

Face Gear Drives: Design, Analysis, and Testing for Helicopter Transmission Applications

by: F. L. Litvin and J.-C. Wang, University of Illinois at Chicago
R. B. Bossler, Jr., Lucas Western, Inc.
Y.-J. D. Chen and G. Heath, McDonnell Douglas Helicopter Co.
D. C. Lewicki, NASA Lewis Research Center

American Gear Manufacturers Association



TECHNICAL PAPER

AVA Documents
Document Sharing Hub

Face Gear Drives: Design, Analysis, and Testing for Helicopter Transmission Applications

F. L. Litvin and J.-C. Wang, University of Illinois at Chicago
R. B. Bossler, Jr., Lucas Western, Inc.
Y.-J.D.Chen and G. Heath, McDonnell Douglas Helicopter Co.
D. C. Lewicki, NASA Lewis Research Center

[The statements and opinions contained herein are those of the author and should not be construed as an official action or opinion of the American Gear Manufacturers Association.]

ABSTRACT:

The use of face-gears in helicopter transmissions was explored. A light-weight split-torque transmission design utilizing face-gears is described. Face-gear design and geometry were investigated. Topics included tooth generation, limiting inner and outer radii, tooth contact analysis, contact ratio, gear eccentricity, grinding and structural stiffness. Design charts were developed to determine minimum and maximum face-gear inner and outer radii. An analytical study showed that the face-gear drive is relatively insensitive to gear misalignment with respect to transmission errors, but the tooth contact is affected by misalignment. A method of localizing the bearing contact to permit operation with misalignment was explored. Two new methods for grinding of the face-gear tooth surfaces were also investigated. The proper choice of shaft stiffness enabled good load sharing in the split torque transmission design. Face-gear experimental studies were also conducted. These tests demonstrated the feasibility of face-gears in high-speed, high-load applications such as helicopter transmissions.

Copyright © 1992

American Gear Manufacturers Association
1500 King Street, Suite 201
Alexandria, Virginia, 22314

October, 1992

ISBN: 1-55589-582-4

FACE-GEAR DRIVES: DESIGN, ANALYSIS, AND TESTING FOR HELICOPTER TRANSMISSION APPLICATIONS

F. L. Litvin and J.-C. Wang (University of Illinois at Chicago)

R. B. Bossler, Jr. (Lucas Western, Inc.)

Y.-J. D. Chen and G. Heath (McDonnell Douglas Helicopter Co.)

D. G. Lewicki (NASA Lewis Research Center)

Introduction

The Advanced Rotorcraft Transmission (ART) program is an Army funded, joint Army / NASA program to develop and demonstrate lightweight, quiet, durable drivetrain systems for next generation rotorcraft [1]. One contract team participant, McDonnell Douglas Helicopter Co. (MDHC) / Lucas Western Inc., developed a novel split torque ART configuration using face-gears [2, 3]. The geometry and design of face-gears and computerized simulation of their meshing have been developed by another member of the team, the University of Illinois at Chicago.

Manufacturing of face-gears was proposed many years ago by the Fellows Corporation. Face-gears have had widespread use in low power applications (fig. 1) but have not had much development for design and manufacturing practices necessary for high power use.

The theory of face-gear drives has not been developed sufficiently for the needs of the designers and manufacturers. Publications in this area in English by E. Buckingham [4], D.W. Dudley [5] can be considered only as a brief description of face-gear drives. J. Davidov [6], F.L. Litvin and L.J. Liburkin [7] have published the results of their investigation of face-gear drives in Russian literature, but these works are not familiar in the western world.

The advantages of face-gear drives are: (i) reduced sensitivity of the bearing contact to gear misalignment, (ii) reduced level of noise due to the very low level of transmission errors, (iii) more favorable conditions of transfer of load from one pair of teeth to the next pair of teeth, (iv) accurate axial location of the pinion is not required in contrast to such re-

quirement for the spiral bevel pinions (fig. 2). Statement (iii) is based on the advantage of involute gearing to have a common normal for those teeth that are finishing and starting the meshing. The analysis shows that the face-gear drives maintain the conjugate action because the face-gear teeth are generated as conjugated to the pinion teeth. The amount of misalignment that can be tolerated is easily controlled by the manufacturing process. The comparison made was for misalignments greater than spiral bevel gears can tolerate.

The majority of the work in this paper has been presented in reference [8], with the exception of face-gear grinding investigations. This paper shows that with proper design face-gear drives can find a successful application in high power applications. The results of computerized simulation of meshing and bearing contact and experimental test of face-gear drives confirm that such drives can be successfully applied.

The paper covers application of face-gear drives in helicopter transmission. The advantage of this design is the possibility to split the torque between two face-gear drives. This results in a significant savings in transmission weight. The design of face-gear drives, simulation of meshing and bearing contact, and grinding of the face-gear tooth surfaces, have been analytically described. Computer programs and design charts have been developed. The torque split has been confirmed by finite element structural analysis. Prototype face-gear drives have been successfully tested at NASA Lewis Research Center.

All types of gears, including face-gears, have niches where their advantages are greater than competing types. This program is an attempt to explore an application where face-gears appear to offer an advantage.

Split Torque Design

The idea of torque split is illustrated in fig. 2. Fig. 2a shows an alternative version of the torque splitting by two spiral bevel pinions, *a* and *b*, designed as one rigid body. Fig. 2b shows the second version of the split of torque when a single spur (or helical) pinion is in mesh with two face-gears. The advantage of the face-gear version is that the same transmitted power results in a reduced load on the bearings in comparison with the spiral bevel version shown in fig. 2a. A second advantage is the pinion is a conventional spur (or helical) gear compared to a complex spiral bevel design with two pinions.

The general configuration of the MDHC/Lucas ART design is illustrated conceptually in fig. 3. There are two engines rated at 2500 HP each which combine to drive the rotor shaft with 5000 HP. The transmission is designed to carry 3000 HP per side for a one engine inoperative condition. Power flows from the engine through an overrunning positive-engagement clutch to a spur pinion, which is lightly restrained radially. The spur pinion drives a downward-facing face-gear and an upward-facing face-gear. The face gear shafts terminate in spur pinions, which drive a large combining gear. The hub of the combining gear is attached to the sun gear of a high contact ratio planetary gear set where the carrier is the output member and is attached to the rotor shaft. A small pinion is driven at the aft side of the main combining gear. This pinion drives another face-gear mounted on the NOTAR™ (notail rotor) driveshaft, which leads aft directly to a NOTAR™ fan.

The concept of torque split appears to be a significant development wherein an input spur gear pinion drives two face-gears arranged to provide an accurate division of power. This division greatly reduces the size and weight of the corner-turning hardware as well as the size and weight of the next reduction stage. The predicted payoff is greatly reduced weight and cost compared to conventional design.

The pinion which serves the two face-gears is a conventional spur gear with an even number of teeth. If the spur gear were rigidly located between the two face-gears, precise torque splitting would be very unlikely. The spur gear has a free-floating mount which allows self-centering between the two face-gears. The effect of normal manufacturing inaccuracy on torque split will be compensated by automatic relocation of the input pinion to its balanced position as long as compliant support is provided for the front end of the pinion shaft. It will be shown analytically (see next section) that precise torque splitting (with $\pm 1.0\%$) will take place.

More importantly, torque splitting between two driven gears by a free-floating spur-gear pinion has been used for many years in truck transmissions. The first known truck application was the experimental Road Ranger transmission produced by the Fuller Transmission Division of the Eaton Manufacturing Company in 1961. Truck transmissions using this principle have been in production since 1963. In addition to accurate torque splitting, it was found that gear noise was reduced and gear life was increased. Thus the use of a free floating pinion as a torque-splitting device is well substantiated.

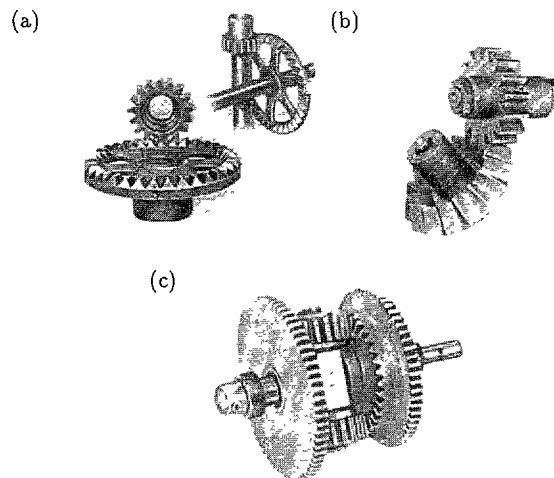


Fig. 1. Examples of the application of face-gear drives.

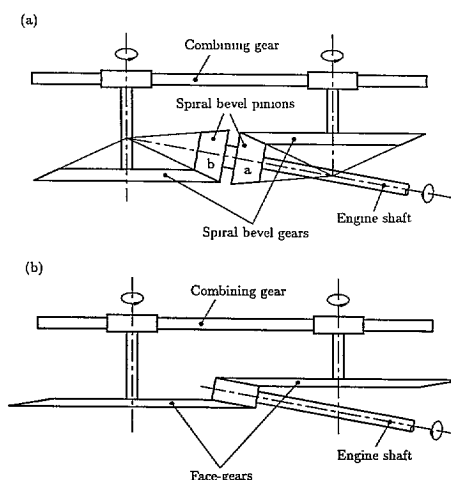


Fig. 2. Different versions of torque split.

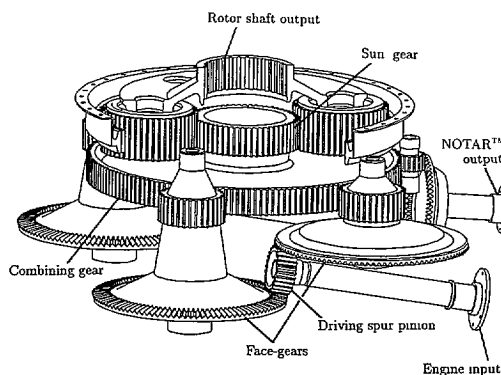


Fig. 3. Three-stage split-torque single planetary transmission.

The gear chain formed by the input pinion, two face-gears, two spur pinions and the combining gear is a *closed loop locked train*. Errors of gear tooth orientation may cause a nonsatisfactory system backlash or even the impossibility to assemble the closed loop train. However, since the face-gears and the combining gear are provided with prime number teeth, indexing by assembly will enable to change the tooth angular orientation inside of the chain and provide the satisfactory system backlash. This is similar to the practice of assembly of locked trains used in marine drives. In reality, there is a need in one test assembly only. Then, the required indexing can be accomplished in accordance with the developed chart.

Finite Element Structural Analysis for the Split Torque Gear Drive

The success of a split-torque gear train design depends on the equal division of the torque to the two output shafts. Conceptually, the floating pinion design makes the system of the pinion shaft a two-force member. The transmitted forces on the two diametrically opposite meshing points on the pinion have to balance each other to achieve equal torque splitting.

The analytical effort to validate the split torque concept was conducted through use of the finite element method. To analyze the deflection and the percentage of torque splitting, the elasticity of the gear structure and pinion shaft support have to be considered first. The finite element model provides an accurate approach to include the stiffness and the deflection of the gear structure. The overall model of the split-torque gear train is shown in fig. 4. This model has been used to analyze the torque splitting percentage for dif-

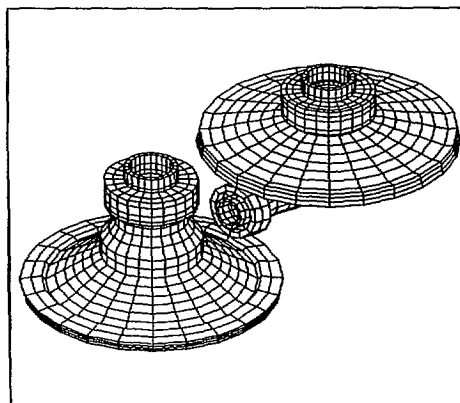


Fig. 4. Finite element model of the split-torque drive.

ferent support conditions as shown in Table I. The stiffness of the front and rear support of the pinion shaft were varied to determine their effect on torque split (cases 1 through 4, table I). The contacts between the pinion and the two face-gears were modeled using gap elements. The torque split was determined using gap element reaction forces as calculated using finite element analysis. Among the cases studied, the most even torque split was provided in case 9 when the stiffness of the shaft's front support was 6×10^4 lb/in. This is an order of magnitude less than a typical bearing support.

The use of backlash control to compensate for the difference in the tangential stiffness between the two output shafts was investigated. In reality, the exact compliance between the teeth of the three gears is not practical. The deviation

Table I. Advanced rotorcraft transmission torque splitting percentages

Case No.	Support of Pinion Shaft		Gear Meshing Clearance (inch) Due to Backlash Adjustment		Split Torque Percentage (%)	
	front-end spring rate (lb/in)	Rear-end spring rate (lb/in)	face-down gear	face-up gear	face-down gear	face-up gear
1	$K = 0$ (free float)	$K = \infty$ (restrained)	0.0	0.0	51.58	48.42
2	$K = \infty$ (restrained)	$K = \infty$ (restrained)	0.0	0.0	56.87	43.13
3	$K = 6.0 \times 10^5$ #	$K = 6.0 \times 10^5$ #	0.0	0.0	53.11	46.89
4	$K = 6.0 \times 10^4$	$K = 6.0 \times 10^5$	0.0	0.0	51.41	48.59
5	$K = 6.0 \times 10^4$	$K = 6.0 \times 10^5$	0.0	0.0005	51.55	48.45
6	$K = 6.0 \times 10^4$	$K = 6.0 \times 10^5$	0.0	0.005	52.86	47.14
7	$K = \infty$	$K = 6.0 \times 10^5$	0.0	0.005	81.66	18.34
8	$K = 6.0 \times 10^4$	$K = 6.0 \times 10^5$	0.003	0.0	50.54	49.46
9	$K = 6.0 \times 10^4$	$K = 6.0 \times 10^5$	0.005	0.0	49.97	50.03

#: 6.0×10^5 lb/in is the translational spring rate for typical bearing and housing support in helicopter transmission.