Hybrid Geared Traction Transmissions*

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Roller traction drives and geared planetary drives have the common characteristic of multiple drive elements clustered symmetrically around a central member. This form of construction, adopted in order to maximize torque capacity, results in the drive input and output members being coaxial.

Geared planetary drives are associated with high torque delivered from a unit of low frontal area, whereas fixed ratio and variable ratio types of traction drives are renowned for their ability to accept high input speeds at moderate torque levels with least noise.

A combination of the two drive systems retains the feature of coaxial construction and results in a geared section which is capable of providing high torque and a traction section which transfers torque from the toothed pinions to the high-speed input member. Direct coupling of the traction planet rollers with the pinions of the geared unit allows generation of a speed ratio much higher than possible from a planetary unit.

This present work explores the basic configuration of geared traction drives, outlines geometric and structural factors to be considered in their construction, and describes current work on hybrid helicopter transmissions rated at 500 and 3000 hp.

Reduction Ratios from Coaxial Drives

A drawback of conventional planetary gears is that, as more planet pinions are fitted to increase torque capacity, the reduction ratio available decreases. Accordingly, highest torque from a planetary unit of given diameter is associated with many planet pinions and a low speed ratio. But a high speed ratio is possible when many pinions are present, if these pinions are driven directly rather than through the sun gear of a planetary unit. This effect is illustrated in figure 1 where a set of toothed pinions is shown between a sun gear and a ring gear. The lower central section is a conventional planetary drive with a stationary ring gear and input to the sun gear. The top left of the figure shows a ring gear driven by pinions, while the top right shows similar pinions driving an externally toothed gear, termed a sun gear for convenience.

In all cases the gear proportions are the same, with $S=4P$ and $A=6P$ where A, P, S denote the pitch diameter of the ring gear, pinion, and sun gear, respectively. Speed ratios obtained from the three arrangements are

Arrangement	Speed ratio	Numerical value
Planetary, fixed ring gear	$(A/S)+1$	2.50
Sun output, input to pinions	$-S/P$	−4.0
Ring output, input to pinions	A/P	6.0

These trends, true for $P<0.6$ S, illustrate that, with a coaxial gear arrangement formed from a sun gear, ring gear, and intermediate pinions, the larger reduction ratios are obtained by supplying input torque to the intermediate pinions and taking the high output torque from either of the two gears on the central axis. It follows that the one central gear not used to supply output torque is redundant.

Symbols

a	radius of sun roller
A,P,S	pitch diameter of ring gear, planet pinion, and sun roller, respectively

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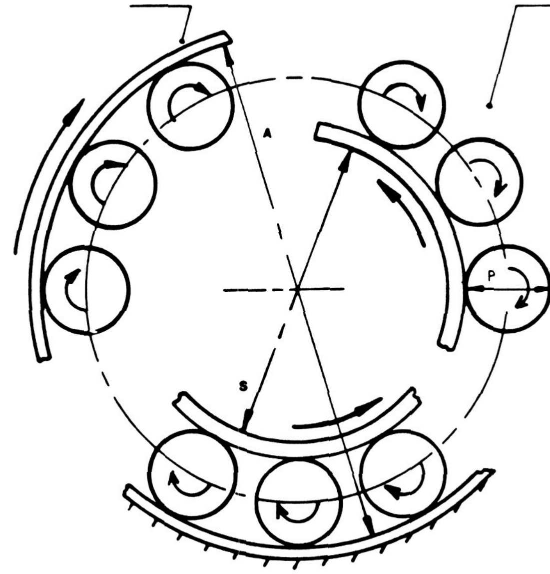

Figure 1. - Comparison of pinion drive arrangements with planetary drive.

B_1, B_2	cone semiangle at first row and second row rollers, respectively
b,c,d	axial spacing of retaining rings
E	Young's modulus
F_1, F_2	radial forces on roller retaining rings
g	radius from central axis of drive to center of ramp contact
k_1, k_2	radial stiffness of retainer rings
n	number of tractive contacts on sun rollers
$P_a, P_{x,1}, P_{y,1}$	normal force at each sun roller contact, x_1 roller contact, and y_1 roller contact, respectively
r	ramp angle of torque loading mechanism, measured from roller axis
S_c	maximum contact stress at roller contacts
T	torque transmitted
t	traction coefficient, defined as ratio of tractive force to normal load
w	roller contact width
x_1, y_1	first-row roller radii (see fig. 2)
x_2, y_2	second-row roller radii (see fig. 2)
α	angular location of second-row rollers (see fig. 2)
φ	pressure angle of gear teeth

Traction Drive to Toothed Planet Pinions

The foregoing section demonstrates that one way of overcoming the ratio limitation of a fixed-ring planetary unit, while still retaining torque capacity, is to drive the planet pinions individually instead of collectively through the sun gear. This procedure is clearly most effective when many pinions can be fitted, a condition at which a planetary unit offers high torque capacity but low ratio.

Given that a high ratio unit can be formed by driving a central output gear through many pinions, there remains the problem of transmitting torque to these pinions and ensuring that each pinion contributes only its designed share of total torque.

In this respect the introduction of a traction unit shows to advantage. The creep phenomenon associated with a traction drive determines that each traction roller receives a torque which is dependent on the creep rate between the sun roller and the planet roller interface. Essentially, the traction unit divides torque between its rollers by fluid film traction. This torque distribution then passes to any geared elements connected to the traction rollers. In the case of these geared pinions driving a collector gear, all the pinions and attached traction rollers are locked at a common speed, and so the only factor affecting creep rate and hence torque distribution is that of film thickness between the sun roller and the nominally identical pinions.

At the interface pressure used in traction drives the incremental stiffness of the fluid film is higher than the radial stiffness of the drive support structure. Moreover, traction roller surfaces can be held to tolerances approaching those associated with rolling contact bearings. These conditions, in combination with the load-balancing properties of a three-point support, lead to equality of film thickness at each roller contact, and, in consequence, ideal matching of torques between as many pinions as drive the gear.

Type of Traction Unit

The traction section of a hybrid drive consists of a central roller surrounded by one or more rows of planet rollers. With the simplest and traditional method of construction only one row or ring of planet rollers is present. The number of torque points then is limited to the number of planet rollers that can be fitted, and there is no benefit in respect of speed ratio.

In addition to distributing torque to geared pinions, however, the traction unit must span the radial distance between the drive central axis and the geared pinions. High ratios at the geared stage are associated with a large diameter output gear, and consequently a greater radial distance to the pinion axes than can be conveniently bridged by a simple sun, planet, and ring roller type of traction unit.

With a double row of planetary traction rollers the complexity of the design increases, but several benefits are realized:

(1) The number of traction contact surfaces is twice that of the number of rollers fitted.

(2) An increased number of rollers can be fitted.

(3) Stepped rollers pack compactly and also provide a gain in speed ratio.

(4) Dual rows of stepped rollers provide axial stability and assist in the fitting of a torque loading mechanism.

(5) An increased radial distance to the outer-row rollers makes it convenient for these rollers to drive geared pinions.

These factors can be seen in figure 2, which shows the structural arrangement of a two-row traction drive.

Speed Ratios

When stepped rollers are used, a speed ratio in the region of 3.5:1 at the traction section can be combined with a ratio between 8:1 and 10:1 at the geared stage. In this way the combined geared-traction assembly can provide overall ratios of about 30:1. This ratio is larger than feasible from two planetary units in series. Accordingly, geared traction assemblies show to best advantage at higher speed ratios than those which are naturally suited to geared planetary units.

For very high speed ratios (over 40:1 in total) three-row traction assemblies are appropriate, especially in the case of speed-increasing applications where the sun rollers of the traction unit have to rotate at speeds above 30 000 rpm.

Similarity with Pure Traction Drive

Previous work on pure traction drives (refs. 1 and 2) has demonstrated successful operation of the two-row roller configuration at high speeds and moderate torques. Experimental results from

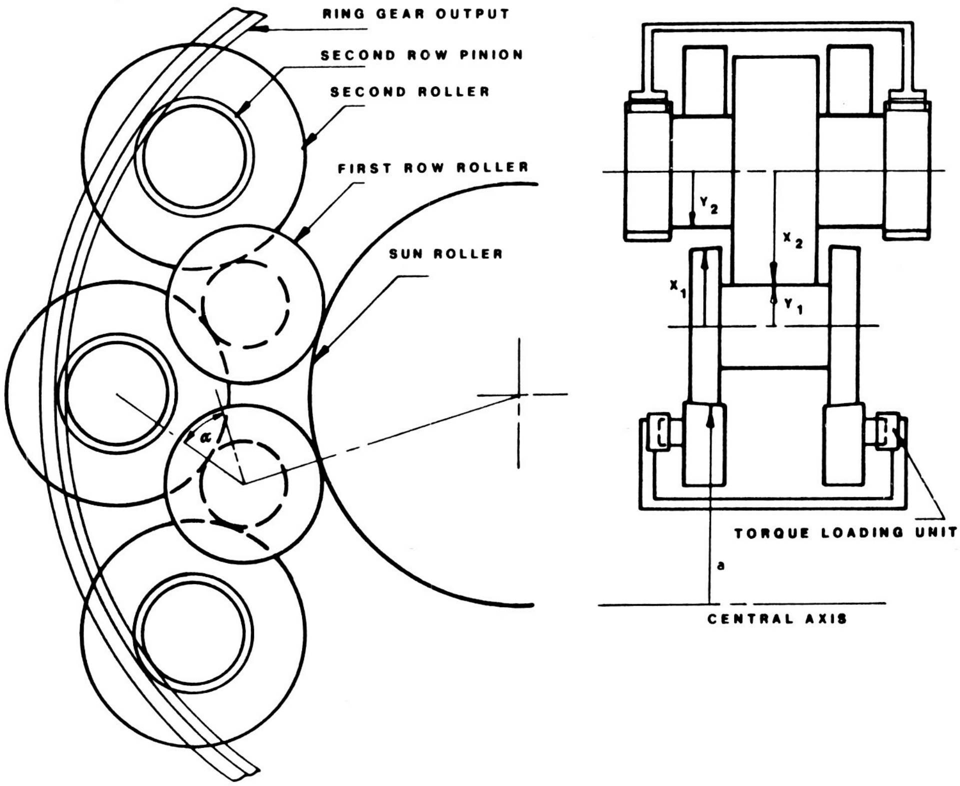

Figure 2. - Structural arrangement of two-row geared traction drive.

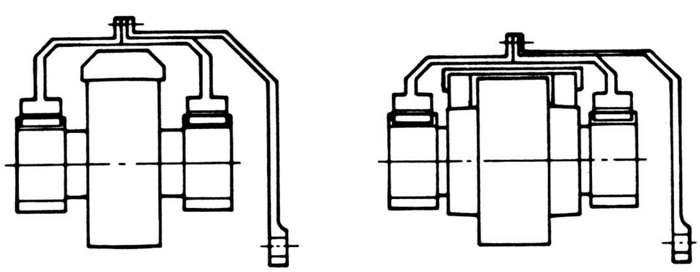

FIG.3A DUAL PINIONS AND SINGLE RETAINING RING

FIG.3B DUAL PINIONS AND LINKED RETAINER RINGS

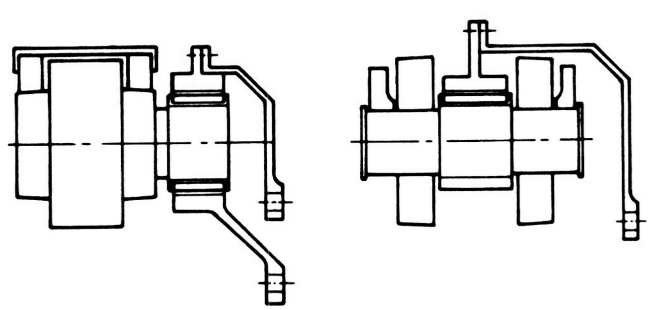

FIG.3C ASYMMETRIC ARRANGEMENT WITH SINGLE PINION AND EITHER INTERNAL OR EXTERNAL OUTPUT GEAR

FIG.3D SINGLE PINION WITH DUAL RETAINER RINGS AND INTERNAL GEAR OUTPUT

Figure 3. - Arrangements of second-row roller toothed pinion.

these designs, obtained in back-to-back tests, provide a background from which to select roller geometry, operating stresses, fluid temperatures, and design traction coefficient. Attempts to provide increased torque capacity from these pure traction drives led to the present work in which the torque capability and overall ratio are extended by combining toothed pinions with the outer-row rollers.

Arrangements of Second-Row Rollers and Geared Stage

Coupling of the outer-row traction rollers to the toothed pinions can take a number of forms (fig. 3). Symmetric arrangements with no resultant skewing motion on a planet shaft involve the use of a ring gear. Figures 3(a) and (b), for instance, make use of dual ring gears and either a single retaining ring or dual retaining rings.

Figure 3(d) is also symmetrical, with a single ring gear, whereas the asymmetric arrangement of figure 3(c) can involve either an internal gear or an external gear to combine the pinion torques. Design work demonstrates that for high torque applications the asymmetric arrangement of figure 3(c) offers the least weight as a result of keeping to a minimum the number of large-diameter torque transfer rings. It is for this reason that the helicopter transmission designs discussed are based on figure 3(c).

Traction Unit Components

Torque Loading Mechanism

Roller contact loads are made proportional to transmitted torque by means of a torque loading mechanism. The reason for introducing such a device is to avoid rollers carrying high contact loads when low torques are being transmitted; a result is a marked increase in fatigue life according to the difference between mean torque transmitted and maximum design torque.

Torque loading mechanisms invariably act on conical members in order that an axial motion of a cone takes up the radial elasticity of the drive. Thus the greater the radial stiffness of the traction section, the less the cone angle of the torque sensing rollers can be, and the lower is the ratio of spin/roll motion at the contacts. Low spin, of course, is associated with low losses and high efficiency.

For any applied torque there is a unique relation linking the ramp angle r of a torque loading mechanism with the radius g at which it acts and the contact radius a of the traction roller or cone. With a traction cone of semiangle B designed to work at a traction coefficient t defined as the ratio of tangential to radial force at each contact,

$$g \sin B = at \tan r$$

This nondimensional expression is independent of torque, as it ought to be. Typically, designing for a traction coefficient t of 0.06 and for $B = 3°$ results in the geometry, $\tan r = 0.872\ g/a$. If g/a is in the region of unity, the ramp angle r is 41°, which is higher than would be initially anticipated.

A further aim is that the torque loading mechanism operate: progressively with no frictional loss; that is, a pure rolling motion in the loading mechanism is preferred.

First-Row Rollers

Each roller in the first row is in equilibrium under the contact forces set up at one sun/roller contact and two second-row roller contacts. The number of these contacts is of course doubled if a two-sided configuration is adopted. Early designs of traction units tended to the use of true roller surfaces on all diameters, with shoulders to restrict axial motion. More recently, however, it has been demonstrated that axial positioning and stability of the first-row rollers can be maintained by certain convex/concave roller surfaces (ref. 1). This technique brings two major advantages. First, there is no possibility of flange scuffing, and, second, any flange power loss, as experienced in alternative designs, is eliminated.

First-Row Contact Conditions

The necessary normal load P_a at each contact point of the sun roller with a first-row roller is set by the torque transmitted T and the operating traction coefficient t. Thus, with n contacts on a sun roller of radius a,

$$T = tnaP_a$$

and the axial force necessary to set up the required normal load on each half of a split sun roller is

Sun roller axial force $= T \sin B/(2at)$

This axial force is imposed by the torque loading mechanism.

A first approximation to surface contact stress is given by assuming that the contacting surfaces are cylinders. Then with roller width w_1 roller radii a and x_1 semicone angle B_1 modulus E and contact stress S_c conventional contact stress analysis gives, with Poisson's ratio = 0.3,

$$w_1 a x_1 S_c^2 = 0.175\, E\, P_a(a+x_1)\cos B_1$$

from which the torque capacity can be expressed as

$$T = 0.57 S_c^2\, w a^2\, x_1 nt/[E(a+x_1)\cos B_1]$$

Second-Row Rollers

Second-row traction rollers transmit torque to the pinions of the geared section. It is through these rollers that torque reaction passes to the transmission housing; in consequence, only these rollers need support by bearings. Radial components of the two contact loads on each second row roller are reacted by the free-floating retaining rings. In this way the second row bearings do not carry radial forces, provided freedom is given to accomodate any radial growth of the retaining rings.

Second-Row Contact Loads

A normal force P_a acting at each sun roller/first-row contact results in a corresponding radial force $P_a \cos B_1$. This force is supported on two second-row rollers, each inclined at $90° - \alpha$ from the line of action of the radial force. The interface load $P_{y,1}$ between the first- and second-row rollers of radius y_1 and x_2, respectively, and semicone angle B_2 then is, ignoring the traction force difference,

$$P_{y,1} = P_a \cos B_1/(2 \sin \alpha \cos B_2)$$

Eliminating force P_a results in a general relation linking contact stress S_c torque T transmitted by the sun cones, and the roller contact width w_2 such that

$$S_c^2 w_2 tna \sin \alpha \cos B_2 = 0.88E\, T \cos B_1(y_1+x_2)/(y_1 x_2)$$

Retaining Ring Conditions

Internal radial forces acting on the traction rollers are contained by floating retaining rings. To avoid the retaining rings carrying edge loads the contact surfaces are made slightly convex, or the two rings are coupled so that any endwise creep and consequent edge loading is avoided. A form of retaining shoulder is included, however, which is normally inactive, but serves to keep the rings normal to the central axis when the unit is unloaded.

The radial load F set up by each second-row roller and reacted by the retaining ring, or rings, is derived from the vector sum of $P_{y,1}$ loads:

$$F = 2P_{y,1} \cos y = T \cos B_1/(tna \sin \alpha \cos B_2)$$

Different positions for the rings can be chosen (fig. 3), depending on any restriction in either the radial or axial direction. If a single retaining ring is chosen, this fits symmetrically around the outer-row rollers. Variable-ratio traction drives based on a spherical outer member adopt this type of construction. But separate retaining rings on each side of the outer roller are effective when axial length is not restricted. Further, this design is essential if each outer roller drives a single pinion; for a single pinion makes the radial forces asymmetric.

Forces Acting on Rings

A general arrangement from which any special case can be derived consists of two retaining rings, distances b and c, respectively, from the centers of the outer-row roller. The midpoint of a toothed pinion is distance d from the centers of the adjacent ring. Forces acting on each outer row roller must be in equilibrium and comprise the reaction forces F_1 and F_2 from the retaining rings, $P_{x,2}$ from the first-row rollers, and the tangential tooth load W_t from the pinion. The radial load from the pinion is accordingly $\pm W_t \tan \theta$, the plus and minus signs being appropriate to a collector gear with external teeth and internal teeth, respectively. Conditions to be satisfied are

Radial equilibrium, $F_1 + F_2 + W_t \tan \varphi = P_{x,2}$

Radial stiffness, k, $F_1/k_1 = F_2/k_2$

Moment balance, $bF_1 = cF_2 + (c+d) W_t \tan \varphi$

Forces on each retaining ring are found to be

$$F_2 = P_{x,2} b/(b+c) - W_t \tan \varphi (b+c+d)/(b+c)$$

$$F_1 = P_{x,2} c/(b+c) + W_t \tan \varphi (d)/(b+c)$$

and the ring stiffness are related by

$$k_1 = k_2(c/b) + k_d(c+d)/b$$

where k_d is the total drive stiffness as measured, for instance, by the ratio of ring radial deflection to applied load.

Equivalent expressions for ring loads in terms of a known torque T applied to the sun rollers are

$$F_2 = \frac{bT \cos B_1}{2(b+c) tna \sin \alpha \cos B_2} - \frac{(b+c+d) W_t \tan \varphi}{b+c}$$

$$F_1 = \frac{cT \cos B_1}{2(b+c) tna \sin \alpha \cos B_2} + \frac{d W_t \tan \varphi}{b+c}$$

Conditions for Equal Ring Forces

Ring forces F_2 and F_1 can be equal at the conditions set by

$$(b-c) P_{x,2} = (b+c+2d) W_t \tan \varphi$$

when b, c, and d must be adjusted appropriately. For the symmetrical condition of $b=c$, the only solution for equal loads is with $d=-c$; that is, with the pinion equally spaced between the two retaining rings.

Helicopter Transmission Design

Present work on geared traction-drives centers on their application to helicopter main gearboxes. These designs are based on the torque rating and overall speed ratio of existing commercial transmissions, principally those of the OH 58 and the UH 60A aircraft. Benefits attempted in these applications can be summarized as

(1) High speed-ratio from coaxial geared/traction section
(2) Elimination of a speed reduction stage
(3) Weight benefit at speed ratios higher than naturally suited to geared planetary units
(4) Equal loading of toothed pinions by tractive creep mechanism
(5) Noise reduction in roller drive section.

Low Ratio 500-hp Transmission

This transmission follows the speeds and torques appropriate to an OH 58 unit uprated to 500 hp. The input speed to the transmission is only 6060 rpm as a result of a double reduction stage being present in the engine reduction gearbox. The overall speed ratio required in the transmission is 17.4:1 for a main shaft speed of 347.5 rpm.

Since a bevel gear train is necessary in the transmission and a speed reduction is taken at this point, there remains only a modest ratio to be generated by the geared traction unit. Speed ratios selected for each section of the drive are

Geared traction section	8.5:1
Input bevel train	2.05:1

While a ratio greater than 8.5:1 would be preferable at the traction section, this is not feasible, given the input speed of 6060 rpm. This ratio, however, contrasts with the 4.7:1 ratio taken in the planetary unit of the OH 58 design.

Figure 4 shows a schematic arrangement of the drive elements. The combined geared-traction section fits around the transmission main shaft exactly like a planetary unit. But the carrier frame for the final row rollers and pinions is held stationary by the housing, against which torque is reacted, while output torque to the main shaft is taken from the ring gear. Input torque to the bevel pinion is carried by the bevel gear to a ramp and roller type of torque loading unit. This unit drives the dual-sun rollers. Compression springs also are included in the torque loading unit so that, at zero and low torques, the traction rollers experience a contact force.

Ten sets of traction rollers in each row results in there being 20 torque transmitting contacts at the sun rollers. Similarly, there are 10 pinions meshing with the output ring gear. This arrangement gives the benefit of conformal tooth contacts at the high-torque section of the transmission. The toothed output pinions are located in hollow, splined outer-row rollers. Earlier designs included pinions which were electron-beam welded to the rollers; but the splined design was adopted to avoid a weld in an area of high cyclic stress and, also, to permit regrinding of the pinion teeth.

Bearings locating the outer-row rollers are mounted in sleeves that can roll slightly in the housing surrounding the bearing. In this way radial growth of the outer roller axis, allowed by expansion of the retaining rings, does not result in a radial load on the bearings. Accordingly, bearings on the outer-row rollers react only tangential loads into the carrier frame.

To minimize height, the shaft carrying the bevel gear is supported at each side of the traction unit. The top bearings on this shaft react bevel gear thrust into the carrier frame and to tie bolts which connect the middle and lower housings of the transmission. Three main housing sections are present, although a two-section design is possible. The top section contains the main lift bearings, and the transmission torque-reaction spigots; the middle section contains the traction unit and allows this unit to be assembled entirely on the bench; the lower housing carries the input bevel pinion, the lubrication pump, and associated items.

Design weight for the low ratio design is 146 lb with an all-steel carrier frame. This weight is competitive with the 141 lb of an uprated OH 58 design, detailed in reference 3, as a result of incorporating lower gear and bearing stresses.

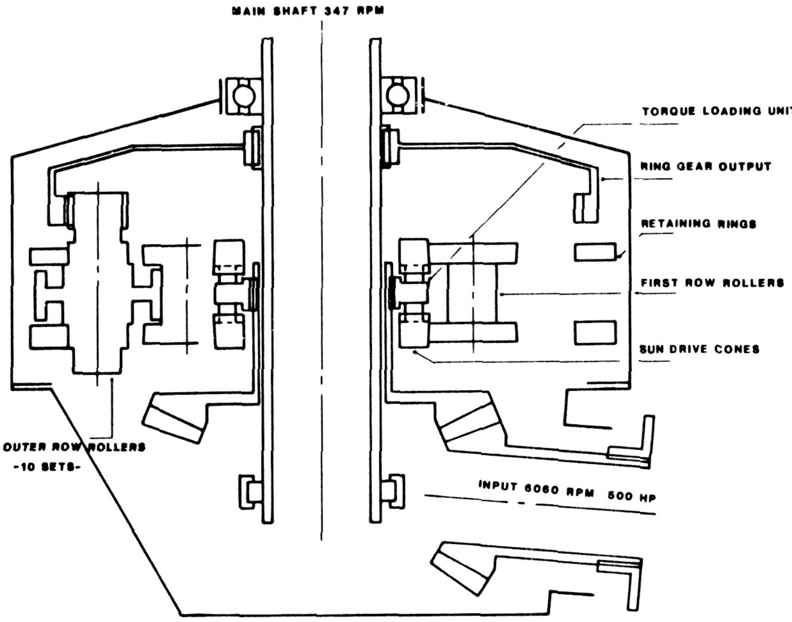

Figure 4. - Schematic of 500-hp helicopter transmission with 6060 rpm input speed.

High Ratio 500-hp Transmission

Parametric studies on designs with different speed ratios but the same output torque capacity have demonstrated that hybrid geared traction drives can provide gains in overall ratio with only minor weight increments. Thus, from a weight standpoint, a geared traction drive shows to best advantage at high speed ratios. On this basis a 500-hp helicopter transmission was investigated based on a main shaft speed of 347.5 rpm and an input speed of nominally 36 000 rpm. These speeds, with an overall speed ratio of 103:1, correspond to those of the OH 58 transmission from the main shaft to engine.

Three reduction stages are present and consist of

(a) A final gear-reduction stage based on seven pinions driving a rotating ring gear
(b) An intermediate stage traction drive in which the outer-row rollers drive the toothed pinions
(c) A bevel reduction stage which accepts an engine speed of about 36 000 rpm.

A schematic arrangement of the transmission is shown in figure 5. Engine input power passes through an overrunning clutch to a bevel train of 3.75:1 ratio. The remaining 27.5:1 reduction ratio is taken in the geared traction section. This high ratio contrasts with the lower ratio of 8.5:1 necessary when the transmission input speed is 6060 rpm as a result of retaining the engine reduction gearbox. With the high ratio design, however, the transmission housing diameter increases in consequence of the larger second-row rollers. Seven sets of rollers and toothed pinions are present in the traction section.

A later design of the high ratio transmission involves inversion of the bevel gear and the traction drive section in order that the transmission can fit conveniently on the same test stand as the low ratio design. An overrunning clutch unit is omitted from the design, and the input shaft is connected directly to the bevel pinion. Bearings for this high-speed pinion consist of a front roller bearing, a tandem pair of angular-contact bearings to react thrust, and a forward-mounted bearing to preload the tandem bearing pair.

Otherwise the design follows closely that of the low ratio unit. The bevel gear is straddle-mounted across the traction section, oil transfer tubes and retaining bolts pass through hollow first-row rollers, toothed pinions are splined to the second-row rollers, and the main housing separates into three sections.

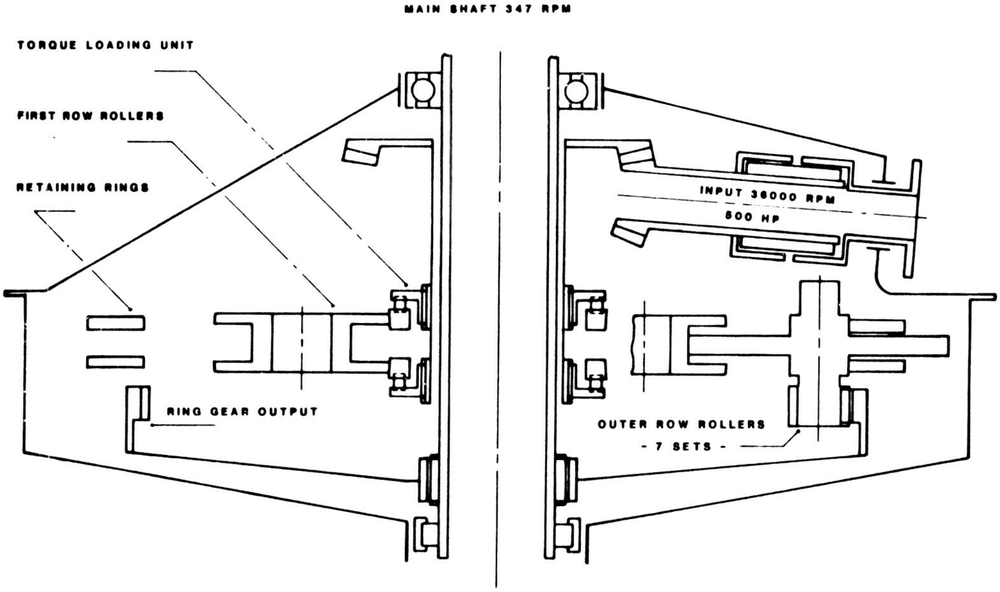

Figure 5. - Schematic of 500-hp helicopter transmission with 36 000 rpm input speed.

Weight estimates for this design give a dry weight of 158 lb for a main rotor torque of 90 680 in-lb. The incremental weight gain from the low ratio design is only 12 lb, a feature that emphasizes the benefit of including all the reduction trains from the engine to the main shaft in one transmission. This trend to eliminate gear trains in an engine reduction gearbox also results in reduced drive train losses.

3000-hp Transmission

Attempts to design traction drives of high power capacity coincided with the development of several twin-engine helicopter transmissions rated at 3000 hp. Accordingly, it proved convenient to adopt the torques and speeds of the UH 60A geared transmission as a design specification for a hybrid traction drive.

The overall speed ratio required is 81:1 between the main shaft at 258 rpm and the engine at 20 900 rpm. This ratio is spanned in three stages consisting of

(1) A final, geared stage with six pinions, 10:1
(2) Intermediate traction unit, six roller sets, 3.39:1
(3) Engine reduction and combining bevel, 2.4:1

The ratio of 34:1 taken in the geared traction section is sufficiently high that only one bevel gear stage is necessary to match the engine speed of 20 900 rpm. In consequence, to avoid a second bevel train, the engine axes intersect the transmission main shaft and are inclined outward at 30° from the centerline of the fuselage. In this way the number of speed reduction stages is kept to the minimum of three, and the engines are widely separated.

Figure 6 shows a schematic layout of a 3000-hp transmission, being a section through the main shaft, one engine input, the traction section, and the tail drive gears. Each engine drives through an overrunning spring clutch to a bevel pinion and a combining bevel gear. The bevel gear thus carries the sum of engine powers.

Tail drive power is extracted from the bevel gear shaft by a train of spur gears. This arrangement ensures that the traction drive carries engine power less tail drive power. With a free choice of tail drive position, one stage of spur gears could be eliminated. The arrangement adopted, however, allows the exact speed ratios and drive shaft positions of the UH 60A transmission to be duplicated.

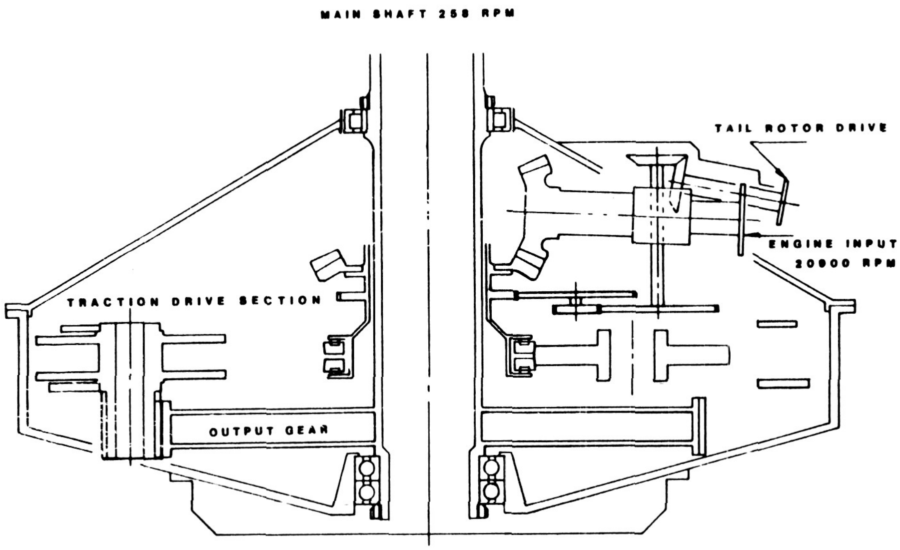

Figure 6. - Schematic of 3000-hp twin-engine helicopter transmission.

Input bevel pinions are larger in diameter than would normally be the case because the bevel gear is limited in minimum diameter by having to pass around the main shaft; the low ratio in the bevel train then sets the bevel pinion at a larger diameter than necessary from a tooth load standpoint. The torque loading unit is driven from the main bevel gear and serves to load equally each side of the split sun rollers. Outer-row rollers are located by bearings which fit directly in the main housing at the lower end and in a subsidiary cover at the top end. In this way the geared traction section is held independently of the top section of the transmission housing. Similarly, the engine input sections and the combining bevel gear and bearings are contained in the upper housing.

The traction unit serves to divide transmitted torque equally between six final-stage pinions which drive the output gear. This output gear is of the external tooth type. The alternative design with internal teeth does not show to advantage from a weight standpoint, since the benefit of a slightly narrower face width on the combining gear is negated by a gain in support structure weight for the ring gear. The combining gear is mounted directly on the main shaft, thus avoiding the need for separate support bearings.

Weight estimates for the 3000-hp design yield a total of 1020 lb for a dry unit without accessory drives. Some 4-percent reduction from this figure appears possible by design refinements in the areas of the housing and the combining gear webs.

Discussion and Summary

Completion of design work on the 500- and 3000-hp transmissions enables general trends to be established for hybrid geared traction drives.

1. A weight advantage can be demonstrated at speed ratios in the geared traction unit, which are higher than those naturally suited to a geared planetary unit. Thus, if the ratio in the coaxial section of the drive is in the region of 3:1 for a single planetary unit, 9:1 for a double planetary unit, and 30:1 for a three-stage unit, the advantage from a weight standpoint lies with planetary units. In these cases, however, the large number of planet pinions present require a corresponding number of support bearings.

If the ratio per planetary stage is greater than about 3.8:1, then the geared traction drive shows a weight advantage.

2. The housing diameter for the geared traction drives is larger than that of a wholly geared unit, especially when high ratios are needed.

3. Noise levels for the geared traction transmission are anticipated to be lower than for an equivalent geared system. Numerical data on this effect are available from pure traction units, but confirmation in the case of high power awaits test results from a 500-hp unit now at the assembly stage.

4. The coaxial arrangement of a geared traction unit enables its installation in a helicopter transmission to follow procedures established by the use of geared planetary units. The trend to high ratios in the geared traction unit means that there is little advantage in employing an epicyclic arrangement in place of the fixed-axis design. Restrictions on the diameter of the main shaft are not present as a result of the sun traction rollers being larger than the sun gear of an equivalent geared drive.

5. Benefits in respect of traction drive sizing and life result from extracting tail drive power prior to the geared traction unit.

6. Torque capacity and overall weight are dependent on the allowable contact stress at the traction surfaces. Test data from smaller traction drives indicate that surface stressing to levels between those experienced in gear teeth and rolling-contact bearings is appropriate. This is a result of the elliptical contacts at roller surfaces, with a minor axis in the direction of rolling, being able to accomodate structural misalinement more easily than gear teeth of wide facewidth.

7. The tendency for geared traction units to show advantage at high speed ratios means that the total drive train from engine to main shaft must be included in a design. The least advantage is present, as with the low ratio 500-hp design, when an existing engine reduction gearbox has to be accomodated.

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