

NASA  
Technical Memorandum 83397

AVRADCOM  
Technical Report 83-C-5

(NASA-TM-83397) ADVANCES IN TRACTION DRIVE  
TECHNOLOGY (NASA) 21 P HC A02/MF A01  
CSCL 151

N83-28455

Unclass  
G3/37 28069

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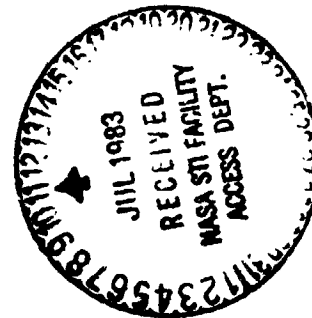
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Prepared for the  
International Off-Highway Meeting and Exposition  
sponsored by the Society of Automotive Engineers  
Milwaukee, Wisconsin, September 12-15, 1983

**NASA**



# Advances in Traction Drive Technology

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## ABSTRACT

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Traction drives are among the simplest of all speed changing mechanisms. Although they have been in industrial use for more than 50 years, their operating characteristics and performance capabilities are not widely known. This paper briefly traces their technical evolution from early uses as main transmissions in automobiles at the turn of the century to modern, high-powered traction drives capable of transmitting hundreds of horsepower. Recent advances in technology are described which enable today's traction drive to be a serious candidate for off-highway vehicles and helicopter applications. Improvements in materials, traction fluids, design techniques, power loss and life prediction methods will be highlighted. Performance characteristics of the Nasvytis fixed-ratio drive are given. Promising future drive applications, such as helicopter main transmissions and servo-control positioning mechanisms are also addressed.

FRICION WHEELS OF UNEQUAL DIAMETER were one of the earliest speed changing mechanisms. It is speculated that their use even predates that of gearing "toothed" wheels, whose beginnings date back to the time of Archimedes, circa 250 B.C. (1). Even today, friction drives may be found in equipment where a simple and economical solution to speed regulation is required. Phonograph drives, self-propelled lawnmowers, or even the amusement park ride driven by a rubber tire are a few of the more common examples. In these examples, simple dry con-

tact is involved and the transmitted power levels are low. However, this same principle can be harnessed in the construction of an oil-lubricated, all steel component transmission which can carry hundreds of horsepower using today's technology. In fact, oil-lubricated traction drives have been in industrial service as speed regulators for more than 50 years. Despite this, the concept of transmitting power via traction is not widely known or understood.

Although traction drives have been available for some time (2-6) it is perhaps since the mid 1960's or so that they have been considered serious competitors to conventional mechanical power transmissions. The earlier drives, particularly those targeted for automotive applications, had their share of durability problems above nominal power levels. As a consequence, relatively few succeeded in the market place. The underlying reason for this was that certain critical pieces of technology were generally lacking. Designs were based on mostly trial and error. No uniform failure theories were available to establish service life or reliability ratings. The drive materials of the day were crude by today's standards. In short, traction drives were in their technical infancy.

Prompted by the research for more efficient automotive transmissions and bolstered by advancements made in rolling-element bearing technology, interest in traction drives has been renewed. Today's analytical tools, materials, and traction fluids are far superior to those available prior to the 1970s. This has led to the re-emergence of traction drives and the technology related to their design.

It is the intent of this review to discuss the basic principles of these traction drives and to trace the evolution of their technology, in a limited sense, from their early development to the efforts underway today.

\*Numbers in parentheses designate references at end of paper.

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### EARLY APPLICATIONS

One of the earliest known examples of a friction\* drive is that patented by C. W. Hunt in 1877 appearing in Fig. 1 (6). The Hunt drive had a single, spoked transfer wheel that was probably covered with leather running against a pair of toroidally shaped metal disks. Judging by the pulley flanges attached to the toroidal disks, the drive was intended to regulate the speed of belt-driven machinery such as that commonly found in factories at the turn of the century.

A similar drive was devised by W. D. Hoffman as shown in an 1899 British patent application (7) (Fig. 2). The toroidal drive arrangement apparently found great favor with traction drive designers through the years. Work continues on this configuration even today, more than 80 years later.

Friction drives also found use on several types of wood-working machinery dating back before the 1870's. For example, Appleton's Cyclopedia of Applied Mechanics (8), published in 1880, reports of frictional gearing being used to regulate the feed rate of wood on machines in which one wheel was made of iron and the other, typically the driver, of wood or iron covered with wood. For driving light machinery, wooden wheels of basswood, cottonwood, or even white pine reportedly gave good results. For heavy work, where from 30 to 45 kW was transmitted by simple contact, soft maple was preferred.

**AUTOMOTIVE SERVICE** - It was not until the introduction of the horseless carriage at the end of the 19th century that the goal of developing a continuously variable transmission (CVT) for a car sparked considerable friction drive activity. Mechanical ratchet, hydraulic, and electro-mechanical drives were all tried, but drives relying on friction, because of their simplicity, were the first automobile transmissions to provide infinite ratio selection. The earliest of these was the rubber V-belt drives that appeared on the 1886 Benz and Daimler cars, the first mass-produced gasoline-engine-powered vehicles. Friction disk drives were used as regular equipment on a number of early motor cars, such as the Lambert as illustrated in a 1907 advertisement (9) (Fig. 3). Others included the 1906 Cartercar, 1909 Sears Motor Buggy, and 1914 Metz Speedster.

\*The term "friction drive" is normally used to refer to a drive that is nonlubricated while "traction drive" refers to one with oil-wetted components. The friction terminology may have evolved from the fact that these drives intentionally use at least one roller that was covered with a high friction material such as leather, rubber, fiber or even wood. This notwithstanding, the term "friction" is somewhat of a misnomer since it is still the traction force responsible for positive motion of the driven element.

The Cartercar had an extremely simple friction drive consisting of a metal disk, driven by the engine crankshaft, in friction contact with a large, fiber-covered spoked wheel mounted on a transverse countershaft. To vary speed ratio, a driver operated lever was used to radially position the output follower wheel across the face of the metal disk - turntable fashion. The smoothness and ease of operation of the Cartercar transmission made it quite popular. It is not well-known that Mr. W. C. Durant, founder and first president of General Motors Company, acquired the Cartercar Company in 1908 because of his expectation that friction drives would soon be universally used in automobiles (10,11). In 1910, the Cartercar Company even produced a Model "T" truck, equipped with their friction drive. Despite its catchy slogan, "No clutch to slip - no gears to strip ... a thousand silent speeds and only one control lever, that's a Cartercar", the Cartercar Company's commercial success was shortlived.

From 1909 until 1912, Sears marketed a two-cylinder, 14-horsepower "Motor Buggy", also equipped with a friction drive (12). "Absolute simplicity, its positiveness under the most severe conditions and its unequalled flexibility", boasted one of the Sear's ads. However, by about 1915, cars equipped with friction drives had virtually disappeared (12), presumably due, in part, to the need to frequently renew the friction material.

Despite the limited success of these earlier attempts, the goal of developing and automotive transmission that smoothly and automatically shifted was not lost. In the late 1920s the Buick Division of General Motors was given the task of developing a continuously variable, oil-lubricated, steel-on-steel traction drive. This transmission was similar in design to the Hayes double toroidal traction drive, patented in 1929. The Hayes Self-Selector Transmission (13), although originally developed in the United States, was later offered as an option on the 1935 British Austin Sixteen (14).

The General Motors toroidal drive, later called the toric transmission, is illustrated in Fig. 4. The geometry of the drive is remarkably similar to the 1877 Hunt drive, with the addition of a second toroidal cavity and a ball differential to balance loading between the two cavities. An extensive test program was conducted on this drive. Seventeen road-test vehicles equipped with the toric drive accumulated over 300,000 miles of road testing (11). A 20-percent improvement in highway fuel mileage was reported. In 1932, General Motors decided to produce this type of transmission (10). However, no cars equipped with the toric drive were ever sold to the public. The reasons for halting production were never really made clear. Some say that there were unresolved discrepancies in service life data obtained during road tests and that obtained from laboratory bench testers. Others believe

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that the availability of premium quality bearing steel, needed in large amounts to make each drive, was simply not great enough at the time to meet expected production requirements. What ever the reasons, Alfred P. Sloan, Jr., then president of General Motors, turned the transmission down for production in the belief that it would simply be too expensive to make (10).

General Motor's New Departure Bearing Division produced an industrial counterpart to the toric drive. By 1935 when production was halted, over 1600 units of the "Transitorque" traction CVT had been marketed (13). The drive's design is credited to Richard T. Erban, an early traction-drive pioneer, who briefly worked for General Motors during this period (2).

In England, after several years of analyzing the Hayes Self Selector drive, Perbury Engineering, Ltd., retrofitted a modified, scrapped Hayes transmission into Hillman Minx sedan in 1958 (14). Fuel savings were reported to be 20 to 25 percent but the concept really never caught on with any of the several dozen companies or so that had expressed initial interest in the drive (14).

In the United States in 1949, Charles Kraus installed a modified version of a toroidal CVT into an American Motors Nash Rambler (2). This unit had a semi-toroidal roller geometry similar to that patented in 1937 by Jacob Arter for industrial service. The Arter drive is still commercially produced in Switzerland. In 1973, Tractor, Inc. demonstrated a Ford Pinto equipped with an improved version of the Kraus drive lubricated with Monsanto's then new traction fluid. Although operational characteristics were established, expected fuel economy improvements were largely negated by the hydraulic losses in the thrust bearings used to clamp the toroids together (15). More recent toroidal drive designs partially overcame this problem by mounting two toroidal drive cavities back-to-back along a common shaft, thereby eliminating these troublesome thrust bearings.

**INDUSTRIAL SERVICE** - Starting with the 1877 Hunt drive, adjustable-speed traction drives have been in industrial service for more than 100 years. The bulk of these drives has been performing a speed matching function for light-duty equipment, such as drill presses.

A sample of representative traction drive configurations appears in Fig. 5. According to Carson's 1975 article (16), more than 100 United States patents on adjustable-speed traction drives are on file. Out of these, perhaps a dozen or so basic geometries are in production. Of those commercially available, few are rated at greater than 10 kW power capacity. An early review of the basic types of adjustable-speed traction drives can be found in (17). Reference 18 gives descriptive information on 24 types of variable speed traction drives that were commercially available in 1963.

Applications for modern traction drives are quite diverse ranging from machine tools, textile machinery to conveyor and pump drives. Specific applications can be found in (2-4, 16, 17).

### BASIC PRINCIPLES OF OPERATION

**FEATURES** - Traction drives can be constructed to give a single, fixed-speed ratio, like a gearbox or, unlike a gearbox, a speed ratio that can be continuously varied by using some means to shift or tilt rollers so they engage at different rolling radii. This latter arrangement is of extreme interest to drive train configurators since it provides them with an essentially "infinite" number of shift points to optimize the performance of their drive system.

Because power transfer occurs between smooth rolling-bodies, generally across a thin, tenacious lubricant film, traction drives possess certain performance characteristics not found in other power transmissions. Traction drives can be designed to smoothly and continuously vary the speed ratio with efficiencies approaching those of the best gear drives. Unlike transmissions with gear teeth, which, even when perfectly machined, generate torsional oscillations as the load transfers between teeth, power transfer through traction is inherently smooth and quiet without any "backlash". A lubricant film trapped between the rollers, tends to protect against wear and to dampen torsional vibrations. The operating speed of some traction drives is limited only by the burst strength of the roller material and the available traction in the contact. In many cases, traction drives can be designed to be as small as or smaller than their nontraction-drive counterparts. When manufactured in sufficient quantity, costs can also be quite competitive because of the similarities in manufacturing traction drive components and ordinary mass-produced ball and roller bearings.

**TRACTION POWER TRANSFER** - A basic understanding of how power is transferred between traction-drive rollers is helpful in reviewing the contributions made in this area. Figure 6 shows a simple, lubricated, roller pair in traction contact. A sufficiently large normal load  $N$  is imposed on the rollers to transmit the tangential traction force  $T$ . The amount of normal load required to transmit a given traction force without destructive gross slip is dictated by the available traction coefficient,  $\mu$ , which is the ratio of  $T$  to  $N$ . Since contact fatigue life is inversely related to the third power of normal load, it is extremely desirable to make use of lubricants that produce high values of  $\mu$ . The search for lubricants having high traction capabilities will be discussed later.

The rollers, as illustrated in the enlarged view of the contact appearing in Fig. 6, are not in direct contact but are, in fact, separated by a highly compressed, extremely thin lubricant film. Because of the presence of high pressures in the contact, the lubrication process is accompanied by some elastic deformation of the contact surface. Accord-

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ingly, this process is referred to as elastohydrodynamic (EHD) lubrication.

This phenomenon, also occurs for other oil-lubricated, rolling-element machine elements such as bearings and gears. The importance of the EHD film in traction contacts lies in its ability to reduce and/or eliminate wear while acting as the principal torque transferring medium.

**TRACTION CURVE** - The tribological properties of the lubricant in the contact, particularly its traction characteristics, are fundamental to the design of traction drives. Figure 7 shows a typical traction-versus-slip curve for a traction fluid. This type of curve is typically generated with a twin-disk traction tester. Imposing a traction force across a lubricated disk contact which is rotating at an average surface velocity  $U$  gives rise to a differential velocity  $\Delta U$ , sometimes referred to as "creep". Three distinct regions can generally be identified on a traction curve. In the linear region the traction coefficient increases linearly with slip. In the non-Newtonian regions it increases in a nonlinear fashion, reaches a maximum, and then begins to decrease. Finally, the curve shows a gradual decay with slip in the thermal region due to internal heating within the oil film. It is the linear region of the traction curve that is of the greatest interest to traction-drive designers. The design traction coefficient, which dictates how much normal load is needed to transmit a given traction force, is always chosen to be less than (by, generally, 20 to 30 percent) the peak available traction coefficient to provide a safety margin against slip. Traction drives are generally equipped with a torque-sensitive loading mechanism that adjusts the normal contact load in proportion to the transmitted torque. Such mechanisms insure that the contact will always have sufficient load to prevent slip without needlessly overloading the contact under light loads.

### ADVANCEMENTS IN TECHNOLOGY

Traction drive technology made relatively little progress for the first half of this century except for the occasional introduction of a new geometric variation. Designs were largely predicated on laboratory or field experience and very little of this information was reported in the open technical literature.

Because of the great similarity in the contact operating condition, traction-drive technology benefited greatly from the wave of technical advancements made for rolling-element bearings. Major advancements in bearing design occurred in the late 1940s with Grubin's work in elastohydrodynamic lubrication (19) and Lundberg-Palmgren's analysis of rolling-element fatigue life (20). In fact, the lubrication principles, operating conditions, and failure mechanisms of traction-drive contacts and bearing contacts are so similar that the design

fundamentals are virtually interchangeable. The same may be said for gear contact design criteria as well.

In view of the durability shortcomings of earlier traction drives, much of the recent research has centered on improving the power capacity and reliability of these devices without sacrificing their inherent simplicity or high mechanical efficiency. Although work has been performed on many fronts, research efforts to date can be loosely categorized under one of several areas: (1) modeling the tractive behavior of the lubricant within the contact and its attendant power losses; (2) predicting the useful torque that can be passed between rollers without surface distress or that amount corresponding to a given fatigue life; (3) determining and improving the durability characteristics of traction-drive materials, primarily bearing-grade steels; (4) developing lubricants that produce higher traction forces in the contact without sacrificing conventional lubricant qualities; and (5) developing drive arrangements that maximize durability, torque capacity, and ratio capability and minimize size, weight, power loss and complexity.

**CAPACITY** - The earliest traction drives generally used leather, wood, rubber, or fiber covered friction wheels running against metal disks. As these soft friction materials aged, they lost flexibility and wore rapidly. The driving surfaces generally had to be renewed or replaced at frequent intervals, depending on the rate of usage. Despite this, friction drives found use in early steam tractors, factory machinery, wood-working tools, and in several vintage cars. These simple, smooth, low-cost speed changers are still in use today for light-duty applications ranging from hand tools, washing machines, and record players to amusement park rides and cement mixers. In these modern drives more durable rubber and reinforced plastic materials have been substituted for the leather and fiber wheels of yesteryear. However, the thermal capacity and wear characteristics of these softer materials still basically set the useful power capacity of this class of transmission. For applications requiring a low-cost speed changer, adjustable-speed friction drives are good choices.

Oil-lubricated traction drives having hardened steel roller contacts appeared in the early 1920s. Carson (6) credits Erban in Austria with the development of the first metal-to-metal, oil-lubricated drive in 1922. The 1923 Arter drive is another example. The presence of hardened steel rollers in these drives significantly increased the allowable operating contact stresses. The purpose of the oil was to protect the contact surfaces from wear while providing cooling. However, the relatively low coefficient of friction of the oil meant that these drives had to carry unusually high-contact loads to inhibit slip. High loading generally leads to early pitting, unless the torque rating of the drive was

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appropriately restricted. Even though the early traction drives tended to be bulky, their relatively high efficiency and smoothness of operation still made them attractive for many applications.

**FATIGUE LIFE** - In the past traction drives were basically sized for some allowable Hertz (contact) stress in the contact zone based on experience. No concerted effort had been made to size traction drives for a certain fatigue life such as the way ball and roller bearings are sized. Hence, while most of the drives shown in Fig. 5 may function well under some conditions, reliability characteristics are generally not well defined. Their expected service lives have not been determined in a manner to allow for comparisons between different types of drives or to predict the effect that widely varying operating conditions might have on fatigue life.

The first modern approach to determining the durability characteristics of traction drives was that published by Coy, Loewenthal and Zaretsky in 1976 (21). Their analysis made use of the fact that the materials, operating stresses, lubrication conditions and failure mechanisms, namely rolling-element fatigue, of properly designed traction drive contacts are virtually identical to those of ball or roller bearings. Thus the methods used to determine service life ratings of rolling bearings, namely Lundberg-Palmgren Theory, should be applicable to sizing traction drives. Lundberg and Palmgren (20) theorized that the probability of encountering a subsurface defect in the material leading to fatigue pitting was statistically related to the contact stress, the depth of the critical stress below the surface and the volume of material being stressed.

In (22), a simplified version of this fatigue-life theory was developed for traction drive contacts and in (23) this method was used to show the effects of torque, size, speed, contact shape, traction coefficient and number of multiple, parallel contacts on predicted drive life. These investigations show that multiple, load-sharing contacts significantly benefit torque capacity and drive life. Also, torque capacity and drive life are proportioned to size to the 2.8 and 8.4 power, respectively as shown in Fig. 8. Figure 9 from (23) clearly illustrates the importance that the oil's coefficient of traction has on performance.

**TRACTION FLUIDS** - Because of the importance that the coefficient of traction has on the life, size, and performance of a traction drive, considerable attention has been given to identifying fluids with high traction properties, starting in the late 50s with Lane's experiments (24). Hewko (25) obtained traction performance data which indicated that the lubricant composition and surface topography had the greatest overall effects on traction and that naphthenic-based mineral oils gave better performance than paraffinic oils.

Some of these early investigations lead to the development of commercial traction fluids.

The research of (26) describes the development of a formulated traction fluid, designated as Sunoco Traction Drive Fluid-86. This fluid evolved into Sun Oil's TDF-88, a commercial traction fluid currently available on a limited basis.

Hamman, et al (27) in examining some 26 test fluids identified several synthetic fluids that had up to 50 percent higher coefficient of traction, depending on test conditions, than those reported for the best naphthenic base oils. This research laid the ground work for the development of Monsanto's family of commercial traction fluids, Santotrac 30, 40, 50 and 70. These fluids are the most widely used traction oils today. The results of accelerated rolling contact fatigue tests (28) indicated that these synthetic traction fluids have good fatigue life performance, statistically comparable with the automatic transmission fluid used in this experiment.

A recent addition to the commercial traction fluid market is that produced by the Mitsubishi Oil Co., Ltd. Their Diamond Traction Fluid is offered in three viscosity grades and is said to offer high traction coefficients, good wear and anti-oxidation properties.

It should be kept in mind that the use of traction fluids is not mandatory, although preferable. This is best illustrated by the experiments of Gaggermeier (29) in which traction coefficients for 17 different lubricants were measured on a twin-disk traction tester at both high and low contact pressures and surface speeds. The traction fluids in his tests showed substantially higher coefficients of traction than all of the commercial naphthenic mineral oils tested. The greatest differences occur at relatively low pressures and high surface speeds (Fig. 10). At relatively high pressures and low speeds the traction fluids show less of an advantage. Under such conditions a good quality naphthenic mineral oil would serve almost as well. However, for most traction drive applications there is considerable incentive to using a traction fluid, with expected traction improvements falling somewhere between the two examples of Fig. 10.

**MATERIALS** - Earlier traction drives were not exploited to their full potential because of uncertainties regarding their longterm reliability. The limited durability characteristics of the materials used in these drives was a major contributing factor. The substitution of oil-lubricated, hardened steel roller components in place of rubber or reinforced plastic running dry against cast-iron parts raises their load capacity by at least an order of magnitude.

Because of the similarity in operating conditions, hardened bearing steels are logical choices for traction-drive rollers. Today's bearing steels are of significantly higher quality than the traditional air-melted, AISI 52100 steels used in rolling-element bearings since the 1920s. The introduction of vacuum remelting processes in the late 1950s has re-

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sulted in more homogeneous steels with fewer impurities and have extended rolling-element bearing life several-fold. Life improvements of eight times or more are not uncommon according to (30). This reference recommends that a life-improvement factor of six be applied to Lundberg-Palmgren bearing life calculations when using modern vacuum-melted AISI 52100 steel. A similar life-improvement factor is applicable to traction-drive life calculations. This improvement in steel quality in combination with improvements in lubricant traction performance have increased the torque capacity of traction drives several-fold.

**PERFORMANCE PREDICTIONS** - The distribution of local traction forces in the contact of an actual traction drive can be rather complicated as illustrated in Fig. 11. This figure shows the distribution of local traction vectors in the contact when longitudinal traction, misalignment, and spin are present. These traction forces will align themselves with the local slip velocities. In traction-drive contacts some combination of tractions, misalignment, and spin are always present. To determine the performance of a traction-drive contact, the elemental traction forces must be integrated over the area of contact.

The power throughout the contact is determined from a summation of the traction force components aligned in the rolling direction times their respective rolling velocities. It is clear that in misalignment only a portion of the traction force is generating useful traction and that the remainder is generating useless side force. For pure spin no useful traction is developed, since the elemental traction forces cancel one another. Since the contact power loss is proportional to the product of the elemental traction forces and slip velocities, the presence of spin and misalignment can significantly decrease the efficiency of the contact. Furthermore, both conditions lower the available traction coefficient and reduce the amount of torque that can be transmitted safely. Designs that minimize spin and side slip can be quite efficient. Contact efficiency of 99 percent or higher are possible.

**THEORY** - In the 1960s and early 70s, numerous papers were presented on the prediction of traction in EHD contacts. Poon (31) and Lingard (32) developed methods that integrated the contact forces to predict the available traction forces of a contact experiencing spin. Poon's method utilized the basic traction data from a twin-disk machine together with contact kinematics to predict the available traction. Lingard used a theoretical approach in which the EHD film exhibited a Newtonian viscous behavior at low shear rates until a critical limiting shear stress was reached. At this point the film yielded plastically with increasing shear rate. This model showed good correlation with experimental traction data from a toroidal, variable-ratio drive of the Perbury type. This same model was also successfully used by Gaggermeier (29) in an

unusually comprehensive investigation of the losses and characteristics of traction drive contacts. In addition to copious amounts of twin-disk traction data for numerous lubricants under various combinations of slip, sideslip, and spin, Gaggermeier (29) also investigated the sources of power losses of an Arter type toroidal drive. His findings were that of the total power losses, the load-dependent bearing and drive idling (no-load) losses were always greater than the losses due to traction power transfer. This result underscores the need to pay close attention to these tare losses in order to end up with a highly efficient traction drive.

A recent and comprehensive traction contact model is that proposed by Johnson and Tevaarwerk (33). Their model covers the full range of viscous, elastic, and plastic behavior of the EHD film. At higher pressures and speeds, typical of traction-drive contacts, the response of the lubricant film is linear and elastic at low rates of strain. At sufficiently high strain rates, the shear stress reaches some limiting value and the film shears plastically as in the case of some of the earlier analytical traction models.

In (39) Tevaarwerk presents graphical solutions developed from the Johnson and Tevaarwerk elastic-plastic traction model. These solutions are of practical value in the design and optimization of traction-drive contacts. By knowing the initial slope (shear modulus) and the maximum traction coefficient (limiting shear stress) from a zero-spin/zero-sideslip traction curve, the traction, creep, spin torque, and contact power loss can be found over a wide range of spin values and contact geometries.

Figure 12 shows that this analysis compares favorably with test data, taken from experiments of Gaggermeier (29). The observed, pronounced reduction in the available traction coefficient with just a few degrees of misalignment, underscores the need to maintain accurate alignment of roller components in traction drives.

**FLUID TRACTION DATA** - To be able to apply the aforementioned traction drive, certain fundamental fluid properties, namely, the lubricant's shear modulus and limiting yield shear stress, must first be known under the required operating speeds, pressures, and temperatures. Because of the difficulty of simulating the highly transient nature of an actual traction contact, the most reliable basic fluid property data have been traditionally deduced from the initial slope and maximum traction coefficient of an experimental traction curve. To obtain experimental traction data for design purposes, a NASA program was conducted for both the Monsanto and Sun Oil traction fluids over a range of speeds, pressures, temperatures, spin, and sideslip values that might be encountered in traction drives (35). A regression analysis applied to the data resulted in a correlation equation that

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can be used to predict the initial slope and maximum traction coefficient at any intermediate operating condition (36).

### RECENT DEVELOPMENTS

During the past 5 years, several traction drives, which incorporate much of the latest technology, have reached the prototype stage. Laboratory tests and design analysis of these drives shows them to have relatively high-power densities and, in some cases, to be ready for commercialization.

**NASVYTRAC DRIVE** - Although light-duty variable-ratio traction drives have been reasonably successful from a commercial standpoint, very few, if any, fixed-ratio types have progressed past the prototype stage. This is somewhat surprising in view of the outstanding ability of traction drives to provide smooth, quiet power transfer at extremely high or low speeds with good efficiency. They seem particularly well suited for high-speed machine tools, pump drives, and other turbomachinery. In other industrial applications they offer potential cost advantages because traction rollers should not be much more expensive to manufacture in quantity than ordinary rollers in roller bearings.

In terms of earlier work on fixed-ratio traction drives, the developmental effort at General Motors Research Laboratories on their Planetary traction drive (as described by Hewko (37)) was perhaps the most complete. Several of these drives were built and tested, including a 6-to-1 ratio, 373-kW unit for a torpedo and a 3.5-to-1 ratio, 75-kW test drive. This last drive exhibited better efficiency and lower noise than a comparable planetary gear set (37).

Interest in fixed-ratio traction drives is also high outside of the United States. Tests were recently conducted in Japan on a planetary traction drive of a construction similar to the General Motors unit for use with a gas-turbine auxiliary propulsion unit (APII) system (38). Planetary traction drives have also been studied in Finland.

The traction drives described thus far have a simple, single-row planet-roller format. For drives like these the number of load sharing planets is inversely related to the speed ratio. For example, a four-planet drive would have a maximum speed ratio of 6.8 before the planets interfered. A five-planet drive would be limited to a ratio of 4.8 and so on.

A remedy to the speed ratio and planet number limitations of simple, single-row planetary systems was devised by Nasvytis (39). His drive system used the sun and ring-roller of the simple planetary traction drive, but replaced the single row of equal diameter planet-rollers with two or more rows of "stepped" or dual-diameter planets. With this new "multi-roller" arrangement, practical speed ratios of 250 to 1 could be obtained in a single stage with three planet rows. Furthermore, the number

of planets carrying the load in parallel could be greatly increased for a given ratio. This resulted in a significant reduction in individual roller contact loading with a corresponding improvement in torque capacity and fatigue life.

In (39) Nasvytis reports the test results for several versions of his multiroller drive. The first drive tested was a 373-kW (500-hp) torpedo drive of three-planet row construction with a reduction ratio of 48.2 and an input speed of 53 000 rpm. The outside diameter of the drive itself was 43 cm (17 in) and it weighed just 930 N (210 lb) including its lightweight magnesium housing. It demonstrated a mechanical efficiency above 95 percent. To investigate ultrahigh-speed operation, Nasvytis tested a 3.7-kW (5-hp), three-row, 120-to-1 ratio speed increaser. The drive was preloaded and operated without torque at 480 000 rpm for 15 min and ran for 43 consecutive hours at 360 000 rpm without lubrication but with air cooling. Two back-to-back drives were operated for 180 hr at speeds varying from 1000 to 120 000 rpm and back to 1000 rpm. They transmitted between 1.5 and 2.2 kW (2 and 3 hp). Another 3.7-kW (5-hp), three-row speed increaser, with a speed ratio of 50, was tested for more than 5 hr at the full rated speed of 150 000 rpm with oil-mist lubrication and air cooling. It successfully transmitted 3.7 kW (5 hp) at 86 percent efficiency (39).

The basic geometry of the Nasvytis traction (Nasyvtrac) drive is shown in Fig. 13. Two rows of stepped planet-rollers are contained between the concentric, high-speed sun roller and low-speed ring rollers. In the drive shown the planet rollers do not orbit but are grounded to the case through relatively low-speed and lightly loaded reaction bearings that are contained in the outer planet row only. The high-speed sun-roller and other planet bearings have been eliminated. The sun-roller and first row rollers float freely in three-point contact with adjacent rollers for location. Because of this floating roller construction, an excellent force balance situation exists even with thermal or mechanical housing distortion or with slight mismatches in roller dimensions.

Based on the inherent qualities of the Nasvytrac drive, a NASA program was initiated (40) to parametrically test two versions of the drive. These drives of nominally 14-to-1 ratio were tested at speeds to 73 000 rpm and power levels to 180 kW. Parametric tests were also conducted with the Nasvytis drive retrofitted to an automotive gas-turbine engine. The drives exhibited good performance, with a nominal peak efficiency of 94 to 96 percent and a maximum speed loss due to creep of approximately 3.5 percent. The drive package size of approximately 25 cm in diameter by 11 cm in width (excluding shafting) and total weight of about 26 kg (58 lb) makes the Nasvytrac drive, with a rated mean life of about 12 000 hr at 75 kW and 75 000 rpm, size competitive with the best commercial gear drive systems (23).



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**TURBOPUMP DRIVE** - A 70 000 rpm, 10.8-to-1-reduction-ratio Nasvytrac drive weighing just 4 kg was designed and built for a long-life, rocket-engine pump drive system to drive the low-speed liquid-oxygen and liquid-hydrogen boost pumps (41). Either an auxiliary turbine or a gear drive off the main pump can be used. Use of an auxiliary turbine complicates the design, while gear drives were not well suited for this application because they wore badly in a matter of 20 min or so in this hostile cryogenic environment. This fell far short of the 10-hr life requirement envisioned for future, reusable rocket engines. The relatively low sliding characteristics of the Nasvytrac drive, coupled with its demonstrated ability to run for long periods of time unlubricated, make it an excellent candidate for this application. Preliminary tests on this drive in liquid oxygen, including tests in which the drive was repeatedly accelerated under full power (15 kW) to 70 000 rpm in 5-sec intervals, showed it to perform satisfactorily (41). Cumulative operating times up to an hour have been recorded. Future work is needed to realize the 10-hr life goal, but the potential of this transmission for this application has been clearly demonstrated.

**HYBRID TRANSMISSION** - Current work with variations of the Nasvytis traction-drive concept are underway at NASA. A recently fabricated, 370-kW (500-hp) helicopter main rotor transmission combines the best features of gears with traction rollers (42). This experimental "hybrid" transmission, which offers potential cost, noise, and reliability benefits, is currently under test (Fig. 14). This transmission carries 58 percent more power in a test package that is only 22 percent heavier than the production OH-58 helicopter gear box it simulates. In a high ratio version, to be tested shortly, the hybrid transmission reduces the engine's power turbine shaft speed of 36 000 rpm down to the rotor speed of 347 rpm in a single hybrid stage plus a bevel gear mesh. This permits the elimination of the nose gearbox on the engine. The net result is a predicted 68 percent increase in system power-to-weight ratio. A 300 percent or greater increase in reliability is also expected based on traction roller fatigue life estimates.

The benefits of this transmission are attributed to the hybrid's unique geometrical configuration. Gear pinions are affixed to the ends of rollers in the outer row. The pinions, in turn, mesh with a collector ring gear (or bull gear) which is normally attached to the low speed output or rotor shaft in this case. The high torque capacity of the drive per unit weight is due, in part, to the high number parallel load paths in the final mesh. Another contributing factor is the large reduction ratio, on the order of 10-to-1, achieved across the final mesh which enables the high speed traction roller section to carry relatively small torque loads.

An additional benefit of integrating gears with traction rollers is the ability of the rollers to equalize the load sharing between the gear pinions through traction "creep". Creep is the small difference in velocity, generally less than 0.5 percent, between the surfaces of the driver and driven rollers due to torque transfer. In the hybrid drive, if one of the pinions is carrying more load than the others then the roller to which it is attached will experience a slightly higher creep rate, allowing the load to equalize. Thus, the traction rollers perform an important secondary function as a torque splitting mechanism. The simultaneous combination of high ratio, high number of redundant load paths and a high degree of load sharing in the final gear mesh is an important feature of the hybrid concept not shared by conventional epicyclic gearing.

**ADVANCED APPLICATIONS** - Work is now underway on hybrid and pure-traction Nasvytis drives for several advanced applications: one such application is for wind turbines where a low-cost but highly reliable speed increase is needed to drive the high-speed alternator. Also, tests are now being conducted on a 160 hp infinitely variable ratio Nasvytis drive, but the performance data has not yet been finalized. Another promising area of investigation for traction drives is as servo-control, positioning mechanisms, such as those used in robotics and in various manufacturing operations. Potential benefits include zero backlash, low torque ripple, high torsional stiffness, ability to run dry (no lubrication system), high reduction ratio in single stage and compact size. Space application of traction mechanism is also under consideration. These are a few of the areas where basic technology is being sought.

**PROMISING VARIABLE SPEED DRIVES** - Taking advantage of the latest technology, several designers have attempted to develop traction CVT's for automotive use. The potential of improving the city fuel mileage 20 to 25 percent, or more, of cars normally equipped with three or four-speed automatic transmissions (43) or of doubling the fuel mileage in the case of flywheel equipped cars (44) has been the major incentive for the resurgence in automotive CVT research and development. These applications represent a significant challenge, since compactness, efficiency, cost, and reliability are all at a premium.

**Perbury CVT** - One such automotive effort is that being conducted by BL Technology, Ltd., formerly British Leyland, on a Perbury, double-cavity toroidal drive. This concept is rather old and well-explored, as mentioned earlier, having been investigated by the General Motors Research Laboratory in the early 30s and late 50s, demonstrated in a 1934 Austin-Mayes, later in a 1957 Hillman-Minx, and also in a 1973 Ford Pinto but with offset rollers.

In 1977 Lucas Aerospace in England adapted the Perbury drive for maintaining constant frequency of the AC generators on the Sidley

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Hawker Harrier, a VSTOL jet fighter. This single cavity, toroidal drive is suitable for driving aircraft generators having output ratings up to 30 kVA. More than 20 years before this, Avco Lycoming, in the U. S., also offered a line of mechanical constant-speed drives based on the toroidal traction-drive principle. They were used on several aircraft in the 50s, including the Douglas A-4E fighter, but have long since been discontinued.

In the case of the Leyland-Perbury automotive CVT (Fig. 15), the double-toroidal drive cavities have six tiltable transfer rollers between input and output toroids. A hydraulically controlled linkage system can tilt these rollers from one extreme position to another. By combining this toroidal drive with a two-range, output planetary gear system, the overall transmission ratio range is greatly expanded.

The BL/Perbury transmission was installed in a medium size test car having a four-cylinder, 60-kW engine. The test car showed fuel mileage improvement of 15 to 20 percent for an average mix of European driving (45). Also, acceleration times were comparable with a manual transmission car driven by a skilled driver having a 10 percent higher power to weight ratio. However, the future production picture for this transmission is not clear.

Vadatec CVT - A promising traction CVT that is of a rather new vintage is the nutating drive being developed by Vadatec Corp. (46). As shown in Fig. 16, a double-conical-roller assembly, complete with an automatic loading mechanism, is mounted at an angle in a drive cylinder that is driven by the input shaft. As the input shaft rotates, the double-cones perform a nutating motion and at the same time are forced to rotate about their own axis as they make drive contact with a pair of moveable control rings. These rings are grounded to the housing but can be axially moved together or apart. A gear pinion attached to the end of the cone shaft meshes with the output ring gear. By varying the axial position of the control rings, the rolling radius of the cones can be synchronously changed. This, in turn, causes a change in rotational speed of the cone shaft pinion and hence varies the transmissions output speed. A couple of Vadatec CVT's have already been built and tested. One of these prototypes has shown successful operation as part of a tractor drive train. Although this transmission is still in the development and acceptance stage, it is a good example of the new breed of traction drive.

Other Traction CVT's - In addition to the aforementioned activities, there are a number of additional traction CVT developments that have been reported on recently. These include the Fafnir planetary, cone roller type CVT, directed toward the mobile equipment market up to about 37-kW (50-hp) (47); the A<sub>1</sub> Research dual cavity toroidal drive (48); and the Bales McCain cone-roller CVT (49). These last two transmissions were carried to a preliminary design stage under a NASA contract for DOE to

develop CVT's for use in advanced electric vehicles (48).

### CONCLUDING REMARKS

The evolution of traction-drive technology has been traced over the past 100 years. Some of the more prominent events in the development of traction drives appears in Table I. This list is by no means intended to be comprehensive but, rather, to give the reader some appreciation of the scope of activities leading to traction drives of today.

The earliest of traction drives, constructed of wood, leather, or fiber-covered disks running in dry contact, found first use in factory equipment before the turn of the century. The ability of traction drives to smoothly and efficiently vary speed made them natural choices as main transmissions for several early vintage cars, such as those produced by Cartecar, Sears, Lambert, and Metz. Apparently, durability problems with the soft-material-covered disks used in these drives foreshortened their commercial success. In the 1920s traction drives equipped with oil-lubricated, hardened, steel rollers started to appear. These drives had much greater power capacity, and by the 1930s several industrial adjustable speed traction drives were being marketed both here and in Europe. About this time, there were several projects to develop toroidal traction car transmissions, notably the Hayes and later, the Perbury efforts in England and the General Motors' work in the United States. In the 1940s modern lubrication and fatigue theories for rolling-contact elements were developed, and these were later adapted to the design of traction drives. In the 1950s work began on identifying fluids with high traction properties and experiments on how these fluids actually behaved within the traction contact. A basic understanding was also obtained on how the fluid film within the contact was compressed into a thin, stiff, tenacious solid-like film across which considerable torque transfer could safely occur. By the end of the 1960s, high-quality bearing steels and traction fluids were commercially available. The power capacity of traction drives using the new steels and fluids virtually tripled. In the 1970s improved traction models and fatigue-life prediction methods were developed. This all led to the development of a new generation of traction drives -- drives with a bright potential role to play in the power transmission industry.

In 1980 the Power Transmission and Gearing Committee of ASME took a major step in recognizing the potential viability of traction drives by establishing a subcommittee to follow the developments in the technology for these transmissions. A primary function of this subcommittee is to assist in the dissemination of technology related to traction drives and to foster their potential use in industry.

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In conclusion, field experience has been gathered by industrial traction-drive manufacturers, some of whom have been making traction drives for more than 40 years. However, traction-drive technology is relatively young. The latest generation of traction drives has reached a high level of technical readiness. As these drives find their way into industrial service and as work continues in the laboratories, further improvements and increased usage of these drives can be expected.

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Table I. - Limited Chronology of Traction Drive Developments

Date	
1870	Wooden friction drives used on wood-working machinery
1877	Hunt toroidal friction drive patented
1906	Friction drive equipped Cartercar introduced
1921	Automatic contact loading mechanism patented by Erban in Germany
1923	Oil lubricated, Arter industrial drive marketed in Switzerland
1926	Carter analyses creep between locomotive wheel and rail
1928 - 1933	General Motors Research Labs. road tests toric traction transmission
1935	Austin Motor Company offers Hayes Self-Selector toroidal transmission on Austin Sixteen
1939	EHD lubrication theory advanced by Ertel and, later, Grubin
1947	Lundberg and Palmgren develop fatigue theory for rolling elements
1949	Kopp Ball Variator commercially introduced in Switzerland
1951	Clark, et al., theorize oil solidifies in EHD contact
1955 - 1962	Braunschweig University experiments on traction contact phenomena in Germany
1955	Vacuum-melted bearing steels introduced
1957	Lane's experiments to identify high traction oils in England
1962 - 1968	Hewko's investigations into traction contact performance
1965	Nasvytis devises fixed-ratio, multiroller planetary drive
1968	Monsanto and Sun Oil introduce commercial traction fluids
1976	Coy, et al., apply Lundberg-Palmgren fatigue life theory to traction drive contacts
1978	Johnson and Tevaarwerk traction model applied to the design of traction drives
1980	ASME establishes traction drive committee

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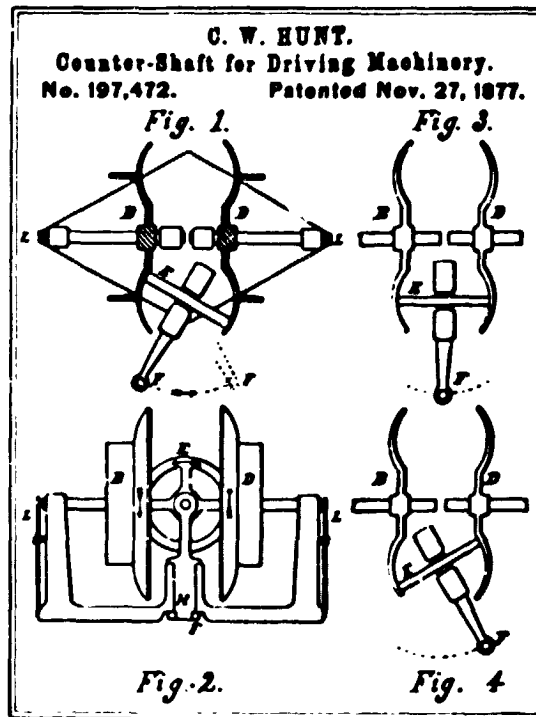


Figure 1. - Hunt's 1877 toroidal friction drive (ref. 6).

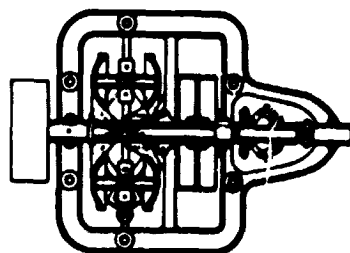


Figure 2. - 1899 Hoffman toroidal friction drive for belt-driven machinery (ref. 7).

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MOTOR AGE      MARCH 7, 1907

"IT'S A STRONGER PROPOSITION THAN YOU HAVE IMAGINED"

**DEALERS**

EXAMINE

**THE LAMBERT**

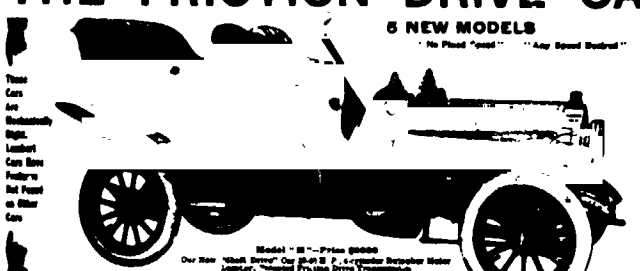
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Model "B" - Price \$2000  
Our New "Friction Drive" Car Built by E. J. Crayton, Buckeye Motor  
Company, "Friction Drive" Transmission

\*The big and strong motor makes for the price - Write for detailed description THE  
L.A. BUCKEYE - 2001 YANKEE AVE. N.Y. - See picture in these columns and you will know the engine of  
this motor is not only powerful, but a great power source. Write, Yonkers or New York

**THE BUCKEYE MANUFACTURING CO., - ANDERSON, IND., U. S. A.**

Good Dealer Agents Wanted in a Few Locations    Members American Motor Car Manufacturers' Association, New York.    Write for Special Art Catalogue

Figure 3. - An early advertisement for a friction-drive car (ref. 9).

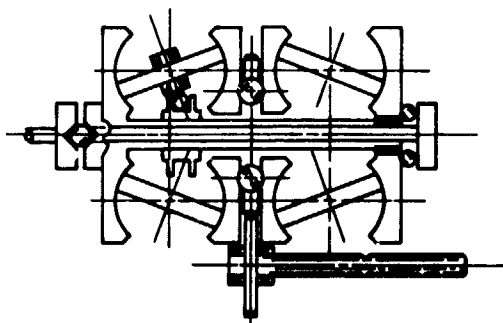


Figure 4. - General Motor's Toric transmission (ca. 1928).

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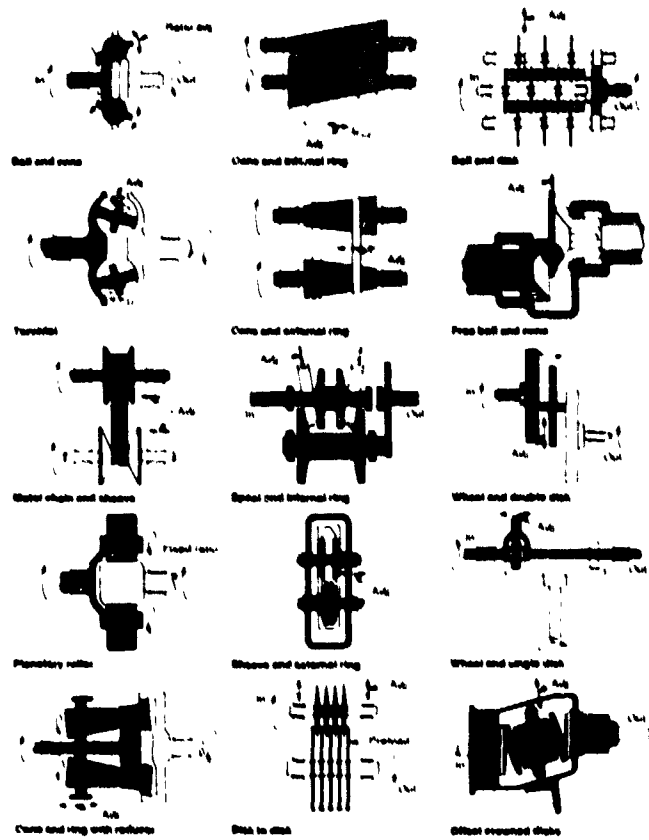


Figure 5. Typical industrial traction drive geometries. (Courtesy of Design Engineering (ref. 2).

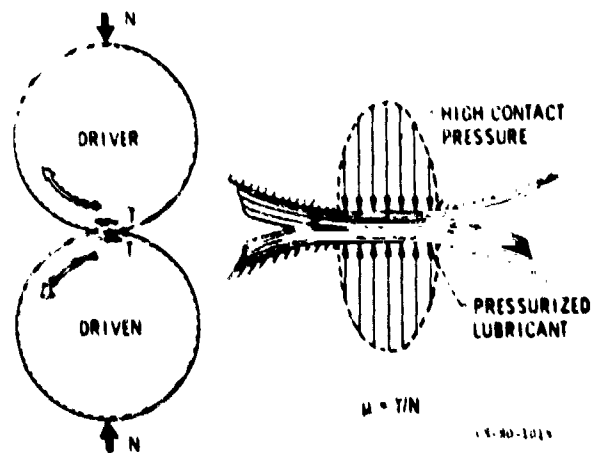


Figure 6. - Power transfer through traction.

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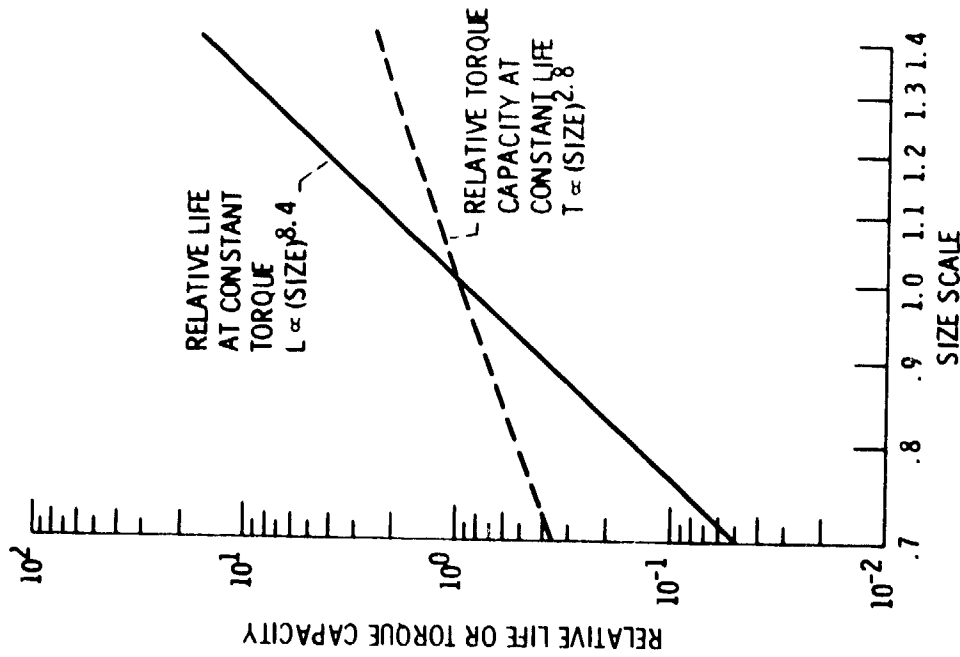


Figure 8. - Relative life at constant torque and relative torque capacity at constant life versus relative size (ref. 23).

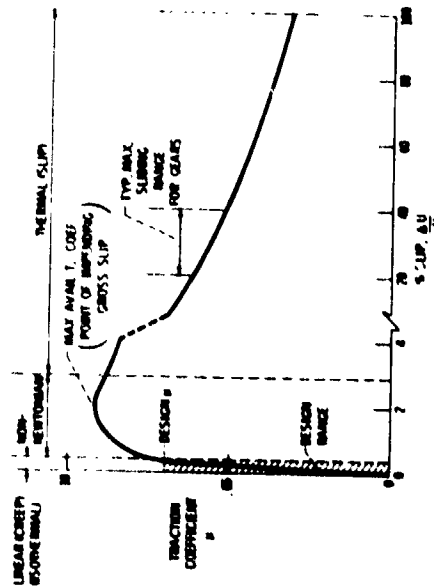


Figure 7. - Typical traction curve showing design range for traction drives.



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# EFFECT OF TRACTION COEFFICIENT ON TRACTION DRIVE LIFE, DIAMETER AND TORQUE CAPACITY

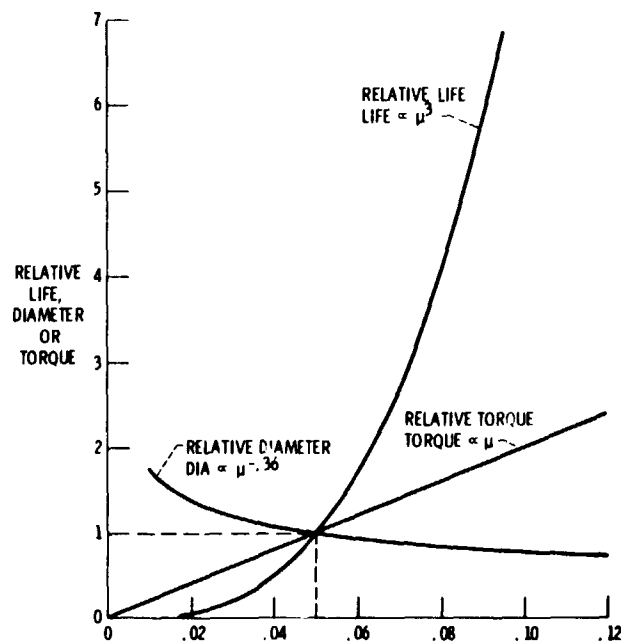


Figure 9. - Relative life, diameter, and torque capacity versus applied traction coefficient (ref. 23).

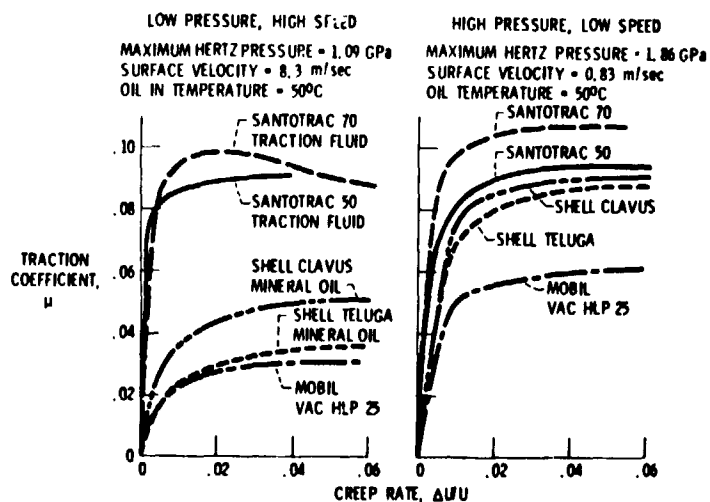


Figure 10. - Traction characteristics of a traction fluid compared with mineral oils (ref. 29).

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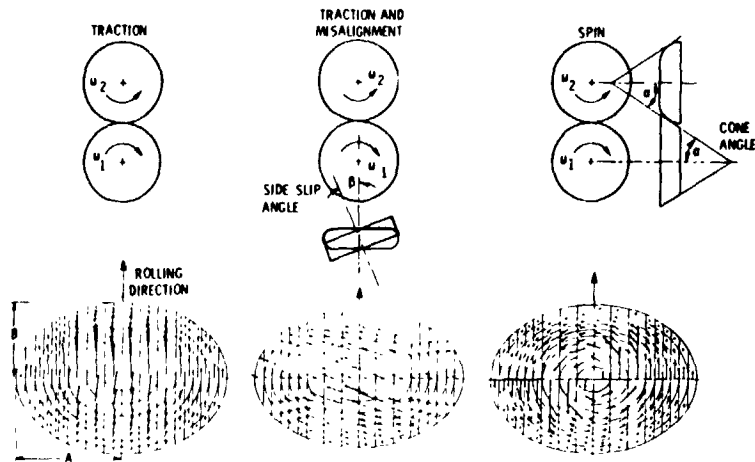


Figure 11. - Effect of misalignment and spin on contact traction force vectors.

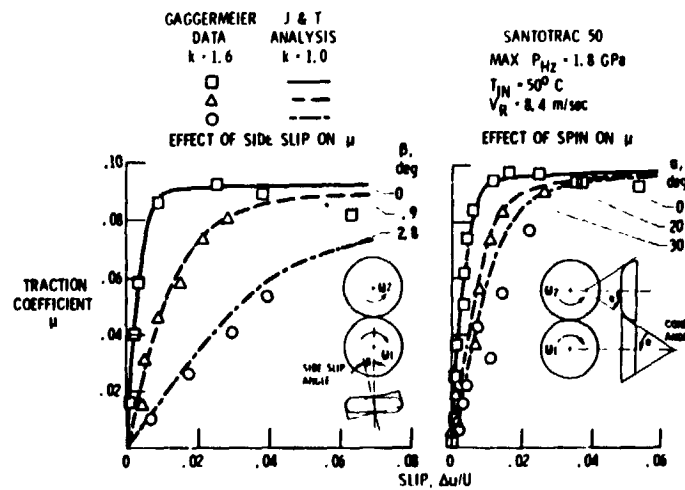


Figure 12. - Comparison of Johnson and Tevaarwerk analysis (ref. 48) with Gaggermeir test data (ref. 29).

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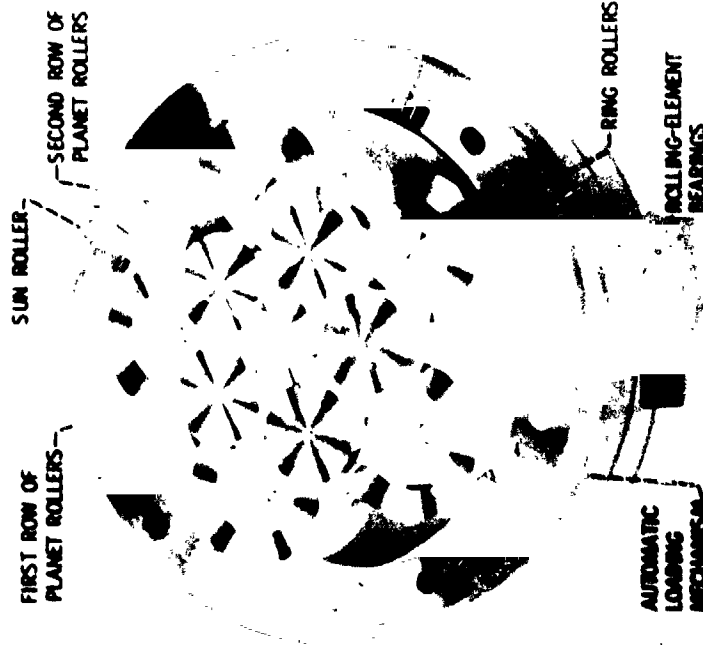


Figure 13. - Geometry of the Nasvytis traction (Nasytrac) test drive (ref. 40).



Figure 14. - 500 HP hybrid helicopter transmission.

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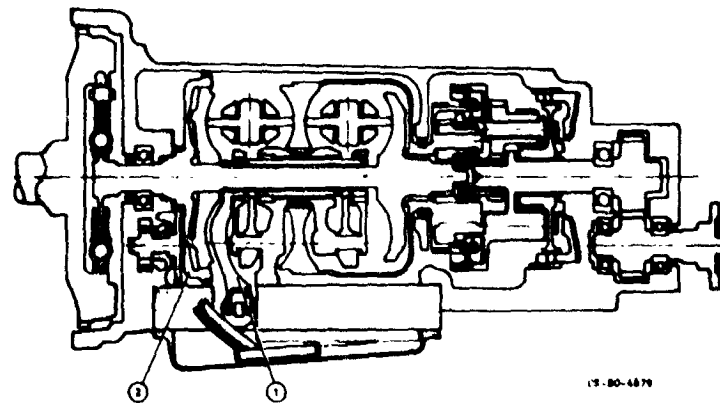


Figure 15. BI Technology/Perbury traction CVT, 60 kW passenger car test installation (ref. 15).

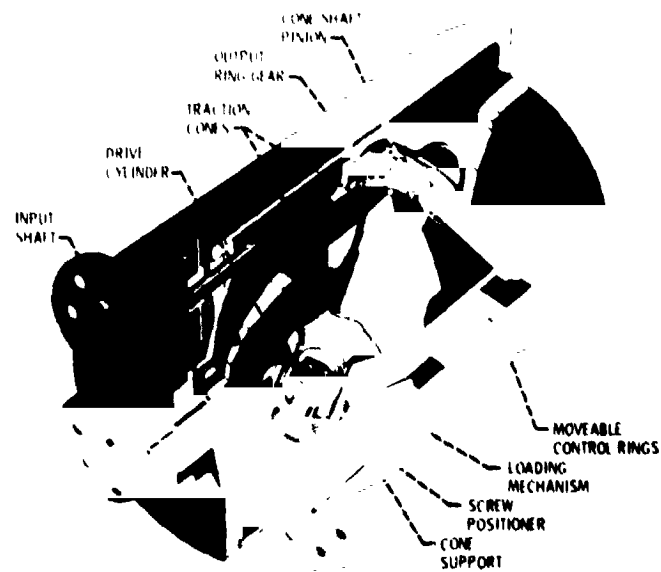


Figure 16. - Vadetec nutating traction CVT. (Courtesy of Vadetec Corp., Troy, Michigan.)