GIMBAL BEARING DESIGN CONSIDERATIONS AND FRICTION CONTROL

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ABSTRACT

The design considerations of bearing selection, bearing fits, bearing installation and thermal control are discussed for a gimbal with a high stiffness, low friction torque requirement. Tradeoffs between a quad set of small diameter spread apart or a large diameter bearing pair resulted in a cleaner, lighter, stiffer unit with the latter selection.

Bearing fits were designed to eliminate clearances with tolerances of 127×10^{-5} mm (50×10^{-6} in) on the bearing shafts and housings. The problems in metrology are discussed and a preferred technique for measurement of small cross-section bearings described. A technique for installation to assure proper seating of the bearing is offered.

Finally, where transient thermal conditions are involved, a method of controlling bearing friction by active control of bearing temperature gradients including the use of bearing unload test curves is described.

INTRODUCTION

When adequate stiffness of a device requires tight bearing fits but other considerations preclude high bearing stresses and high bearing friction, and on top of that there exists high launch loads, a dynamic thermal environment and weight and space limitations, how does one handle this delimma? This paper describes an approach that evolved for handling a case of this nature.

BEARING SELECTION

The problem concerned a two axis gimbal, Figure 1, in which the structure was required to have a stiffness of approximately 40 Hz. At the same time the friction had to be kept to a minimum for proper servo operation. The size, weight and power available were also critical. The azimuth bearing was the key element of the gimbal design in controlling both the stiffness and the friction values. Inasmuch as these two characteristics are direct functions of each other, i.e., an increase in preload for greater stiffness results in an increase in friction, the situation was like trying to walk a tightrope between adequate stiffness and acceptable friction torque levels.

Initial design studies looked at quad sets of bearings with the upper and lower pairs spaced apart to provide overturning moment stiffness. See Figure 2. However one large bearing pair with a high contact angle provided a much cleaner, more compact, lighter weight and stiffer design. The bearing selected was 26.7 cm (10.5 inch) I.D. x 29.5 cm (11.6 inch) O.D. x 2.54 cm (1.00 inch) wide with ninety (90) .635 cm (.250 inch) diameter balls in each row. The material is 440C stainless steel and the bearings are assembled in a DB arrangement with a preload of 125 pounds. For the elevation axis, where the distance between bearings is approximately 20 inches, two DF pairs having 12.7 cm (5.0 inch) bores with a .95 cm (.375 inch) crossection were selected.

The bearings were dry film lubricated with molybdenum disulfide and Rulon A. And a Rulon A plus 5% MoS_2 retainer provides a lubricant reservoir.

BEARING FITS

In order to maintain the stiffness throughout the temperature range an interference fit of 760 to 890 x 10^{-5} mm (300 to 350 x 10^{-6} inch) was specified with the shaft of the azimuth bearing and 0 to 127 x 10^{-5} mm (0 to 50 x 10^{-6} inch) clearance fit with the housing, prior to preloading. For the elevation bearing, 127 to 254 x 10^{-5} mm (50 to 100 x 10^{-6} inch) clearance was specified for both inner and outer diameters prior to preloading. Since the I.D. contracts and the O.D. expands approximately 254 x 10^{-5} mm (100 x 10^{-6} inch) when the preload is applied, all clearances are removed, and stiffness is maintained.

In order to achieve the bearing fits specified, it required very accurate measurements of the bearing diameters and then fabrication of the shafts and housings to fit the selected bearings. However, it was soon found that measuring the diameters of small crossection bearings is not a simple thing to do. As an experiment three sets of bearings were sent for measurement to two sources which are reputed to be experts in metrology and the results were compared with the bearing vendor-supplied data. A typical set of results are shown in Table I.

TABLE I

	O.D. mm(in.)	I.D. (-1) mm (in.)	I.D.(-2) mm(in.)
VENDOR	294.63550 (11.599823)	266.69652 (10.499863)	266.69619 (10.499850)
A	294.63218 (11.599692)	266.69937 (10.49975)	266.69683 (10.499875)
B	294.63111 (11.599650)	266.69594 (10.499840)	266.69467 10.499790)
MEAN	294.63294 (11.599722)	266.69728 (10.499893)	266.69589 (10.499838)
MEAN DEVIATION	0.00170 (0.000067)	0.00140 (0.000055)	0.0081 (0.000032)
MAX. DIFFERENCE	0.00439 (0.000273)	0.00343 (0.000135)	0.00216 (0.000085)

In another experiment, one of the sources was asked to measure the same diameter each day for five successive days using the same operator and same instruments. The difference in the results averaged approximately 100 x 10^{-4} mm (40 x 10^{-6} inch) with a total spread of about 315 x 10^{-4} mm (125 x 10^{-6} inch).

It was found that picking up a small crossection bearing is very much like picking up a wet noodle that has been formed into a ring. The diameter will move in the direction in which hand pressure is applied. To overcome this effect, the procedure for measurement which was adopted provides an integrated average of the radius around the entire perimeter. This is done by utilizing a set of gage blocks stacked to the nominal diameter to be measured. This stack is centered on an Indiron table. A two gram force indicator probe is used to plot the location of the ends of the stack on the Indiron Chart with the scale set for 254 x 10^{-6} inch) per division. A circle drawn between the two diametrically opposed low points represents the nominal diameter, see Figure 3. Without changing the setting, the gage blocks are removed and the ring to be measured centered on the Indiron Table. The ring diameter is then run with the two gram probe and plotted on the same chart. The average difference between this plot and the nominal diameter circle is then determined from which the bearing diameter is established.

BEARING INSTALLATION

Bearing installation was carefully supervised to be sure that the bearings were properly oriented and aligned but of most concern was making sure that the bearings were fully seated without applying excessive forces to the bearings. There is no problem seating an unloaded bearing. With the threads lubricated, the torque to be applied to the screws or threaded retainer which will result in a reasonable force to be applied to the bearing can readily be calculated. For a DB bearing pair, the housing can be heated, the bearing inserted and the retainer torqued down. The housing and bearing subassembly is then heated and installed on the shaft. The shaft retainer is then torqued down. On the surface that would appear to be all there is to it. However, due to the difference in temperature between the bearing and its mating part at the time it is clamped by the retainer, the bearing may not be centered or properly preloaded. Therefore, the procedure that was used was to allow the assembly to cool down until it was close to room temperature. The retainers were then loosened to allow the bearings to center themselves and then retightened. For the outer races and housing, this is adequate. However, for the inner races and shaft, with a DB pair, the preload in the bearing tends to separate the races. When the retainer is retightened, it is necessary to overcome the friction between the bearing and shaft as well as the preload and friction in the threads. This does not lend itself readily to calculation since the normal force between bearing and shaft is difficult to determine. The approach that was used was to tighten the retainer in incremental steps and measure the friction torque of the bearing at each step. A running plot of the friction torque versus the tightening torque was made, see Figure 4. When the slope of the plot flattened out so that the friction was essentially the same for three values of tightening torque the bearing was considered to be seated. For the DF bearings, the procedure requires that the shaft be installed first and then the housing added.

THERMAL CONTROL

The above procedures may be adequate for stable temperatures, however transient thermal conditions can create large temperature gradients across the bearings. These can induce conditions of excessive bearing loads, stresses and friction at one extreme and cause the bearings to become unloaded resulting in a loss of stiffness at the other extreme. To overcome this problem, active thermal control of the temperature gradients can be applied. In this form of thermal control, the bulk temperature is permitted to vary, but the temperature difference between the shaft and housing is maintained at a selected value which will result in a moderate positive load on the bearings.

As shown in Figure 5, the effect on friction and stress of the temperature gradient, ΔT, across a bearing is much greater than that of the bulk temperature. In the subject gimbal, thermal analysis indicated that the azimuth bearing shaft always tended to be cooler than the housing. Therefore heaters were required only on the shaft to prevent the bearings from becoming unloaded. With the elevation bearings the reverse was true, therefore heaters were required only on the housings to prevent excessive loads.

To establish the levels of temperature gradient (ΔT) which should be maintained, heaters and temperature sensors were applied to the housings and shafts and bearing unload test curves were plotted, see Figure 6. This is a technique where the temperature gradient across the bearing is varied and the bearing friction torque values (T_F) measured and plotted. ΔT is varied from zero or some negative value, where the shaft is warmer than the housing to increasing positive values. As the housing becomes warmer, T_F will continue to decrease until the bearing becomes unloaded after which T_F will remain constant. From the unload curve a value of ΔT can be selected where the friction is some safe value above unload and the associated stresses can be computed.

A system of thermal control of ΔT was then applied to maintain the selected value. Provision was also made to increase ΔT if for some reason an undesirable increase in friction torque occurred. See Figure 7. Conversely, ΔT may be decreased in the event that the bearing approaches an unloaded condition.

CONCLUSIONS

When the requirements for a gimbal specify a stiff structure and low friction, the bearing will probably be the most critical element. Careful selection, measurement, fitting and installation are essential. Careful analysis of thermal conditions may reveal a need for active thermal control. Control of bearing temperature gradients provides a means for controlling friction.

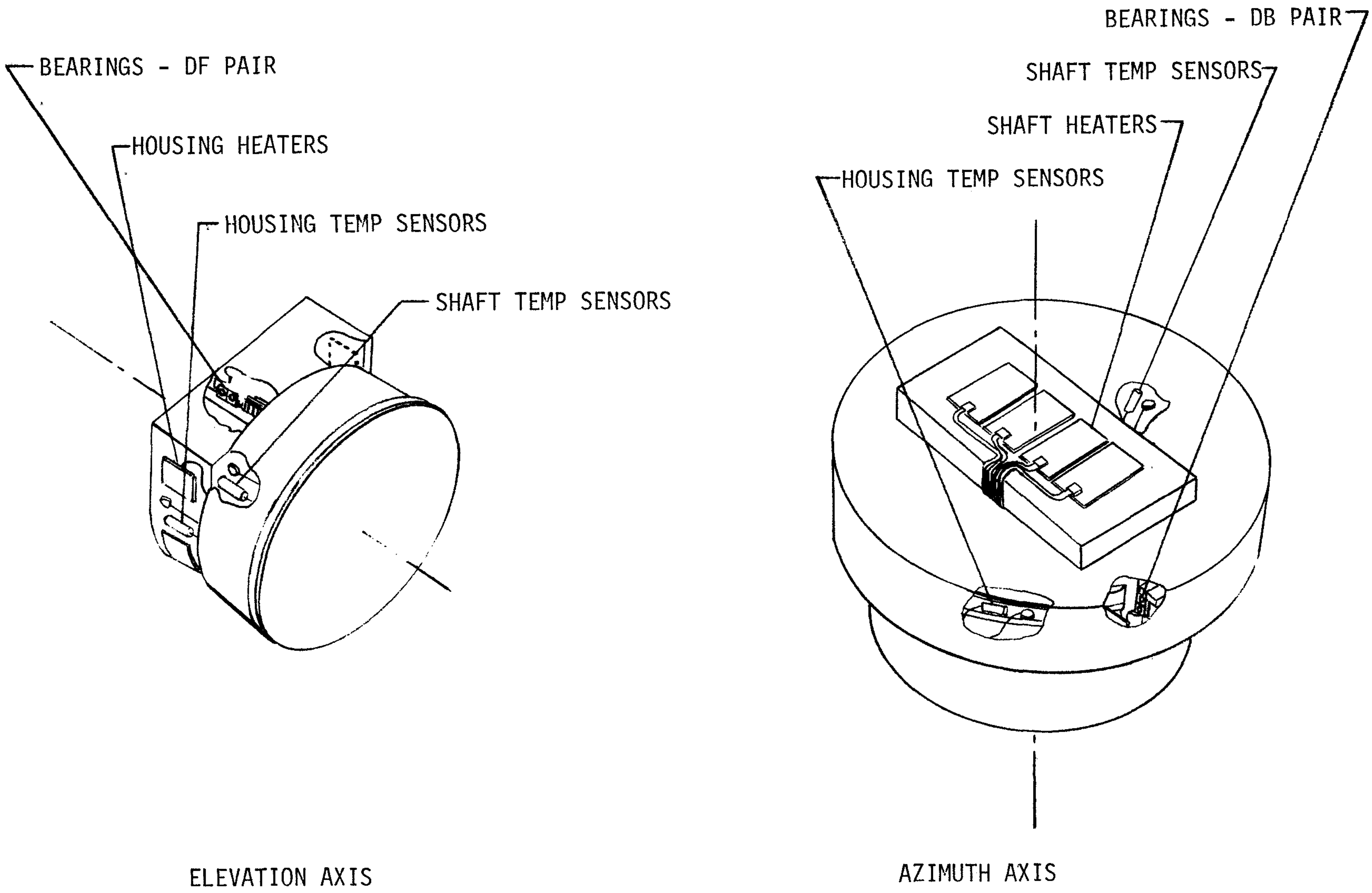

FIGURE 1 - TWO AXIS GIMBAL

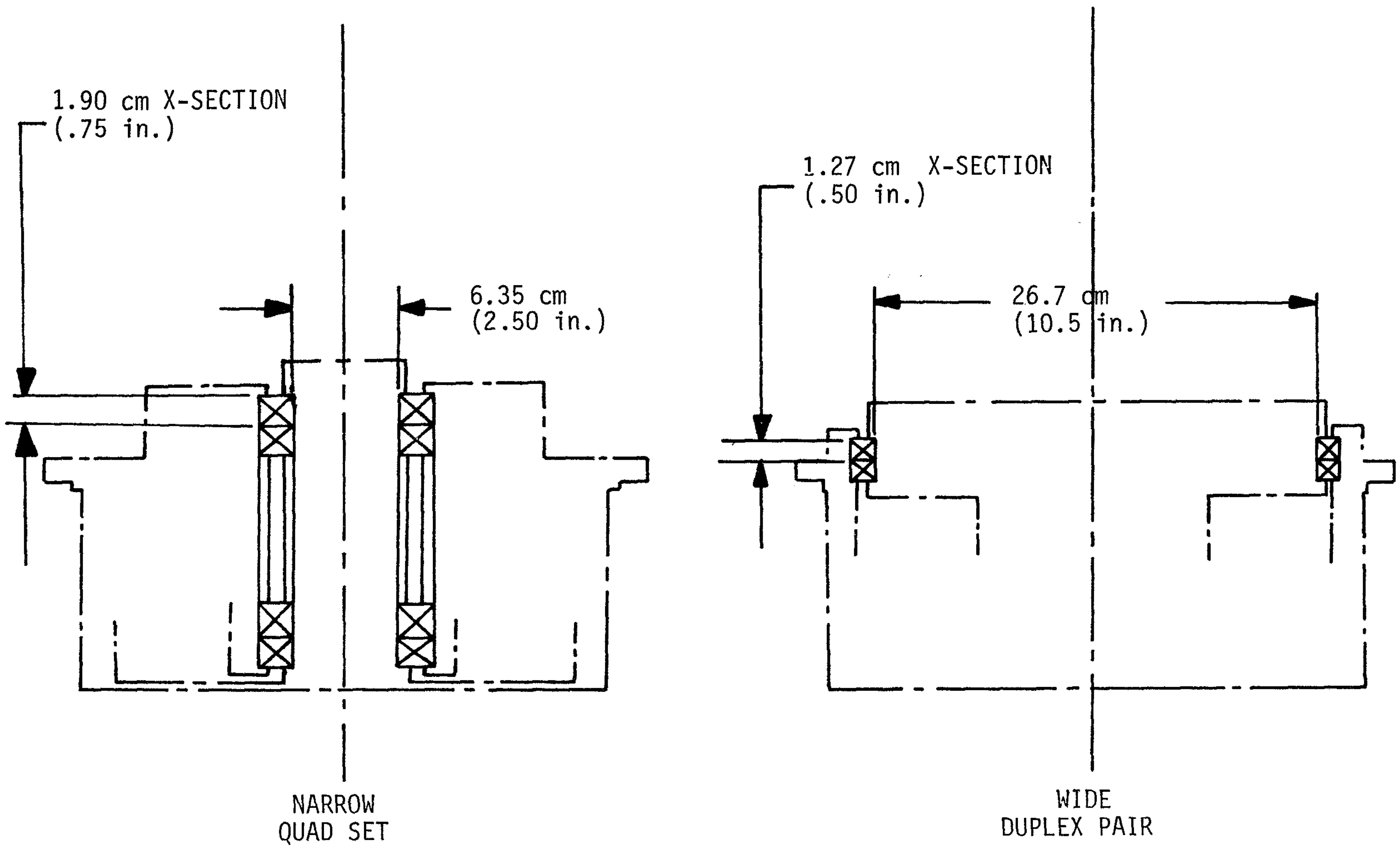

FIGURE 2 - BEARING SELECTION

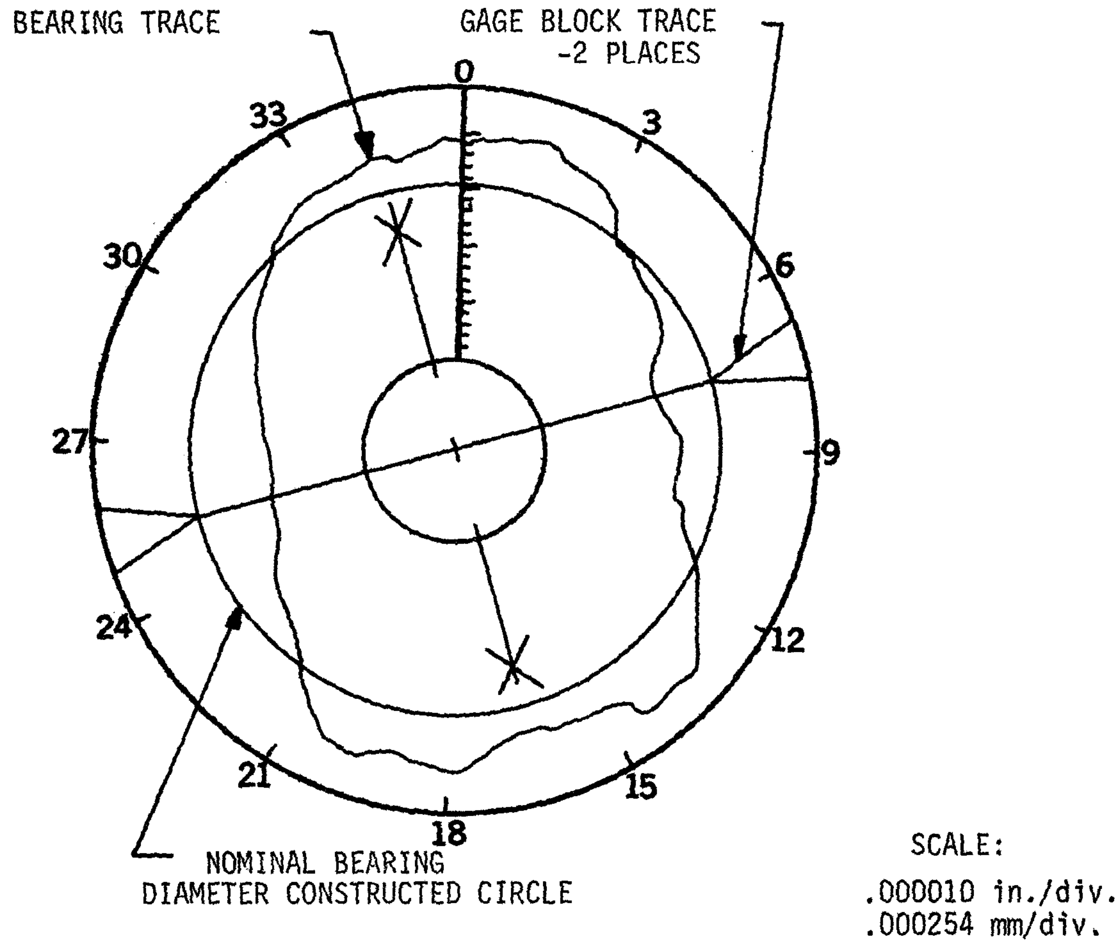

FIGURE 3 - INDIRON PLOT

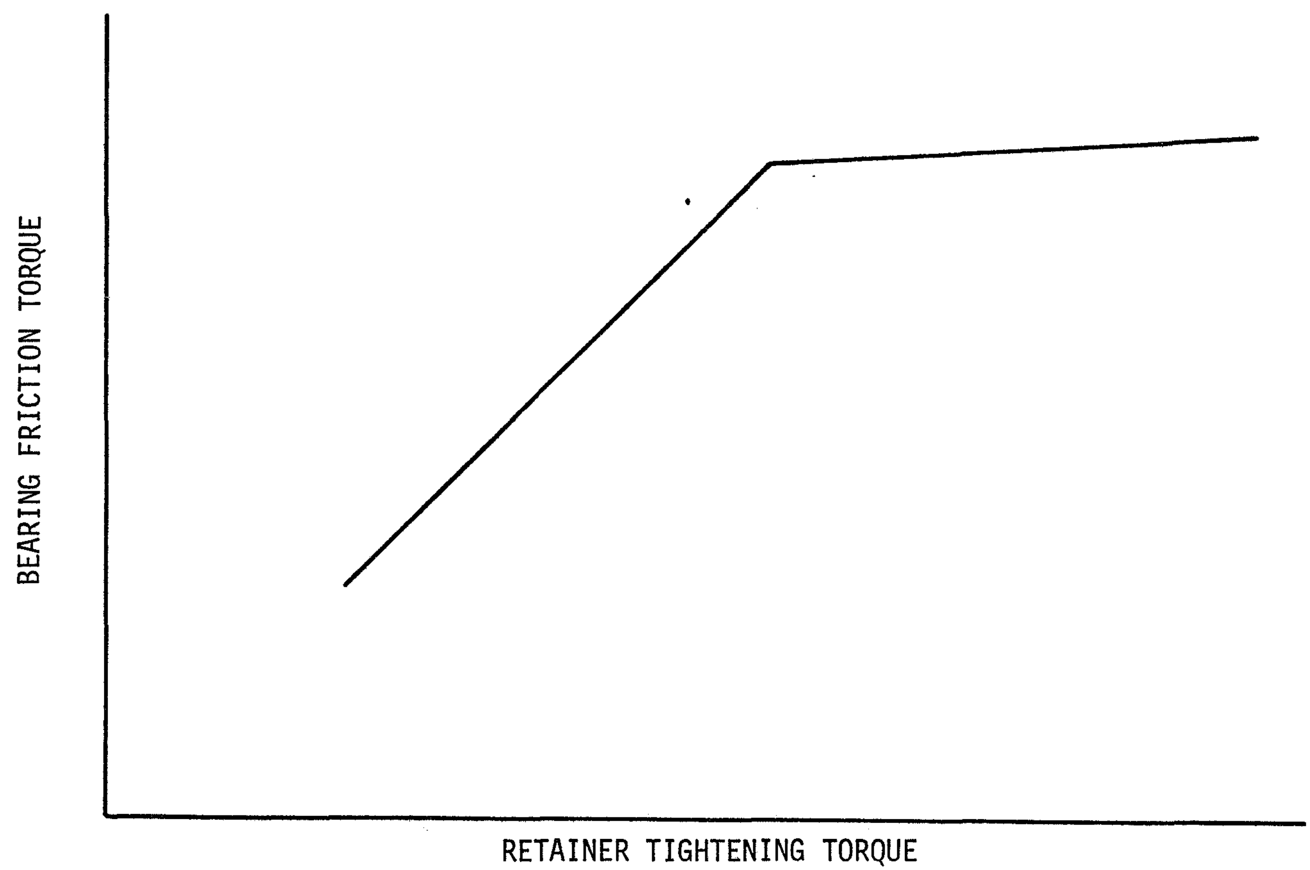

FIGURE 4 - BEARING INSTALLATION PLOT

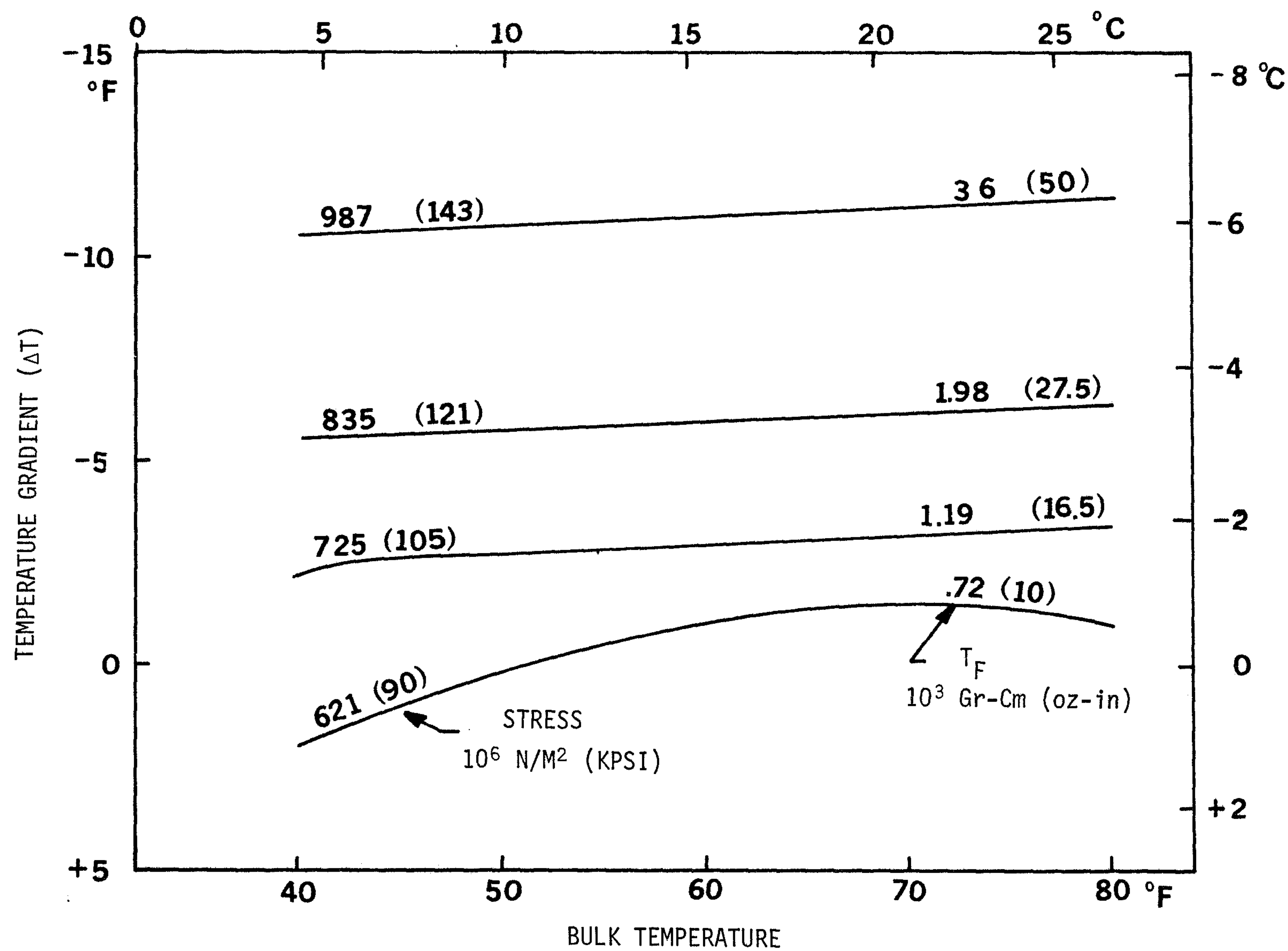

FIGURE 5 - TEMPERATURE EFFECTS (THEORETICAL)

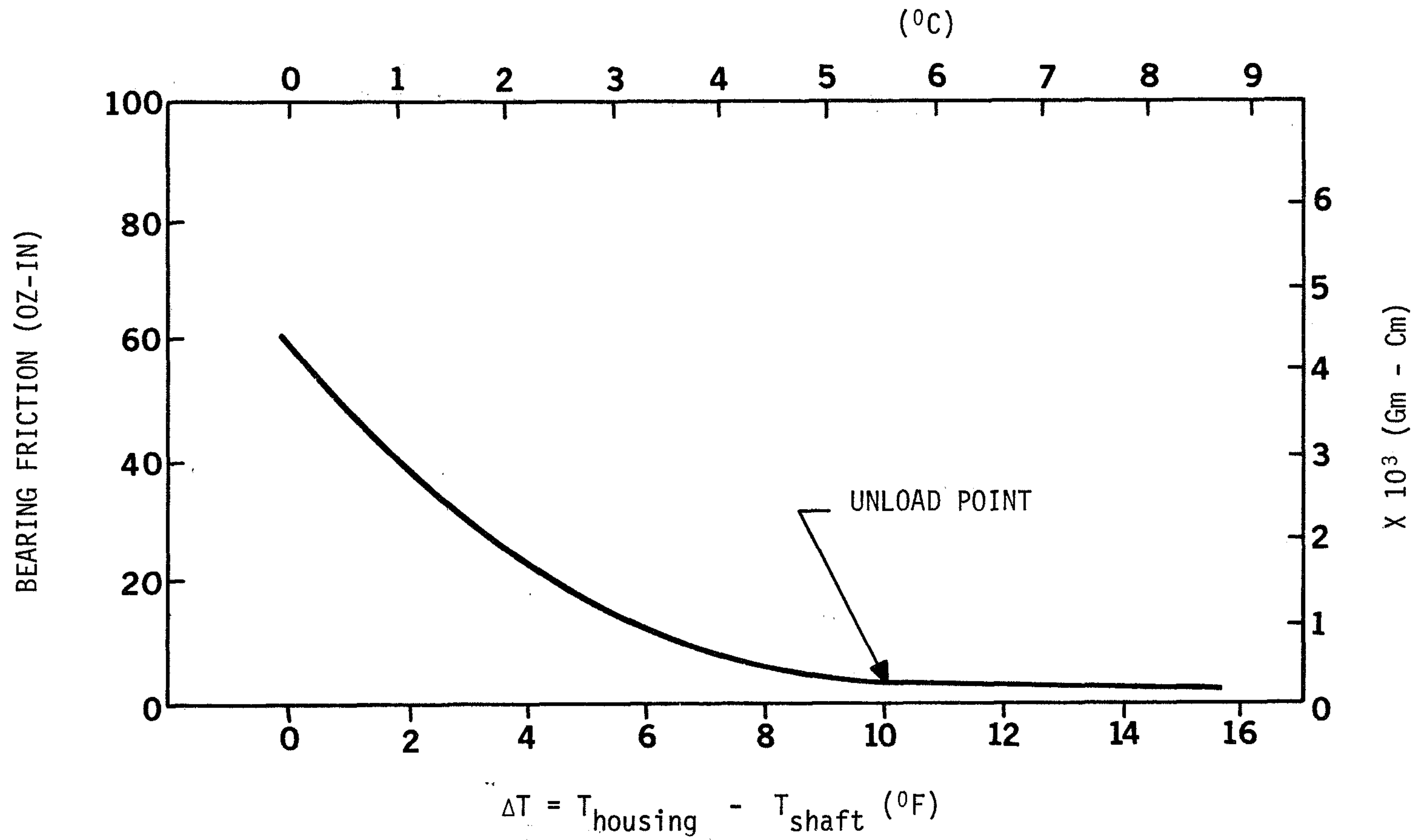

FIGURE 6 - UNLOAD CURVE

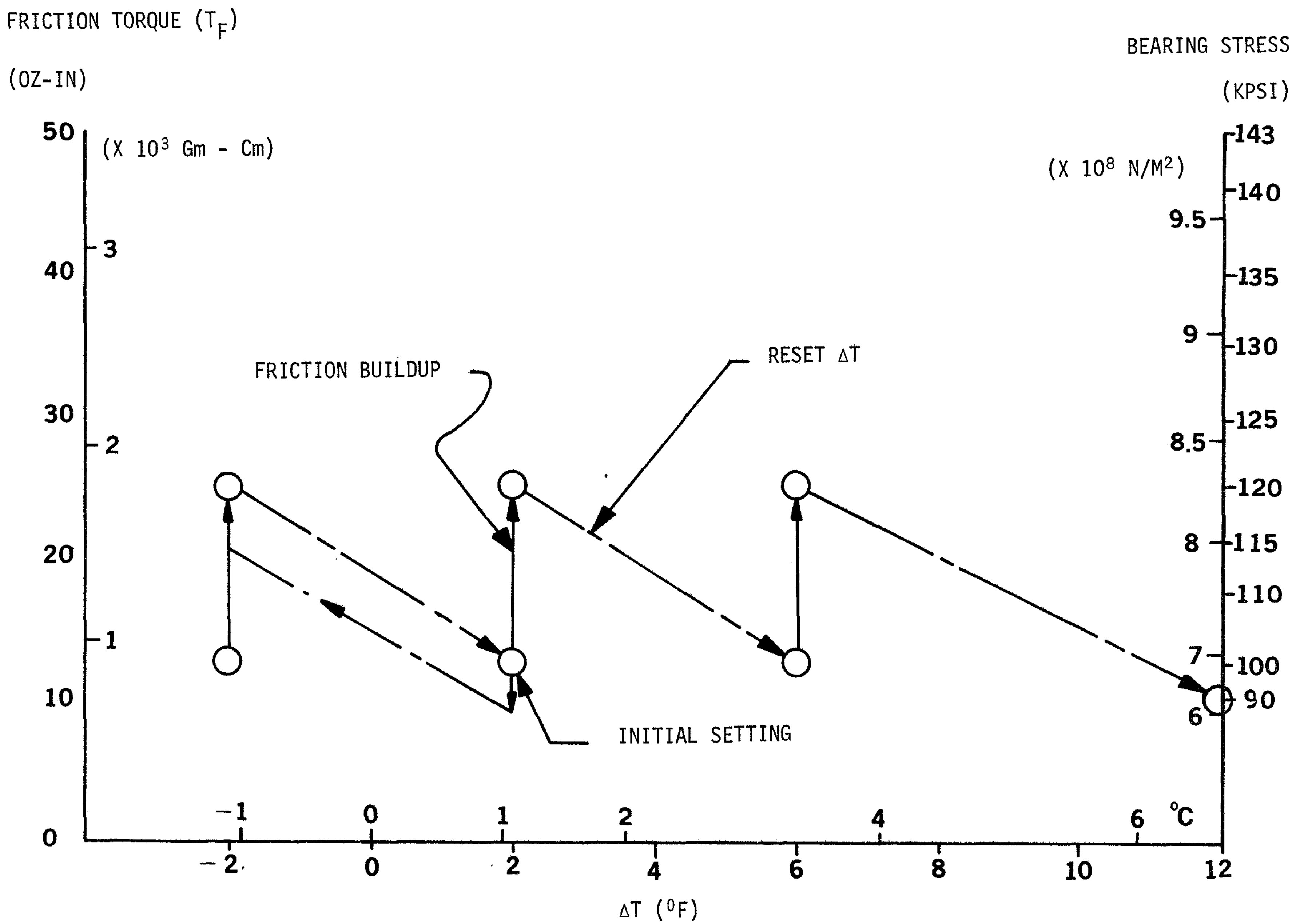

FIGURE 7 - THERMAL CONTROL