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EVALUATION OF A HIGH-TORQUE BACKLASH-FREE ROLLER ACTUATOR

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ABSTRACT

NASA Lewis Research Center recently began a research program to investigate mechanism positioning systems that would be suitable for space vehicles. High-torque-density efficient systems are required that operate smoothly without mechanical backlash and without liquid lubrication systems. Roller drives inherently have many of these required properties because power is transmitted through continuously rolling drive elements. The roller-driven mechanisms investigated range from a smooth dry-running drive designed for the Space Station alpha joint (Loewenthal and Schuller, 1986) to a backlash-free traction robot joint tested in NASA's telerobotic research program (Kuban and Williams, 1987).

This paper presents results of a test program that evaluated the stiffness, accuracy, and torque ripple of a 16:1, 320-ft-lb planetary roller drive for a potential space vehicle actuator application.

The drive's planet roller supporting structure and bearings were found to be the largest contributors to overall drive compliance (reciprocal of stiffness), accounting for more than half of the total. In comparison, the traction roller contacts contributed only 9 percent of the drive compliance based on an experimentally verified stiffness model. The drive exhibited no backlash although 8 arc sec of hysteresis deflection were recorded because of microcreep within the contact under torque load. Because of these load-dependent displacements, some form of feedback control would be required for arc-second positioning applications. Torque ripple tests showed the drive to be extremely smooth, actually providing some damping of input torsional oscillations.

INVESTIGATION OBJECTIVES

Under a cooperative program with industry, a 16:1, 320-ft-lb output torque roller actuator was evaluated experimentally (Steinetz et al., 1986) to determine its potential suitability as a high-torque space vehicle actuator, such as for a control moment gyro (CMG) gimbal drive. Analytical predictions of the torsional stiffness of the drive were compared with static torsional stiffness measurements, and "torsionally soft" drive components were identified. Data were also obtained on the drive's zero backlash, torque ripple, hysteresis characteristics, and positional accuracy performance.

- **EVALUATE SUITABILITY OF ROLLER DRIVE TECHNOLOGY FOR POTENTIAL SPACE ACTUATOR APPLICATION**
- **DETERMINE DRIVE'S TORQUE RIPPLE, HYSTERESIS, POSITIONING ERROR, BACKLASH, AND CREEP CHARACTERISTICS**
- **ANALYTICALLY MODEL DRIVE SYSTEM TORSIONAL STIFFNESS AND EXPERIMENTALLY IDENTIFY "TORSIONALLY SOFT" COMPONENTS**
- **EVALUATE DRY PERFORMANCE OF GOLD ION-PLATED ROLLERS**

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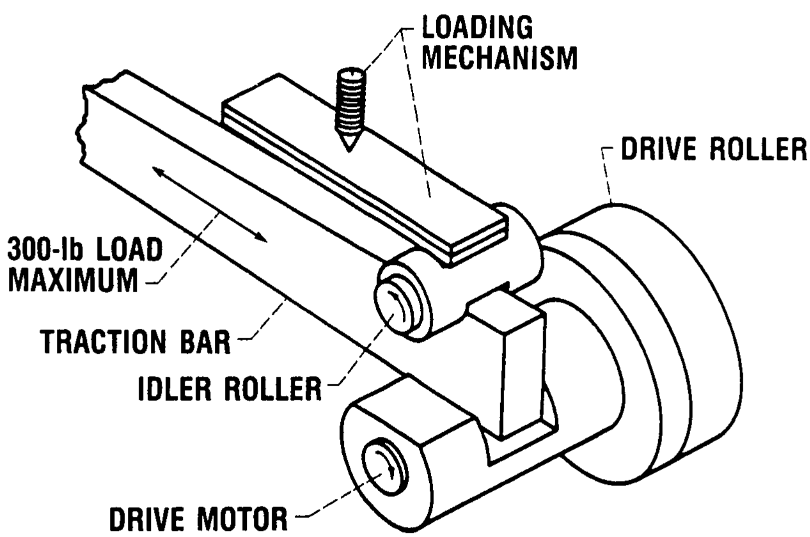
ULTRAPRECISE TRACTION ROLLER FEED MECHANISM

Roller actuators have been used in positioning mechanisms where gear or ball screw systems could not be used as in this ultraprecise traction roller feed mechanism of the Lawrence Livermore Laboratory (Bryan, 1979). In this device a traction roller drives a translating traction bar that positions the parts to 0.2-μin. accuracy using a closed-loop laser interferometry feedback control system.



ACCURACY 0.2 μin. (0.05 arc sec)

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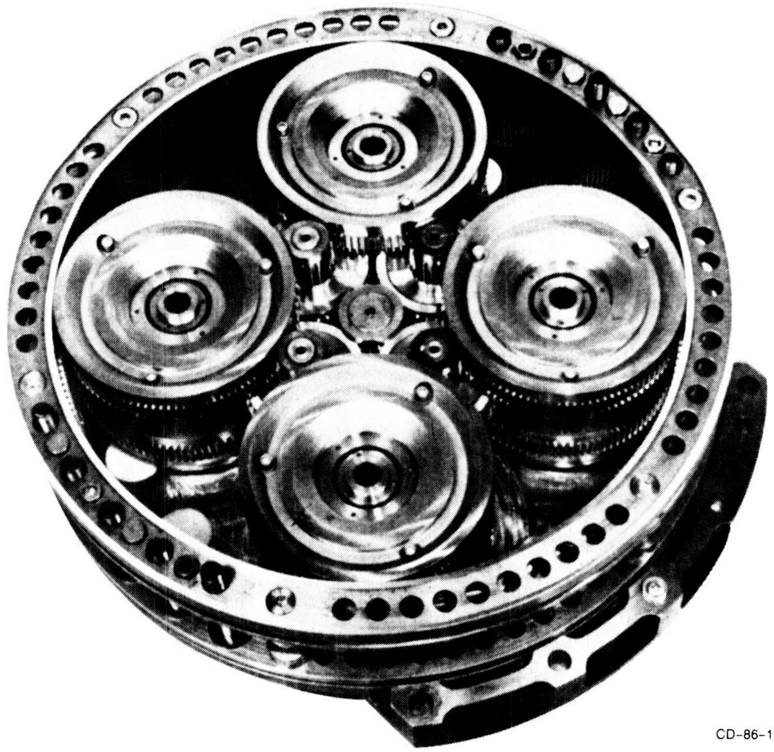
26:1 CONTROL MOMENT GYRO ROLLER-GEAR DRIVE

An exceptional output torsional stiffness of 500 000 ft-lb/rad was demonstrated in this combined roller-gear drive designed and built for a satellite control moment gyro application (General Electric, 1972). In this drive design the rollers and gears transmit the load in parallel, combining in a compact package the excellent torsional stiffness and backlash-free behavior of traction rollers with the high-torque carrying capability of gears.



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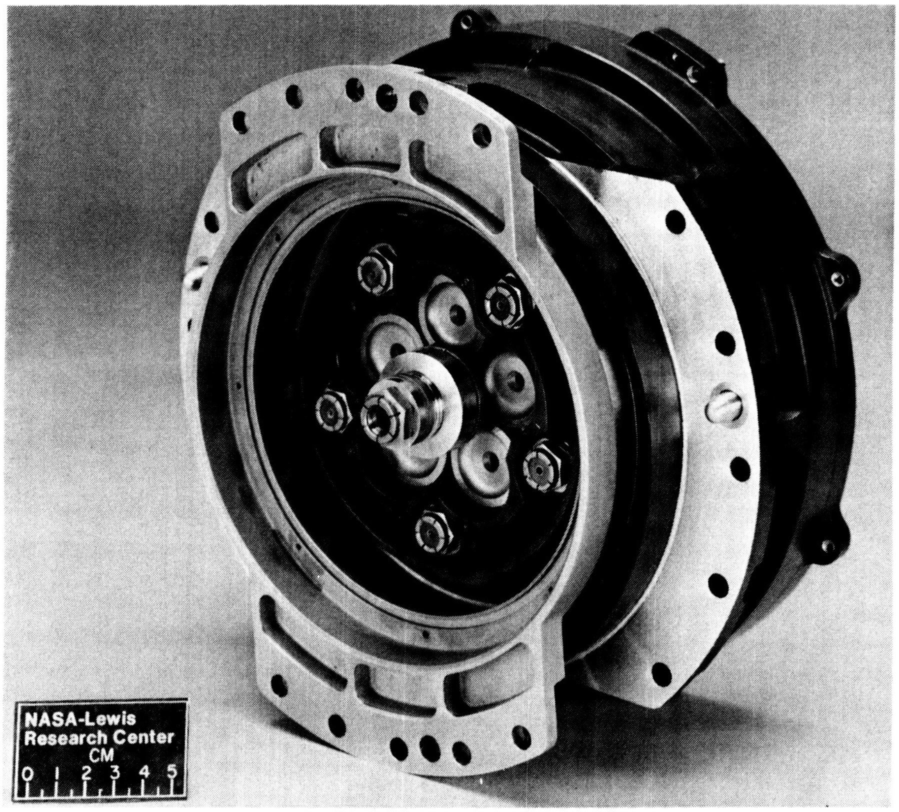
PROTOTYPE SPACE VEHICLE ROLLER ACTUATOR

Critical prototype drive system design requirements include a minimum drive torsional stiffness of 250 000 ft-lb/rad at the output, a low weight, and a minimum design life of 1600 hr. Maximizing the drive's torsional stiffness while minimizing the drive weight and achieving the required life were the paramount considerations guiding the design. Structural design tradeoffs and material selections were consistent with flight hardware requirements.

The drive was designed to operate without liquid lubrication with a design traction coefficient of 0.1, which is at least 20 percent below the maximum available traction coefficient of the gold-ion-plated sun rollers against their steel first-row planet rollers (Spalvins and Buzek, 1981). A layer of gold, 7.8 μin. thick, was ion plated onto the sun roller surface as a dry film lubricant to prevent the sun and first-row planet rollers from cold welding in the vacuum environment. Life limitation in this design is one of wear of the gold layer. Based on NASA sliding friction data (Spalvins, 1985), the gold thickness was determined for a minimum of 1600 hr of operation.

The drive is nominally 9.84 in. in diameter by 8.66 in. in length and weighs 22.1 lb. This compact Nasvytis planetary drive (Nasvytis, 1966) packages well in the small design envelope.



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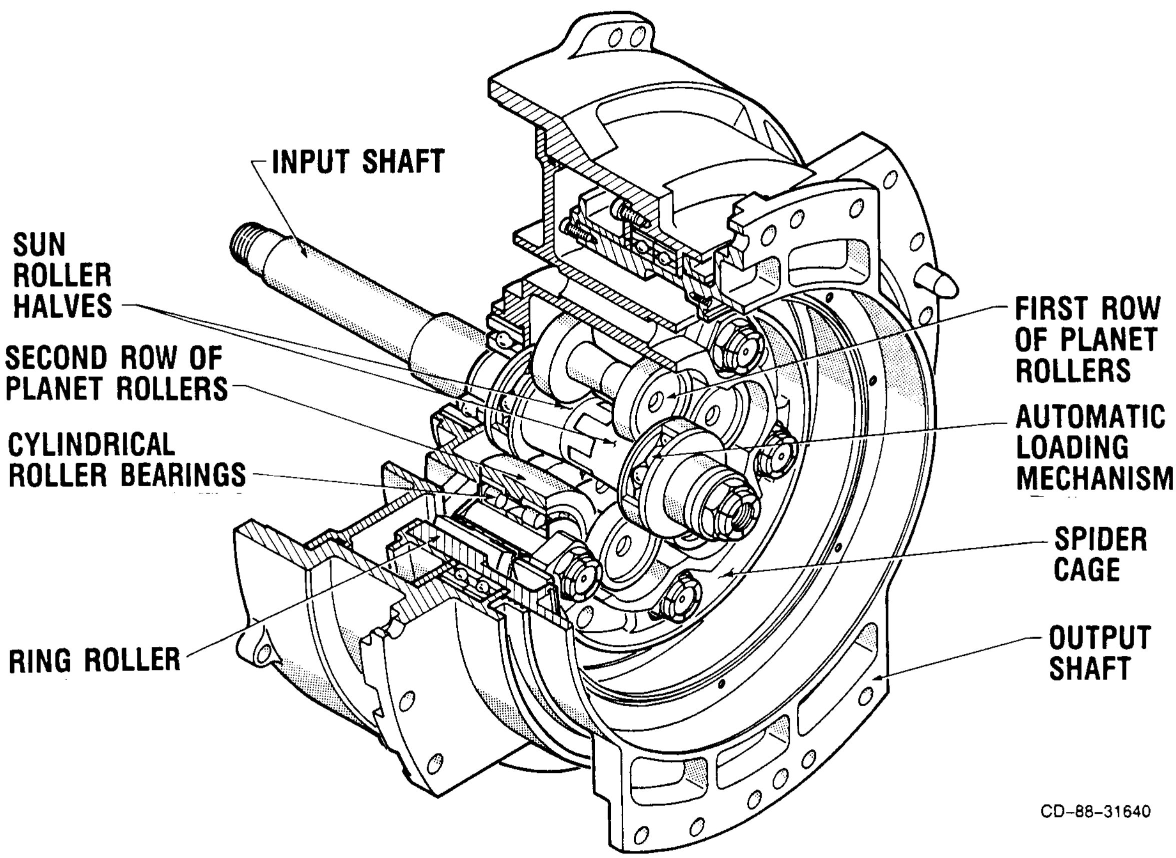
16:1 ROLLER ACTUATOR TEST UNIT

The servomotor-driven input shaft transmits torque to the two halves of the sun roller through two sets of ball-ramp loading devices. The sun roller, in turn, drives five stepped first-row planets that drive five second-row planets. These outer planets carry the torque to the ring roller attached to the output shaft. Because of the double-end geometry of the rollers, ten, twenty, and then ten parallel contact paths occur, respectively, at successive contacts. The number of planet rollers per row, number of rows, and relative step sizes are design parameters to be optimized for a given application.

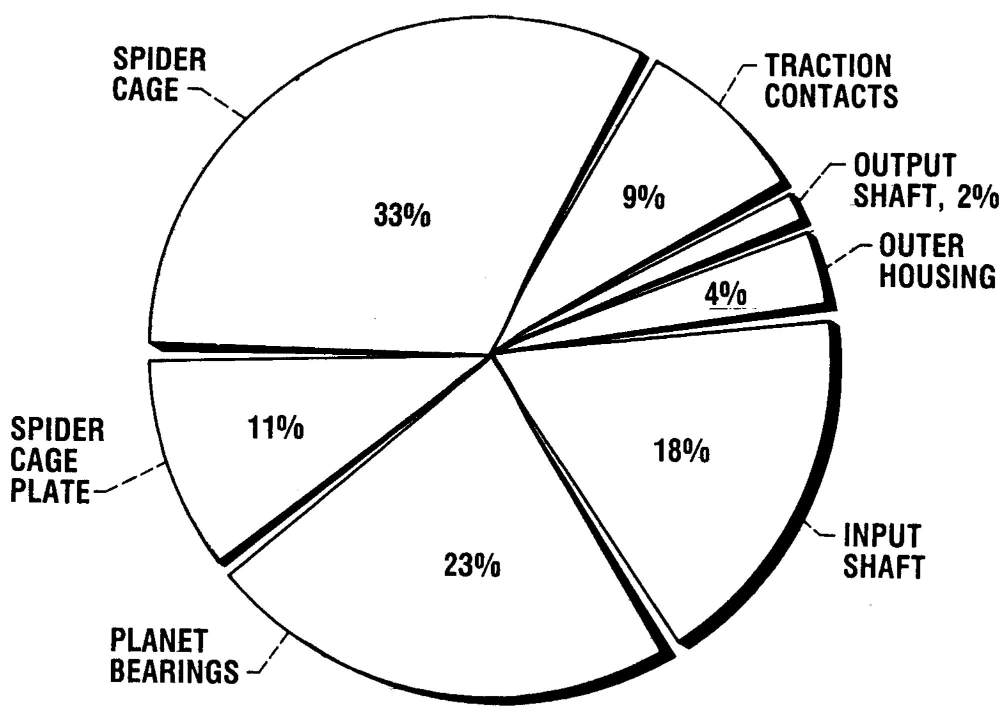
The torque loading mechanism increases the normal load between the conically shaped rollers in direct proportion to the applied torque by causing inward axial motion of the sun roller halves. The potential for slip is not only eliminated by incorporating a roller loading mechanism, but also the normal loads on the rollers do not have to be set at maximum at all times, thereby extending coating wear life and minimizing frictional losses.



PREDICTED DRIVE COMPLIANCE

The torsional stiffness of the drive (or its reciprocal, compliance) was analytically determined by modeling each of the drive's major components. The compliance of each element was found and is presented as a percentage of the total drive compliance. The spider cage support structure, planet bearings, and input shaft torque loader mechanism were the major contributors to the drive compliance. The traction contact compliance analyzed using a comprehensive technique developed at NASA Lewis (Rohn and Loewenthal, 1985) contributed only 9 percent to the overall drive compliance.

Adding the elemental compliances and taking the inverse resulted in an overall predicted drive torsional stiffness at the output shaft of 500 000 ft-lb/rad - twice the design target stiffness value.



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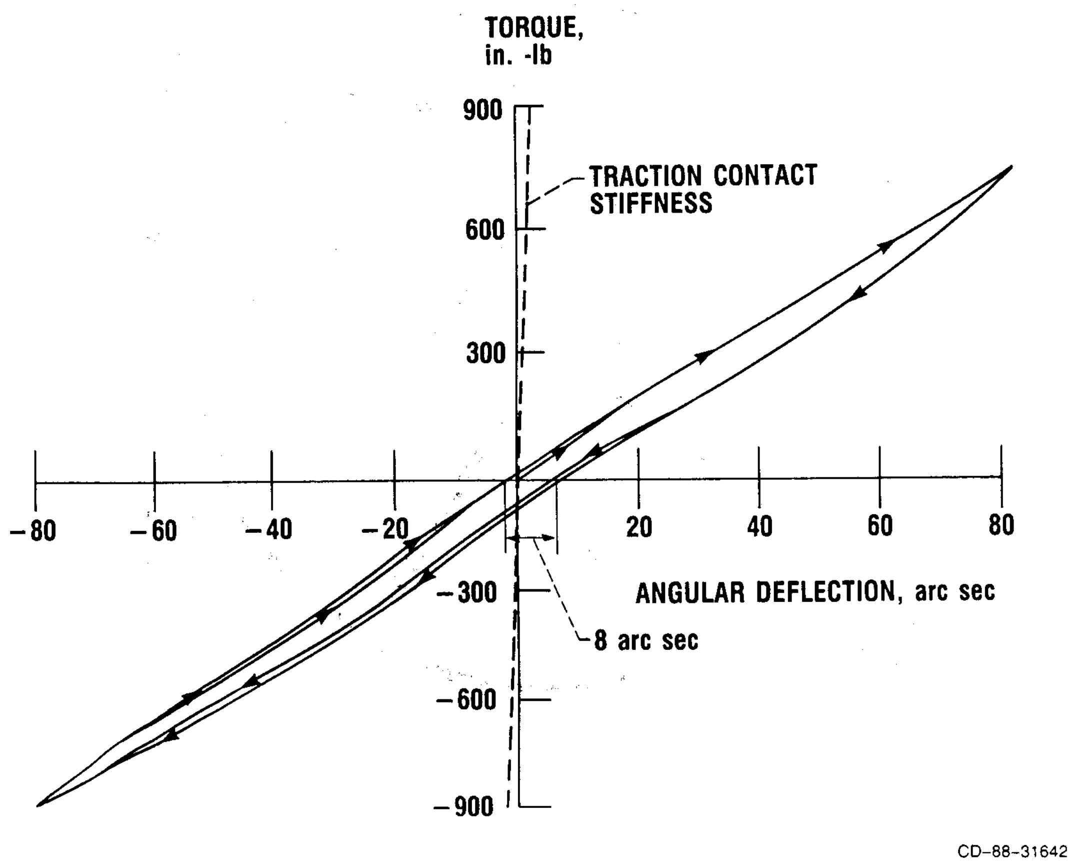
ROLLER ACTUATOR OUTPUT STIFFNESS AND HYSTERESIS

The drive's static torsional stiffness and hysteresis were measured at the output shaft. The slope of this curve reveals a torsional stiffness of 170 000 ft-lb/rad. This stiffness is just over two-thirds of the design value, or one-third of the predicted value.

Note, however, that the slope of the curve is constant across the zero torque line indicating that no backlash is present - a decided benefit for mechanism control systems that typically must position a load around a desired set point. Backlash would appear in this trace as a horizontal or "zero stiffness" line.

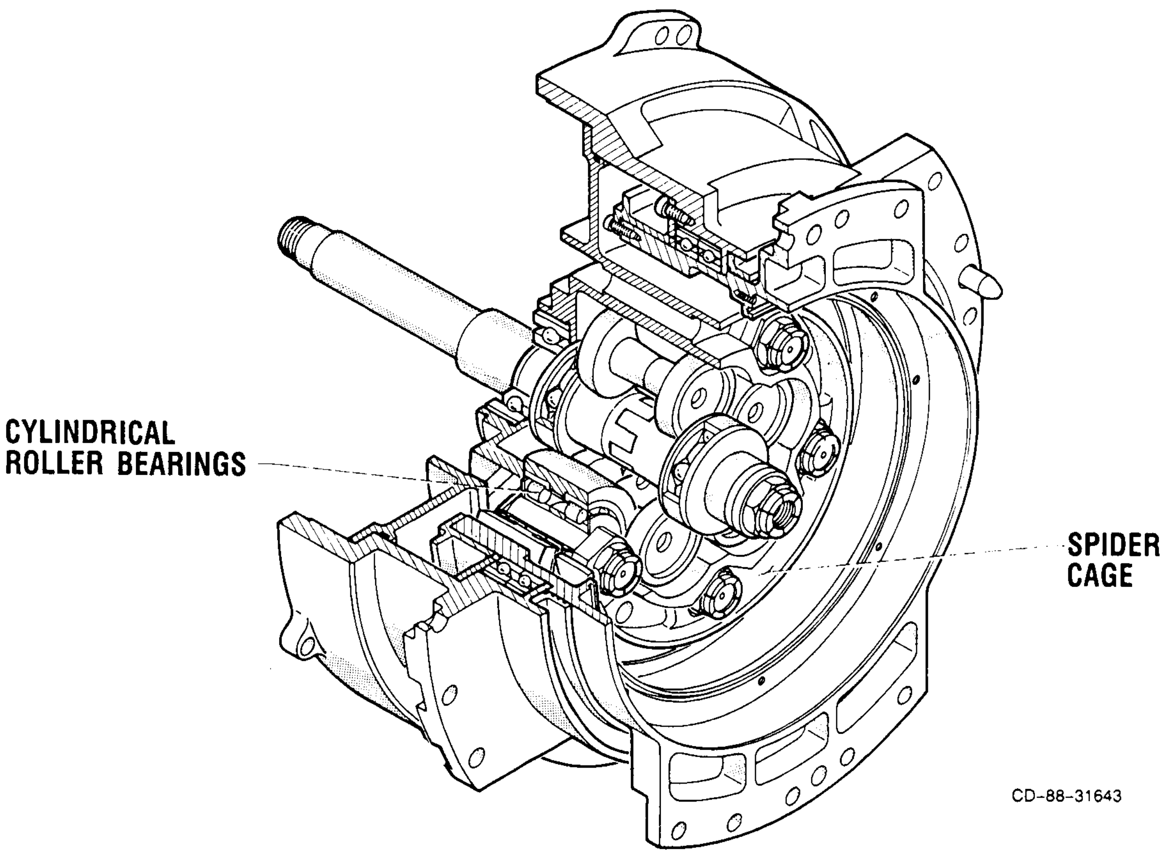
Inelastic displacements from contact microslip resulted in a small 8-arc-sec hysteresis loss during torque reversals. With feedback control systems, which are standard for these types of space vehicle actuators, this small hysteresis is considered acceptable.



MAIN SOURCES OF STRUCTURAL COMPLIANCE

Individual stiffness measurements of key drive components were made to investigate the discrepancy between the measured and predicted drive system stiffnesses. The two largest contributors to drive compliance are the spider-cage support structure and the second-row planet support bearings. The stiffness of the spider cage was measured while installed in the drive by applying a static torque to the output shaft and measuring the angular deflection of the spider-cage about the centerline of the drive. The measured spider-cage support stiffness reflected to the output shaft was approximately half that predicted analytically using the simple beam model. Evidently, a detailed finite element model would be needed to provide better stiffness estimates for the relatively complicated spider cage.

Radial stiffness measurements of four of the drive's cylindrical roller planet bearings were made in a specially designed loading fixture with the same setup that exists in the second-row planet rollers. Near zero load, the measured radial stiffness of the cylindrical roller bearing was less than a third of that predicted by standard roller bearing theory (Harris, 1966) because of an experimentally observed settling-in phenomenon not reflected by the bearing deflection model.

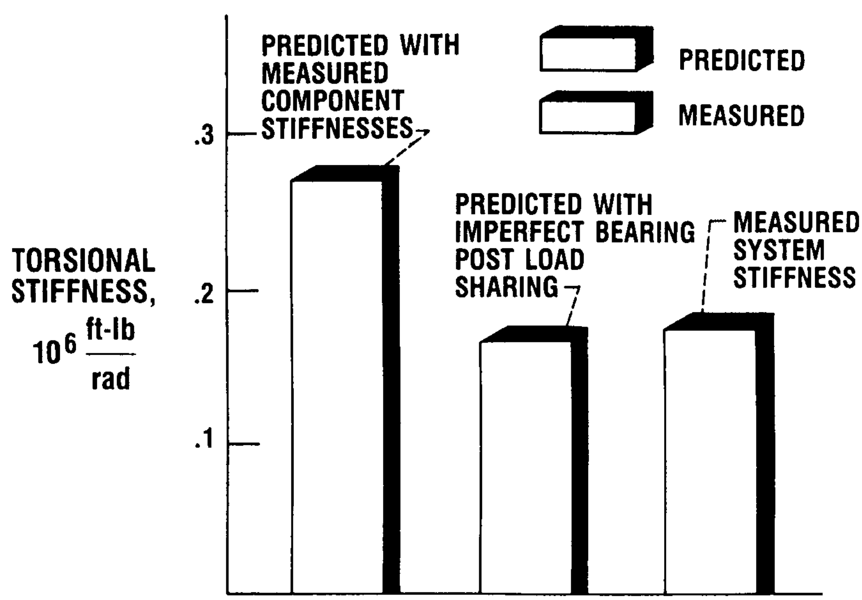


DRIVE SYSTEM STIFFNESS

The measured stiffnesses of the spider-cage support structure and the planet bearings were used in place of their original predictions (left bar shown in figure) to recalculate an overall predicted drive stiffness at zero-torque load. This stiffness of 270 000 ft-lb/rad is now 60 percent higher than that measured for the drive system.

Because of manufacturing tolerances in bearing post locations, it is possible the bearings on the test drive are out of perfect position. Thus, at initial load application only one or two of the supports may be, in fact, loaded. In view of this nonideal load sharing, a decrease in the effective planet bearing system stiffness would be expected. For instance, if it were assumed that only two of the five bearing supports were active at the initially applied torques, then the effective bearing support compliance would increase by a factor of 2.5, resulting in a recalculated drive stiffness of 170 000 ft-lb/rad (center bar shown). This stiffness agrees exactly with that measured.

Based on these results, drive stiffness improvements resulting from a redesign of the second-row planet support structure were analytically considered. Machining the spider cage from beryllium with more rigid connections for planet bearing posts would be expected to improve this component stiffness by 90 percent. Using preloaded or line-to-line fit cylindrical roller bearings would remove the initial "setting in" behavior observed, giving an appreciable higher stiffness at zero load (zero drive torque).



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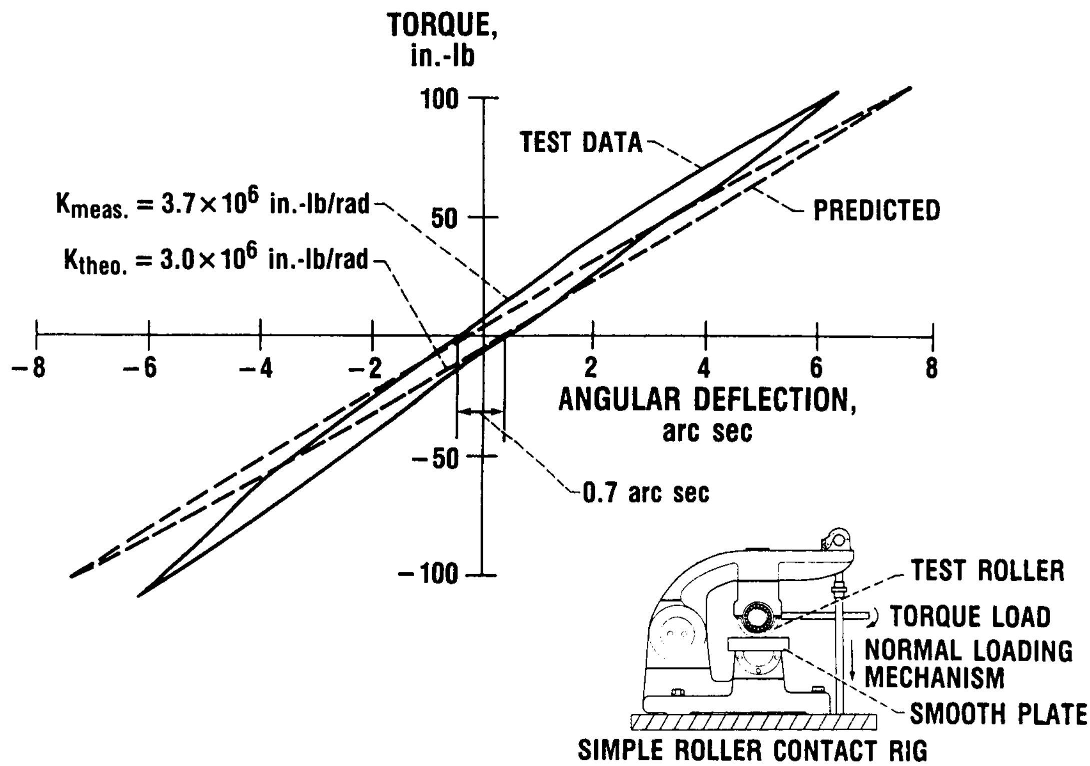
VERIFICATION OF TRACTION COMPLIANCE/HYSTERESIS MODEL

To corroborate the traction contact compliance analysis procedure, tests on a simple roller configuration were conducted at NASA Lewis. The apparatus consisted of a 3-in.-diameter crowned roller normally loaded against a flat plate. Static torque applied to the roller caused small angular displacements (resulting from tangential straining of the contact) as plotted here.

The measured and predicted contact stiffness agree well validating the analysis technique used for the overall drive. The general shapes of the curves also agree well indicating that the model accounts accurately for small inelastic microslip occurring in the contact resulting in the 0.7-arc-sec hysteresis losses shown.



SIMPLE ROLLER CONTACT RIG

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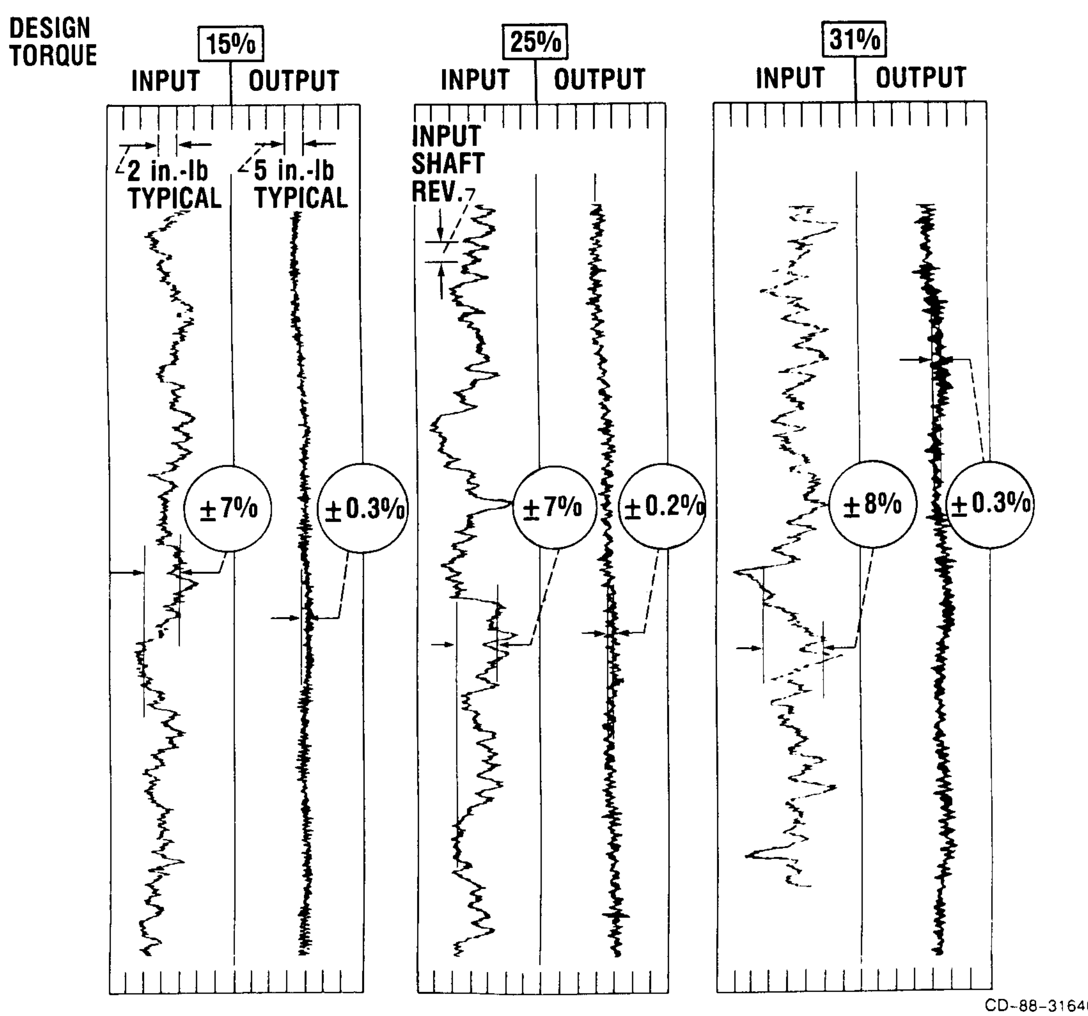
ROLLER ACTUATOR TORQUE RIPPLE

Tests were conducted to approximately determine the torsional ripple/attenuation characteristics of the roller actuator. A variable-speed dc motor drove the high-speed shaft of the actuator while steady torsional loads were applied to the output shaft.

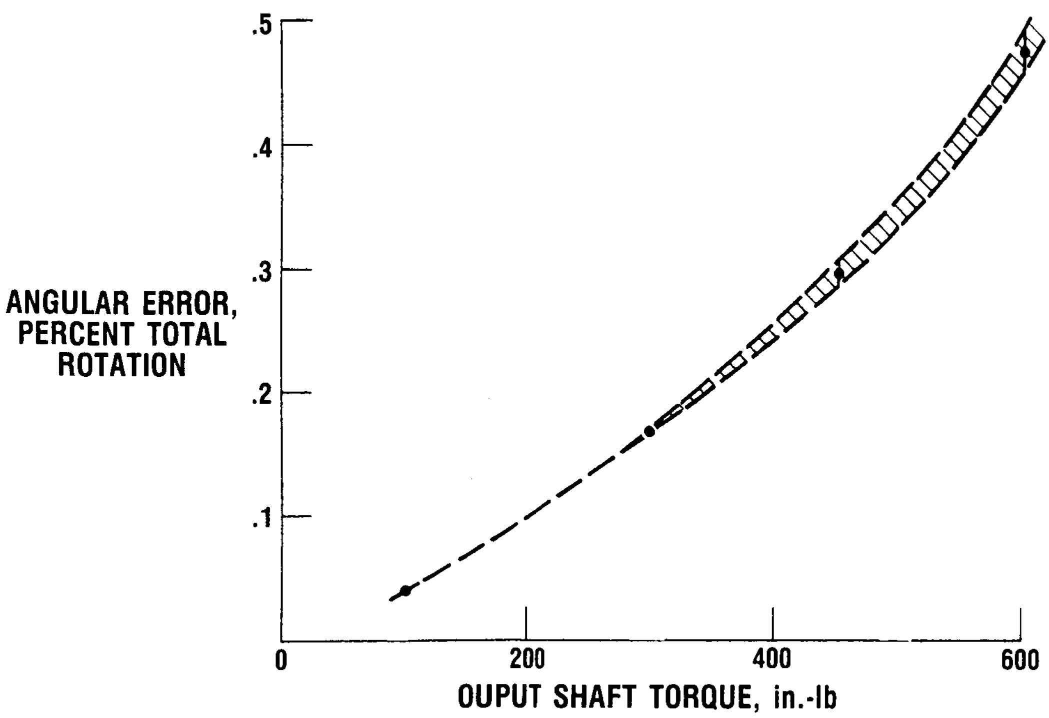
The variations of roller actuator input and output torque signatures for one complete output shaft revolution are shown here. The actuator was driven at 10 percent of maximum speed and at three torque levels corresponding to 15, 25, and 31 percent of maximum torque at 50 percent preload. Torque ripple is shown as a percent variation (plus or minus) of the noted steady-state torque. The input torque varied approximately 7 to 8 percent while the output torque varied on the order of 0.3 percent. If no attenuation (damping) was present, then input and output torque percent fluctuations would be expected to be about the same. This suggests that the drive does not excite or amplify torsional oscillations but, in fact, helps to attenuate vibration through coulombic damping. The traces shown were taken with prerun rollers having less than perfect surface condition. Thus, these traces are considered to represent a conservative view of the smooth torque-transmitting capability of the test drive.



OPEN-LOOP POSITIONAL ACCURACY

A simple test was devised to determine the positional accuracy of the test drive under load in an open-loop control mode (point A to point B and back to point A). Tests were conducted by driving the input precisely 64 revolutions under four steady torque load levels at constant speed in one direction. The system was then "unwound" by rotating the input shaft back to its initial position while maintaining torque in the same direction. Dividing the difference in output shaft angular position before and after rotation by the 8 (128 input revolutions/drive ratio) total output rotations made, results in the percent angular error shown.

These small errors in reproducing commanded input position are caused by two unavoidable characteristics of roller drives. The phenomenon of rolling creep under torque loads is the major contributor to open-loop positional inaccuracy. As each pair of rollers roll over each other under a steady torque, there is a small relative speed difference which is seen at the output as lost motion. At low torques, or when the drive operates unloaded, very small kinematic errors due to imperfectly ground rollers (diameter tolerance, out-of-roundness, lobing, waviness, etc.) can be present. Hence, for critical point-to-point positioning applications the control system must be closed loop in order to feedback output position when using roller actuators. This is not uncommon for such precision positioning mechanisms.



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SUMMARY AND CONCLUSIONS

The suitability of roller drive technology for key space vehicle actuator needs has been shown. The drive system evaluated herein exhibited many required features of precision positioning mechanisms, including absence of mechanical backlash with minimum hysteresis, high-torque capability in a small, lightweight package, and the ability to run smoothly and operate without a liquid lubrication system.

The absence of mechanical backlash or deadband improves the resolution of the system under load-reversal conditions and simplifies the control system. The planetary drive configuration packs a high ratio in a small design envelope. The smooth running and attenuating characteristics are ideal for sensitive minimum vibration positioning applications. The traction rollers have inherently high torsional stiffness (important for high system bandwidth) since torque is transmitted through tangential shearing of the traction interface. Overall drive system stiffness, however, was compromised by support structure compliance. Operating the drive dry with only a thin gold film applied to the traction surfaces prevents cold welding of the rollers in a space vacuum environment and eliminates the need for cumbersome liquid lubrication systems that must pass difficult space qualifications tests.

- **ROLLER ACTUATOR EXHIBITS ZERO BACKLASH WITH ARC SEC HYSTERESIS ERROR**
- **INPUT TORQUE RIPPLE IS ATTENUATED BY COULOMBIC DAMPING**
- **ROLLERS HAVE INHERENTLY HIGH TORSIONAL STIFFNESS COMPROMISED BY STRUCTURAL COMPLIANCE**
- **GOLD FILM LUBRICATION APPEARS FEASIBLE**

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