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PARAMETRIC TESTS OF A TRACTION DRIVE RETROFITTED TO AN AUTOMOTIVE GAS TURBINE

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Douglas A. Rohn and Stuart H. Loewenthal
National Aeronautics and Space Administration
Lewis Research Center

and

Neil E. Anderson
Propulsion Laboratory
AVRADCOM Research and Technology Laboratories
Lewis Research Center

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U.S. DEPARTMENT OF ENERGY
Conservation and Solar Applications
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Prepared for
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National Aeronautics and Space Administration
Lewis Research Center
Cleveland, Ohio 44135

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Neil E. Anderson
Propulsion Laboratory
AVRADCOM Research and Technology Laboratories
Lewis Research Center
Cleveland, Ohio 44115

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by Douglas A. Rohn and Stuart H. Loewenthal
Lewis Research Center

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Propulsion Laboratory
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Cleveland, Ohio

ABSTRACT

Efficient and economical speed reducers are necessary for small, high speed, automotive type gas turbine engines. The results of a test program to retrofit a high performance fixed ratio Nasvytis Multiroller Traction Drive in place of a helical gear set to a gas turbine engine are presented. Parametric tests up to a maximum engine power turbine speed of 45,500 rpm and to a power level of 110kW were conducted. Comparisons were made to similar drives that were parametrically tested on a back-to-back test stand.

The drive showed good compatibility with the gas turbine engine. Specific fuel consumption of the engine with the traction drive speed reducer installed was comparable to the original helical gearset equipped engine.

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INTRODUCTION

Current developments in automotive gas turbine design emphasize low cost and improved efficiency for future engines. Other equally important considerations are size, weight, noise and reliability. Developmental efforts with these goals in mind are well underway on nearly all gas turbine components. One important component which is receiving increased study is the power turbine speed reducer. Current twin shaft, automotive gas turbines have power turbine speeds between 50,000 and 75,000 rpm. Single shaft turbine engines are being designed for speeds of about 100,000 rpm. The speed reduction gearsets required for these extremely high speeds require extremely accurate, high quality and consequently expensive gearsets. The pinions for these reducers are rapidly approaching bending strength and scoring operational limits. Noise, durability and vibration become major concerns for these fine-pitched, small, high pitchline velocity pinions. A new but promising alternative for this application are traction type speed reducers.

Traction drives use smooth rollers to transmit power so the operating problems and noise associated with gear teeth at extremely high speeds are eliminated. There have been several efforts in the past to develop fixed-ratio traction drives [1, 2]. Back-to-back parametric tests on a pair of identical 15 to 1 fixed-ratio, planetary traction drives with two rows of stepped planets showed good performance [3]. This transmission, the Nasvytis Multiroller Traction or NASVYTRAC Drive, appeared to be a good candidate for a primary speed reducer for an automotive gas turbine engine. The work presented in this report describes the results of a test program to evaluate the operational characteristics of the NASVYTRAC drive for this application.

BACKGROUND

High Speed Gearing

Fig. 1 shows three criteria used in high speed gear design, namely dynamic tooth loading, scoring and tooth breakage [4]. Dynamic tooth loads which increase with surface speed (Fig. 1(a)) are present to some extent even on perfectly cut gears due to the way the load is transferred from single tooth to double tooth contact and vice versa. Dynamic tooth loads give rise to increased vibration and noise. Backlash and any errors in tooth profile, lead or spacing tend to aggravate the situation. Modifications to the addendum and dedendum are made to improve meshing action. Fig. 1(a), based on the simplified Buckingham formula [4], shows that the dynamic loading for finer-pitched gears, that is those with more gear teeth, are less sensitive to increases in pitch line velocity, PLV.

Similarly, gears become more vulnerable to scoring with increased surface speed. Scoring, gross metal-to-metal surface distress and overheating, occurs when the sliding energy content of the elastohydrodynamic lubricant film separating the teeth becomes high enough to cause a sudden collapse of the oil film. Increased surface speeds will raise the shear rates and surface temperatures and thus increase the chances of scoring as shown in Fig. 1(b). For the same pitch line velocity, higher tooth numbers (i.e.: smaller gear teeth) have both a shorter tooth contact time and lower relative sliding speeds, producing a lower relative scoring index.

Fig. 1(c), shows the relative tooth breakage index which is a function of the bending fatigue strength of an individual tooth as a cantilever beam. Again as speed increases, tooth breakage increases. However, while the dynamic tooth loading and scoring index benefit from making the teeth smaller and more numerous, the tooth breakage index becomes worse since the fine teeth are too small to carry the load.

Philosophically, extrapolating the trends shown in Fig. 1 to the limit, results in a "gear" with an infinite number of infinitely small teeth, or a roller which would have no dynamic loads to contend with and no suseptibility to scoring or tooth breakage. In essence, the result is a traction roller which can transmit power smoothly without many of the speed limitations of gear teeth. Speed limits for traction contacts are not well defined since these limits are established primarily by the traction characteristics of the lubricant at extremely high speeds where little data exists.

Traction Power Transfer

In a traction drive, torque is mainly transmitted by shear forces acting through a thin, elastohydrodynamic, EHD, lubricant film which separates the driving and driven rollers as shown in Fig. 2. Under the high pressures (1 to 2GPa) and severe shear rates (10^3 sec^{-1}) present within a typical traction contact, the lubricant is thought to be transformed into an amorphous solid or plastic-like material [5]. Because of this lubricant transformation within the EHD film, significant torque transfer can occur without appreciable metal-to-metal contact or wear.

The amount of tangential force that a certain lubricant can transmit is related to the normal load imposed on the rollers by a traction coefficient, μ where: $\mu = T/N$.

Traction drive lubricants can typically develop peak μ values from 0.05 to 0.1 depending on operating conditions such as surface speed, contact pressure, temperature, contacting element shape and surface finish. These special synthetic lubricants have about 50 percent greater μ than conventional oils [6] and therefore significantly improved traction drive power density and life.

NASVYTRAC DRIVE

Concept

Because of the relatively high normal loads between rollers, traction drives should be constructed in a planetary configuration in order not to severely overload support bearings. Fixed-ratio, planetary traction drives with a simple, single row planet roller format have internally balanced loads but are generally limited in speed ratio, particularly for a large number of planets. The ring-to-sun diameter ratio, which determines the drives' speed ratio, decreases with an increase in the number of planets. For a high power-density traction drive it is essential to maximize the number of multiple, load sharing contacts working in parallel. The NASVYTRAC drive concept circumvents the planet number limitation of simple planetary systems by using two or more rows of dual diameter planets. Much higher ratios can be obtained; up to 150 to 1 with three rows of stepped-planets [3, 7]. This geometry, for a given speed ratio, enables a large number of planets to be located in each row, splitting the power through many parallel paths and thus reducing the loading on each contact. This results in a drive with higher torque capacity and improved fatigue life.

The basic geometry of the NASVYTRAC test drive is shown in Fig. 3. Two rows of five stepped-planet rollers, grounded by bearings to the case, are contained between the concentric high speed sun and low speed ring rollers. The reaction bearings, mounted only in the outer planets, are in a favorable position since the reaction forces and operating speeds are relatively low.

The sun roller and first row of planets float freely, relying on three-point contact with adjacent rollers for location. The drive functions like a large roller bearing eliminating the need for any high speed sun roller support bearings. In addition, the floating three-point contact planet support system accommodates slight mismatches in roller dimensions, housing deflections under load, or thermal distortions. These errors in position merely cause a slight imbalance in roller loading and a small shift in roller orientation until the internal force balance is re-established. Because of this roller cluster flexibility, roller dimensional tolerances can be relatively generous compared with those set for conventional rollers in roller bearings.

The number of planet-roller rows, the number of planet rollers in each row, and the relative diameter ratios at each contact are variables to be optimized according to the overall speed ratio and the uniformity of contact forces. In general, drives with two planet rows are suitable for speed ratios to about 35, and drives with three planet rows are suitable for ratios to about 150.

Table I summarizes the main features of the NASVYTRAC drive geometry relative to gear systems.

Parametric Testing

References [3 and 8] report the results from earlier prototype tests on the NASVYTRAC drive performed at the NASA Lewis Research Center. In these earlier tests, two drives of slightly different geometrics and ratios were parametrically tested to 180kW on a back-to-back test stand shown in Fig. 4 to determine key performance factors. The work reported here describes the dynamometer tests of one of these units which was retrofitted to the power turbine assembly of an automotive gas turbine engine.

Service Life

Traction drives, like rolling-element bearings, are generally sized on the basis of rolling-element fatigue life. This is because the operating speeds, contact pressures, temperatures, roller materials and stress cycles are equivalent to those of ball and roller bearings. The risk of failure of traction drive contacts from wear or scoring can be greatly minimized through the use of proper material and lubricating design practices such as those that have been successfully applied in bearing and gear design. In view of this similarity in the failure mechanism, namely rolling-element fatigue, the fatigue life theory of Lundberg and Palmgren can be used to size traction drives. This is the same method that bearing manufacturers use to establish the load capacity ratings for rolling-element bearings. The basic equations for traction drives have been developed and applied to a toroidal type traction drive and the NASVYTRAC configuration [9, 10]. Using these methods, the theoretical B_{10} (90-percent survival) life ratings for the NASVYTRAC drive (from [8]) that was retrofitted to the gas turbine engine are shown in Fig. 5. The roller cluster for this unit measured approximately 21 cm in diameter by 6 cm in width and weighed 7.6 kg. The data in Fig. 5 were calculated for a constant sun roller speed of 50,000 rpm.

Material and lubrication life adjustment factors recommended in [11] were applied to the life ratings along with an estimated, arbitrary life penalty factor of 0.5 to account for the potentially adverse effects of traction on rolling-element fatigue. Currently, insufficient data are available to properly quantify the effects of traction on life.

Continuous power capacity shown in Fig. 5 ranged from 42 kW for 10,000 hours system life to 185kW for a minimum of 100 hours at sun roller speeds of 50,000 rpm (or about 3,500 rpm on the low speed shaft). Since the weighted duty cycle load for a gas turbine vehicle is expected to be significantly less than 42 kW, this drive should be more than adequate. However for applications requiring more torque capacity, the drive size can be readily scaled. Since torque varies with size to the 2.8 power, according to [8], increasing the test drive size by 28 percent, that is increasing the roller clusters' overall diameter to 27 cm and width to 8 cm, will theoretically double torque capacity at a given life level, as shown in Fig. 5.

TEST ENGINE, TEST DRIVE, TEST PROCEDURE

Test Engine

The power plant used in this investigation was the Chrysler Corporation sixth generation automotive gas turbine engine. This engine was used as the baseline engine in an EPA program started in 1972 to demonstrate an experimental gas turbine which had competitive emissions, fuel economy, performance, reliability and potential cost with a conventional internal combustion piston engine [12].

This baseline engine is classified as a free power turbine, low pressure ratio, regenerative design. Manufacturer's engine performance curves showing the relationship of output power to output speed for lines of constant gas generator speed are shown in Fig. 6. Gas generator speed is roughly equivalent to throttle setting. Details of the calculation of corrected (or standardized) values of power, torque, speed and fuel consumption shown in this figure are given in reference [13].

Test Drive

The 14.1 to 1 fixed-ratio Nasvytis Multiroller Traction Drive used in this investigation consists of two rows of five planet rollers each, contained between concentric ring and sun elements.

The drive was equipped with an automatic roller loading mechanism which was incorporated into the sun roller. This loading mechanism adjusts the normal contact load between the rollers in proportion to the transmitted torque, effectively maintaining a constant traction coefficient. This torque-responsive loading mechanism insured that sufficient normal load was applied under all conditions to prevent slip, without needlessly overloading the contracts at light loads. The mechanism was designed to operate above some preselected, mechanically adjusted minimum load setting. If required, a constant level of roller normal loads could also be applied by locking the mechanism. This mechanism is shown in Fig. 7 together with the overall integration of the traction drive and power turbine. The geometry of the loading mechanism was chosen to impose a constant applied traction coefficient of 0.05.

All rolling traction elements were ground, case carburized, consumable vacuum-melted (CVM) SAE-9310 (AMS-6265) steel. The traction lubricant used in the tests was Santotrac 50 whose properties are described in reference [6].

Integration

The power turbine, 9.6875 ratio, 14.0 cm offset helical gear reducer was replaced by a 14.1 ratio, concentric NASVYTRAC drive as shown in Fig. 7. In the stock engine, two radial, fluid film bearings supported the power turbine rotor shaft, two radial fluid film bearings supported the pinion, and a fluid film thrust bearing reacted rotor thrust. For the NASVYTRAC drive installation, only the front fluid rotor bearing was retained. The remaining three radial and one thrust fluid film bearings were replaced by the NASVYTRAC drive and single, split inner race, 25 mm bore-diameter angular contact ball bearing. The self supported sun roller of the traction drive eliminated the need for the fluid film bearings which straddled the pinion and reacted gear separating forces. A spline coupling was used to connect the traction drive to the power turbine shaft, in place of the helical pinion.

An external temperature and flow regulated lubrication system supplied the traction drive and power turbine support bearings with SANTOTRAC 50 lubricant. The kinematic viscosity and other key properties of this fluid at operating temperatures are sufficiently similar to those of the automatic transmission fluid normally used in the engine so that no noticeable performance differences were expected.

Details of the integration of the traction drive and power turbine assembly can be found in reference [14]. A photograph of the power turbine assembly which incorpoates the test traction drive appears in Fig. 8.

Test Stand

A photograph of the test engine installed on the NASA automotive gas turbine test stand appears in Fig. 9. The output shaft from the traction speed reducer is directly connected via a propshaft to the power absorber. The black case containing the traction drive appears on the left side of Fig. 9. Two large exhaust ducts connected to the regenerators on each side of the engine are also apparent.

Instrumentation for these tests included: oil supply and return thermocouples, oil flow meters, thermocouples to measure key bearing and roller temperatures, and proximity probes to record rotor shaft dynamics. Proximity probes were also used to determine sun roller loading mechanism action, and to accurately measure power turbine speed and output speed in order to determine traction contact creep. Creep is the small relative motion between driving and driven rollers in a traction drive which accompanies torque transfer. It is primarily elastic in nature as opposed to "slip" which is non-elastic. Details of the type and location of instrumentation can be found in [14].

Test Procedure

Two parametric tests of the gas turbine engine-traction drive assembly were run: a fixed-preload test series where the roller loading mechanism was locked at a constant (near maximum) roller load and a variable roller loading test series. The testing sequence used was to maintain a constant (dynamometer controlled) output speed while step-wise increasing torque by adjusting the engine throttle setting. To insure steady state conditions in the engine and drive, 10 to 15 minutes of running was allowed after speed changes and 5 to 10 minutes after torque changes before recording a data point.

The fixed-preload test was conducted to check out the engine-drive interactions with a simplified (no loading mechanism) traction drive system. The total lubricant flow rate to the traction drive was set at 0.13 l/s and the power turbine journal and ball bearings received 0.04 l/s. The inlet temperature on both streams was 328 K. Maximum output torques at output speeds of 800, 1600, 2400, and 3150 rpm were 407, 373, 407, and 244 N-m respectively.

After a successful fixed-preload test, the variable roller loading mechanism was activated. Minimum roller preload was set at about 25 percent of maximum in order to guard against the possibility of roller skidding at light loads and high speeds. The lubricant flow rates were the same as in the fixed-preload tests. Lubricant inlet temperatures were set at 339 K. Maximum output torques at output speeds of 800, 1600, 2400, and 3150 rpm were 407, 475, 407, and 271 N m respectively.

For comparison purposes, a test series was also conducted on the engine with the original helical gearset reinstalled. To make a reasonable performance comparison, test points for the gearset test were chosen to correspond to the same gas generator speed and power turbine speed points recorded during the variable roller load parametric tests. All test utilized an engine oil inlet temperature of 339 K.

RESULTS AND DISCUSSION

Traction Drive Performance

Fig. 10 shows the comparative power loss results from the two traction drive test series. The traction drive power loss values shown in this figure were calculated from the amount of heat rejected to the lubricant as it passed through the drive cavity. Using a similar heat balance method, power loss was calculated for the power turbine shaft's fluid film journal and angular contact ball bearings based on the increased heat energy of the bearings' lubricant. No attempt was made to quantify the heat convected to the atmosphere through the traction drive casing, but this was judged to be relatively small due to the insulation that encapsulates the power turbine housing assembly.

Fig. 10 shows that when operating in a fixed-preload mode, the traction drive power loss is mildly dependent on torque and is nearly linearly dependent on speed. This suggests that the drive's drag torque is essentially constant. This is in contrast to the "variable" drag torque associated with the variable roller loading system. At low torques, the normal contact loads, hence losses, for the variable loading system are significantly less than for fixed-preload. As the drive torque level increases, the variable roller loading mechanism increases contact loads to approximately the same levels set by the fixed-preload mechanism and thus the respective power losses are nearly the same.

A comparison of traction drive and power turbine bearing efficiences for the fixed-preload and variable roller loading test series is shown in Fig. 11. These efficiencies are calculated based on measured output shaft power and the heat balance power loss. As would be expected from the power loss data of Fig. 10, the efficiency of the fixed-preload drive is lower than the variable roller loading drive at part load conditions. The efficiencies merge at higher torques. The bands of traction drive efficiency reflect a slight dependence on speed, with the higher efficiencies generally occurring at low output speeds at a given output torque. Power turbine bearing efficiency was essentially independent of speed and type of loading mechanism. Also shown on Fig. 11 is the average efficiency of the traction drive (as a speed reducer) measured on the back-to-back test stand over the range of engine test speeds [8]. Acceptable agreement is obtained between the stand measured efficiency and that determined from a heat balance on the lubricant in the engine tests.

The effect of operating torque and speed on traction drive creep rate is presented in Fig. 12. Included for comparative purposes are the back-to-back stand data from [8] for the variable loading drive at 2500 rpm output speed. Good agreement exists between both sets of creep data.

Also shown in Fig. 12 is a comparison between fixed and variable loading creep data. The lower creep rate (about 0.6 percentage points) associated with fixed-preload operation is due to the higher initial normal contact load. The upward, curved trend of the fixed-preload creep data of Fig. 12 suggests that the peak traction coefficient point is being approached. On the other hand, the variable roller loading curves show a tendency to level off at a nearly constant traction coefficient. With a further increase in torque, the creep rate for fixed-preload operation would exceed that for variable roller loading operation. Although at part loads the creep rate for the fixed preload drive is less, nevertheless, the overall efficiency is inferior to that obtained with variable roller loading operation (as shown in Fig. 11) due to the losses associated with contact overloading.

To determine the effectiveness of the roller loading mechanism, a proximity probe was installed in the drive. The probe monitored the axial position of one side of the two piece sun roller assembly. In this drive, as the tapered sun roller halves moved together, the normal load on the roller cluster would be correspondingly increased. During the variable roller loading test, as shown in Fig. 13, the sun roller halves moved inward together in a nearly linear fashion with increasing torque indicating satisfactory roller loading action.

Roller temperatures in the traction drive increased with operating speed. The sun roller temperature, the hottest roller in the drive, never exceeded 42 K above the oil supply temperature. Planet roller support bearings had lower operating temperatures. The power turbine rotor's fluid film journal and ball bearings reached maximum temperatures of 401 and 389 K respectively.

In general, the Nasvytis drive demonstrated good operational compatability and performance with the gas turbine engine throughout the engine's torque and speed range. Orthogonal, radial proximity probes mounted at two axial positions along the power turbine shaft, showed that the coupled rotor-traction drive system was reasonably stable from engine idle to maximum speed. No rotor instabilities were encountered.

Engine Performance

Fuel flow measurements and specific fuel consumption calculations were made at each test point for both the traction drive and gearset engine tests. Fig. 14 shows the specific fuel consumption (SFC) versus gas generator speed for three nominal power turbine speeds. In this plot, the values of SFC and gas generator speed are corrected values, whereas those for power turbine speed are not. For this comparison, the power turbine and gas generator speeds were chosen for the gearset test so that the operating conditions of the engine itself (not including the speed reducer) would be similar to that of the traction drive test. This was required because the output shaft torque and speed were different for each test series due to the traction drive's higher reduction ratio. Fig. 14 shows that the SFC for the traction drive equipped engine is approximately equal to that for the gearset equipped engine.

Cross-plotting the data from Fig. 14 as a function of power turbine speed with lines of constant gas generator speed, the points of maximum power for each gas generator speed can be identified. The SFC for each of these maximum power points can then be plotted versus gas generator speed as shown in fig. 15. This figure shows that the SFC of the engine with variable roller loading traction drive speed reducer was comparable to that of the baseline engine.

SUMMARY AND CONCLUSIONS

Parametric tests were conducted on a 14.1 ratio Nasvytis Multiroller Traction (NASVYTRAC) Drive retrofitted to an automotive gas turbine engine in place of the stock, 9.69 ratio helical gear reducer. The traction drive had a planetary configuration with two rows of stepped-planet rollers containing 5 planets per row. The sun roller assembly was equipped with an automatic roller loading mechanism. A traction fluid was used to lubricate the modified power turbine assembly and traction drive. The effects of speed and torque on drive power loss, efficiency, creep rate, temperature distribution, and loading mechanism operation were investigated. Tests were conducted to full engine power turbine speed of 45,500 rpm and to estimated power input levels of 110 kW. Comparisons were made to earlier parametric data from back-to-back tests. Drive performance under fixed-preload operation was compared to that under variable roller loading operation. Comparisons were also made between the specific fuel consumption of the traction drive equipped engine and the engine with the original helical gearset. The following results were obtained:

1. The NASVYTRAC drive showed good operational compatibility with the automotive gas turbine engine throughout the range of test conditions.

2. Specific fuel consumption of the engine with the traction drive speed reducer installed was comparable to the original helical gearset equipped engine.

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TABLE I - FEATURES OF NASVYTRAC

- o HIGH SPEED OPERATION
  - ELIMINATES SPEED-LIMITED GEAR TEETH
- o QUIET, SMOOTH
  - ELIMINATES GEAR MESH NOISE AND VIBRATION
- o HIGH RATIO SINGLE COMPACT STAGE
  - ELIMINATES MULTISTAGE GEARING & EXTRA WEIGHT
- o ACTS AS BEARING
  - ELIMINATES INDIVIDUAL BEARINGS
- o SIMPLE ROLLERS
  - ELIMINATES PRECISION GEAR TEETH

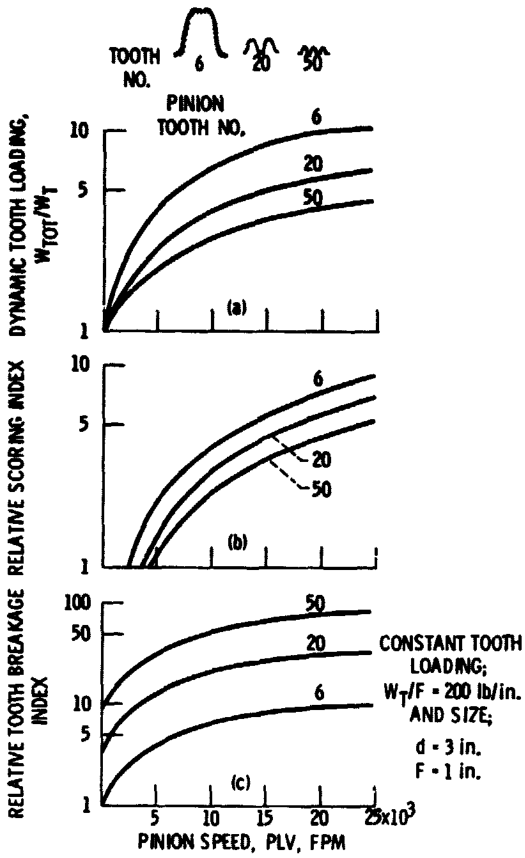

Figure 1. - High speed gear design criteria.

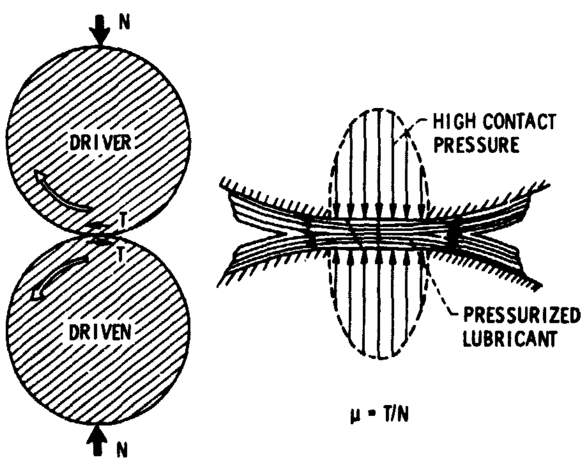

Figure 2. - Power transfer through traction.

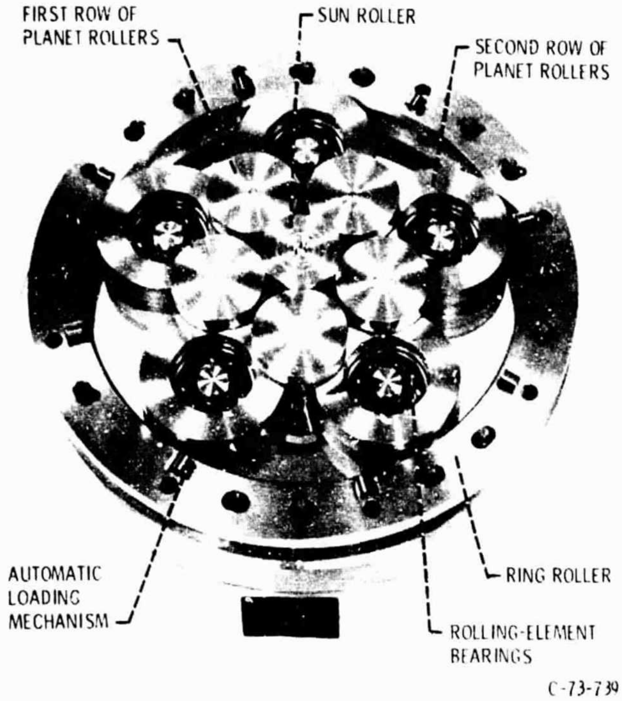

Figure 3. - Geometry of the Nasvytis traction (NASVYTRAC) test drive.

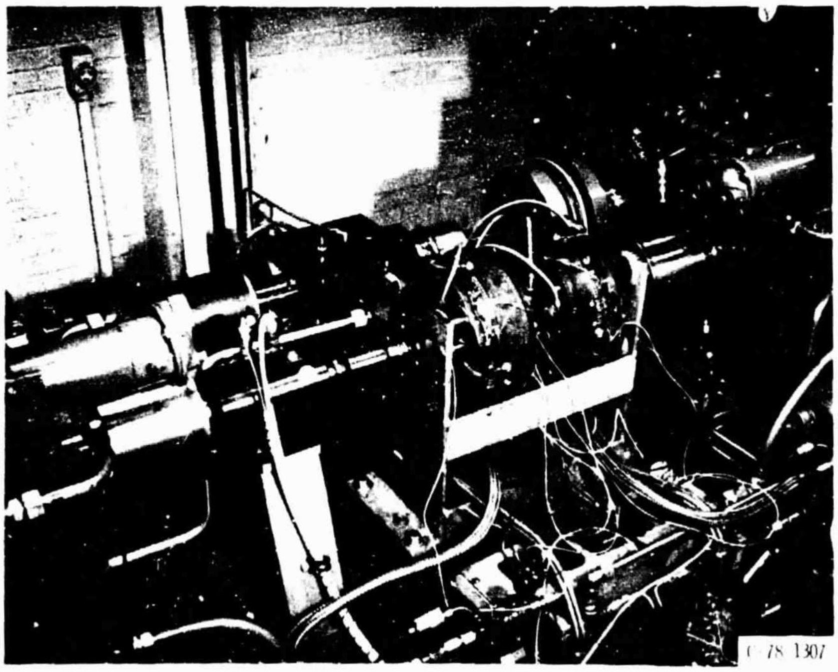

Figure 4. - Back-to-back traction drive test stand.

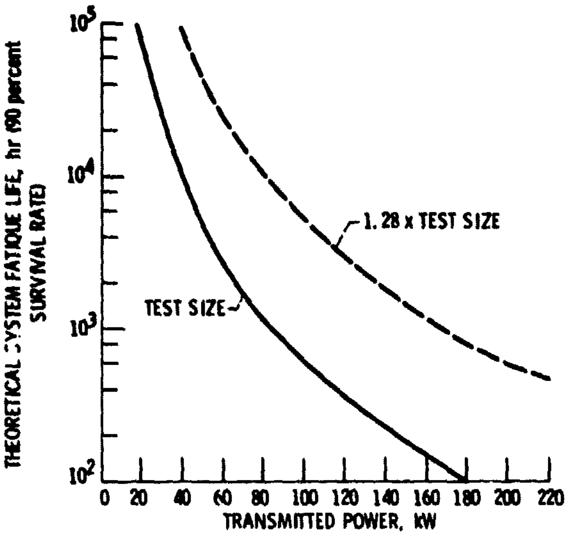

Figure 5. - Traction drive fatigue life at 50 000 rpm input speed.

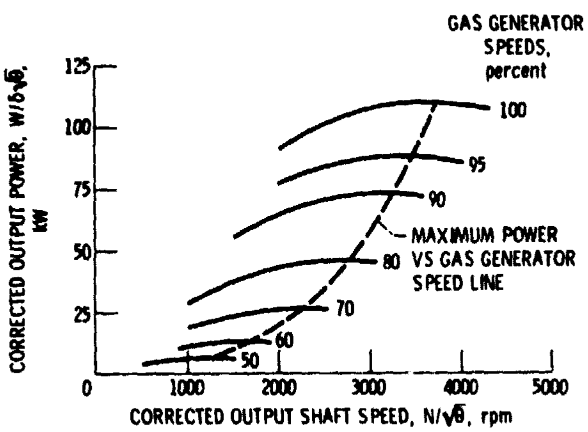

Figure 6. - Gas turbine engine power vs output shaft speed.

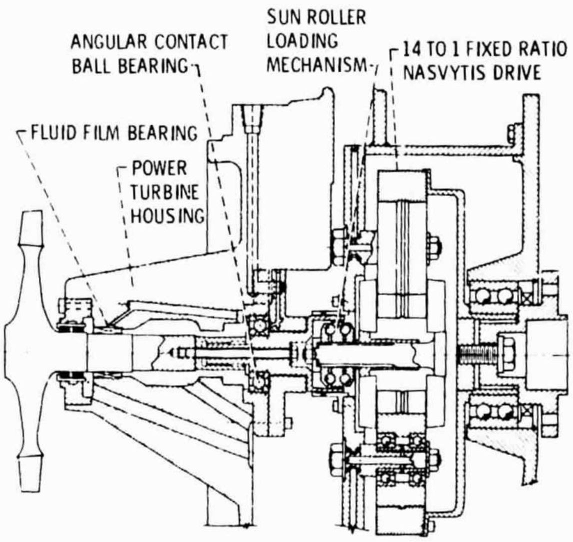

Figure 7. - Installation of Nasvytis traction drive with power turbine assembly.

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Figure 8. - Power turbine assembly. Traction drive case shown on rear of housing.

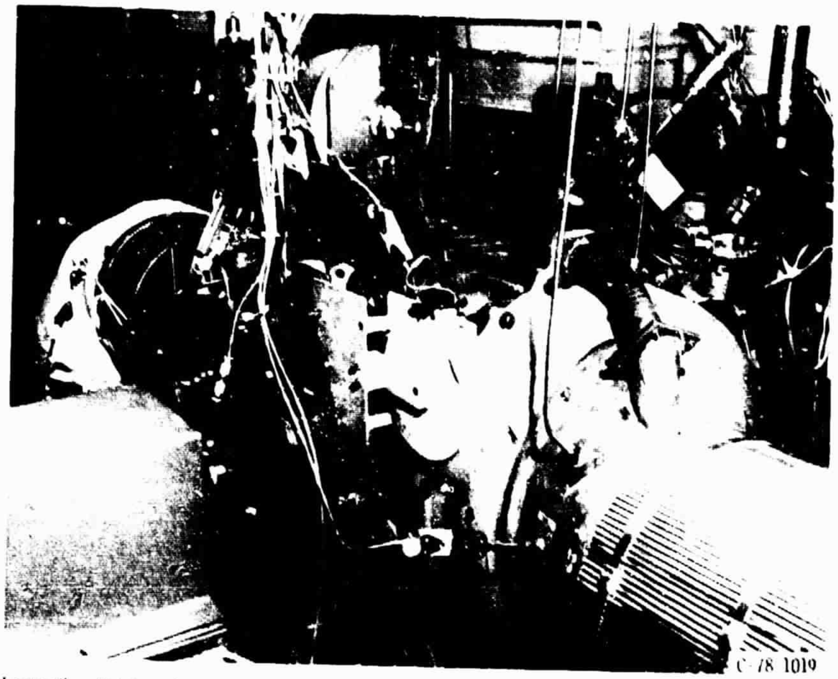

Figure 9. - Traction drive equipped engine on NASA engine test stand. Traction drive case shown on rear (to left) of engine.

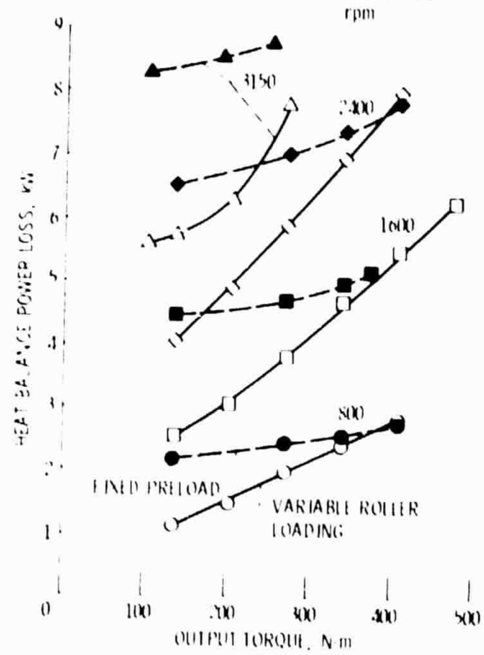

Figure 10. Traction drive power loss.

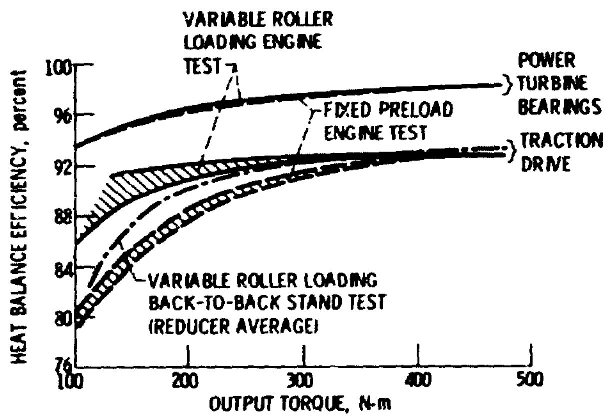

Figure 11. - Traction drive efficiency.

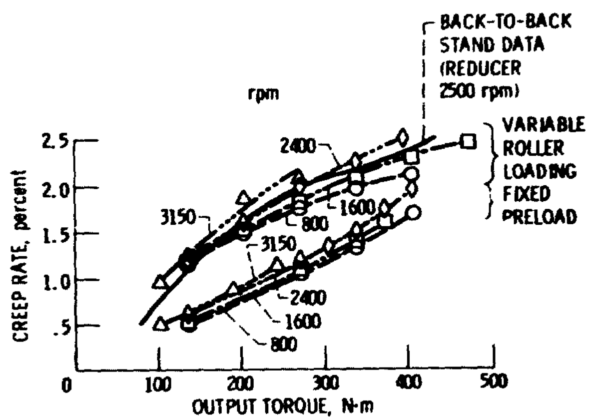

Figure 12. - Traction drive creep.

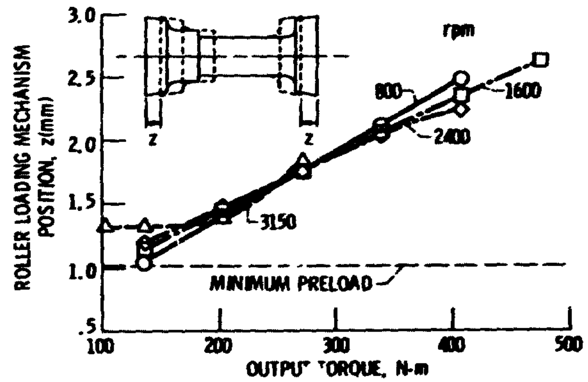

Figure 13. - Variable roller loading mechanism operation.

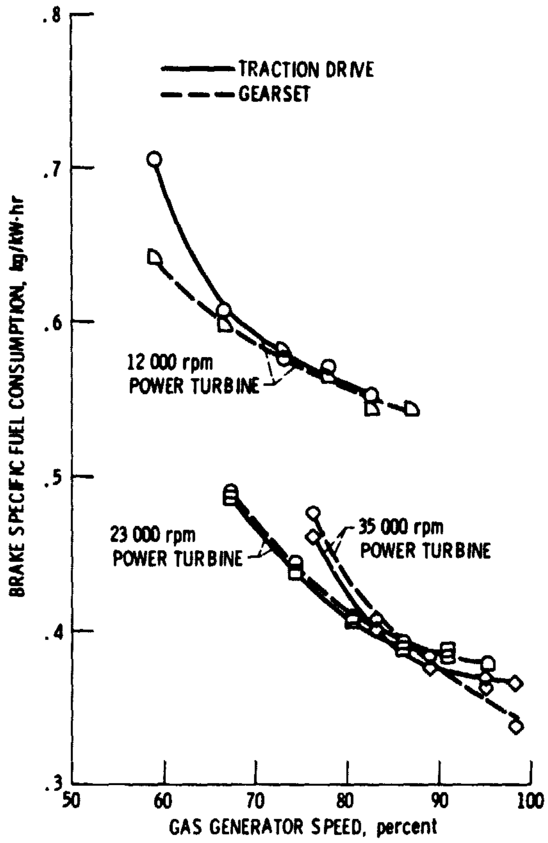

Figure 14. - Comparison of SFC for gearset and traction drive on turbine engine.

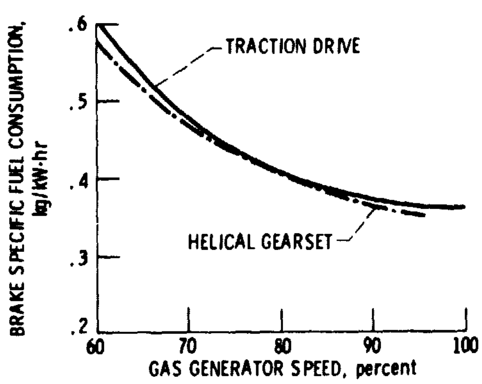

Figure 15. - Comparison of SFC at maximum power for gearset and traction drive on turbine engine.