stiffness of the drive shafts to the pinions.

Figure 1 illustrates a flexible-beam mechanism for dividing an input torque between two co-axial 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.

(b) 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 2 as used in Mil helicopter transmissions to divide the torque from each engine between a pair of bevel pinions. The ratio of 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 half input torque.

Radial equivalents to Figure 2 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 favours the selection of rod-end bearings. A future possibility with Figure 2, 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 deformation 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 allowing only 2 mm amplitude are included, 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 a failure in either drive path transfers the total torque to the remaining drive path.

(c) Zero stiffness mechanisms: Unlimited motion

Alternatives to the pivot-beam mechanism of Figure 2 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 a vehicle rear axle, Figure 3a. General terms for the output torques are summarised in Table I from which it is clear these torques are equal when the side bevels B_1 and B_2 are the same diameter.

Figure 3b illustrates the spur gear equivalent of Figure 3a. 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; thus with $S_1 = S_2$ the output torques are each equal to half input torque. A similar result obtains, Table I, if the two sun gears are removed and replaced by two annulus gears as in Figure 3c. The double-annulus design is somewhat shorter than the double sun design on account of the reduced facewidth (conformal contact) required at the annulus/pinion mesh.

Lesser-known torque dividers include the idler-pinion differential shown in Figure 3d. The inclusion of idler pinions allows the sun gear and the planet carrier to deliver equal output torques, each half 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 3e is developed from Figure 3c by omitting all but one set of pinions. It can also be recognised as a rotary

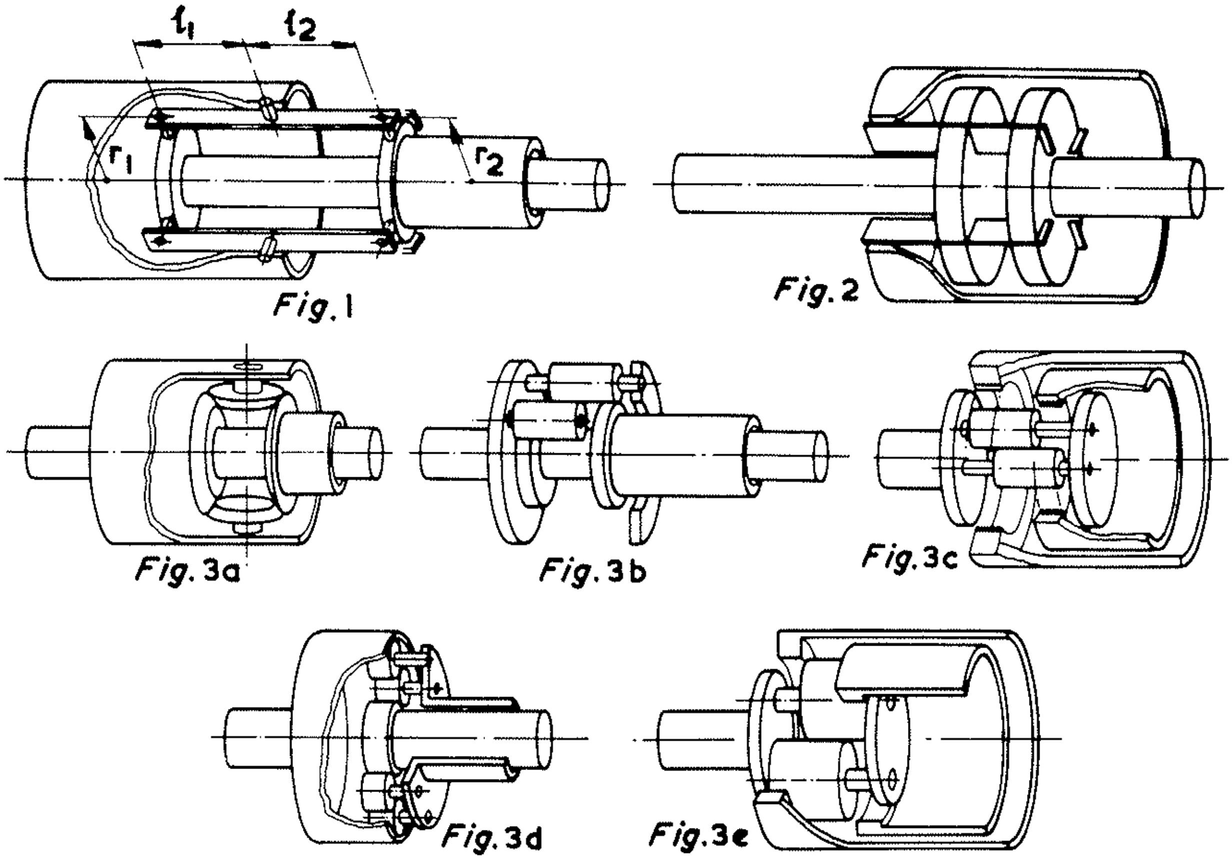


Fig. 1

Fig. 2

Fig. 3a

Fig. 3b

Fig. 3c

Fig. 3d

Fig. 3e

Figs. 1-3 Static-Load Torque Dividers

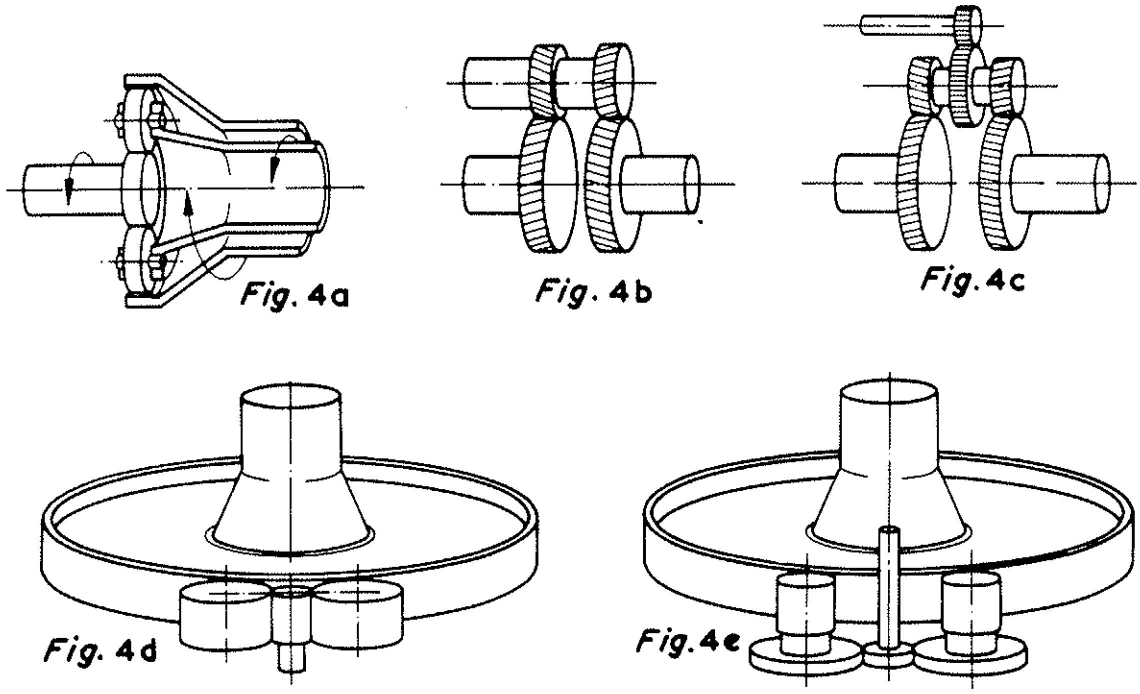


Fig. 4a

Fig. 4b

Fig. 4c

Fig. 4d

Fig. 4e

Fig. 4 Rotary Torque Dividers

equivalent of the pivot-beam arrangement shown in Figure 2. But while Figure 3e brings theeconomy of a minimum total of gears and bearings, the use of a single gear-mesh for reacting the transmitted torque leads to increased size in comparison with the other geared torque splitters.

TABLE I TORQUES DELIVERED BY ZERO-STIFFNESS MECHANISMS

Figure No	Member carrying input torque	Output torques obtained	Condition for equal output torques
2	beam carrier	$T_1 = T_c/\ 1+(\ell_1/\ell_2)(r_2/r_1)$ $T_2 = T_c/\ 1+(\ell_2/\ell_1)(r_1/r_2)$	$(\ell_2/\ell_1) = (r_2/r_1)$
3a	planet carrier	$T\ \ = T_cB_1/(B_2 + B_1)$ $T_{B2} = T_cB_2/(B_2 + B_1)$	$B_1 = B_2$
3b	planet carrier	$T_{S1} = S_1T_c/(S_1 + S_2)$ $T_S2 = S_2T_c/(S_1 + S_2)$	$S_1 = S_2$
3c	planet carrier	$T_{A1} = A_1T_c/(A_1 + A_2)$ $T_{A2} = A_2T_c/(A_1 + A_2)$	$A_1 = A_2$
3d	annulus	$T_c = T_A\ \{1-(S/A)\}$ $T_S = T_A\ (S/A)$	$A = 2S$
3e	planet carrier	$T_{A1} = T_cP_1/(P_2 + P_1)$ $T_{A2} = T_oP_2/(P_2 + P_1)$	$P_1 = P_2$
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

(d) General property of torque dividers

A general property of all the static-load torque dividers outlined (Figures 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 4a 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.

5. COMBINED TORQUE-DIVIDER AND SPEED REDUCTION GEAR

Development of static-load torque-dividing mechanisms leads to the examination of gear trains which perform the dual functions 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 summarised below; these types are incorporated in main rotor transmission arrangements described later.

(a) Epicyclic torque divider

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

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 (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 4a 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 counter-rotation 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.

(b) 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 4b 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 realise 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 4b 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 4c as a means of obtaining axial float and equality of torque division between the helical gears.

(c) 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 4d. 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 centres lie on a straight line, for then perfect load sharing between the idler pinions is 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-centres in line, as in Figure 4d, 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 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 4e, 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.

6. 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:

i) 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,

ii) reducing component sizes and internal loads by dividing torque between parallel drive paths,

iii) obtaining increased reduction ratios in a single stage by the use of either a counter-rotating epicyclic unit or a floating idler-pinion having two mesh points,

iv) where possible, achieving torque division from the gear trains themselves in preference to adding a separate torque dividing unit,

v) maximising the number of final-stage pinions by the use of up to four pinions per engine in conjunction with multiple engines,

vi) minimising the height of the transmission by radial stacking of the reduction stages.

7. TWO FINAL-STAGE PINIONS PER ENGINE

Final-stage torque split

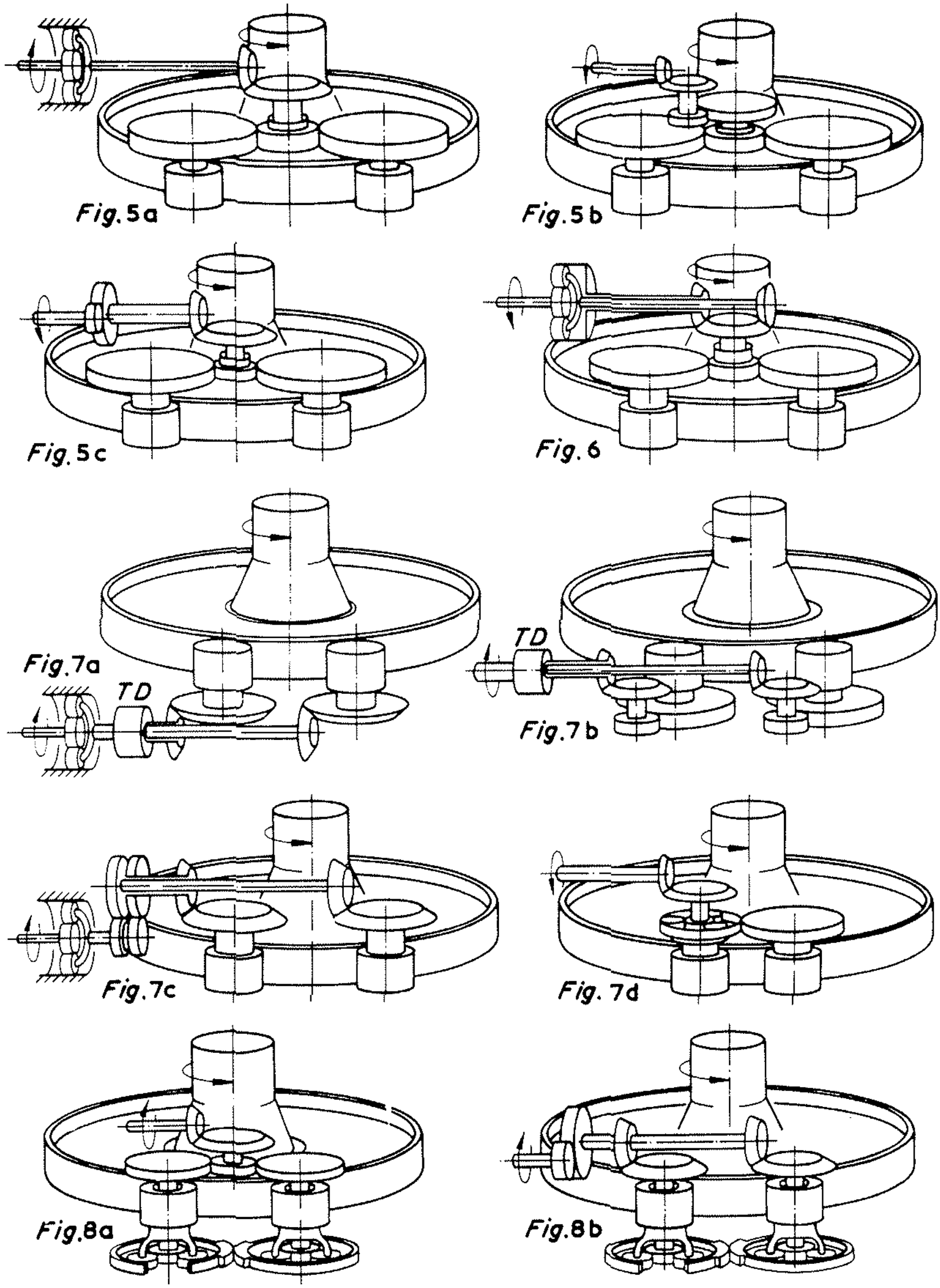
The arrangement of Figure 5a, based on torque division through a floating idler pinion, is appropriate to light helicopters in which the engine reduction planetary provides a bevel pinion speed of about 6600 r/min. The scheme has obvious attractions from a reliability standpoint since only eight gears 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 the 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 5b and 5c replace the planetary gear at the engine by a bevel and spur reduction unit respectively, so reducing to ten tne total number of gears required. Figure 5c 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 counter-rotating outputs from the epicyclic unit are combined on a single bevel gear. This input section is followed by the idler pinion stages of Figure 4e. Thus while Figure 6 requires only one additional bevel gear in comparison with Figure 5a, increased engine speeds 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



Figs. 5-8 Gear Train Configurations With Two Final-Stage Pinions Per Engine

accepting the fixed-annulus planetary unit from an engine manufacturer and adapting it to form a counter-rotating, high-ratio reduction gear.

Alternative arrangements

Further arrangements for obtaining two final-stage pinions per engine are shown in Figure 7. Figure 7a 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. Figure 7b eliminates the engine reduction gear of Figure 7a and, in consequence, drives the dual bevel-pinions at engine speed. Figure 7c employs floating helical gears to divide torque. But the counter-rotating epicyclic in Figure 7d provides the basis for a higher overall reduction ratio than Figures 7a to 7c, eliminates the torque-balance units, and employs the same number of gears as does Figure 7a.

Balanced annulus gears

Reacting torque between identical annulus gears provides a further method of torque dividing. Figure 8a 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 8b, developed from Figure 8a, 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 8a, structural deflections of the gear trains, and backlash, are accommodated by small rotations of the annulus gears until the bevel pinions are equally loaded. Figures 8a and 8b show the annulus torques reacted through gear teeth; but an alternative design, Figure 10e, 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.

8. 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 9a is developed from Figure 8 by allowing the annulus gears to react on to 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 engine power and each final-stage pinion transmits one-third of engine power.

The combination of linked epicyclic torque dividers and helical gears shown in Figure 9a 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 Ref.(1). α is defined as the ratio of power through the planet carrier to power through the annulus.

Figure 9b, extracted from (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 9a. Thus with Figure 9b, from (1), $\alpha = \frac{1}{2}$ and $(s/c) = 3\ \{(A/S) + 1\}$.

9. FOUR FINAL-STAGE PINIONS PER ENGINE

Figures 10a to 10f illustrate 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

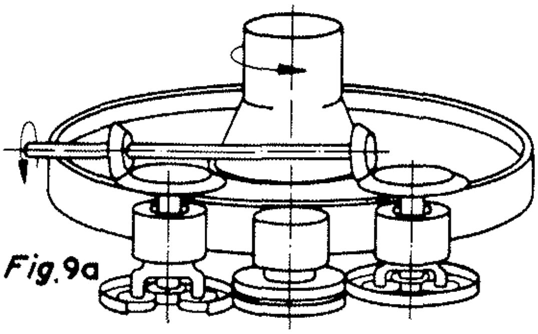


Fig.9a

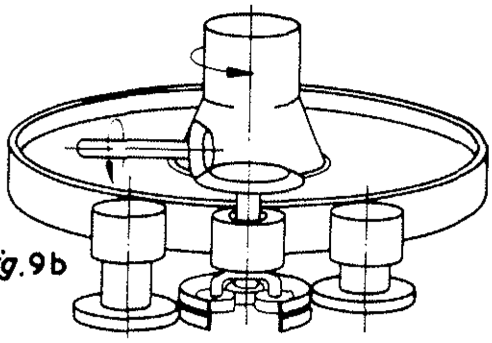


Fig.9b

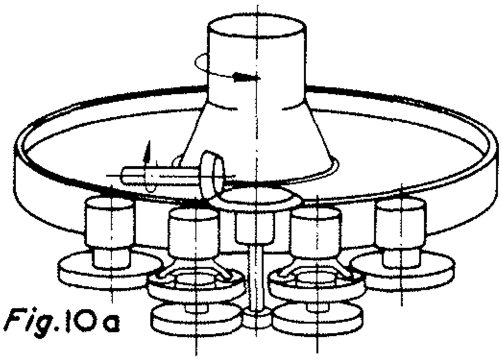


Fig.10a

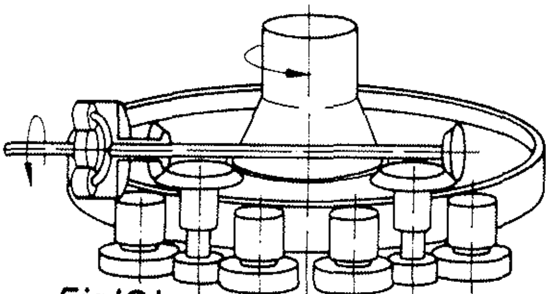


Fig.10b

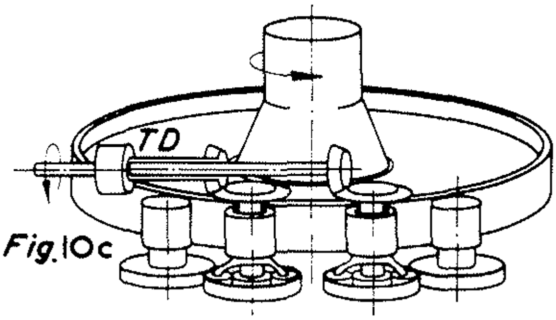


Fig.10c

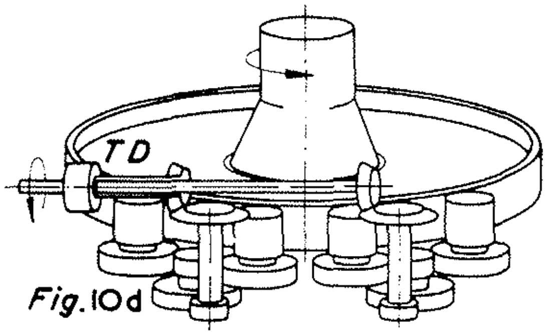


Fig.10d

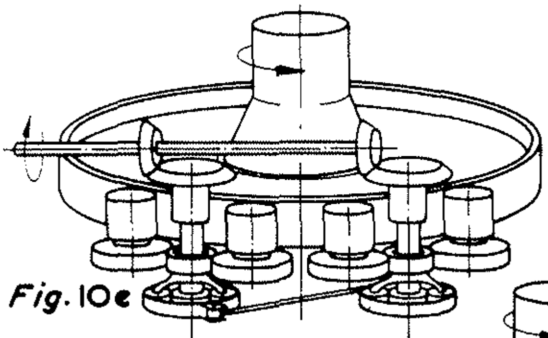


Fig.10e

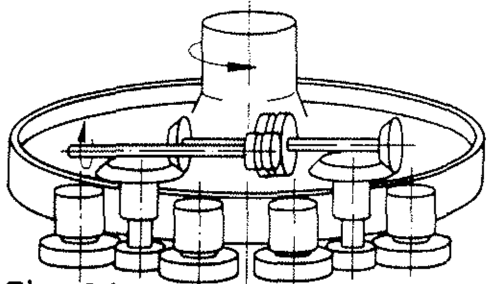


Fig. 10f

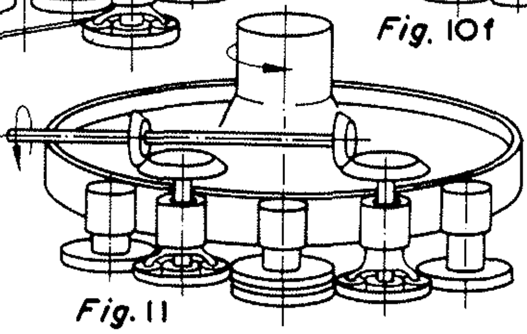


Fig.11

Figs. 9-11 Gear Train Configurations With Three Or More Final-Stage Pinions Per Engine aircraft, that is, for torques up to 1.1 MNm (10^7in lb). The potential for transmitting torques of this magnitude is best illustrated by considering a 1.37 m diameter combining gear driven at 125 r/min by four pinions per engine.

With each engine rated at 3.73 MW (5000 hp) the main rotor receives 282 kNm torque per engine. Gear loading to the accepted levels of 1100 MN/m^2 compressive stress and 360 MN/m^2 bending stress leads to pinions of only 140 mm diameter and 127 mm facewidth. 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 MW and 14.9 MW with aircraft having three and four engines respectively.

10. 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 input power. Concurrently, the torque balance reacts back to the input bevels such that each bevel mesh always carries only one-half input power. In this case the planet carrier/annulus power split ratio is $\alpha = 2/3$ with the result, from (1), that the reduction ratio from 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.

11. CONCLUSIONS

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 favourable circumstances, the ability to eliminate one speed-reduction stage. The principal devices proposed for boosting the reduction ratio per stage are:

(i) a counter-rotating epicyclic unit,
(ii) a floating pinion driving through two diametrically opposed mesh-points,
(iii) multiple pinions driving a combining gear.

Given these mechanism elements the designer's problem is to devise drive train configurations which 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 clearly can be accommodated at the final reduction stage should the reliability of this tooth form be confirmed by further successful development.

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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