Split Torque Gearboxes: Requirements, Performance and Applications

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1. Introduction

Although the simplest gear systems are those with just one gear engagement area between a pair of gears, alternatives are available for applications where it is necessary to transmit a very high torque in a very small space. One option to increase power density is to use the split torque systems that were mainly developed for the aviation industry. These gear systems are based on a very simple idea: division of the transmission of force between several contact areas, thereby increasing the contact ratio. This gives rise, however, to the problem of meshing four gears (Fig. 1).

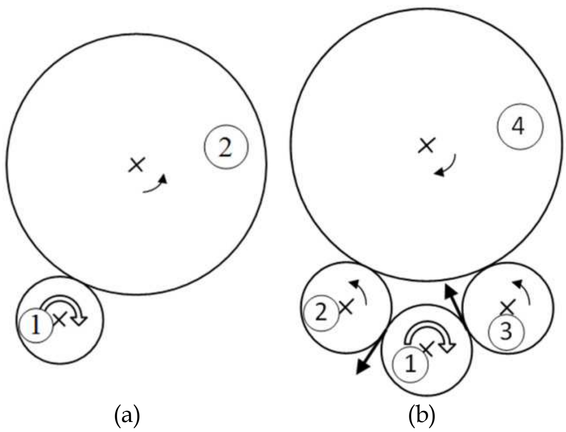


Fig. 1. (a) Standard gearbox assembly; (b) Split torque gearbox assembly

Split torque gearboxes are configurations where a driving pinion (1) meshes with two intermediate idler pinions (2, 3), which simultaneously act on another gear (4). From now on, this assembly will be called four-gear meshing. In this case, the torque split is from gear (1) to gears (2) and (3) which engage gear (4). This gear assembly results in the reduction in gear speed causing an increase in available torque; hence, the split torque transmission means we can use smaller gears.

The greater the number of gears that engage the same crown, the lower the torque exercised by each pinion. Gear assembles can have up to 14 gears engaging a single crown, as happens, for example, in tunnel boring machines.

This chapter explores four-gear meshing in a gear assembly that ensures a 50%torque split for each meshing area. Split torque gears are studied from two perspectives: first, the most common applications of split torque gearboxes in the aeronautical sector and second, the two most restrictive aspects of their application, namely:

- The geometric limitation of the four-gear assembly that requires simultaneous engagement for all four gears. Note that the four gears do not mesh correctly in just any position, although they may seem to do so initially. We will describe the conditions for simultaneous meshing of the four gears in general terms below.
- Torque split between the two gearbox paths must be as balanced as possible to ensure that neither of the paths is overloaded. The technology available to ensure proper torque split between two paths will be discussed below.

2. Applications

Gear transmission requirements for aircraft are very demanding, with a standard gear ratio between engine and rotor of 60:1 (Krantz, 1996). Moreover, the gear transmission system should be safe, reliable, lightweight and vibration-free. One of the most limiting factors is weight and there are three fundamental transmission parameters that greatly affect this factor:

1. The number of transmission stages. The greater the number of stages used to achieve the final gear ratio, the heavier the transmission, given that more common elements such as shafts and bearings are necessary.
2. The number of transmission paths, the basis for split torque gearboxes. Torque is divided between several transmission paths, resulting in a contact force in the smaller gear that means that smaller, and consequently lighter, gears can be used.
3. The final stage transmission ratio. Using a greater transmission ratio in the final stage enables weight to be reduced. This is because torque in previous steps is lower, making it possible to use smaller gears.

In helicopters, planetary gear systems are typically used for the final transmission stage, with planets consisting of between 3 and 18 gears and with planetary gearing transmission ratios between 5:1 and 7:1 (Krantz, 1996; White, 1989).

Using split-path arrangements with fixed shafts in the final transmission stage is a relatively recent development that offers a number of advantages over conventional systems, being several of them based on weight reduction for the overall transmission:

- It allows torque to be transmitted through various paths. This is a major advantage because when torque is split, the contact force between teeth is less and, hence, smaller, lighter gears can be used, therefore reducing the overall weight. Split torque however has the disadvantage that the torque must be shared equally between the paths. The problems associated with split torque are discussed in Section 4.

- It allows transmission path redundancy. Thus, if one transmission path fails during flight, operation can always be assured through another path. In many cases, consequently, gear transmissions are sized so that a single path can handle 100% of engine power.
- It achieves final-stage transmission ratios of around 10:1 to 14:1 (Krantz, 1996; White, 1989). This improvement over the 5:1 to 7:1 ratios for planetary gearboxes (Krantz, 1996; White, 1989) is reflected in a corresponding reduction in the weight of the transmission system.

Several patents for transmission systems that apply split torque have been filed by Sikorky Aircraft Corporation and McDonnell Douglas Helicopters (Gmirya & Kish, 2003; Gmirya, 2005; Craig et al., 1998) that refer either to complete or improved power transmission systems from the rotorcraft or aircraft engine to the rotor or propeller. Other studies that describe various aspects of split torque transmission systems, particularly their use in helicopter gearboxes (White, 1974, 1983, 1989, 1993, 1998), conclude that such gears have a number of advantages over traditional gear systems.

Below we describe two helicopter transmission systems that use multiple path gearboxes. The first is a helicopter gearbox used for laboratory tests of torque divided into two stages, and the second is a commercial helicopter three-stage gearbox that combines bevel, spur and helical gears.

2.1 Helicopter gearbox for laboratory testing

The gear transmission described below was used to perform numerous tests on the operation of split-torque transmissions (Krantz et al., 1992; Krantz, 1994, 1996; Krantz & Delgado, 1996), which can be considered a standard for aeronautical applications. The full assembly is depicted in Fig. 2.

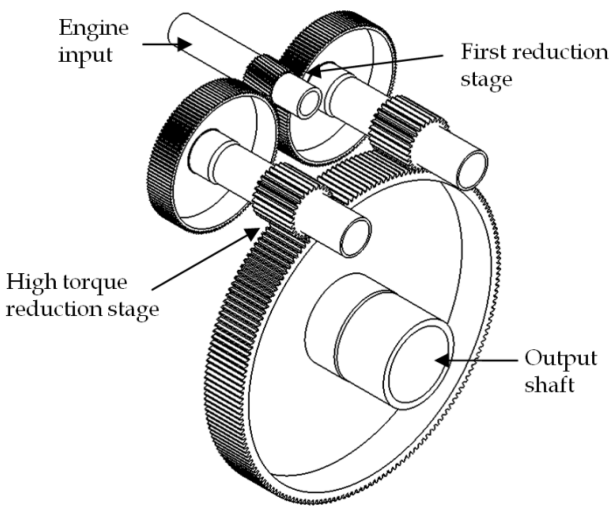


Fig. 2. Helicopter transmission for laboratory testing

The transmission is sized for input of 373 kW at a speed of 8780 rpm. As can be observed in Fig. 2, the transmission has two stages:

- First reduction stage. The first stage is a helical gear with a input pinion with 32 teeth and two output gears with 124 teeth each. The gear ratio is 3.875:1, resulting in an output speed of 2256.806 rpm. This is the stage where torque is split between the input pinion and the two output gears.
- High torque reduction stage. The output shaft is driven by a gear which is driven simultaneously by two spur pinions, each coaxial to the gear in the first reduction stage. The ratio between the gear teeth is 27/176, so the transmission ratio is 6.518:1, resulting in an output shaft speed of 347.6 rpm.

This configuration results in torque of 9017.56 Nm. being transmitted through two paths.

2.2 Commercial helicopter transmission

This gear transmission, studied in depth by White (1998), is sized for two engines, each with a continuous rating of 1200 kW turbine at a nominal speed of 22976 rpm. The main rotor speed is 350 rpm for an overall speed reduction ratio of nearly 66:1. Fig. 3 depicts a plan view of the gear transmission and Fig. 4 is a three-dimensional view showing the gears.

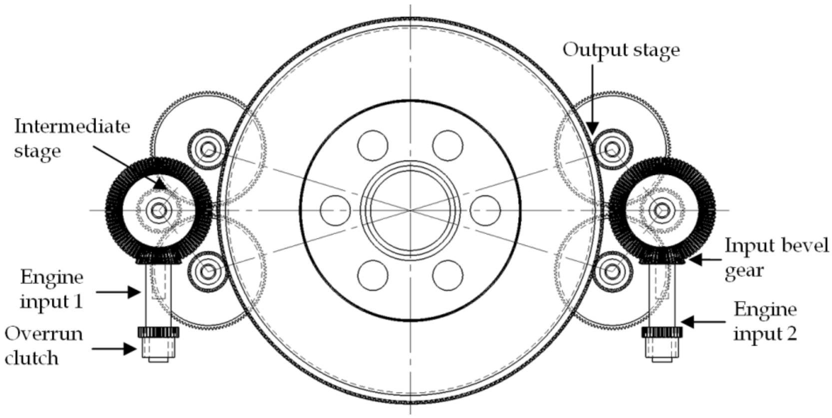


Fig. 3. Arrangement of gear trains between engines.

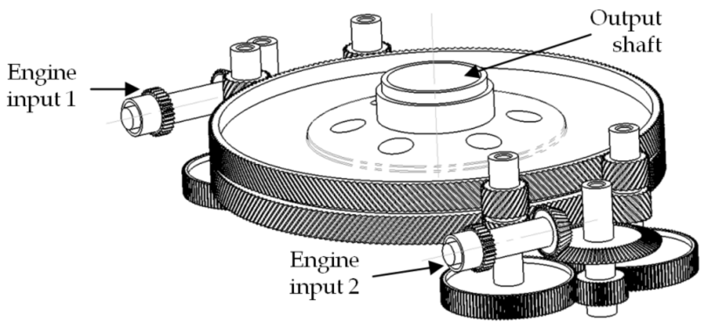


Fig. 4. Three-dimensional view of the gear train arrangement

Total transmission reduction is achieved by three gearing stages, clearly depicted in Fig. 3 and Fig. 4:

- Engine input. Engine torque is accepted by an overrun clutch, mounted with a bevel pinion. This bevel gear, with a between-teeth ratio of 34/84, produces a transmission ratio of 2.470:1. In this stage, the output velocity is 9299 rpm.
- Intermediate stage. Dual offset spur gears are driven by a single pinion. The between-teeth ratio of 41/108 produces a transmission ratio of 2.634:1, resulting in an output shaft speed of 3530 rpm. This meshing results in the first split in torque between the two intermediate gears.
- High-torque output stage. A double-helical gear is driven by a pinion coaxial with each intermediate stage gear. In this stage, the torque is split again between the two helical pinions, with the result that the output shaft simultaneously receives torque from four pinions for each bevel gear. In this transmission it is very convenient to combine torque split with reduction, as greater torque is transmitted in each stage.

The between-teeth gear ratio is 23/232, so the transmission ratio is 10.087:1, resulting in an output shaft speed of 350 rpm.

This configuration uses double-helical gearing at the output stage to drive the output shaft. The helical pinions have opposing angles, which ensures equilibrium between the axial forces. When a double gear operates on the output shaft, the area of support is twice that of a simple gear. This causes a reduction in contact force, which in turn results in a reduction ratio that is twice that of the simple case, with the corresponding reduction in weight and mechanical load.

Overall, this constitutes a transmission ratio of 65.64:1, with the total torque in the output shaft exercised by each engine of 28818Nm, split between the four pinions that engage the output shaft crown. This calculation is based on estimating overall losses, with each input engine operating independently, of 12%.

One of the main problems in split torque transmission is ensuring equal torque split between the paths. To ensure correct torque split, a long, torsionally flexible shaft is used between the intermediate-stage spur gear and the output-stage helical pinions. Section 4 describes the methods most frequently used to ensure correct torque split between paths.

3. Feasible geometric configurations

To ensure simultaneous meshing of four gears (Fig. 1), configuration must comply with certain geometric constraints. A number of studies describe the complexity of simultaneous gearing in split torque gearboxes (Kish, 1993a) and in planetary gear systems (Henriot, 1979, Parker & Lin, 2004); other studies approach the problem generically (Vilán-Vilán et al., 2010), describing possible solutions that ensure the simultaneous meshing of four gears.

For four gears to mesh perfectly, the teeth need to mesh simultaneously at the contact points. The curvilinear quadrilateral and the pitch difference are defined below in order to express the meshing condition. From now on we will use this nomenclature of our own devising -that is, curvilinear quadrilateral - to indicate the zone defined by portions of pitch circles in the meshing area (Fig. 5). The pitch difference is the sum of pitches in the input and output gears minus the sum of pitches in the idler gears at the curvilinear quadrilateral. For perfect engagement between the four gears, the pitch difference must coincide with a whole number of pitches.

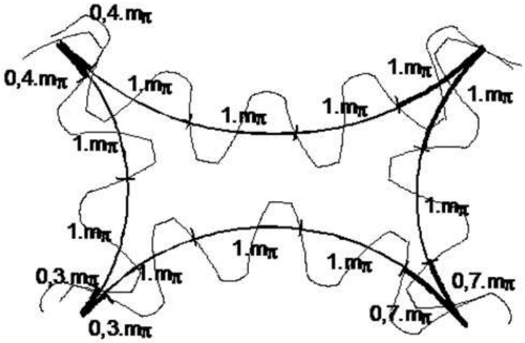


Fig. 5. General conditions for simultaneous meshing of four gears

A relationship is thus established between the position of the gears, as defined by the relative distance between centres, and the number of teeth in each of the gears. Below we explore two possible cases of over-constrained gears:

- CASE 1. Four outside gears.
- CASE 2. Three outside gears and one ring gear.

3.1 Case 1. Four outside gears

For a gearbox with the geometry illustrated in Fig. 6, it is possible to locate the different positions that will produce suitable meshing between gears, in function of the number of teeth in each gear, by defining the value of the angles α, β, δ and γ.

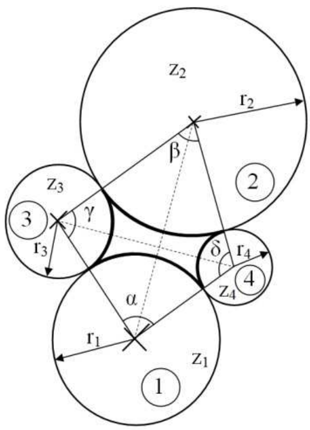


Fig. 6. Nomenclature for the four-gear case

The condition described in the previous section can be mathematically expressed as follows (see Nomenclature):

$$r_1 \cdot \alpha + r_2 \cdot \beta - r_3 \cdot \gamma - r_4 \cdot \delta = n \cdot (m\pi) \quad n \in Z \tag{1}$$

where n is the pitch difference in the curvilinear quadrilateral. As previously mentioned, n must be a whole number to ensure suitable meshing between gears.

We thus obtain an equation with four unknowns (α, β, γ, δ). The three remaining relationships can be obtained from the quadrilateral that joins the centres of the pitch circles (this quadrilateral will be denoted the rectilinear quadrilateral). Finally, we come to a transcendental equation (2) from which α can be obtained according to the number of teeth in the gears.

$$e_1 - f \cdot \cos\left[A_1 \cdot \boldsymbol{\alpha} + B_1 \cdot \arccos\left(\frac{c_1 - a_1 + b_1 \cdot \cos\boldsymbol{\alpha}}{d_1} \right) + C_1 \right] = g_1 - \\ -h_1 \cdot \cos\left[A'_1 \cdot \boldsymbol{\alpha} + B'_1 \cdot \arccos\left(\frac{c_1 - a_1 + b_1 .\cos\boldsymbol{\alpha}}{d_1} \right) + C'_1 \right] \quad (2)$$

Once the angle α has been determined, we can calculate:

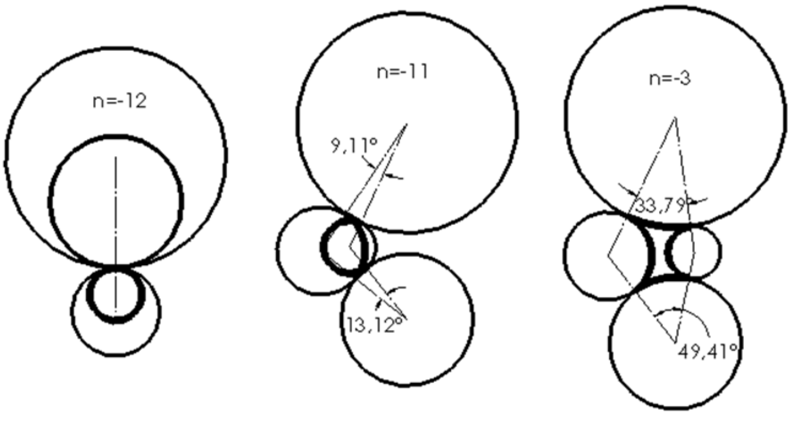
$$\beta = \arccos\left(\frac{c_1 - a_1 + b_1 \cdot \cos\alpha}{d_1} \right) \quad (3)$$

$$\gamma = A_1 \cdot \alpha + B_1 \cdot \beta + C_1 \quad (4)$$

$$\delta = A'_1 \cdot \alpha + B'_1 \cdot \beta + C'_1 \quad (5)$$

a_1, b_1, c_1, d_1, e_1, f_1, g_1, h_1, A_1, B_1, C_1, A_1', B_1' and C_1' are numerical relationships among the teeth number from each wheel that must mesh simultaneously. The value of each coefficient is listed in the Appendix.

The transcendental equation for obtaining α has several solutions, all representing possible assemblies for the starting gears. For example, for four-gear meshing with the next teeth numbers: z_1=30, z_2=50, z_3=20 and z_4=12 (see Nomenclature), all the possible solutions for the gear can be encountered. In this case solutions are n = -12, -11, -3, -2, -1, 0, 1, 2, 3, 4, 7, 29 and 30, where n is the pitch difference between the two sides of the curvilinear quadrilateral (a whole number that ensures suitable meshing). Fig. 7 shows some of the possible meshing solutions.



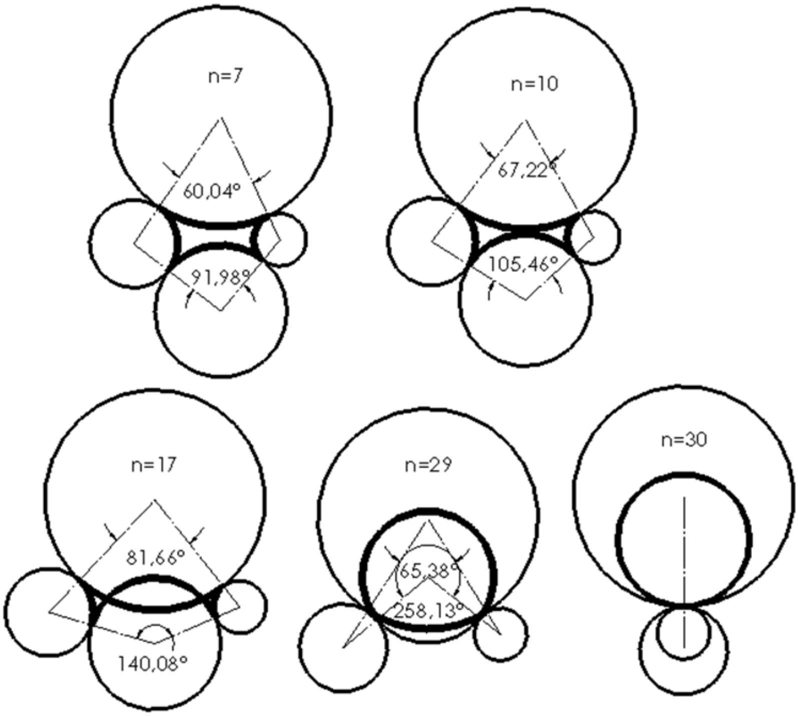


Fig. 7. Feasible solutions for given numbers of teeth

3.2 Case 2. Three outside gears and one ring gear

In this case torque is transmitted from a driving pinion (1) to a ring gear (2) through two idler pinions (3) and (4). Two solutions are available depending on the geometry of the rectilinear quadrilateral that joins the centres of the pitch circles, either crossed (Fig. 8 (a)) or non-crossed (Fig. 8 (b)). The starting equation is different for each of these cases.

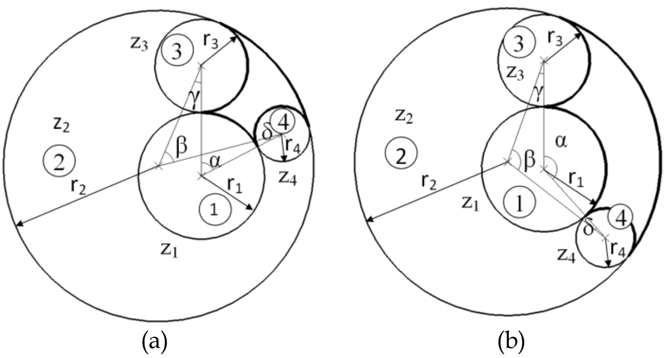


Fig. 8. Solutions for three outside gears and one ring gear: (a) crossed quadrilateral (b) non-crossed quadrilateral

For the crossed quadrilateral configuration, the starting equation is (see Nomenclature):

$$z_1 \cdot \alpha + z_2 \cdot \beta + z_3 \cdot \gamma - z_4 \cdot \delta = \pi \cdot \left(2 \cdot n + z_3 + z_4\right) \tag{6}$$

Finally, we come to the same transcendental equation (2), where the coefficients are a_2, b_2, c_2, d_2, e_2, f_2, g_2, h_2, A_2, B_2, C_2, A_2', B_2' and C_2', whose values are listed in the Appendix.

For the non-crossed quadrilateral configuration, the starting equation is:

$$z_1 \cdot \alpha - z_2 \cdot \beta - z_3 \cdot \gamma - z_4 \cdot \delta = \pi \cdot \left(2 \cdot n - 2 \cdot z_2 + z_3 + z_4\right) \tag{7}$$

Finally, we come to the same transcendental equation (2), where the coefficients become a_2, b_2, c_2, d_2, e_2, f_2, g_2, h_2, A_3, B_3, C_3, A_3', B_3' and C_3', whose values are listed in the Appendix.

3.3 A particular case: Outside meshing with equal intermediate pinions

A common split torque gear assembly is one with two equally sized idler pinions (Fig. 9).

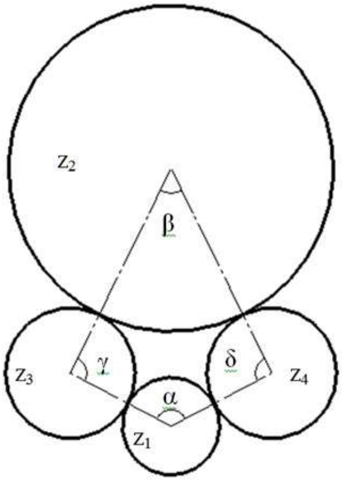


Fig. 9. Idler pinions in an outside gear

The solution is obtained by particularizing the general solution for four outside wheels and imposing the condition $z_3 = z_4$, or $\gamma = \delta$. The following equations are defined for the curvilinear quadrilateral:

$$z_1 \cdot \alpha + z_2 \cdot \beta - 2 \cdot z_3 \cdot \gamma = n \cdot 2\pi \tag{8}$$

$$\alpha + \beta + 2 \cdot \gamma = 2\pi \tag{9}$$

$$\left(z_1 + z_3\right) \cdot \sin\left(\frac{\alpha}{2}\right) = \left(z_2 + z_3\right) \cdot \sin\left(\frac{\beta}{2}\right) \tag{10}$$

Resolving the system, the following transcendental function in α is obtained:

$$\frac{z_1 + z_3}{z_2 + z_3} \cdot \sin\left(\frac{\alpha}{2}\right) = \sin\left[\frac{z_3 + n}{z_2 + z_3} \cdot \pi - \frac{z_1 + z_3}{z_2 + z_3} \cdot \left(\frac{\alpha}{2}\right)\right] \tag{11}$$

The solutions for the other angles can now be obtained:

$$\beta = 2 \cdot \arcsin\left[\frac{z_1 + z_3}{z_2 + z_3} \cdot \sin\left(\frac{\alpha}{2}\right)\right] \tag{12}$$

$$\gamma = \pi - \frac{\alpha}{2} - \frac{\beta}{2} \tag{13}$$

4. Load sharing

The main problem in the design of split torque gearboxes is to ensure that torque is equally split between different paths. Small deviations in machining can result in one of the paths with 100% of torque and the other path operating entirely freely (Kish & Webb, 1992). This situation causes excessive wear in one of the paths and renders the torque split system ineffective.

Below we describe approaches to ensuring equal torque split between different paths in split torque gear arrangements. The main types are:

1. Geared differential. This differential mechanism, frequently used in the automotive sector, delivers equal torques to the drive gears of a vehicle.
2. Pivoted systems. These use a semi-floating pinion constrained both to pivot normal to the line of action and to seek a position where tooth loads are equal.
3. Quill shafts. A torsion divider with a separate gear and pinion, each supported on its own bearings, are connected through the quill shaft, which allows torsional flexibility.

The use of any of these systems to ensure correct torque split makes the gearbox heavier and assembly and maintenance more complex, which is why a number of authors do not support the use of systems that ensure torque split. Described below are the main systems that ensure correct torque split and discussed also are the proposals of authors who advocate for not using special systems.

4.1 Geared differential

One way to ensure correct torque split between two branches is to use a differential system. The great disadvantage of this system, however, is that resistive torque lost in one branch leads to loss of the full engine torque. Different differential mechanisms can be used, with assemblies very similar to those in vehicles or to the system depicted in Fig. 10. Assembled at the entry point to the gearbox is an input planetary system that acts as a differential that ensures load sharing. This transmission accepts power from three input engines, each of which has a differential system that ensures balanced torque splitting. Power is input from each engine to the sun gear of the differential planetary system. The carrier is the output to a bevel pinion that drives one torque splitting branch while the ring gear drives the other torque splitting branch. As the carrier and the ring gear rotate in opposite directions, the bevel pinions are arranged on opposite sides to ensure correct rotation direction. Each output bevel gear drives one pinion which then combines power into the output gear.

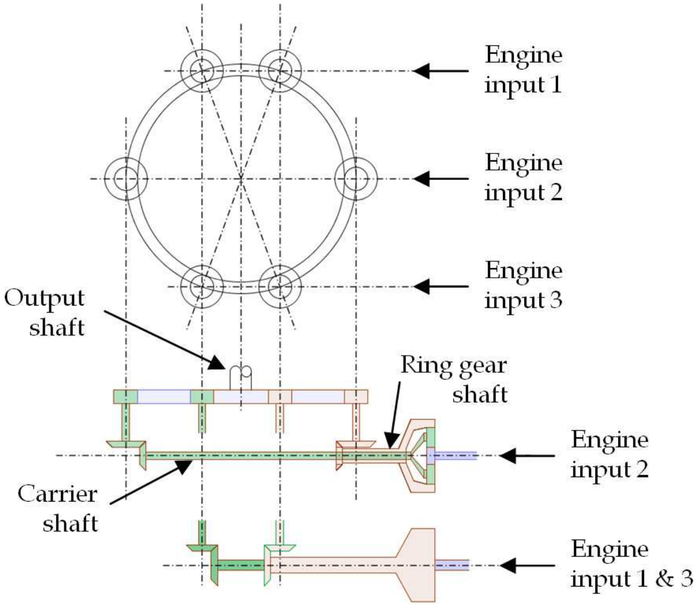


Fig. 10. Schematic view of split torque main transmission

4.2 Pivoted systems

One type of pivoted systems is described in detail in a patent (Gmirya, 2005) for split torque reduction applied to an aerial vehicle propulsion system (Fig. 11). "*The input pinion (64) engages with gears (66) and (68). The input pinion is defined along the gear shaft A_G, the first gear*

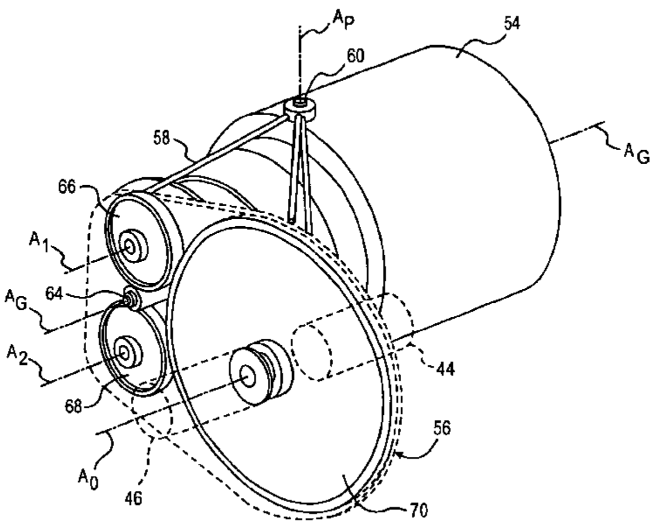


Fig. 11. Perspective view of the split torque gearbox with pivoted engine support (Gmirya, 2005)

(66) defines a first gear rotation shaft A_1 and the second gear (68) defines a second gear rotation shaft A_2. The axes A_G, A_1 and A_2 are preferably located transversally to the pivot axis A_p. The first gear (66) and the second gear (68) engage an output gear (70). The output gear (70) defines an output rotation shaft A_0 and is rotationally connected to the translational driveshaft (44) and the rotor driveshaft (46) to power, respectively, the translational propulsion system and the rotor system".

The assembly transmits torque from the pinion (64), which operates at very high revolutions, to the output shaft (44 -46) via two paths. The pivot system works as follows: since the input pinion (64) meshes with two gears (66) and (68), the pivoted engine arrangement permits the input pinion (64) to float until gear loads between the input gear (64), the first gear (66) and the second gear (68) are balanced. Irrespective of gear teeth errors or gearbox shaft misalignments, the input pinion will float and split torque between the two gears.

4.3 Quill shafts

Below we describe assemblies used in systems that allow some torsion in the split torque shafts (Smirnov, 1990; Cocking, 1986) in order to minimize the difference in torque split between paths. These systems achieve their goal in several ways:

- Conventional systems (Kish, 1993a) assemble intermediate shafts with some torsional flexibility so that angular deviation produced between the input and output pinions adjusts the torque transmitted via the two paths.
- Other systems are based on elastomeric elements in the shaft (Isabelle et al., 1992, Kish & Webb, 1992) or materials with a lower elastic modulus (Southcott, 1999), such as an idler pinion constructed of nylon or a similar material (Southcott, 1999). This solution is not explored here because the torque transmitted is reduced.
- Yet other systems operate on the basis of spring elements (Gmirya & Vinayak, 2004).

The use of such elements in the design adds weight and makes both initial assembly and maintenance more complex, thereby losing to some degree the advantages of split torque gearboxes. Described below are the most representative types of quill shaft.

4.3.1 Conventional quill shafts

Conventional quill shaft design involves assembly on three different shafts (Fig. 12). The

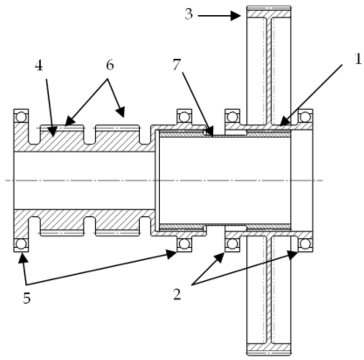


Fig. 12. Conventional assembly of a quill shaft

input shaft (1) is assembled with two separate bearings (2) and the input gear (3).The output shaft (4) is assembled with two separate bearings (5) and, in this case, two output pinions (6). The quill shaft is a third shaft (7) that connects the other two shafts. Due to a lower polar moment of inertia, it admits torsional flexibility, resulting in a small angular deviation between the input and output shafts. The value of the angular deviation is proportional to the transmitted torque; thus, if one path transmits more torque than the other, the angular deviation is greater, allowing the shaft that transmits less torque to increase its load.

4.3.2 Quill shafts based on elastomeric elements

Elastomeric elements are frequently used in quill shafts given their low elastic modulus. For example, one system (Isabelle et al., 1992), based on using elastomers (Fig. 13), consists of *"an annular cylindrical elastomeric bearing (14) and several rectangular elastomeric bearing pads (16). The elastomeric bearing (14) and bearing pads (16) have one or more layers (60); each layer (60) has an elastomer (62) with a metal backing strip (64) secured by conventional means such as vulcanization, bonding or lamination"*.

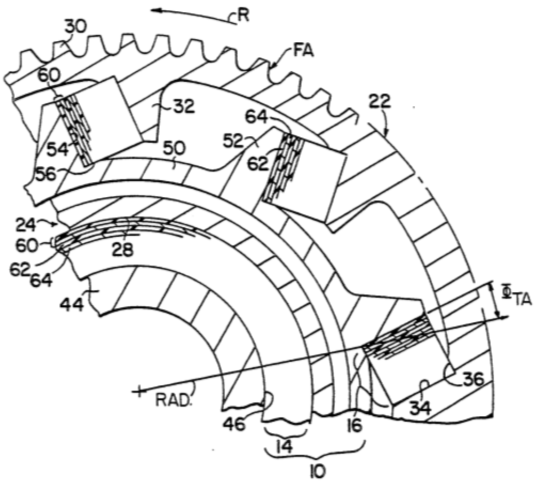


Fig. 13. Elastomeric load sharing device (Isabelle et al., 1991)

The annular cylindrical elastomeric bearing (14) absorbs possible misalignments between shafts resulting from defects in assembly. The rectangular elastomeric bearing pads (16) are responsible for providing torsional flexibility to the shafts of the possible gear paths in order to ensure equal torque transmission.

Another elastomer-based system (Kish & Webb, 1992) (Fig. 14) consists of an assembly with *"a central shaft (21) and a pair of bull pinions (22) and (23). The shaft (21) is supported by the bearings (24) and (25); a gear flange (26) at the end of the shaft has bolt holes (27) and teeth (28) on the outer circumference. A spur gear (29) is held to the flange (26) using upper and lower rims (30) and (31), consisting of flat circular disks (32) with bolt holes (33) and an angled outer wall (34). Gussets (35) between the wall and the disk increase rim stiffness to minimize deflection. One or more elastomer layers (36), bonded to the outer surface (37) of the wall (34), act as an elastomeric torsional isolator"*.

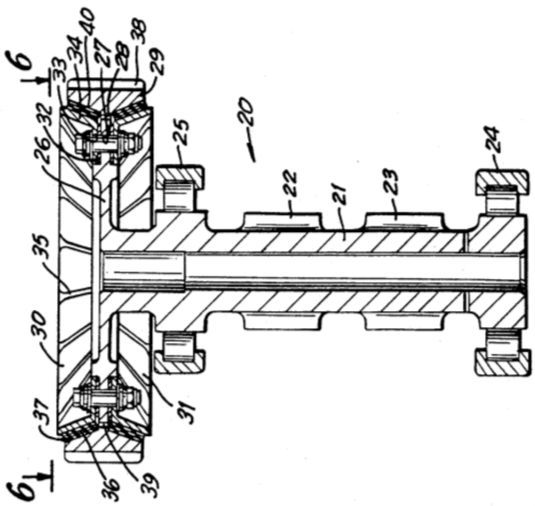


Fig. 14. Gear assembly using an elastomeric torsional isolator (Kish & Webb, 1992)

This assembly was tested in the Advanced Rotorcraft Transmission project (Kish, 1993a), by comparing it with conventional quill shafts. It was concluded that the torque split was excellent and also had other advantages such as lower transmission of force to supports, less vibration and less noise during operation.

The main problem with using elastomers to achieve proper torque split is their degradation over time, especially when used in high-torque gear transmissions where temperatures are high and there is contact with oil. Some authors therefore propose the use of metallic elements to achieve the same effect as the quill shaft.

4.3.3 Quill shafts based on spring elements

Some authors propose the use of metallic elements to achieve the same effect as the quill shaft. One such system (Gmirya & Vinayak, 2004) (Fig. 15) is based on achieving this effect by using *"at least one spring element (30) placed between and structurally connecting the gear shaft (32) and the outer ring of gear teeth (34). The gear shaft (32) has flange elements (36) that project radially outboard of the shaft. The ring of gear teeth (34), similarly, has a flange element (38) that projects radially inward towards the gear shaft"*. In this case, a pair of spring elements (30) is arranged on each side of the gear teeth flange element (38)".

This assembly is designed in such a way that the spring elements absorb torsional deflection between the gears, thereby ensuring proportional torque split between paths.

4.4 No use of special systems

Split torque gearboxes are used in order to reduce the weight of the gear system, so the simplest option is assembly without special systems for regulating torque split. Several authors support this option, for example, Kish (1993a, 1993b), who concluded from tests that acceptable values can be achieved without using any special torque split system, simply by

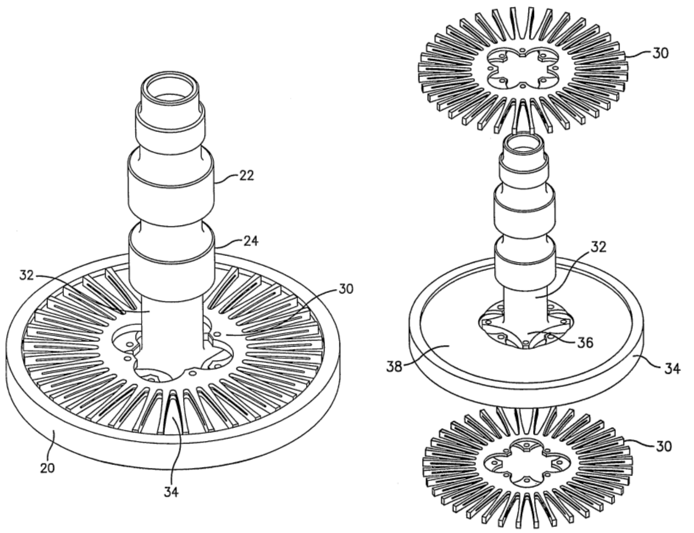


Fig. 15. Load sharing gear in combination with a double helical pinion (Gmirya & Vinayak, 2004)

ensuring manufacturing according to strict tolerances and correct assembly. Krantz (1996) proposed the use of the clocking angle as a design parameter to achieve adequate torque split between paths. This author has studied the effects of gearshaft twisting and bending, and also tooth bending, Hertzian deformations within bearings and the impact of bearing support movement on load sharing.

Krantz (1996) defined the clocking angle as β and described the assembly prepared for measurement (Fig. 16): "*The output gear is fixed from rotating and a nominal counter-clockwise*

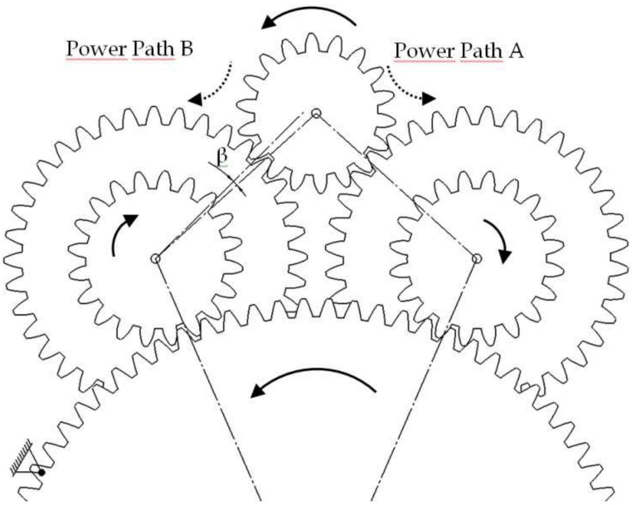


Fig. 16. Assembly for measurement of the clocking angle

torque is applied to the input pinion so that the gear teeth come into contact. When all the gear teeth for both power paths come into contact, then the clocking angle β is, by definition, equal to zero. If the teeth of one power path are not in contact, then the clocking angle β is equal to the angle that the first-stage gear would have to be rotated relative to the second-stage pinion to bring all the teeth into contact".

The tests show that suitable (47 per cent/53 per cent) load sharing can be achieved merely by taking into account the clocking angle and ensuring proper machining and assembly.

This research into the clocking angle has been followed up by subsequent authors (Parker & Lin, 2004) who have studied how contact between different planetary gears is sequenced.

5. Conclusion

Choosing the correct assembly for aircraft power transmission is a key factor in the quest for weight reduction. Although technological advances in mechanical components can help achieve weight reduction in gear systems, their influence is much less than that of choosing the correct gear assembly.

Planetary gear or split torque systems are typically used in helicopter gear transmissions. The fundamental advantage of the split torque systems is that less weight is achieved by equal torque transmission and gear transmission ratios. This advantage is based primarily on arguments as follows:

- In the final transmission stage where the greatest torque is achieved, the use of several paths for the transmission ratio means that, given equal torque and stress levels in the teeth, the ratio between output torque/weight will be better in torque split gear systems than in planetary gear systems.
- In the final transmission stage, transmission ratios of around 5:1 or 7:1 are achieved by planetary gearboxes used with a single stage, compared to 10:1 or 14:1 for split torque gears used in the final stage.
- The possibility of achieving higher transmission ratios in split torque gearboxes makes it possible to use a smaller number of gear stages, resulting in lighter gear systems.
- Split torque gearboxes need fewer gears and bearings that planetary gearboxes, which means lower transmission losses.
- A key factor for aircraft use is that split torque gearboxes improve reliability by using multiple power paths; thus, if one path fails, operation is always assured through another path.
- The main disadvantage of the split torque gearboxes is when torque split between the possible paths is uneven; however, several solutions are available to ensure correct torque split.

These arguments would indicate the advisability of using this type of transmission in aircraft gear systems.

6. Nomenclature

m gear module
r_i radius of the pitch circle of wheel i

z_i number of teeth in wheel i
α angle formed by the lines between centres, between wheels 3 4 1
β angle formed by the lines between centres, between wheels 3 2 4
γ angle formed by the lines between centres, between wheels 2 3 1
δ angle formed by the lines between centres, between wheels 1 4 2
n pitch difference between the two sides of the curvilinear quadrilateral

7. Appendix

The numerical relationships among the teeth number used in the text are listed below. C_1, C_1', C_2, C_2', C_3 and C_3' are functionS of n, a whole number which represents the pitch difference in the curvilinear quadrilateral.

$$a_1 = (z_1 + z_3)^2 + (z_1 + z_4)^2 \tag{14}$$

$$b_1 = 2 \cdot (z_1 + z_3) \cdot (z_1 + z_4) \tag{15}$$

$$c_1 = (z_2 + z_3)^2 + (z_2 + z_4)^2 \tag{16}$$

$$d_1 = 2 \cdot (z_2 + z_3) \cdot (z_2 + z_4) \tag{17}$$

$$e_1 = (z_1 + z_3)^2 + (z_2 + z_3)^2 \tag{18}$$

$$f_1 = 2 \cdot (z_1 + z_3) \cdot (z_2 + z_3) \tag{19}$$

$$g_1 = (z_1 + z_4)^2 + (z_2 + z_4)^2 \tag{20}$$

$$h_1 = 2 \cdot (z_1 + z_4) \cdot (z_2 + z_4) \tag{21}$$

$$A_1 = \frac{z_1 + z_4}{z_3 - z_4} \tag{22}$$

$$B_1 = \frac{z_2 + z_4}{z_3 - z_4} \tag{23}$$

$$C_1 = 2\pi \cdot \frac{z_4 + n}{z_4 - z_3} \tag{24}$$

$$A_1' = \frac{z_1 + z_3}{z_4 - z_3} \tag{25}$$

$$B_1' = \frac{z_2 + z_3}{z_4 - z_3} \tag{26}$$

$$C_1' = 2\pi \cdot \frac{z_3 + n}{z_3 - z_4} \tag{27}$$

$$a_2 = \left(z_1 + z_3\right)^2 + \left(z_1 + z_4\right)^2 \tag{28}$$

$$b_2 = 2 \cdot \left(z_1 + z_3\right) \cdot \left(z_1 + z_4\right) \tag{29}$$

$$c_2 = \left(z_2 - z_3\right)^2 + \left(z_2 - z_4\right)^2 \tag{30}$$

$$d_2 = 2 \cdot \left(z_2 - z_3\right) \cdot \left(z_2 - z_4\right) \tag{31}$$

$$e_2 = \left(z_1 + z_3\right)^2 + \left(z_2 - z_3\right)^2 \tag{32}$$

$$f_2 = 2 \cdot \left(z_1 + z_3\right) \cdot \left(z_2 - z_3\right) \tag{33}$$

$$g_2 = \left(z_1 + z_4\right)^2 + \left(z_2 - z_4\right)^2 \tag{34}$$

$$h_2 = 2 \cdot \left(z_1 + z_4\right) \cdot \left(z_2 - z_4\right) \tag{35}$$

$$A_2 = \frac{z_1 + z_4}{z_4 - z_3} \tag{36}$$

$$B_2 = \frac{z_2 - z_4}{z_4 - z_3} \tag{37}$$

$$C_2 = \pi \cdot \frac{2 \cdot n + z_3 + z_4}{z_3 - z_4} \tag{38}$$

$$A_2' = \frac{z_1 + z_3}{z_4 - z_3} \tag{39}$$

$$B_2' = \frac{z_2 - z_3}{z_4 - z_3} \tag{40}$$

$$C_2' = \pi \cdot \frac{2 \cdot n + z_3 + z_4}{z_3 - z_4} \tag{41}$$

$$A_3 = \frac{z_1 + z_4}{z_3 - z_4} \tag{42}$$

$$B_3 = \frac{z_4 - z_2}{z_3 - z_4} \tag{43}$$

$$C_3 = \pi \cdot \frac{2 \cdot n - 2 \cdot z_2 + z_3 + 3 \cdot z_4}{z_4 - z_3} \tag{44}$$

$$A_3' = \frac{z_1 + z_3}{z_4 - z_3} \tag{45}$$

$$B_3' = \frac{z_3 - z_2}{z_4 - z_3} \tag{46}$$

$$C_3' = \pi \cdot \frac{2 \cdot n - 2 \cdot z_2 + 3 \cdot z_3 + z_4}{z_3 - z_4} \tag{47}$$

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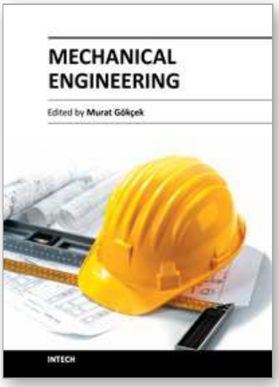

Mechanical Engineering
Edited by Dr. Murat Gokcek

ISBN 978-953-51-0505-3
Hard cover, 670 pages
Publisher InTech
Published online 11, April, 2012
Published in print edition April, 2012

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