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NASA TRANSMISSION RESEARCH AND ITS PROBABLE EFFECTS ON HELICOPTER TRANSMISSION DESIGN

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Abstract

NASA transmission research is oriented either to advance the state-of-the-art in mechanical power transfer technology or to add to the fundamental body of knowledge belonging to bearings, gears, lubrication, rolling-element fatigue, life prediction, traction phenomena, and mechanical power transfer. Transmissions studied for application to helicopters in addition to the more conventional geared transmissions include hybrid (traction/gear), bearingless planetary, and split torque transmissions. Research is being performed to establish the validity of analysis and computer codes developed to predict the performance, efficiency, life, and reliability of these transmissions. Results of this research should provide the transmission designer with analytical tools to design for minimum weight and noise with maximum life and efficiency. In addition, the advantages and limitations of new and novel drive systems as well as the more conventional systems will be defined.

Introduction

The helicopter, more than any other contemporary aerospace or industrial innovation, has placed severe performance demands on power transmission components such as bearings and gears. Well-designed mechanical components, good materials, and lubrication systems which are integrated into helicopter drive train systems can make the difference between a helicopter's reliable, economic operation and failure.

It has long been a requirement to provide technology to obtain long-life, efficient, lightweight, and compact mechanical power transmissions that are also low-cost and quiet for both commercial and military helicopter applications. In general, current state-of-the-art transmission systems are disturbingly noisy to the pilot and passengers. The maintenance rate on these transmission systems is high. The time between overhaul (TBO) and mean time between failures (MTBF) on present-day helicopters is much lower than that required for economical commercial operation. The helicopter drive system is generally heavier than desired [1].

The realization of technological improvements for future helicopter drive systems can only be obtained through advanced research and development. Hence, NASA transmission research is oriented either to advance the state-of-the-art in mechanical power transfer technology or to add to the fundamental body of knowledge belonging to bearings, gears, lubrication, rolling-element fatigue, life prediction, traction phenomena, and mechanical power transfer [1]. A considerable amount of work is required to establish the validity of analysis and computer codes developed to predict the performance, efficiency, life, and reliability of transmission systems.

Relatively new concepts may be required to achieve significant technological advances. NASA transmissions studied for application to helicopters in addition to the more conventional geared transmissions include hybrid (traction/gear) [2], bearingless planetary [3], and split torque transmissions [4].

The NASA Lewis Research Center in cooperation with the U.S. Army Aviation Research and Development Command's Propulsion Laboratory devised a comprehensive helicopter transmission technology research program beginning October 1977 [1]. This paper reviews the results of this research and its probable effects on helicopter transmission design.

Gears

Life and Reliability

A reliability model for the compound planetary gear train (Fig. 1) has been derived for use in the probabilistic design of this type of transmission [5,6]. This gear train has the ring gear fixed, the sun gear as input, and the planet carrier as output. The input and output shafts are assumed to be coaxial with the applied torques and each other; no side or moment loading is considered.

The reliability model is based on the reliability models of the bearing [7,8] and gear mesh components [9-11] which are two dimensional Weibull distributions of reliability as a function of life. The transmission's 90-percent reliability life and basic dynamic capacity are presented in terms of input sun rotations and torque (Fig. 2). Due to the different Weibull distributions for the bearing and gearing components, the Weibull model for the planetary transmission is an approximate model. This model includes the transmission's 90-percent reliability life, Weibull exponent, basic dynamic capacity, and load-life exponent. The life and reliability model allows the designer to obtain both qualitative and quantitative comparisons between transmission designs and applications. For an example, the analysis shows that due to the nature of the component life distributions, reducing the loading in the transmission makes the bearings more important in the life characteristics of the transmission. Increasing the loading makes the sun gear life more important in the overall life distribution of the transmission. In addition, adding a fourth planet gear more than doubles the life of the transmission [6].

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Spur Gear Design

The design procedure for designing gear sets as shown in Figs. 3 and 4 so they will have a minimum center distance was developed [12,13]. A minimum center distance design will be lighter weight, which is critical in aircraft applications. Another advantage is that smaller gears will have less noise due to the smaller pitch line velocities.

The derivation of the design procedure identified a two-dimensional design space whose coordinates are number of teeth and diametral pitch. Constraint boundaries for pitting, scoring, and bending fatigue failure, as well as the geometric constraint of involute interference were identified as shown in Fig. 5. The region of acceptable design choices is labelled in the upper left of the design space. Lines of constant slope through the origin represent the locus of points in the design space for which the center distance is constant. The slope of the line represents center distance; the smaller the slope, the smaller the center distance. The minimum sized gear design would then be a gear with diametral pitch and number of teeth which correspond to point A on the plot.

A design approach sometimes found in gear handbooks is to use the interference limit and the tooth bending strength limit to define the "best" design. This gives point B in the design space. This point will give a smaller gear set than point A but it is not a balanced design since it ignores the pitting and scoring problems that will be encountered in service.

The procedure was expanded to include the effect of nonstandard unequal addenda gearing shown in Fig. 6 [14]. Unequal addenda gearing makes it possible to reduce the size (number of teeth) for the same ratio as standard gears without running into kinematic interference. Unequal addenda gearing is shown in Fig. 6.

Conventional practice holds that unequal addendum geometry is better than standard geometry because the short addendum pinion teeth are stronger in bending fatigue than the smaller standard teeth. The research results show that for minimum center distance gear sets (which meet the design limits on strength and kinematic interference) there is no appreciable size reduction. The results apply in general, since geometric similarity and strength similitude are maintained for the dimensionless result. The critical factor in sizing is the Hertzian contact stress.

This research clarifies the main attributes of long and short addendum gearing and extends work on minimum center distance gearing.

A comprehensive method was developed for the design of spur gears with improved efficiency over the full range of gear operating conditions [15]. Previously available methods were intended to provide an estimate of only full load efficiency. The new method was then utilized to show the effects of spur gear size, pitch, ratio, lubricant viscosity, face width, pitch-line-velocity, and load efficiency.

Fig. 7 shows the effect of given gear diameter, tooth number, and pitch-line-velocity on efficiency at a given load. This moderate to heavily loaded gear set is most efficient with fine-pitched, large diameter gears operating at high pitch-line velocities. With this method the gear designer can now design a gearset for optimum efficiency at any operating condition desired.

Fig. 8 shows the predicted power loss for three gears designed for the same application. Gear L is a standard gear while K and M are both high contact ratio gears, but of difference size. From the analysis based on gear design K, it is possible to design a high contact ratio gearset with an efficiency comparable to a standard gear design.

The finite element method is often used to do stress and deflection analysis of gears. A major difficulty in the use of the FEM programs is sizing the grid spacing in the region of the load. The effect of Hertzian deformation contributes up to 20 percent to the total deflection at the gear contact point. Research is reported in [16] that relates to choosing the FEM grid for this type of problem, in order to properly account for Hertzian deflections.

In [17] a study of the effect of rim thickness and fillet radius on gear stresses is reported. It was found that compressive stresses opposite the loading side of the tooth are most sensitive to rim thickness. Partially supported rim gears such as in lightweight aircraft applications have a decrease in the stress with increase in rim thickness, whereas for fully supported rim gears, the opposite is true. The root stresses increased with decreasing fillet radius.

Dynamic Analysis

A high contact ratio gear dynamic analysis was developed to determine the dynamic loads, stresses, and deflections for spur gears [18]. The analysis determines the effect on the gear tooth dynamics with various tooth profile modifications, tooth spacing errors, system mass, and system damping.

The analysis was first developed for internal and external high contact ratio and standard spur gears [19]. The analysis was expanded to include multiple gear meshes and planetary gears with up to 20 planets [20]. A computer code was developed that determines the gear dynamic loads, stresses, and deflections. The code plots the results for both single tooth mesh and several teeth in series to show the effect of tooth spacing errors. The program is being expanded to include helical gears.

Spiral Bevel Gear Design

The surface geometry of circular cut spiral bevel gears was developed [21,22]. Earlier work was done for the "ideal" case of a logarithmic spiral shaped involute tooth [23]. The work is complimentary to the earlier work in that it addresses the problem of circular cut spiral bevel gears which are practical to make on existing production machines.

The emphasis on the tooth surface analysis is on determining the principal radii of curvature of the surface. The principal radii, which are the maximum and minimum radii at the point of contact between mating gear teeth are needed to calculate contact stress, tooth contact patterns shown in Fig. 9, and the elastohydrodynamic lubricant film thickness. The formulae and procedures are general and well suited to use in computer algorithms and specific results are easily obtained by symbolic manipulative computer programs. The work specifically considers involute, straight, and hyperbolic cutter profiles. The results may be used in analysis of gear sets such as in the input stage of helicopter main rotor gearboxes and tail rotor gearboxes.

Further work was performed to describe spiral bevel gear sets with two different mesh contact patterns [24]. Two different methods give tooth contact paths in different directions. One path is across the profile direction of the tooth, the other is along the length of the tooth. Each contact path offers certain advantages for increased life, better lubrication, and reduced noise and vibration. The methods described in [24] indicate approaches to take in further analysis of the implications of two different geometries.

Gear Materials

Several gear materials have been evaluated for endurance life on the NASA LeRC Spur Gear Fatigue Tester [25]. A comparison of the life of various gear materials is shown in Fig. 10. The Vasco X-2 has a life statistically equivalent to AISI 9310 but has less fracture toughness. The CBS 600 material is somewhat better than AISI 9310 but also has less fracture toughness. The best material tested to date is the EX-53 material which has twice the life of AISI 9310 and an equivalent fracture toughness. Shot peening of AISI 9310 gears gave a life improvement of 60 percent over the standard gears without shot peening [26]. This improvement in life is attributed to the subsurface compressive residual stress induced by shot peening.

Gear Lubrication

An analysis for into mesh oil jet lubrication was performed with an arbitrary offset and inclination angle from the pitch point for the cases when the oil jet velocities are less than, equal to, or greater than gear pitch line velocity [27,28]. Equations were developed for minimum and maximum (optimum) oil jet impingement depths. The analysis includes the minimum oil jet velocity required to impinge on the gear or pinion and the optimum oil jet velocity required to obtain the maximum impingement depth. The best lubrication and cooling is obtained with maximum impingement depth when the oil jet velocity equals the gear pitch line velocity. Less than optimum gear lubrication and cooling is obtained at oil jet velocities that are less than or greater than the pitch line velocity. In addition, the pinion may be completely missed at jet velocities that are much lower or much higher than the pitch-line velocity. These analyses allows the designer to locate the oil jets and optimize lubrication flow and volume for maximum efficiency and, hence, lower heat generation.

An analysis and computer program called TELSGE were developed to predict the variations of dynamic load, surface temperature, and lubricant (elastohydrodynamic) film thickness along the contacting path during the engagement of a pair of involute spur gears [29]. The analysis of dynamic load includes the effect of gear inertia, the effect of load sharing of adjacent teeth, and the effect of variable tooth stiffnesses which are obtained by a finite-element method. Results obtained from TELSGE for the dynamic load distributions along the contacting path for various speeds of a pair of test gears show high loads near the pitch line where pitting failures are observed experimentally. Effects of damping ratio, contact ratio, tip relief, and tooth error on the dynamic load can be examined. A lubricant film thickness analysis for the OH-58 transmission sun-pinion gear set is shown in Fig. 11.

Gear Noise

Gear noise in helicopter transmissions is a major contributor to the overall noise inside the passenger areas of most helicopters. Gear noise is a direct result of the deviations in the gear tooth profiles from the true involute form (or in the generalized case, the true conjugate form). It is a matter of necessity that the gear profile be altered from the ideal (mathematically) conjugate form. This is to provide a compensation effect to allow for deflections in the gear support and the gear teeth themselves which are caused by the normal operating loads. The essence of noise minimization is to balance the noise producing negative effects of nonconjugacy with the positive effects of desensitizing the gear system to the effects of deflections which can cause gear misalignments and eccentricities. It is clear that extensive system modelling of the gear tooth action, gear support stiffnesses, and dynamic behavior is required in order to design gears with the appropriate noise compensating tooth profiles.

One such approach has been completed [30]. A transfer function method for predicting the dynamic responses of gear systems with several meshes was developed. The model was applied to the NASA spur gear fatigue test apparatus. An optimum profile modification design method was developed and applied to the NASA gear rig. The profile modification chart is shown in Fig. 12. The NASA test gear is shown in Fig. 13. Noise tests and fatigue tests to evaluate the performance of the minimum noise design will begin in the near future.

Measurement of the noise with conventional microphones to measure sound pressure levels is difficult for most gearing installations because of the reverberant conditions of the room surrounding the gear installations. Greater success can be expected from measurements obtained via accelerometers mounted on the gearbox housing. However, the effect of the radiation mechanism that transfers vibrations into sound pressure levels is neglected.

A method that circumvents this problem has been developed [31]. The method is based on the measurement of acoustic intensity by two closely spaced microphones. A robotic acoustic intensity measurement system (RAIMS) was designed and constructed to automate the analysis of sound fields surrounding gearboxes in actual installations (Fig. 14). The acoustic intensity measurement method does not depend on having an anechoic chamber around the gearbox and may be used in the reverberant, multisource noise environment found in a typical test cell. The RAIMS system will be used in the NASA helicopter transmission test cells and in the NASA spur gear test rig for the minimum noise gears.

Noise is related to the kinematic precision of gear trains. A theory for the kinematic precision of gear trains was developed [32,33]. The kinematic accuracy defines the actual ratio of input speed to output speed for every instant in time (fig. 15). The small deviations from the ideal or steady ratio are the source of high noise levels and large dynamic loads.

The theory defines kinematic accuracy as a function of machine settings used in the gear grinding process and as a function of gear eccentricity and deflections. There are two principles that are used to derive the theory. The first is that the vector surface normals of the gear teeth must coincide, and the second is that the vector velocity of the contact point on each gear tooth must coincide (Fig. 16).

Rolling-Element Bearings

The input pinion of main helicopter transmission is typically supported on stacks of angular contact ball bearings or combinations of ball and cylindrical roller bearings (Fig. 17). The use of tapered-roller bearings for this application should eliminate the problems of load sharing and lubrication associated with stacked bearing assemblies. Additionally, tapered-roller bearings are ideally suited for the large combined radial and thrust loads from the input bevel pinion.

Speed limitations on standard tapered-roller bearings demand that suitable modifications and careful lubrication be applied for successful operation in high-speed pinion applications. In this program, the performance of commercially available tapered-roller bearings, modified for high-speed operation, was verified both analytically and experimentally. The bearing design and the arrangement of the pinion shaft were selected by computer analysis. The bearing selection was verified experimentally and an optimum lubrication system and flow rates were determined [34].

The experimental activity established that automotive pinion quality tapered-roller bearings are capable of reliable operation under load and speed conditions anticipated in an advanced helicopter transmission. The conditions of the tests (100 percent power) were a shaft speed of 36,000 rpm (1.3 million DN), a thrust load of 1583 pounds (7041 N), and a radial load of 723 pounds (3216 N). (DN is a speed parameter equal to the bore of the bearing in millimeters multiplied by the shaft speed in revolutions per minute.) The computer analysis accounted for thermal and mechanical interactions of the bearings and their environment. The predicted life of the selected 33 mm bore tapered-roller bearing at 60 percent prorated load conditions was in excess of the desired 2500 hours.

The design and lubrication of large bore (4.75 in.) tapered-roller bearings for operation at speeds up to 2.4 million DN under combined radial and thrust loads has been demonstrated [35-37]. The bearing design was computer optimized for high-speed operation. Lubricant was supplied to the bearing through the shaft and directly to both the large end and the small end of the rollers.

The advanced high-speed bearing ran with less heat generation and ran cooler than the baseline bearing to which it was compared as shown in Fig. 18. It also was capable of higher speed operation; 20,000 rpm as opposed to the 15,000 rpm limit on the baseline design bearing. Four of the advanced design bearings made of CBS-1000M material ran to 24 times rated catalog life without failure.

The high-speed bearing was designed for lower stress and heat generation in the critical contact of the roller large end and the cone rib. The baseline bearing was only modified to supply lubricant to this critical contact.

Predictions by the computer program CYBEAN, for cylindrical roller bearing analysis, have been verified with experimental data [38,39]. The experimental verification was conducted with 118-mm bore cylindrical roller bearings at speeds up to 25,500 rpm. Calculated bearing temperatures and heat generation agree very well with the experimental data. The program also calculates roller dynamics and bearing life considering lubrication and thermal effects. CYBEAN is a valuable tool for the design and analysis of cylindrical roller bearings for difficult and critical applications.

Spherical roller bearing analysis predictions by computer program SPHERBEAN have been verified by experimental data [40]. The program calculates roller dynamics, heat generation, temperature, and bearing life. It has capability to simulate performance of a planet bearing in planetary gear systems.

Experimental verification was conducted with 40-mm bore, double-row, spherical roller bearings at speeds up to 19,000 rpm. Predicted temperatures correlated well with experimental measurements. Predicted trends in temperature with bearing geometry changes were consistent with experimental observations.

The usefulness of SPHERBEAN was demonstrated by its ability to accurately simulate spherical roller bearing performance in conventional and high-speed ranges. The program can assist in the design and analysis of spherical roller bearings for difficult applications such as the planetary stage of a helicopter main transmission or in geared fan or turboprop speed reduction units.

Transmissions

Transmission Evaluation

The data in the open literature defining current state-of-the-art transmission technology, supported by tests under carefully controlled conditions, are virtually nonexistent. If changes are to be made in gear and bearing technology as applied to transmission systems, the effect of this technology must be assessed. Hence, the operating parameters of current state-of-the-art transmissions must be evaluated. This would allow improvements in components and new transmission concepts to be quantified with respect to noise, vibration, efficiency, stresses, and thermal gradients.

Four state-of-the-art transmissions are being evaluated: (a) the 317-hp OH-58 three planet gear transmission, (b) the 317-hp OH-58 four planet gear transmission, (c) the 3000-hp Sikorsky UH-60A transmission, and (d) the 3000-hp Boeing UTTAS transmission. In addition, three advanced geared transmission concepts are being investigated under this program: (a) advanced components transmission (Fig. 19), (b) bearingless planetary transmission (Fig. 20), and (c) split-torque transmission (Fig. 21).

These transmissions are being evaluated on the NASA Lewis Research Center's 500-hp (Fig. 22) and 3000-hp (Fig. 23) transmission facilities. These facilities are unique in that they can test both conventional geared and hybrid (traction/gear) transmissions. Initially, these facilities are being used to establish baseline information on transmissions designed using current state-of-the-art design techniques. Advanced analytical techniques which include those previously discussed will be used to examine the many parameters that affect the service life, efficiency, noise generation, and reliability of these transmissions. Further, the experimental results will be used either to verify or to modify existing theory and computer codes.

Some of the results of the evaluation of the efficiency and operating characteristics of the 317-hp, three planet, OH-58 transmission are shown in Figs. 24 and 25. Fig. 24 shows the baseband frequency spectrum of the OH-58 transmission showing the spiral bevel gear amplitude compared to the spur gear amplitude. The noise is traceable to tooth kinematical error. It is generally true that larger amplitudes of vibration occur at tooth mesh frequencies of the spiral bevel gears. This was determined from measurements on several different transmissions.

The effect of lubricant type on transmission efficiency was determined for 11 different lubricants in the OH 58 transmission [41]. In addition, six of the lubricants were tested in the NASA gear fatigue tester to determine the effect of these lubricants on gear life. Fig. 25 shows the results of these tests. Among the 11 different lubricants, the efficiency ranged from 98.3 to 98.8 percent, which is a 50-percent variation relative to the losses associated with the maximum efficiency measured. Of the six lubricants shown in Fig. 25, there was no correlation between efficiency and life. The lubricants had no significant effect on the vibration signature of the transmission. Additional work is being conducted to define the lubricant chemistry, additive package, and rheological and physical properties in order to determine which of them affect the results shown.

Transmission Concepts

Based on fundamental research performed in mechanical components, an advanced 500-hp transmission was designed and fabricated (Fig. 19). The concept is a high-contact-ratio four planet-gear transmission for improved load capacity and life. The high-contact-ratio gears are expected to result in lower noise and reduced dynamic loads. The main bevel gear has been straddle-mounted to improve deflection of the gear mounting, thereby improving load sharing in the gear mesh. This, too, is expected to result in lower noise and improved life. The planetary ring gear has been cantilever-mounted to relieve problems inherent in the ring-gear-to-case spline interface. Rolling-element bearings will be manufactured from vacuum-induction-melted, vacuum-arc-remelted (VIM-VAR) AISI M-50 material. The VIM-VAR AISI M-50 will result in longer bearing life. The bevel gear set was manufactured from VIM-VAR AISI 9310. The lubrication system is the latest technology of positive radial-jet lubrication to the sun gear and spline [42]. This will reduce wear and increase the load-carrying capacity of the gear set.

This advanced transmission which weighs 144 lb has a weight-to-power ratio of 0.29 lb of transmission weight per horsepower, as compared with the standard 120 lb 317-hp OH-58 transmission of 0.38 lb/hp. Preliminary tests have been conducted with the transmission on the 500-hp test stand.

The transmission design has been modified to allow for the replacement of the ball bearings with tapered-roller bearings. Tapered-roller bearings on the output and input transmission shafts offer greater load capacity and longer life than the ball bearings.

The self-aligning bearingless planetary transmission (Fig. 20) covers a variety of planetary-gear configurations, which share the common characteristic that the planet carrier, or spider, is eliminated, as are conventional planet-mounted bearings. The bearings are eliminated by load balancing the gears, which are separated in the axial direction. All forces and reactions are transmitted through the gear meshes and contained by simple rolling rings. The concept was first demonstrated by Curtis Wright Corp. under sponsorship of the U.S. Army Aviation Research and Development Command [43]. The 500-hp bearingless plan etary transmission for the NASA program is being designed to be comparable with the OH-58 baseline transmission. The transmission weight-to-power ratio is approximately 0.27 lb/hp.

A means to decrease the weight-to-power ratio of a transmission or to decrease the unit stress of gear teeth is by load sharing through multiple power paths. This concept is referred to as the split-torque transmission [4]. Feasibility studies were conducted on two variants of this concept (Fig. 21).

The first variant is in the 500-hp range with a single-engine input (Fig. 21(a)); the second is in the 3000-hp range with a two-engine input (Fig. 21(b)). Instead of a planetary-gear arrangement, the input power is split into two or more power paths and recombined in a bull gear to the output power (rotor) shaft.

Preliminary weight estimates of the split-torque concept indicate that the weight-to-power ratio is approximately 0.24 lb/hp. This concept appears to offer weight advantages over conventional planetary concepts without using high-contact-ratio gearing. The effects of incorporating high-contact-ratio gearing into the split-torque concept is expected to further reduce transmission weight.

A remedy to the speed-ratio and planet number limitations of simple, single-row planetary systems was devised by A.L. Nasvytis [44]. His drive system used the sun and ring-roller of the simple planetary traction drive, but replaced the single row of equal diameter planet-rollers with two or more rows of stepped, or dual diameter, planets. With this new multiroller arrangement, practical speed ratios of 250 to 1 could be obtained in a single stage with three planet rows. Furthermore, the number of planets carrying the load in parallel could be greatly increased for a given ratio. This resulted in a significant reduction in individual roller contact loading with a corresponding improvement in torque capacity and fatigue life.

To further reduce the size and the weight of the drive for helicopter transmission applications, NASA incorporated with the second row of rollers, pinion gears in contact with a ring gear (Fig. 26). The ring gear is connected through a spider to the output rotor shaft. The number of planet-roller rows 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. The traction-gear combination is referred to as the hybrid transmission.

Preliminary tests were conducted with a 500-hp low ratio variant of the hybrid transmission (Fig. 26). The transmission, which has a weight-to-power ratio of 0.27 and a speed reduction ratio of 17:1, could retrofit the OH-58 helicopter. The second variant of the hybrid transmission is referred to as the 500-hp high-ratio variant. This transmission has a speed reduction of 101:1. the low-ratio hybrid is designed for a speed input of approximately 6020 rpm, and the high-ratio variant is designed for an input speed of approximately 36,000 rpm. Because the transmission can accommodate the higher input speed, the 40-lb, 6:1 reduction gear box on the engine can be eliminated. Hence, the power train weight-to-power ratio can be as low as 0.20 lb/hp.

Concluding Comments

The helicopter, more than any other contemporary aerospace or industrial innovation, has placed severe performance demands on power transmission components such as bearings and gears. Well-designed mechanical components, good materials, and lubrication systems which are integrated into helicopter drive train systems can make the difference between a helicopter's reliable, economic operation and failure. NASA transmission research is oriented either to advance the state-of-the-art in mechanical power transfer technology or to add to the fundamental body of knowledge belonging to bearings, gears, lubrication, rolling-element fatigue, life prediction, traction phenomena, and mechanical power transfer. Transmissions studied for application to helicopters in addition to the more conventional geared transmissions include hybrid (traction/gear), bearingless planetary, and split torque transmissions. Considerable amounts of work are required in these areas to establish the validity of analysis and computer codes developed to predict the performance, efficiency, life, and reliability of these transmissions. Real-time data recording, control and analysis for transmission testing are available for this purpose on the NASA 500-hp and 3000-hp helicopter transmission test stands. Both test stands are capable of testing conventional and traction type helicopter transmissions. Results of this research should provide the transmission designer with analytical tools to design for minimum weight and noise with maximum life and efficiency. In addition, the advantages and limitations of new and novel drive systems as well as the more conventional systems will be defined.

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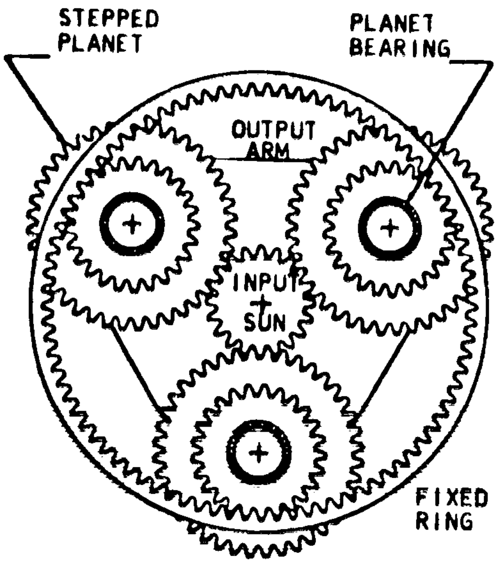

FIGURE 1. -COMPOUND PLANETARY GEAR TRAIN.

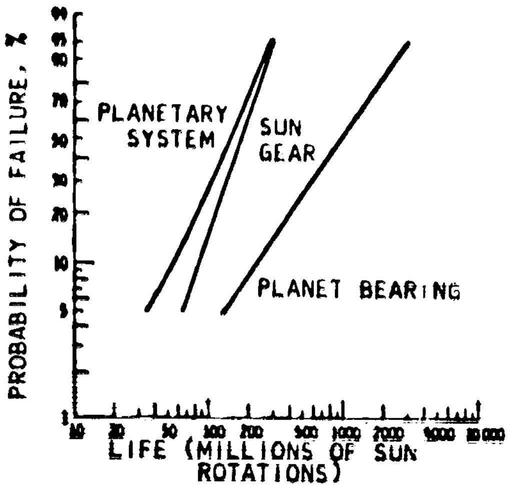

FIGURE 2. -WEIBULL DISTRIBUTIONS FOR SUN GEAR, PLANET BEARING, AND TRANSMISSION FOR A PLANETARY WITH BALANCED COMPONENTS.

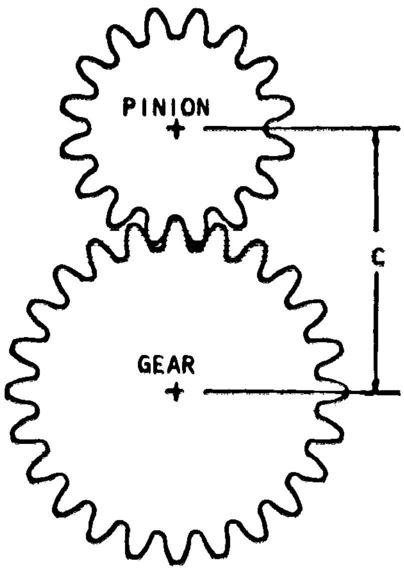

FIGURE 3. -EXTERNAL GEAR MESH.

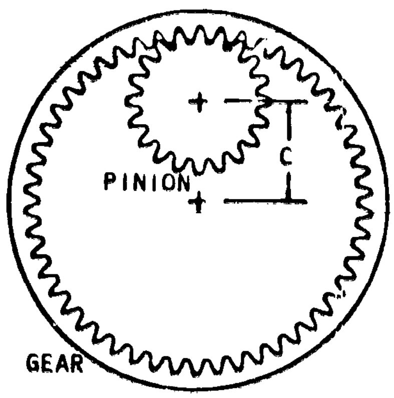

FIGURE 4. -INTERNAL GEAR MESH.

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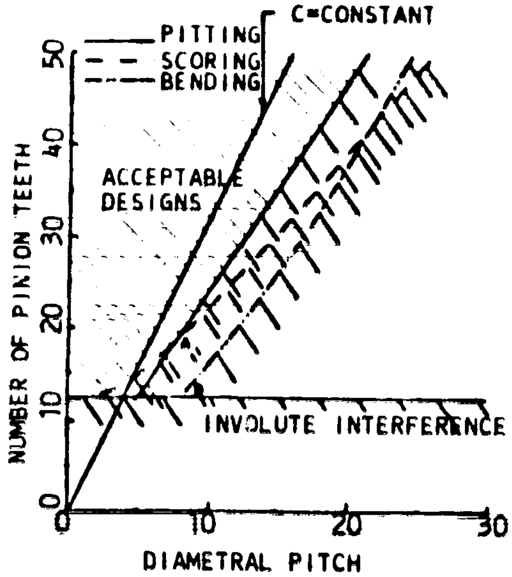

FIGURE 5. -DESIGN SPACE FOR MINIMUM CENTER DISTANCE SPACING IN SPUR GEARS.

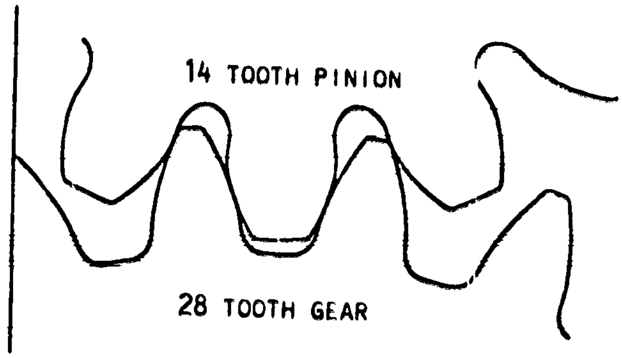

FIGURE 6. -UNEQUAL ADDENDA MESH GEOMETRY.

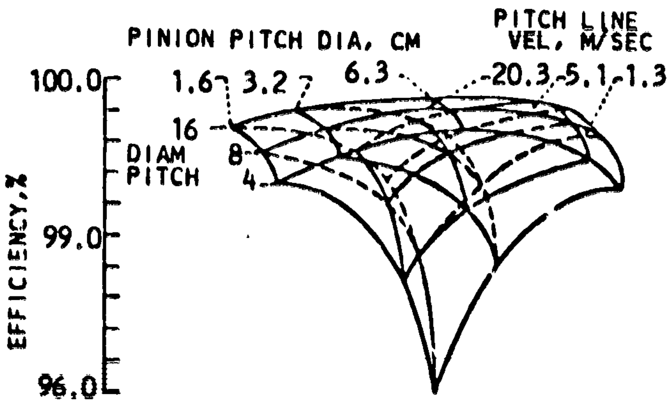

FIGURE 7. -EFFECT OF PINION DIAMETER, DIAMETRAL PITCH, AND PITCH LINE VELOCITY ON GEARSET EFFICIENCY AT A K-FACTOR OF 300; RATIO, 1.0; PRESSURE ANGLE, 20°; PINION WIDTH/DEPTH RATIO, 0.5; LUBRICANT VISCOSITY, 30 CP.

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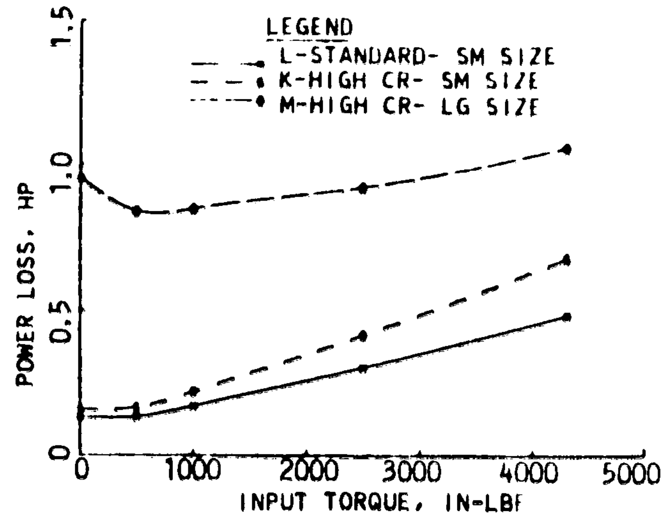

FIGURE 8. -POWER LOSS PREDICTION OF NORMAL & HIGH CONTACT RATIO SPUR GEARS.

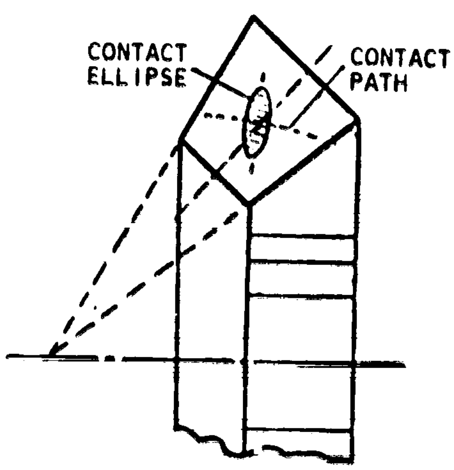

FIGURE 9. -TOOTH CONTACT PATH DESCRIPTION.

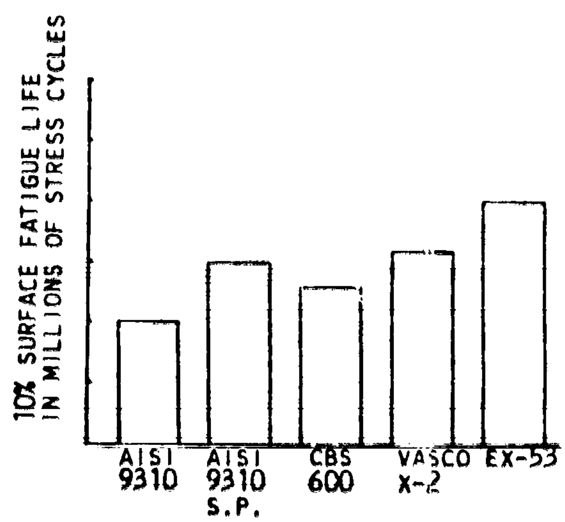

FIGURE 10. -FATIGUE LIFE FOR FOUR GEAR MATERIALS COMPARED TO BASE-LINE AISI 9310 MATERIAL.

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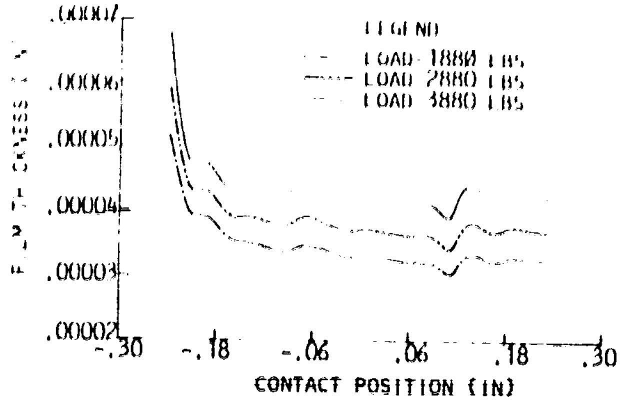

FIGURE 11. -OH-58 SUN & PLANETARY GEAR ANALYSIS, FILM THICKNESS VS. CONTACT POSITION.

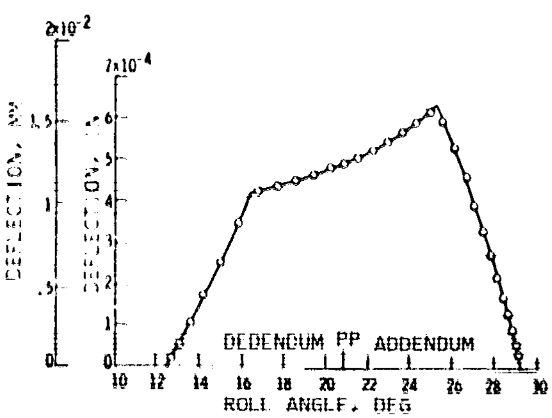

FIGURE 12. -PROFILE MODIFICATION FOR MINIMIZATION OF VIBRATION EXCITATION.

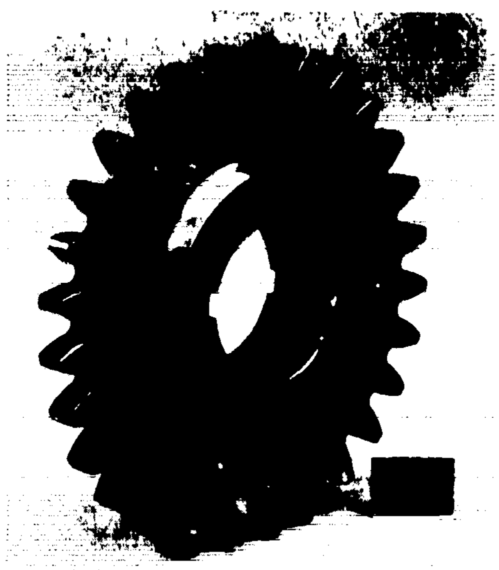

FIGURE 13. -NASA TEST SPUR GEAR, 3.5 INCH PD, 8 PITCH, 28 TEETH.

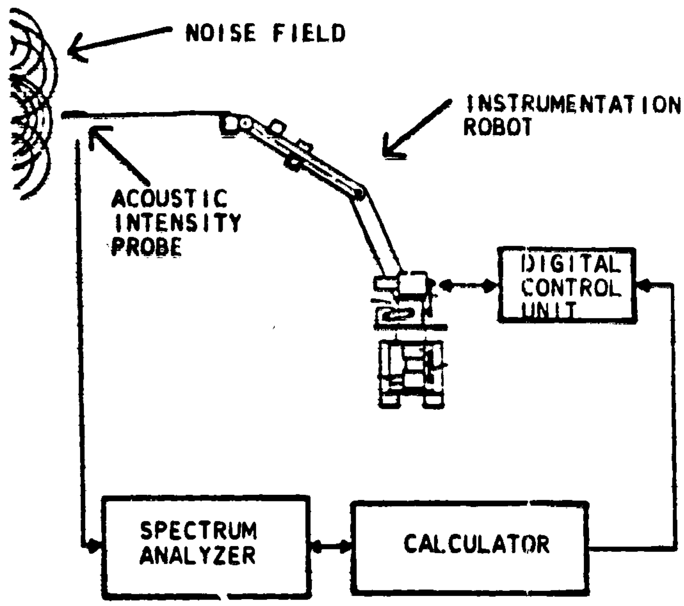

FIGURE 14. -SCHEMATIC OF RAIMS SYSTEM.

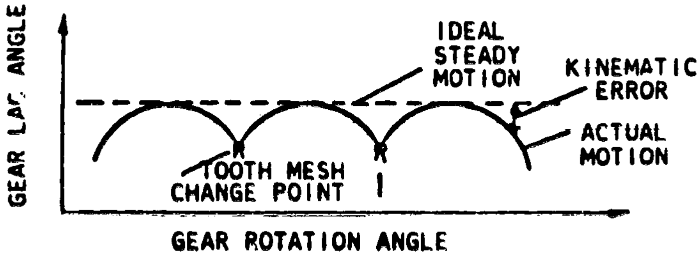

FIGURE 15. -KINEMATIC ERROR FUNCTION FOR THE MESH OF SEVERAL GEAR TEETH.

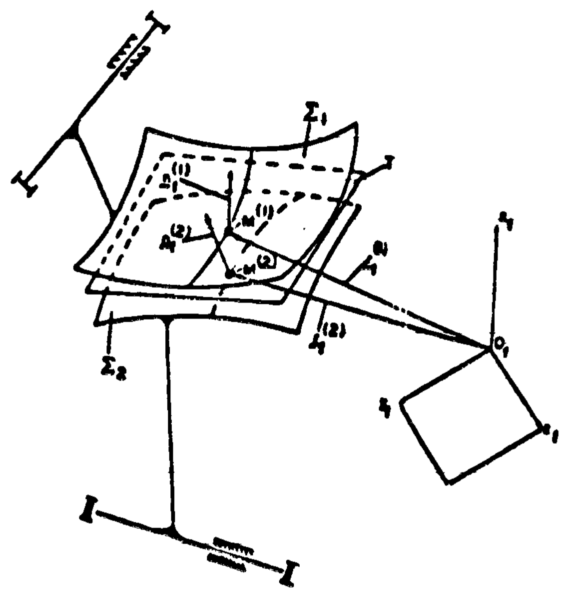

FIGURE 16. -TOOTH SURFACES WITH CLEARANCE INDUCED BY ERRORS.

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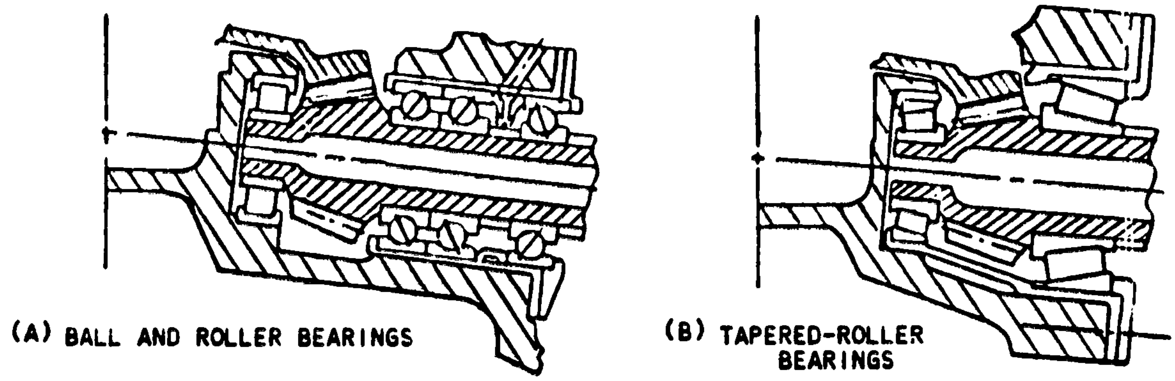

FIGURE 17. -TAPERED-ROLLER BEARINGS REPLACE BALL AND CYLINDRICAL ROLLER BEARINGS ON INPUT PINION FOR HELICOPTER TRNSMISSION.

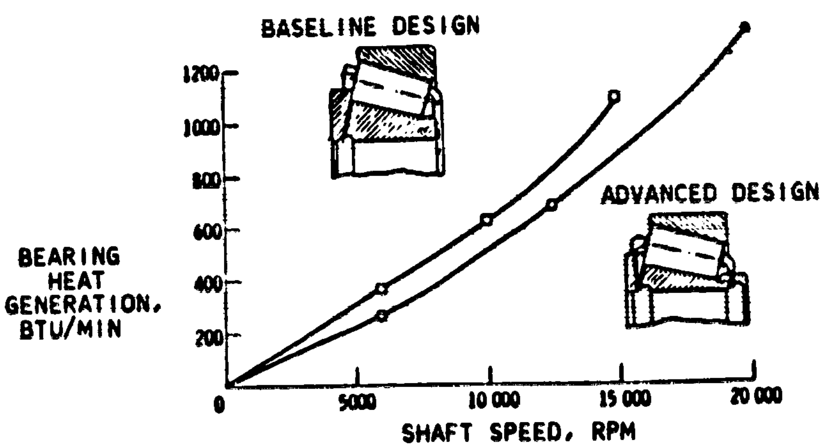

FIGURE 18. -IMPROVED PERFORMANCE OF ADVANCED HIGH-SPEED TAPERED ROLLER BEARINGS.

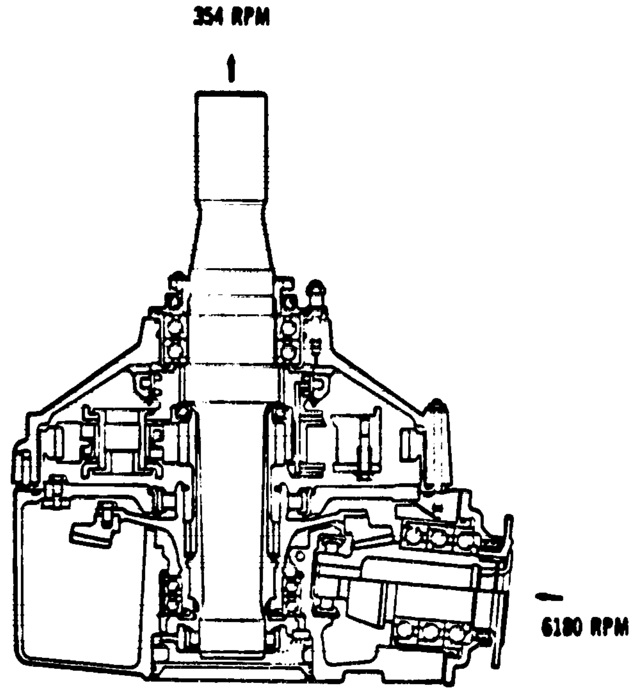

FIGURE 19. -500 HP ADVANCED COMPONENTS TRANSMISSION.

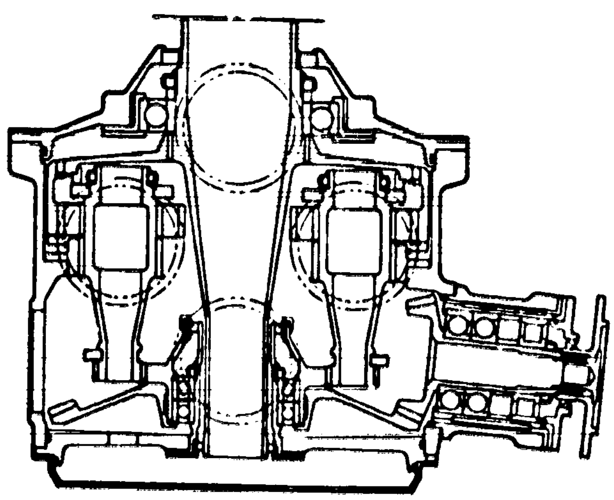

FIGURE 20. -SELF ALIGNING BEARINGLESS PLANETARY (LOW RATIO VERSION).

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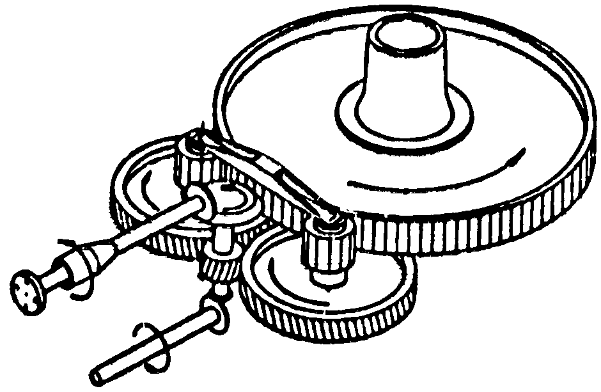
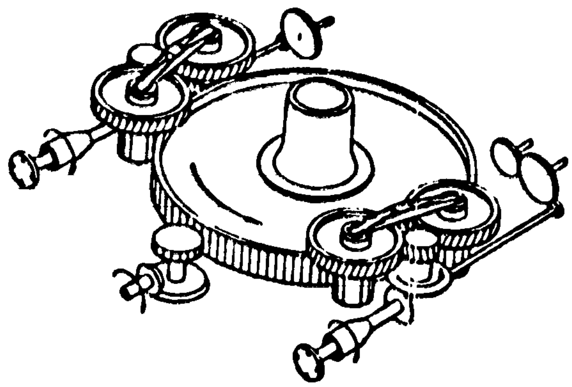


(A) SINGLE INPUT.



(B) DUAL INPUT.

FIGURE 21. -CONCEPTUAL SKETCH SPLIT-TORQUE TRANSMISSION.

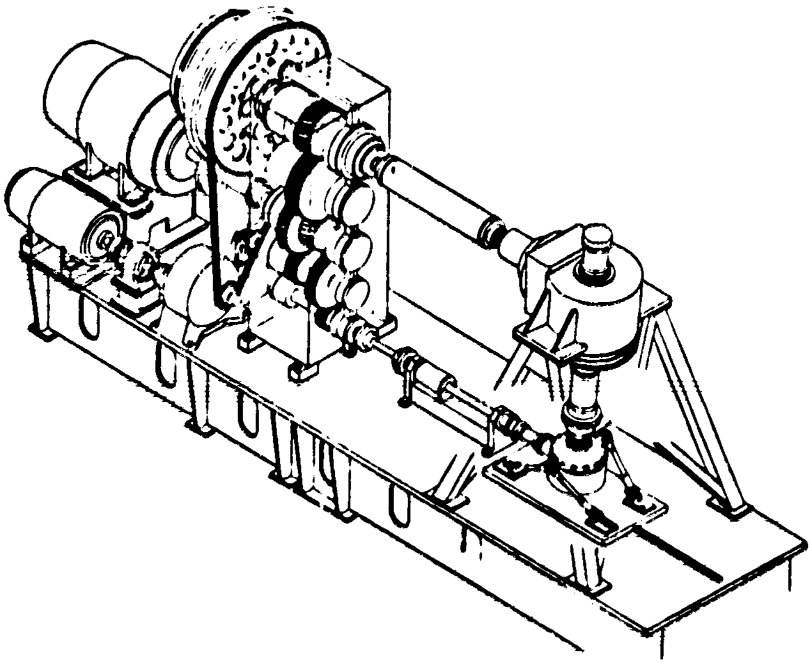


FIGURE 22. -NASA 500 HP HELICOPTER TRANSMISSION TEST STAND.

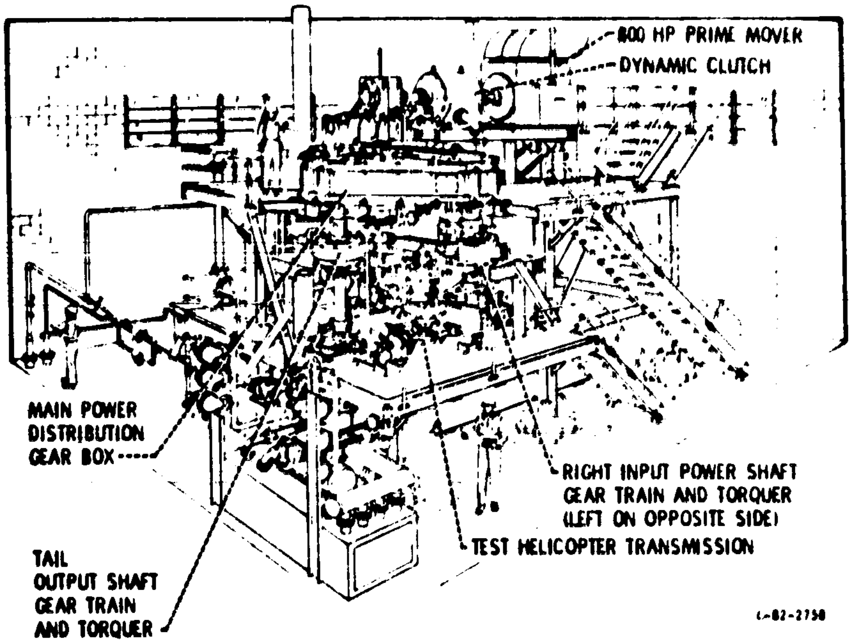


FIGURE 23. -NASA 3000 HP HELICOPTER TRANSMISSION TEST FACILITY.

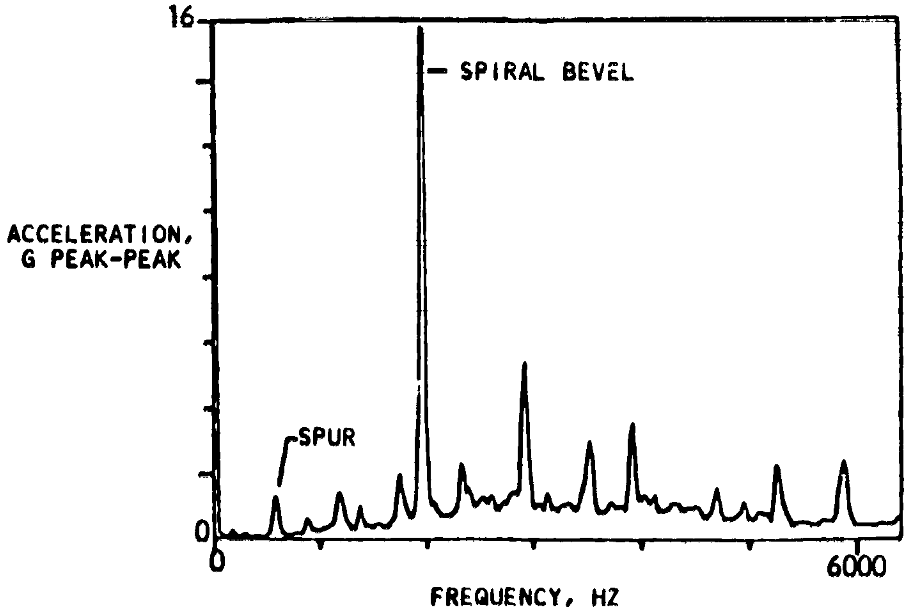


FIGURE 24. -BASEBAND FREQUENCY SPECTRUM SHOWING BEVEL AMPLITUDE COMPARED TO SPUR.

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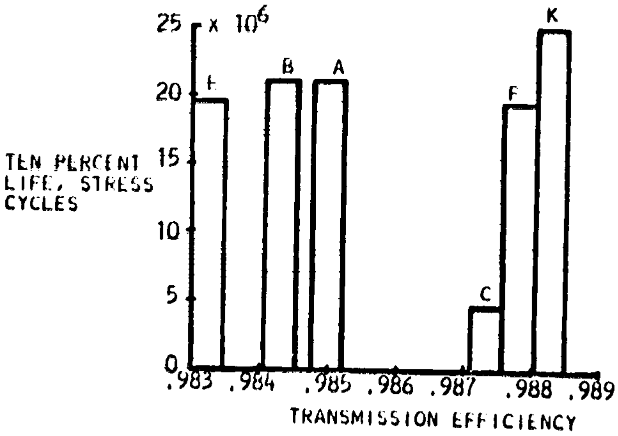

FIGURE 25. -LIFE FROM GEAR FATIGUE TESTS COMPARED TO TRANSMISSION EFFICIENCY FOR SIX LUBRICANTS.

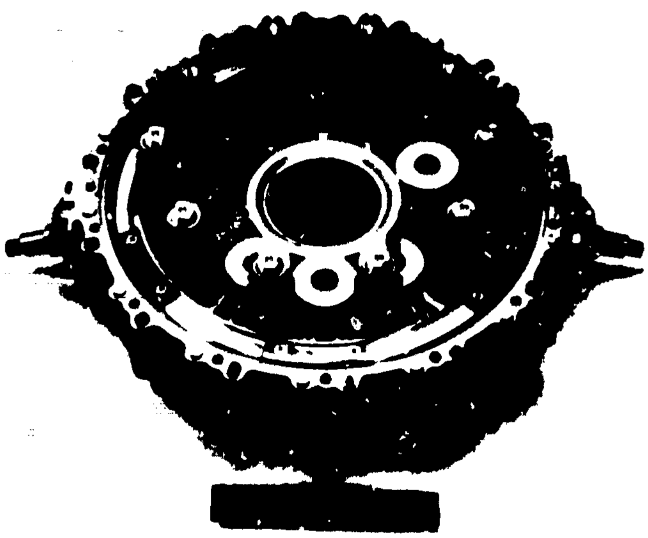

FIGURE 26. -500 HP HYBRID HELICOPTER TRANSMISSION.