Roller Traction Drive Unit for Extremely Quiet Power Transmission

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In driveline applications, where the noise and vibrations are not tolerable, roller traction drives can provide speed reduction with high efficiency and virtually no noise. This paper discusses the principle of operation of the General Motors (GM) roller traction drive and presents dynamometer test results illustrating the effect of speed and torque on efficiency, reduction ratio, traction coefficient, and noise pattern. The unit is a planetary drive with preloaded, barrel roller planets for transmittal of power. A torque-actuated preloading mechanism prevents slippage, even under severe overload and greatly improves life and efficiency of the drive. Examples of roller drives ranging from 500 to 3 hp, actually built and tested for underwater application, are also briefly discussed.

Introduction

Friction drives, more correctly traction drives, are outstanding in their capacity for quiet and vibration-free operation. Other important features included are the absence of backlash, inherent damping, excellent efficiency, capability to run at extremely high speeds, simplicity, and nearly perfect internal-force balance. However, limited torque capacity, excessive weight, and inadequate rolling surface durability have prevented traction drives from being utilized widely in the past.

Recent advances in metallurgy, lubrication, analytical methods, high-speed computation, and better understanding of the mechanics of metal fatigue and elasto-hydrodynamic phenomena within the rolling contacts, warranted a new look at the traction drive principle. As a result of several years of extensive work at GM, it has been possible to successfully couple the inherent advantages of traction drives with good durability, smaller weight and size to provide an efficient and practical drive system.

Generally, traction drives can be divided into those capable of varying drive ratio, and those having essentially fixed ratio. Variable ratio drives are generally somewhat less efficient, bulkier, and considerably more complicated. The GM roller traction drive, which is the main topic of this discussion, is a fixed ratio type.

GM Roller Traction Drive Principle

Friction between the contacts of ball and roller bearings can be used to transmit substantial amounts of power. If the coefficient of friction, or more correctly the coefficient of traction, and the normal load on the rolling bodies are sufficient to prevent slipping, any preloaded ball or roller bearing can become a traction drive. The coefficient of traction is not the same as the coefficient of rolling resistance of rolling contact bearings, and is usually 20 to 30 times higher. Drives with fixed preload suffer from at least two severe limitations. First, such a drive slips under torque overload conditions. Second, at part load, contacts are (in effect) overloaded in the normal plane. This considerably lowers over-all efficiency and useful life. Ideally, the preload in the direction normal to the contact plane should be proportional and slightly higher than the tangential torque forces transmitted through the contact at any given time. Then, the torque demand automatically generates sufficient preload for efficient driving and makes slipping impossible. The GM roller traction drive has such nonslip characteristics.

Figure 1 shows schematically the basic geometry of the drive. Essentially, the unit is a large barrel roller bearing with split inner race, in which the cage and the races are designed to be capable of transmitting power. The special split inner race also incorporates a mechanical, torque actuated, axial loading feature for controlling the preload of the unit in porportion to torque transmitted. Identical planets, six in this case, are equally spaced within a rigid outer ring. The planet carrier incorporates a set of shafts that serve to transmit the torque from the planets, through suitable planet journals, to the planetary carrier. The shafts are free to move radially with respect to the carrier.

The torque loading device consists of two sets of three balls confined between the back face of each of the two inner races and the two bidirectional ramps, shown in the central portion of Fig. 1. Slight initial preload is built into the unit to assure proper operation of the torque loading mechanism at very low torques.

In the following discussion, the input to the unit is at the sun, the output at the carrier, and the ring is the stationary reaction member. (Other combinations of input-output arrangements are also possible and will be discussed later.) The operation of the drive is as follows: the input torque is equally divided between the two suns and causes the balls of the torque loading device to climb up the ramps on the back face of each sun. This in turn causes the suns to move axially toward each other. Since the outside diameters of the suns have a slight cone angle (identified as α in Fig. 1), the axial motion causes the planets to move radially outward and to press against the inside diameter of the reaction ring. The resulting radial force between the sun, planet, and ring, when multiplied by the coefficient of traction, becomes the usable, tangential torque force. By properly selecting the ball ramp and sun cone angles and by taking into consideration the contact traction coefficient, it is possible to generate a slight excess of normal load at the contacts. This insures that the drive will never slip at the contacts, regardless of how high the torque may become. In fact, should the torque become

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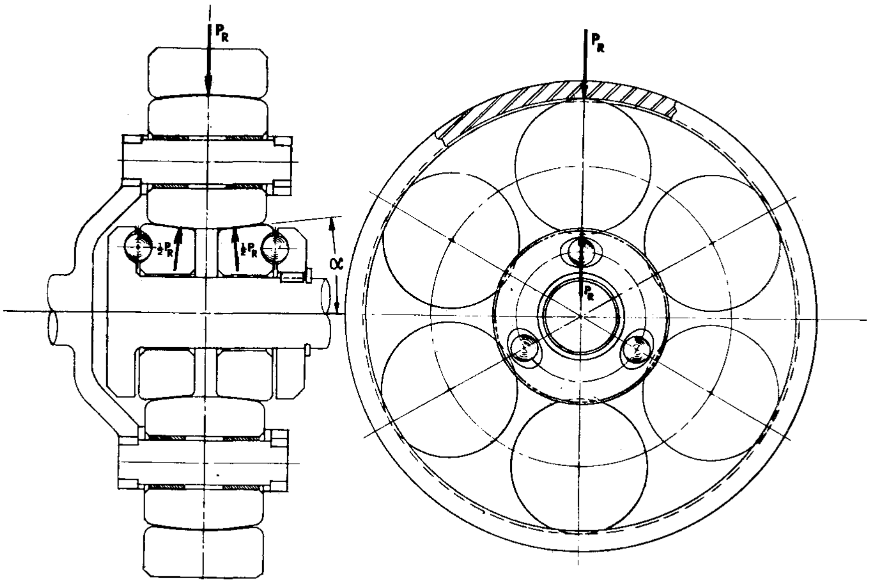


Fig. 1 Basic configuration of the GM roller traction drive. Ball-ramp type mechanism preloads the split sun roller in proportion to torque transmitted making the unit slip-proof, more efficient, and more durable.

excessive, the input shaft would be twisted off or the contacts become plastically deformed, but slippage would be impossible. This simple torque loading mechanism is a very rigid mechanical system and is capable of extremely fast response to torque fluctuations, torque reversals, or shock loading.

Unique Characteristics

In such a geometric configuration, each planet is in a nearly perfect force equilibrium, and the whole unit has all of its radial and axial forces internally balanced, as indicated by the opposing force vectors in Fig. 1. As a result, the drive can be readily isolated from the supporting housings to provide an additional means for noise reduction. However, the major reason for the quiet operation of a roller drive unit is related to the kinematics of power transfer through the smooth and continuous action of the tractive forces between round, lubricated rolling components, machined to high precision. The steady-state power flow through elasto-hydrodynamic contact interface results in the absence of periodic, vibration disturbances and also adds an effective damping component into the driveline dynamics. This is unlike gears where the tooth mesh frequency excitations may result in severe torsional and axial vibrations of the driveline. Even a theoretically perfect gear set, with several teeth in simultaneous contact, will generate torsional oscillations as part of the torque is transferred from one tooth to the next.

Roller traction drives are essentially positive drive devices. Even though the tractive forces are transmitted through shear stresses in the lubricant film and not by positive engagement between rollers, the fluid film is only a few hundred molecules thick and is under such extremely high local pressures that it acts essentially as a solid. Consequently, under normal operating conditions, as will be shown later by actual test data, the drive ratio of the unit remains virtually constant from no load to full load. The lubricant film also eliminates wear of the rolling elements, carries the heat away from the contact zone, and adds significantly to the torsional damping of an essentially rigid power train. The fact that a plurality of planets press with high force against the ring and sun simultaneously further increases damping and improves quiet performance.

Because the rolling elements are under constant preload, the roller drive is not subject to backlash when torque is suddenly removed or reversed. This results in quieter operation of the roller drive and permits its use in precise power control and position servocontrol systems. The transfer function for the roller drive can be accurately defined mathematically which makes the over-all servoloop analysis and performance more accurate.

In addition to transmitting torque quietly, the roller drive, being essentially a large rolling contact bearing, can also support substantial radial shaft forces. When suitably designed, the drive can also support limited thrust forces equally well.

Because power is transferred through contacts that are essentially in pure rolling motion, the power losses in the traction drive are very low. This results in high mechanical efficiency and enables roller drives to be operated at very high speeds. Being essentially torque sensitive devices, the power rating of a roller drive can be increased simply by increasing its speed. This is of special importance in gas-turbine applications, where speeds are high but torques are low.

Ratio

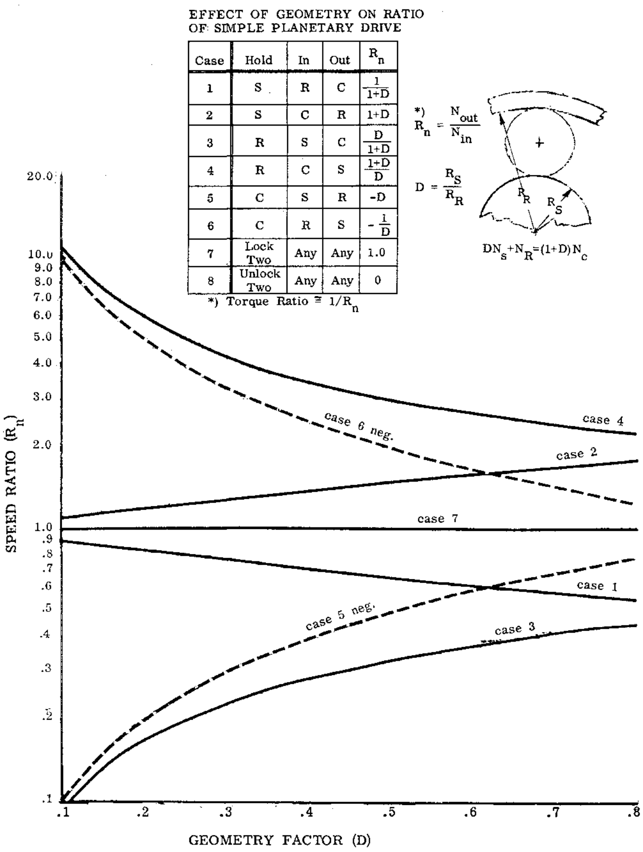
The ratio pattern of a simple planetary drive can be studied by plotting ratio vs geometry for combinations of input and output. The relationship between the speed of each of the three planetary members and the geometry is

$$DN_S + N_R = (1 + D)\, N_C \qquad (1)$$

where D = ratio of sun radius to ring radius; N_S = sun speed; N_C = carrier speed; and N_R = ring speed. By solving for the ratio of any of the two speeds, the reduction (or speedup) ratio of the planetary can be determined. The practical range for D is usually $0.10 < D < 0.80$, with corresponding ratio limits of about 1.1 to 11 reduction (or speedup). Higher reduction ratios require interconnecting two or more planetary units in series. The eight possible ways in which a given planetary drive can produce different ratios are plotted in Fig. 2 in terms of reduction ratio (R_n) vs geometry factor (D). With roller drives, an infinite number of planetary ratios are possible, because no consideration needs to be given to the integral number of teeth in relation to pitch. This also broadens the extreme ratio limits for roller drives by eliminating the necessity of allowing for the height of the gear teeth. For the same reason, a greater number of planets can be inserted into the space between the sun and the ring, thus increasing the life and power capacity of a given size unit.

Traction and Durability

The torque that can be transmitted through a given rolling contact is a function of normal force acting on the contact and the coefficient of traction between the two rolling bodies.



Case	Hold	In	Out	R_n
1	S	R	C	$\frac{1}{1+D}$
2	S	C	R	1+D
3	R	S	C	$\frac{D}{1+D}$
4	R	C	S	$\frac{1+D}{D}$
5	C	S	R	-D
6	C	R	S	$-\frac{1}{D}$
7	Lock Two	Any	Any	1.0
8	Unlock Two	Any	Any	0

*) Torque Ratio = $1/R_n$

Fig. 2 Effect of geometry on drive ratios for all possible input-output combinations that can be obtained in a simple, planetary drive.

The latter is affected by the type of lubricant used, as illustrated in Fig. 3 from Ref. 1. Indirectly, the coefficient of traction also affects the fatigue life of a traction drive because the life of a rolling contact element is inversely proportional to a power of the load, whereas the torque capacity is only directly proportional to the load. As a result, increasing the torque capacity by increasing the coefficient of traction is far more desirable than by increasing the normal load. From a rather thorough study, we have found that naphthenic base oils and certain synthetic fluids have highest tractive properties.

Other variables affecting coefficient of traction are the rolling speed, the lubricant temperature, the surface finish quality, the Hertzian contact pressure, the degree of contact spinning, and the geometry of the rolling bodies. All these variables, together with the properties of the material used in manufacturing of the rolling components, interact to produce a drive with predictable power capacity, size, fatigue durability, and reliability.

The relationships governing the rolling contact durability in a traction drive closely resemble those used for rolling contact bearings. Relationship between the several parameters mentioned previously and their model effect on life prediction are

$$L_1/L_2 = (T_2/T_1)^c \times (R_1/R_2)^d \times (N_{p1}/N_{p2})^e \times (S_2/S_1)^f \quad (2)$$

where L = life; T = torque; R = characteristic dimension; N = speed; N_p = number of planets; S = Hertz stress; c, d, e, f = experimentally or analytically determined exponents; and 1, 2 = subscripts two different sets of conditions.

The durability of the traction drive can be brought to any level by properly adjusting the previously listed parameters. Reliability of an assembly of many rolling elements can be computed by the use of Miner's cumulative damage theory and from known Weibull distribution for the population. We have had considerable success in obtaining experimental data and then applying analytical techniques that provide rather accurate life predictions. A more detailed description of the statistical and mathematical approaches used is given in Refs. 2 and 3.

A typical example of size effect on the B_{10}† life of the unit at different power levels is shown in Fig. 4. The 100-hp drive with B_{10} life of 1500 hr was arbitrarily given a size factor of 1. The dashed line is the expected brinelling limit and defines the smallest permissible unit for given power levels. A relationship between life in hours and reliability expressed as percent survival of the units, for any level of transmitted power, is shown in Fig. 5. At the design point indicated, 90% of the units survived at least 1500 hr while transmitting 100 hp at 3600-rpm input speed. Even at the brinelling limit of 330 hp, the B_{10} life of the unit is a statistically predictable 42 hr. For higher reliability‡ B_1 life may be considered. For example, drive has B_1 life of 275 hr while

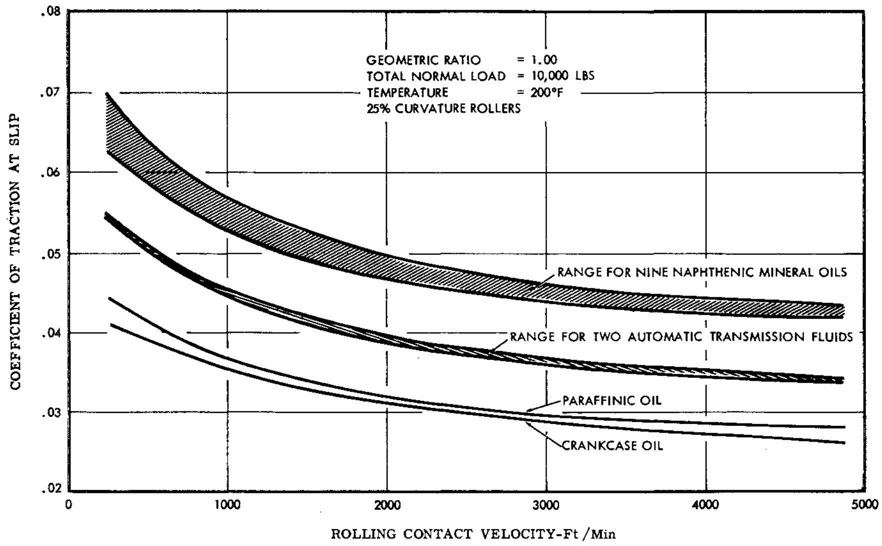


Fig. 3 Traction characteristics of representative lubricants as measured in traction unit.

† B_{10} life is defined as hours of operation which will be exceeded without failure by at least 90% of units transmitting given power.
‡ B_1 life is defined as hours of operation which will be exceeded without failure by at least 99% of units transmitting given power.

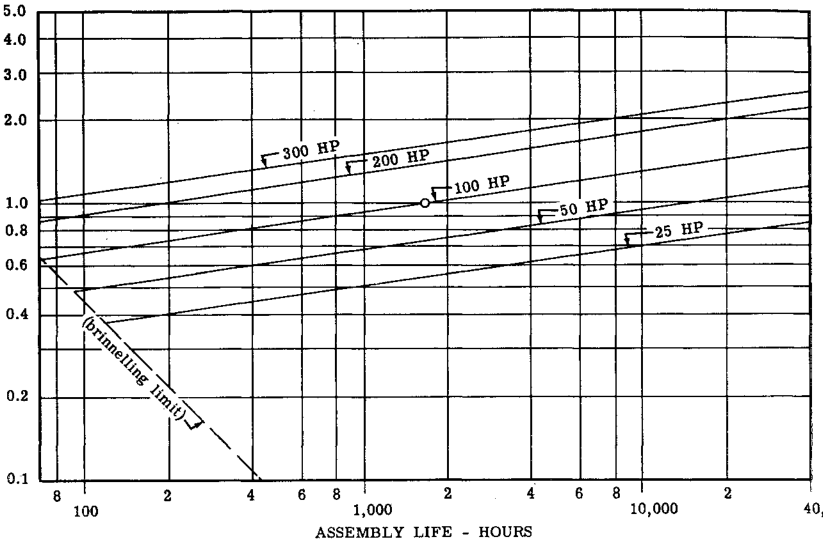


Fig. 4 B_{10} life of a typical traction drive operating at several different power levels plotted as a function of size. Reduction ratio = 3.5:1; input speed = 3600 rpm.

transmitting 100 hp. These results apply only to the life of the power transmitting rolling contacts, and other components, such as bearings, shafts, or springs, may have shorter lives.

To determine an optimum traction drive configuration, it is necessary to vary each parameter individually between permissible physical limits, to observe effects of the change on life and size, and then to repeat the calculations for all other parameters until the optimum combination is found. Such procedures have been programmed for a digital computer.

Performance Testing of the Drive

A roller drive of 100-hp capacity and 3.5:1 reduction and approximately 7.0-in. o.d. has been designed and tested at the GM Research Laboratories in Warren, Mich. A planetary gear set was also run in the same housing. The latter generally conformed to American Gear Manufacturers Association (AGMA) class 8 specifications and was randomly selected from an existing production line. Table 1 lists some of the specifications for the gear set.

The gear set was judged by a jury of gear noise experts as being "average-to-quiet" in its performance. Both the roller drive and the gear drive were of approximately the same size, over-all reduction ratio, life, and power capacity. After performance comparison tests, both drives were subjected to an extensive noise and vibration testing conducted at the GM Proving Ground Noise and Vibration Laboratory in Milford, Mich.

The longitudinal cross section of the traction drive used is shown in Fig. 6. Basic planetary geometry is as shown in Fig. 1. The torque loading assembly is above the centerline in Fig. 6. For experimental testing of ultimate tractive capacity, an adjustable calibrated spring load was used. This assembly is shown below the centerline in Fig. 6. Special ball spline assures that the axial spring forces are transmitted to the sun race with a minmum friction hysteresis. A photograph of the unit with front cover removed is shown in Fig. 7.

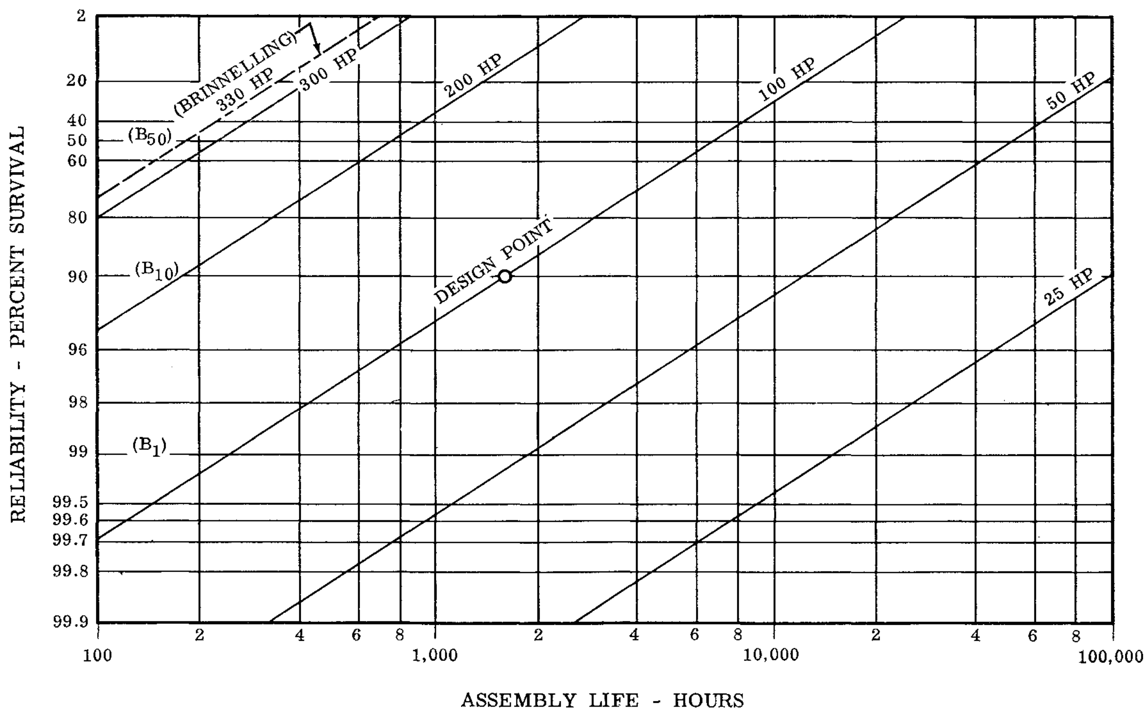


Fig. 5 Reliability of a typical traction drive operating at several different power levels. Size factor = 1.0; reduction ratio = 3.5:1; input speed = 3600 rpm.

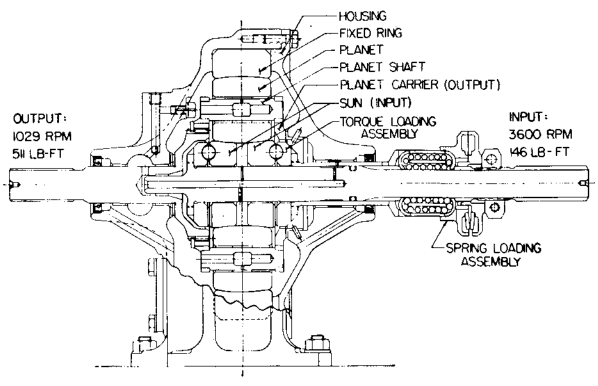


Fig. 6 Schematic cross section of the 100-hp roller traction drive used in performance tests. Reduction ratio = 3.5:1; rated power = 100 hp; rated input speed = 3600 rpm; rated B_{10} life = 1500 hr; overload capability = 330 hp.

Efficiency and Ratio Tests

These tests were run using the torque loading device rather than spring preload in the roller drive. The driver dynamometer was set on automatic speed control, whereas the torque on the absorber was increased in steps up to full load. Then, the speed was changed and the test repeated. Instrumentation accuracy was $\pm 0.125\%$ torque measurement and $\pm 0.05\%$ speed measurement. In all tests, GMR-1204 oil, which is a special automatic transmission fluid [(ATF) used in automotive transmissions] having naphthenic base with oxidation inhibitor and VI improver, was used. The housing was completely filled with oil kept at 150°F by circulation through an external heat exchanger.

Typical results representing the over-all efficiency vs input torque at 3000 rpm are plotted in Fig. 8. At all torques, the traction drive was quite efficient. Because of torque energized preload, this is particularly true at part-load operation. Data obtained at lower speeds showed traction drive efficiency as high as 97.8%. The gear drive efficiency under the same conditions was lower, particularly at low power levels. This efficiency data may however, be obscured by the bearing and seal losses, as well as by the fact that the housings were completely filled with oil which possibly resulted in more churning losses with gears than with rollers. Figure 9 plots ratio vs input torque at constant input speed of 3000 rpm. The change was only about $\frac{1}{2}\%$ from no load to full load, and is a combined result of rolling body deflections and elastohydrodynamic contact creep.

Of great interest in traction drive design is the ultimate tractive capacity of the rolling contacts. This can best be measured as coefficient of traction,

$$f_t = P_t/P_n \quad (3)$$

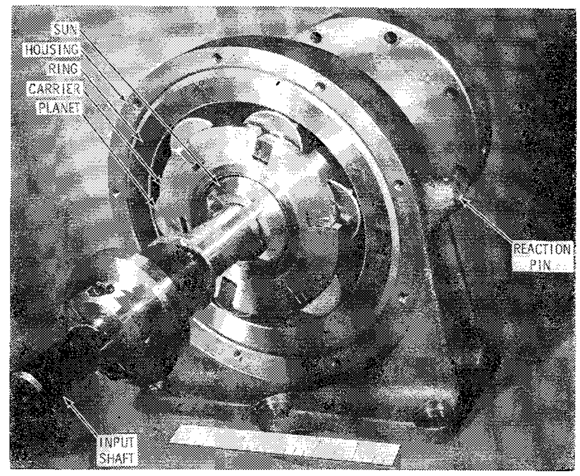


Fig. 7 Roller traction drive assembly used in performance tests is shown with housing cover removed.

where f_t = coefficient of traction; P_t = tangential tractive force; and P_n = radial normal force. Since the increase in torque is always accompanied by a slight increase in creep, it is possible to plot the two for a series of speeds. The creep, being essentially a speed ratio change, can be defined as

$$C = [(N_{\text{in}} - RN_{\text{out}})/N_{\text{in}}] \times 100$$

where C = creep, %; R = ideal (geometric) ratio at no load; N_{in} = input speed; and N_{out} = output speed.

To measure experimentally the coefficient of traction and creep, a known normal force was applied to the contacts at a constant input speed, and the torque was increased until impending gross slip was observed. Typical plot for a 3000-lb constant preload force is given in Fig. 10. In the range tested, the speed does not significantly affect traction-creep characteristics up to about 0.6% of creep. In all cases, at about 1% creep, a characteristic "knee" occurred in the curves, indicating that the contacts were on the verge of gross slip. As the torque was further increased, creep increased rapidly, and the operation became unstable and the drive would slip if torque were not removed. Normally, the ball ramp angle in the torque loading device would be chosen so as to force the drive to operate at a point indicated by the dashed line, thus limiting the creep and allowing ample safety factor.

The curves in Fig. 10, in reality, constitute empirical solutions to an important part of the classical elasto-hydrodynamic problem of the lubricated rolling contacts. In recent decade, this problem occupied numerous researchers who attempted to formulate it mathematically and subsequently verify the results by experimental tests. Excellent work has been done by Cameron, Johnson, Crook, Cattanneo, and

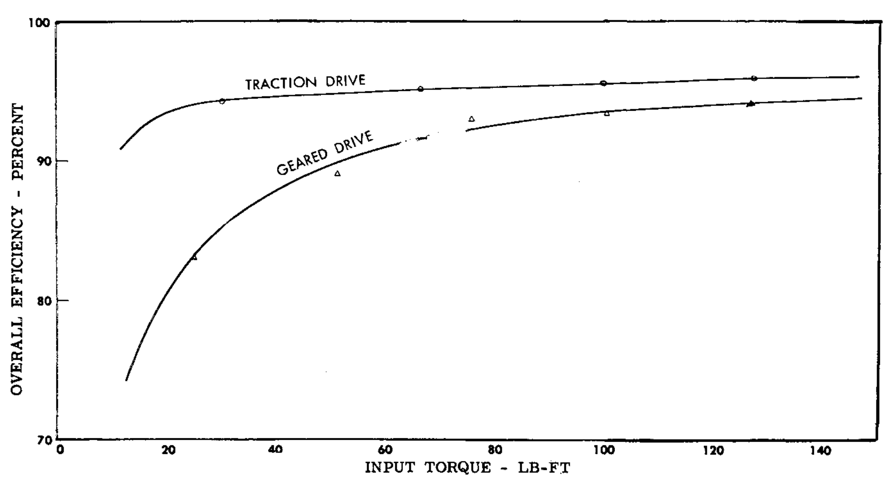


Fig. 8 Over-all efficiency of roller traction drive unit plotted as a function of input torque (upper curve) at 3000-rpm input. Planetary geared drive having similar size, ratio, and power capacity showed lower efficiency (lower curve). Both drives were operated with housings completely filled with GMR-1204 oil at 150°F. The data include losses due to churning, viscous shear, seals, bearings, and flexible couplings.

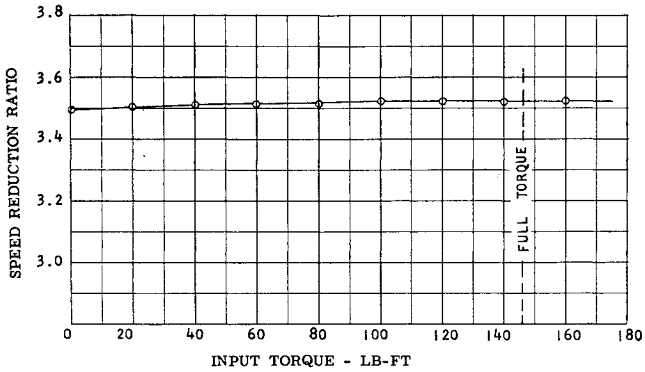


Fig. 9 Effect of torque on reduction ratio of the roller drive running at constant input speed of 3000 rpm.

Buffler in Europe, as well as Reichenbach, Sternlicht, Mindlin, Sibley, and Cheng in this country, to list but a few.[4–8] To solve such a complex problem, where laws of elasticity, kinematics, fluid dynamics, thermodynamics, and chemistry interact simultaneously, several simplifying assumptions were made. These, by necessity, introduced error into the solutions which made quantitative results difficult to assess.

The limiting traction coefficient or the slip coefficients from Fig. 10 is replotted as a function of contact normal load for a series of speeds in Fig. 11. The traction coefficient is strongly affected by speed and contact normal load. In all cases, increasing speed tends to lower the coefficient of traction. Increasing load, however, results in a peak coefficient at intermediate loads with subsequent drop at lower and higher load levels. This result points out that for optimum operation, the ball ramp loading device must be considerably nonlinear.

Noise and Vibration Tests

A series of tests were made using the noise and vibration test facilities of the GM Proving Grounds at Milford, Mich. To measure noise, vibration pickups were attached to the housing in all three planes while microphones were located at maximum noise intensity point 3 in. from the unit. Noise and vibration recordings were made while running the drive at various speed and torque combinations. First, the traction drive was tested, and then the tests were repeated with the gear components placed into the same housing, and all pickups and instrumentation setting unchanged. Typical results are shown in Figs. 12–14.

Table 1 Specifications of the planetary gear set used in the comparison study with the roller traction drive

Speed reduction ratio, fixed ring	2.55:1
Number of planets	4
Planet pitch diameter, 13 teeth	0.9946 in.
Sun pitch diameter, 47 teeth	3.5960 in.
Ring pitch diameter, 73 teeth	5.5853 in.
Ring outside diameter	7.000 in.
Normal diametral pitch	14
Normal pressure angle	20°
Helix angle	21°
Face width	0.845 in.
Total cumulative error	0.0005 in.
Rated input torque	150 ft-lb
Rated input speed	3000 rpm

The frequency spectra and over-all case vibration levels of the test cell alone, the traction drive and the gear drive, both transmitting 100 lb-ft of torque at 2000 input rpm, are given in Fig. 12. The over-all vibration level for the test gear (shown on the left) is 24 db higher than that of the traction drive. Disregarding input unbalance frequencies below 200 cps, it can be seen that the traction drive has a nearly "white" noise pattern, without predominant discrete frequency peaks. The peaks at 240 cps (lube pump), 410 cps (pillow block bearing noise), and 1000, 1500, 2000 (natural frequencies of housing elements), are external and could be changed by design, being really not a part of either drive.

The various frequency peaks indicated in Fig. 12 were identified as follows: 1, 2, 3 = unbalance frequency and harmonics; 4, 5, 6, 7, 8, 9 = tooth mesh frequency and harmonics; 10 = lube pump tooth mesh frequency; 11 = external pillow block bearing noise; 12 = housing cover natural frequency; 13 = ring natural frequency; and 14 = housing natural frequency. The gear drive exhibits a sharp peak at the tooth mesh frequency of 960 cps which was present up to its eighth harmonic. A very similar pattern is observed for airborne noise spectrum for the two devices, as shown in Fig. 13. Figure 14 shows the effect of speed and torque on the over-all noise levels of the two devices. It is interesting to note the quieting effect of torque on the noise of the roller drive.

Typical Drive Units

The results of the drive tests of the 100-hp prototype unit were very encouraging and led to the design of additional

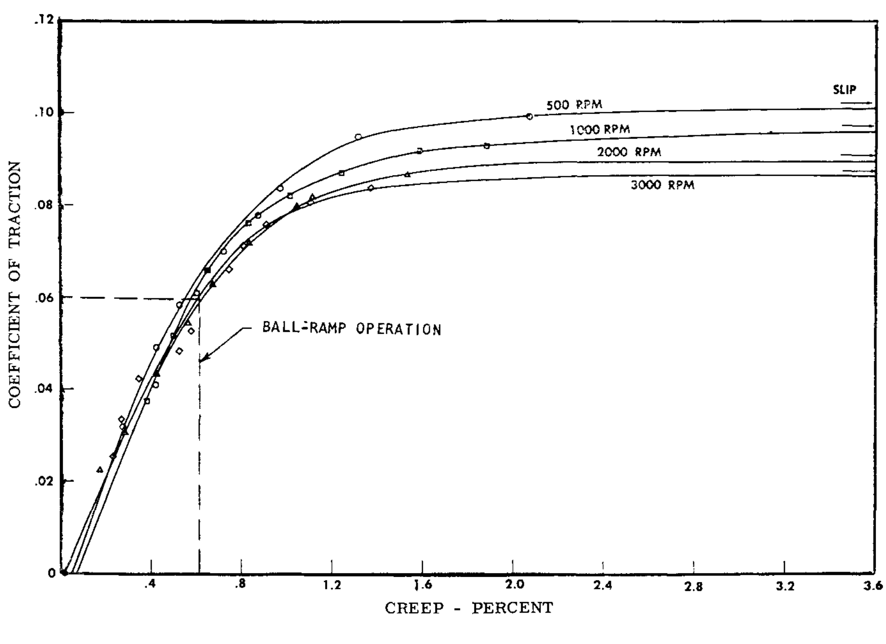


Fig. 10 Coefficient of traction vs creep at 3000-lb contact normal load and four different test speeds, GMR-1204 oil at 150°F.

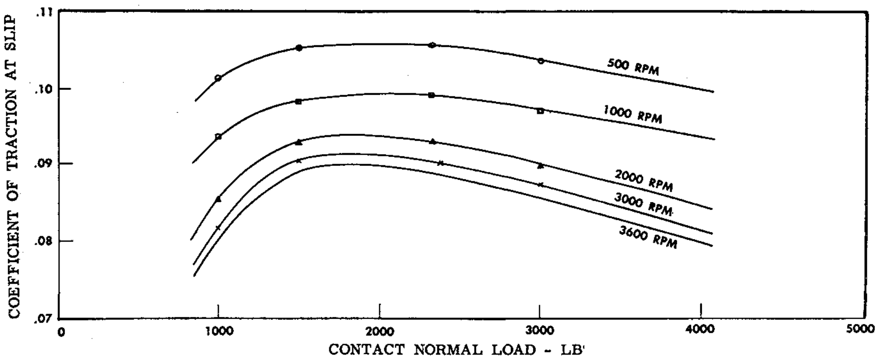


Fig. 11 Effect of contact preload on coefficient of traction at several input speeds, GMR-1204 oil at 150°F.

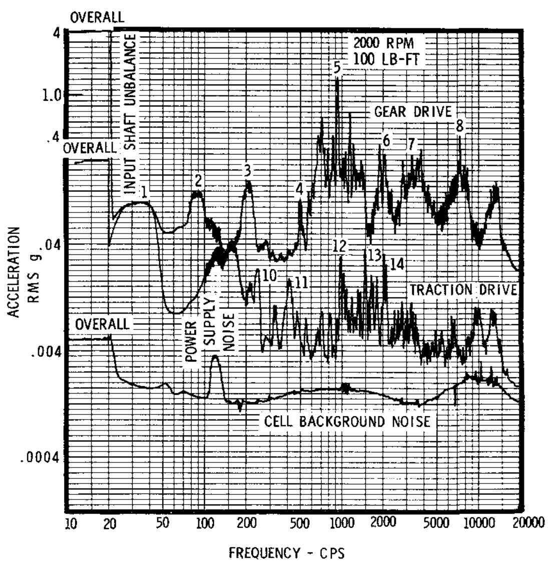


Fig. 12 Frequency spectrum of case vibrations for traction drive and gear drive running at 2000 rpm and transmitting 100 lb-ft of input torque (over-all levels are indicated on the left). Lower curve is test cell background noise with nothing running.

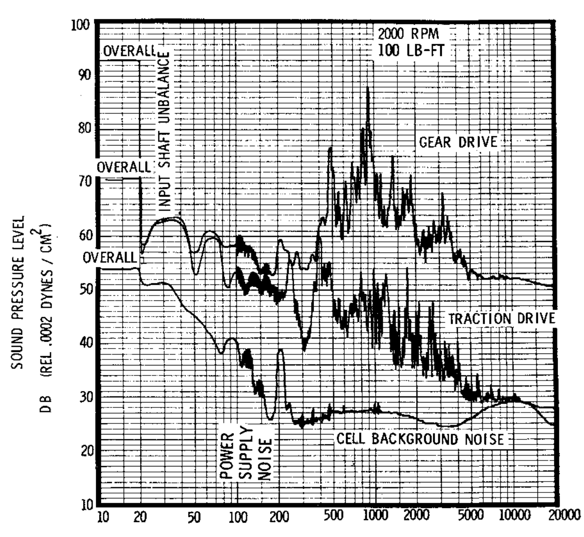


Fig. 13 Frequency spectrum of airborne noise for traction drive and gear drive running at 2000 rpm and transmitting 100 lb-ft of input torque (over-all levels are indicated on the left).

drives for applications requiring extremely quiet speed reduction, as indicated below. A 500-hp unit with a 5.65:1 speed reduction at 8000 input rpm is shown in Fig. 15. The unit is approximately 6 in. long, has the outside diameter of 12 in. and weighs 95 lb with oil. In full power tests, the unit performed at high efficiency with very low noise.

A reduction unit for 100-hp capacity at 2500-rpm input speed and 2.9:1 ratio with reversal of rotation, is shown in Fig. 16. The sun element is the input shaft, the carrier is the fixed reaction member, whereas the rotating ring and the outer housing are the output member. The unit is shown with rear cover removed.

A 6-hp unit mounted in the end frame of an electric motor is shown in Fig. 17. The drive has a 6.25:1 reduction ratio. In addition to transmitting power, it is also used as a bearing to support one end of the rotor. This drive unit is being currently used for main propulsion of the General Motors oceanographic submarine deep ocean work boat (DOWB). In addition to the preceding low-horsepower units, others of 5000, 10,000, or even 15,000 hp could be constructed.[9] Indeed, preliminary analyses indicate that larger units would be more efficient and would have improved weight-to-power ratio.

Summary

Roller traction drives developed by GM were subjected to a series of extensive dynamometer tests in which efficiency, power capacity, noise, and vibration patterns were investigated. The traction drive exhibited up to 97.8% over-all efficiency and showed extremely quiet operation, both in terms of airborne noise and vibration. In addition to low noise, the traction drive noise frequency spectrum was of the "white" type, without significantly outstanding frequency

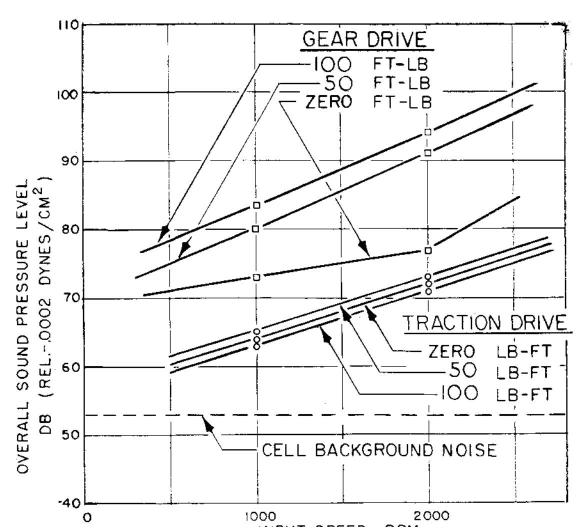


Fig. 14 Effect of speed on over-all airborne noise at three different torque levels. Microphonic pick-up was located at maximum intensity point 3 in. away from the housing.

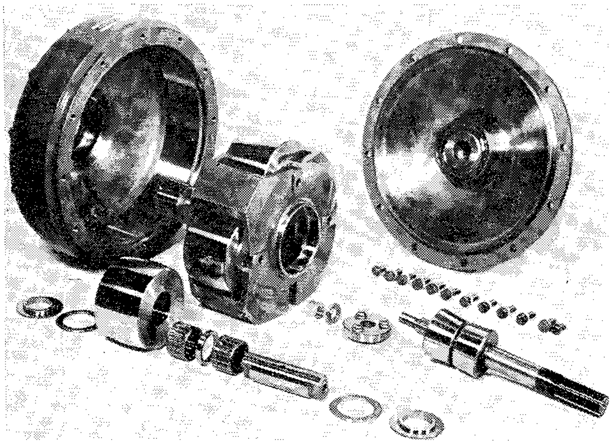


Fig. 15 Exploded view of the 500-hp roller traction drive having 5.65:1 reduction ratio at 8000 input rpm. The unit with oil weighs 95 lb.

peaks. Similar data obtained with an AGMA class 8 planetary gear at the same speeds and torques showed a very sharp peak at the tooth mesh frequency (and its harmonics) and 24-db higher over-all noise level.

Several unique design and geometry features make the life of the GM roller drive considerably longer than the life of similar traction drives. The torque energized preloading mechanism further improves the reliability, efficiency, and life.

Traction drives of this type can serve as practical speed reducers where low noise, low vibration levels, and high efficiency are of primary importance. These characteristics make the unit especially desirable for underwater propulsion and high-speed applications.

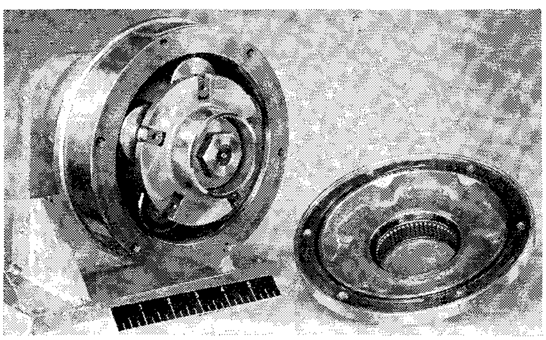


Fig. 16 100 hp speed reducer with housing cover removed. Rated input speed = 2500 rpm; rated input torque = 220 ft-lb; reduction ratio = 2.91:1 (rev.); o.d. = 7.0 in.

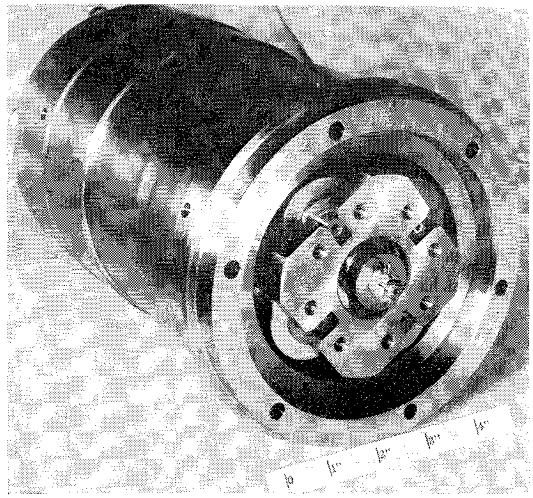


Fig. 17 6-hp roller drive having 6.25:1 reduction ratio at 3000 rpm is mounted in the housing of an electric motor. The drive also serves as bearing for the rotor support. The unit is used on the General Motors oceanographic submarine "DOWB."

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