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Comparison of Analysis and Experiment for Gearbox Noise

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COMPARISON OF ANALYSIS AND EXPERIMENT FOR GEARBOX NOISE

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ABSTRACT

Low-contact-ratio spur gears were tested in the NASA gear-noise rig to study the noise radiated from the top of the gearbox. Experimental results were compared with a NASA acoustics code to validate the code for predicting transmission noise.

The analytical code is based on the boundary element method (BEM) which models the gearbox top as a plate in an infinite baffle. Narrow-band vibration spectra measured at 63 nodes on the gearbox top were used to produce input data for the BEM model. The BEM code predicted the total sound power based on this measured vibration.

The measured sound power was obtained from an acoustic intensity scan taken near the surface of the gearbox at the same 63 nodes used for vibration measurements. Analytical and experimental results were compared at four different speeds for sound power at each of the narrow-band frequencies over the range of 400 to 3200 Hz. Results are also compared for the sound power level at meshing frequency plus three sideband pairs and at selected gearbox resonant frequencies. The difference between predicted and measured sound power is typically less than 3 dB with the predicted value generally less than the measured value.

INTRODUCTION

A major source of helicopter cabin noise (which has been measured at over 100 dB sound pressure level) is the gearbox. Reduction of this noise is a NASA and U.S. Army goal. Gearmesh noise is typically in the frequency range of 1000 to 3000 Hz, a range important for speech. A requirement for the Advanced Rotorcraft Transmission project is a 10-dB noise reduction compared to current designs. A combined analytical and experimental effort is underway to study effects of design parameters on noise production. An important part of the project is performing experiments in the NASA Lewis gear-noise facility to verify analytical codes.

Various analytical codes are available to simulate gear dynamics, vibration, and noise. These include DANST (dynamic analysis of spur-gear transmissions) (Lin et al., 1987a, 1987b, and 1989),

GEARDYN (gear dynamics) (Boyd and Pike, 1987 and Pike, 1981); and GRD (geared rotor dynamics) (Kahraman et al., 1990). Finite element methods may also be used to predict the structural vibration properties of a gearbox.

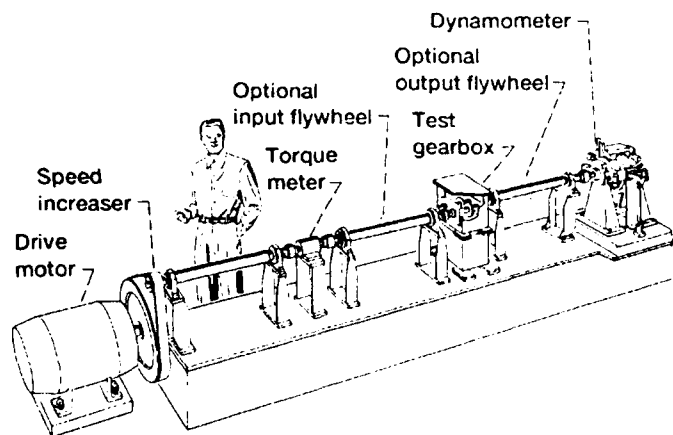
The acoustic modeling code used to predict the noise produced by gear vibration is the boundary element method for acoustic prediction (BEMAP). BEMAP uses the technique of boundary element analysis to predict gearbox noise (or other structural noise) from vibration data (Seybert et al., 1990).

Part of the effort of NASA gear-noise research is directed toward validating computer codes so that the codes may be used as design tools. Earlier work in this project produced dynamic load and stress data for validation of a popular NASA gear dynamics code (Oswald et al., 1991) and dynamic friction measurements (Rebbechi et al., 1991). A related project provided validation data for BEMAP code predictions for radiated sound power using a simple laboratory apparatus (Seybert et al., 1992).

The goal of this effort was to provide experimental data from an actual gearbox to verify BEMAP predictions. Results presented include narrow-band spectra comparing measured and computed sound power, and sound power comparisons at mesh frequency and at several modal frequencies.

TEST FACILITY AND TEST SPECIMENS

The facility used for these tests is the NASA Lewis gear-noise rig (Fig. 1). This rig consists of a single-mesh gearbox driven by a 150-kW (200-hp) variable speed electric motor. An eddy-current dynamometer loads the output shaft. The gearbox can be operated at speeds up to 10 000 rpm input or 6000 rpm output. The rig was built to carry out fundamental studies of gear noise and the dynamic behavior of gear systems. It is designed to allow testing of various configurations of gears, bearings, dampers, and supports. To reduce unwanted reflection of noise, acoustic-absorbing foam baffles covered the test cell walls, floor, and other nonmoving surfaces. The material used attenuates reflected sound by at least 20 dB for frequencies of 500 Hz and above.



(a) Layout.



(b) Test gearbox showing measurement grid and RAIMS robot with sound intensity probe.

Figure 1.—NASA gear-noise rig.

A poly-V belt drive was used as a speed increaser between the motor and gearbox input shaft. A soft coupling was installed on the input shaft to reduce input torque fluctuations, which were caused by nonuniformity at the belt splice.

The test gears were identical spur gears (operating at 1:1 ratio) machined to AGMA Class 15 accuracy. The gears have linear tip relief chosen for optimum operation at a design load. Test gear parameters are shown in Table I.

TABLE I.—TEST GEAR PARAMETERS

Spur gear type	Standard involute, full-depth tooth
Number of teeth	28
Module, mm (diametrical pitch in. ⁻¹)	3.175 (8)
Face width, mm (in.)	6.35 (0.25)
Pressure angle, deg	20
Nominal (100 percent) torque, N-m (in.-lb)	71.77 (635.25)
Theoretical contact ratio	1.64
Modification amount, mm (in.)	0.018 (0.0007)
Modification start, deg	24.5

A 63-node measurement grid was marked out on the top of the gearbox. The grid covers an area 228 by 304 mm (9 by 12 in.) centered on the 286- by 362-mm (11.25- by 14.25-in.) top. The gearbox with its measurement grid and some instrumentation are shown in Fig. 1(b).

INSTRUMENTATION AND TEST PROCEDURE

An experimental modal test was performed to determine the modes of vibration and natural frequencies of the gearbox top. An 800-line, 2-channel dynamic signal (FFT) analyzer collected frequency-domain data. Commercial modal software running on a personal computer was used for the analysis. The modal tests were performed with the top installed on the gearbox and with the gearbox heated to operating temperature. The structure was excited sequentially at each of the 63 nodes using a load-cell-equipped modal hammer to measure excitation forces. The response was measured with a small piezoelectric accelerometer mounted at a reference location near the center of the gearbox top.

Gearbox vibration was measured by two piezoelectric accelerometers with a frequency range of 5 to 10 000 Hz. A fixed accelerometer was located at the reference node and a movable accelerometer was used for measurements at each of the 63 nodes on the top.

The analyzer captured the magnitude spectrum for the movable accelerometer and the phase spectrum between the accelerometers. The data were transferred to a desktop computer and converted to real and imaginary components of velocity. These velocity components were used to produce an input data file for BEMAP.

Immediately after the accelerometer scan (under stable operating conditions), an acoustic intensity scan was performed with the aid of a computer-controlled robot designated RAIMS (robotic acoustic intensity measurement system). The RAIMS software (1) commanded the robot to move an intensity probe over a prescribed measurement grid; (2) recorded acoustic intensity spectra in the analyzer for each node of the grid; and (3) transmitted the spectra to the computer for storage on disk. RAIMS is described more completely in Flanagan and Atherton (1985) and in Atherton et al. (1987).

The acoustic intensity probe consists of a pair of phase-matched 12-mm (0.5-in.) microphones mounted face-to-face with a 12-mm (0.5-in.) spacer. The probe has a frequency range (+1 dB) of 125 to 5000 Hz. Measurements were made at a distance of 75 mm (3 in.) between the acoustic center of the microphones and the gearbox top.

The intensity spectra from each of the 63 measurement locations collected at each operating condition were averaged, then multiplied by the radiation area to compute an 800-line sound power spectrum. The radiation area was assumed to be the area of the grid plus one additional row and column of elements or 0.0910 m². The actual area of the top is 0.1034 m². The measurement grid did not extend completely to the edges of the gearbox top because (1) the edge of the top was bolted to a stiff mounting flange which would not allow much movement and (2) measurements taken close to the edge of the top would be affected by noise radiated from the sides of the box. (To keep the experimental effort reasonable, neither noise nor vibration measurements were attempted on the gearbox sides.)

Vibration and intensity measurements were made at four test conditions: 3000, 4000, 5000, and 6000 rpm and at a constant 94 percent of the reference torque level of 71.8 N-m (635 in.-lb)

During measurements, speed was held to within ± 8 rpm and torque to ± 1 N-m. All frequency-domain data were recorded over the bandwidth 0 to 3200 Hz.

ANALYTIC MODEL

The BEMAP program is based on the boundary element method (BEM) (Seybert et al., 1985, 1986, 1987, 1990, 1991, and 1992). BEMAP can compute several acoustic parameters on, inside, or outside the structure of interest including sound intensity, sound pressure level, sound radiation efficiency, and the total sound power. For this work, a public-domain PC version of BEMAP was used. This version is called BEMAP100 and will be available through the Computer Software Management and Information Center (COSMIC) at the University of Georgia. BEMAP100 (referred to here as BEMAP) is limited to 100 nodes and 70 elements.

Using BEMAP, the gearbox top was modeled as a plate in an infinite baffle. Inputs to BEMAP include the geometry of the structure and, for each mode, the vibration frequencies and the nodal velocity components. Hence, for 63 nodes and 800 frequencies, 50 400 pairs of velocity values were required. BEMAP computed an 800-line sound power spectrum for the gearbox top based on the accelerometer data. Several specialized computer programs were developed for this project. These programs collect and transform measured data, produce input files, extract the required output data (from a large BEMAP output file), and plot spectra comparing BEMAP predictions with measured sound power.

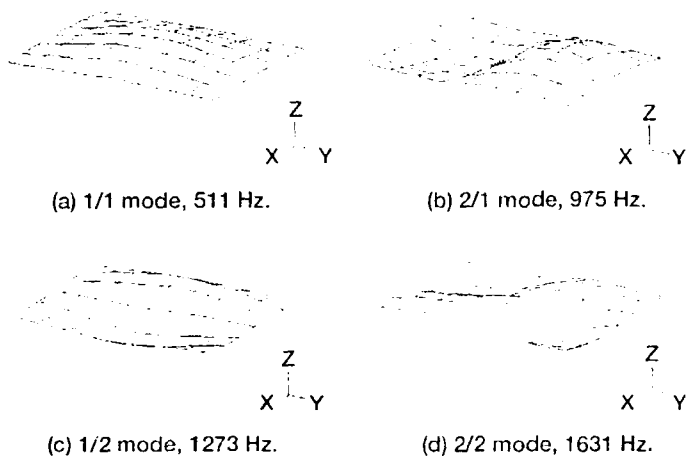


Figure 2.—Gearbox vibration modes.

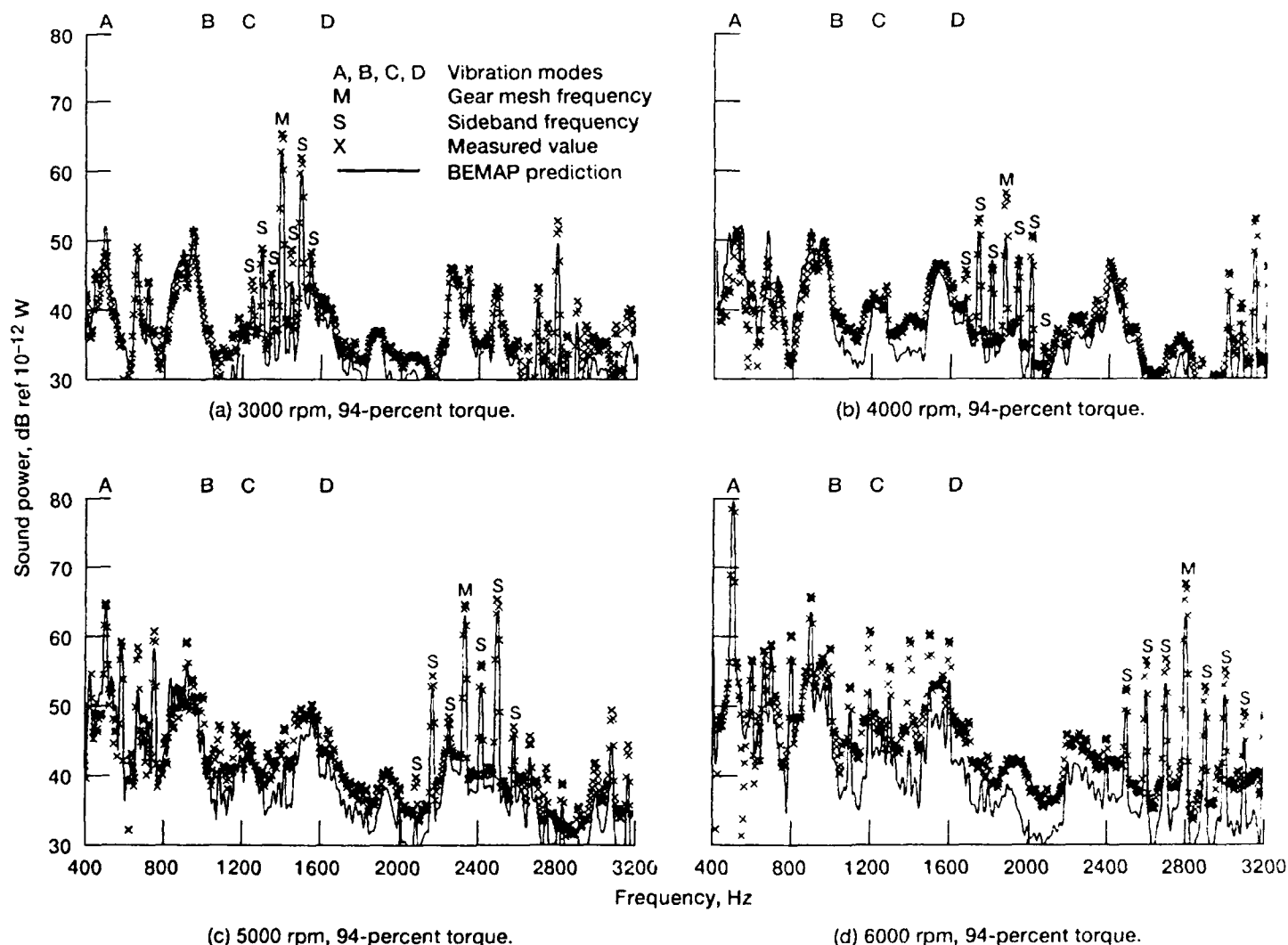


Figure 3.—Comparison of BEMAP prediction and measurement for sound power.

The 63-node grid (7 nodes x 9 nodes) produced 48 quadrilateral linear elements (6 elements x 8 elements). The BEMAP model requires that the spacing between nodes be not more than $1/4$ of the acoustical wavelength. The nodal spacing was 38 mm, hence the highest frequency meeting this requirement is 2250 Hz.

RESULTS AND DISCUSSION

The first four vibration modes (from modal tests) are shown in Fig. 2. Seybert et al. (1991) reported modes at slightly different frequencies. This difference is due to different clamping conditions for the top. A soft cork gasket under the gearbox top has been replaced by a rubber O-ring in a machined groove. (The O-ring allows stiffer clamping.)

Measured and computed sound power spectra for the four test conditions are shown in Figs. 3(a) to (d). The gearmesh frequency, three pairs of sidebands around the mesh frequency, and the frequencies of the first four modes are indicated in the figure. The measured sound power is generally about 3 dB higher than predicted values. The spectra from lower speeds show better correlation than those from higher speeds.

The acoustic data used for the spectra in Fig. 3 were used to compute the mesh sound power. The mesh sound power is defined here as the sum of the sound power at gearmesh frequency plus at three pairs of sidebands (i.e., the sum of seven values). The measured and computed mesh sound power are compared in Fig. 4.

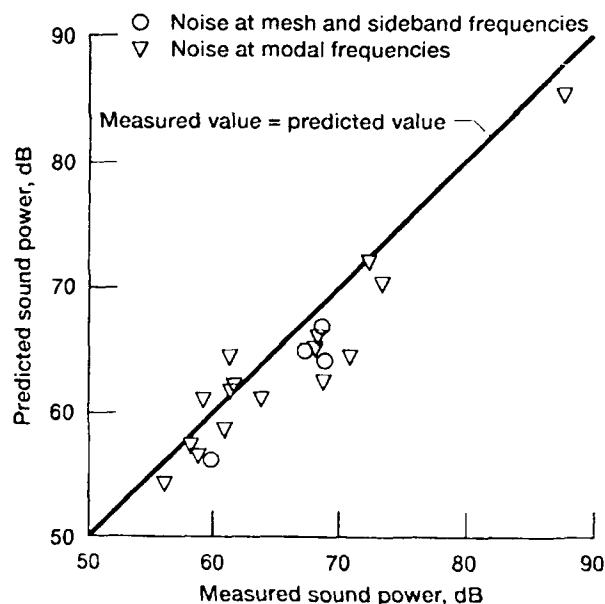


Figure 4.—BEMAP predictions for sound power compared to experimental data.

Also compared in Fig. 4 are values of the sound power at modal resonance frequencies. (These values are computed over an 80-Hz bandwidth centered at each of the first four modal resonance frequencies.) The absolute difference between measured values and BEMAP predictions averaged 2.4 dB. Measured values were typically higher than predicted values.

At 3000 rpm, the 0 to 3200 Hz measurement bandwidth contains the second harmonic of gearmesh frequency (2800 Hz) as well as the fundamental (1400 Hz). A noise peak at the second harmonic frequency can be seen (but is not marked) in the traces shown in

Fig. 3(a). At speeds higher than 3000 rpm, all higher harmonics lie outside the bandwidth. For consistency of analysis, no higher harmonics were included in the calculated mesh sound power.

The vibration data used by BEMAP were taken only from the gearbox top. This limitation was imposed by the 100-node limitation and also to keep the effort required for collecting and processing the measurement reasonable. (It required about 2 hr of gearbox operating time to collect data for each operating condition. The 800-frequency BEMAP analysis took 11 hr of computing time on a desktop computer.)

Using BEMAP with the measured vibration data provides a prediction for noise radiated solely from the top of the gearbox. The measured sound power also includes some low-frequency noise refracted or reflected from the motor, shafting, or other noise sources. Low-frequency noise would be expected to contaminate the measurements because the acoustic baffles decline in absorption ability at low frequencies and because low-frequency noise is more likely to be refracted to the intensity probe. Because of this low-frequency noise, frequencies below 400 Hz are not shown in the spectra of Fig. 3. Even somewhat above 400 Hz, there were a few frequency bands in which the sound intensity flow direction was toward the gearbox; hence, the intensity was negative. These frequencies show up as gaps in the spectra.

The coarseness of the measurement grid is a source of error for frequency measurements above 2250 Hz. Other noise sources (such as the motor and the shafting) contribute to error primarily at low frequencies (below approximately 400 Hz). Noise from the sides of the gearbox will make the measured sound power higher than the BEMAP predicted value. These errors are caused by limitations of the measurement and not by the analysis code itself.

CONCLUSIONS

Measurements of gearbox noise were made in the NASA gear-noise rig for comparison with predictions from the acoustic computer code BEMAP. The computer predictions are based on measured vibration data. The analytical and experimental results for sound power were compared for four different operating speeds to validate the code as a design tool for predicting radiated transmission noise. The following conclusions were obtained.

1. The spectral traces for computed sound power are similar to spectra of measured sound power. This indicates that the analysis successfully predicts the sound radiation properties of the gearbox.
2. The predicted and measured sound power were compared at gearmesh frequencies and at modal resonance frequencies. These values agree, within an average error of 2.4 dB, with the predicted sound power generally less than the measured value.

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