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# DYNAMIC BEHAVIOUR OF AN AIRCRAFT POWER TAKE-OFF SHAFT DRIVE-SYSTEM

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## ABSTRACT

Aircraft power take-off shafts operate at high speeds and in a difficult operating environment. This paper describes the results of an investigation into the dynamic behaviour of the F/A-18 Aircraft Mounted Accessory Drive gearbox drive system. The investigation was carried out to determine the cause of failure of the input bearing in this gearbox. Measurements of gearbox vibration and shaft dynamic orbits were carried out to determine the origin of the high dynamic loads, and to evaluate the system critical speeds. These measurements indicate that the system behaviour was determined by a large initial unbalance due to component clearances, and a dynamic magnification due to the proximity of running speed to the system critical speed. Possible system improvements and proposals for design changes are discussed, as are ways to provide an interim alleviation of the problem.

## INTRODUCTION

Failure of the input bearing of the F/A-18 AMAD (Aircraft Mounted Accessory Drive) gearbox has necessitated in-flight engine shut-down in several of these aircraft. Initial investigations failed to yield an obvious cause of failure such as bearing material defects, or lubrication deficiencies [1]. Accordingly, the investigation was widened to include the possibility of mechanical overload of the bearing, and this paper describes the resulting initial investigation into the dynamic behaviour of the AMAD gearbox drive system, and the influence that this behaviour may have on bearing failure.

The use of drive-shafts operating at high speed is attractive from the viewpoint of size and weight, as for a constant power requirement, the torque is inversely proportional to speed.

However, the disadvantages are likely to be a reduced tolerance for balance and misalignment, and a closer proximity to system critical speeds, which also may be lowered due to the size reduction that follows the lower torque requirement. The current configuration of AMAD gearboxes generally requires an external shaft, and to accommodate misalignment flexible couplings are required, and axial movement also has to be accommodated.

Several investigations of aircraft external drive systems are described in refs [2] - [5]. References [2] and [3] describe the development of a power takeoff system for the X-29A PTO/AMAD, which was powered by the same engine (a General Electric F404), and runs at the same rpm as the F/A-18, but utilised a considerably longer driveshaft. References [4] and [5] describe an unstable subsynchronous vibration encountered in development of the F-16 high-speed shaft and gearbox of the engine start system. Reference [6] contains relevant general design information for high-speed drive systems. These references show that seemingly quite small design changes can have a large effect on the system dynamic behaviour.

With the exception of a set of measurements carried out on the AMAD gearbox post-overhaul test rig at Lucas Aerospace Sydney, all of the measurements described here were carried out on RAAF aircraft, and include casing vibration, and driveshaft dynamic mode shapes as measured by proximity probes at six locations. Video recordings of shaft strobed motion were also made.

## AMAD GEARBOX SYSTEM DESCRIPTION

Figure 1 shows the location of the two AMAD gearboxes in the F/A-18 aircraft. The AMAD gearbox is powered by a high speed drive from the General Electric F-404 engine. The drive is geared at a ratio of 1:1 through two right-angled drives from the high-speed engine rotor (16810 rpm being 100% speed). The gearbox has several accessory pads on the front face which mount the electrical generator, hydraulic pump, fuel boost pump and air turbine starter. Figure 2 shows the overall layout, Fig. 3 a cross-section through the input housing, and Fig 4 the input PTS and sleeve. When starting, the power flow is in a ‘reverse’ direction from the air turbine starter through the AMAD gearbox to the engine. The gearbox is mounted in two steel spherical mounts on each side, with an upper elastomeric mount. The lower inboard mount is fixed, the outer one sliding on a spindle. Additional restraint to the gearbox is brought about by the fuel and hydraulic pipe connections to the fuel boost pump and hydraulic pump. Figure 5 shows the transmission drive-shaft. The drive-shaft is composed almost entirely of titanium, the only steel components being the drive flange at the engine end, which also incorporates shear bolts, and the captive flange attachment bolts.

The PTS input shaft is free (within limits) to move axially within a sleeve (Figs. 3 and 4). The sleeve is supported by two identical ball bearings; and is isolated from gear side loads, as the drive gear is itself supported on two separate bearings. The housing dimension in the axial direction is checked and adjusted before assembly so that there is between 0.002 and 0.004 in (0.051 to 0.102 mm) end-clearance for the bearing outer races. The bearings are deep-groove plain ball-bearings. The cage is bronze, with steel pins. Ref. [1] provides further details on bearing construction. There is no specific control over bearing axial preload. The PTS shaft is located within the sleeve at the diameters A and B in Fig. 4; manufacturing tolerances of the components result in radial clearances at these two points ranging between 0.0013 and 0.0028 inch (0.033 to 0.071 mm). This clearance is large in comparison with the balance tolerance of the PTS shaft which is 0.002 oz ins at the flange end, equivalent to a centre-of-mass eccentricity of 0.00012 ins (0.0031 mm), as the flange end weighs 16 oz.

## TEST PROCEDURE

Initial measurements were made on an aircraft selected at random, which was available because of an unrelated unserviceability. The initial measurements were carried out using a tri-axial group of accelerometers on the PTS input bearing housing (Fig. 6) and an additional point on the base of the gearbox. Upon completion of the initial recordings and analysis, the drive-shaft motion was visually observed and video-taped, using a strobe-light running at a slight slip frequency, so as to give motion visualisation. The results showed that a standard video camera produced good results (tests with a high shutter speed resulted in gaps in the image). The video-taping was then carried out by connecting a timing signal from a shaft tacho to the strobe-unit, and adjusting the slip frequency to produce the desired effect. In order to quantify this motion, the drive-shaft displacement was recorded with an eddy-current proximity probe. A once/revolution tachometer signal, using reflective tape on the drive-shaft, was also obtained.

These initial recordings of the drive-shaft motion were limited to measurements at one location. In order to gain a more complete picture of the drive-shaft dynamic behaviour, in particular the shaft orbits at various axial locations, a fixture was attached to the airframe as in Fig. 7. Two eddy-current proximity probes were mounted at one time at each of the three axial positions. The axial mounting dimensions are shown in the drive-shaft illustration (Fig. 5). The probes were mounted at 90 degrees relative to one another, and were quite large at 9.5 mm (0.375 ins) tip diameter so as to enable a sufficiently large gap from the driveshaft to prevent the possibility of drive-shaft damage. Accelerometers were mounted on the probe bracket adjacent to the displacement probes, so as to monitor absolute probe motion in case of resonance of the mounting frame which would change the displacement measurement. For all the recordings the reflective tape used with the optical tacho remained in the same position on the drive-shaft, so as to enable consistent phase relationship as the probes were shifted to the different axial locations.

Further observations of drive-shaft dynamic behaviour and casing vibration were made on the AMAD gearbox post-overhaul test stand at Lucas Aerospace Sydney. The test stand is operated at input speeds up to 18,000 rpm during acceptance tests, a significantly higher speed than possible in the aircraft (typically limited to 94% speed - 15800 rpm). Tests were carried out with the standard test rig shaft (a shaft with single flexible coupling at each end), and with the F/A-18 drive-shaft.

## RESULTS AND DISCUSSION

### Initial aircraft vibration measurements

The initial measurements were made on an aircraft selected at random, using a tri-axial group of accelerometers on the PTS input bearing housing (Fig. 6) and an additional point on the base of the gearbox. The vibration frequency vs amplitude (in/s rms) for the three accelerometers is given in Fig. 8(a)-(c). The vibration amplitude (in/s rms) of the once/per revolution frequency component is plotted vs speed in Fig. 9 for the left-side, and Fig. 10 for the right-side.

The vibration levels recorded during this initial trial were exceptionally high, reaching 5.0 in/s rms (Fig. 9), and were much greater than considered acceptable for this class of machinery. Levels tolerated in aircraft tend to be higher than that considered acceptable for industrial equipment, however a level of 1.5 in/s peak velocity is generally considered to be an upper limit. For example, levels for the GE F-404 engine over 1.0 in/s average, (which is equivalent to 1.57 in/s peak) are considered an upper limit. The input housing vibration was predominantly once-per-revolution, almost certainly caused by a large unbalance in the rotating assembly. Misalignment could conceivably be a factor, but in view of the absence of large vibration at twice/rev, and the absence of a large axial component, it is not likely. Also, checks of the alignment of this assembly showed it to be within tolerance.

Following recording of these high vibration levels, video-taping of the drive-shaft motion, using a strobe synchronised with the shaft (but with a small slip frequency for motion visualisation) was carried out. This provided graphic evidence of sizeable run-out of the drive-shaft and of the input PTS shaft. During engine run-up in the region 75-80% of full speed, the diaphragm coupling flanges passed through several modes. Subsequent analysis of shaft vibration amplitude and phase showed little connection with this mode, so that it appears to be localised to the coupling assembly. Visual observation suggested that the dynamic shaft motion could be as large as 0.060 inches peak-peak, and this was confirmed by the eddy-current proximity probe measurement (Fig. 11). The shaft motion comprised almost entirely of a once per revolution frequency component over the entire speed range. There was no evidence of any significance non-synchronous vibration, and it is presumed (although this could not be absolutely confirmed at this stage) that the vibration is forward whirl, as there is no reason to expect otherwise.

Notable features of the vibration measurements during engine run-up were an absence of resonances in casing or shaft motion, as confirmed by amplitude and phase measurements. However, the rapidly increasing amplitude of both casing vibration and shaft motion with engine rpm, suggest that the drive-train is approaching its first significant critical speed.

The input housing assembly of the AMAD gearbox on the left-side of aircraft A21-116 was subsequently removed, and the input casing was found to be cracked in two locations, on the housing flange. Also, the housing retaining nuts were found to have low tension, and there was evidence of heavy fretting on the input housing and mating gearcase flange. Later examination of the aft PTS bearing showed loose cage pins, heavy bearing on the cage, and contamination on the raceway.

## Drive System and AMAD Gearbox Dynamic Behaviour

The shaft dynamic mode shape was measured using the multiple probe locations as illustrated in Fig. 5. The mode shape, including the shaft orbit (1X component, slow-roll[1] corrected) is plotted in Fig. 12, for an engine speed of 15 300 rpm. A line is drawn joining each orbit to indicate the point where the reflective tape on the shaft is coincident with the optical tacho probe. Orbit directions are shown with the arrow. Notable features are the circularity of the orbits at the AMAD gearbox end. Also apparent was the small change in phase of the shaft amplitude with increasing speed, indicating that the critical speed is significantly higher than the speed reached during these tests. These plots also confirm the forward whirl of the orbits.

The harmonic content of the shaft orbit motion is low, for example at probe position 1 (Fig 5), and a shaft frequency of rotation of 258 Hz, the 1X component is .025 in p-p, the 2X component .0023 in p-p and the 3X component 0.0013 in p-p. Also, from examination of dynamic shaft displacement frequency spectra there was no evidence of significant non-synchronous vibration, other than at the engine end of the drive-shaft. It was presumed that at this end, where amplitude is lower, that there is influence from the engine vibratory behaviour.

## Test stand drive system dynamic behaviour

The AMAD gearbox post-overhaul test stand was utilised to carry out further tests on the drive system dynamic behaviour. Test results of drive-shaft radial amplitude are summarised in Fig. 13. The Nyquist plot (amplitude vs phase) shows a large increase in amplitude at the higher speeds, together with the phase change (lag) of vibration relative to the tacho signal, which indicates that critical speed is not far beyond the 18000 rpm reached in testing.

## Reduction of dynamic loads

Rotation of the drive-shaft relative to the AMAD input PTS can have a significant effect on vibration levels, as evidenced by measurements which were carried out on aircraft A21-39 left-side AMAD gearbox. The input housing vibration ranged from 1.2 in/s, 0.89 in/s and 0.69 in/s peak in each of the three possible positions of the drive-shaft relative to the input PTS. The subsequent fleet wide survey by the RAAF [7] has shown the results to be repeatable in nearly all cases, and reductions in input housing vibration of over 50% on some aircraft have been achieved by this technique.

Further improvements can be expected if the clearances between the input PTS and sleeve (Fig. 4) could be reduced to the minimum possible consistent with permitting axial movement. A longer term solution is to increase the size of the input PTS bearing, which will both increase its load carrying capacity, and also increase the stiffness, which will raise the critical speed of the drive system, and reduce the dynamic load increment.

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[1] slow-roll refers to the shaft run-out errors evident at very slow speed, as a result of features such as shaft eccentricity and mounting errors.

## CONCLUDING REMARKS

(a) AMAD gearbox input housing vibration measurements have shown very high vibration levels on some aircraft. The vibration appears to be largely due to unbalance in the driveshaft assembly.

(b) Measurements of drive-shaft motion have confirmed synchronous forward whirl of the drive-shaft which couples the engine to the AMAD gearbox The driveshaft orbits are nearly circular.

(c) The drive system appears to operate below its first critical speed, but the rapid increase in vibration levels at the higher speeds indicates that the first critical speed may not be far above running speed. There is no evidence of significant driveshaft system resonance during the operating speed range of idle to full military power.

(d) The primary cause of unbalance appears to result from clearances in the AMAD gearbox input shaft assembly. These clearances will bring about an initial unbalance of the assembly much greater than individual component balance factors.

(e) Rotation of the drive-shaft relative to the input PTS assembly can bring about significant reductions in vibration levels.

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7. RAAF TFLM AMAD Vibration Data Base. 1996.

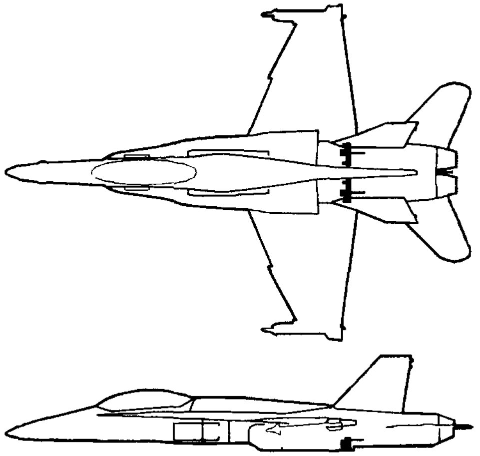

Figure 1 AMAD gearbox location

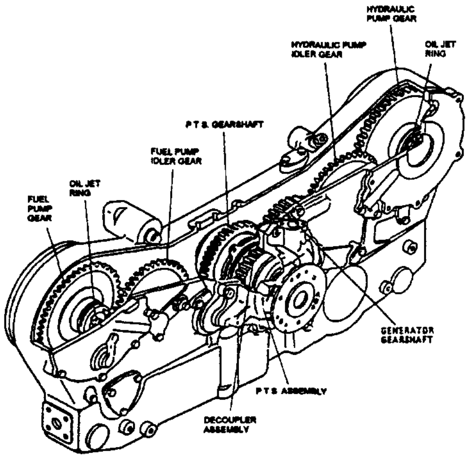

Figure 2 AMAD gearbox

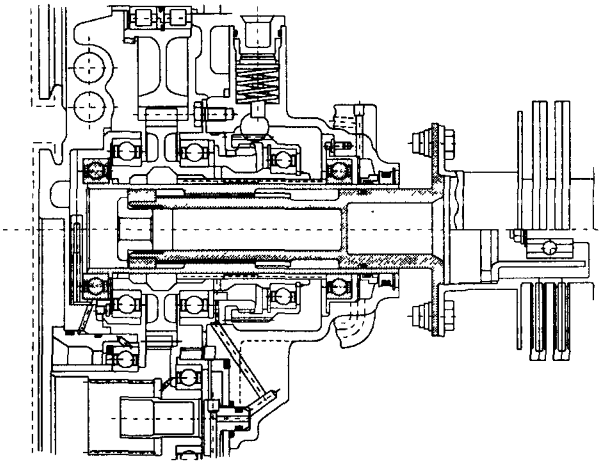

Figure 3 Input Housing

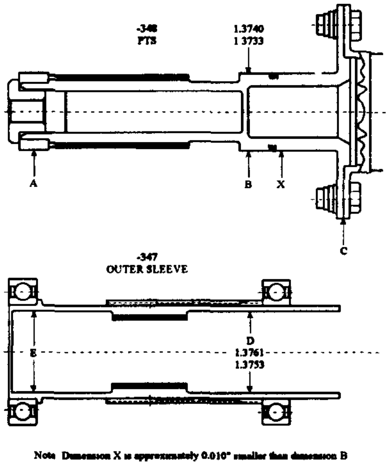

Figure 4 Input PTS and sleeve

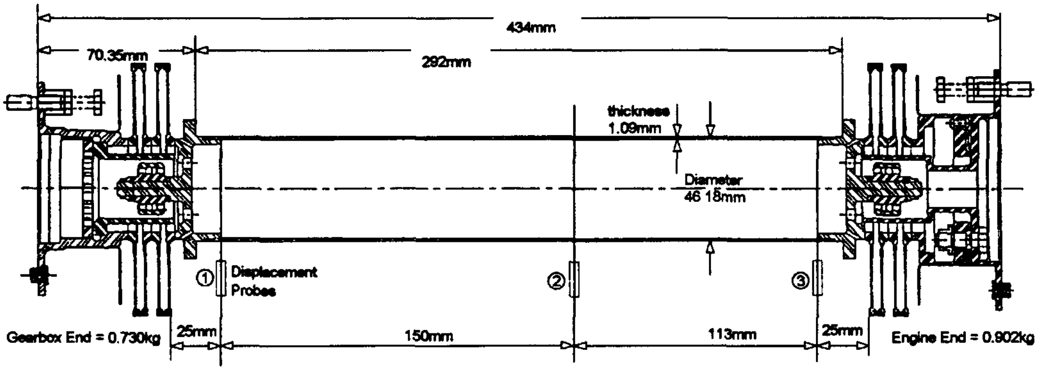

Figure 5 Transmission drive-shaft

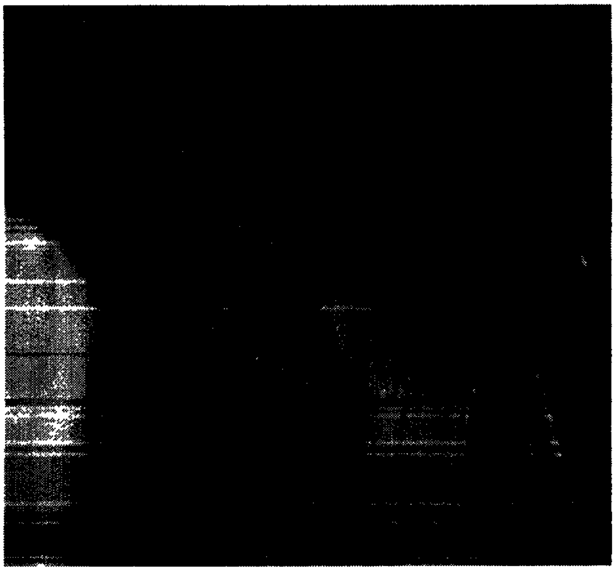

Figure 6 Tri-axial accelerometer group

Figure 7 Proximity probe mounting bracket

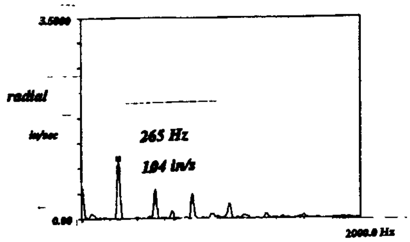

Figure 8a Radial Vibration

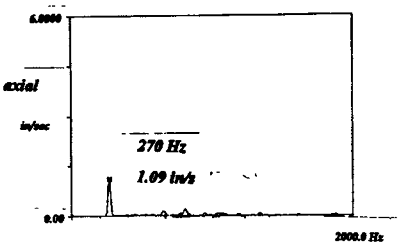

Figure 8b Axial Vibration

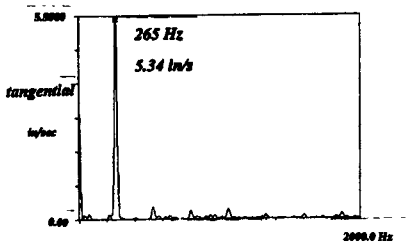

Figure 8c Tangential Vibration

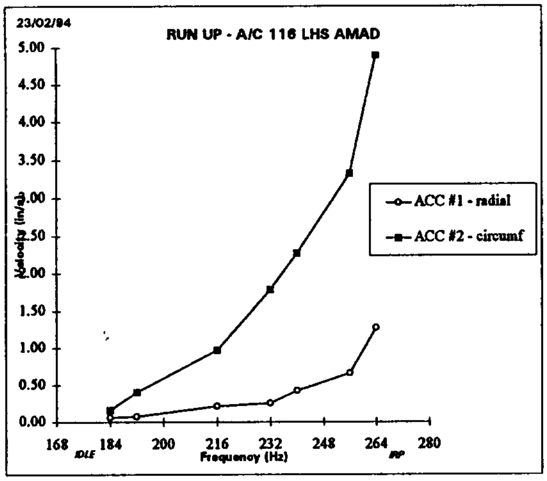

Figure 9 Left side 1X vibration

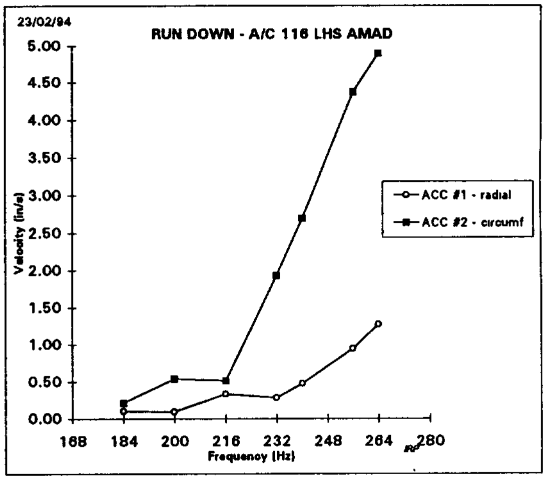

Figure 10 Right-side 1X vibration

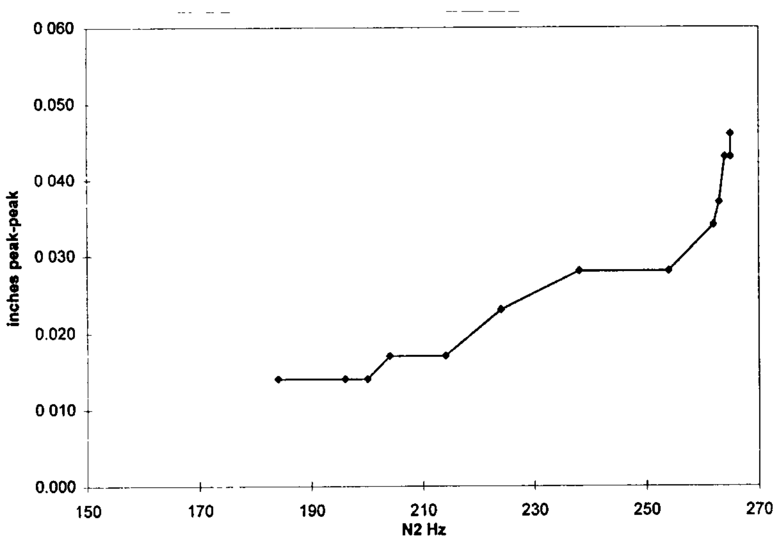

Figure 11 Drive shaft radial run-out peak-peak in A/C 116 LHS

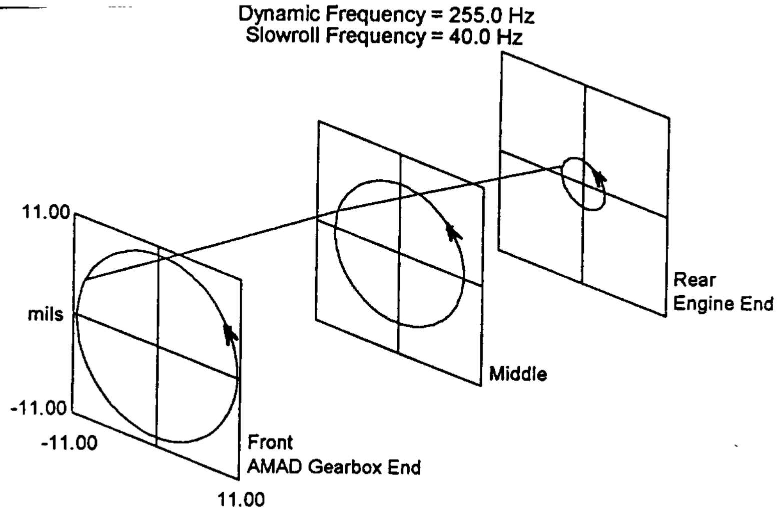

Figure 12 Drive shaft dynamic response (1X component) corrected for slow-roll errors

| Shaft freq<br>*Hz* | Ampl P-P<br>*mils* | Phase<br>*Degrees* | x-coord<br>*mils* | y-coord<br>*mils* |
|---|---|---|---|---|
| 20 | 10.5 | 21 | 0.00 | 0.00 |
| 55 | 11.1 | 29 | -0.09 | 1.62 |
| 80 | 11.2 | 34 | -0.52 | 2.50 |
| 102 | 12.2 | 39 | -0.32 | 3.92 |
| 127 | 13.4 | 44 | -0.16 | 5.55 |
| 152 | 14.1 | 46 | -0.01 | 6.38 |
| 175 | 15 | 49 | 0.04 | 7.56 |
| 195 | 19 | 53 | 1.63 | 11.41 |
| 225 | 21.7 | 47 | 5.00 | 12.11 |
| 262 | 29.1 | 39 | 12.81 | 14.55 |
| 282 | 38 | 28 | 23.75 | 14.08 |
| 305 | 49 | 14 | 37.74 | 8.09 |

corrected for slow-roll run-out

Shaft rotated 240 deg relative to initial position

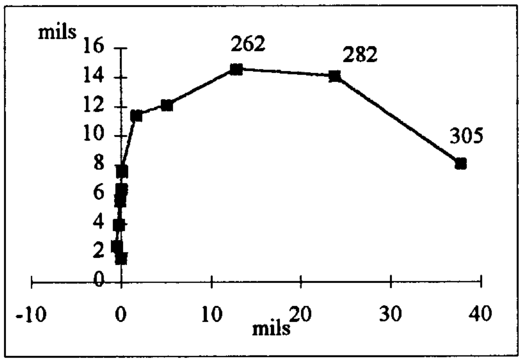

Figure 13 Nyquist plot of drive shaft dynamic response on post-overhaul test stand