Helicopter Transmission Arrangements with Split-Torque Gear Trains*

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A need for progressive improvement in the performance of helicopter propulsion systems leads the transmission designer to examine alternative methods for transferring power from the engines to the rotors. Predominant in this effort is the problem of designing main rotor gearboxes having higher power to weight ratios with increased life and reliability.

Development and uprating of components used in production gearboxes has, in the past, enabled transmission designers to match improvements elsewhere in the propulsion system. This short-term form of development is attractive since the design loads and problem areas are known in advance and the improved design can be incorporated into an existing transmission with minimum changes. But the uprating of conventional components results in progressively smaller gains and, as is historically demonstrable, contributes little to the problem of weight reduction for there is a natural tendency to trade-off any potential weight benefit with component life and reliability.

As an alternative to component development, this work argues the case for improved drive-train configurations. In particular, the use of torque-splitting gear trains is proposed as a practicable means of improving the effectiveness of helicopter main gearboxes; the work thus makes use of and extends concepts presented earlier in references 1 and 2. The potential for major weight reduction exists, but since significant changes to the propulsion configuration may be required, the development of such gear trains is necessarily longterm. Of equal importance, however, is that no advance in technology level is required, for split-torque gear trains merely involve novel arrangements of well developed components such as gears, shafts and bearings.

Areas of Application

Traditional gear train arrangements for main rotor gearboxes comprise spur, bevel, and planetary gear units. These units must be sized for main rotor power if located between the main rotor and the point of engine power combination; otherwise the drive units must be sized for single-engine power.

Now split-torque gear trains provide parallel drive paths for the transmission of power, and, as shown later, the torque split can be achieved through spur, bevel, or planetary gear trains. In consequence the transmission designer has more options to call on than if the choice were limited to conventional gear arrangements which transmit all power through a single path. The challenge to the designer then lies in exploiting the advantages of torque-splitting devices at an appropriate point in the drive train.

While weight reduction remains the principal aim, split-torque drive trains provide a means of forestalling technology limitations set at one extreme by advanced, high-speed engines and, at the other extreme, by the high torques required by the main rotors of heavy lift helicopters (HLH). Further, the ability in some arrangements to maintain separation of individual engine drive trains until the final speed-reduction stage provides redundancy between drive paths from each engine.

Consideration of such properties in relation to transmission designs spanning the light observation helicopter (LOH) to HLH aircraft allows indentification of the following areas of application for split-torque gear trains:

(1) Offset, final-stage reduction gears based on a combining gear driven by multiple pinions

(2) Intermediate speed-reduction stages which torque-balance drive trains upstream and downstream through balanced reaction members

(3) With high-speed engines of low torque capacity, an engine reduction stage based on a high-ratio, counter-rotating epicyclic unit

(4) With engines of high torque capacity, an engine reduction stage based on dual bevel pinions driven through a torque dividing mechanism

(5) Coaxial, final-stage reduction gears for transmission of very large torques to the main rotor. These gear trains are appropriate to HLH transmissions with output torques greater than so far encountered on a production aircraft. Owing to the shift in emphasis from items (1) to (3), however, these coaxial drive trains are eliminated from discussion in the present work.

Methods of Torque Splitting

Mechanisms for achieving torque division can be subdivided into

(1) Static-load torque dividers characterized by having no internal relative motion of the loaded elements other than the small adjustments necessary to accommodate manufacturing tolerances and structural deflections. In consequence, this type of torque-divider provides no change in speed ratio across its input and output members. Typical examples of static-load torque dividers, discussed later, include torque tubes, pivot beams, flexible beams and a number of three-shaft geared mechanisms.

(2) Rotary torque-dividers having continual relative motion within the mechanism. In this way a speed-reduction ratio can be generated between the input and output members concurrently with the division of torque. This group of mechanisms includes epicyclic torque dividers, double-helical gears with axial freedom, and floating idler pinions.

(3) Reaction-balance torque dividers which combine the properties of static-load and rotary-torque dividers to achieve a speed reduction while retaining the advantage of torque equalization through a static reaction member.

Static-Load Torque Dividers

Elastic deformation

Elastic deformation of structural members provides a simple means of dividing torque between a pair of gears while avoiding the introduction of wearing elements such as bearings: examples include flexible cantilever-beams and torsionally flexible drive shafts. The equality of torque division provided by elastic members depends on

(1) Indexing accuracy on assembly

(2) Attainment of equal and low stiffness of each drive path

(3) Over a range of torques, equal deflection of the housings and bearings supporting the gears.

Low stiffness in the drive paths allows the gears and their support bearings to experience sensibly equal loads despite the runout errors arising from machining tolerances. But to achieve confidence in load sharing between the gears over a period of operation necessitates continual monitoring or periodic checking of the torque levels in each drive path. High-performance marine reduction gears provide an example of load sharing through torsionally flexible drive shafts. In this application a combining gear is driven by pairs of pinions, near equality of tooth loads being assisted by the low torsional stiffness of the drive shafts to the pinion.

Figure 2 illustrates a flexible-beam mechanism for dividing an input torque between two coaxial members. Variants of the mechanism have been developed commercially since the device offers simplicity and the ability to attain high torque capacity by duplication of the beam elements.

Zero-Stiffness Mechanisms: Limited Motion

Ideal torque distribution between a pair of gears is attained with a torque-dividing mechanism having zero stiffness. This characteristic contrasts with the finite stiffness of elastic torque dividers. Examples of a zero-stiffness device include the axial pivot-beam arrangement of figure 1 as used in Mil helicopter transmissions to divide the torque from each engine between a pair of bevel pinions.

The ratio of the torque division varies with the length of the pivot arms and the radius of the pivot points as noted in table I; but equal-length arms at a common radius clearly result in each output torque being one-half of the input torque.

Radial equivalents to figure 1 can be devised, but each pivot beam still requires three bearings. These bearings, of course, are the critical elements of a torque divider for they are loaded and have no relative motion across the races other than small-amplitude oscillation, a condition which favors the selection of rod-end bearings. A future possibility with figure 1, however, involves the use of elastomeric bearings on the pivot beams to eliminate possible wear and life problems. In this case the torque-divider would become a member of the elastic group.

In normal operation, with drive paths of equal stiffness, each end of a pivot-beam in the torque divider experiences a motion amplitude not exceeding about 1 mm. Then, if motion limit stops are included allowing only 2-mm amplitude, a complete failure in one of the gear drive paths leads to the total input torque being carried by the remaining drive path. In this way the motion limit stops, possibly in the form of a large-clearance spline, provide redundancy such that failure in either drive path transfers the total torque to the remaining drive path.

Zero Stiffness Mechanisms: Unlimited Motion

Alternatives to the pivot-beam mechanism of figure 1 include a family of geared torque dividers. These mechanisms comprise three-shaft differentials capable of providing two torque outputs, each equal to half the torque applied to the input member. The commonest example is, of course, the bevel differential as used in vehicle rear axles (fig. 3(a)). General terms for the output torques are summarized in table I from which it is clear that these torques are equal when the side bevels B_1 and B_2 are the same diameter.

Figure 3(b) illustrates the spur-gear equivalent of figure 3(a). In this case the pairs of pinions mesh with each other, but only one sun gear meshes with each pinion. The torques delivered by each of the output sun gears are given in table I; when $S_1 = S_2$ the output torques are each equal to one-half of the input torque. A similar result obtains (table I) if the two sun gears are removed and replaced by two annulus gears as in figure 3(c). The double-annulus design is somewhat shorter than the double sun design on account of the reduced face width (conformal contact) required at the annulus-pinion mesh.
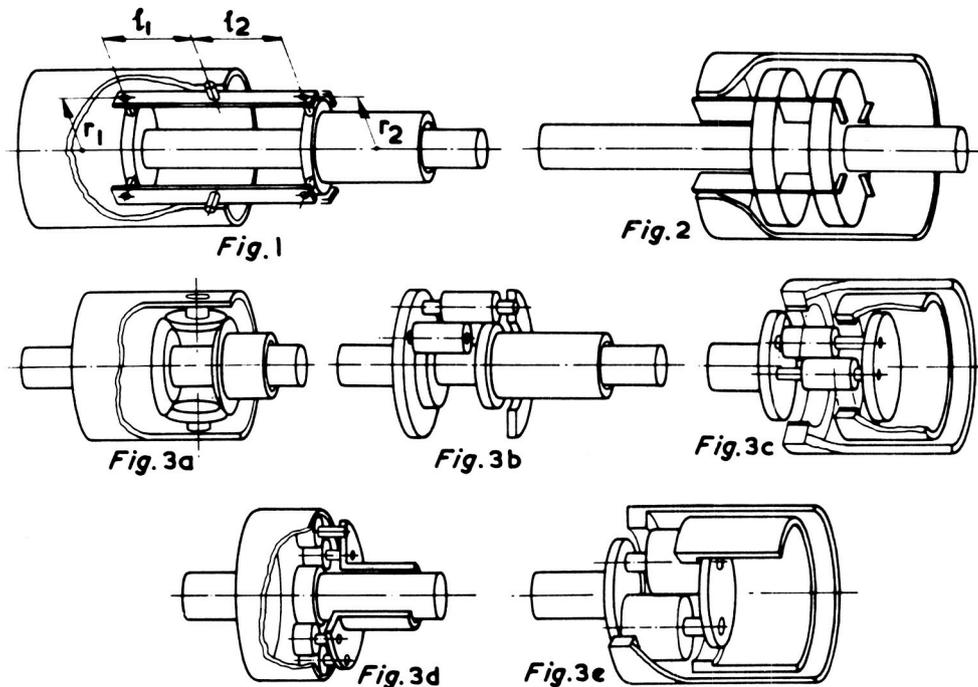
Figures 1 - 3. - Static-load torque dividers.

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TABLE I. – TORQUES DELIVERED BY ZERO-STIFFNESS MECHANISMS

Figure	Member carrying input torque	Output torques obtained	Condition for equal output torques
1	Beam carrier	$T_1 = T_c/[1 + (L_1/L_2)(r_2/r_1)]$ $T_2 = T_c/[1 + (L_2/L_1)(r_1/r_2)]$	$(L_2/L_1) = (r_2/r_1)$
3a	Planet carrier	$T_{B1} = T_c B_1/(B_2 + B_1)$ $T_{B2} = T_c B_2/(B_2 + B_1)$	$B_1 = B_2$
3b	Planet carrier	$T_{S1} = S_1 T_c/(S_1 + S_2)$ $T_{S2} = S_2 T_c/(S_1 + S_2)$	$S_1 = S_2$
3c	Planet carrier	$T_{A1} = A_1 T_c/(A_1 + A_2)$ $T_{A2} = A_2 T_c/(A_1 + A_2)$	$A_1 = A_2$
3d	Annulus	$T_c = T_A[1 - (S/A)]$ $T_S = T_A(S/A)$	$A = 2S$
3e	Planet carrier	$T_{A1} = T_c P_1/(P_2 + P_1)$ $T_{A2} = T_c P_2/(P_2 + P_1)$	$P_1 = P_2$
4a	Sun gear	$T_A = T_S(A/S)$ $T_c = T_S[(A/S) + 1]$	Not possible

Notation:

A	annulus diameter	T_A	annulus torque
B	bevel gear diameter	T_B	bevel gear torque
P	planet pinion diameter	T_c	carrier torque
S	sun gear diameter	T_S	sun gear torque

Lesser-known torque dividers include the idler-pinion differential shown in figure 3(d). The inclusion of idler pinions allows the sun gear and the planet carrier to deliver equal output torques, each one-half of the input torque, when the annulus diameter is twice that of the sun gear; then $(A/S) = 2$ as shown in table I.

The two-pinion differential of figure 3(e) is developed from figure 3(c) by omitting all but one set of pinions. It can also be recognized as a rotary equivalent of the pivot-beam arrangement shown in figure 1. But while figure 3(e) brings the economy of a minimum total of gears and bearings, the use of a single gear mesh for reacting the transmitted torque leads to a larger size in comparison with the other geared torque splitters.

General Property of Torque Dividers.

A general property of all the static-load torque dividers outlined (figs. 1 to 3) is that, if the member used for torque input is held stationary, a motion of one output member produces a reverse motion of the remaining output member. Further, if equal output torques are required, the forward and reverse motions noted must have the same amplitude. It follows that, for equal output torques, the mechanisms of figures 1 to 3 must provide a speed ratio of 1:-1 across the output members when the input member is held stationary.

The above property also holds true for the rotary torque dividers of figure 4, discussed in section 5. But in the case of figure 4(a), any gear trains downstream of the epicyclic unit must be considered as part of the torque dividing mechanism, for these gear trains rectify the torque imbalance between the planet carrier and annulus members.

Combined Torque-Divider and Speed-Reduction Gear

The development of static-load torque-dividing mechanisms leads to the examination of gear trains that perform the dual function of speed reduction and torque division. An attraction of such gear trains is the need for fewer components than with a separate torque-dividing mechanism and gear train. Three distinct types of torque-dividing gear trains are summarized below; these types are incorporated in main rotor transmission arrangements described later.

Epicyclic Torque Divider

Epicyclic speed-reduction gears in helicopter transmissions normally have the annulus grounded to the transmission housing. But by accepting two counterrotating outputs, with a single-input drive, the torque normally reacted to the transmission housing can be utilized as a secondary drive path as shown in figure 4(a).

Recombination of the separate drive paths prior to driving the main rotor can be achieved in a number of ways, some of which are outlined in references 1 and 2. The gear trains driven through the planet carrier and the annulus are always torque-balanced by the epicyclic unit so that the gear tooth loads in each drive path are predictable. The division of torque in the epicyclic torque divider of figure 4(a) is not equal but corresponds to the ratios given in table I. It follows that the output torques differ in the ratio $T_c/T_A = (S/A) + 1$.

A further property of the epicyclic torque divider, and the principal attraction of the device, is the large speed-reduction ratio generated by counterrotation of the planet carrier and the annulus. In this way high input-speeds can be accepted by the sun gear while keeping the speed of the planet carrier sufficiently low to avoid bearing life being absorbed by centrifugal load from the planet pinions.

Helical Gear Torque Divider

The ability of double-helical gears to divide load between a pair of gears by balancing the axial thrusts is well known. Figure 4(b) illustrates this technique as a means of obtaining two output drive paths transmitting equal torques. But free axial motion of the double helical gear is difficult to realize for, if driven by a spline, friction in the spline prevents axial float for all but large differences in end thrust on the helical gears. Thus, the input gear of figure 4(b) must be driven by a diaphragm coupling with low stiffness in the axial direction. But a preferable solution, if circumstances allow, is to employ the additional spur-gear train of figure 4(c) as a means of obtaining axial float and equality of torque division between the helical gears.

Dual Idler Pinions

An effective method of torque dividing to increase the torque capacity of a final-stage combining gear consists of introducing dual idler-pinions as shown in figure 4(d). This approach doubles the torque capacity at the critical output stage without increasing the diameter of the combining gear. Floating the primary pinion between a pair of idler pinions has obvious advantages when the three centers lie on a straight line, for then perfect load sharing between the idler pinions is

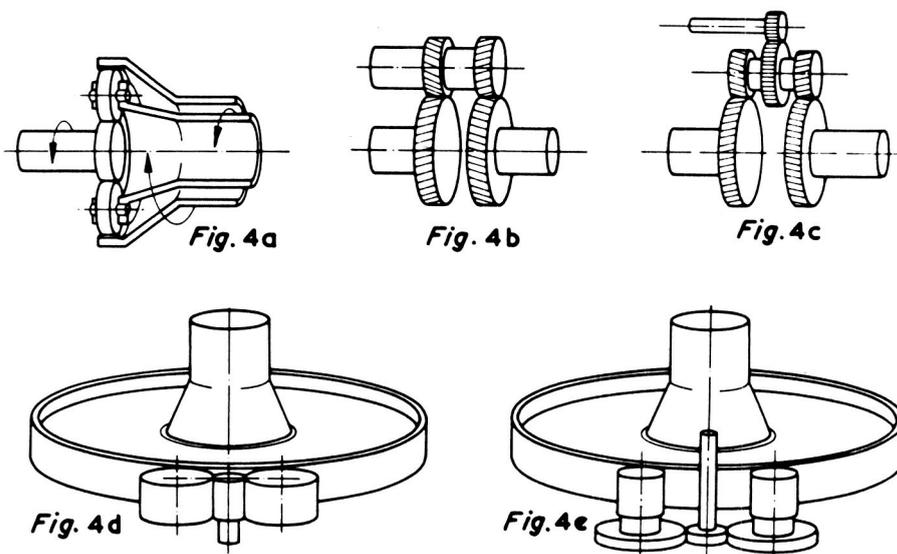Fig. 4a   Fig. 4b   Fig. 4c

Fig. 4d   Fig. 4e

Figure 4. - Rotary torque dividers.

ensured by allowing the primary pinion to float with the tooth forces. Location of the pinion by a light bearing mounted in a low-stiffness (rubber) support ring gives heavy damping to the primary pinion with negligible resistance to the small motions required for load-sharing between the primary pinion, the idler pinions and the combining gear.

With the three gear centers in line (as in fig. 4(d)) the idler pinions need to be larger than the primary pinion in order that teeth on the primary pinion clear teeth on the combining gear. Now the compressive stress at the tooth contacts decreases with an increase in the diameter ratio of idler pinion-primary pinion; hence, it is preferable from a tooth stress standpoint that the idler pinions be at least twice the diameter of the primary pinion. But idler pinions of such a diameter bring a significant weight addition; hence, the justification of figure 4(e), still with ideal torque division, in which the use of compound pinions allows a speed ratio to be taken at the primary-idler pinions. The weight of the compound pinions is then partly offset by the transmission of reduced torques upstream.

Transmission Arrangements Based on Final-Stage Combining Gear

The different methods of torque splitting available at the output, input, and intermediate reduction stages of a transmission can be permutated to give a large number of drive-train configurations. A number of configurations can be eliminated on the basis of weight and complexity; hence, only the more practicable arrangements are discussed below. For uniformity the arrangements are assumed to be for a single-rotor aircraft having horizontal engines. These arrangements aim to reduce the weight of the main gearbox by combinations of the factors:

(1) Achieving a high reduction ratio at the final stage by driving a combining gear through multiple, torque balanced pinions. In this way the transition to low torque levels is made through a minimum number of simple components.

(2) Reducing component sizes and internal loads by dividing torque between parallel drive paths.

(3) Obtaining increased reduction ratios in a single stage by the use of either a counterrotating epicyclic unit or a floating idler pinion having two mesh points.

(4) Where possible, achieving torque division from the gear trains themselves in preference to adding a separate torque dividing unit.

(5) Maximizing the number of final-stage pinions by the use of up to four pinions per engine in conjunction with multiple engines.

(6) Minimizing the height of the transmission by radial stacking of the reduction stages.

Two Final-Stage Pinions Per Engine

Final-Stage Torque Split

The arrangement of figure 5(a), based on torque division through a floating idler pinion, is appropriate to light helicopters in which the engine reduction planetary provides a bevel pinion speed of about 6600 rpm. The scheme has obvious attractions from a reliability standpoint since only eight gears are required after the engine reduction unit.

Preliminary designs for single-engine LOH transmissions demonstrate the ability to obtain a 6.4:1 reduction ratio at the final stage preceded by about 3:1 speed reduction at the idler-pinion mesh. This ability to obtain a 19:1 speed reduction from six gears of the last two reduction stages highlights the advantage of torque dividing as a means of moving to low torque levels through a minimum number of components.

Figures 5(b) and (c) replace the planetary gear at the engine by a bevel and spur reduction unit, respectively, so reducing to 10 the total number of gears required. Figure 5(c) avoids a high-speed bevel pinion and therefore presents a minimum-risk arrangement.

Input-Stage Torque Split

Torque dividing at the engine reduction stage can be achieved as shown in figure 6; in this case the counterrotating outputs from the epicyclic unit are combined on a single bevel gear. This input section is followed by the idler pinion stages of figure 4(e). Thus, while figure 6 requires only one

additional bevel gear in comparison with figure 5(a), increased engine speed can be accepted on account of the doubled reduction ratio made possible by counter rotation of the epicyclic members. A useful long-term development is clearly that of accepting the fixed-annulus planetary unit from an engine manufacturer and adapting it to form a counterrotating, high-ratio reduction gear.

Alternative Arrangements

Further arrangements for obtaining two final-stage pinions per engine are shown in figure 7. Figure 7(a) employs a static-load torque divider (TD) after the engine reduction gear; in consequence, the transmitted torque is split equally between the dual-bevel pinions. In figure 7(b) the engine
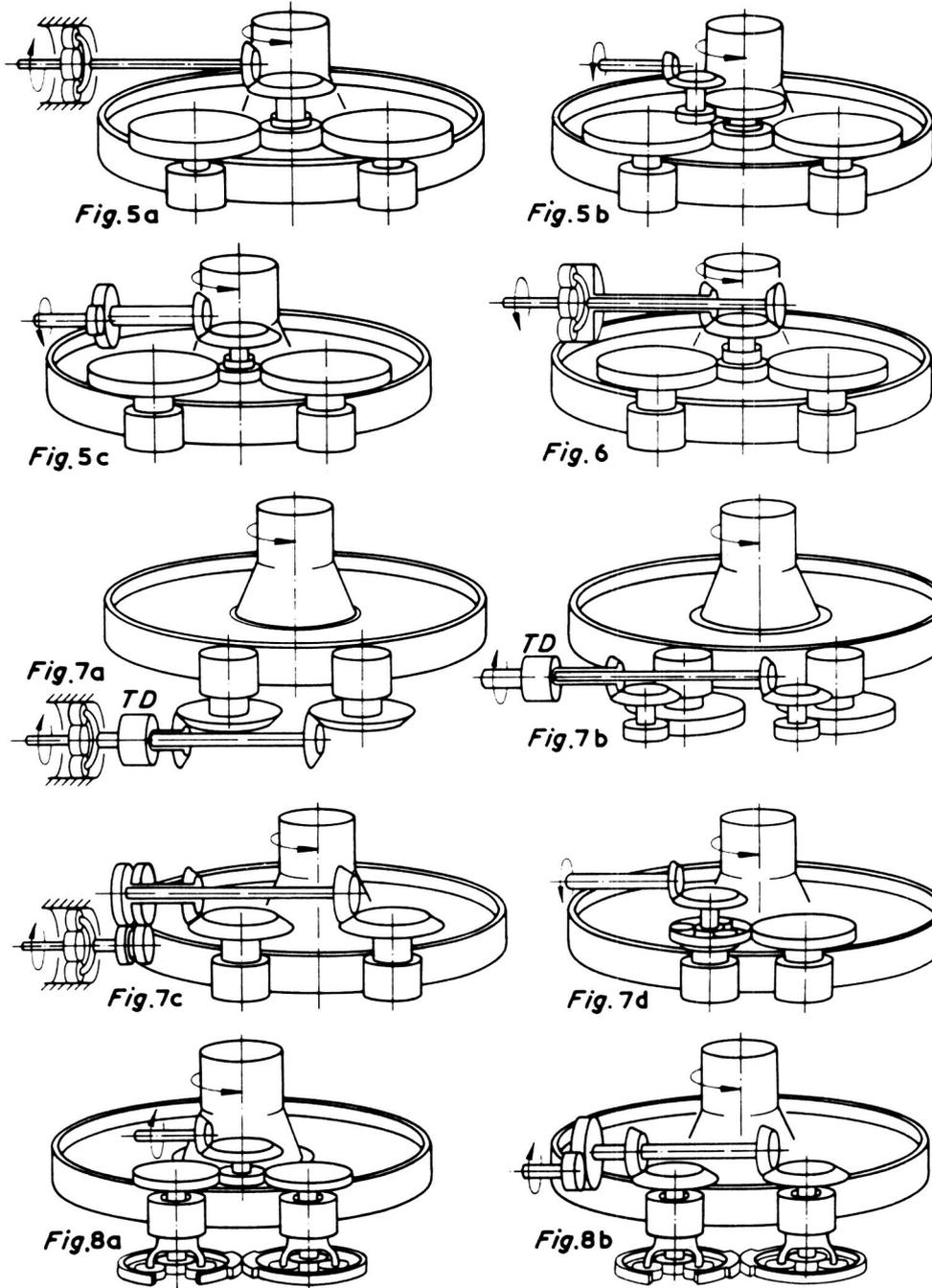
Figures 5 - 8. - Gear-train configurations with two final-stage pinions per engine.

reduction gear of figure 7(a) has been eliminated, so that the dual bevel-pinions are driven at engine speed. The engine of figure 7(c) employs floating helical gears to divide torque. But the counter-rotating epicyclic unit in figure 7(d) provides the basis for a higher overall reduction ratio than figure 7(c), eliminates the torque-balance units, and employs the same number of gears as figure 7(a).

Balanced Annulus Gears

Reacting torque between identical annulus gears provides a further method of torque dividing. Figure 8(a) shows such a gear train in which the annulus gears are not clamped to the transmission housing but are geared together and allowed to rotate a few degrees until the tooth forces, and, hence, the annulus torques, self-balance. This self-balancing action then imposes equal torque division between the spur gears driving the sun gears.

Figure 8(b), developed from figure 8(a), depicts a gear train which offers the ability to split the torque from a high-power engine between a pair of torque-balanced bevel pinions. As in the case of figure 8(a), structural deflection of the gear trains, and backlash, are accommodated by small rotations of the annulus gears until the bevel pinions are equally loaded. Figures 8(a) and (b) show the annulus torques reacted through gear teeth; but an alternative design (fig. 10(e)) involves linking the annulus gears with a tie bar, the connections being such that any motion of the tie bar causes the annulus gears to rotate in opposite directions.

Three Final-Stage Pinions Per Engine

Efforts to increase the torque capacity of the combining gear lead to drive-train arrangements which *torque-balance between more than two final-stage pinions per engine*. For example, figure 9(a) is developed from figure 8 by allowing the annulus gears to react onto a floating double-helical gear. In this way the annulus gears become power flow paths such that, by appropriate choice of gear ratios, each annulus transmits one-sixth of the engine power and each final-stage pinion transmits one-third of the engine power.

The combination of linked epicyclic torque dividers and helical gears of figure 9(a) has sufficient degrees of freedom to provide ideal torque division between the three final-stage pinions and the two input bevel pinions. Reduction ratios across the epicyclic are found to be $(s/c) = 1.5 [(A/S) + 1]$ and $(s/a) = -2A/S$, these ratios corresponding to $\alpha = 2$ in equations (16) and (17) of reference 1, where α is defined as the ratio of power through the planet carrier to power through the annulus.

Figure 9(b) (extracted from ref. 1) also places three torque-balanced pinions per engine on the combining gear. But in this case the single input bevel changes the split ratio and effectively doubles the speed reduction across the epicyclic in comparison with figure 9(a). Thus, with figure 9(b), $\alpha = 1/2$ and $(s/c) = 3[(A/S) + 1]$ (ref. 1).
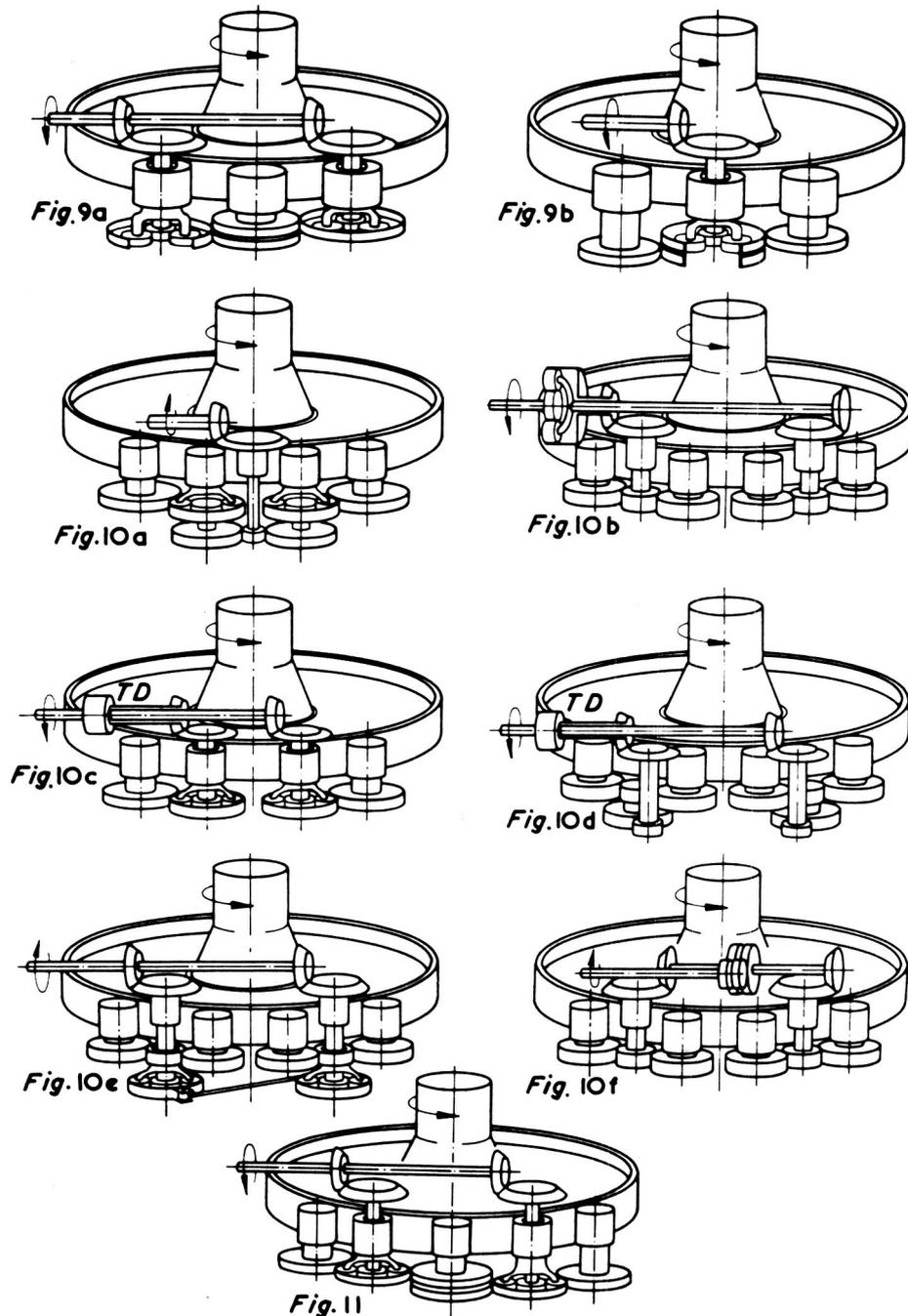
Four Final-Stage Pinions Per Engine

Figure 10 illustrates drive-train arrangements which allow the power from each engine to be divided equally between four final-stage pinions. These arrangements are seen to be extensions of the torque-dividing schemes previously described; the options available include input sections based on torque-balance units, single- and dual-bevel pinions, and epicyclic torque dividers. With four pinions per engine, the torque capability of the combining gear can be raised sufficiently to accommodate the main rotor torques appropriate to HLH aircraft, that is, for torques up to 1.1 MN·m (10^7 in·lb). The potential for transmitting torques of this magnitude is best illustrated by considering a 1.37-m-diameter combining gear driven at 125 rpm by four pinions per engine.

With each engine rated at 3.73 MW (5000 hp) the main rotor receives 282 kN·m of torque per engine. Gear loading to the accepted levels of 1100 MN/m² compressive stress and 360 MN/m² bending stress leads to pinions of only 140-mm diameter and 127-mm face width. In consequence the final reduction ratio is raised to no less than 9.8:1, the equivalent of two planetary stages, and the main rotor can be rated at 11.2 and 14.9 MW with aircraft having three and four engines, respectively.

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More Than Four Final-Stage Pinions Per Engine

Gear trains can be devised which divide the transmitted torque equally between more than four final-stage pinions. Figure 11 shows an example of a five-pinion arrangement in which dual epicyclic torque dividers and a double-helical gear pair ensure that each final drive pinion carries one-fifth of the input power. Concurrently, the torque balance reacts back to the input bevels such that each bevel mesh carries only one-half of the input power. In this case the planet carrier to annulus power split ratio is $\alpha = 2/3$, with the result (ref. 1) that the reduction ratio from the sun gear to planet carrier is raised to $2.5[(A/S)+1]$. But it is evident that the complexity associated with torque balancing five pinions nullifies the advantage of a high reduction ratio.

Figures 9 - 11. - Gear-train configurations with three or more final-stage pinions per engine.

Recent Design Work

Recent efforts in conjunction with NASA Lewis have resulted in the selection of two split-torque drive-train configurations for further detailed analysis and design.

A lower powered single-engine design rated at 500 hp is based on figure 5(b) with the offset spur-gear train omitted. Thus the 103:1 reduction ratio from an engine speed of 35 560 rpm to a main shaft speed of 347.5 rpm is achieved is only three stages from a total of eight gears. Two additional gears are required to form a tail rotor drive. The last reduction stages are of split-torque form; these gear trains provide a speed reduction ratio of almost 25:1, leaving a ratio of 4.1:1 at the engine reduction bevel.

In comparison with uprated OH 58 designs, preliminary results show reductions in total weight, gear and bearing numbers, drive-train losses and noise. These advances are achieved at the expense of an increased diameter on the main housing.

A twin-engine 3000-hp transmission design is based on the OH 60A speeds of 258 and 21000 rpm at the main shaft and engine input respectively. Four pinions driving a final-stage combining gear result in this gear being only 12 percent larger than the ring gear of an equivalent planetary unit. Yet the speed ratio achieved at the final reduction stage is over 7:1, which should be compared with the 4.7:1 available from a five-pinion planetary unit.

This high ratio at the final stage allows the overall ratio of 81:1 to be generated in only three reduction stages of fixed-axis gears. Consequently the drive-train losses and numbers of gears and bearings are as low as with any comparable design, and yet, in comparison with a planetary unit, the combining gear and its four pinions show a major weight benefit.

Different drive-train options are available according to the engine input positions required. Accordingly, the configuration and method of torque division selected changes as a result of engine separation being less than or greater than about 100 cm. Detail design work on the 3000-hp transmission is not yet sufficiently advanced to allow weight predictions of any accuracy, but in comparison with conventional designs, benefits of at least 15 percent appear realistic.

Conclusion

Split-torque drive trains allow the grouping of multiple pinions around a combining gear. Such a combining gear always provides a higher speed-reduction ratio than a final-stage planetary unit and achieves this advantage at reduced weight with fewer gears and bearings.

Torque dividing at intermediate reduction stages also generates higher speed-reduction ratios per stage than do conventional fixed-axis and planetary gear trains. A consequence is reduced torque levels through the transmission and, in favorable circumstances, the ability to eliminate one speed-reduction stage. The principal devices proposed for boosting the reduction ratio per stage are

(1) A counterrotating epicyclic unit
(2) A floating pinion driving through two diametrically opposed mesh points
(3) Multiple pinions driving a combining gear.

Given these mechanism elements, the designer's problem is to devise drive-train configurations that harness the high reduction ratios while ensuring equality of tooth loading at the multiple-gear meshes. Figures 5 to 10 illustrate a number of configurations worthy of investigation. Current technology components are assumed in the configurations described, but conformal gear forms can be accommodated at the final reduction stage.

The above factors, in combination with the potential for weight reduction, drive-train redundancy, and a transmission of low overall height, provide a case for further evaluation of split-torque gear trains as a means of improving the effectiveness of helicopter transmissions.

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