

ACTIVE CONTROL OF BEARING PRELOAD USING PIEZOELECTRIC TRANSLATORS

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ABSTRACT

In many spacecraft applications, mechanisms are required to perform precision pointing operations or to sometimes dither about or track a moving object. These mechanisms perform in a predictable and repeatable manner in benign temperature environments. Severe thermal gradients experienced in actual space applications however, cause assemblies to expand and contract around their bearings. This results in unpredictable changes in bearing preload, and hence bearing friction. This becomes a limitation for servos controlling pointing accuracy. Likewise, uncontrollable vibrations may couple into fixed preload (hence, fixed stiffness) mechanisms and limit pointing accuracy. Consequently, a complex problem we face today is how to design mechanisms that remain insensitive to changing thermal and vibrational spacecraft environments. Research presented in this paper involves the simplified modeling and test results of an actuator module that used piezoelectrically preload-controlled bearings. The feasibility of actively controlling bearing preload was demonstrated during this study. Because bearing friction is related to preload, a thermally active system designed with aluminum components and a 440C bearing, was friction tested at temperatures ranging from 0 to 70°C (32 to 158°F). Effectiveness of the translators were demonstrated by mapping a controllable friction range throughout tested temperatures. We learned that constant preload for this system could be maintained over an approximate 44°C (79°F) temperature span. From testing, it was also discovered that at the more deviate temperatures, expansions were so large that radial clearances were taken up and the duplex bearing became radially preloaded. Thus, active control of bearing preload is feasible but may be limited by inherent geometry constraints and materials used in the system.

INTRODUCTION

Mechanisms that can adapt to their environment are becoming more and more of a spacecraft necessity. High demands for gimbal performance and life are driven by future mission requirements as evidenced in Reference [1]**. No longer are passive systems able to meet requirements of the 1990's. Extremely tight pointing requirements dictate that actuators must not only operate reliably and predictably, but must be able to compensate

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**Numbers in square brackets refer to references at the end of this paper.

for environmental effects. This becomes particularly apparent when cryogenic payloads must be gimballed and actuator assemblies are operated at extremely low temperatures. Coefficients of thermal expansion for these housings and shafts generally differ to some extent with their bearings. The temperature changing precision fits can have a dramatic effect on the bearing preload, which will always manifest itself as a friction torque increase.

Rotary actuators are generally modeled as single degree-of-freedom (DOF) joints. They can be better represented, in a dynamic sense, as six DOF bodies. Assuming rigid housings and shafts, a duplex bearing will dictate constitutive relationships for at least five of the six DOF. Provided that sensible strain energy in the actuator is focused in the bearing, a robotic manipulator or gimbal assembly can be designed such that damping or vibration isolation is achievable in a variety of directions. Spacecraft mechanisms and their payloads are often susceptible to low frequency spacecraft vibrations during launch and in orbit. Typically we think of attenuating the disturbance functions by increasing damping or isolation of the driven payload. We may also "tune" the actuator in a modal stiffness sense so that it and its payload are uncoupled from high energy modes. This becomes a complex problem because designing to avoid modes that may occur in orbit may yield an actuator stiffness that is parametrically excited during launch. Actuators specifically engineered for one dynamic environment or operation may not be well suited for other environments. It would be a technology breakthrough if an actuator could exhibit adaptive compliance. Thus, an actuator would be actively stiffness-tuned to uncouple it and its payload from disturbances, or alternatively be used to damp-out vibrations entirely by acting as a soft, energy-absorbing joint.

The key parameter that makes for this adaptable actuator is controllable preload in a duplex bearing. This single parameter plays a crucial role in establishing frictional and transverse stiffness behavior. Although preload is controllable by using piezoelectric devices, it is unfortunately not directly observable and must be inferred by friction or stiffness measurements.

Piezoelectric materials exhibit crystalline lattice distortion when subject to an electric field. This distortion will manifest itself as an apparent strain that is repeatable and controllable. Piezoelectric wafers, arranged in a bonded stack, form a "pusher" or "translator." These translators have been employed in numerous applications [2,3] in commercial industry and at least one conceptual design in a bearing has been patented [4]. These translators can provide useful functions since expansion can be controlled precisely and significantly large forces can be generated. Thus, with bearing preload being extremely sensitive to raceway axial stickout, piezoelectrics are ideal devices for actively regulating this bearing parameter.

BEARING SENSITIVITIES

During the design of the active preload control module, a bearing parametric sensitivity study was performed. Many design cases were examined using software based upon theoretical relations presented in [5] and [6]. Relationships studied were temperature effect on preload, preload effect on friction, and preload effect on transverse stiffness. Observations were made on friction and stiffness relationships to ball size, raceway curvature, and contact angle.

Differential expansion between a bearing and its mounting assembly will always act to increase preload. Provided there is no significant change in axial dimensions, increasing temperature would, for example, cause a 440C bearing to expand faster than a titanium housing, thus increasing preload. Decreasing temperature would cause a 440C bearing to shrink faster than a titanium shaft, also increasing preload. For a typical 79 mm (3.125 in) bore, 440C stainless-steel thin race bearing mounted in a titanium housing/shaft, a change of 140°C (252°F) will double the preload from 34 to 68 N (75 to 150 lbs) as shown in Figure 1.

Increasing preload will always increase friction torque proportionally. Harris in [6] gives Palmgren's statistically based relationship between preload and friction.

$$M_i = f_1 F_B d_m \quad (1)$$

Where for a duplex pair,

$$f_1 = z \left(\frac{F_s}{C_s} \right)^y \quad (2)$$

and

$z = .001$, a constant for 30° contact angle

$y = .33$, a constant

F_s = static equivalent load

C_s = basic static load rating

F_B = preload

d_m = diametral pitch

For the 79 mm (3.125 in) bore bearing mentioned earlier, Figure 2 shows its theoretical friction-preload relation. Note that the friction torque more than doubles as preload is increased from 34 to 68 N.

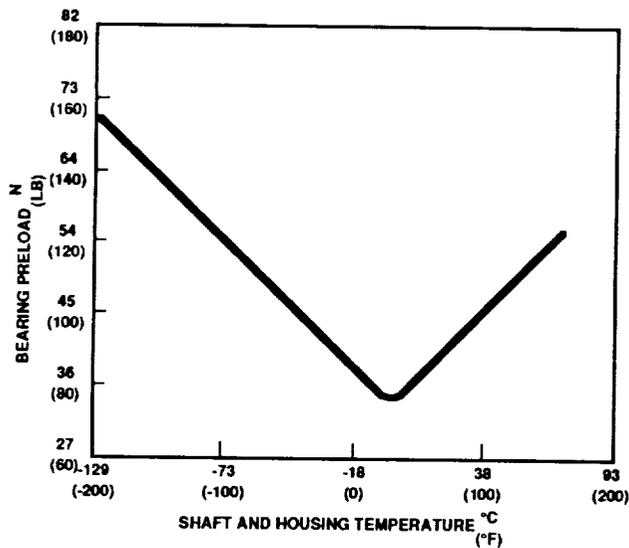


Figure 1. Theoretical Bearing Preload Versus Shaft and Housing Temperature

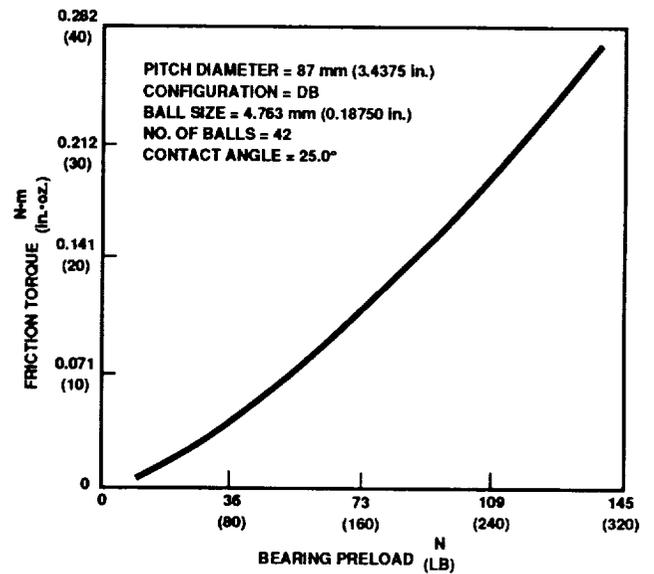


Figure 2. Bearing Friction from Preload

Preloading effects on bearing transverse stiffness is a complex phenomenon. Figure 3 shows some characteristic curves as theoretical stiffness changes with applied moment. Moment stiffnesses are typically the most crucial on electromechanical actuators since these assemblies tend to always cantilever their payloads and/or are subjected to bending type loads. For smaller moments, stiffness is highly dominated by preload. Conversely, preloading diminishes as external loads begin to increase in size and mask the small preload. Data in Figure 3 is relevant to a duplex pair with a 89 mm (3.5 in) axial spread between the races. For an adjacent duplex pair, doubling the preload can often times result in doubling the stiffness.

Ball size, raceway curvature, and contact angle all exhibit influences over stiffness as shown in Figures 4, 5, and 6. Minimum ball sizes, as a rule of thumb, tend to maximize stiffness but minimize load carry capability. Friction tends to decrease for bearings with larger balls primarily because the number of sliding interfaces against the retainer are less. Raceway curvatures approaching .500 will be stiffer since the raceway wrap-around the ball is increasing and the ball footprint or contact area increases. A larger footprint with decreased race curvature will cause more sliding at the race/ball contact zone and hence, friction would be expected to increase. Contact angle increasing for a double-back (DB) pair allows for better moment carrying capability as shown in Figure 6. Contact angle effects on friction behavior do not enter into the friction modeling function and are expected to be very slight for slow rotating systems.

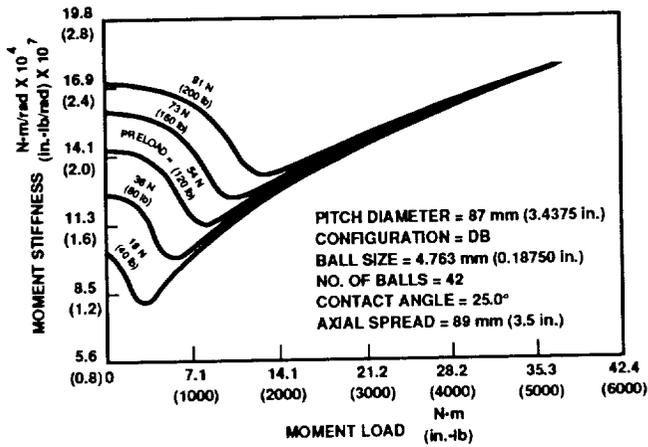


Figure 3. Effect of Preload on Bearing Moment Stiffness

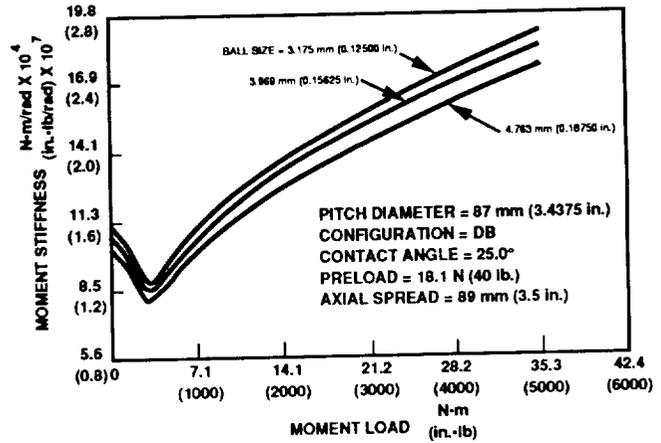


Figure 4. Effect of Ball Size on Bearing Moment Stiffness

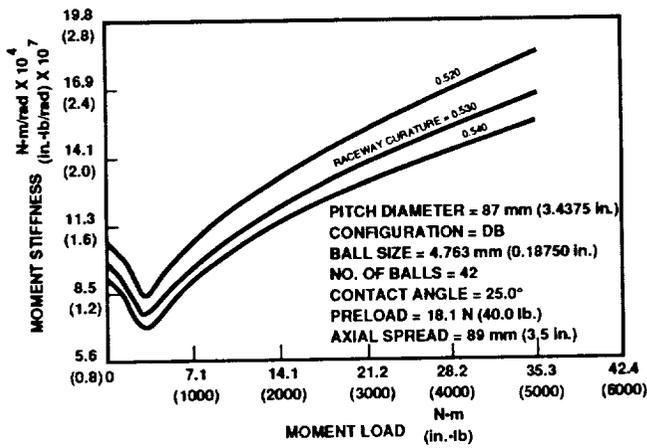


Figure 5. Effect of Raceway Curvatures on Bearing Moments Stiffness

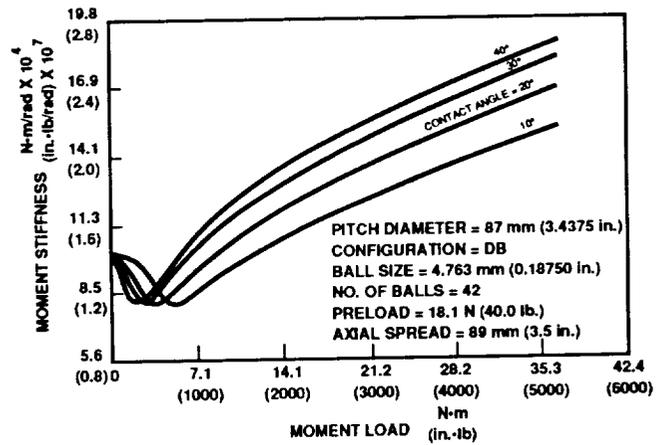


Figure 6. Effect of Contact Angle on Bearing Moment Stiffness

We also know that typical free strains for these translators are on the order of .1%

$$\epsilon = \frac{\Delta L}{L} = .001 \quad (5)$$

Combining (3) (4) and (5), we can solve for the free state translator length.

$$L = \frac{\delta_n}{\left(.001 - \frac{F}{AE} \right)} \quad (6)$$

Using this relation and referencing Figures 9 and 10 which show specific data for our module, we can analyze our preloading system. It was estimated that we needed .0216 mm (.00085 in) of expansion, against a load of 76 N (167 lbs). A translator of approximately 51 mm long by 1 cm diameter was then sized. For six translators on one side of the bearing, a net preload force of over 454 N (1000 lbs) was possible.

Deflection of the housing and mounting assemblies ideally should be at least ten times stiffer than the bearing being controlled. This insures that these extraneous compliances are kept to below 10% of the total system deformation. This was designed into the module housing with some additional margin because of uncertainties in the translator stiffnesses.

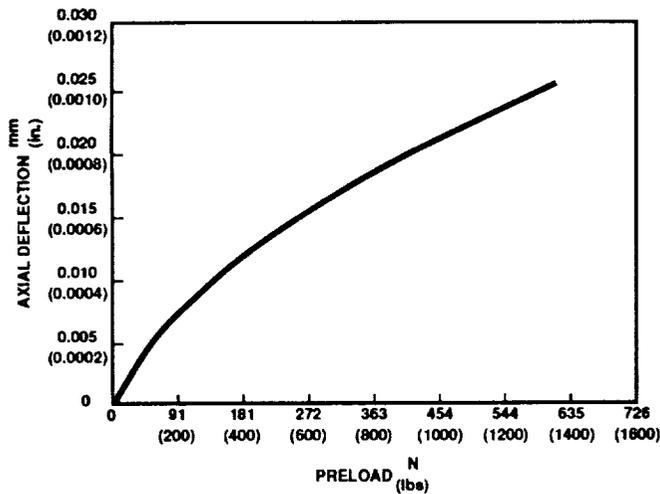


Figure 9. Axial Deflection of Bearing Race Due to Preloading

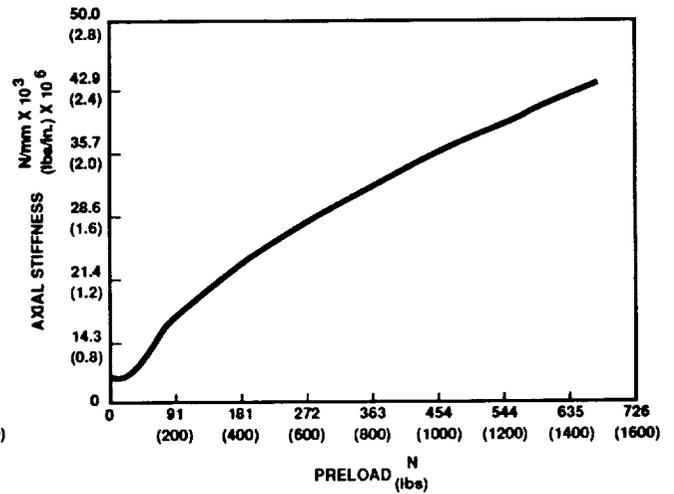


Figure 10. Axial Stiffness of Bearing Race from Preload

To test the effectiveness of the translators, expansion measurements were made as a function of voltage. Maximum applied voltage for these devices was 1000 volts per the manufacturers recommendations. By design, the translators averaged .0510 mm of free state expansion. After building and testing the module, the axial deflection of the outer race against bearing preload was measured and averaged approximately .0203 mm (.0008 in) as predicted. Voltage-expansion curves for the translators are shown in Figure 11.

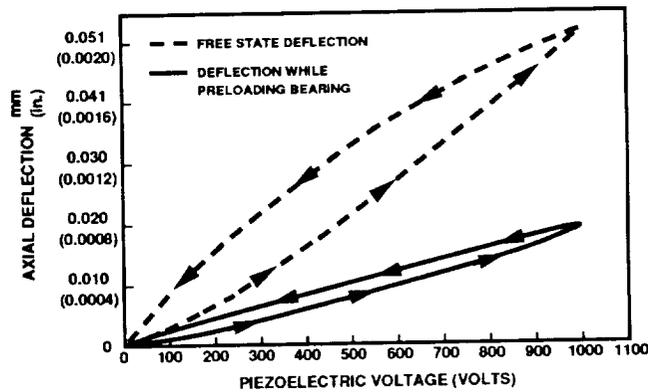


Figure 11. Axial Deflection of Piezoelectric Translators in Free State and Mounted

TEST SETUP

The intent of this project was to demonstrate that a constant friction device could be built that relied on piezoelectric translators for controlling preload. The bearing module test setup, shown in Figures 12 and 13, was subjected to temperatures ranging from 0 to 70°C while friction was being monitored (and controlled). Dry lubed bearings were used since wet lubricants would introduce error due to viscosity effects. An input drive motor rotated an internal shaft at about 1 RPM while frictional reaction torques were measured by a dynamometer attached to the module housing.

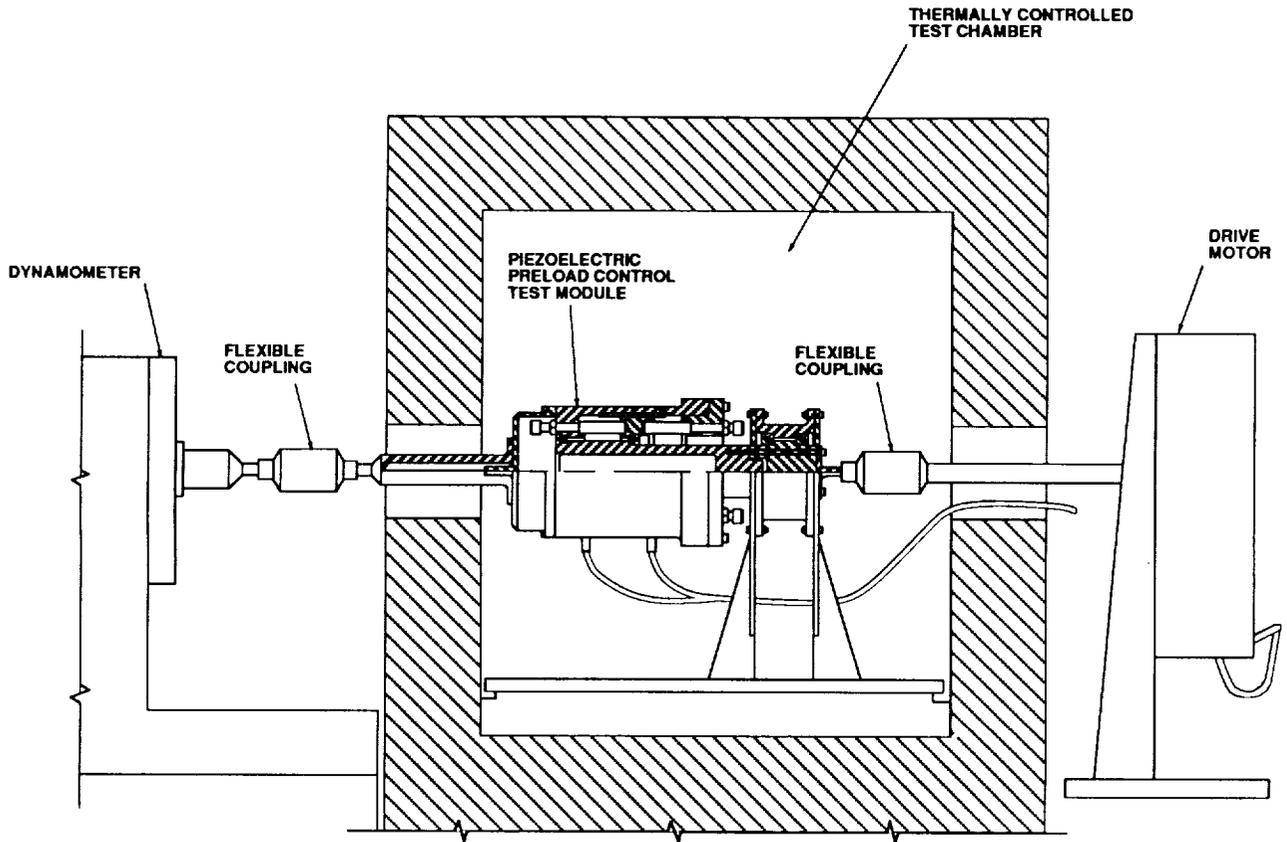


Figure 12. Test Module Setup

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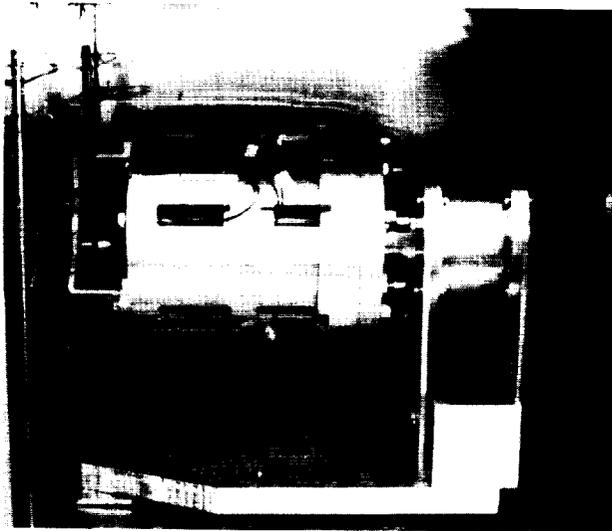


Figure 13. Bearing Preload Module Inside Environmental Chamber

The preload drive module consisted of a 6061 aluminum shaft and housing to produce a worst case thermal expansion mismatched system. Bearings consisted of a 79 mm (3.125 in) bore duplex pair, made from 440C stainless steel spaced 89 mm (3.5 in) apart. Bearings were mounted with line-to-line fits on both their bore and outside diameters. For this bearing module, unlike typical spacecraft products, little attention was paid to packaging, volume, or weight constraints to save on time and effort. Particular effort was focused on minimizing piezoelectric costs while ensuring that the translators would perform as expected.

EXPERIMENTAL RESULTS

As a first task, the module sensitivity to temperature, independent of compensation, was investigated. Dummy translators were made from aluminum rod stock and were used to replace the piezoelectric translators. Friction torque measurements at a variety of temperatures showed extreme inherent temperature sensitivity as shown in Figure 14. This data was in general agreement with theoretical results except that theoretical frictional magnitudes were much higher. An explanation for this follows in later discussion.

The compensated system behaved somewhat as expected showing very good control of friction (hence preload) throughout a 44°C temperature range. As shown in Figure 15, a friction-controlled envelope was measured as a function of temperature and translator energization. Controlling friction torque became limited by several unforeseen phenomena.

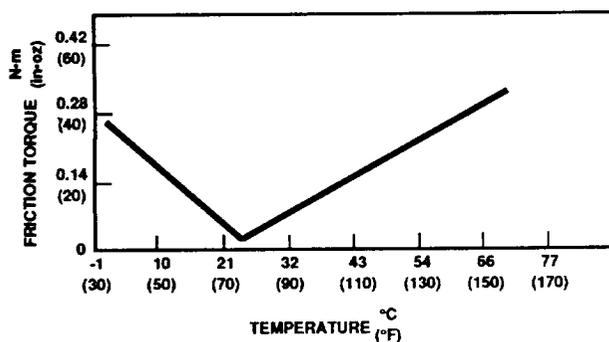


Figure 14. Inherent Friction Sensitivity of Tester Using Dummy Translators

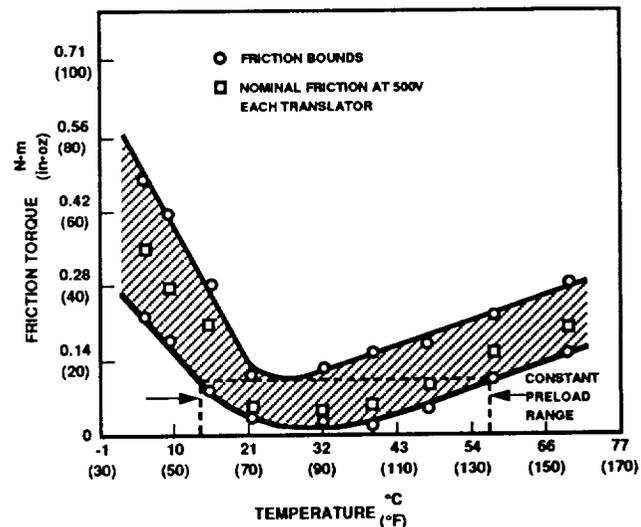


Figure 15. Controllable Friction Envelope for 440C Bearing and 6061 Aluminum Housing and Shaft

First, because of the severe mismatch of thermal expansion between aluminum and 440C stainless steel, radial interferences on the bearing fits at slightly deviate temperatures became overwhelming. Elastic deformations were so large that the thin race bearing became "radially" preloaded. Any attempts to slide races axially with the translators only served to increase preload. This behavior was thought to be responsible for the bounds of Figure 15.

Second, at room temperature, maximum friction torque measured was only .113 N.m (16 in.oz). If we look at Figure 2, based on the Palmgren friction relation, only 73 N (160 lbs) of preload was reached. It is believed much higher preloads were achieved however. Figure 11 shows the stickout was controlled properly with axial race displacement of approximately .0203 mm (.0008 in). Thus, the 454 N (1000 lbs) of preload was most likely achieved. Palmgren's frictional relationship is based on statistical measurements of many bearings, but without reference to lubricant. From various characterization tests, we have found the Palmgren relationship of equation (1) gives a good indication of relative bearing frictional behavior, but a rather poor indication of absolute frictional behavior. Thus, the theoretical friction relationship is very conservative.

Third, the piezoelectric devices themselves are not immune to temperature effects. Piezoelectrics have a large coefficient of thermal expansion of 5×10^{-6} (m/m)/°C. This most likely contributed to some of the preloading and must be taken into account for designs with severe temperature variation.

Lastly, hysteresis in piezoelectrics is usually present to about 20% of the maximum expansion in the free state. For this design, the translators pushing against each other cut this hysteresis in half. It was also discovered however, that as shown in Figure 11, when only energizing one side of the translators, measured hysteresis was not as large as expected. This suggests that a future design would work satisfactorily having only translators that preload, then allow the bearing axial force to "unload" the race.

SUMMARY AND CONCLUSIONS

The feasibility of actively controlling bearing preload was demonstrated by fabricating and testing a piezoelectrically controlled bearing module. Because bearing friction is related to preload, a thermally active system, designed with aluminum components and a 440C bearing, was friction tested at temperatures ranging from 0 to 70°C (32 to 158°F). Effectiveness of the translators was demonstrated by mapping a controllable friction range throughout tested temperatures. We learned that constant preload for this system could be maintained over a temperature span of approximately 44°C (79°F).

From testing, it was also discovered that at the more deviate temperatures, expansions were so large that radial clearances were taken up and the duplex bearing became radially preloaded. Thus, active control of bearing preload is feasible but may be limited by inherent geometry constraints and materials used in the system.

Although not tested, we learned by analysis that bearing moment stiffness corresponding to the controllable preload could have more than doubled. With moment stiffness being the most crucial for spacecraft applications, a large part of this stiffness for the test module was provided by the bearing axial spread. Thus, the inherent design desensitized the moment stiffness to preload, but still showed that stiffness could be controlled.

Controlling a structural joint for friction and stiffness behavior is both feasible and practical. To meet precision pointing applications of the future this technology will be necessary and will likely be incorporated into future state-of-the-art actuator designs.

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