Identification and Correction of Damaging Resonances in Gear Drives

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Abstract

As a result of extensive research into the vibration characteristics of gear drives, a systematic approach has evolved, by which damaging resonances can be eliminated. The method combines finite element techniques with experimental signature and modal analyses. Implementation of the bulk of the method can be carried out early in the design stage.

A step-by-step description of the approach, as it was applied to an existing accessory drive, is given in the text. It is shown how premature bearing failures were eliminated by detuning the torsional oscillations of a gearshaft. A dramatic reduction in vibration levels was achieved as a result of detuning the problem gear.

The proposed approach can be extended to other types of rotating machines.

Introduction

During the endurance test of an Aircraft Mounted Accessory Drive, premature failure of the ball bearings on the starter shaft was discovered. The test, intended to last 8,000 hours without failure, was abruptly halted after only 900 hours of testing. The failed bearings were of the single row deep groove type with split riveted cages.

When the drive was disassembled and the failed bearings examined, the rivets were torn off and the cage was split open. Further analysis showed evidence of plastic deformation and pitting at the cage interface. Dimensional check of the bearing showed that load carrying elements were within tolerance and that no measurable defects existed. Also, no discoloration, such as due to lack of lubricant, was evident.

All results of the failure analysis were pointing towards vibration as being the cause of the cage failure. The fact that material was being upset on either side of the ball pockets, and the indications of pitting at the cage interface, could only have derived from ball repetitive hammering, thus inducing tensile stresses in the cage rivets. When sufficient number of loading cycles have been accumulated, and the fatigue limit of the rivet material has been reached, tensile failure of the rivets would ensue.

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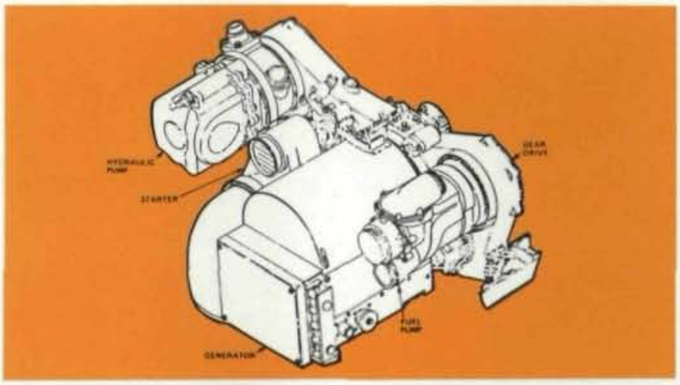


Fig. 1—Aircraft mounted accessory drive with accessories

The literature is full of examples where gear vibrations lead to more dramatic failures than the one presented here. Drago and Brown[1], for example, refer to a helicopter transmission gear that exploded during operation, because one of the resonant frequencies coincided with an excitation frequency. A number of case histories of gear-excited torsional vibrations are illustrated in reference[2]. In the latter paper, Rieger showed that torsional modes may be excited by low order harmonics of shaft rotation. The magnitude of excitation is directly related to gear machining errors as clearly analyzed by Mark[3]. He identified three types of transmission error: those due to tooth spacing errors, tooth-to-tooth random error other than tooth spacing, and tooth elastic deformations combined with mean profile deviations.

The purpose of this paper is to illustrate how the gear resonance problem was identified through the use of waterfall diagrams and finite element techniques. A systematic procedure is then proposed to eliminate damanging gear resonances from the operating range, early in the design stage.

Waterfall Diagram Survey

The accessory drive under consideration is shown schematically in Fig. 1. The outline of the gear case is represented by a dash-dot line. The various accessories consist of an hydraulic pump (HP), an air turbine starter (ATS), a variable speed constant frequency generator (VSCF), a fuel boost pump (FP), and two lube pumps (not shown). A frontal section of the drive shows the gear arrangement in Fig. 2.

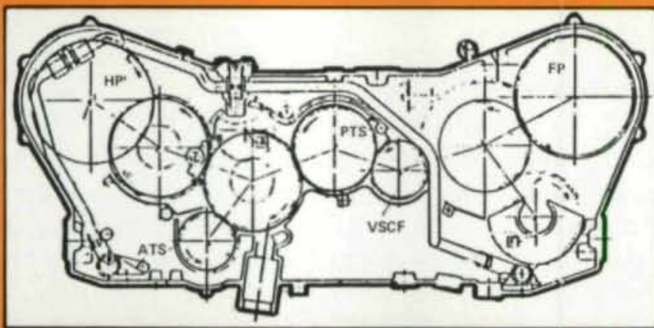


Fig. 2—(top left)
Frontal section
of gear drive

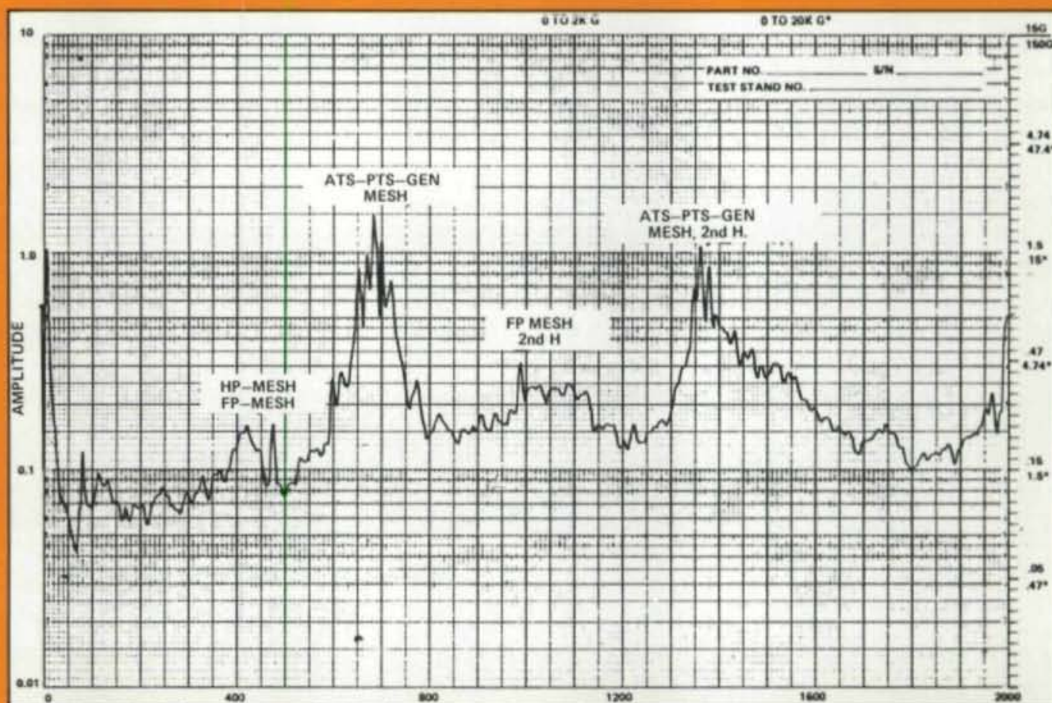


Fig. 3—(center)
Typical vibration signature
measured off gear case

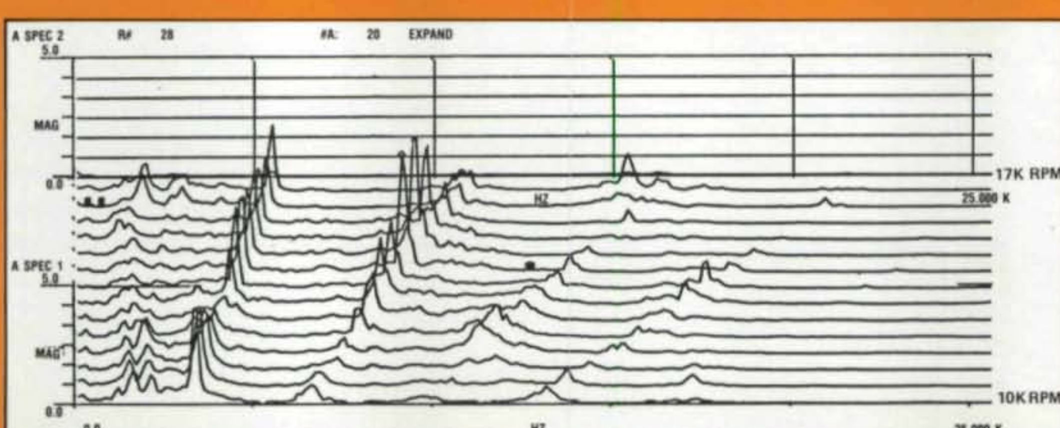


Fig. 4—
Waterfall
diagram of base-
line configuration

Typically during an acceptance test, the vibration signature, taken off an accelerometer mounted on top of the drive, is analyzed over a 2 KHZ band and a 20 KHZ band. Because gear mesh excitation occurs over frequencies higher than 2 KHZ, only the 20 KHZ signature is shown in Fig. 3. The signature is characterized by peaks occuring at mesh frequencies associated with the different gear meshes in the drive. Also indicated are the second harmonic as well as amplitude-modulated side bands associated with shaft and bearing rotations. Those side bands are usually indicative of excessive shaft misalignments and bearing defects[4, 5]. All these peaks are generally present to some extent in all gear drives and a single speed record, such as shown in Fig. 3, is far too inadequate in identifying a vibration problem.

If similar signatures, over a given speed range, are collected at equal speed increments and then arranged in tandem as shown in Fig. 4, the so-called waterfall diagram is obtained. This waterfall diagram was generated using an autosequence program on the HP5423A Hewlett Packard Structural Dynamics Analyzer. Some features are now clearly apparent from Fig. 4. Peaks associated with gear impact propagate along straight lines defined by the mesh orders of the drive, i.e. their frequencies vary linearly with the speed of the corresponding gears. If a peak frequency is speed independent, then it must be either associated with a constant speed shaft or with a stationary resonance, such as that of the casing or the mounting structure. On the other hand, if a peak, excited by one of the gear meshes, attains a maximum at a speed within the test range, then the likelihood of a rotating element resonance exists. Such is the case with the gear mesh excitation of the ATS, VSCF, and PTS (Power Take-off Shaft).

An equally illustrative method of representing the data encompassed in a waterfall diagram is shown in Fig. 5. This plot is generated for the same drive using a Gen Rad dual channel analyzer. The fanning lines represent orders of excitation, the values of which are indicated on the right ordinate. The abscissa and the left ordinate represent the PTS speed in RPM and the vibration frequency in HZ respectively. The varying size square symbols shown in the figure are indicative of the vibration amplitude at the corresponding point.

It is interesting to note that, again a resonant point is detected at the same speed (14,000 RPM) and frequency (9.1 KHZ) as observed in Fig. 4. The order of excitation in this case is 39 which is the number of teeth of the PTS gear, as would be expected. The problem now is to determine which one of those three gears is the culprit. The PTS gear geometry, shown in Fig. 6, lends itself well to impulse hammer testing; while the other two, shown in Fig. 7, because of their compactness, are not suitable for impluse hammer testing. The former acts more like a plate in transverse vibrations, while the latter mostly behave as torsional members. An analytical finite element approach is, therefore, adopted for the ATS and VSCF gears.

Impulse Hammer Testing of PTS Gear

Use of a calibrated impulse hammer, in modal structural, analysis, has proven to be very effective in a great number of applications. The main reason is that the frequency spectrum of the time-varying impulse signal is nearly flat over a wide frequency range (up to 10 KHZ with a hard tip). Thus, all resonances of the structure within this frequency band can be excited.

The test procedure involves impacting the object with the hammer at many points, and measuring the motion at one critical point (or vice versa). These tests supply the stimulus and response information to compute the classical transfer functions. An example of such transfer functions is shown in Fig. 8.

The input and output signals are fed into a HP5423A dual channel FFT spectrum analyzer. An autosequence program is written to automate the conversion of the transfer function data into natural frequencies, modal damping, and mode shapes. A Hewlett Packard 9872 X-Y plotter is used to reproduce a plot of the animated mode shapes. Fig. 9 shows a plot of the first four modes. It is significant to note that none of the obtained modes coincided with the 9.1 KHZ observed in the waterfall diagram. It is, therefore, unlikely that the PTS gear is the source of the vibration problem.

Finite Element Modal Analysis of the ATS-Gear

The MSC/NASTRAN program is used to compute the natural frequencies and mode shapes of the ATS gear. Because gear and spline teeth contribute very little to the circumferential stiffness of the gearshaft, it is safe to assume

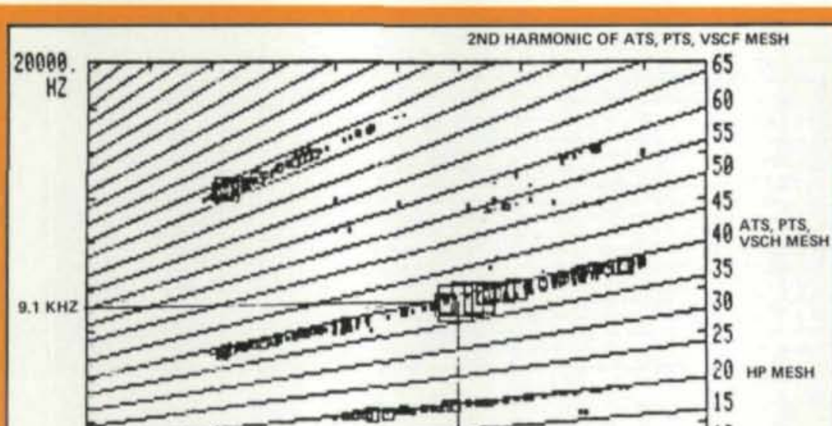
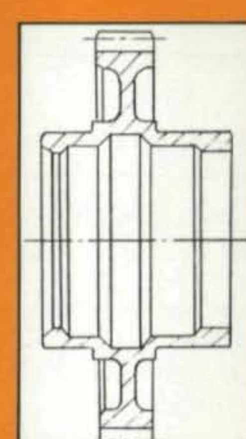


Fig. 5—(above) Excitation order diagram of baseline configuration

Fig. 6—(below) PTS Gearshaft



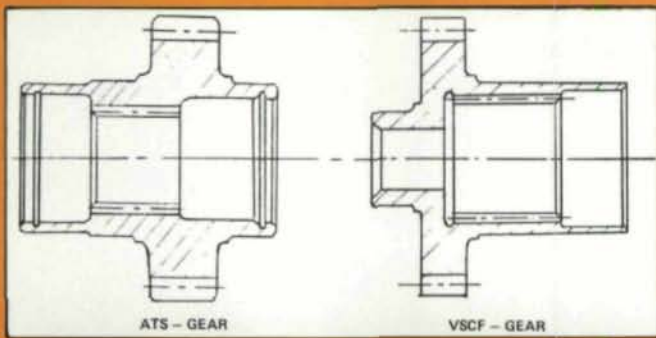


Fig. 7—(above left) ATS and VSCF Gearshafts

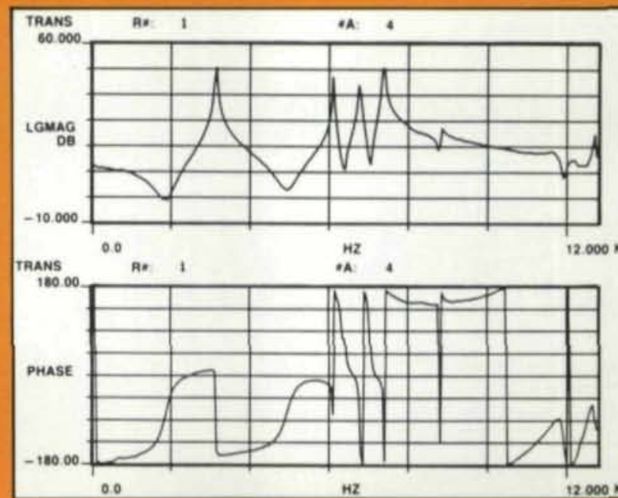


Fig. 8—(top right) Transfer function amplitude and phase for PTS—Gear

Fig. 9—(below center) Mode shapes, frequencies, and damping of PTS—Gear

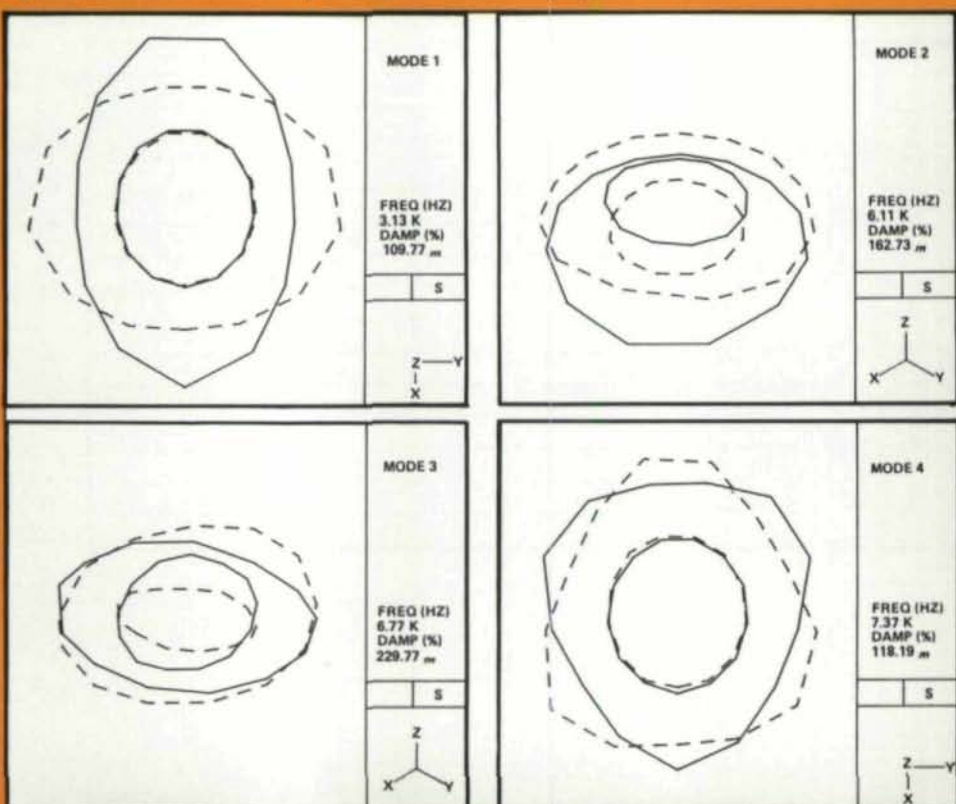


Fig. 11—(bottom right) Mode shapes and frequencies of baseline ATS—Gear

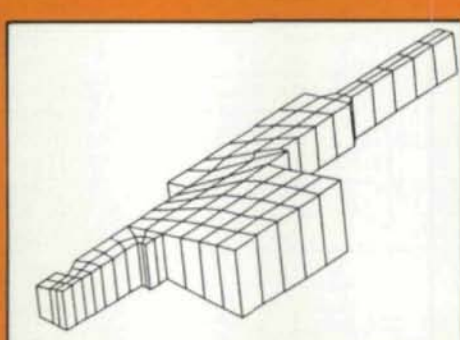


Fig. 10—Finite element model of ATS—Gear

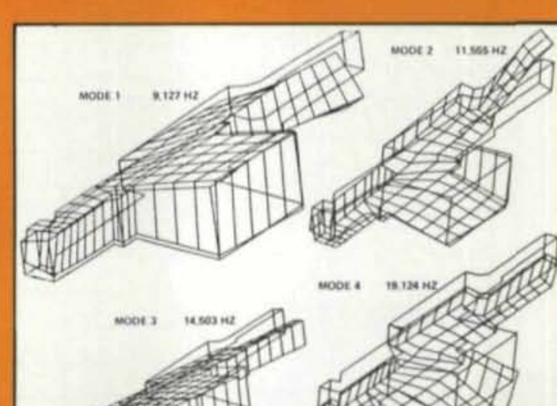
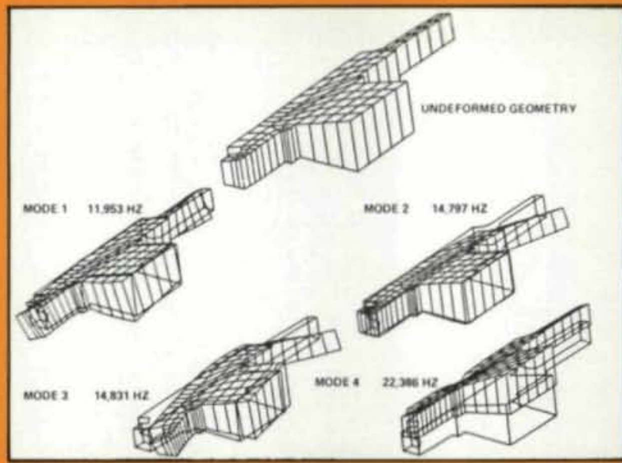


Fig. 13—(top right) Finite element model, mode shapes, and frequencies of modified ATS—Gear



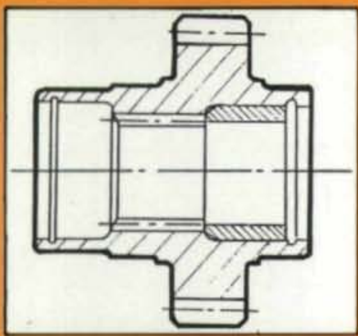


Fig. 12—(above left) Modified ATS Gear

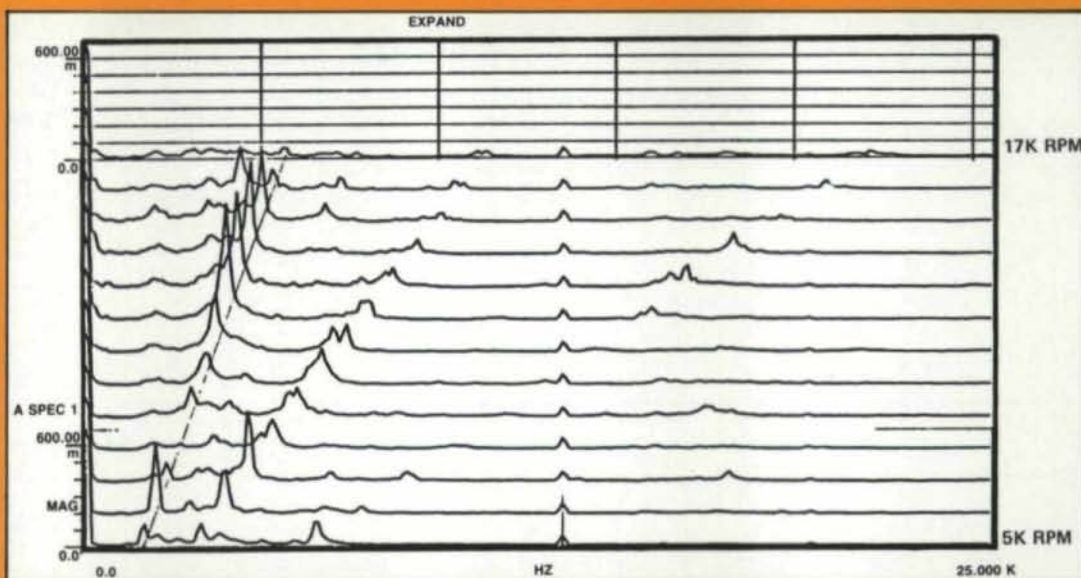


Fig. 14—(center) Waterfall diagram of drive with ATS—Gear removed

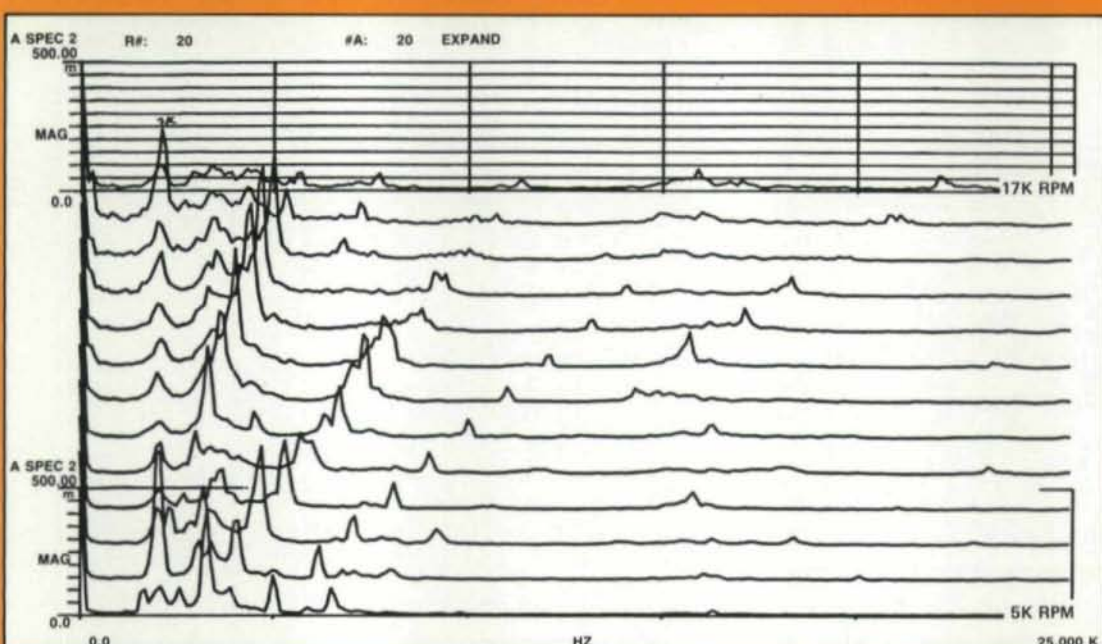


Fig. 15—(bottom left) Waterfall diagram of drive with modified ATS—Gear installed

the latter is an axisymmetric structure. The cyclic symmetry option in the program is, therefore, utilized and a plot of a 15° segment is reproduced in Fig. 10 with the hidden lines removed. The model is composed of 80 hexahedrons over 230 grid points. The model was constrained from rotation about the axis of rotation at one end of the spline. The analysis results in the plotted mode shapes and associated frequencies shown in Fig. 11. It is interesting to note that the fundamental frequency of torsion for the ATS gearshaft is 9.1 KHZ which coincides with the resonance observed in the waterfall diagram of Fig. 4.

Similar analysis of the VSCF gear showed that none of its modes coincided with the indicated resonance. Since the ATS gear has proven to be the culprit, a structural modification of the same is necessary. The object of the modification is to detune the subject gear, moving its first torsional frequency beyond the operating range (higher than 10.93 KHZ). A number of modified configurations were analyzed before a successful fix was reached. The recommended configuration entailed pressing a sleeve into the drive end of the gearshaft as shown in Fig. 12. The finite element model of the new gear and resulting mode shapes and frequencies are shown in Fig. 13. The NASTRAN program predicts the first torsional of the modified gear to be 11.95 KHZ, sufficiently above the operating range.

Verfication Testing of the Modified Configuration

To confirm the results of the above analysis, a waterfall diagram is generated for the subject drive with the ATS gear removed. Fig. 14 shows such a diagram. Indeed, as theoretically conjectured, the ATS gear must have been the resonating element in the drive because the amplitude along the corresponding excitiation order dies down with speed when the ATS gear is no longer in the gear train.

The modified gear was subsequently installed and another waterfall diagram developed as shown in Fig. 15. This clearly shows that the 9.1 KHZ resonance has now disappeared. The diagrams of Figs. 14 and 15 are almost identical proving the effectiveness of the detuning process.

Conclusions

The above investigation clearly illustrates the importance of performing a detailed modal analysis of the gear elements in a high speed drive. As can be seen, the alternatives may be very costly. A systematic procedure to avoid the damaging consequences of gear resonance may be outlined as follows:

1. Identify excitation mechanisms in the drive.
2. In the design stage, evaluate natural frequencies and mode shapes of individual gears using finite element techniques.
3. Eliminate torsional modes from operating range.
4. For helical and bevel gears, minimize axial vibrations by detuning and/or damping treatments.
5. Test gear blanks to verify analytical results.
6. Introduce additional fixes if necessary.
7. Generate waterfall diagrams of assembled drive.
8. Identify remaining resonant problems, if any, with other elements of the drive such as housings and accessories.
9. Correct remaining problems accordingly.
10. Generate new waterfall diagrams after all fixes have been instituted.

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