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LINEAR PROOF MASS ACTUATOR

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ABSTRACT

This paper describes the mechanical design, analysis, fabrication, testing, and lessons learned by developing a uniquely designed spaceflight-like actuator. The Linear Proof Mass Actuator (LPMA) was designed to attach to both a large space structure and a ground test model without modification. Previous designs lacked the power to perform in a terrestrial environment while other designs failed to produce the desired accelerations or frequency range for spaceflight applications. Thus, the design for a unique actuator was conceived and developed at NASA Langley Research Center. The basic design consists of four large mechanical parts (Mass, Upper Housing, Lower Housing, and Center Support) and numerous smaller supporting components including an accelerometer, encoder, and four drive motors. Fabrication personnel were included early in the design phase of the LPMA as part of an integrated manufacturing process to alleviate potential difficulties in machining an already challenging design. Operational testing of the LPMA demonstrated that the actuator is capable of various types of load functions.

INTRODUCTION

With the development of large space structures, a means of control to eliminate vibrations induced into the structures by disturbances such as plume impingement, docking forces, and crew activity is desired. Also of interest is the ability to study structural behavior of a flight article in a terrestrial environment so that analyses can be verified prior to launch. Hence, it became necessary to design an actuator with multiple functions for spaceflight and terrestrial environments.

The actuator was to satisfy four major requirements: 1) Typically, space structures are extremely lightweight, and some structures can not support their own mass in a terrestrial environment; therefore, the actuator should be lightweight, yet rigid; 2) A major function of the actuator is to damp out vibrations, but equally as important is its ability to excite oscillations in a structure so that reactionary dynamics can be studied; 3) The actuator must be functional in spaceflight and terrestrial environments without being modified; 4) The actuator must perform in the gravity oriented axis and two axes orthogonal to the gravity axis. Together, the above requirements resulted in a challenge that, if successfully met, would contribute significantly to the design of lightweight actuators that could operate in gravity or zero-g environments. An actuator called LPMA was designed, fabricated, and tested that met the above requirements. As a result of this work and its inventive nature, U.S. Patent 5,150,875 entitled "Linear Mass Actuator" was granted on September 29, 1992.

DESIGN

To avoid replicating the deficiencies of previous actuators, a unique and original design was required. After an intense conceptual design phase, a linear mass actuator with a friction drive, powered by DC torque motors, was selected and subsequently called the Linear Proof Mass Actuator (LPMA). A photograph of the assembled LPMA is shown in Figure 1. Designing a friction drive system utilizing DC torque motors solved several

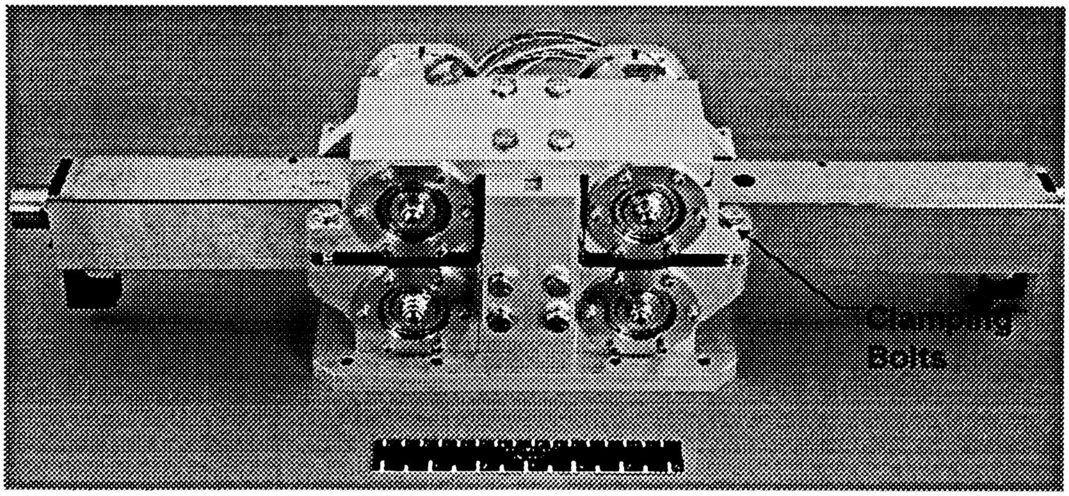

Figure 1. Linear Proof Mass Actuator (NASA Photograph L-89-2126)

problems of previous actuators. Several actuators possessed cogging problems because of gear drives while others were power limited by magnetic field drivers. The combination of DC torque motors and friction drive eliminated these deficiencies, yet the friction drive system requiring tolerances of 2.54×10^{-6} meters (0.0001 inches) created more challenges. The minute tolerances presented a difficult design and fabrication task. With the use of the American National Standards Institute's Dimensioning and Tolerancing document, ANSI Y-14.5M-1982, the difficult tolerancing task was accomplished.

As previously mentioned, the design consists of four major mechanical parts with the Mass being the most precise and critical, fifty-six smaller parts, and fasteners which are shown in Figure 2. The Mass weighs approximately 98 N (22 lb) which is roughly half of the total system weight. Sandwiched between four motor driven shafts, called Rollers, the Mass translates linearly to deliver a force. Friction contact between the Rollers and the Mass is the only drive mechanism. When the four clamping bolts (see

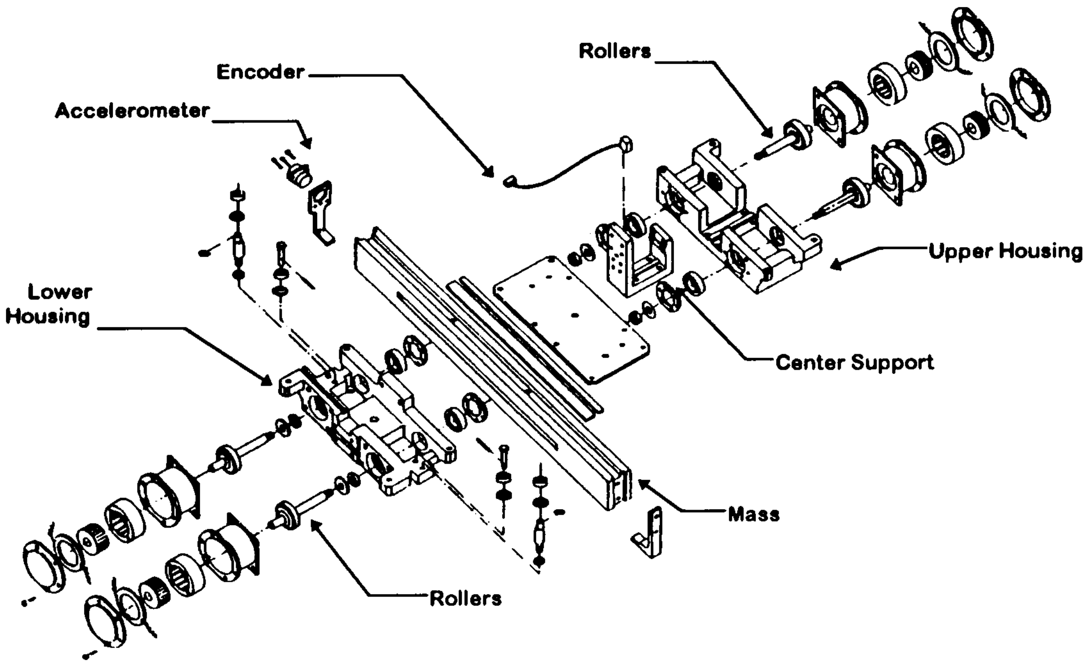

Figure 2. Components of LPMA (2)

Figures 1 and 3) are torqued, the Upper and Lower Housings spring about the Center Support which causes the Rollers to engage the Mass. After the Mass is engaged, an additional preload of 111 N (25 lb) gives the Mass/Roller interface the necessary friction to move the Mass without slipping. The clamping bolts are screwed into locking helical inserts to maintain the proper torque values. The top and bottom Motors, which drive the Rollers, counter-rotate to move the Mass one direction, then they reverse rotation to move the Mass the opposite direction. Figure 3 pictorially illustrates the functioning system. By deleting gears, belts, and hydraulics, this design resulted in improved performance because the Mass traveled in a smoother fashion with no cogging effects. Deleting belts eliminated a mechanism to compensate for decreased belt tension as the belt aged and eliminated concerns over belt breakage. Omitting hydraulics solved potential fluid leakage problems and removed a substantial weight penalty that was not desirable for spaceflight. As a result, the friction drive system for the Mass provides a smooth traveling linear actuator. Since the Mass and the DC Torque Motors determine the ultimate amount of force to be applied to a space structure, Newton's Second Law becomes the governing design equation.

$$F=ma$$

Force Delivered = (mass of the Mass) x (Torque Motor Acceleration)

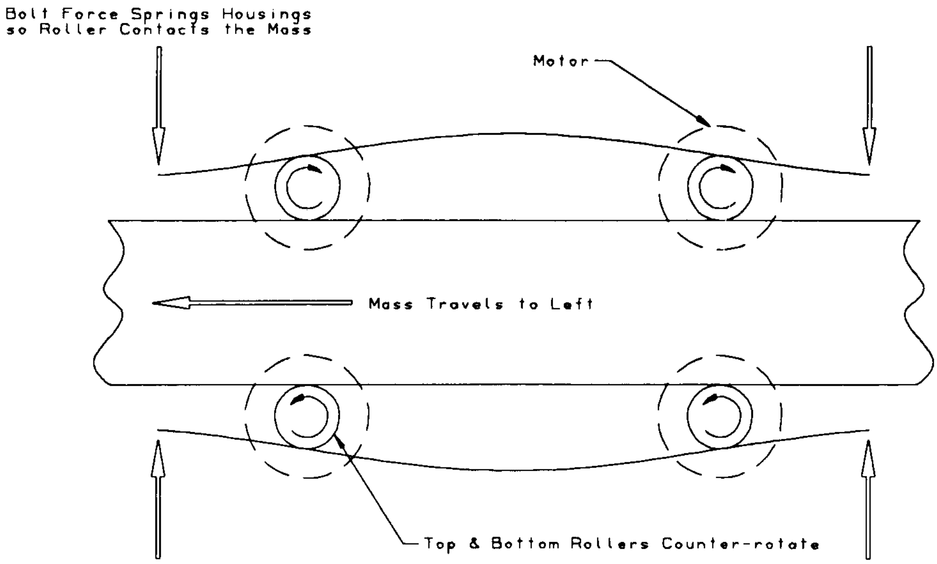

Figure 3. Schematic Representation of Actuator Operation

The actuator was then designed around the Motor-Roller and Mass interface. The length of the Mass determines the available stroke or distance of travel which maximizes the amplitude of motion. The volume of the Mass determines the mass which accelerates to produce the total available force. Therefore, after the Mass was sized, only the motors could vary the force by changing the acceleration.

The Mass is sized at 10 kilograms (22 lb), so when the assembly is oriented vertically, the LPMA delivers 30 N (6.75 lb) of force. When the LPMA is oriented horizontally, it possesses additional force capability which allows delivery of 128 N (28.78 lb). The full stroke of the LPMA is $\pm$15 cm (5.9 inches) with a position resolution of 10 micrometers (3.94×10^{-4} in).

The LPMA can operate in four different modes. The first mode is a position mode where force is delivered to the Mass so the Mass may maintain its positon at the commanded position relative to the LPMA base. This mode is limited by an upper frequency limit which is a function of excitation amplitude and maximum available force. Figure 4 illustrates how the stroke, which is defined in decibels because of its compatibility with the controls analysis, varies with frequency. Stroke is defined in decibels as: db=20log(x/15), where x is the stroke length in centimeters. The second mode is a force mode which is limited at low frequencies by the stroke capability and excitation amplitude. Figure 5 illustrates how the forces vary with frequency. The force

LPMA Stroke Curves

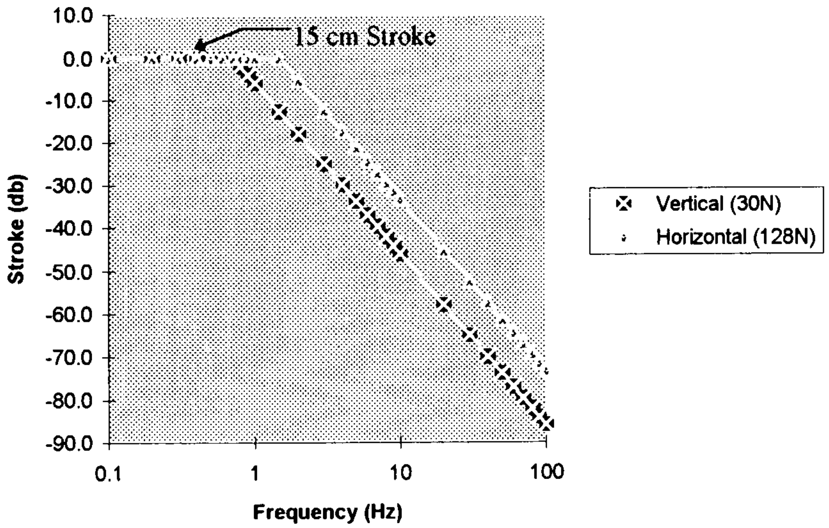

Figure 4. Stroke Versus Frequency

LPMA Force Curves

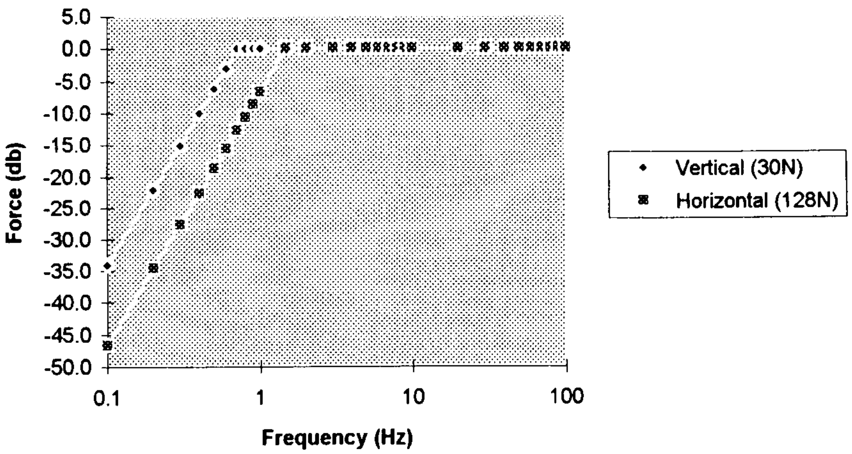

Figure 5. Forces Versus Frequency

curves are again displayed in decibels because of their compatibility with the controls analysis. The third mode is a combination of the position and force modes which gives the LPMA increased performance across the frequency range. The fourth and final mode is the velocity mode which is used in rare occurrences where constant velocity is required.

The friction drive concept requires a preloaded metal to metal contact between the Mass and the Rollers. This type of contact presents problems with galling and wear. To avoid galling, 17-4 PH Corrosion Resistant Steel (CRES) heat treated to 496°C (925°F) to achieve a Rockwell Hardness of C47 was selected for the Mass, and 15-5 PH CRES heat treated to 538°C (1000°F) to achieve a Rockwell Hardness of C36 was selected for the Rollers (shown in Figure 6). The durability of 17-4 on 15-5 is considered very good and the heat treatments were chosen so that the Mass, which is very costly to produce, is much harder than the Rollers. Therefore, any detrimental wear will occur on the Rollers, which are much cheaper to replace.

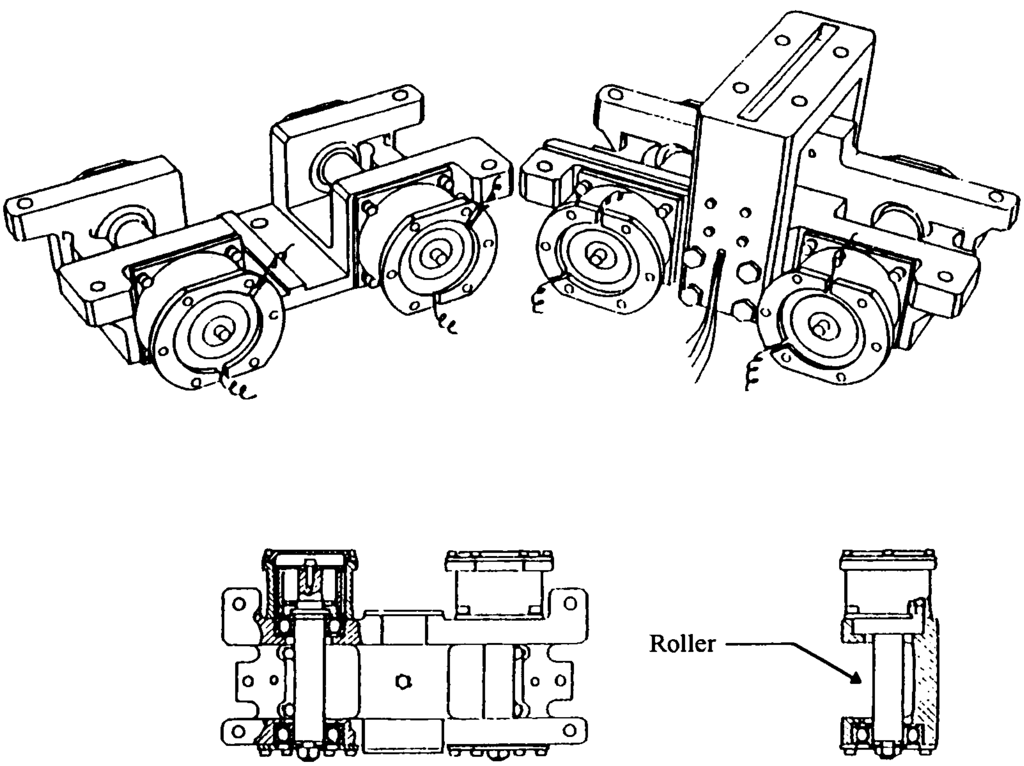

Figure 6. Lower Housing (Left) and Upper Housing (Right) (1)

ANALYSIS

The structural analysis of the LPMA consisted of a finite element model created using solid elements for the Upper Housing, Center Support, and Mass, beam elements for the Rollers, and spring elements for the Roller Bearings. The Lower Housing is more massive and obviously stiffer than the Upper Housing, so the Lower Housing was assumed to be a rigid structure in this analysis. As a result, the modeling concentrates on the Upper Housing and its mating parts.

Because the LPMA relied on friction to drive the Mass, the Mass and Rollers necessitated a metal to metal non-slipping interface. The design requirements specified zero tolerance which ensures a metal to metal fit that is nearly impossible to manufacture; therefore, the parts were toleranced so the minimum tolerance was zero and the maximum tolerance between the Rollers and the Mass was 0.029 cm (0.0114 in). For any situation other than a perfect fit, a certain amount of bolt preload is necessary to clamp the Upper and Lower Housings, which contain the Rollers, to the Mass for engagement. The worst case preload of 15,123 N (3400 lbs) is required for the maximum tolerance of 0.029 cm (0.0114 in) to guarantee metal to metal engagement. Figure 7 illustrates the displacements and Figure 8 illustrates the deformations of the Upper Housing and Center Support under the 15,123 Newton (3400 lbs) preload assuming an infinitely stiff Lower Housing. The Von Mises stresses

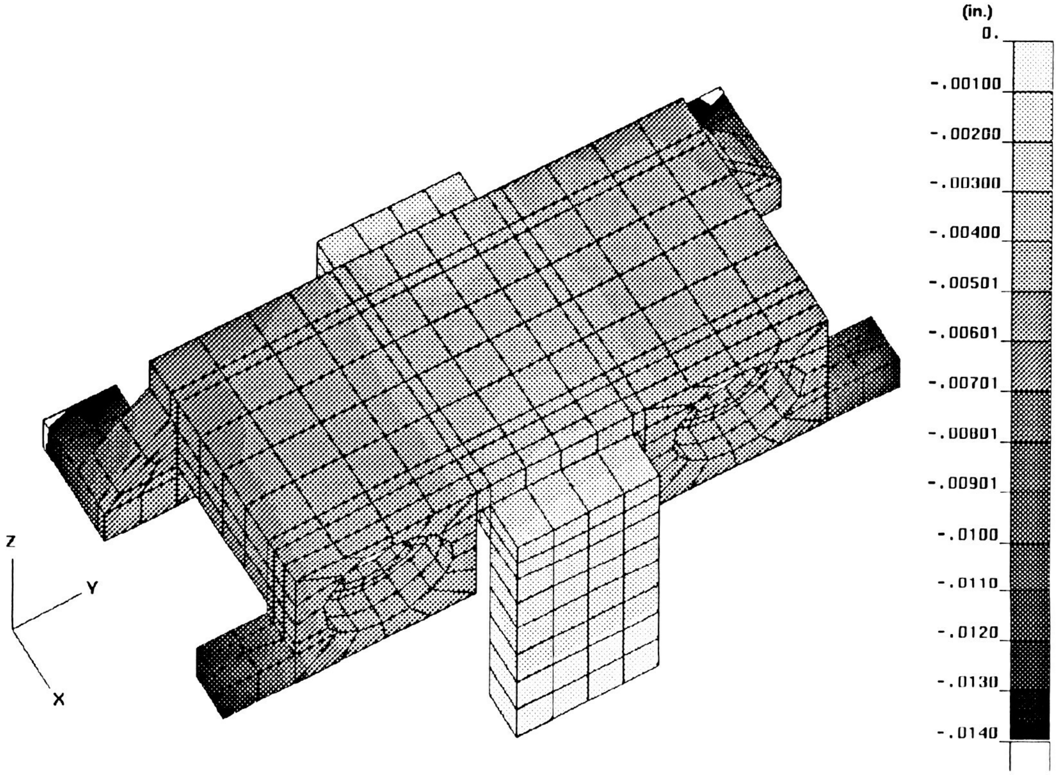

Figure 7. Upper Housing and Center Support Displacements for 3400 lbs Preload

for the Upper Housing were below the yield 241,325 kPa (35 ksi) of Aluminum 6061-T6 except at two bolt locations. The stresses at these bolt locations are artificially high because the bolt loads are applied as point loads. Figure 9 shows the stresses for the preload state indicating the higher stresses at the attachment feet of the Upper Housing. Detailed calculations at the bolt interface area indicate that the actual stress

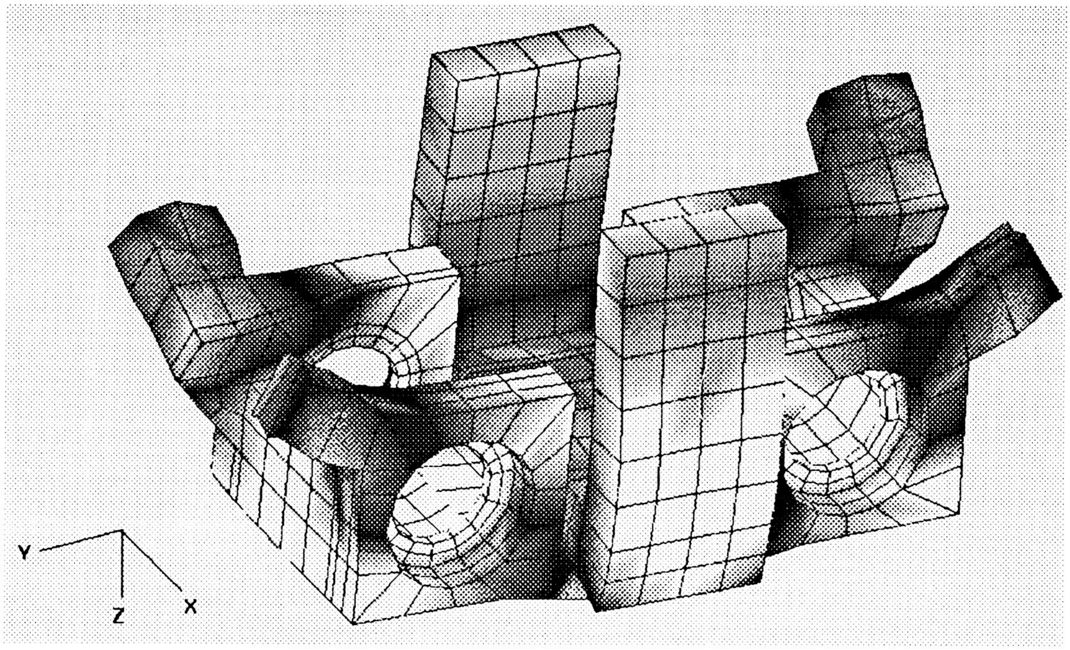

Figure 8. Upper Housing and Center Support Deformations for 3400 lb Preload

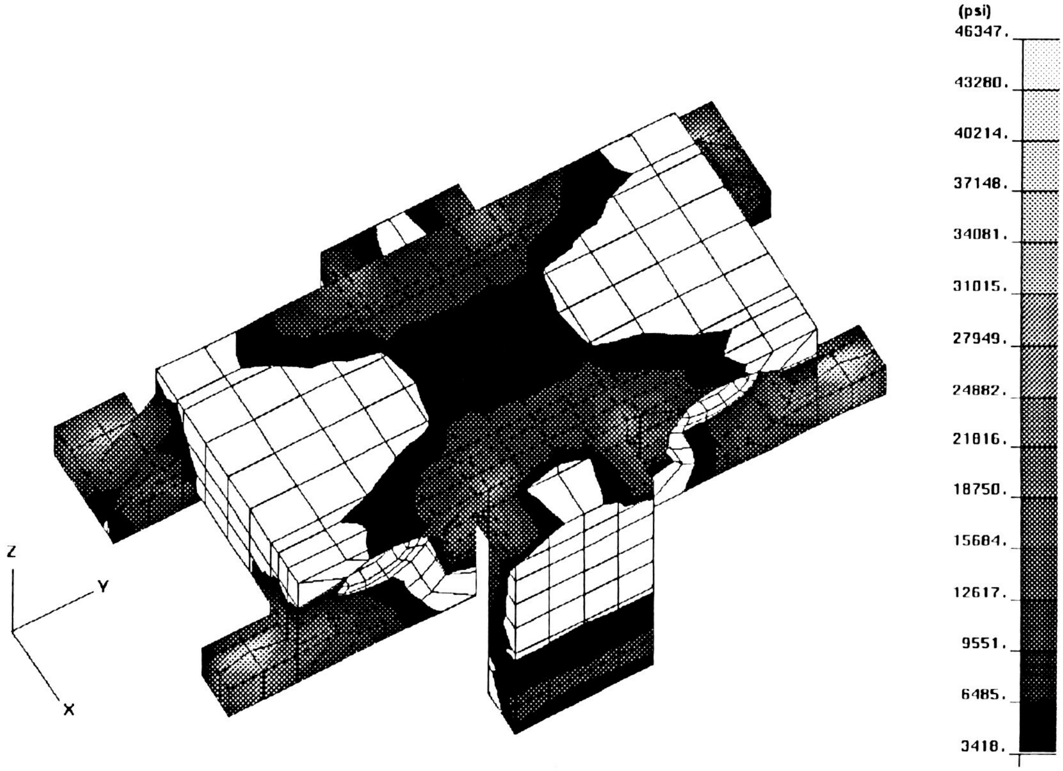

Figure 9. Upper Housing and Center Support Von Mises' Stresses for Preload Condition

was 166,859 kPa (24.2 ksi) compared to the artificially high stress of 319,0238 kPa (46.3 ksi) in Figure 9.

As mentioned previously, a metal to metal non-slip interface was the requirement for the friction drive to work. The analysis determined the bolt loads necessary for metal to metal contact. The Mass must deliver the maximum force without slipping on the Rollers. A conservative coefficient of friction of 0.3 was chosen versus the typical value of 0.58 for steel on steel. Using this extremely conservative value, only 111N (25 lbs) of additional bolt force were required to produce sufficient contact force, which produces minimal stresses in the rollers, for operation at maximum acceleration. As a result, a maximum load of 15,234 N (3425 lbs) was required to obtain a metal to metal non-slip interface for the LPMA for its worst case tolerance where the maximum gap existed between the Rollers and the Mass.

FABRICATION and ASSEMBLY

The philosophy of the program was geared to manufacturing one spaceflight engineering model and five subsequent spaceflight prototypes for ground testing. Utilizing spaceflight design and fabrication practices for the engineering models increased manufacturing costs. The project rationale demanded this philosophy so that a flight article only necessitated fabrication and flight qualification if an opportunity for a space mission became available.

One of the newest engineering phrases of the 1990's is "integrated product design." This is a concept where all disciplines involved in the final product work the task together from start to finish to produce a cheaper and higher quality product in a shorter amount of time. The LPMA project team practiced this philosophy from the start of the program in early 1986 to its completion in June of 1990. The Lead Fabrication Technician participated in the design phase recommending materials, tooling cuts, and design changes. The monetary savings were numerous and significant, and the model was produced quicker and cheaper than envisioned. As an example, to reduce the high costs of spaceflight fabrication, all small parts were purchased or machined simultaneously for all six assemblies prior to completion of the design of the Upper and Lower Housings. This course of action was risky because the possibility of scrapping hardware because of design changes to the housings existed; yet, the smaller pieces were simpler and less likely to change. The gamble paid off as the bulk machining of the smaller hardware pieces saved tooling costs because repetitive tooling set up operations were avoided. In addition to the fabrication time saved, Quality Assurance (QA) time to inspect the hardware was also reduced for the same reasons. As a result of project budget cuts, only two complete prototype assemblies were manufactured; therefore, four assemblies of smaller hardware became spare parts.

After assembling the engineering model, the Rollers were noticeably scored after a press fitting through the precision bearings. Using the integrated product design philosophy within the project team, a new assembly method was developed using the spare parts. The new process submerged the Rollers in liquid nitrogen for five minutes to shrink the Roller diameters. The cold Rollers were assembled with the warm Barden Precision Bearings producing a loose fit. After insertion through the bearings, the Rollers warm up to their original dimensions which produce the desired interference fit, and the Rollers remain undamaged.

In order to ensure smooth travel and optimum performance, the Mass must be designed precisely and machined accurately within the miniscule tolerances provided. The Mass' design required an expert machinist for fabrication to hold the rectangular 56.6 cm (22.3 in) long piece of 17-4 PH corrosion resistant steel flat and parallel to 0.0002 cm (0.0001 in), so it would maintain frictional contact with the Rollers at all times.

TESTING

Due to low priority, the LPMA was developed over a period of approximately five years. The customers of LPMA finally established a drop dead delivery date for the hardware. This delivery date, coupled with the fact that the electronics/software design and fabrication lagged the mechanical development, left little time for testing. The only mechanical testing accomplished was that testing done while verifying the software and control system. Even through this limited testing, the LPMA demonstrated that the Mass would deliver forces by several methods over a frequency range of 0 to 100 Hz. The LPMA displayed capabilities to move in response to commanded displacements, velocities, or accelerations. The input curves could also simulate sinusoidal, step, and saw-tooth functions. Despite the limited testing, the LPMA proved this friction drive concept and was used at a ground test facility to be discussed later in this paper.

LESSONS LEARNED

Since the engineering model was intended to be a learning experience for subsequent prototypes, the project team was able to improve the design and fabrication deficiencies. Numerous improvements, which could be applied to future prototypes, were derived from the experience.

The first and most productive lesson learned was using the integrated product design philosophy. By integrating the lead fabrication technician into the design team, many problems were eliminated before they became problems. The engineer and fabrication technician frequently reviewed the design in a "Coyote Team" manner. Coyote Teams differ from "Tiger Teams" in that a Tiger team attempts a solution to an existing problem; whereas, a Coyote team searches for potential problems before they impact the work. Time and money were saved due to the technician's suggestions on material selections and tooling cuts on the hardware. He also offered a suggestion on the dimensional verification process of the housings and center support. Since aluminum alloy tends to warp after an abundance of machining, he suggested checking the dimensions while the parts were in their tooling fixtures. The warpage was very small but large enough to fail the QA inspection; yet, when assembled, torquing the fasteners in the assembly eliminated the warpage. His suggestion saved the time and money of rework or a new set of hardware.

For spaceflight aluminum parts, anodization and chemical film are two widely used protection processes. Anodization offers good abrasion resistance while chemical films protect better against corrosion. Typical thicknesses of sulfuric anodization are 0.0001-.002 cm (0.00005-0.001 in) while chemical film thickness is zero. Sulfuric anodization was chosen for its inherent abrasion protection since many of the LPMA parts were metal to metal fits. One problem with sulfuric anodization was entrapment. It is common knowledge that the solution can cause problems in screw threads and inserts, and that the threaded holes should be protected from the solution. After anodizing the Upper and Lower Housings, a film of 0.1 cm (0.04 in) was measured in the bearing bore holes. The hole diameters were 4.7 cm (1.85 in) and entrapment was never envisioned in a hole this large. Figure 10 illustrates the area of entrapment. The parts had to be remachined to remove the excessive film and the protection process was switched to chemical film treatment which has zero dimensional change. So, for hardware with three place decimal tolerances, careful consideration should be given to choosing anodization over chemical film treatment.

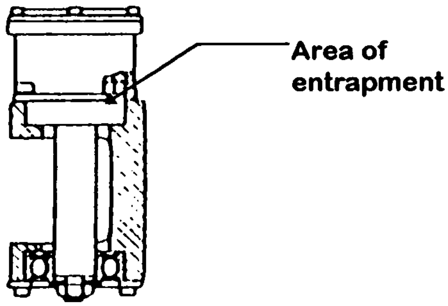

Figure 10. Area of Anodization Entrapment. (2)

The Roller and Mass interface was the most important interface of the actuator. Heat treatments were chosen to avoid galling and to ensure that excessive wear would not occur to the most expensive part, the Mass. The heat treatments were chosen using average values. After fabrication, it was discovered in MIL-HDBK-5, Metallic Materials and Elements for Aerospace Vehicle Structures, that each heat treatment has a range of hardness values for a particular temperature. As a result, it was possible for the Mass to be softer than the Rollers which was not desirable. Both parts were checked, and their intended hardness values were in a desirable range. To ensure that the Roller is always softer, the heat treatment should be changed to H1150 or H1150M, which yields hardness values no higher than C37 and C30, respectively.

After pressing the precision bearings on the Rollers, scoring was witnessed on the Rollers. The scoring would have a detrimental effect on the friction contact between the Rollers and the Mass, so a new installation procedure, which thermally decreases the Roller diameters by submerging them in liquid nitrogen, was utilized. While the

Rollers are still cold, they are inserted through the bearings with no contact. As the Rollers warm up, contact is achieved. For spaceflight, a better solution would be to redesign the Roller by stepping down the diameter where the Mass contacts the Rollers. As a result, the Mass height would increase by twice the radial decrease of the Rollers. This solution avoids the liquid nitrogen process which causes the Rollers to frost after installation, and the bearing lubricants may be contaminated from the melted ice.

An adjustable shaft with a bearing was located on the Lower Housing with the bearing running in a groove on the lower portion of the Mass to eliminate lateral motion created by tolerance stackup. The adjustable shaft was a cam by design so the full diameter of the cam was utilized at the maximum tolerance. After adjusting the cam on the assemblies, it became apparent that the cam should have been much larger in diameter to give a better feel for alignment adjustment. Figure 11 shows the cam which should have its diameter increased for better performance.

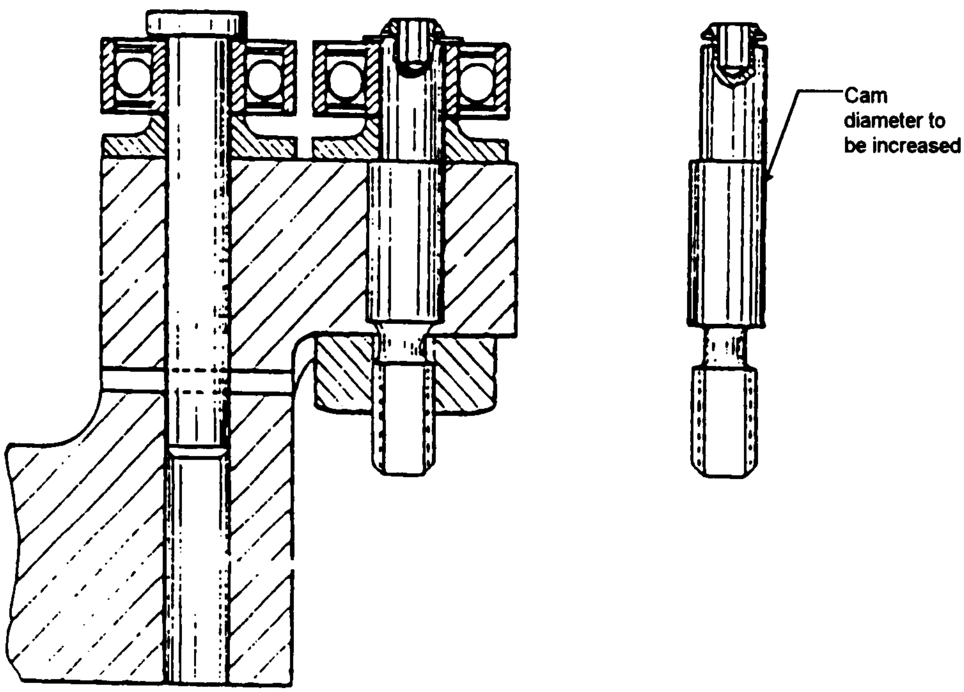

Figure 11. Views of Cam and Cam Assembly. (1)

Another lesson learned pertains to using commercial parts. On the LPMA, the D.C. torque motor parts were supposed to be interchangeable which was a bonus because the LPMAs were designed for total interchangeability. After dimensionally checking the lot of twenty-four motors, it was documented that the motor parts were not dimensionally interchangeable. These commercial parts were verified early in the fabrication process which allowed time to re-bore the inner diameters of the motors so that the whole set was completely interchangeable. As a result, commercial parts should always be verified for interchangeability despite the claims of product literature.

To acquire the flatness needed for the Mass to translate properly, a uniform flatness requirement of 0.0005 cm (0.0002 in) was needed on the Roller sides of the Mass. To accomplish this requirement, the Mass was fixed to a granite table where bluing ink was applied to one Roller surface. The surface was then hand-lapped with a 0.00038 cm (0.00015 in) flatness lapping stone until the ink was removed; thus, meeting the flatness requirement. Since new machines can achieve tighter tolerances, most hand operations of the past have become obsolete. Today's current machining technology brought the Mass within 0.002 cm (0.001 in) of the Mass' goal, so the lapping process was shortened by only having to remove 0.002 cm (0.0008 in) of material. As a result, the art of hand-lapping still had not lost its niche in today's high technology world.

CONCLUSION

The LPMA was used as a frequency exciter for a large space structures support fixture at the top of Building 1293 at NASA Langley Research Center. To ensure structural soundness of the support fixture, the LPMA was attached and operated to excite this instrumented fixture. By observing and comparing frequency responses of the fixture, the structural integrity was verified without the cost of load testing. As long as the LPMA is connected to the fixture, structural verification could be done by turning on its power.

ACKNOWLEDGEMENTS

I would like to acknowledge those who contributed to the design and analysis of the LPMA. Irby Jones conceived the mechanical design and Jim Miller conceived and engineered the electrical design. Vaughn Behun and Lewis Goodrich were instrumental in developing a preliminary design concept. Ed Crossley served as a consultant during final design, and he checked all mechanical design drawings. Calvin Davis, Lead Technician, directed the LPMA fabrication, and he contributed suggestions on material selections, tooling cuts, and assembly procedures which made the fabrication and asembly processes much simplier. Genevieve Dellinger completed the structural analysis using finite element model techniques. She created Figures 7,8, and 9 in this paper during her analysis. Without the contributions of the above, this paper would have not been possible.

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