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Vibration Measurements on Planetary Gears of Aircraft Turbine Engines

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Planetary reduction gear stages are an efficient, compact, and lightweight means of speed reduction and are, therefore, used in many aircraft turbine engines. In the PT6 engines, of which more than 13,000 have been produced, the planetary reduction gearboxes allow a speed reduction from more than 30,000 to about 6000 rpm in turboshaft and 1210-2200 rpm in turboprop applications. These gearboxes have been developed to a high level of reliability. The continuing demand for uprated or new higher powered designs requires a good understanding of the design factors that play a role in the dynamic gear loads and motions. The theoretical dynamic analysis of a planetary gear stage is quite complex due to the multiple, nonlinear gear meshes. Therefore, the development of these gearboxes depends to a high degree upon engine running experience and detailed dynamic measurements. A variety of dynamic measurements taken on PT6 reduction gearboxes over a number of years is reviewed. Peculiar behavior found in these tests is discussed, such as load sharing among planets, responses due to gear errors, and a dynamic instability.

Introduction

THE application of high speed turbine engines in propeller aircraft requires reduction gearboxes with high speed ratios in compact, lightweight designs. Planetary gear stages have proved to be excellent choices for this purpose. The turboprop models in the PT6 class of engines, with a range of 550-1170 shp (410-875 kW), have two planetary reduction gear stages. These engines, which have now been in production for more than 15 years, are used primarily in general aviation, most often in twin-engine aircraft.

The power turbine speed is 30,000 rpm or higher while the propeller speed at rated power varies between 1210 and 2200 rpm. In more recent applications where low noise levels are desired, the propeller speed is low. The four-engine De Havilland Dash-7 has a propeller speed at cruise of about 1000 rpm.

The general layout of the PT6, in this case the A-27, is shown in Fig. 1. This model has one power turbine stage and produces a maximum of 500 kW. Other models have two power turbine stages and have a higher maximum power, e.g., the A-50 has a maximum of about 840 kW.

Several PT6 models are also produced with a single planetary gear stage. These are turboshaft models and are used in installations where the output shaft speed is much higher, such as compressor and generator sets, marine or land transportation drive units, and in helicopter installations.

In all models reliable operation of the planetary gears over long periods of time is essential. The reliability of the gearbox of a new production engine is checked in part by the vibration level measured on the gearbox casing during the first run. A simple *g* level is used as a limit for production acceptance purposes. In order to allow a rapid identification of the parts that have caused the rejection of a build, and also in the interest of future design efforts, a deeper understanding of the vibration measurements is desirable. For this reason numerous gearbox vibration tests have been performed on PT6 models. It is the purpose of this paper to present some typical results of these tests and to review the significant aspects of the data.

Related Work

The vibrations of planetary gears have been the subject of many theoretical and a few experimental investigations. Some recent references, chosen to illustrate the wide range of interest, are given in the list of references. Additional useful references can be found in each of these papers.

Of course, many studies on the dynamics of simple gear trains are directly applicable to planetary gears. Examples are Refs. 1-3, in which are discussed respectively the dynamic factor of tooth loads, the meshing error function and its effect on the sidebands at gear mesh frequency in vibration spectra, and various noise sources. Theoretical dynamic analyses of planetary gears, concerning especially the dynamic load sharing between the planets, are presented in Refs. 4 and 5. Vibration aspects in particular types of planetary gears are discussed in Refs. 6 and 7.

The existence of resonances in the highly nonlinear planetary gear systems is difficult to establish. A linearized analysis of PT6 type planetary gears is given in Ref. 8. No strong evidence of the importance of resonances is available. Probably, these gear systems must be considered highly damped. This view is supported by the fact, noted in Ref. 9, that 3-5% of the maximum engine power is absorbed in the lubrication system, primarily in the cooling of the planet journal bearings and gears.

Gearbox vibration levels depend strongly upon the quality of the gears, in particular the transmission error in planetary gears. One component of the transmission error in planetary gears has been shown in Ref. 10 to be responsible for a significant part of the vibration. This part can be minimized by relative phasing of the planets.

PT6 Reduction Gearbox

The two planetary gear stages in PT6 turboprop engines are illustrated in Fig. 2. Only the features of interest in the dynamics of the system are shown. The first stage has three planets, a floating sun gear, and a spline-supported ring gear. The gears are spur gears. The ring gear spline is helical which allows the determiniation of the transmitted torque by means

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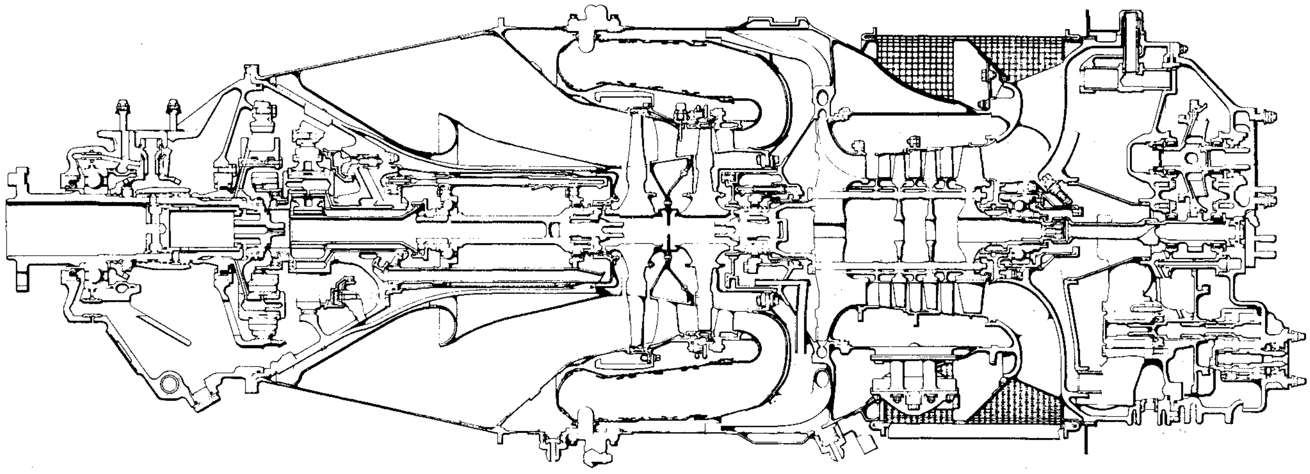

Fig. 1 Cross-section of PT6A-27.

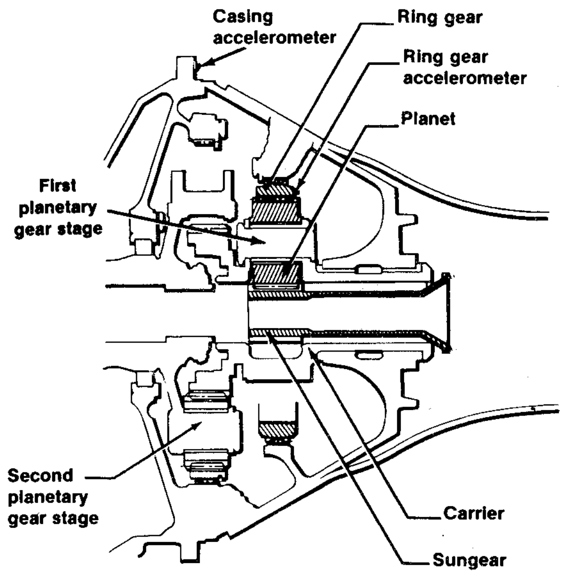

Fig. 2 Main components of reduction gearbox.

of the measurement of the axial force component. This is done with a hydraulic system which for simplicity is not shown in Fig. 2. The planets are supported by journal bearings on the planet pins which are fixed in the carrier. The carrier is supported on one side and drives the floating sun gear of the second stage on the other side. The second planetary stage is designed similarly. It will not be considered in this paper because its vibrations are generally less important and do not have peculiar features.

Typical PT6 first stage gear components are shown in Figs. 3 and 4. The carrier is in operating condition subject to a torque which twists the two carrier plates with respect to each other. As a result the planet pins are then not truly parallel to each other, which may cause axial misalignment of the gears. This static load case requires careful attention during the design phase.

Of all vibrations possible in the first gear stage only the in-plane effects will be considered in this paper. No evidence of any serious out-of-plane vibrations exists. It is the in-plane system of loads and deflections that transmits the power. Also, the main design problems occur in the in-plane system. These are how to ensure both load sharing among the planets and a minimum dynamic factor, and how to design planet journals with an adequate life under the highly demanding loading and thermal conditions to which they are exposed.

For a good appreciation of some of the measurements it should be noted that the gear teeth in these planetary gears are machined to a high degree of accuracy, case hardened, and honed. The gears are of classes 30 and 31 which have a tolerance in the pitch of 0.0002 in. and in the profile of 0.0001 in. This is of the order of the calculated deflection of a sun gear tooth at the pitch point under maximum load. The planet journal bearings have an extremely small minimum film thickness of the order of 0.0001 in. under maximum load. Because of the high stiffnesses of these bearings this is still a multiple of the actual bearing deflection under maximum load.

Some particulars of two types of first stage reduction gears used in four PT6 models are shown in Table 1. Planet rotation is given with respect to the carrier. The planet gears are meshing synchronously. In the A-27 and the A-41 there are nine groups of teeth in the gears, each group containing one-ninth of the total number of teeth. This follows from the common factor 9 in the tooth numbers. All teeth in each group mesh with each other but never with those in the other groups. In the A-45 and the A-50 there are only three such groups.

Table 1 Some characteristics of two types of first stage reduction gears

PT6 Model	A-27 A-41	A-45 A-50
Number of planets	3	3
Number of teeth		
Sungear	27	27
Planet	45	51
Ring gear	117	129
Frequency relative to sun		
Sungear rotation	1	1
Carrier rotation	0.1875	0.17307
Planet rotation	0.4875	0.4378
Gear mesh	21.9375	22.327
Period relative to gear mesh		
Gear mesh	1	1
Sungear rotation	21.9375	22.327
Carrier rotation	117	129
Planet rotation	45	51
Mesh repeat period		
Sun planet	135	459
Planet ring gear	585	2193
Sun planet ring gear	1755	19737

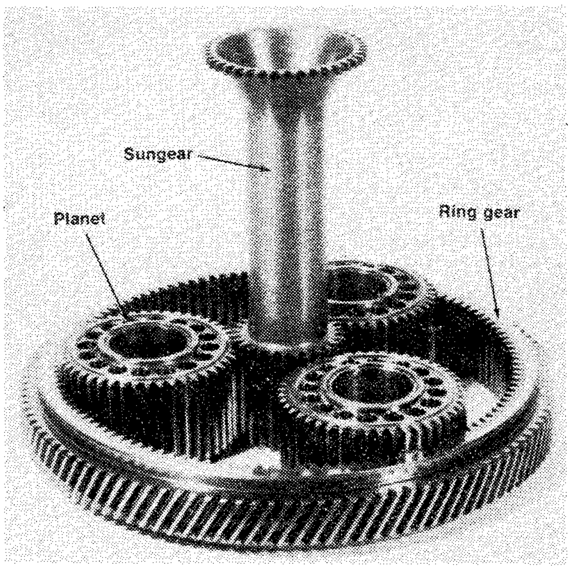

Fig. 3 First stage gear components.

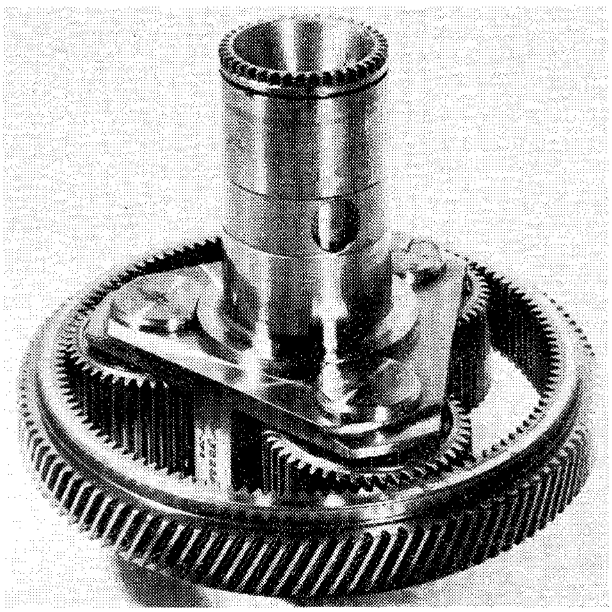

Fig. 4 First stage gear assembly.

Test Instrumentation

Tests have been performed on rotating and nonrotating gearbox components. The nonrotating ring gear and gearbox casing are easily accessible for measurements with strain gages and accelerometers. The casing accelerometer attached to the casing on the outside at the flange is the instrument that is always present in gearbox vibration measurements. It is the standard indicator of the vibrations in production engines. Other measurements on nonrotating parts, such as oil pressure and thermal measurements, have been performed, but these are not of direct interest in this paper.

Slip rings are the only practical means for measurements on high speed rotors when many channels are required. The slip ring assembly must be designed and installed carefully. Several propeller and gearbox parts have to be modified to accommodate such an arrangement. Sun gear vibratory stresses have been measured with strain gages at various locations on fillets between teeth (Fig. 5), on the shaft and near the spline. Sun gear and carrier motions have also been measured with variable inductance proximity probes supported from nonrotating parts.

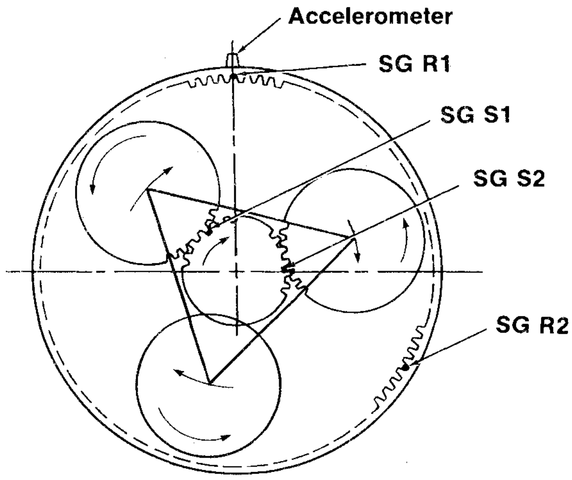

Fig. 5 Typical instrumentation.

In a typical test the data are recorded in a routine manner on multichannel FM magnetic tape and displayed selectively for the benefit of test personnel. Most data reduction after a test consists of replaying the tape and processing the data in digital form through special analysis equipment to the required format. This equipment can store a limited amount of simultaneous data from two channels so that simultaneous events, e.g., gear meshes, can be investigated in groups of two. Additional capability of recording data on digital tape is available for subsequent special data reduction on the digital computer.

Test Conditions

Most engine tests take place in test cells that are designed with sufficient attention to airflow conditions to satisfy overall power and performance requirements. In the case of PT6 engines the power absorber is a propeller or a dynamometer. The propeller has variable pitch for endurance or performance tests that must be run at a variety of speeds and power settings. Runs with the gearbox slip ring arrangement were possible only with a propeller with fixed pitch. Hence, in that configuration the freedom to investigate various power and speed combinations is more limited. In some cases this limitation was considered unacceptable and additional investigations were carried out with the slip ring arrangement in a wind tunnel. Special conditions that could be investigated there included low power settings at speeds which may occur in certain flight conditions. Tests on engines without a slip ring but with gearbox instrumentation on nonrotating parts have also been performed on test aircraft in actual flights.

Vibration Data

The vibration data collected over the years concern many different PT6 models and for each of these many different builds. The results shown here have been obtained on A-27, A-41, A-45, and A-50 models. For the purposes of this paper the differences in the vibration phenomena between the various models are not significant and can be ignored. The test results will be discussed in terms of four particular aspects: an apparent instability, unequal sidebands, load sharing between the planets, and minimization of gear error.

Vibration Instability

The usual measurement of vibration with an accelerometer on the gearbox casing is simply a single reading of the peak vibratory acceleration. This is convenient because it does not require complex data reduction equipment. In production acceptance tests the maximum value over the engine speed range is the only criterion. In more elaborate experimental tests the peak casing acceleration can be plotted directly as a

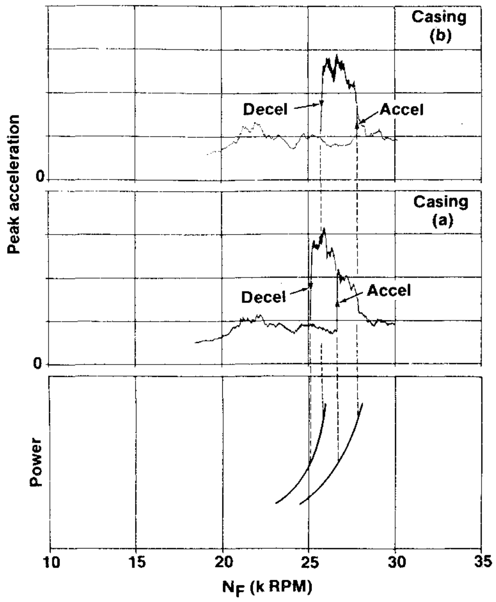

Fig. 6 Casing accelerometer vibration levels.

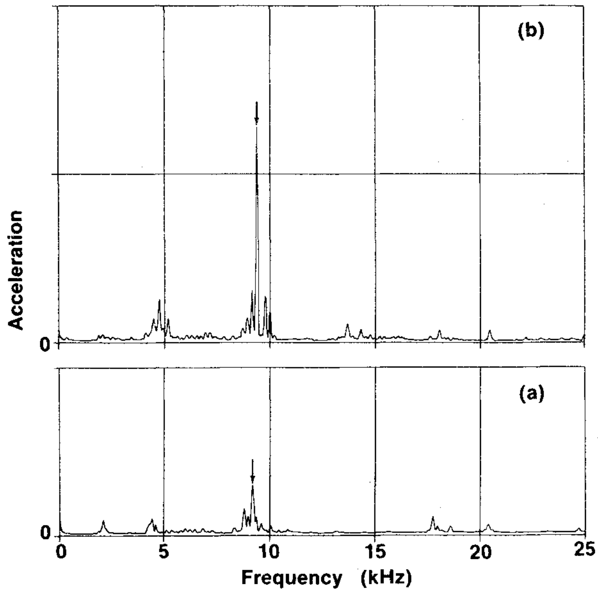

Fig. 8 Casing accelerometer vibration spectra.

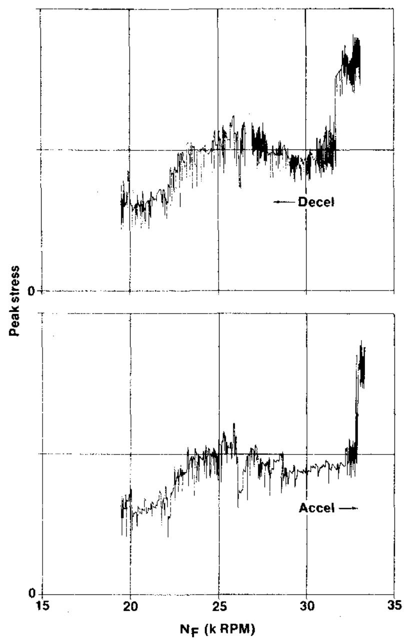

Fig. 7 Sun gear tooth stress levels.

function of speed (Fig. 6). It was noted that this graph occasionally showed a sharp increase at a certain speed, that this speed depended upon the torque level, and that this speed depended upon whether the engine speed was slowly increased (accel) or slowly decreased (decel). Graph (a) in Fig. 6 shows these differences for an engine running at relatively low power.

The curve marked decel matches the one marked accel perfectly except where shown. This indicates that the effect is not due to data processing, for example the processing of a nonstationary signal (accel too fast) as if it is a stationary signal. Graph (b) in Fig. 6 shows the effect for the same engine at higher power levels. In general the effect disappears at high power levels.

In order to determine whether the effect is felt by the first stage gear components, strain gage measurements available for another PT6 model were examined. The peak stress levels in a sun gear tooth fillet are shown in Fig. 7 for a slow accel and a slow decel at a low power level. The effect appears to be present and can be shown to depend on the power level as illustrated in Fig. 6.

A satisfactory explanation for this effect is not yet available. It is likely that it will be found once a complete dynamic analysis of the gear stage has been made, including the nonlinearity of the gear meshes and considering the independent motions of all planets. More tests would also be useful to measure the simultaneous behavior of all gear components at this speed. Since the instability is not known to affect the life of the gearbox adversely, a deeper investigation awaits a future research effort.

Unequal Sidebands

Spectrum analysis of vibration signals to isolate the frequency components is useful to determine sources or causes of vibration. Typical spectra of casing acceleration are shown in Fig. 8. The spectrum just below the instability is presented in (a) and that above in (b). The major peaks in both spectra at first stage gear mesh frequency are indicated with an arrow. In both graphs there are a number of groups of smaller peaks at second stage gear mesh frequency and its harmonics. The fourth harmonic of the second stage gear mesh frequency corresponds approximately to the first harmonic of the first stage mesh frequency. Disregarding these second stage effects it can be noted that the most significant difference between graphs (a) and (b) in Fig. 8 is the level at the first harmonic of the first stage mesh frequency and its sidebands.

The sidebands turn out in a more detailed analysis to occur at frequencies of gear mesh plus or minus multiples of the carrier rotation frequency. This is a well-known modulation phenomenon in the vibrations of planetary gears. In a simple harmonic analysis of modulated signals, the sidebands are always found to be symmetrical in amplitude with respect to the center frequency. In the many vibration spectra of casing accelerations measured on gearboxes which have been analyzed, the sidebands are rarely if ever symmetrical. Furthermore, the amplitude of one of the sidebands is often larger than that of the center frequency. This phenomenon is believed to be the result of the basic nonlinearity of the gear mesh. A satisfactory explanation is expected to follow from the complete dynamic analysis of the gear stage.

It may be noted in Fig. 8 that a relatively small change in the engine speed has a strong effect upon the amplitudes and the sidebands, not only in the first stage but also in the second stage. The harmonics of the second stage mesh frequency,

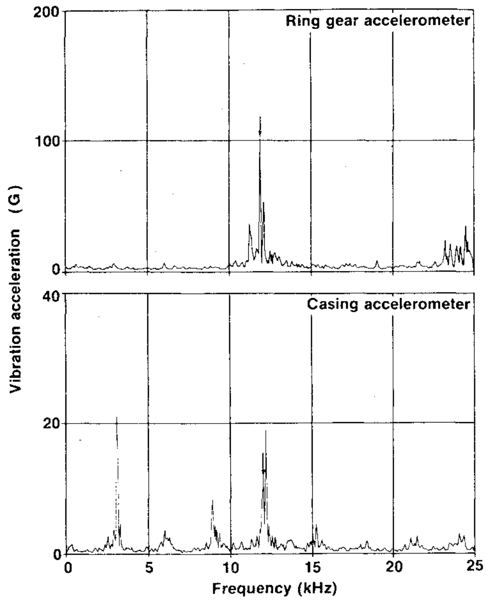

Fig. 9 Casing and ring gear vibrations.

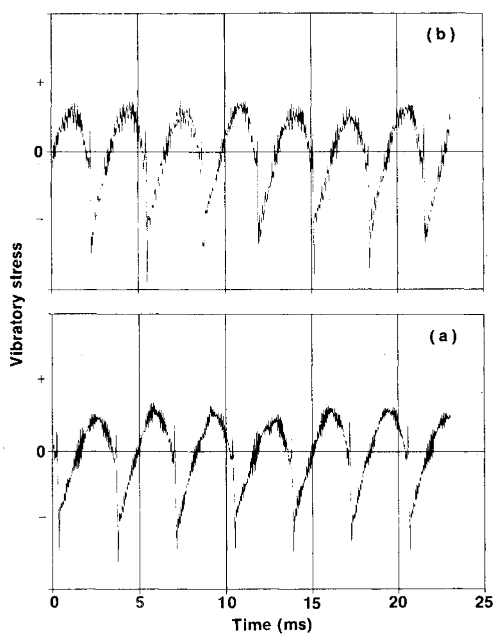

Fig. 10 Ring gear tooth stresses.

especially the second harmonic in graph (b), have changed significantly. An example of the spectrum of casing acceleration in another engine model is given in Fig. 9. The second stage frequency components at gear mesh and its harmonics are here more prominent. First stage mesh frequency (at the arrow) again has unequal sidebands. The spectrum of the acceleration measured directly on the first stage ring gear at the same engine conditions, but not simultaneously, is also shown. The second stage vibrations are virtually absent. The first stage mesh frequency and its sidebands are more pronounced but with different relative sideband levels. This may be due to small differences in engine speed between the two graphs. It indicates that spectrum peaks are sensitive indicators of the dynamic behavior of the gear stage components. It may be noted also that the second harmonic of the first stage mesh frequency is measured by the ring gear accelerometer and not in the casing. These are unusually high frequencies for mechanical systems, generally above 20 kHz, and are therefore subject to measurement limitations.

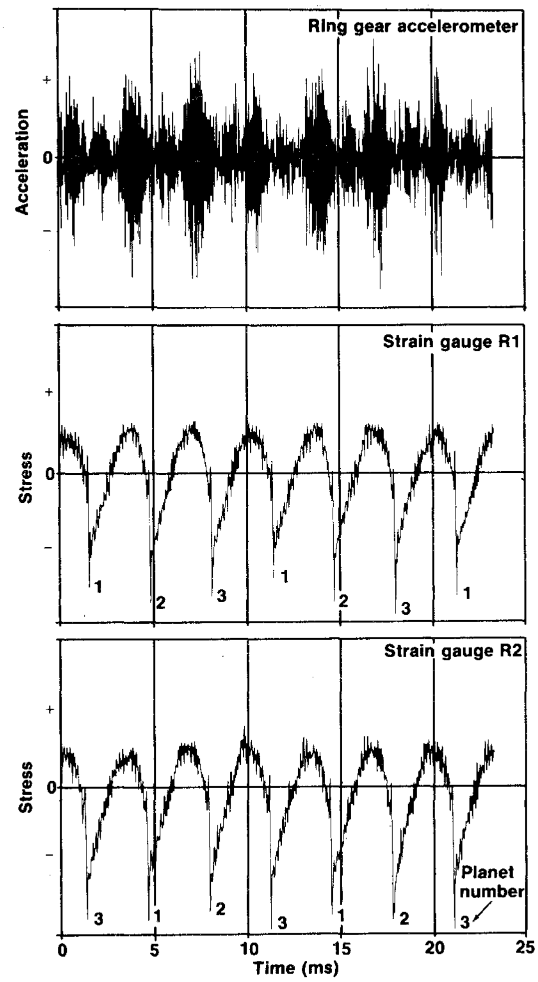

Fig. 11 Load sharing at ring gear.

Load Sharing between Planets

Load sharing between planets is one of the main objectives in the design of a planetary gear stage. Only by proper load sharing can the superior power transmission of planetary gear stages be realized. A good indication of load sharing can be obtained from the time histories of stresses measured in the root fillets of ring and sun gear teeth. The only true measure of load sharing is the simultaneous measurement of the loads on all planets. The next best indicator is the consecutive measurement at one tooth fillet of the loads caused by the passing of the planets. This is shown in Fig. 10 for a ring gear tooth fillet strain gage. The vertical lines represent the abrupt change from a tensile stress to a compressive stress when the planet tooth mesh moves from the ring gear tooth which precedes the strain gaged root fillet to the one following it.

In between two actual tooth meshes the strain gage measures ring gear bending caused by the loads on neighboring teeth. Note that the tensile stress due to bending is larger than that due to meshing. Also, maximum ring gear bending does not occur exactly halfway between two successive tooth meshes.

The variation in gear mesh loads for the different planets is evident in graph (a) of Fig. 10. The graph (b) is for the same strain gage but at a slightly higher speed, just above the instability. There are no remarkable differences between the two. In graph (b) the actual tooth loads appear to vary more widely than in (a), but those on neighboring teeth appear more regular than in (a) where they seem subject to modulation.

A record of simultaneous loads occurring on two ring gear strain gages which are 120 deg apart is presented in Fig. 11. Comparison of the maximum compressive stresses in the strain gages 1 and 2 shows that load sharing is subject to variations. In the case illustrated it appears that planet 1

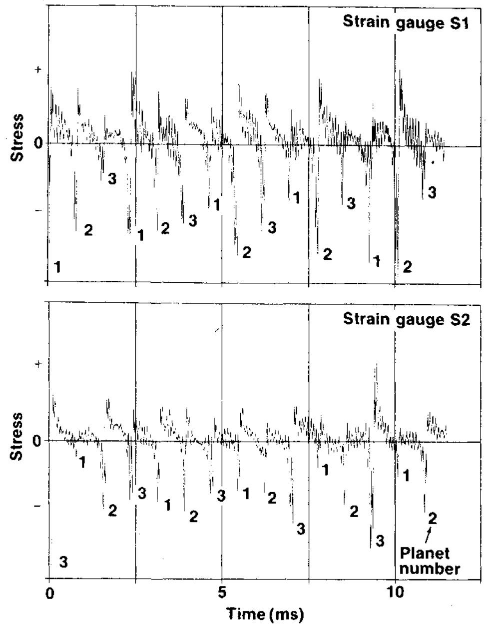

Fig. 12 Load sharing at sun gear.

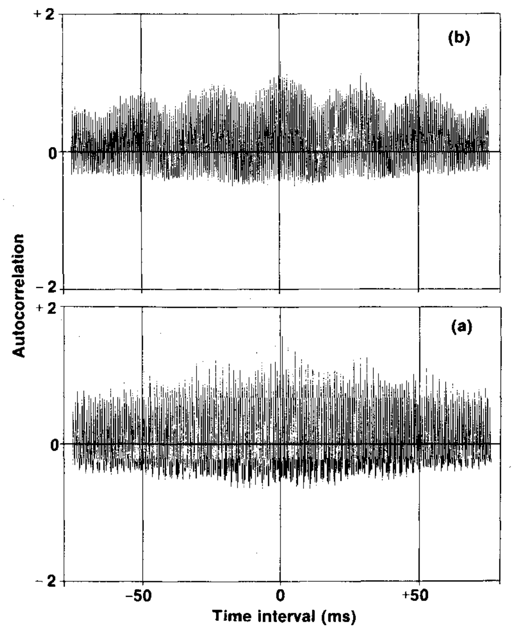

Fig. 13 Periodicity in sun gear stress.

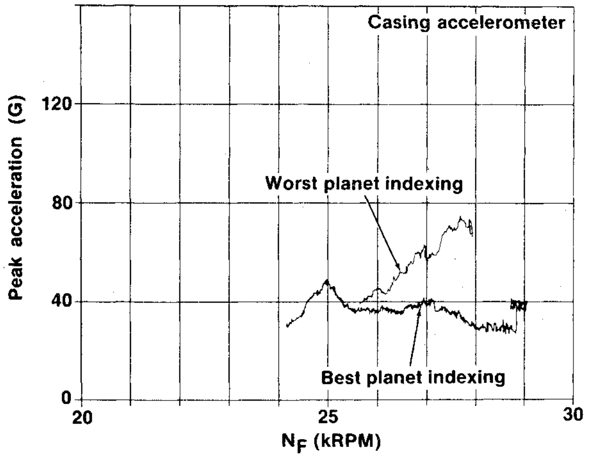

Fig. 14 Casing vibrations affected by indexing.

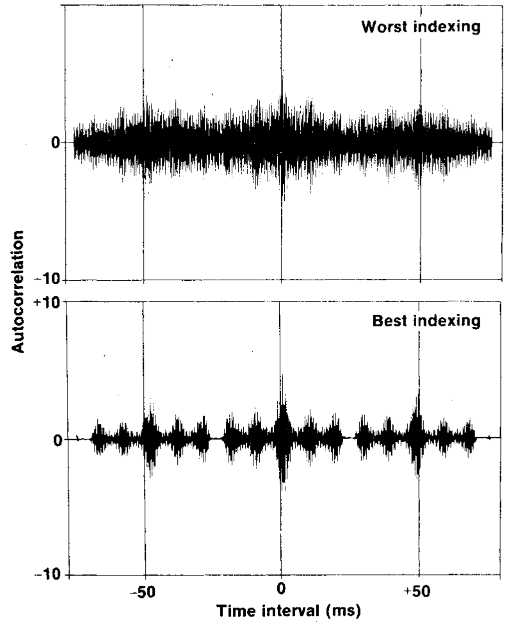

Fig. 15 Periodicity in casing acceleration.

carries more often a smaller load and planet 3 a higher load. A more detailed statistical analysis over a longer period of time is necessary to confirm such a conclusion.

The vibratory acceleration of the ring gear recorded simultaneously with the stresses is also shown in Fig. 11. This graph clearly shows the modulation of the gear mesh frequency at once and twice planet pass frequency, caused by the tooth loads of the passing planets and the ring gear bending in between.

Load sharing between planets at the sun gear meshes has also been measured with two strain gages at sun gear tooth fillets 120 deg apart. This is shown in Fig. 12. The stress at the mesh changes abruptly from compressive stress to tensile stress when the planet tooth mesh moves from the sun gear tooth which precedes the strain gaged root to the one following it. The variation in load sharing between the planets at the sun gear meshes is remarkably great. Clear evidence has been found in similar records of planets carrying occasionally no gear load at all. Note also that many meshes generate two high compressive or tensile peaks. Equal load sharing between planets has apparently not been achieved, particularly at the sun gear, at least for the engine conditions at which the records shown here were measured. The fact that nevertheless the gearbox is very reliable suggests that the load carrying capability of these designs could possibly be increased if load sharing could be ensured.

An attempt to look at the tooth loads in a statistical manner was made by producing the autocorrelation function of the sun gear tooth fillet stress. This would show periodicities over a longer period of time. Two examples are given in Fig. 13. Graph (a) shows some regularities. However, graph (b) made of the same strain gage signal at a slightly higher engine speed just above the instability shows a much more pronounced periodicity with a period close to the planet pass period. Comparison of graphs (a) and (b) suggests that the instability represents a realignment of the various gears such that sun gear meshing is more periodic, or less subject to tolerances and gear errors at the higher speed.

Minimization of Gear Error

Gearbox vibrations increase in general with an increase in the various manufacturing errors that may occur in the teeth. These errors are known collectively as the transmission error of the gear stage. One particular error of importance in the PT6 type of planetary gear stage is the first harmonic of the pitch error made in the machining of the planet teeth. The pitch of the teeth or the distance between neighboring teeth should be constant but turns out to have small variations. In the first harmonic of the pitch error these variations occur with an amplitude that varies sinusoidally around the planet. This type of variation will also occur in a planet with perfect teeth which rotates around a point slightly away from its geometric center. Generally, the pitch error is minimized in production hardware by strict quality control.

It is shown in Ref. 10 that planetary gears can tolerate a pitch error to some degree without increasing the vibration levels by properly indexing, or angular phasing, of the various planets. The pitch errors in the planets will then cancel each other. An example of the benefit to be gained by indexing is presented in Fig. 14, which shows two graphs of casing accelerations for two gearbox builds differing only in the relative indexing of the planets. It is obvious in this particular example that indexing is remarkably effective in reducing the vibration level. Another indication of this can be seen in Fig. 15, where the autocorrelation functions of the casing accelerations in the two cases at a particular speed are compared. After the most effective indexing was introduced the vibration autocorrelation function shows practically only periodicities due to the second stage which in this build had five planets.

Conclusion

The planetary gears in the reduction gearboxes of the various PT6 models share a reputation of reliability with the other engine components. From the viewpoint of the dynamics analyst the motions of the gears and the loads on the teeth and on the bearings are of great interest. These dynamic effects play an important role in the reliability of the parts and should be understood better for the benefit of future designs. The test results shown here indicate some of the areas where deeper investigations will be useful. These are the occurrence and nature of an apparent instability in the vibration level at certain speeds, the geometric causes of the amplitudes of the sidebands of the vibrations at first stage gear mesh frequency and in particular the usual lack of symmetry, the factors that will insure better load sharing between the planets, and the relationship between vibration levels and the various components of the transmission error.

It is clear that the experimental work performed so far has been of considerable help in understanding these dynamic effects. However, for a complete resolution of the problems it is necessary to perform the experiments in a more systematic manner with many more simultaneous measurements under well-controlled conditions. In addition, the experimental work must be guided by theoretical analyses which include all these effects. These analyses are inherently complex because of the large number of nonlinear gear meshes in a planetary gear stage. These efforts are in progress.

Acknowledgment

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