Automated Acoustic Intensity Measurements and the Effect of Gear Tooth Profile on Noise

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Abstract:

Acoustic intensity measurements were made at NASA Lewis Research Center on a spur gear test apparatus. The measurements were obtained with the Robotic Acoustic Intensity Measurement System developed by Cleveland State University. This system provided dense spatial positioning and was calibrated against a high quality acoustic intensity system. The measured gear noise compared gearsets having two different tooth profiles. The tests evaluated the sound field of the different gears for two speeds and three loads. The experimental results showed that gear tooth profile had a major effect on measured noise. Load and speed were found to have an effect on noise also.

Introduction

The NASA Lewis Research Center investigated the effect of tooth profile on the acoustic behavior of spur gears through experimental techniques. The tests were conducted by Cleveland State University (CSU) in NASA Lewis' spur gear testing apparatus. Acoustic intensity (AI) measurements of the apparatus were obtained using a Robotic Acoustic Intensity Measurement System (RAIMS). This system was developed by CSU for NASA to evaluate the usefulness of a highly automated acoustic intensity measurement tool in the reverberant environment of gear transmission test cells.

The purpose of this article is to report on the results of noise tests of two different spur gear profile configurations which included a total of 12 different speed and load conditions. Also, the useful features of an automated acoustic intensity measurement system are demonstrated through the presentation of the test results.

RAIMS

RAIMS consists of a two-channel spectrum analyzer (FFT), a desktop computer, an instrumentation robot arm, a digital control unit for the robot and an acoustic intensity probe as shown in Fig. 1. The computer, analyzer and digital control unit module are connected via an IEEE-488 interface bus to provide computer coordination of the robot and data acquisition system. A description of the components and an evaluation of this automated system have been reported by Flanagan and Atherton.[1,2]

RAIMS measures acoustic intensity by the two microphone techniques, utilizing the imaginary part of the cross-power spectrum. Other researchers[3,4] have used this technique for

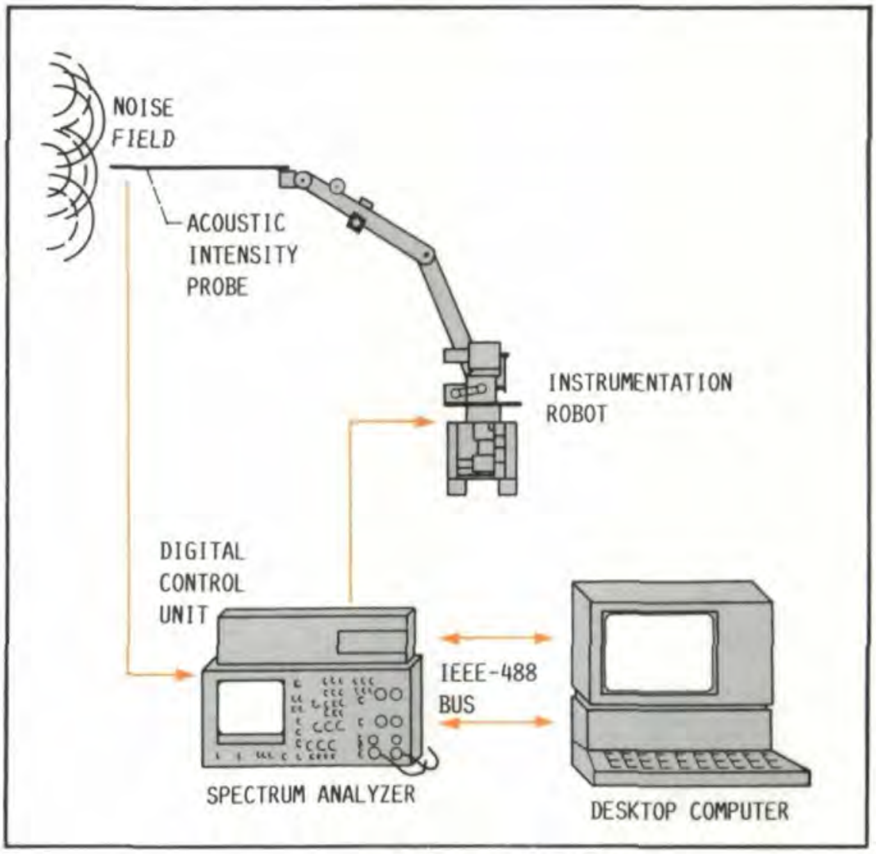

Fig. 1—Schematic of the RAIMS system

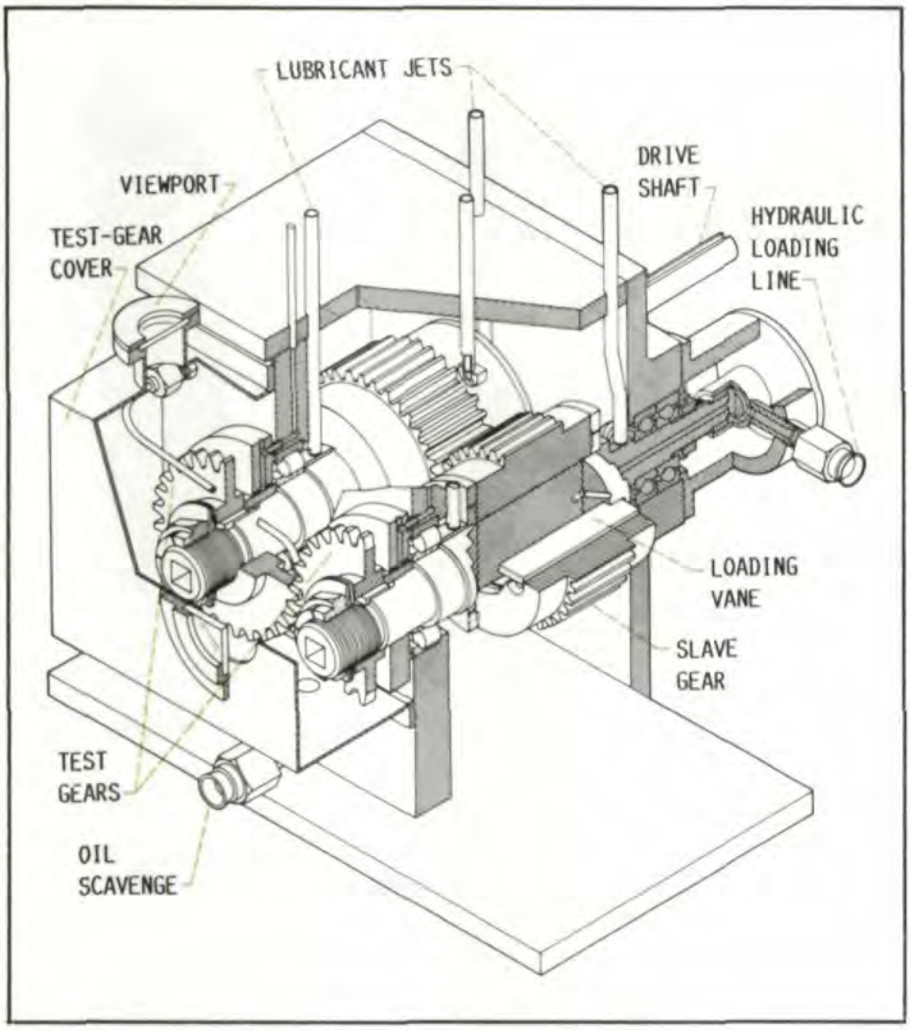

Fig. 2—NASA Lewis Research Center's gear fatigue test apparatus.

acoustic measurements in reverberant environments and have investigated the source and effect of various measurement errors in this method.[5,6] The microphones and the instrumentation consist of high quality, commercially available equipment. Because the acoustic intensity algorithm was programmed in the desktop computer, the system was calibrated against a commercially available, precision system which computes acoustic intensity directly. This process involved the comparison of the acoustic intensity from a noise source by the two systems and provided verification and calibration of RAIMS in the frequency domain.

Acoustic intensity is the net flow of sound power per unit area as measured at a point in space. It contains both magnitude and direction information and is generally presented in the frequency domain to display the frequency content of the intensity vector. Measurements at points which form an inclusive envelope around a noise source can provide information on the source location and total emitted sound power. The total sound power is a useful quantity because it is a characteristic of the noise source and is unaffected by the environment.

The AI emitted from the spur gear test apparatus depends on the nature of the excitation and the manifestation of the surface vibration into the acoustic farfield. Surface mounted accelerometers are frequently used to identify vibration amplitudes, but they cannot characterize the noise field phenomena. Also, accelerometers are limited to measuring the vibration at the attachment point to the structure. At the beginning of testing, it is difficult to determine the accelerometer placement to pick up the most dynamically active points of the structure. Accelerometers on thin, flexible housings can mass load the structure and corrupt the dynamic response as well as the resulting acoustic field. Sound pressure measurements from single microphones can also be biased by the reverberation and acoustical absorption characteristics of the surrounding environment. Because acoustic intensity is a vector quantity and does not measure standing waves, it has been shown to be a viable technique to characterize radiated sound power and identify acoustic sources in reverberant environments.

Apparatus and Test Hardware

The experimental noise tests were performed in the NASA Lewis Research Center gear fatigue test apparatus shown in Fig. 2. The test apparatus is of the four-square power loop type with the torque preload supplied by a rotary hydraulic actuator built into one of the shafts. An electric motor attached at the extension of the second shaft provides the power to drive the system. Changes in load conditions are made by adjusting the hydraulic pressure on the actuator loading vanes. Speed changes are made by exchanging sheaves. The slave gears are simply supported by the bearings and the test gears are overhung cantilever fashion at the front of the apparatus. The two bearings are supported by vertically rigid mounting plates. A metal cover with a transparent viewport in the center encloses the test gears.

Two sets of gears were tested. The first set consisted of the NASA standard fatigue tester gears whose dimensions are shown in Table 1. This set of gears has tip relief starting at about 27° roll angle, which is just before the start of single tooth contact during mesh. As the roll angle increases, the tooth profile has a linear deviation from the true involute. The teeth of the second set of gears had tooth profile modifications consisting of slightly more tip relief and the addition of root relief. The gear tooth profiles were measured on an involute checking machine and their traces are shown in Fig. 3.

Test Program

The acoustic intensity measurement program was carried out at the operating conditons indicated in Table 2. Implementation of the 12 tests was accomplished by adjusting the pressure to the hydraulic actuator of the test rig and ex-

(continued on page 26)

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(continued from page 23)

changing the sheave diameters of the input shaft. For the two operating speeds of the test, the meshing frequencies of the 28 teeth test gear and 35 teeth slave gear are indicated in Table 3.

To carry out the acoustic tests, RAIMS was placed in front of the test rig facing the test gear cover. The robot was then programmed to measure AI in the four planes to the left, front, right, and top of the cover in square patterns of 2.54 cm extent as indicated in Fig. 4. The tip of the acoustic intensity probe was held between 5 to 10 cm from the surface of the test rig. A total number of 163 scan positions were used.

Taking the average of 32 measurements, calculating the acoustic intensity, storing the data, and positioning of the robot required about one min. This automated sequence was repeated for all of the 163 spatial points of the total scan. The 163 points did not represent a complete enclosure scan. Consequently, the spatial integration of the acoustic intensity represents only a partial measurement of the total sound power. A complete closure scan was not possible due to piping obstructions and limited access space. The partial sound power is still a useful measurement for the comparison test of the two gear pairs.

Test Results

The automated test program produced a great deal of data, portions of which are presented in the following graphs. Fig. 5 shows the AI spectrum at one of the 163 points for the standard test gears operating at 10 160 rpm and 1615 N tangential load.

The spectrum of Fig. 5 shows three regions of high amplitudes which are present (to a greater or lesser degree) in all of the AI spectrums from each of the 163 measurement points. The first region extends from 500 to 1500 Hz, and is characterized by several peaks that are separated by the operating speed of 170 Hz. The high amplitudes in this region are attributable to the excitation from the bearing passing frequencies which may be amplified by the test gear cover.

The second region extends from 2800 to 3500 Hz. The amplitudes in this region are caused by the coincidence of the torsional natural frequency predicted by Mark[7] to be at 3500 Hz. Region three extends from 4300 to 6000 Hz, and

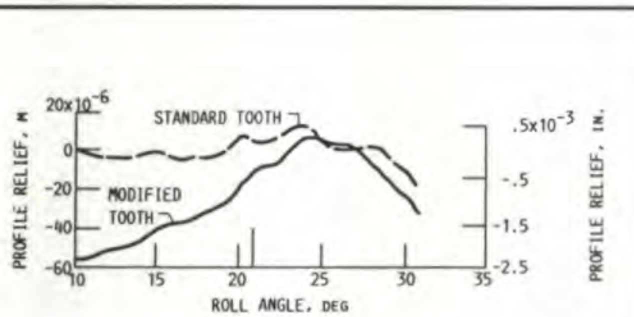

Fig. 3 – Tooth profiles of test gears (zero profile relief corresponds to a perfect involute profile).

it represents the contribution from the fundamental meshing frequencies of the test and slave gears. The sidebands are caused by the errors of the gear tooth profiles.

The fundamental and first harmonic meshing frequencies

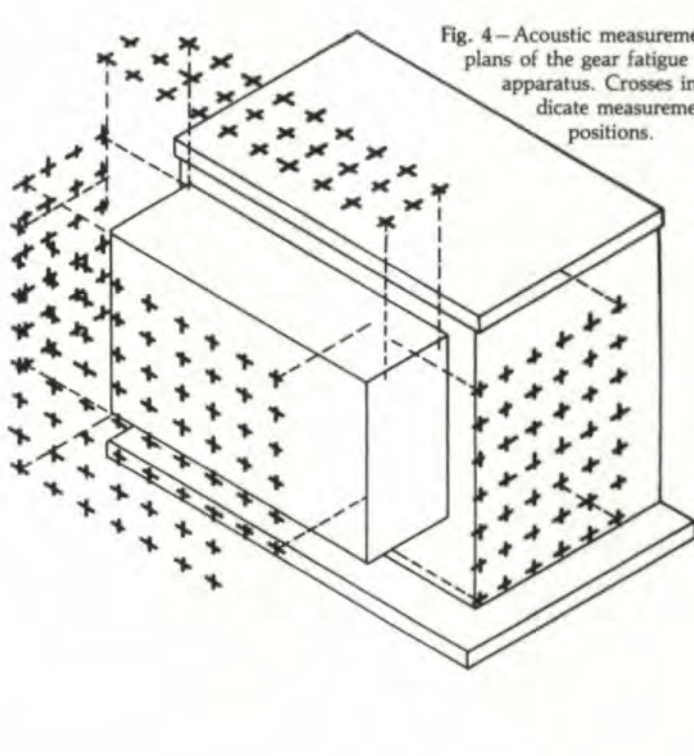

Fig. 4 – Acoustic measurement plans of the gear fatigue test apparatus. Crosses indicate measurement positions.

Table 1 — Spur Gear Data

Test gear — standard:	
Number of teeth	28
Diametral pitch	8
Whole depth, cm (in.)	0.762 (0.300)
Addendum, cm (in.)	0.318 (0.125)
Pressure angle, deg	20
Pitch diameter, cm (in.)	8.890 (3.500)
Tooth width, cm (in.)	0.635 (0.250)
Outside diameter, cm (in.)	9.525 (3.750)
Root fillet, cm (in.)	0.102 to 0.152 (0.04 to 0.06)
Measurement over pins, cm (in.)	9.603 to 9.630 (3.7807 to 3.7915)
Pin diameter, cm (in.)	0.549 (0.216)
Backlash, cm (in.)	0.0254 (0.010)
Tip relief, cm (in.)	0.001 to 0.0015 (0.0004 to 0.0006)
Test gear — Modified: see Fig. 3 for modifications	
Slave gear	
Number of teeth	35
Diametral pitch	10
Tooth width, cm (in.)	3.81 (1.5)

Table 2 — Load and Speed Conditions

Test number	Test gear	Tangential load, N (lb)	Speed, rpm
1	Standard	1615 (363)	10 160
2		1615 (363)	7 470
3		1292 (290)	10 160
4		1292 (290)	7 470
5		969 (218)	10 160
6		969 (218)	7 470
7	Modified	1615 (363)	10 160
8		1615 (363)	7 470
9		1292 (290)	10 160
10		1292 (290)	7 470
11		969 (218)	10 160
12		969 (218)	7 470

Table 3 — Meshing Frequencies

Operating speed, rpm	Test gear, Hz	Slave gear, Hz
10 160	4741	5927
7 470	3486	4358

of the test gear are at 4741 and 9482 Hz. The fundamental mesh frequency of the slave gear is at 5927 Hz. Note the strong presense of the slave gears. This is not unexpected, since the test apparatus is used for fatigue testing and the slave gears are lightly loaded.

An indication of the housing dynamic behavior can be obtained by plotting lines of constant intensity at a given frequency for a complete measuring plane. Fig. 6 shows such an iso-intensity plot for the right side of the spur gear testing apparatus using the same operating condition as in Fig. 5 at 5927 Hz. This planar representation shows concentration of high and low amplitudes across the plane which appear to derive their origin from the structural dynamics of the housing.

Figs. 7 to 10 are the results when, at a given load/speed condition, the acoustic sound power of the scanned area is

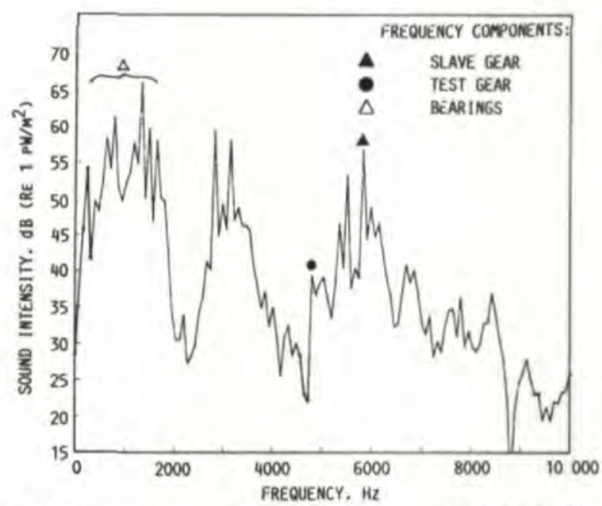

Fig. 5 — Frequency spectrum of standard gear in middle of right side — at 10 160 RPM and 1615 N load.

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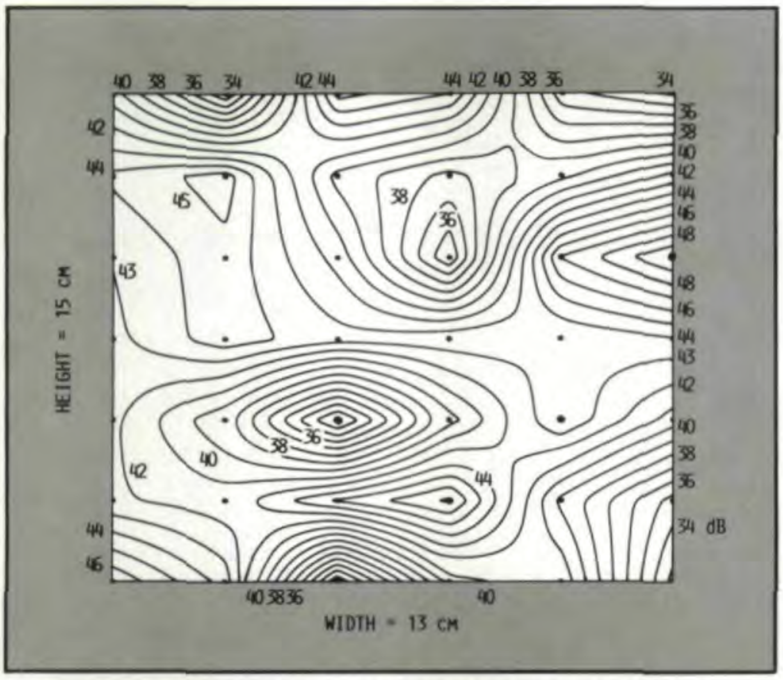

Fig. 6 – Iso-intensity plot of the right scan at 5927 Hz for the standard gears, 10 160 RPM, and 1615 N tangential load.

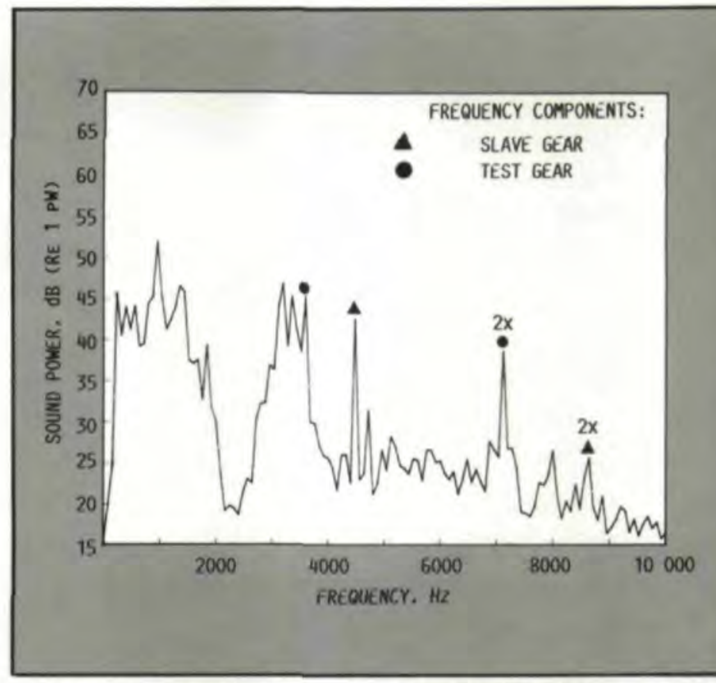

Fig. 8 – Sound power for the four planes investigated (conditions: standard gears, 7470 RPM, 1615 N).

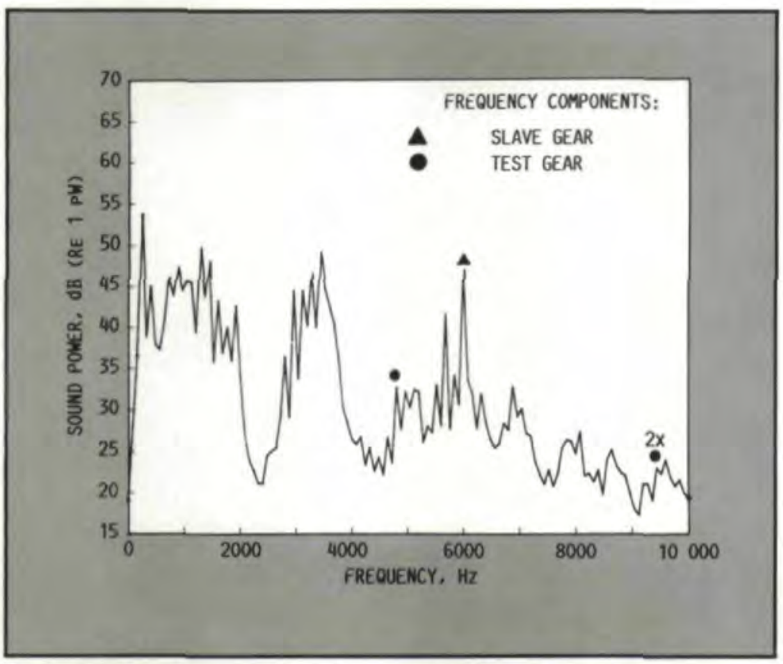

Fig. 7 – Sound power for four planes investigated (conditions: standard gear, 10 160 RPM, 1615 N).

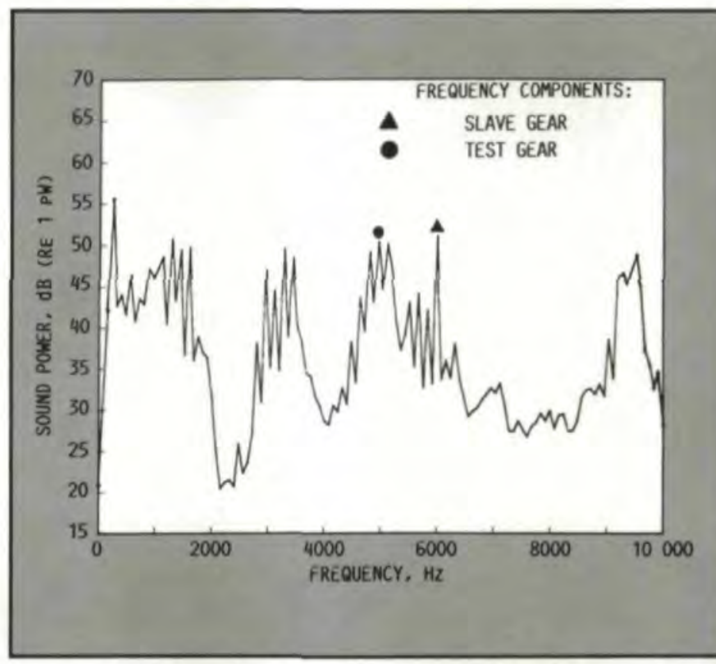

Fig. 9 – Sound power for the four planes investigated (conditions: modified gears, 10 160 RPM, 1615 N).

determined. The plots show the sound power from standard and modified gears at the two speed conditions and 1615 N load.

Notice that the gear mesh and sideband frequencies and bearing passing frequencies are still present. Inspection of the four plots shows the strong signal from the slave gears and bearings. For the standard test gear cases, the amplitude increased by 5 db from the low to the high speed tests at the slave gear fundamental frequency (Figs. 7 and 8). For the modified test gear cases, the increase was nearly 10 db at the slave gear fundamental frequency (Figs. 9 and 10). The cause for this higher increase could be the influence of the modified test gears, which have a relatively high amplitude, on the slave gears (cross-coupling).

The standard test gears show a 10 db increase in amplitude from the high to the low speed test (Figs. 7 and 8). The explanation is that the test gears operated near the predicted torsional natural frequency range of 3500 Hz at the low speed tests. Finally, this effect is noticeable even in the modified gear test data. The difference between the amplitudes of the standard and modified test gears at the high speed test is nearly 16 db.

Fig. 11 is a comparison of the six tests performed for each gearset. The measurements indicate a significant difference

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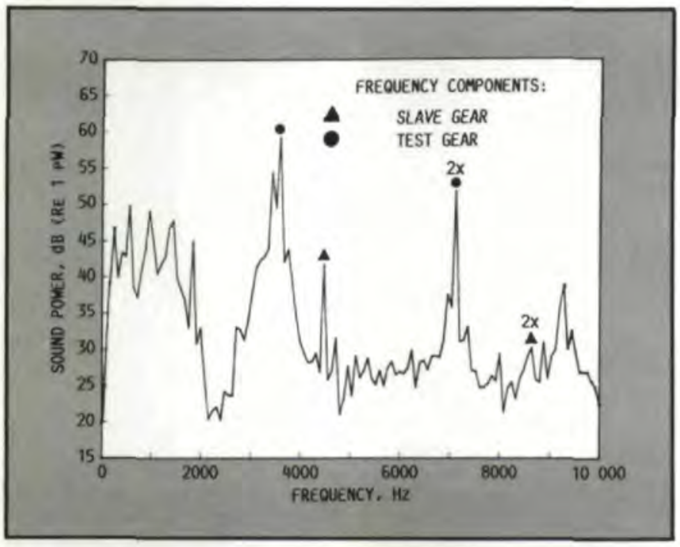

Fig. 10 – Sound power for the four planes investigated (conditions: modified gears, 7470 RPM, 1615 N).

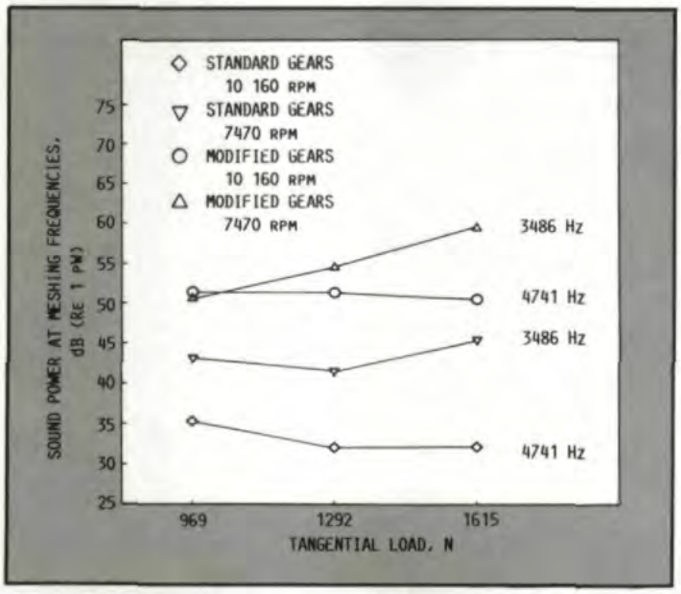

Fig. 11 – Sound power versus mesh frequencies and loading.

in the performance of the standard and modified gearsets and slight variations due to load and speed.

Conclusions

Review of the experiments and the theories for acoustic intensity and spur gears leads to the following conclusions:

1. Acoustic intensity identified dominant frequencies.

2. Robotic acoustic intensity measurements allow determination of total sound power from a noise source not withstanding the difficulty in getting around obstructions.

3. The acoustic intensity method can locate "hot spots" (surface sources and leaks). However, the measured acoustic intensity is related only to the surface phenomena while the item of interest is the source excitation. Identification of the excitation from the surface phenomena is dependent upon the dynamic behavior of the structure.

4. The test data indicates that the modified test gears are noiser than standard test gears. This shows the marked sensitivity of gear noise to the influence of tooth profile.

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