Design and Test of a Propfan Gear System

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A propfan gear-system technology development program was undertaken to meet the needs of future geared propfan propulsion systems. The design and initial testing of a prototype, 13,000 hp, counterrotating gear system are described. Emphasis was placed on utilization of modern design techniques and advanced gear and bearing technologies during the design phase of this program. A highly instrumented gear-system test was conducted to verify design calculations. A new gear-system test facility is described that allows simulation of flight-operating conditions.

I. Introduction

THE turbofan engine has been widely used over the past several decades for high-speed flight. Recent advancements in propeller design, however, have led to industry's development of an alternative to turbofan propulsion. The propfan offers a substantially more efficient system, up to 30% fuel savings compared with the most advanced turbofan, without sacrificing high-speed flight. Major engine and aircraft manufacturers expound that propfan systems offer a cost-savings potential in both commercial and military applications.

Gear-system technology has progressed significantly, especially in support of the helicopter industry. When it was realized that a critical element in the propfan program was missing, a development program with the purpose of applying advanced technology to the propfan gear system was initiated. The program was designated AGBT[1], standing for advanced gearbox technology. The design and test of this gear system are described herein.

II. Design Requirements

The goal in this AGBT program was to develop an advanced technology, counterrotating gear system with the following characteristics: high reliability—30,000 h mean time between removal (MTBR); high efficiency—greater than 99%; low weight; improved maintainability.

Future propfan gear-system life must equal or surpass the standards accepted for turbofan engines in commercial service today. The 30,000 hp MTBR life requirement represents a high level of reliability that strongly influenced the gear-system design.

In addition to these design goals, specific design requirements were established based on engine specifications, propfan operating characteristics, and aircraft installation requirements. These values are listed in Table 1. Studies conducted in Ref. 1 concluded that propfans are most cost-effective when used for short- to medium-range flights. Considering thrust requirements for aircraft in this category and propulsive efficiency of the propfan, it was determined that engine power levels of 10,000–15,000 hp were required. Performance studies were conducted on various counterrotating pusher and tractor configurations. The aft-mount, pusher configuration was found to be more favorable in commercial applications due to lower cabin noise levels. These studies led to selection of the speed/power/ratio values shown in Table 1.

The gear system was designed to support a propfan moment of 731,000 in./lb and 25,000 lb thrust. The moment load results from a combination of propfan aerodynamic loads and maneuver *g* loads.

III. Configuration

The configuration selected for the counterrotating gear system was the differential planetary. Of the five other arrangements analyzed (see Table 2), the differential planetary was found to have the fewest number of parts, weigh the least, and have the highest efficiency. The differential action allows a gearset, which would transmit only 7000 hp as a simple planetary, to carry 13,000 hp to the two counterrotating shafts. Higher efficiency is obtained by minimizing the number of gear meshes (two), including a highly efficiency internal contact.

Another aspect in the configuration selection was the method for propfan pitch control. The differential planetary splits the torque between the two output shafts in relation to the numbers of teeth in the gearset, allowing speed to vary. The torque split selected in this design provides 44% of the forward prop and 56% to the aft prop. The differential planetary system requires the pitch-control mechanism to regulate prop speed rather than torque as in several of the alternate configurations listed in Table 2. Independently, the propfan manufacturer decided that it would be easier to control prop speed rather than torque. For many reasons, the differential planetary is ideally suited for this application.

The general arrangement of the AGBT gear system is shown in Fig. 1. Specific areas of the design will be addressed in the following sections.

Planet Bearing and Gear Type

The next major consideration in this design was selection of a gear and planet-bearing type. These two tasks are highly interrelated in a planetary gearset. In this design, gear-system life was the strongest consideration. In a planetary gearset this equates to planet-bearing life, which in turn determines the gear-system size. Thus, it is critical to maximize planet-bearing life in the smallest volume possible.

A study of various planet-bearing configurations showed that a cylindrical roller planet bearing would have the longest life of any type given a fixed envelope size (corresponding to a planet-bearing bore). These were Antifriction Bearing Manufacturers Association (AFBMA) calculations used only to determine capacity. In each case, it was assumed that each

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bearing type could be made to work in a planet-bearing environment. This information by itself did not lead to the selection of a cylindrical roller planet bearing, but later it was found to match the requirements of the gear type selected.

A similar study was undertaken to determine the best gear type for this system. The criteria for evaluation in order of importance were load capacity (life), vibration, development risk, power loss, ease of manufacture, and weight. Both low- and high-contact ratio spur, single-helical, and double-helical gears were considered.

In the 1960's, Allison was involved in several development efforts to increase the capacity and reduce vibration levels in the T56 gear system. Much success was achieved with single- and double-helical gearing (Refs. 2 and 3) in these experimental units. These advantages were derived from increased load sharing due to helical gearing, which leads to smoother meshing action. Recently, high-contact ratio spur gears have been investigated for similar reasons.

Calculations were made for bending and contact stress and flash temperature for single-helical, double-helical, and high-contact ratio spur gears as shown in Table 3. Face width, center distance, and gear ratio were held constant, whereas other parameters were varied to obtain the best configuration for each type. These results did not demonstrate any advantage for high-contact ratio spur gears. Single- and double-helical gears were very similar. The final decision on gear type was made based on planet-bearing complexity. In single-helical gears, an overturning moment is created that must be reacted by the planet bearing. Although a cylindrical bearing (preferred as a result of the study just described) could be used, spacing between rows would be excessive. Double-helical gears, however, will work naturally with cylindrical bearings. These gears require the axial freedom inherent in a cylindrical bearing to achieve load sharing between the sun and planet gears.

From a manufacturing standpoint, the cylindrical roller bearing is the easiest to manufacture as an integral planet gear/bearing. Fabrication of double-helical gears was considered to be more difficult than single-helical gears, but less risky than high-contact ratio spur gears. Since Allison has successfully manufactured double-helical gears in the past, development risk was determined to be low. For many reasons, the double-helical gear and cylindrical roller planet bearing appeared to be the best solution.

Table 1 Gear-system design requirements

Configuration	Single-input, dual, counterrotating output
Propfan orientation	Pusher
Rated power	13,000 hp
Input speed	9500 rpm
Input torque	7187 ft-lb
Reduction ratio	8.333
Life	MTBR = 30,000 h
Maximum propfan load	25,000 lb thrust
	731,000 in.-lb at propfan flange

Table 2 Potential counterrotating gear-system arrangements

Differential planetary
Compound planetary
Split-path planetary
Triple-compound idler
Differential epicyclic
Split-path parallel offset

Double-Helical Gear-Loading Sharing

An important aspect of double-helical gearing is the technique used to achieve load sharing between the two rows on each gear and among the planets. Double-helical planetaries have been attempted before without success in some cases due to unequal load sharing. Most unsuccessful designs have attempted to use splines to allow axial movement at the sun- or ring-gear location. As shown in Table 4, if we assume a friction coefficient of 0.15, spline friction will not allow the required movement. The solution utilized in this design was to locate the ring-gear assembly on an axially soft diaphragm. With the sun gear fixed axially on preloaded ball bearings, the planet free to move on the planet roller bearings, and the ring gear free to track with the planet gear, all contacts will continually readjust to distribute the load equally. A potential problem with the flexible diaphragm, however, is the possibility of modal vibration if an excitation source is present.

Load sharing from planet to planet was addressed just as it has been in the Allison T56 planetary gear system for many years. The ring gear is soft in the radial direction to allow a load balance to occur around the ring. A stiff ring gear would,

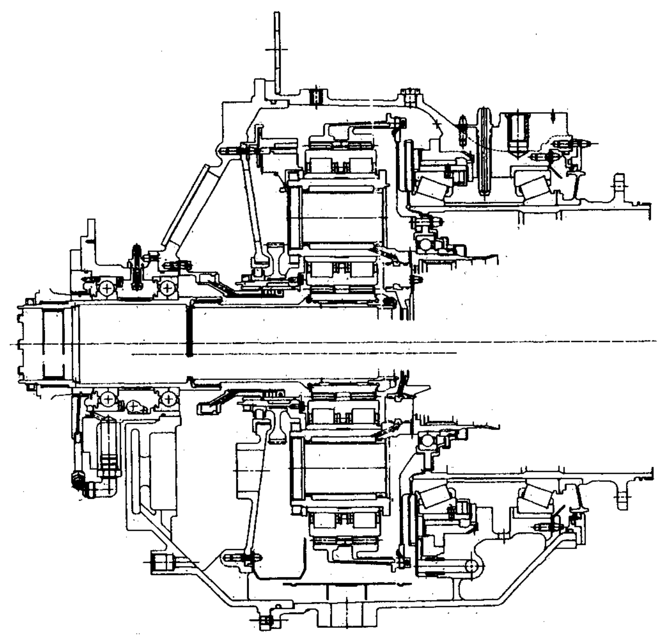

Fig. 1 Gear-system general arrangement.

Table 3 Comparison of gear types

Gear type	Contact stress, lb/in.^{2}	Bending stress lb/in.^{2}	Flash temperature, °F
Single-helical	146,000	32,400	276
Double-helical	143,000	33,700	300
High-contact ratio spur	152,000	37,500	329

Table 4 Effectiveness of load-sharing techniques

Technique	Load sharing between tooth rows, %
Spline at sun gear	68/32
Spline at ring gear	65/35
Ring gear on flexible diaphragm	51/49

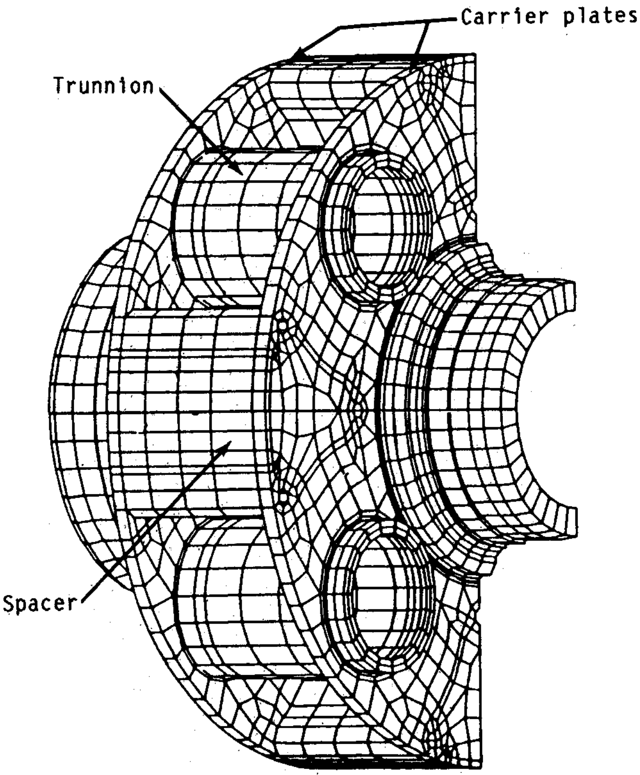

Fig. 2 Planet-carrier FE model.

by comparison, resist the high load resulting in a high individual tooth load. Load sharing in the AGBT system was calculated to be within 13% of perfect balance at any time.

Advanced Gear-System Materials

Another area of technological advance has been that of materials. There are several new high-hot-hardness gear materials to choose from today, such as EX-53, CBS 600, CBS 1000, VASCO X-2, and M-50 NIL. These materials allow operation at much higher temperatures and help prevent damage in the event of loss of oil. CBS 600 was selected for the sun and planet gears in this program. A material known commercially as Latrobe UT-18 was chosen for the ring gears. This material has mainly been used as a high-strength shaft steel. It is very stable dimensionally when nitrided, a requirement in heat-treating large ring gears. The counterrotating, carrier ball-bearing material was M-50 NIL.

Tapered Roller Bearings for Propeller Support

Tapered roller bearings were selected to support the propeller due to their high-thrust capability and relatively short required mounting distance. A more common roller/ball-bearing arrangement requires three bearings and a wide spread to reduce radial forces from moment loads. These large-diameter tapered roller bearings have not previously been utilized in aircraft gear systems, but appear to be well suited to the propfan arrangement. Surface finish requirements were more strict here (8 μ-in. rms) than normal due to the poor lubrication conditions, but techniques were developed to meet this requirement. The use of an aluminum housing created thermal growth concerns between the steel shafting and the aluminum mount. A spring system was developed to accommodate these dimensional changes under temperature variations. The bearings were designed with ribbed cups instead of the more common ribbed cone to provide lubrication in an oil-off situation. Centrifugal force traps oil in the critical rib area so that the system can operate for limited time without an oil supply. Predicted power loss in these bearings is quite acceptable, 6.6 hp at takeoff power conditions (13,000 hp, 25,000 lb thrust).

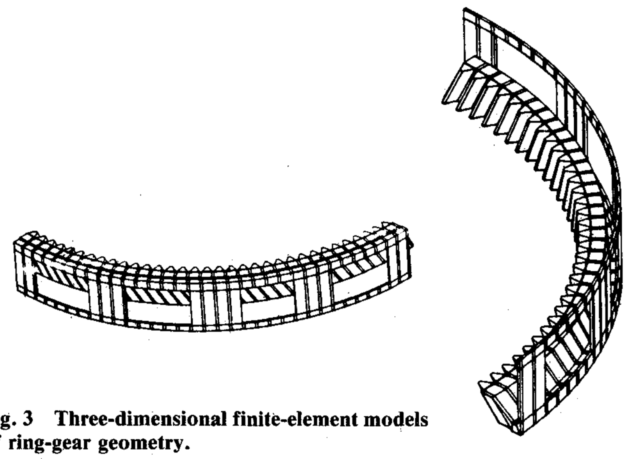

Fig. 3 Three-dimensional finite-element models of ring-gear geometry.

Detailed Bearing Analysis

All gear-system bearings were analyzed with the aid of large-scale computer programs (Refs. 4–6) to determine capacity, optimize geometry, and select lubricant flow rates. The planet bearing was analyzed with several programs since there is presently no comprehensive planet-bearing computer program. The tapered roller bearings were analyzed by the vendor with proprietary in-house programs. System life was consistent with the 30,000 h MTBR based on commercial turboprop reliability analysis.

Planet Carrier

The planet carrier was designed to minimize deflection in the planet bearing. Typically, the carrier trunnion deflects under the bearing load due to the cantilever arrangement of the trunnion on the drive plate. The carrier was stiffened in the circumferential direction by placing spacers between the trunnions (see Fig. 2). To minimize deflection due to radial loads, a plate was attached to the ends of the trunnions. Planet-bearing deflection at maximum load was calculated to be 0 deg, 2 min.

Housing

The housing was designed as test equipment. The structure was analyzed to provide the correct stiffness and strength, but features required for flight, such as cored oil passages and minimum weight, were not part of this design effort.

IV. Finite-Element Analysis

Extensive use was made of finite-element analysis to determine deflections and stresses in gear-system components. All gears were modeled using a gear-mesh generator program originally developed by Drago[7], but modified to accommodate the new geometry. These models were primarily used to determine deflections. Tooth root stresses were evaluated with AGMA procedures. Major structural components—housing, carrier, and prop shaft—were analyzed for both stresses and deflections.

The ring-gear model shown in Fig. 3 was used to optimize radial deflection across the gear tooth to prevent misalignment. The maximum error predicted by this model was 0 deg, 4 min.

The planet gear and integral roller bearing were analyzed to optimize gear deflections both across the cylindrical roller-bearing length and around the circumference to improve the bearing load distribution. Three-dimensional loads were applied across the gear tooth and then reacted by a series of radial springs to simulate the rollers (see Fig. 4). The resulting load distribution is shown in Fig. 5. Vibration analyses of the planet gear showed that the center rib was required to eliminate the possibility of resonant vibration during operation.

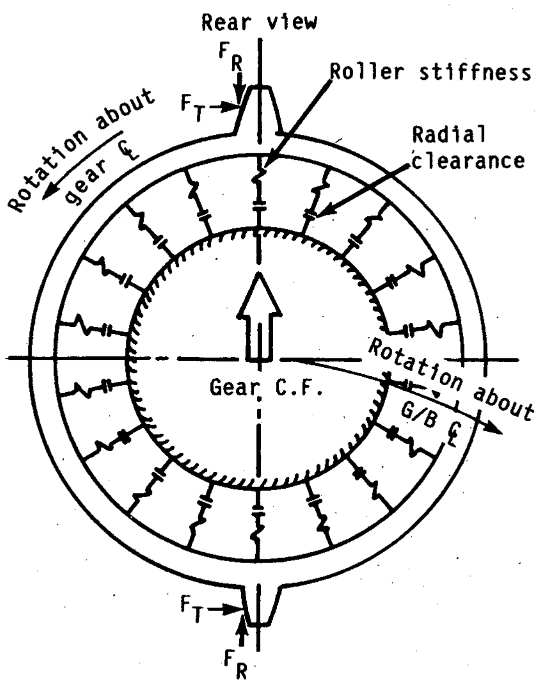

Fig. 4 Model used to determine bearing load distribution.

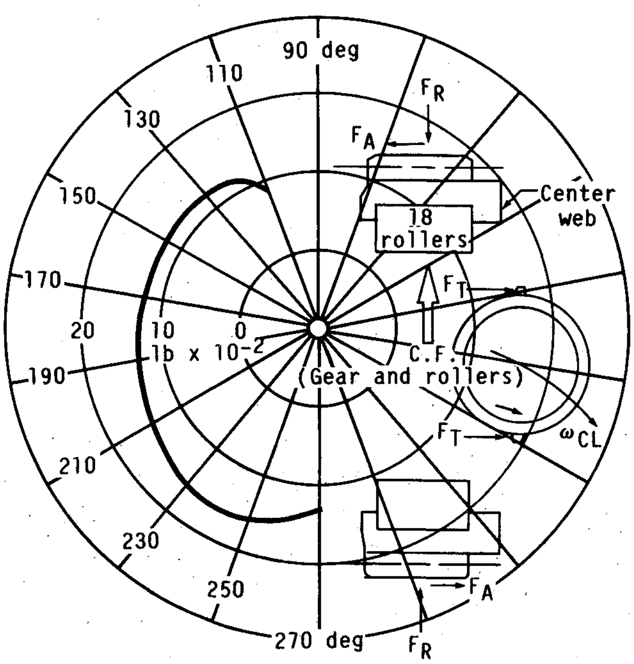

Fig. 5 AGBT planet gear-bearing reaction loads.

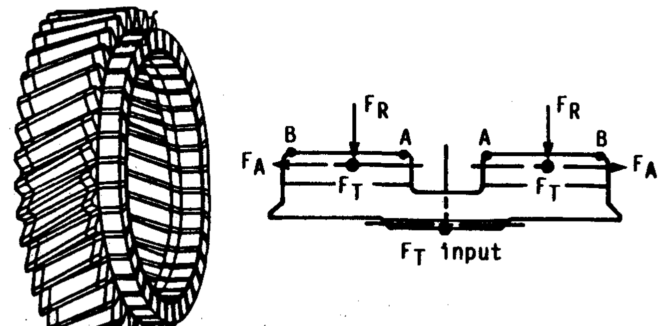

Fig. 6a Sun-gear shaft finite-element model.

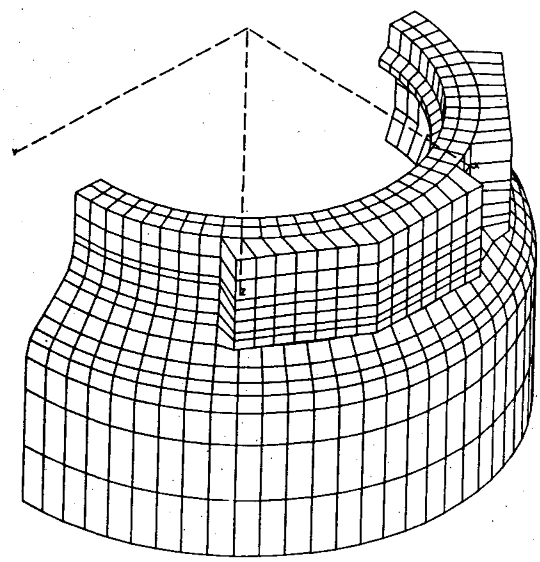

Fig. 6b Three-dimensional finite-element housing model.

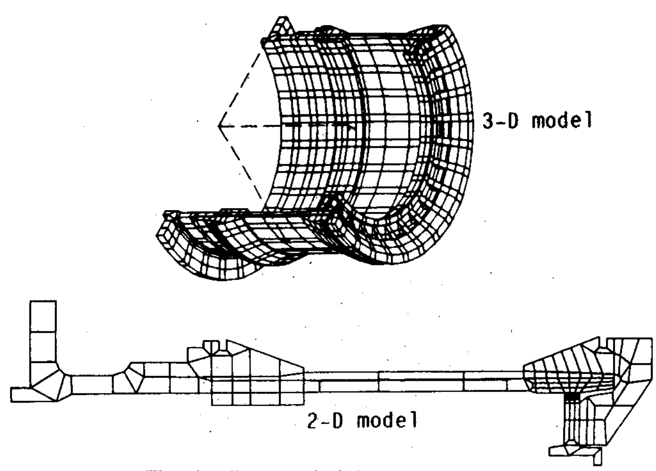

Fig. 6c Prop shaft finite-element model.

Other finite-element models used to design the gear system are shown in Figs. 2 and 6. The carrier model became very complicated due to the difficult geometry and three-dimensional loading. This model was used to determine stresses and deflections and required fastener clamp loads. The sun-gear rim thickness was optimized with the model shown in Fig. 6a. Radial deflections were limited to prevent gear-tooth mesh misalignment. The housing and prop shaft geometries were optimized for both stress and deflection characteristics.

V. Test-Facility Description

The counterrotating AGBT gear system was tested in-house in a newly constructed back-to-back test facility. Back-to-back gear-system testing consists of operating two gear trains of equal ratio in opposition to each other. With the input and output shafts connected, one gear system operates as a speed increaser and the other as a reducer. In this case, the reducer is the test gear system. By introducing torque into the system, power will recirculate between the two.

A new facility was constructed in 1985 specifically for development testing of propfan gear systems. The counterrotating gear-system test stand is shown in Figs. 7 and 8. As shown, the facility is capable of running systems to 16,000 hp. The power to drive the rig was supplied by two 500 hp motoring dynamometers. Because the maximum speed of the dynamometers is 3000 rpm, the output of the dynamometers drives through a 5 : 1 speed-increaser gear system into the torque applier. The torque-applier gear system reduces the

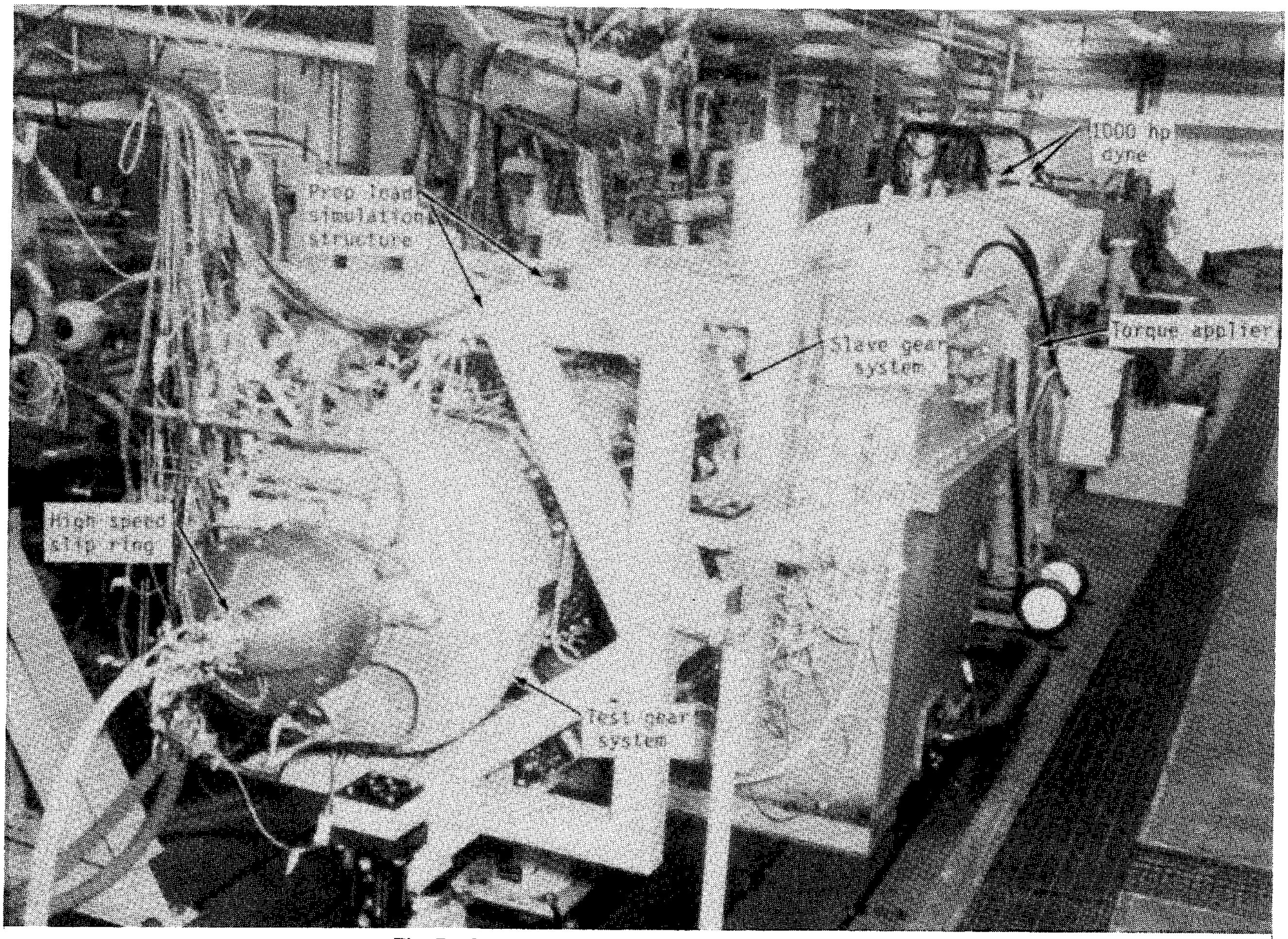

Fig. 7 Counterrotating propfan gear-system test stand.

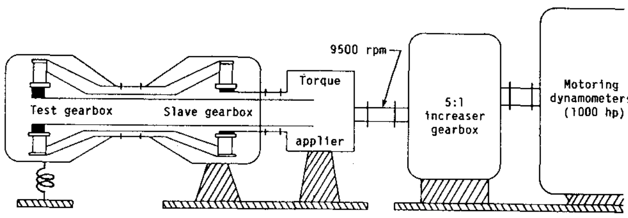

Fig. 8 Schematic diagram of the counterrotating propfan gear-system rig.

speed internally by a 2.6 : 1 ratio to drive two Rotac hydraulic torque appliers (maximum operating speed of 5000 rpm) and then steps the output (two concentric output shafts) back up by the same ratio. The two output shafts of the torque applier are connected to the input shafts of the test and slave gear systems. With the test and slave systems coupled to the outputs of the torque-applier gear system, the torque loop of the rig is completed. A schematic diagram of the torque loop is shown in Fig. 9.

The slave gear system is mounted rigidly to the torque applier, while the test gear system is cantilevered from the slave. Collars are mounted to the test and slave gear-system housing mount pads. Hydraulic rams between the two collars are used to apply thrust, moment, and side loads.

Because the AGBT gear system is a differential planetary system, a method was required to fix the two output shaft speeds. (The differential fixes the torque split between the two output shafts.) A speed-coordinating gear train was built into the slave accessory housing that connects the input sun-gear shaft to the plant carrier. When these two shafts are properly

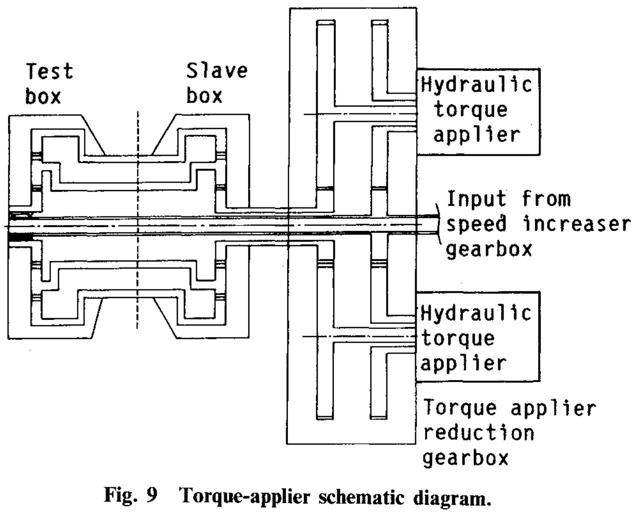

Fig. 9 Torque-applier schematic diagram.

connected, the ring gear (outer prop shaft) must rotate at the same speed and in the opposite direction to the carrier (inner prop shaft). In addition to fixing the two prop shaft speeds, the speed-coordinating gear shaft was instrumented with a torque bridge to measure the amount of power required to force the prop shafts to run at equal speed. This speed-control torque was then used as a health monitor of the test and slave gear systems throughout the testing. A schematic of the test rig with the speed-coordinating gear train is shown in Fig. 10. Power flow in the test rig is indicated by arrows.

The test and slave gear systems were lubricated with remote oil systems. Special features of the lubrication systems in-

Table 5 Gear-system temperatures (°F) at full power (13,000 hp)

Oil-out temperature, 223; average housing temperature, 217

Bearing	Bearing temperatures: Inner race	Outer race
Inboard prop	—	211
Outboard prop	193	199
Carrier ball	209	—
Carrier roller	218	219

Table 6 Gear-system vibration levels (in./s) at full power (13,000 hp)

Input housing	
Vertical	0.31
Lateral	0.28
Axial	0.14
Main housing	
Vertical	0.23
Lateral	0.20
Axial	0.21

cluded the following: 3 μ abs filtration ($\beta_3 > 200$) (supply and scavenge); Lubriclone* air/oil separator; Quantitative Debris Monitor (QDM)* system.

The oil-distribution plumbing for the test and slave gear systems was totally external. The oil supply to each area of the gear system could be controlled by changing orifices upstream of the internal oil jets.

Propfan Gear-System Testing

A series of tests were conducted to verify the structural integrity of the new design and provide a comparison of test data to the performance predicted. The test gear system was designed to accept a large number of sensors including strain gages, thermocouples, proximity probes, and accelerometers as well as an internal telemetry system. In total, more than 140 parameters were continuously recorded throughout the test on the test gear system, slave gear system, and test equipment.

The following operating conditions were attained: steady-state speed and torque, 0–11,000 rpm input speed and −500 to +7200 ft-lb; steady-state speed and torque with propfan loads, −10,000 to +15,000 lb, 100,000 in.-lb moment load; transient conditions including accels and decels to and from 10,000 rpm input speed with loads from −500 to +2000 ft-lb input torque.

VI. Test Data

A considerable amount of data was recorded during testing. To simplify this discussion, only the test gear-system data will be presented. Slave gear-system performance was similar to that of the test unit.

The data presented in the first several sections were run without prop loads. The effect of prop loads will be discussed later.

Temperatures

Gear-system temperatures were well behaved as shown in Figs. 11 and 12 (oil-in temperature was controlled to 180°F). Temperatures are shown as a function of speed since they did not change appreciably with torque. Table 5 shows these temperatures at full design power, 13,000 hp at 9500 rpm. The maximum temperature at any of these locations at full design power was 223°F.

*Trade name, Tedeco Division, Aeroquip Corporation.

Table 7 Gear-system efficiency

Condition	Speed (rpm)	Power (hp)	Efficiency: Predicted	Measured
AGBT design point	9500	13,000	99.5	99.4
578-DX takeoff	10,290	10,400	99.3	99.1
578-DX cruise	10,290	4700	99.1	98.6

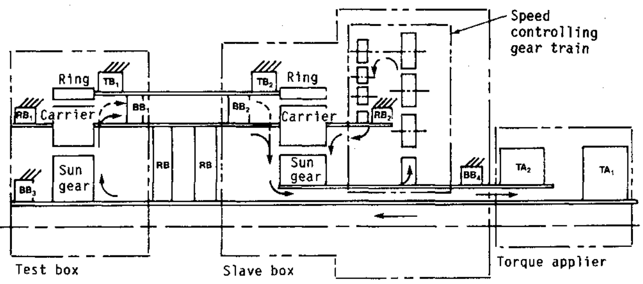

Fig. 10 Schematic diagram of the test rig and speed-control gear train.

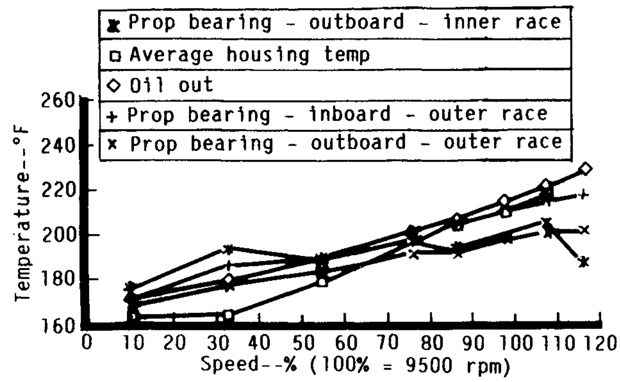

Fig. 11 Prop bearing, housing, and oil temperatures as a function of speed.

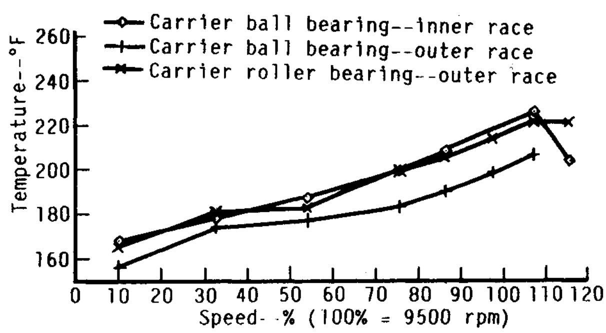

Fig. 12 Carrier ball- and roller-bearing temperatures as a function of speed.

Data from the internal telemetry system were not usable due to a transmitter malfunction. This system included the planet-bearing temperatures and carrier ball- and roller-bearing inner race temperatures. Observations during the test indicated that these temperatures were similar to those in Fig. 11.

Vibration

Figure 13 shows the average gear-systems vibration level at six locations on the test gear-system housing as a function of speed. These levels are quite low and represent smooth operation. Vibration levels were primarily a function of speed just as were the temperatures. The dominant frequencies correspond to input- and output-shaft rotational speeds. Vibration levels at the AGBT design point, 13,000 hp, are shown in Table 6.

Figure 14 shows a comparison of vibration levels at discrete frequencies in various Allison gear systems. Vibration at the gear-mesh frequency, the most objectionable type of vibration, was quite low in this gear system. The use of double-helical gearing significantly reduced gear-mesh vibration by improving the meshing action, i.e., contact ratio.

Efficiency

Gear-system power loss was determined by calculating the heat transferred to the oil. Housings were not insulated in these tests, but convective losses were determined to be low. If we use the highest convective heat-transfer coefficient applicable to this situation, 3 Btu/hr ft^2 °F, 0.5 hp would be lost to the atmosphere at the peak housing temperature. Compared to the total power loss at 13,000 hp (83.6 hp), this is a small percentage.

The results of these calculations are shown in Table 7. Also shown are the predicted values corresponding to takeoff and cruise power levels of the future Allison propfan demonstrator system, the 578-DX. The high power points agree well with the predicted values. Part power efficiency was lower than predicted due to higher tare (no-load) losses. It is thought that oil was not being effectively removed from the housing during this test due to the remote location of the test equipment scavenge pump.

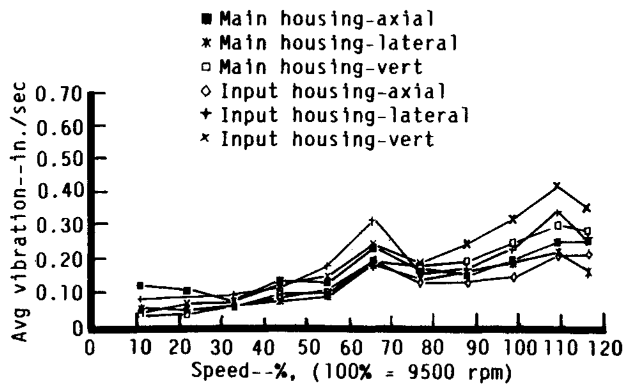

Fig. 13 Gear-system vibration as a function of speed.

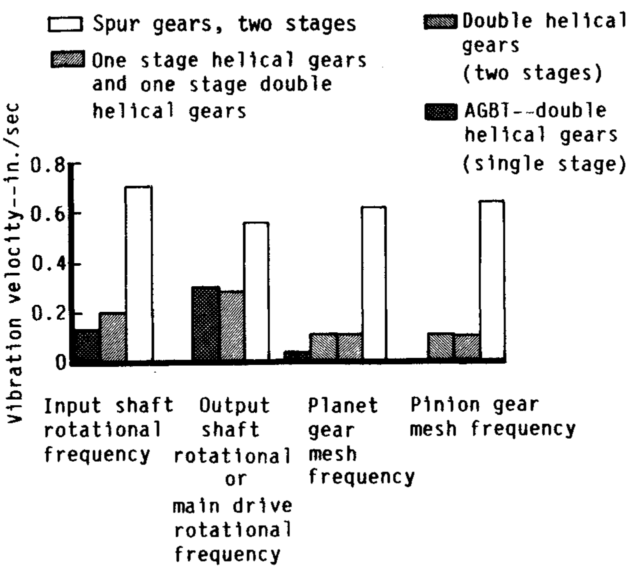

Fig. 14 Vibration levels for large Allison gear systems.

Effect of Prop Loads

Prop loads were applied to the gear system at two operating speeds while transmitting approximately 50% power. In the initial phase of testing described here, the prop load system was not capable of developing the full design loads. Thrust from −10,000–+15,000 lb and moment loads to 100,000 lb were achieved, however. Temperatures and average vibration levels were affected more by the change in speed than prop loads. The maximum temperature increase at any location due to these loads was 6°F. The only exception was 25°F increase at the inner race of the outboard tapered roller bearing due to the 15,000 lb thrust load at 110% speed. Most vibration levels varied by no more than 0.1 in./s as the prop loads were changed.

Strain Gages

Stress levels at 26 locations were monitored on-line during this test. Most of these gages were positioned to detect the modal vibration of either the flexible diaphragm or the ring gears. The stress levels measured during the test and afterward in a more thorough analysis indicated that all components were functioning as predicted. These data validate the finite-element models used to design these critical parts and help to substantiate the 30,000 h life requirement.

Gear-System Condition After Teardowns

The condition of the gear system at teardown was excellent. Contact patterns on the gear teeth were good. The sun and planet gear had full tooth contact. The planet bearings were like new except for a few circumferential scratches on several rollers. The tapered roller prop bearings had a milky appearance around the roller track. This is representative of a poor λ ratio due to either lack of oil film or a rough surface finish. The 8 μ-in. rms surface finish will be improved to alleviate this condition.

VII. Summary

Requirements for a high-power propfan gear system were addressed in an advanced technology gear-system program, the objective of which was to design and test an efficient, long-life, counterrotating gear system in the 10,000–15,000 hp range. Extensive finite-element analyses were used to predict and optimize performance of all gears and main-gear system structures. All bearings were evaluated with large-scale computer programs. Row-to-row tooth load sharing, a problem in some previous double-helical gear systems, was successfully resolved by utilizing a flexible diaphragm, allowing the gears to seek their optimum operating location. Planet-to-planet load sharing was accomplished through ring-gear radial flexibility. High hot hardness steels, which extend fatigue life and provide protection in case of oil loss, were selected for all gears and bearings. The gearbox was tested to the design point of 13,000 hp during a highly instrumented test in newly constructed propfan gear-system test facility. All measured values were well within acceptable limits.

Acknowledgments

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