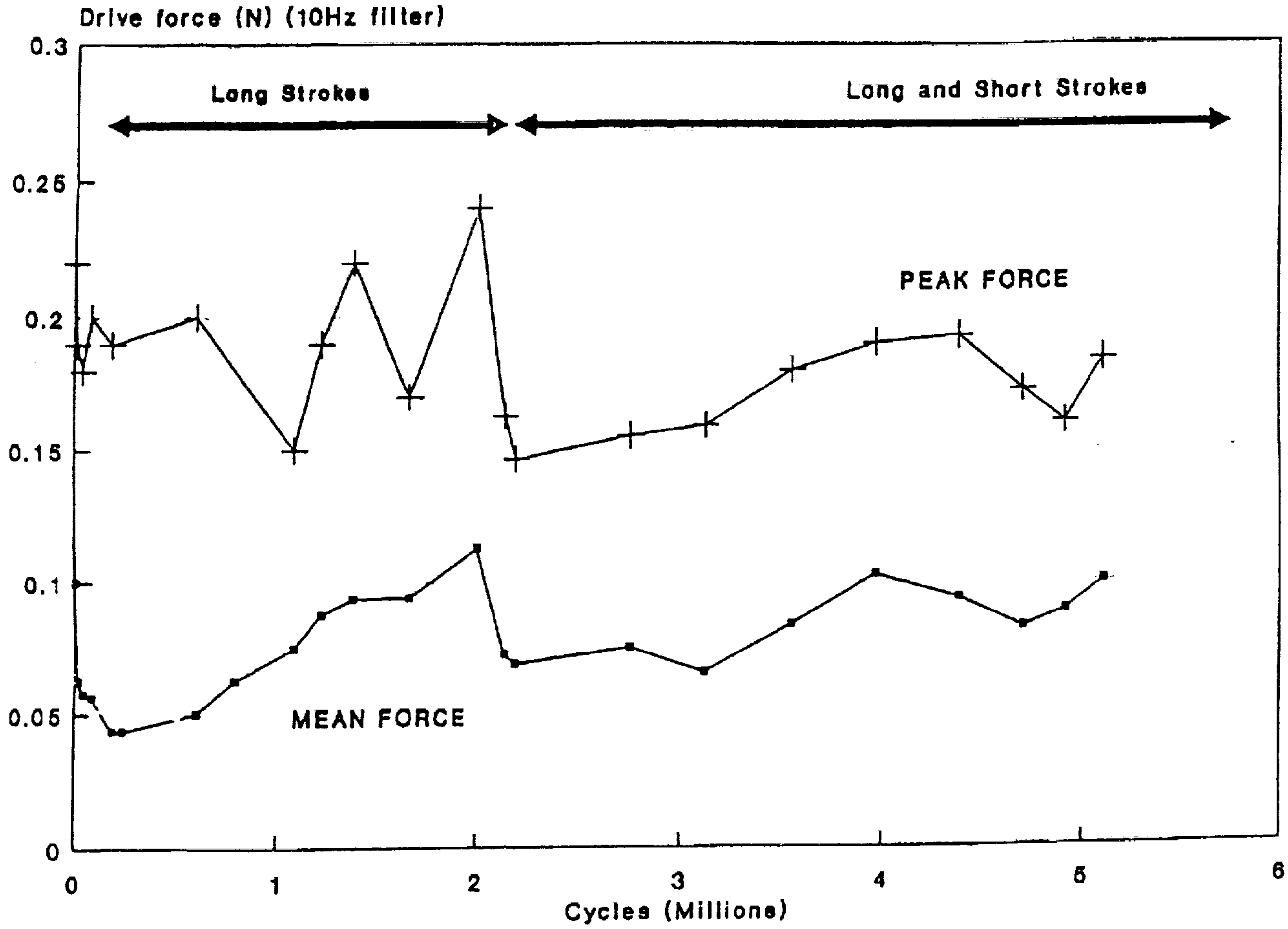


FIG.9 Roller Guide System - MIPAS
Drive force vs. no. of cycles