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## APPLICATION OF TRACTION DRIVES AS SERVO MECHANISMS

by

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## ABSTRACT

The suitability of traction drives for a wide class of aerospace control mechanisms is examined. Potential applications include antenna or solar array drive positioners, robotic joints, control moment gyro (CMG) actuators and propeller pitch change mechanisms. In these and similar applications the zero backlash, high torsional stiffness, low hysteresis and torque ripple characteristics of traction drives are of particular interest, as is the ability to run without liquid lubrication in certain cases. Wear and fatigue considerations for wet and dry operation are examined along with the tribological performance of several promising self-lubricating polymers for traction contacts. The speed regulation capabilities of variable ratio traction drives are reviewed. A torsional stiffness analysis described in this study suggests that traction contacts are relatively stiff compared to gears and are significantly stiffer than the other structural elements in the prototype CMG traction drive analyzed. Discussion is also given of an advanced turboprop propeller pitch change mechanism that incorporates a traction drive.

## INTRODUCTION

An interesting but not particularly well known class of speed changing mechanisms utilize traction as the means to transfer torque. Applications range from dry contacts such as the locomotive wheel against the rail and elastomer coated rollers in paper handling equipment to lubricated contacts in industrial adjustable speed traction drives.

As power transmissions, few mechanical drives match their low noise, smooth torque transfer characteristics and speed regulating accuracy. However, their suitability as servo drive mechanisms has not been given wide attention. For some control system applications, their ability to provide a smooth transfer of motion with relatively low hysteresis losses and high torsional stiffness while producing no detectable "backlash" upon direction reversal are obviously beneficial qualities. On the other hand, traction drives are unsuitable for point-to-point (PTP) positioning applications where it is impractical to feedback output position.

The discussion which follows is intended to provide greater insight into some of the performance characteristics of traction drives which may lend themselves to certain servo mechanism applications. A number of examples will be cited, as known to the authors, to help illustrate some of the prior art along with a few applications presently under consideration. Due to the broad

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nature of this technology and the space limitations set forth here, this review can only hope to couch upon some of the more relevant design considerations. While published information on traction drives is considerably more limited than other types of transmissions, particularly in terms of servo mechanism applications, a significant design data base does exist, as can be found in references 1 to 3.

#### BACKGROUND

The earliest of speed changing mechanisms used smooth wheels of unequal size in frictional contact. According to his writings, Leonardo Da Vinci devised and experimented with several machines which incorporated frictional wheels. One such machine was for the grinding of lenses or polishing of mirrors where the repetitive toothing errors from gearing would produce imperfect surfaces.

Because of their simplicity and their unique ability to smoothly and continuously regulate speed, friction drives or traction drives, as they are now known, found use in a host of applications. Some early examples include wood working machines (circa 1870) in which the wood feed rate was regulated with friction drive rollers and turn-of-the-century motor cars equipped with continuously variable, friction disk transmissions (ref. 1). More modern examples of feed regulating and loading applications can be found in paper and film handling equipment including printing and copying machines and computer plotting devices. In these applications, the processed material is normally "pinched" between an elastomer-coated drive roller and another that freewheels, such as that used in multicolor printing operations to register the paper. Analogous space vehicle applications include a pinch roller drive mechanism for deploying a spool-wrapped antenna for the Helios B satellite (ref. 4) and one that dispenses and stores a transfer boom for Skylab (ref. 5).

Philosophically, "linear" traction drives include a wide class of vehicles ranging from bicycles to locomotives. This later example of the wheel/rail contact vividly illustrates that traction power transfer need not be limited to trivial load levels. In fact, advanced, light-weight helicopter transmissions (Fig. 1) which incorporate steel traction rollers lubricated with synthetic traction fluids have been tested at the authors' laboratory at power levels to 370 kW (ref. 1).

Although not a mature discipline, design techniques for traction drives have made good progress in recent years. Based on lubrication, fatigue and performance models developed originally for rolling-element bearings, analytical methods are now available to size traction drive contacts for a specified reliability level and to make reasonable estimates of power loss and traction capacity limits. Metallurgical improvements in the fatigue resistance of bearing steels coupled with the commercial introduction of high traction fluids have increased the power capacity of traction drive contacts by several fold. While an extended discussion of these technical advancements is not central to this review, helpful design information can be found from several sources (refs. 1 to 3).

## TRACTION DRIVES AS SPEED REGULATORS

Apart from their basic mechanical simplicity, the most sought after characteristic of traction drives is their ability to provide mechanical stepless speed control over wide ratio ranges, with some units being infinitely variable. In this regard, traction drives have been performing light to medium duty (up to 75 kW) speed matching for industrial machinery for more than 50 years (refs. 1 and 2). Applications range from knitting and fiber-spinning equipment to milling machines and high-speed grinders (ref. 2). There are more than 2 dozen commercial manufacturers of variable ratio traction drives, with some designs dating back to the 1920s. Operating lives of 20 years or more are not uncommon in factory environments.

A sample of commercially available, adjustable speed traction drive geometries appears in Fig. 2. Speed regulation is achieved by altering the rolling radii ratio between input and output member. This is accomplished by either tilting or translating an intermediate transfer element or through direct translation of input or output members. Normally a manual vernier control element is used for this positioning, although feedback control has been used on some aerospace applications. With manual control, output speed regulation and repeatability is often with  $\pm 0.1$  percent under uniform loading. Under extreme no-load-to-full-load fluctuations, maximum output speed variations are typically 1 to 2 percent. However, with suitable servo controls, accuracy can be restored back to the 0.1 percent level (ref. 6).

## TRACTION CURVE

The speed variation between traction rollers due to torque transfer is generally referred to as "creep" (ref. 3). Traction drives are designed to operate along the linear ascending portion of the traction curve (Fig. 3) at some point below the peak. A ball or roller-ramp loading mechanism (Fig. 4) is commonly used to increase the clamping force between roller elements in proportion to the transmitted torque. The angle of the ball ramp is tailored to the lubricant-contact condition. It is selected so that the ratio of the applied tangential to normal force, or traction coefficient, is about 20 to 30 percent less than the maximum available traction coefficient under the least favorable operating condition. This provides some margin against slip. With a torque sensitive loading mechanism, gross slip will not occur, even at torque levels which exceed the elastic limit of the rollers or structural material. This loading action is not unlike the behavior of a roller ramp or sprag-type clutch. Improved part load efficiency and extended service life are additional benefits from modulating the clamping force with the demand load. Maximum efficiencies of adjustable speed traction drives typically range from 85 to 93 percent. However, multiple contact fixed ratio traction drives have efficiencies of about 96 to 97 percent, with single contact systems being as high as 99 percent.

Operating creep rates of traction contacts are relatively small. They range from 0.1 to 0.2 percent for dry contacts or those lubricated with traction fluids at low speeds (Fig. 3), to 3 or 4 percent for lightly loaded, high speed contacts lubricated with mineral oils. This speed difference is

not due to slip between driver and driven roller but is, in fact, the accumulated lost motion due to the tangential stretching and compressing motion or compliance occurring at the roller contact interface. It is well known that a "locked" or "zero slip" region exists at the leading edge of the contact and only at the peak traction point, i.e., point of impending slip, will this region completely disappear. Nevertheless, this lost motion due to creep precludes the use of traction drives for open-looped positioning devices. However, open-looped speed regulation accuracies of  $\pm 0.5$  percent or better are achievable with an automatic loading mechanism over a wide range of speeds and torques based on test data for a 14:1 fixed ratio traction drive reported in reference 7.

It is instructive to note that the creep or slide-to-roll rates associated with traction drives are generally one to two orders of magnitude smaller than those associated with gear teeth as they enter and leave mesh. Under difficult to lubricate conditions, such as those that exist for most space mechanisms, the fundamental rolling nature of a traction contact is a decided benefit in terms of reduced wear and heat generation. This was clearly demonstrated at the authors' laboratory where a 10.8:1 ratio planetary traction drive (Fig. 5) survived 1 hour of repeated full power acceleration (11 kW at  $9 \times 10^4$  rad/sec<sup>2</sup>) tests from 0 to 70,000 rpm while submerged in LOX (ref. 1). Test life was equivalent to  $2 \times 10^6$  revolutions of the high-speed shaft. Expected gear life under comparable conditions would be a maximum of about 15 or 20 minutes due to excessive wear under this hostile lubrication condition. Furthermore, a similar but 120:1 ratio traction drive was tested fully preloaded for 43 hours at 360,000 rpm without failure using no liquid lubrication whatsoever (ref. 1). Such speeds are, in all likelihood, beyond the realm of even oil lubricated gears.

## DURABILITY

### Fatigue

The failure modes of traction drive contacts are very similar to those of ball and roller bearings. This is not surprising since the operating stresses, materials (normally hardened bearing steels), contact geometries and surface finishes as well as lubrication conditions are also very similar. As with rolling-element bearings, fatigue or pitting is the likely failure mode when the quality of lubrication is good, that is when the ratio of lubricant film thickness to composite surface roughness or lambda ratio is above about 1.5. Statistical fatigue life can then be predicted from a modified form of the Lundberg and Palmgren theory that is traditionally used to determine service life ratings of commercially available ball and roller bearings. Details of this method can be found in reference 3. In general, fatigue life  $L$  is related to size, torque  $T$  and available traction coefficient  $\mu$  as follows:

$L \propto \text{size}^{8.4}$  at constant  $T$  and  $\mu$

$L \propto T^{-3}$  at constant size and  $\mu$

$L \propto \mu^3$  at constant size and  $T$

Thus a 9 percent increase in size or a 26 percent increase in  $\mu$  will double the fatigue life at constant torque capacity or, in turn, cause a 26 percent increase in torque capacity at constant fatigue life.

#### Wear

As the quality of lubrication or lambda ratio diminishes, the degree of asperity contact and surface distress increases and the failure mode shifts from predominantly fatigue to one that is predominantly wear. It is this mixed or boundary lubrication regime where most low speed space mechanism contacts must operate.

Solid lubricant films formed from graphite,  $\text{MoS}_2$ , PTFE or those from soft metals such as lead, gold or silver are potential substitutes for liquid lubricants and greases in the low lambda regime. The general increase in friction coefficient attendant with some dry film lubricants is obviously beneficial for traction drive contacts, allowing some reduction in the required clamping loads. However, the normal concerns of using dry film lubricants still remain, namely their non-renewable nature, the integrity of the film/substrate bond and the detrimental accumulation of wear debris. Also, the somewhat erratic nature of a dry film's friction coefficient may preclude their use for certain torque-sensitive mechanisms. In such cases, low vapor pressure greases, such as the perfluoroether type, are normally preferred provided that potential contamination of optical surfaces are not a factor and evaporation rates are within operational life requirements. hard-faced, ion-sputtered or vapor-deposited coatings such as TiN and TiC avoid some of these problems (ref. 3), but their long-term fatigue resistance still requires better definition.

Polymers/elastomers. - Perhaps the most promising self-lubricating materials for traction drive actuators are some of the polymers and elastomers that have been successfully used for a variety of industrial roller type components. These range from molded polyurethane casters and wheels to polyamide (nylon), acetal and polyamide-imide (Torlon) thermoplastic bearings and gears. Many of these materials can be reinforced with glass fibers for strength and filled with graphite or PTFE for reduced friction. In addition to their self-lubricating abilities, these maintenance-free materials offer significant vibration damping, quietness of operation, low weight and corrosion resistance. The general tribological properties of a large number of these polymers is relatively well defined (ref. 9), but their specific performance as traction drive materials for use in space mechanisms is not as well understood. Specifically, their cold temperature and vacuum, traction and wear characteristics need better definition as does their stress relaxation (cold flow), radiation resistance, and outgassing behavior. Because contamination by liquid lubricants can markedly reduce their traction capability, use of these materials must be restricted to relatively clean environments, not unlike those required for other types of precision mechanisms. Despite these concerns, a surprisingly large class of polymeric materials have been applied successfully as structural, protective, dielectric

and sealing materials on spacecraft (ref. 10). Furthermore, because of their ability to self-lubricate and to resist cold welding even in a hard vacuum, they have also been used in mechanisms at points of sliding. Typical examples include phenolic ball bearing cages impregnated with dry or liquid lubricants and fine-pitched gears made from polyamide or acetal resin.

The load capacity of some commercially available polymers in sliding/rolling contact is surprisingly large. For example, table I lists some of the results of test programs sponsored by the authors' laboratory to define the traction capacity limits of several engineering thermoplastics. In these tests, the rolling failure load limits of six different thermoplastic rollers 50 mm in diameter in contact with equal sized steel rollers were determined in air at ambient temperature and pressure. From the rolling tests, two of these materials, an unfilled polyamide-imide and a cotton fiber phenolic, were selected for torque transfer testing near their peak traction capacity limits. The polyamide-imide resin withstood a remarkable 220 N/mm (1250 lb/in) of unit normal loading at a rolling speed of 10 m/s with 2 percent sliding for 30 minutes of testing without damage. A narrow roller, just 5 mm wide, of this material transmitted 4.8 kW of power continuously for 10 minutes at 20 m/s without failure. While the performance of some of these commercially available thermoplastics clearly exceeds any of those anticipated for servo mechanisms applications, these results do suggest that plastics could be used in place of steel for certain low-cost, moderate power, traction speed regulators.

Polyimides are another class of polymer which offers promise as a low-wear, high-traction material. Polyimide films have excellent UV radiation resistance and vacuum stability, being used extensively on spacecraft for thermal blanket insulation and a variety of external semistructural applications (ref. 10). In addition, some polyimides have unusually high-sliding wear resistance. For example, ambient pin-on-disk experiments with an experimental, partially fluorinated polyimide pin sliding against a steel disk showed it to have an average friction coefficient of 0.8 and an average wear rate of  $5 \times 10^{-15}$  m<sup>3</sup> per meter of sliding at sliding speeds of 2.7 to 3.1 m/s (ref. 11). At this wear rate, it would take  $1 \times 10^9$  revolutions for a 10 mm wide, relatively heavily loaded traction roller made of this polymer to wear off a 1 mm thick layer of material, assuming a relatively high 1 percent slide-to-roll ratio.

Another potentially useful class of materials for traction rollers are vulcanized natural and synthetic rubbers. Some guidelines for the speed/load limits of these elastomers for general traction drive service can be found in reference 3. Polyurethane rubbers are particularly durable, having been used in difficult service conditions such as when molded to forklift truck wheels and subway train tires. Furthermore, polyurethane molded rollers have already been used without failure for one type of traction drive, space qualified, rotary actuator known to the authors. Its traction capacity in a vacuum is high. Sliding friction tests of a molded polyurethane wheel, meeting MIL-R-3065 specifications, exhibited a friction coefficient from 0.7 to 1.3 against an aluminum or magnesium concave slider in a vacuum of  $10^{-5}$  torr at room temperature (ref. 12). Since all elastomers stiffen to some extent at

low temperatures, working traction coefficients would have to be less than this if a temperature controlled environment was impractical.

### SERVO MECHANISM APPLICATIONS

While traction drives have had a well established history of use as power transmitting speed changers, their use in servo systems has not been as broad in comparison. This is somewhat unexpected since the typical low power, low speed, low backlash and low starting friction requirements of such applications are particularly well suited to the capabilities of traction drives. In many positioning applications such as CMG gimbal drives, the torques may be high but because the speeds are typically low, the buildup of wear or fatigue stress cycles is also low. In addition, the available traction coefficient of a lubricant is greater at low operating speeds. Under these conditions, the traction drive elements can be made relatively small and still provide adequate service life.

Furthermore, smooth rollers in driving contact avoid the meshing errors and torque discontinuities of gear teeth as the load is transferred from one pair of teeth to the next. Since rollers are always in continuous driving contact no matter the direction of tangential loading, backlash is precluded. However, the accumulation of microslip in the contact gives rise to a hysteretic type torque-deflection loop that requires some form of feedback control for high precision positioning systems. As previously discussed, speed regulation accuracies as low as  $\pm 0.1$  percent can be achieved manually under favorable conditions, although about  $\pm 1$  percent is probably more representative of a well-designed, open-loop traction drive.

#### Representative Servo Applications

One prominent example of a traction type aerospace servo mechanism is the Lucas, constant speed drive flying on the AV8-A Harrier VSTOL jet fighter operating in North America and Europe (ref. 1). This toroidal-type roller drive, similar in geometry to that shown in Fig. 4, maintains a constant 400 Hz frequency a.c. generator output within  $\pm 1$  percent from a variable turbine speed input which ranges from 3250 to 8000 rpm. It delivers 13 kW for driving aircraft generators having output ratings up to 30 kVA. It does so with a power-to-weight ratio that is 10 percent better than comparable equipment with considerably lower maintenance costs. In the 50s, Avco Lycoming offered a similar line of toroidal traction drive type constant speed drives. These flew on several commercial and military aircrafts, including the Lockheed C-121, Douglas A-4E and Grumman S-2D. The servo mechanism performance characteristic of toroidal traction drives were investigated in reference 6. Experimental performance data on a 1 hp prototype model indicated that a velocity control accuracy of  $\pm 0.1$  percent can be readily obtained with transient-response times varying from 20 to 40 ms depending on signal amplitude and load conditions.

Another example of a speed regulation traction mechanism, but a nonservo type, is a ball type continuously adjustable variable ratio drive which

synchronizes the digital counter readout of a commercially available liquid flow volume meter. The drive couples the positive displacement fluid rotor to the meter's counter. A vernier adjustment screw is used to vary the speed ratio in 0.02 percent increments between the metering rotor and the counter so that accurate readings are obtained when the meter is calibrated against some "master" meter or other liquid volume test standard.

Traction type linear actuators are also commercially available. These units are useful for a wide range of positioning and motion applications particularly where jamming and overload protection are required. In one design, three equally spaced, skewed rollers located on each end of a two piece, spring-loaded, clamping block are loaded against the surface of the smooth, rotating drive shaft. The angle of roller skew relative to the drive shaft axis is directly related to the amount of linear travel of the block per drive shaft revolution or "lead." The thrust capacity at slip can be easily varied by adjusting the screws which vary the spring clamping load. If the linear element would hit its stop or jam, the rollers would automatically slip at the set thrust level and be automatically reset for operation when the jam was relieved.

#### OPERATING CHARACTERISTICS

In comparison with gear type servo mechanisms, traction drives possess several desirable operating characteristics. These can be summarized as follows:

- (1) Zero backlash
- (2) High torsional stiffness
- (3) Low starting friction or breakaway torque
- (4) Low torque ripple
- (5) Low velocity errors
- (6) High torque density and compactness
- (7) Infinite resolution of speed or ratio adjustment

In addition to these features, the roller contact due to its low sliding nature can be designed to operate for extended periods of time without liquid or grease lubrication. As previously discussed, this can be accomplished by either using high traction solid film coatings such as ion-plated gold or silver or using advanced, low-wear polymers such as partially fluorinated polyimides or polyamide-imides. The elimination of a depletable liquid lubrication is a decided benefit for spacecraft mechanism applications.

Furthermore, rollers, unlike gears, have the ability to harmlessly slip at predetermined traction limits. The over torque, release-clutch-tendency can prevent catastrophic damage if jamming should occur at some point in the mechanism drive train.

Finally, in the case of dual fault or multiply redundant servo drive mechanisms, such as those for some spacecraft applications, independent, multiple drive rollers can be readily engaged and disengaged from drive



systems with conventional linear actuators without the need to stop the system from rotating or the need to provide proper tooth mesh phasing.

#### Stiffness

To achieve minimum response time for a point-to-point motion control or to increase responsiveness of a servo controller in general, it is necessary to maximize drive system torque-to-inertia ratio and stiffness. In reference 13 a method was presented for analyzing the torsional stiffness and hysteresis effects of a traction drive contact of arbitrary geometry under a wide variety of loading conditions.

Torsional stiffness  $K_T$  was found to increase with an increase in normal load  $N$ , available traction coefficient, shear modulus of the material  $G$  and roller radius  $R$  while decreasing with an increase in transmitted torque  $T$ . The exact expression for the stiffness of a crowned roller pair of equal size was given as:

$$K_T = \frac{\pi G a}{\Gamma} \left( 1 - \frac{T}{\mu N R} \right)^{1/3} R^2 \quad (1)$$

where  $a$  is the semiwidth of the elliptical contact and  $\Gamma$  is a contact geometry parameter.

Another finding of the study performed in reference 13 was that the traction contacts themselves are considerably stiffer than the bearings and other structural elements in a complete traction drive system, typically accounting for only several percent of the drive's total compliance.

A comparison of the theoretical stiffness of equally sized gear and traction roller pairs made of steel showed that traction contacts were typically two or three times stiffer than gears under comparable loads. This is illustrated in Fig. 6 where the stiffness of traction contacts relative to comparably sized and loaded gears is plotted as a function of gear tooth size (diametral pitch). The method used to analyze gear stiffness considers the local Hertzian normal compliance and tooth beam bending. It also considers standard undercut and fillet bending and shear as well as foundation flexibility (ref. 13).

It is apparent from Fig. 6, that gear mesh stiffness is relatively insensitive to the number of teeth or the torque level and fluctuates between two relatively discrete levels as the load is transferred between single tooth and double tooth contact. Apart from the kinematic (unloaded) errors, this stiffness fluctuation contributes to velocity fluctuations in a loaded gear train. In contrast, the stiffness of a traction drive contact is essentially constant for a given load, although some degradation with increased transmitted torque will occur as shown.

## Hysteresis/Backlash

A major advantage of a traction positioning mechanism is the absence of backlash. However, due to the inelastic displacements or microslip that occurs in certain regions of the contact, some hysteretic losses will be present during torque reversals. A comparison of predicted and measured hysteresis and stiffness of a 9.5 mm steel ball against a flat appears in Fig. 7. The experimental data were taken from the investigation appearing in reference 14. For a maximum applied torque value of 100 N-mm, which is 67 percent of the peak slipping torque, the width of the predicted hysteresis loop was 8.2 arc seconds.

In the case of gears, the total lost motion along the displacement axis at zero torque is due to backlash. As a practical matter some backlash allowance is required for all gear sets to prevent meshing interference from gear runout, tooth-to-tooth spacing errors, center distance variations, and thermal and structural deflections. This group of backlash sources is also responsible for the bulk of transmission error (gear train position error). Increasing the quality or precision of the gear set as well as using finer pitched (smaller) gear teeth will help to minimize but not totally eliminate both backlash and transmission error. Philosophically speaking, increasing diametral pitch to the limit, one ends up with a gear having an infinite number of infinitely small teeth, that is, a roller having neither backlash or transmission error.

For servo positioning mechanisms that "hunt," such as most pointing and tracking systems and most gimbal drives, the stiffness and backlash characteristics of the drive train at or near the torque reversal crossing point is of prime importance. At this crossing point, some means must be employed to eliminate the precipitous loss in both stiffness and output position sensitivity as the teeth move freely through the backlash clearance. Otherwise a serious discontinuity in the control system algorithm will be created. The means of eliminating or minimizing backlash take various forms, with spring loaded split gears and preloaded, auxiliary antibacklash gear trains being among the more common.

The traction contact, on the other hand, shows no such backlash discontinuity or "deadband," as illustrated in Fig. 7. In fact, stiffness is not only preserved at the zero torque crossing point but actually reaches a maximum there (Figs. 6 and 7). The noticeable but steadily continuous drop in stiffness at the higher torque levels is usually of lesser importance from a servo viewpoint. However, in some applications it may be desirable to "marry" the best servo drive characteristics of rollers and gears.

## Roller-Gear Drives

A novel geometric arrangement that combines traction rollers with gears in parallel was devised by Dr. A. L. Nasvytis in the early 1960s (ref. 15). In the roller-gear drive, as it is termed, the center portion of the roller is replaced with a spur gear. The rollers not only serve to transmit a portion of the torque but also provide a support bearing function since their diameters are equal to the pitch diameter of the gear. Helicopter transmissions of 370, 820, and 2760 kW using a multiple-row roller-gear drive planetary stage have been built and tested. Unusually high efficiencies of up to 99 percent have been recorded due to the nearly ideal positioning of the roller gear elements along the gear tooth's pitch point.

The performance of a 26:1 ratio, 400 ft-lb CMG rotary actuator which incorporated a roller-gear drive (Fig. 8) in combination with a brushless d.c. motor was evaluated in reference 16. The rollers, which share 25 percent of the torque in parallel with the gears, eliminated the effects of backlash and helped to minimize breakaway torque while adding substantial stiffness to the drive system. Static output torsional stiffness values of  $680 \times 10^3$  N-m/rad ( $500 \times 10^3$  ft-lb/rad) or greater were measured at the zero torque crossing point. Breakaway or starting friction varied from 0.4 to 2.5 percent of rated torque. While no backlash was detected, a small degree of lost motion was measured that ranged from 2.9 to 4.7 arc seconds up to 25 percent of maximum torque. Tests of a similar but smaller 106 N-m (78 ft-lb) torque, 15:1 ratio roller gear/brushless d.c. motor servo drive were conducted in reference 17.

## CMG Traction Drive

A 16:1 ratio, 435 N-m (320 ft-lb) output torque, CMG gimbal drive having two rows of stepped planet rollers is being prepared for tests at the NASA Lewis Research Center. This drive, which is shown in Fig. 9, incorporates traction rollers ion-plated with about 3000 angstroms of gold in dry rolling contact. It was designed to provide a minimum output torsional stiffness of  $680 \times 10^3$  N-m/rad ( $500 \times 10^3$  ft-lb/rad) and to survive  $1.6 \times 10^6$  operation cycles over a three year operating life. The drive itself is nominally 25 cm in diameter by 11 cm in length and weighs 10 kg. It is equipped with an automatic loading mechanism, similar to that illustrated in Fig. 4, that will be activated above some threshold torque value or initial preload level. The initial preload level can be set anywhere from nearly 0 to 100 percent of the full rated torque. High initial preload values will increase roller normal loads and, in turn, increase the initial torsional stiffness according to equation (1). However, some increase in breakaway torque will also occur. The initial preload setting for optimal response will be selected during system tests.

At the time of this writing, test results are unavailable. However, a breakdown of predicted drive component stiffness and compliance values appears in table II. It can be seen that the planet bearings are the single most torsionally soft element in the drive system, accounting for more than 33 percent of the total drive compliance. By comparison the combined effect of all the traction roller contacts themselves contribute only 14 percent of the drive's compliance.

## TRACTION DRIVE PITCH CHANGE MECHANISM

A NASA program was started in the early 1970s to develop technology for an advanced turboprop (ATP) commercial aircraft capable of accommodating up to 150 passengers at a cruise Mach number between 0.7 and 0.8. Fuel savings for an ATP aircraft are projected to be on the order of 20 percent for single rotation (SR) and 30 percent for counter rotation (CR) propeller systems relative to comparable state-of-the-art turbofan engines without a significant compromise in engine noise or emissions. A technology area that has been identified as being critical to the successful development of an ATP aircraft is a highly reliable, efficient gearbox/pitch change mechanism (PCM) propeller drive system (ref. 18).

Due to the unusually high power levels (on the order of 10,000 kW) and large size propellers (on the order of 4 m in diameter) the blade reaction torque which must be carried by the PCM is more than an order magnitude larger than previous turboprop PCM systems. In addition, a goal has been established to increase the operating reliability of the drive system from today's 4000 to 8000 hour mean time between unscheduled removal (MTBUR) to greater than 20,000 hour MTBUR. Furthermore, the propeller blade angle control system must be sufficiently precise to synchrophase the clock position of multiple engine propellers and thus minimize the "beating" noise hitting the fuselage. In view of these new and challenging requirements, PCMs which embody advanced technology features are currently being investigated.

One promising approach being studied is the electromechanical PCM illustrate in Fig. 10. This system incorporates a compact, high-reduction ratio (210:1) hybrid traction drive, powered by an advanced technology, electronically commutated, permanent magnet (PM) synchronous motor. A similarly constructed PM synchronous alternator which is colocated with the motor inside the rotating spinner provides the electrical power. This alternator receives mechanical power across the rotating interface through a power takeoff shaft extending from the main propeller gearbox. Thus, high maintenance, rotating power transfer devices such as electrical slip rings or hydraulic fluid transfer seals, in the case of hydraulic actuators, are eliminated.

The hybrid traction drive (Fig. 11) couples the high-speed servo motor (nominally 20,000 rpm at maximum blade slew rates) to the ball screw. The ball screw, in turn, articulates the individual blades through stiff links, connected at one end to an eccentric pin at the blade root and at the other end to the translating ball screw nut. The motor input to the hybrid drive occurs at the sun roller while the output to the ball screw is taken from the toolhed output ring gear. The planet carrier frame is grounded to the casing while the outer ring roller is allowed to freely rotate, serving as a "containment" ring to react the traction drive rollers' clamping loads.

## Blade Angle Control

An autonomous digital electronic control module, located in the nose of the spinner for ease of maintenance, commands blade angle position based on control signal information from the cockpit. This control signal data passes across the rotating interface through a novel fiber optic coupling with superior noise rejection and data bandwidth characteristics. The drive mechanism is actuated only when a change in blade position is commanded and then only in a low speed, hunting mode to achieve synchrophasing. Since the cruise condition occupies the bulk of the mission, the drive system experiences only intermittent periods of high slew rate when either reverse pitch or emergency feather operations are commanded. However, the drive mechanism carries nearly full torque under practically all operating conditions since it must always react the blades' centrifugal force twisting moment. In the event of PCM failure this twisting moment would back drive the blades toward a flat pitch position which, if left unchecked, could lead to a potentially dangerous overspeed condition. To prevent this situation from occurring, an independently driven pitchlock screw mechanism which tracks the position of the ball screw nut has been incorporated into the system. In the event of any mechanical or electrical component failure downstream of the ball screw, the pitchlock will lock the propeller blades within  $1^{\circ}$  to  $2^{\circ}$  of their last position and allow engine operation in a "fixed prop" mode without any danger of overspeeding.

## Hybrid Traction Drive

The traction drive was selected for this application for several reasons. Its high ratio capabilities in a compact "pancake" package enables it to be conveniently installed as a module near the nose of the spinner for ease of maintenance and/or replacement with the engine still on the wing. Because of its rolling nature, the roller drive has the ability to handle intermittent bursts of input speeds to 20,000 rpm without the need for liquid lubrication cooling. The unit can be conveniently packed with synthetic "traction" grease (formulated grease with a high coefficient of friction) and then "sealed for life."

The traction drive's low mechanical hysteresis and high torsional stiffness increases blade positioning resolution. While gear type differential speed reducers can be designed to achieve the same 210:1 ratio in a single stage, their expected efficiencies in the neighborhood of 80 percent are decidedly inferior to the mid-90 percent range expected for the hybrid traction drive. With the proposed friction, low inertia mechanism coupled to a high resolution digital controller, blade position accuracies of better than 3 arc min of a degree are projected for the system (ref. 18). Although the level of precision necessary to achieve multiengine synchrophasing has yet to be established, the PCM described here should be significantly more accurate than the current industry standard of about  $\pm 5^{\circ}$  on blade clock position.

## SUMMARY

Adjustable speed traction drives have been in commercial service for more than 50 years, performing a speed regulation function for a wide range of industrial machinery. Speed regulation accuracies typically range from  $\pm 1$  percent with open loop control to about  $\pm 0.1$  percent with feedback control or with open loop control when the loads are relatively constant.

Despite their smooth torque transfer characteristics, low rolling friction and undetectable backlash, traction actuators have not been widely applied to aerospace servomechanisms. One notable exception are the constant speed generator drives that have been in service on several commercial and military aircraft. Also "pinched" roller drive systems similar to those commonly used in printing and paper handling machinery have been incorporated in some spacecraft deployment mechanisms.

In this paper, reference is made to the traction-creep characteristics, lubrication principles, material considerations and fatigue life sizing relationships associated with traction drives. The low sliding nature of a traction contact lends itself to the application of solid film lubricants and to the use of soft metal coatings and self-lubricating polymers. Certain classes of thermoplastics, such as polyimides and polyamide-imides have low wear, high traction characteristics and can sustain very high loads without failure. Although several of these polymers have been used as structural and sliding joint materials on spacecraft, their long term tribological performance in a space environment needs greater definition.

The torsional stiffness characteristics of a traction contact were found, from the method presented, to compare favorably with those of gears, particularly at the zero torque crossing point. The typical torque-angular displacement loop of a statically strained traction contacts revealed no backlash, although a small hysteresis displacement, on the order of arc seconds, was observed. High stiffness and the absence of backlash are important to certain "hunting" type positioning mechanisms, such as CMG actuators. In this regard, "roller-gear" CMG drives where traction rollers share the load in parallel with gears have shown excellent performance. A current investigation with a pure traction roller CMG drive suggests that overall system stiffness is primarily dictated by the planet support bearings and structure. The traction contacts themselves have a relatively minor contribution.

Finally, a conceptual design is provided of an electromechanical propeller pitch change mechanism which incorporates a hybrid traction drive. This system, proposed for an advanced turboprop commercial aircraft, contains advanced electric motor and fiber optic control technology. Projected blade positioning accuracies of three arc minutes of a degree should be sufficiently precise to achieve low noise synchrophasing of multiengine propeller blades.

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TABLE I. - EXPERIMENTAL TRACTION CAPACITY LIMITS OF THERMOPLASTIC ROLLERS  
AGAINST STEEL ROLLERS AT AMBIENT CONDITIONS (50 mm diameter  
cylindrical rollers from 3 to 20 mm)

Material description	Steady state <sup>a</sup> thermal failure load limit, N/mm		Steady state <sup>b</sup> peak trans- mitted power, kW	Typical range of traction coefficient at slip
	Rolling	2 to 3 percent sliding		
Polyamide-imide resin (unfilled Torlon)	215	220	4.8	0.25-0.35
Phenolic w/cotton fibers	115	85	2.2	.3- .5
Polyamide-imide w/graphite and PTFE (filled Torlon)	145	---	---	.15- .25
Extruded Polyamide 6/6 (Nylon 6/6)	39	---	---	.15- .4
Mold-Polymerized Polyamide (Monocast Nylon)	38	---	---	.15- .25
Acetal Copolymer	32	---	---	.1- .2

<sup>a</sup>Failure observed during 30 minute test period at 10 m/s rolling speed.

Capacity limits are greater at rolling speeds less than 10 m/s.

<sup>b</sup>Power transmitted by 5 mm wide rollers for a minimum of 10 minutes at 20 m/s rolling speed.



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TABLE 11. - THEORETICAL STIFFNESS OF A 16:1 RATIO CMG TRACTION  
DRIVE AT ZERO TORQUE AND 25 PERCENT INITIAL  
PRELOAD - MEASURED AT OUTPUT

Element	Stiffness, $10^6$ ft-lb/rad	Compliance, $10^{-6}$ rad/ft-lb	Percent of total compliance
Planet bearings (cylindrical roller)	2.35	0.426	33
Input shaft, spline, and torque loader	4.52	.221	17
Spider cage (guided end condition)	5.12	.195	15
All traction contacts ( $\mu = 0.1$ )	5.44	.184	14
Roller support housing (aluminum)	6.54	.153	12
Outer housing (aluminum)	10.5	.095	7
Output shaft	29.8	.034	2
Totals	760	1.308	100

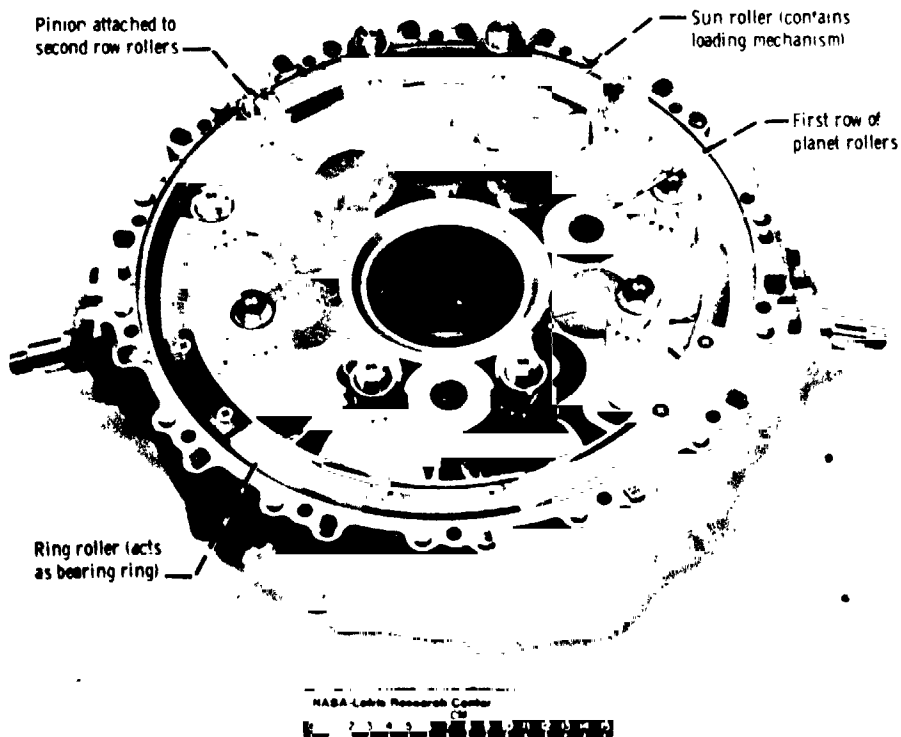


Figure 1. - 500 HP hybrid traction helicopter transmission.

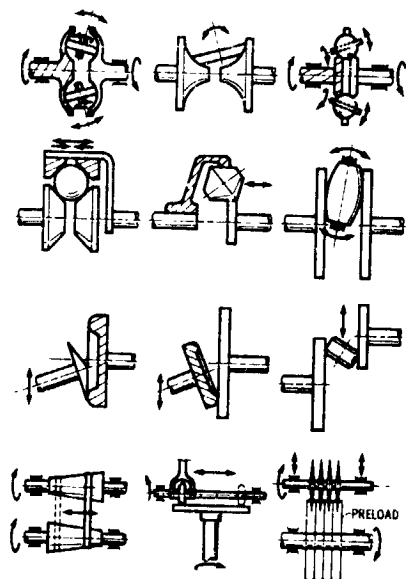


Figure 2. - Typical adjustable speed traction drive geometries.

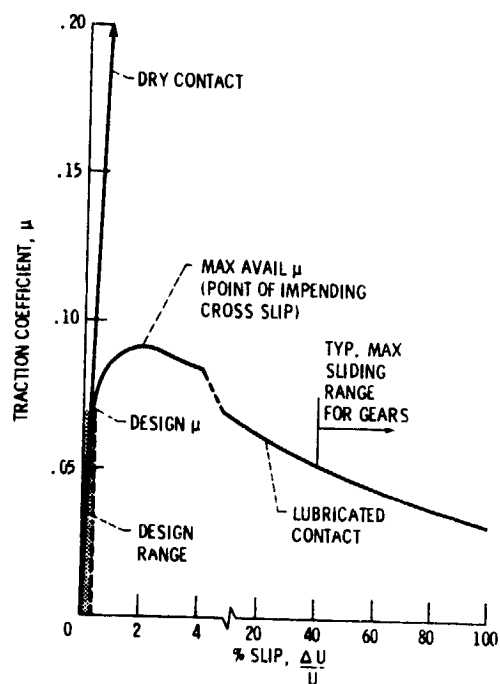


Figure 3. - Typical traction curve showing design range for liquid lubricated and dry traction drive contacts.

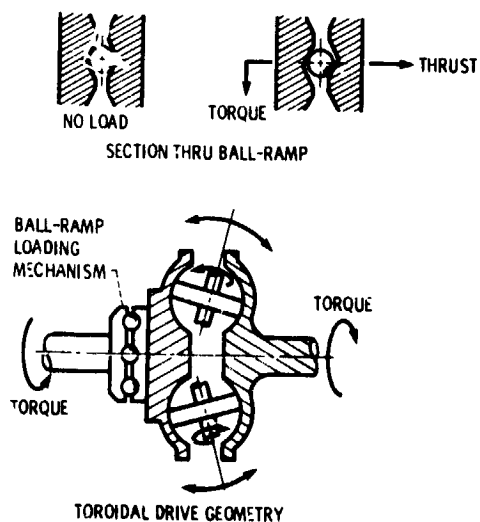


Figure 4. - Typical traction drive torque sensitive loading mechanism.



Figure 5. - Cryogenic turbopump roller drive.

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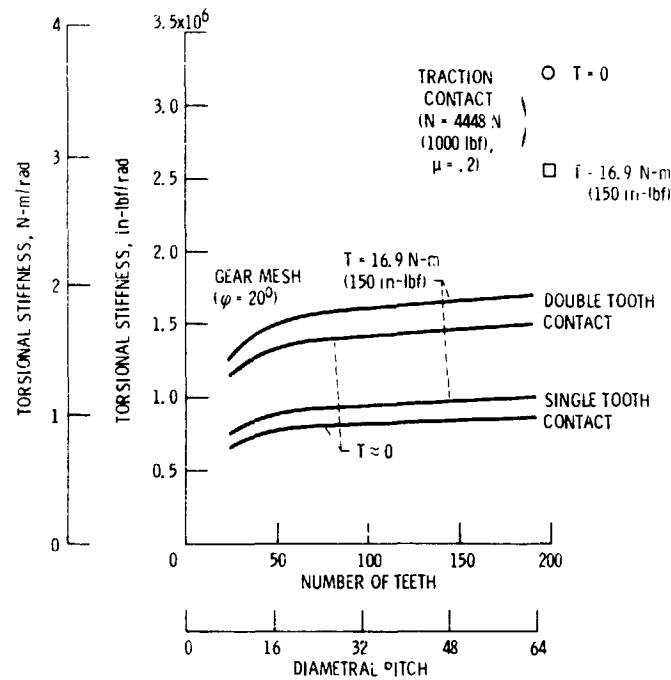


Figure 6 - Theoretical comparison of gear mesh and traction roller contact torsional stiffness at equal diameters, 76.2 mm (3.0 in), widths, 4.57 mm (.18 in) and loads.

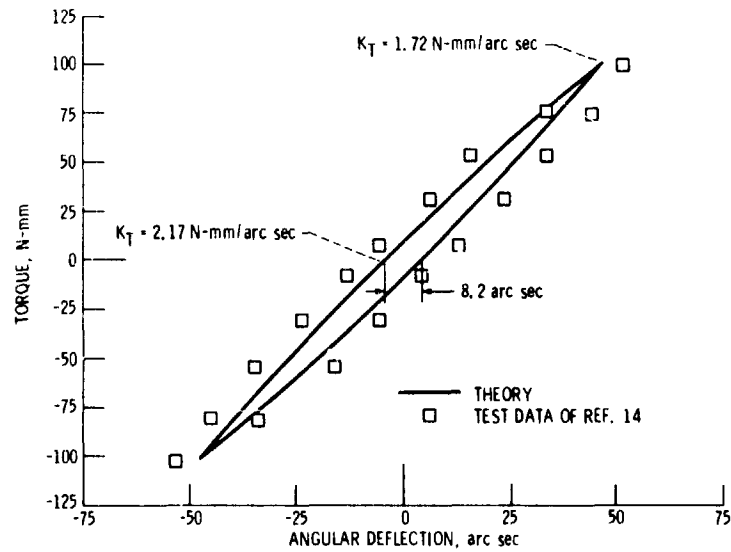


Figure 7 - Comparison of predicted and measured traction contact hysteresis and stiffness of a 9.5 mm steel ball against a flat under a 61.8 N static load from data of ref. 14;  $\mu = .51$ .

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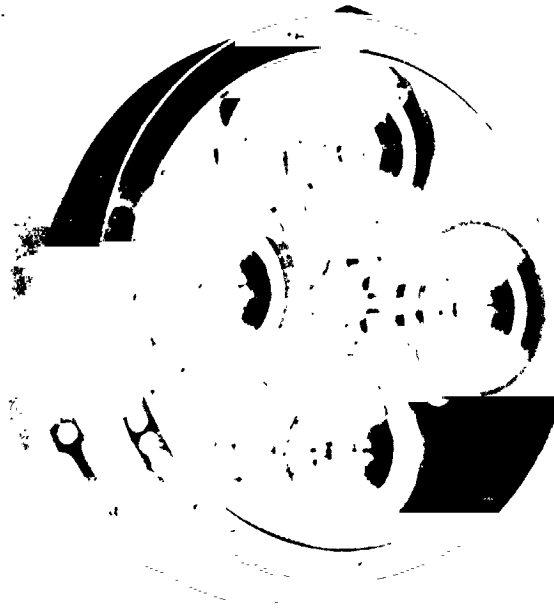


Figure 8. - Control moment gyro roller gear drive.



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Figure 9. - Traction roller control moment gyro drive having ion-plated gold rollers.

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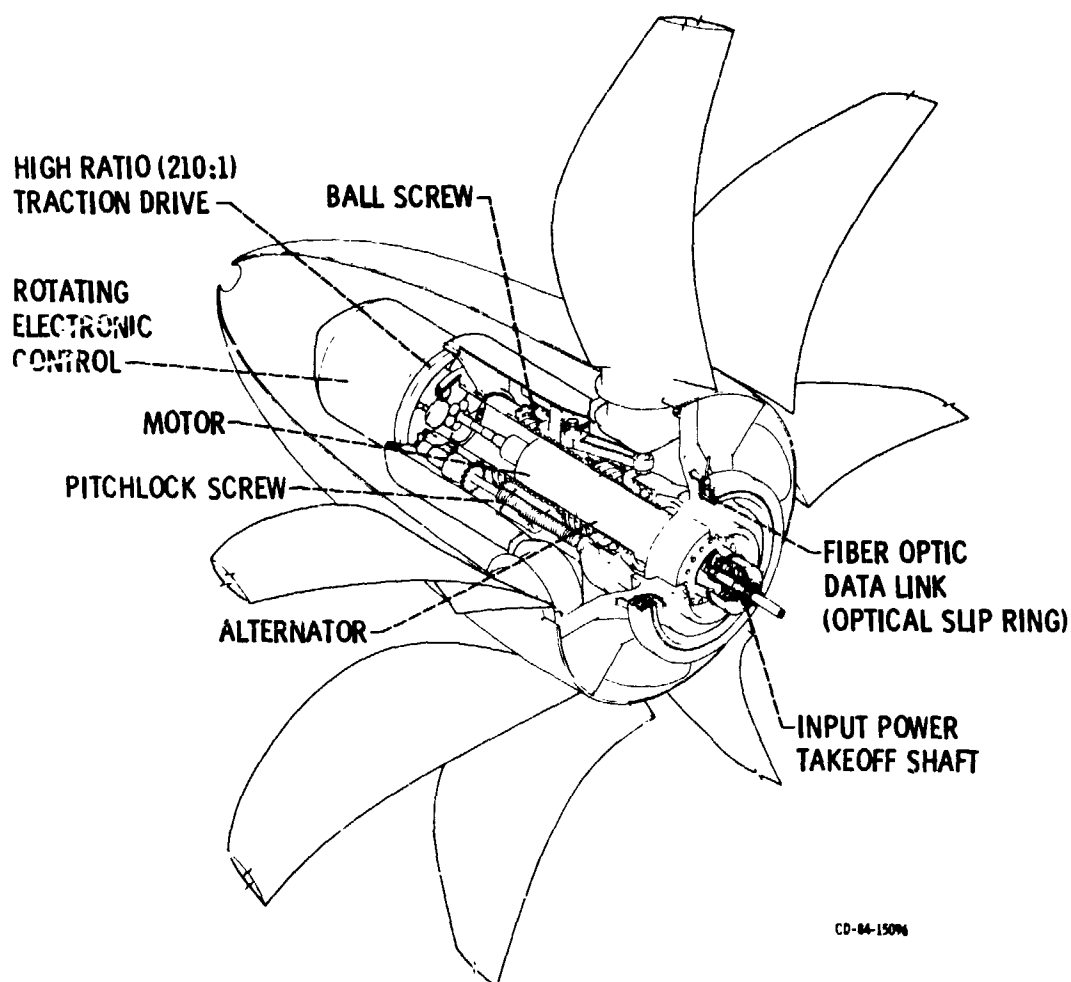


Figure 10. - Conceptual design of an advanced turboprop propeller pitch change mechanism which incorporates a hybrid traction drive.

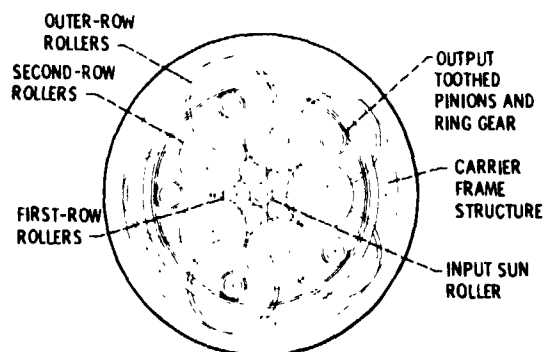


Figure 11. - 210 to 1 ratio, hybrid traction drive geometry.