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HELICOPTER GEAR NOISE AND ITS TRANSMISSION TO THE CABIN

J.S. POLLARD

WESTLAND HELICOPTERS LTD.
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1. INTRODUCTION

Helicopter soundproofing treatments have been fairly successful in reducing the transmission of airborne sound, but the attenuations obtainable are limited to 15-20dB for military aircraft and 25-30dB for civil aircraft (1). Recent measurements have shown, however, that much of the noise reaching the cabin is due to structural radiation of gear noise in the 500Hz to 4kHz frequency range. In this case soundproofing schemes are not very effective and further noise reductions can only be obtained by (a) reduction of gear noise at source, (b) isolation of the gearbox/airframe interface and (c) structural alterations to the airframe. Each of these solutions is the subject of separate long term investigations being pursued by Westland Helicopters Ltd. To provide support for these studies and to enable short term solutions to be devised, a number of experimental studies have been carried out to further the understanding of the mechanisms of gearbox noise generation (with regard to helicopters) and its transmission via the gearbox and airframe structures to the cabin. Such tests include gearbox rig monitoring, gear component resonance surveys, transmission error measurements, dynamic absorbers, airframe shake tests and in flight noise and vibration measurements. This paper reviews some of the more important results of these tests and associated theoretical studies and pays particular attention to the Lynx gearbox.

2. NOISE GENERATION MECHANISMS

The noise radiated by the gearbox casing is a result of the force fluctuations from the elastic deformation of the gear teeth under load and tooth manufacturing errors. These cause the gears to rotate in a non-uniform manner and dynamic forces are set up at the gear meshing frequencies and harmonics. The forces vibrate the gear shafts in the torsional, axial and lateral modes and hence displace the bearings so that the gearbox casing vibrates and radiates noise.

The Lynx main gearbox (Figure 1) consists of two port and starboard input pinion assemblies, which are identical in operation, each providing a step down in RPM in two stages. The input drive from the two engines is taken through free wheel assemblies to involute form spiral bevel pinions which mesh with spiral bevel crownwheels. This provides the first stage reduction of 2.48:1 and turns the drive through 79^{o} to the conformal input pinion assemblies. The spiral bevel pinion and crownwheel have 21 and 52 teeth respectively giving a meshing frequency of 2110Hz (denoted as 1B). The crownwheels are bolted to the conformal input pinions which mesh with the conformal wheel making the second stage reduction of 7.63:1. The conformal pinion and wheel have 11 and 84 teeth respectively giving a meshing frequency of 446Hz (denoted as 1C). The conformal gears and the spiral bevel gears are the two dominant noise sources of the Lynx gearbox. A photograph of the Lynx gearbox complete with casing and mounting arrangement is shown in Figure 2.

Figure 3 taken from Lynx gearbox rig data illustrates the marked similarity between the harmonic content of the vibrations measured on the casing via accelerometers and the noise radiated to a nearby microphone. Both sets of spectra exhibit prominent discretes at the gear meshing frequencies (1B and 1C), their harmonics and associated sidebands. Harmonics of the conformal meshing up to the 10th can be clearly seen and in many cases the harmonic levels are higher than the fundamental level. These discrete frequencies dominate the helicopter cabin noise spectrum in flight. Wide differences in levels exist between boxes of the same type, however, as illustrated in Figure 4. Variations of up to 15dB in noise level and over 20dB in vibration level are possible for a given meshing frequency. These variations are due to a number of parameters including transmission error differences and thus offer scope for significant noise reductions.

3. GEAR GEOMETRIC EFFECTS

Improvements in spiral bevel gear noise have been obtained over the last few years by modifications to tooth geometry. These changes (stage 1 gears to stage 3 gears) were made as part of the normal tooth development programme to maximise load capacity. Figure 5 shows that significant reductions were obtained in the casing vibration, and hence the radiated noise, at bevel meshing frequency. This improvement at source, together with the fact that cabin soundproofing treatments are more effective at high frequencies has meant that bevel gear noise on the Lynx helicopter is no longer the problem it once was. Attention has now been directed instead to conformal gear noise reduction and the fact that the harmonics of the conformal meshing are of similar levels to the fundamental suggests that improvements in conformal gear geometry should be beneficial.

Conformal gear noise arises from variations in tooth deflection, which increase with increasing load and torque, and pitch and helical errors. Pitch errors indicate inaccuracies in spacing between successive teeth and the helix error determines the deviation of tooth lead across the face width. Unlike involute gears, conformal gears do not use the transverse profile to give uniform angular motion, this being obtained by the helical action alone. In general higher helix angles should give quieter gear operation but a change from narrow faced conformal gears to wide faced gears, in which the face width was increased from 3 inches to 3.7 inches and the helix angle was changed from 19^{o} to 15^{o} 13', made no appreciable difference to the noise levels. It is possible, however, that the helix angle changes were not large enough to cause significant variations.

Recent transmission error measurements by WHL have concentrated on conformal gear meshing in relation to helix angle differences and tooth deflections. The difference in helix angle between the pinion and the wheel to compensate for pinion bending produces a predicted transmission error of the form shown in Figure 6(a), assuming no tooth deflection and a perfect straight tooth profile. The pinion first leads the wheel, then lags the wheel as the contact moves across the face width of the rotating gears. As torque is applied the teeth are deflected resulting in pinion lead variation and a predicted transmission error curve as shown in Figure 6(b), assuming no helix angle errors. Including the helix angle difference inclines the curve as shown in Figure 6(c). For comparison, Figure 6(d) is an average curve of measured data although it only applies to single gearbox input shaft measurements only. The similarity between the measured and predicted curves suggests that benefits may occur from changes in tooth lead profile to correct variation of tooth deflection and the use of offset bearings to control helix angle differences due to gear case deflections.

4. GEAR COMPONENT AND CASING RESONANCE EFFECTS

Modifications to gear geometries (apart from helix angle changes) may be difficult to specify and manufacture and also there may be great difficulty in optimising gears for smooth quiet operation at different power levels. For these reasons many helicopter manufacturers have concentrated on reducing system resonances in the gear trains, although this does not remove the source of the noise. For example if there is a critical shaft frequency resonance near gear mesh frequency the shaft response could be very large. System resonances can be avoided or shifted in frequency by relocating the bearings, changing the bearing stiffness and altering the shaft stiffness and mass distributions. Such modifications can be studied by firstly developing a mathematical model of the gearbox dynamic system. This approach has been used by WHL (2) on a Wessex Tail Rotor Gearbox (single stage spiral bevel gear) where the gearbox was considered as a dynamic system under displacement excitation from the transmission error. Static deflections due to applied torque were predicted and showed good correlation with measured data. It is hoped to apply similar modelling techniques to the Lynx main gearbox. One experimental application being considered is the attenuation of conformal meshing vibration by dynamic abosrbers inside the gears. Preliminary laboratory measurements (Figure 7) have shown that a symmetrical 'dumbell' type absorber tuned near to conformal meshing frequency and fitted inside the conformal pinion significantly reduced inplane pinion vibration. Further tests are proposed with the absorber mounted in a complete Lynx gearbox during rig running. Other modifications under consideration are changes in shaft materials, the addition of damping materials to the gears themselves and the use of damping rings in which a high damping capacity material is placed between gear rim and hub.

Gear component alterations can only really be carried out after extensive mechanical vibration analysis of an existing gearbox. For this reason WHL are at present conducting both experimental and theoretical studies to determine the natural frequencies and mode shapes of vibration of isolated gear components, gear assemblies and the complete casing of the Lynx gearbox. This has proved extemely complex since cross coupling of in-plane and out-of-plane modes occurs for the single components and these experimental results show little correlation with those for the components assembled together. In spite of the problems, a number of natural frequencies have been found to be close to forcing frequencies as illustrated in Figure 8. This figure shows that the conformal wheel and hub assembly has a natural frequency of 940Hz close to twice conformal meshing, whilst the conformal pinion and bevel ring assembly has a natural frequency of 2145Hz near to bevel meshing. To date the theoretical studies have been confined to simple components such as ring gears. Reference 3 considers the flexural vibrations of circular rings of any given cross-section, including the effects of rotatory inertia and shear deformation of the ring, and shows that in general two distinct modes of vibration occur. These are either purely 'in-plane' or 'perpendicular-to-plane' and the natural frequencies can be calculated in terms of moments of inertia and torsional rigidity of the cross-sectional shape. A family of ring gears taken from different helicopter gearboxes is being tested to see how the theory correlates with size of ring gear. Also a Wessex helicopter main bevel gear is being structurally altered (e.g. teeth removed) to see if the resonant frequencies are shifted significantly. Finite element methods are capable of predicting the stiffness of normal mode characteristics of gearbox casings and applying these to the Lynx main gearbox has resulted in good agreement between measured and predicted natural frequencies of the casing (4).

In a similar manner gearbox dynamic resonances are being studied on the rig by measuring the noise and vibration variation with input rotational speed. Standard RPM tracking analysis using an azimuth signal to tune a filter unit enables the conformal and input bevel meshing frequencies and associated harmonics to be tracked at constant torque. Preliminary results for conformal meshing are shown in Figure 9 for two accelerometer positions on the gearbox casing and it is clear that major resonances of the gearbox occur near forcing frequencies. A number of design features for reducing resonance effects can be incorporated into the gearbox case structure, e.g. stiffness and mass changes, double walls, etc. and these are being considered for future work. In the meantime WHL intend to experimentally coat a Lynx gearbox with damping material to a thickness of about 1/8" to increase case damping and reduce vibration and noise.

Since helicopter gearboxes are at present essentially rigidly mounted to the airframe (see Figure 2), the vibrations originating from gear meshing are transmitted directly to the airframe. Vibration levels up to 31.6g at gear meshing frequencies have been measured on the Lynx structure during flight. Recent airframe shake tests, in which a Lynx airframe was subjected to vibration excitation at the gearbox feet (to simulate the in flight environment), showed that the vibration levels on the airframe were of the same order as the input levels at the gearbox feet. Also the shape of the vibration response curves of the airframe were similar to the shape of the noise field curves measured inside the cabin (5). It would appear, therefore, that vibrations are being transmitted to the main frames of the cabin with little reduction in level causing individual panels to vibrate at large amplitudes and hence radiate noise into the cabin. Preliminary in-flight data appears to confirm this as shown in Figure 10, since acceleration levels measured on the gearbox casing, gearbox mounting points and cabin roof are generally of the same level (i.e. no attenuation) at both conformal meshing and bevel meshing frequencies. It is generally believed, therefore, that much of the high frequency noise reaching the helicopter cabin is structure-borne and thus attention is being directed to high frequency isolation of the gearbox from the airframe, based on developments of low frequency devices. There are a number of possible designs one of which is shown in simplified form in figure 11. This is a fixed frequency device which has isolation properties partially controlled by a bob-weight in parallel with a spring arrangement. Such a system has an advantage over the conventional isolator in that the required stiffness is not controlled by the system resonant frequencies and that relatively high stiffnesses are possible whilst maintaining good vibration attenuating characteristics.

A number of practical problems need to be overcome, however, in that the isolation mounts must be able to cope with large flight loads and high torque reaction with static deflections within acceptable limits for control inputs and shaft alignments. Thus parallel studies are considering the response of the airframe itself to the high frequency vibrations transmitted through the gearbox mounting points. A theoretical programme of work is being formulated to study the passage of travelling waves in the fuselage and the sound radiated from the structural elements such as panels, stringers, and frames. This will include the coupling of flexural and longitudinal waves at corners and joints, the effect of blocking masses, structural modal densities, and detuning methods. This will be supported by experimental data obtained from in flight measurements of noise and vibration and from a laboratory evaluation of the response of vibrating panel treatments.

5. CONCLUDING REMARKS

The helicopter cabin noise spectrum is dominated by gear noise consisting of prominent discretes at gear meshing frequencies, harmonics and associated sidebands. Reductions in cabin noise can be obtained by combined attention to noise at source, the gearbox/airframe interface and the airframe itself. Studies on the Lynx helicopter in all three areas are being actively pursued and it is believed that with our present understanding of gear noise and its transmission to the cabin, significant improvements in cabin noise environments will be possible in the future.

6. ACKNOWLEDGEMENTS

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7. REFERENCES

1. J.S. Pollard, A Preliminary Study of Helicopter Internal (Cabin) Noise, WHL Research Paper 514. March 1976.

2. M.L.W. Salzer, Gearbox Noise - Analysis of System Dynamics. WHL Research Paper 545. March 1977.

3. D.L. Hawkings, A Generalised Analysis of the Vibration of Circular Rings. Journal of Sound and Vibration Vol.53, Part IV. August 1977

4. A.J. Barnard, The Finite Element Analysis of a Lynx Main Gearbox. WHL Research Memorandum 377, Sept. 1976

5. C.R. Wills, Vibration Transmission Paths. WHL Research Paper 523, June 1976.

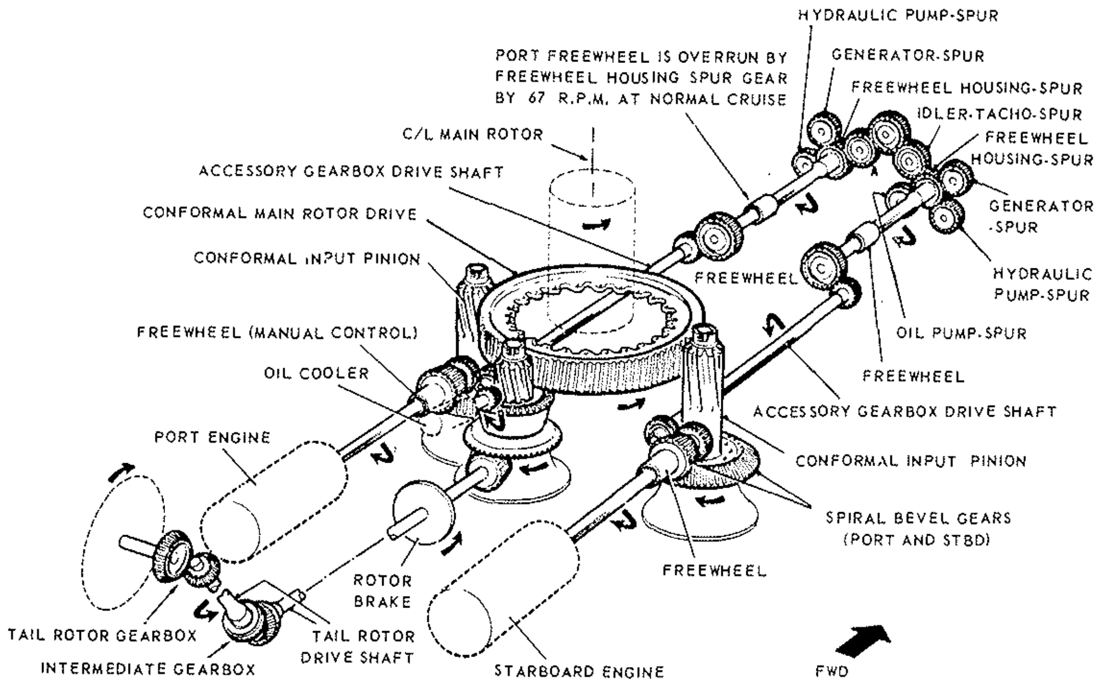


FIG. 1. LYNX MAIN GEARBOX GEAR ARRANGEMENT

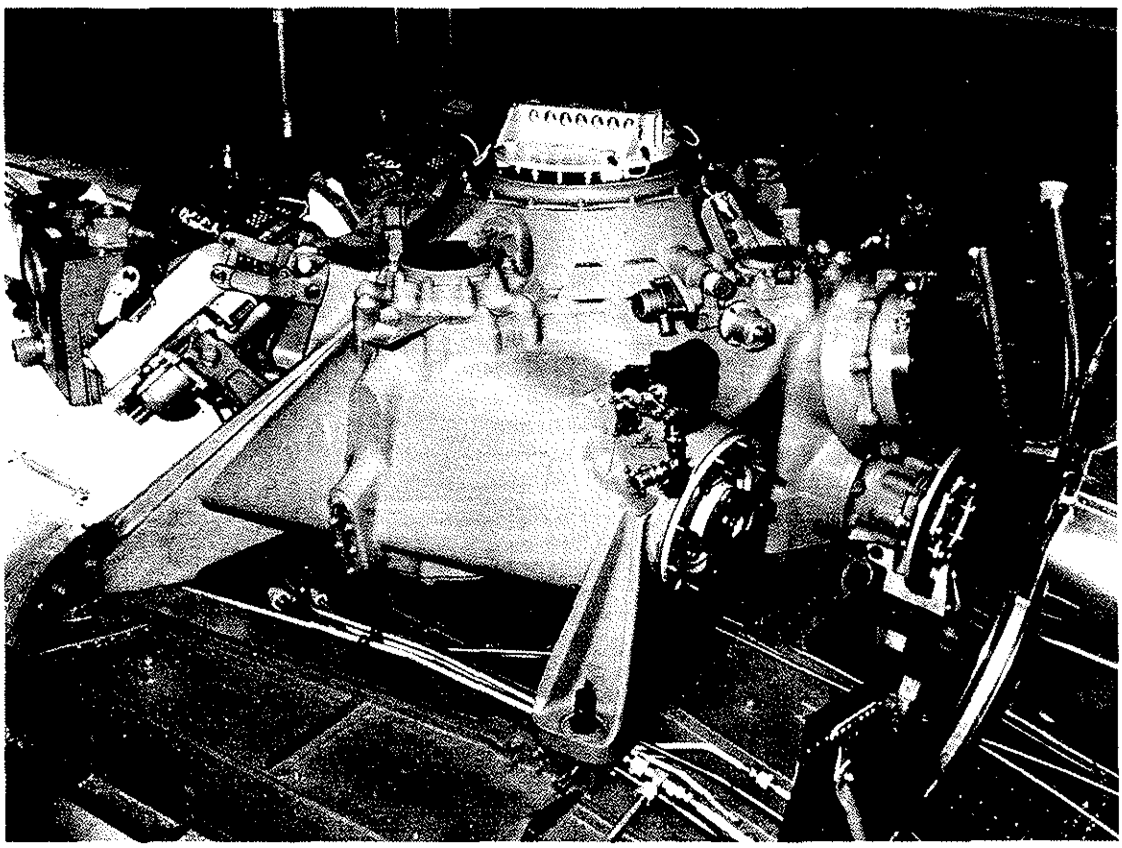


FIG. 2. LYNX GEARBOX VIEWED FROM PORT SIDE

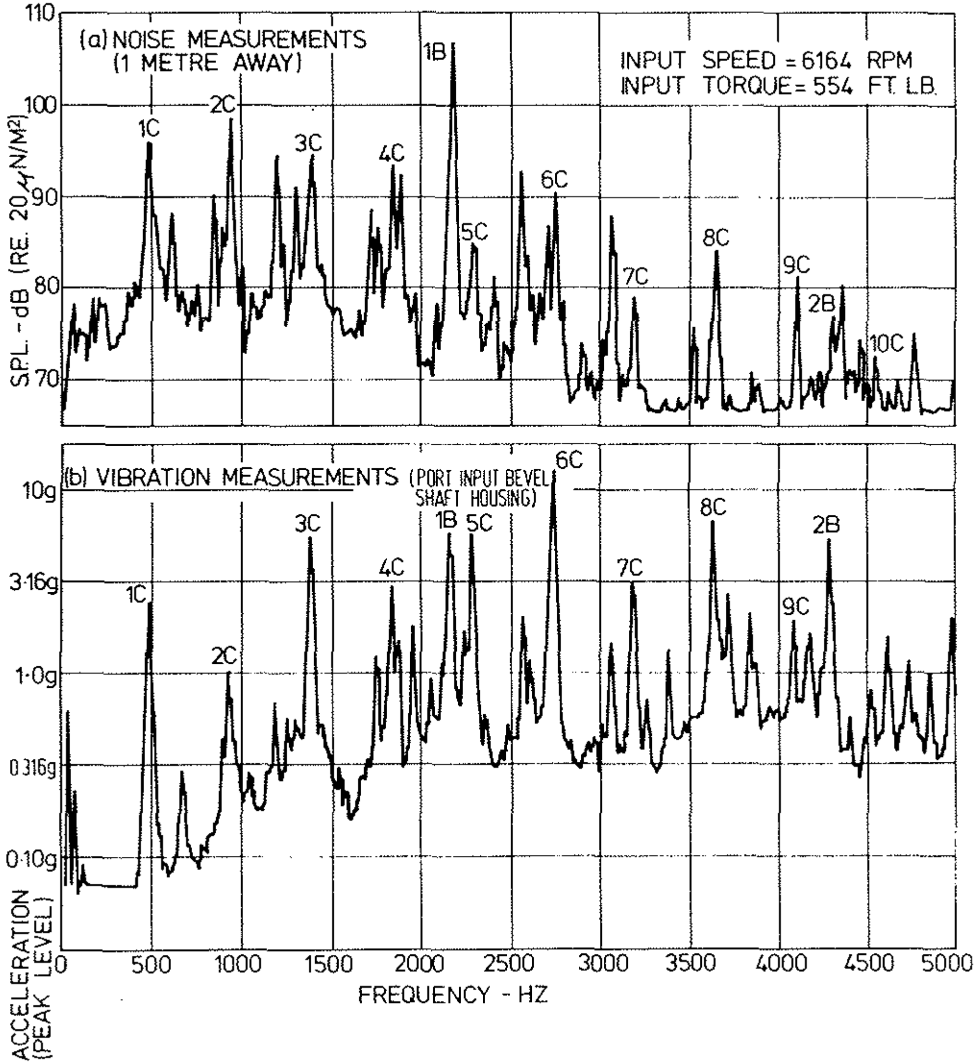


FIG. 3. NOISE & VIBRATION SPECTRA MEASURED ON LYNX GEARBOX RIG

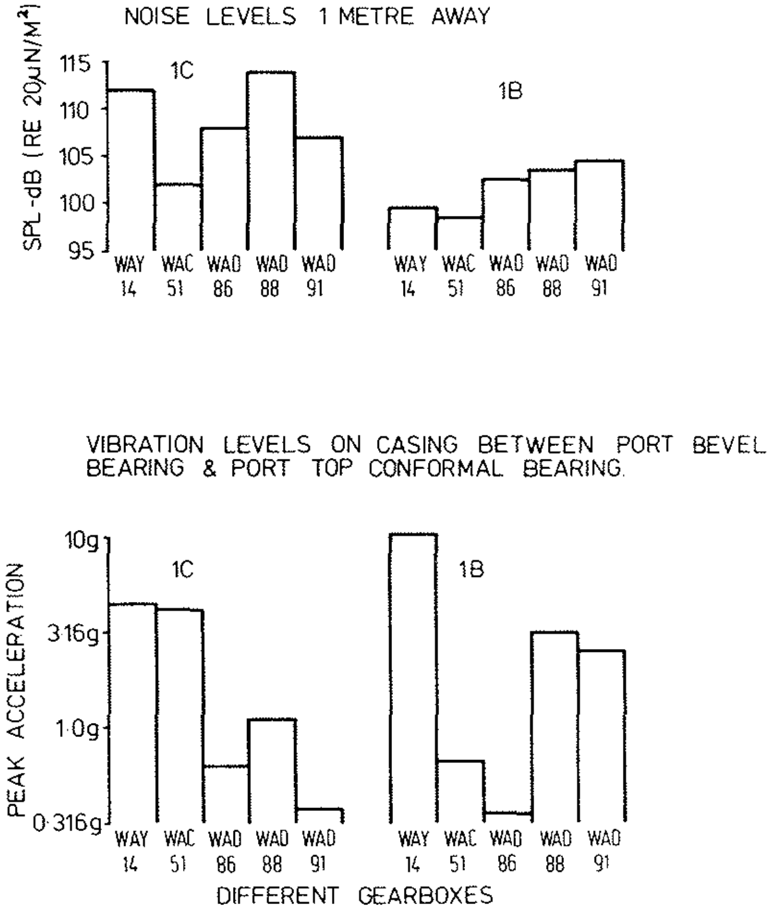
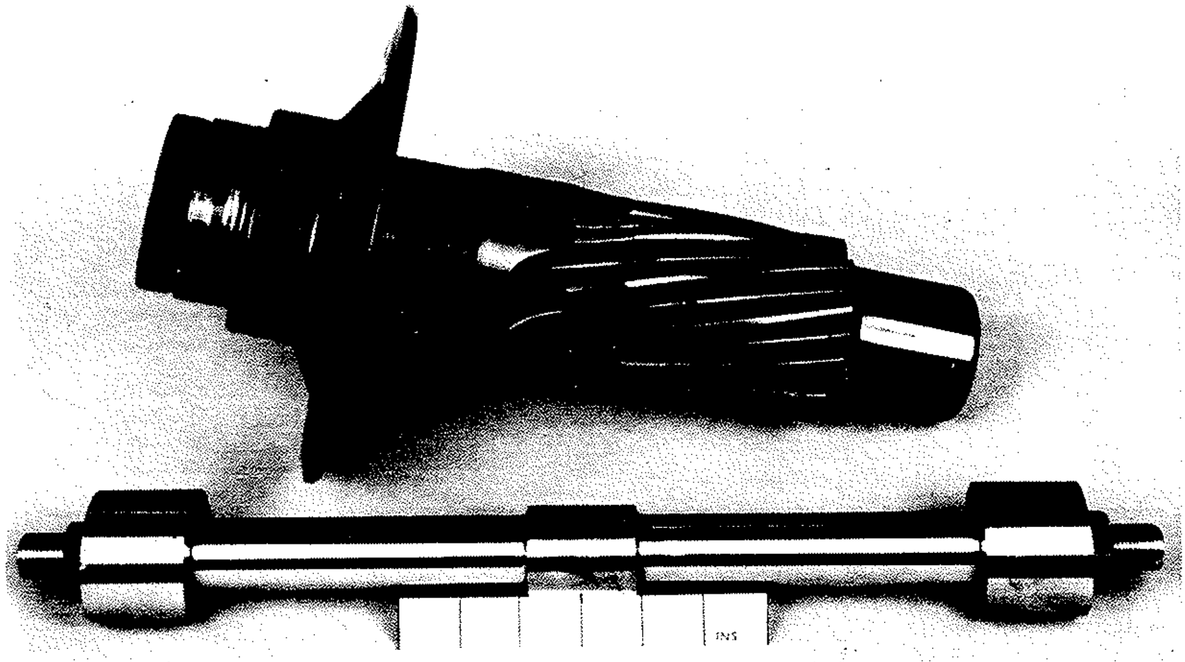


FIG. 4. NOISE & VIBRATION LEVEL VARIATIONS BETWEEN LYNX GEARBOXES FROM RIG DATA



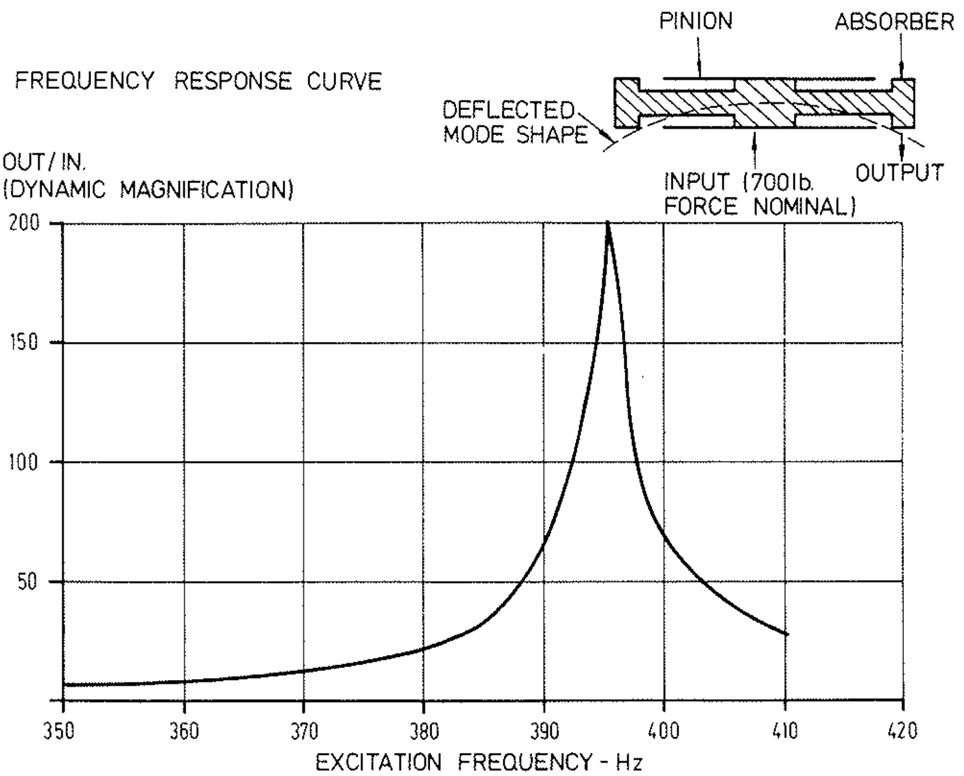


FIG. 7. CONFORMAL PINION ABSORBER AND FREQUENCY RESPONSE CURVE

FIG. 5(a). TOOTH CONTACT PATTERNS FOR STAGES 1 & 3 INPUT BEVEL PINIONS

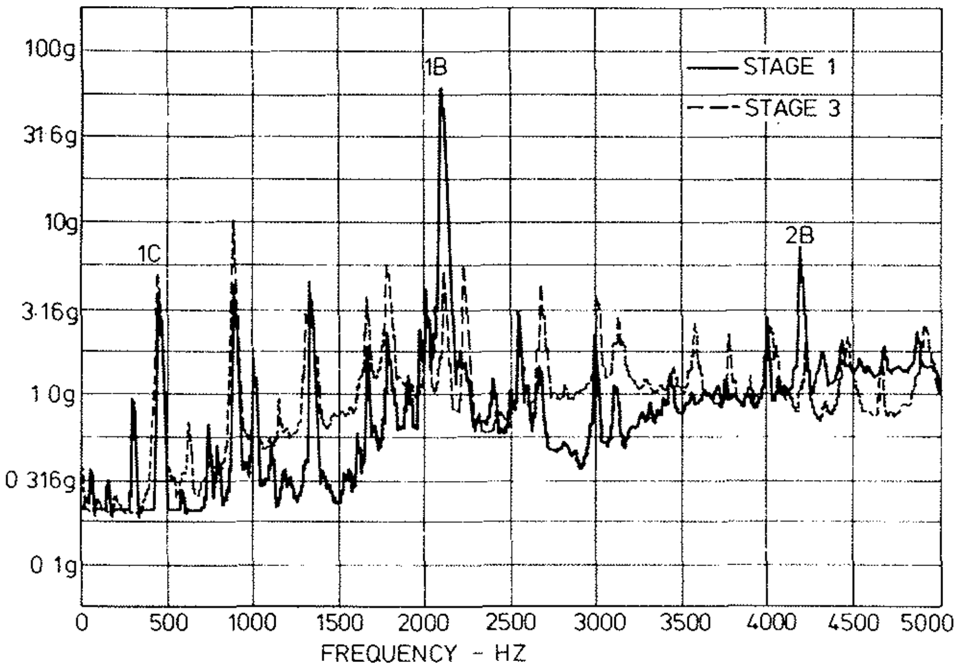


FIG. 5(b). COMPARISON OF VIBRATION SPECTRA FOR DIFFERENT INPUT BEVEL TOOTH CONTACT PATTERNS ON CASING BETWEEN STARBOARD BEVEL BEARING & STARBOARD TOP CONFORMAL BEARING

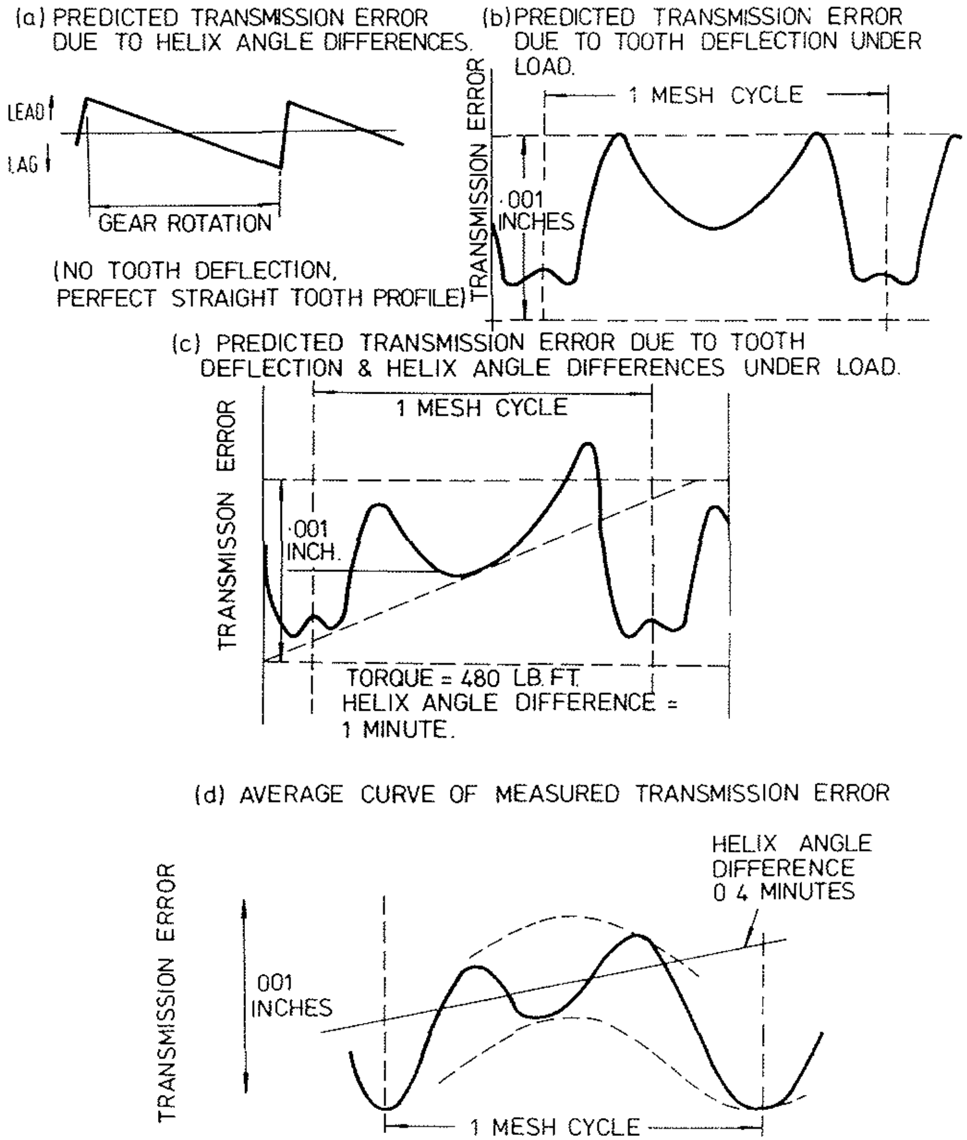


FIG. 6. COMPARISON OF MEASURED & PREDICTED TRANSMISSION ERROR CURVES FOR LYNX CONFORMAL TOOTH MESHING

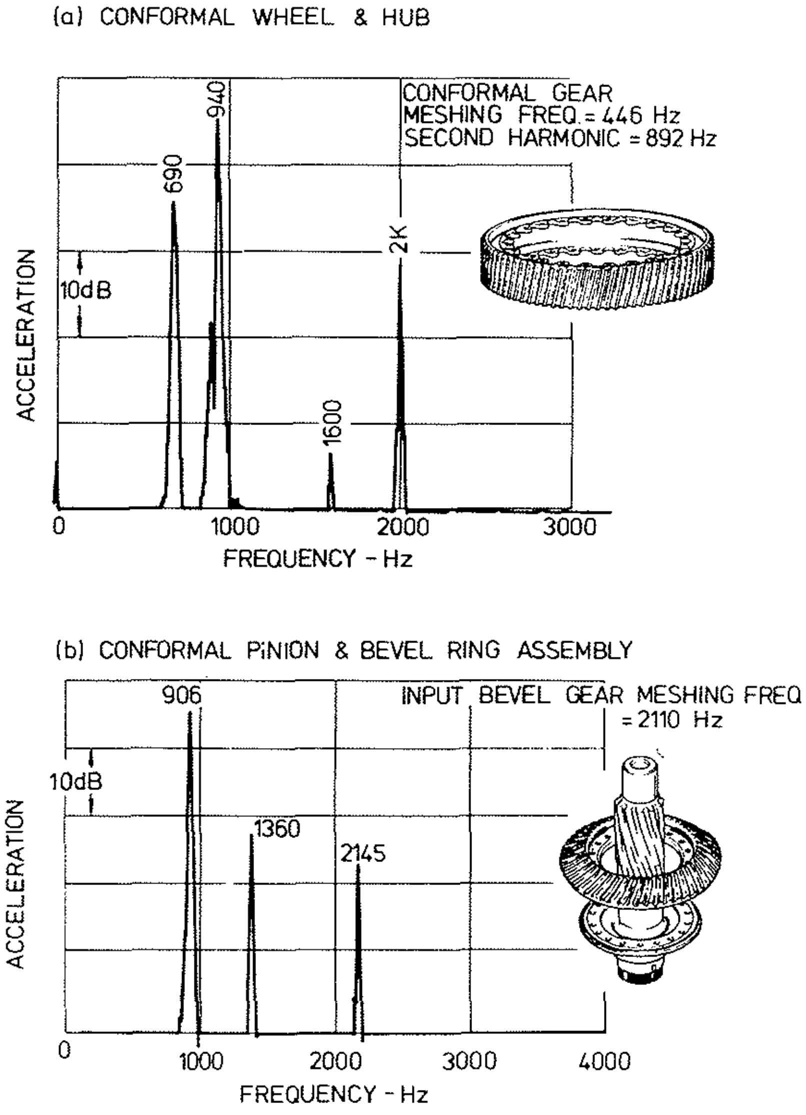


FIG. 8. NATURAL FREQUENCY DETERMINATION OF LYNX GEARBOX COMPONENTS

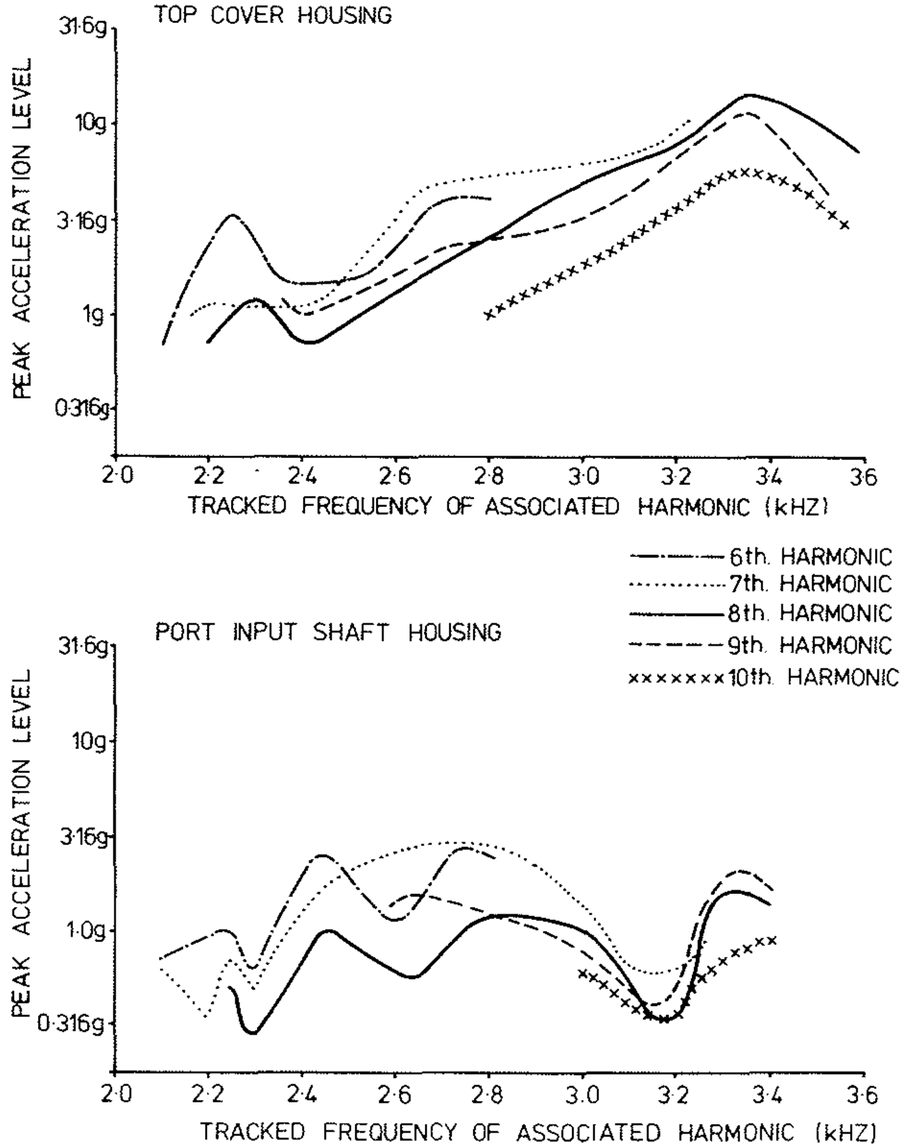


FIG. 9. R.P.M. TRACKING OF LYNX CONFORMAL GEAR MESHING VIBRATION

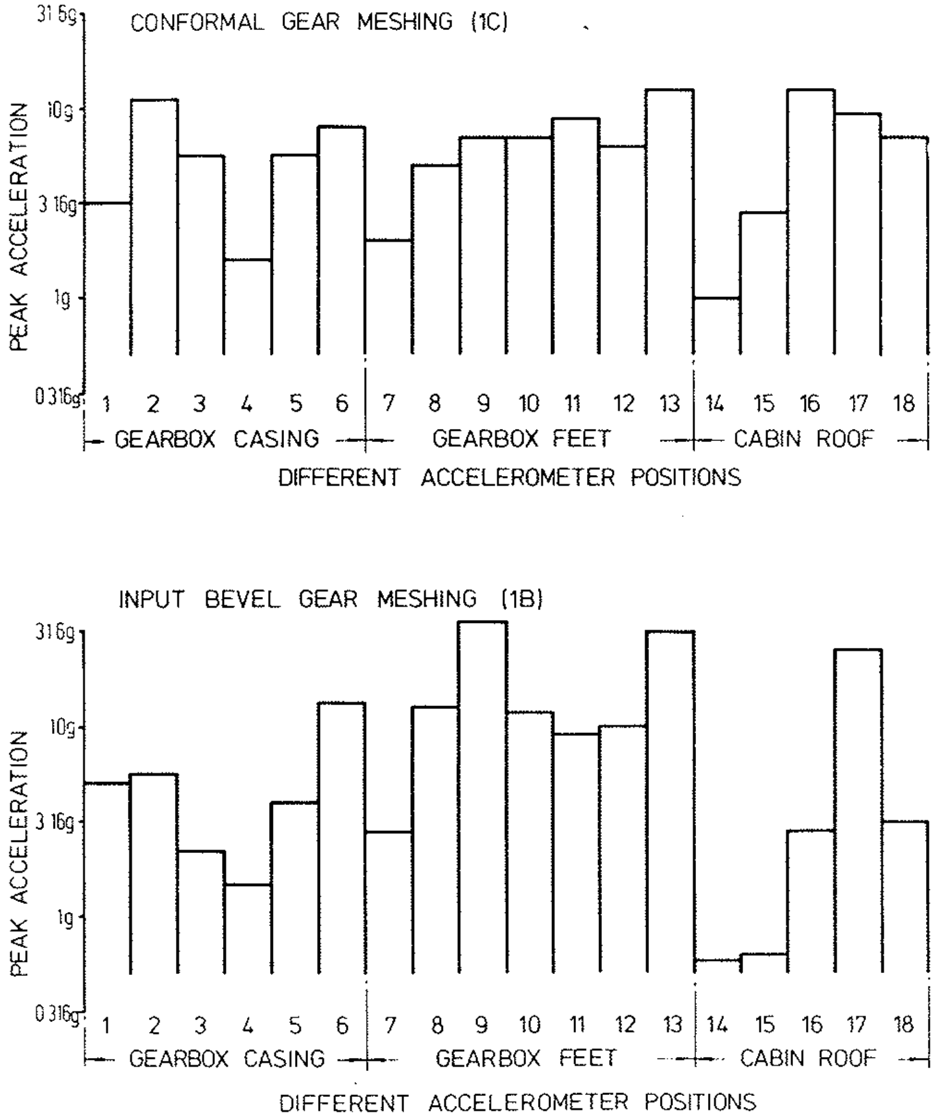


FIG. 10. COMPARISON OF IN FLIGHT VIBRATION LEVELS AT SEVERAL POSITIONS ON THE LYNX STRUCTURE

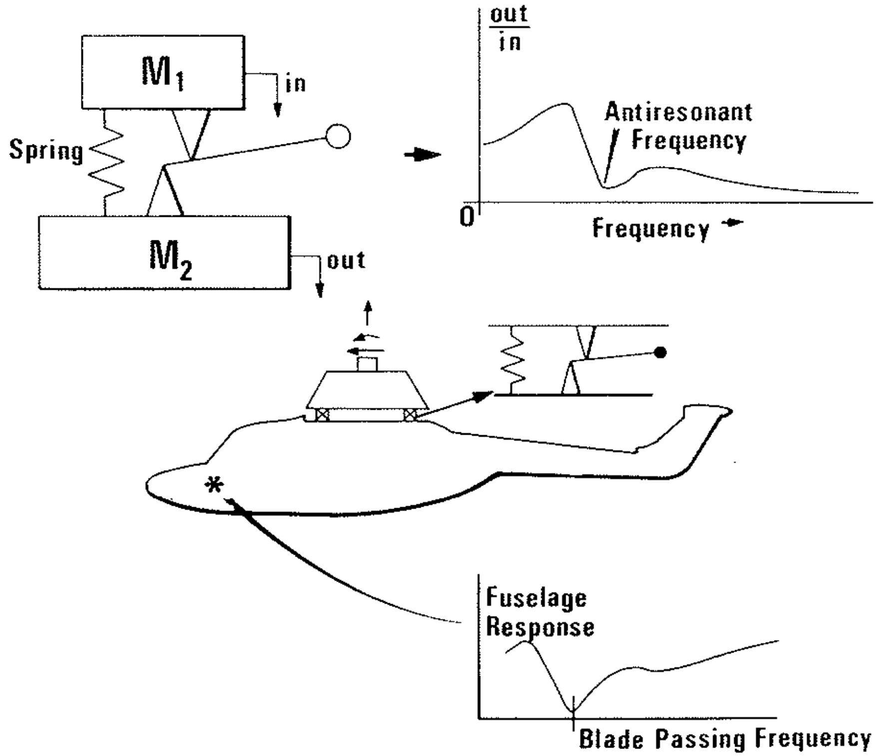


FIG. 11. ISOLATORS APPLIED TO A HELICOPTER