

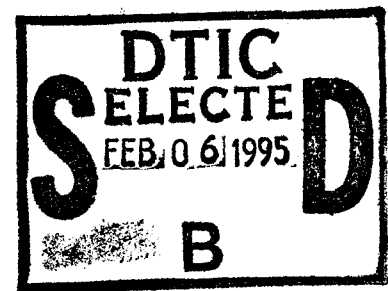
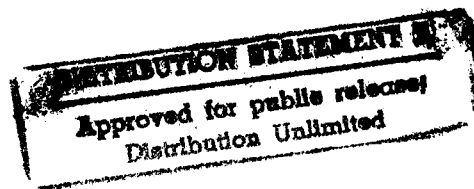
Influence of Gear Design Parameters on Gearbox Radiated Noise

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SYNOPSIS Spur and helical gears were tested in the NASA gear-noise rig to compare the noise produced by different gear designs. Sound power measurements were performed under controlled conditions for a matrix of operating conditions. Sound power was computed from near-field acoustic intensity scans taken just above the top surface of the gearbox.

Test gears included four spur and five helical gear designs. The gears were designed to be as nearly identical as possible except for deliberate differences in tooth geometry and contact ratio. Test results are presented as narrow-band sound power spectra and as charts comparing the various designs.

1 INTRODUCTION

A major source of helicopter cabin noise (which has been measured at over 100 decibels sound pressure level) is the gearbox. Reduction of this noise is a NASA and US Army goal. A requirement for the Army/NASA Advanced Rotorcraft Transmission project was a 10 dB noise reduction compared to current designs.

The main exciting forces which produce gear noise are the meshing forces of the gear teeth in the transmission. While many factors influence transmission noise, the simple fact remains that if the basic exciting forces are reduced and no amplifying factors are present, the overall noise level of the system will be reduced.

Among the several ways in which the gear tooth meshing forces may be reduced, two of the most directly applicable to helicopter transmissions are the form of the teeth and the overall contact ratio. Both approaches are attractive for an aerospace application since, unlike sound absorbing treatments, these approaches have the potential for reducing noise without reducing performance or increasing overall system weight. Both approaches also offer the possibility of improving gear performance in terms of longer life, higher load capacity, greater reliability, and reduced weight while simultaneously reducing noise levels.

Helical gears, as compared to spur gears, typically produce lower noise levels. Winter [1] provides a concise summary on the variation of excitation levels with face contact ratio. There is little other definitive data, for accurate, ground gears, which defines the noise advantage of helical gears. Similarly, anecdotal information indicates that higher contact ratios, both face and profile, also tend to reduce noise levels but, again, hard data was not readily available.

While helical gears provide some noise reduction, their use also generates a thrust load which must be

dealt with in the design of the overall system, especially the support bearings, gear blank design, and housing structure. Double helical gears, which cancel the thrust loads from each helix within the gear blank, provide relief from net thrust problems. However, the noise properties of double helical gears have not been reported.

Noninvolute tooth forms have been investigated for possible use in helicopter transmissions in recent years. Testing of high profile contact ratio, noninvolute tooth form gears, (HCR-NIF), has shown that the load capacity can be substantially higher than that of conventional involute gears and the bending load capacity (at high loads) was at least equal to that of the involute gears [2]. These investigations, however, have centered almost universally on the load capacity and not on noise generation.

This program was conducted as part of the Advanced Rotorcraft Transmission project [3]. Its objective was to define, by controlled testing, the effect on noise levels due to changes in the profile and face contact ratios and the gear tooth form. These factors were varied both separately and in combination.

The test gear configurations were selected to be representative of those used in helicopter transmissions. The test gear designs include four different types of spur gears (low- and high-contact-ratio in both involute and non-involute profiles) as well as five different helical (single and double) gear designs with various profile and face contact ratios. The gears were designed to be as nearly identical as possible except for deliberate differences in tooth geometry and contact ratio.

Testing was conducted under controlled conditions (torque, speed, oil flow, temperatures, etc.). Acoustic intensity measurements were taken with the aid of a robot to insure repeatability of measurements between gear sets and to minimize the influence of operator technique. Results presented here include trends of the

sound power at mesh frequency and narrow-band spectra of sound power. Preliminary results from this program were earlier presented by Drago [4].

2 TEST GEARS

Eight sets of test gears were designed. Four of these are spur gears. Two sets have an involute tooth form and two utilize a noninvolute, constant radius of curvature tooth form. The four helical gear sets include various profile and face contact ratios. All gears were designed in accordance with standard aerospace practice so that, except for size, they are representative of typical helicopter gears. The eight gear designs are summarized in Table 1 and are shown in Fig. 1. Additional test parameters are shown in Table 2.

Figure 1 also shows a gear set which is not listed in Table 1. This was not one of the planned test variants.

During the manufacture of the test gears, the double helical gear drawings went out with a drafting error such that both helices were manufactured with the same hand. The resultant gear set (known officially as "spread single helical gears" and unofficially as "OOPS" gears), are shown in the upper right corner of Fig. 1. Although these gears probably would not be used in a production environment, we decided to test one pair of them anyway.

3 APPARATUS AND PROCEDURE

3.1 Test Facility

The NASA Lewis gear noise rig (Fig. 2) was used for these tests. This rig features a single-mesh gearbox powered by a 150 kW (200 hp) variable speed electric motor. An eddy-current dynamometer loads the output

Table 1 Test Gear Configurations

Configuration	Tooth Form	Type	Transverse Pressure Angle	Contact Ratio		
				Profile	Face	Total
1. Spur Baseline	Involute	Spur	25	1.3	0.0	1.3
2. HCR Spur	Involute	Spur	20	2.1	0.0	2.1
3. Helical Baseline	Involute	20° Helical	25	1.3	1.15	2.5
4. Double Helical	Involute	35° Helical	25	1.3	2.3	3.6
5. Helical	Involute	27° Helical	25	1.3	1.6	2.9
6. HCR Helical	Involute	34° Helical	20	2.1	2.1	4.2
7. NIF Spur Baseline	NonInvolute	Spur	25	1.3	0.0	1.3
8. NIF-HCR Spur	NonInvolute	Spur	20	2.1	0.0	2.1

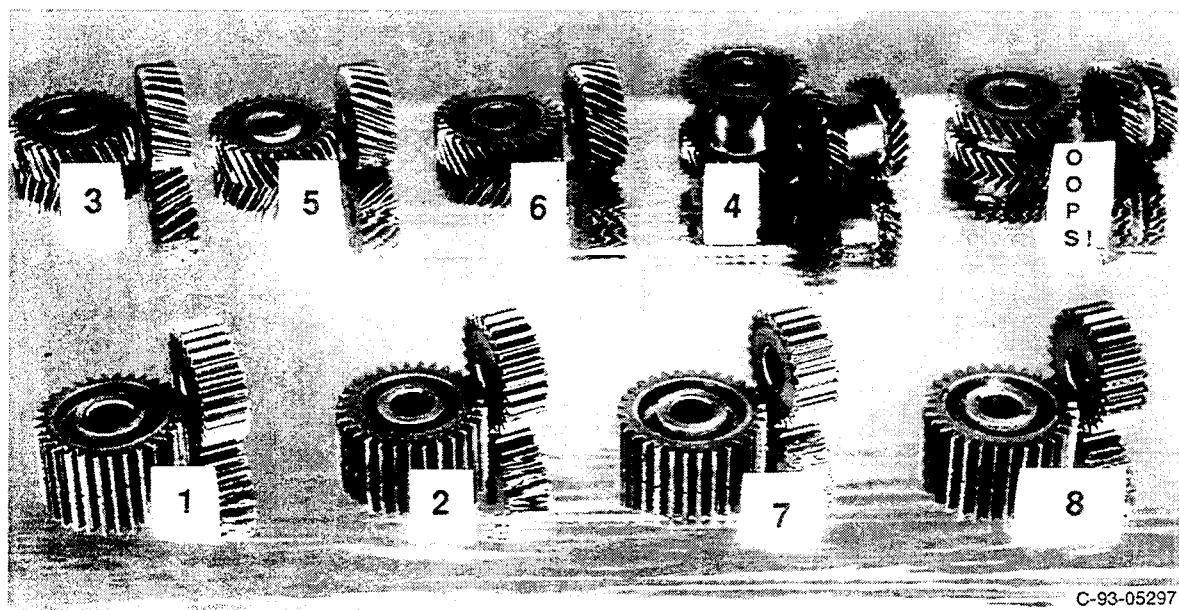


Fig. 1 - Test gears

Table 2 Test gear parameters

No. Teeth	25 and 31
Transverse module, mm (diametrial pitch, in ⁻¹)	3.175 (8)
Face width, mm (in)	31.8 (1.25)
100% input speed, rpm	5000
100% input torque, N-m, (in-lb)	256 (2269)
100% power, kW (hp)	134 (180)

shaft. The gearbox can be operated at speeds up to 6000 rpm. 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, acoustical baffles covered test cell walls, floor, and other nonmoving surfaces. The material attenuates reflected sound by 20 dB or more for frequencies of 500 Hz and above.

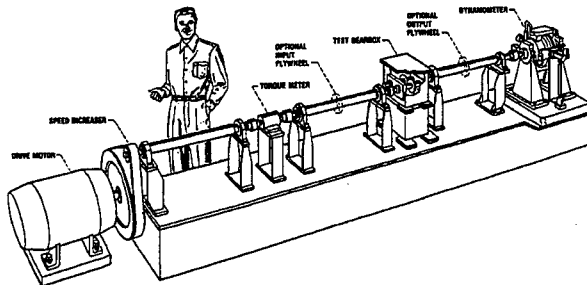


Fig. 2 - Gear noise rig

3.2 Instrumentation and Test Procedure

Experimental modal test results from a previous testing program [5] provided the first five natural frequencies and modes of vibration of the gearbox top. The natural frequencies were checked to assure that gear mesh frequencies did not coincide with important modes of the gear box. Also, from previous analytical work, we know that torsional modes of the gear system are well above the 6000 rpm speed limit of the rig.

Acoustic intensity measurements were performed, under stable, steady-state operating conditions, 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. The gearbox, robot and intensity probe are illustrated in Fig. 3. RAIMS is more completely described in references [6] and [7].

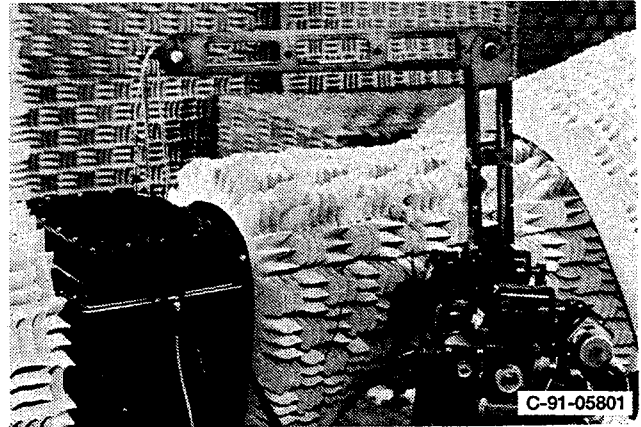


Fig. 3 - Test gearbox and RAIMS robot

The acoustic intensity probe consists of a pair of phase-matched 6 mm microphones mounted face-to-face with a 6 mm spacer. The probe has a frequency range (± 1 dB) of 300-10 000 Hz. Measurements were made at a distance of 60 mm between the acoustic center of the microphones and the gearbox top.

At each operating condition, the intensity spectra collected from the twenty nodes of the grid were averaged, then multiplied by the area to compute an 801-line sound power spectrum. The area was assumed to be the area of the grid plus one-half additional row and column of elements or 0.0910 m². The actual area of the top is 0.1034 m². We did not extend the measurement grid 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.

Noise measurements from the gearbox sides were not attempted for the following reasons: (1) the top is not as stiff as the sides; thus, noise radiation from the top dominates at most frequencies; (2) the number of measurement locations were reduced; and (3) shafting and other projections made such measurements difficult.

Sound power measurements were made over a matrix of nine test conditions: 3 speeds (60, 80, and 100 percent of 5000 rpm) and at 3 torque levels (60, 80 and 100 percent of the reference torque 256 N-m (2269 in-lb)). During each intensity scan, the speed was held to within ± 5 rpm and torque to ± 2 N-m. At least five complete sets of scans were performed on each gear set.

Acoustic intensity data were recorded over the bandwidth 896-7296 Hz. On the 801 line analyzer, this produced a line spacing of 8 Hz. We chose this frequency range because it includes the first three harmonics of gear meshing frequency for the speed range (3000-5000 rpm).

3.3 Processing Sound Power Data

The sound power data captured by the method outlined above consists of many datafiles of sound power spectra.

A-1

Sample spectra for the four spur gear configurations are shown in Fig. 4 and spectra for the five helical gear configurations are shown in Fig. 5. Each spectrum includes the first three harmonics of gear mesh frequency. The harmonic frequencies are marked with a "•" on the top border. Each harmonic is surrounded by several sidebands. The most prominent sidebands were related to the pinion shaft frequency. Gear shaft sidebands were not prominent.

To characterize the measurements, we decided to reduce each 801-line sound power spectrum to a few numbers that would represent the gear mesh noise. We call these numbers the harmonic sound power levels. We considered five methods for determining the harmonic sound power level:

(1) Record only the value at the mesh frequency harmonic. This means to ignore sidebands even though they were often significant.

(2) Check the harmonic frequency and several sidebands and record the highest value.

(3) Add together the values within a fixed-width frequency band centered on the mesh frequency. This means more sidebands would be included at lower speeds where the sideband spacing is less.

(4) Similar to (3) except the size of the frequency band would vary with speed. This means the number of values added together would not be constant.

(5) Add the values at the mesh frequency and at a fixed number of sidebands on each side of the mesh frequency.

Alternative 5 was chosen for calculating harmonic sound power levels. We used three pairs of sidebands at pinion shaft spacing (i.e., 7 peaks). Sound power values were converted to Watts prior to calculation of sums.

To reduce effects of speed drift and signal leakage we took the value at the peak plus two frequency lines on each side. In other words, we added together 5 values at each peak. Since seven peaks were used, 35 values (5x7) were added together to produce each harmonic sound power level. Figure 6 shows the data (marked with symbols "*" and "+") used to compute one harmonic sound power level. This is from the top trace in Fig. 4 near the first harmonic at 2083 Hz. (We deliberately chose an unusual example where one sideband is higher than the mesh frequency.) The sideband spacing (at 5000 rpm) is 83 Hz., thus there are about 10 analyzer lines per sideband. At lower speeds, there are fewer analyzer lines per sideband.

3.4 Data Sampling

To be assured that data from each gear set can be reliably compared with data from other gears, we needed to have sufficient records to establish a 95% confidence level of ±1 dB. This is well beyond the practical difference (i.e., a change of about 3 dB) which normal hearing can detect.

We performed at least five complete sets of scans on each gear pair tested. From these sets of measurements, we computed mean values and confidence limits of the harmonic sound power level. (For the calculation of mean and confidence limit, dB values were used. We did not convert back to Watts.) The confidence limit was calculated from:

$$C_l = t(\delta/\sqrt{n})$$

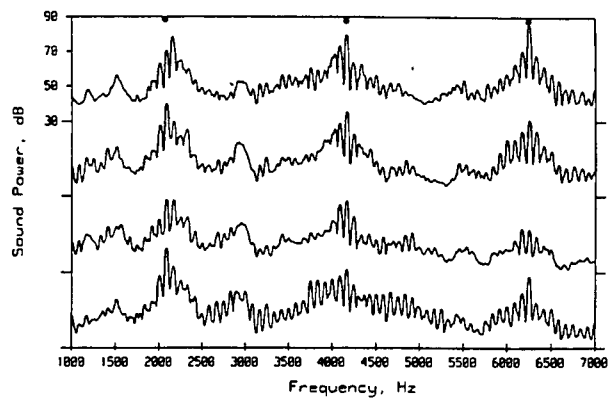


Fig. 4 - Spectra for spur gears (from bottom, configurations 1,2,7,8) at 100% speed, 100% torque.

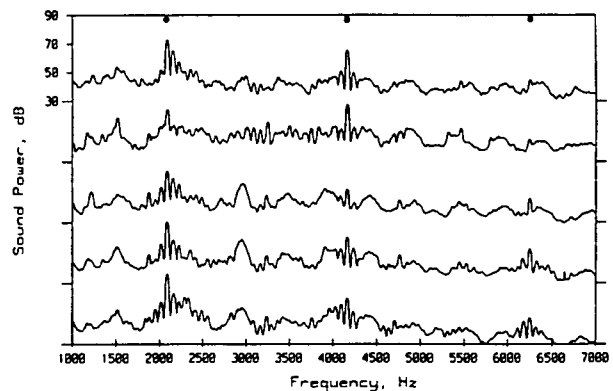


Fig. 5 - Spectra for helical gears (from bottom, configurations 3,5,6,4 and "OOPS") at 100% speed, 100% torque.

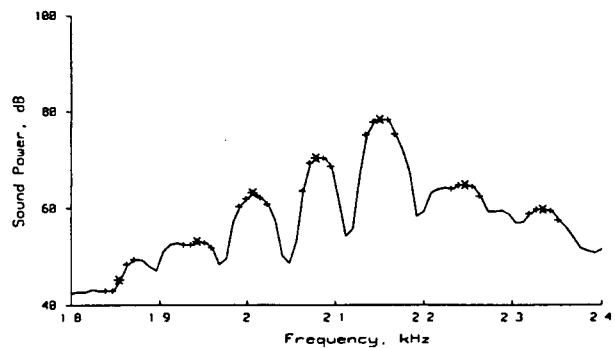


Fig. 6 - Enlargement of portion of top spectrum in Fig. 4.

where

- C_l = confidence limit, dB
- t = probability distribution ("Student t" distribution)
- δ = standard deviation of data, dB
- n = number of samples (typically 5)

Values for the "t" distribution can be found in any standard statistics text. We chose a 95 percent confidence level which corresponds to a probability level of 0.05. The number of degrees of freedom in the t distribution is the number of samples minus 1.

The mean values of the three harmonic sound power levels were used to compute a single "composite" noise level for each test condition by adding the sound power (in

Watts) and then converting to dB. It is these composite values that we compare for the various gear configurations.

To estimate the effect due to sample-to-sample variation, two sets of gears for each design were fabricated and tested. Each gear was inspected in accordance with typical production helicopter standards. The overall accuracy of the gears was consistent with production helicopter gears of similar size and configuration. The variation between the sets of gears is reasonably typical of normal production for gears in the same manufacturing lot. Lot to lot variations (not tested here) may be higher but the overall trend of the effect should be about the same.

A large difference in noise level is sometimes observed on production gear boxes simply as a result of rebuilding them after disassembly for inspection, even though no parts were changed. Considering this effect, in addition to the manufacturing variability checks, we also checked for variability due to disassembly and reassembly.

We checked for variability by testing three "builds" of the first gear set. Each build used exactly the same parts and each was accomplished by the same technician using the same tools, and parts.

4 RESULTS

A very large amount of data was collected during this test program. An overview of all the data is presented in the composite noise level bar charts of Figs. 7-8.

4.1 Spur Gears

We tested gears with both involute and noninvolute tooth form, and with both standard and high profile contact ratio. Though the noise levels (Fig. 7) generally increased with speed, in general, the high contact ratio spur gears (configs. 2,8) were 2 dB quieter than the standard contact ratio gears (configs. 1,7) regardless of the tooth form. Similarly, the involute tooth form gears (configs. 1,2) were quieter (by 3-4 dB) than their noninvolute counterparts (configs. 7,8).

4.2 Helical Gears

The single helical gears include three different helix angles and both standard and high profile contact ratios. As in the spur gears, an increase in the contact ratio correlates with a decrease in the noise level. Increasing the face contact ratio from about 1.15 (config. 3), to 1.6 (config. 5), decreases the noise level substantially in every case, though the results at higher speeds are more dramatic than at lower speeds. Also, at every operating condition, the composite noise level of a helical gear (Fig. 8) is less than the level for a spur gear with similar profile contact ratio.

The combination of a high profile and high face contact ratio further decreases gear noise. Indeed, the high profile and high face contact ratio design (config. 6) with profile and face contact ratios of 2.1, and 2.1 respectively was the lowest noise generator at almost every operating condition.

Helical gears used in helicopters tend to have relatively low face contact ratios (helix angles are kept low to minimize thrust loading and the extra weight associated with

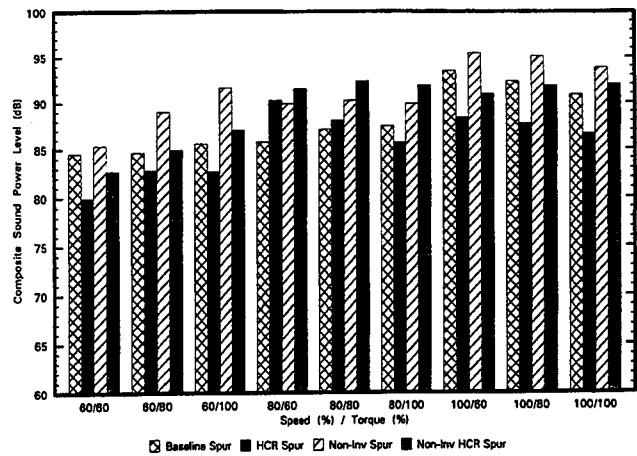


Fig. 7 - Spur gear composite noise levels

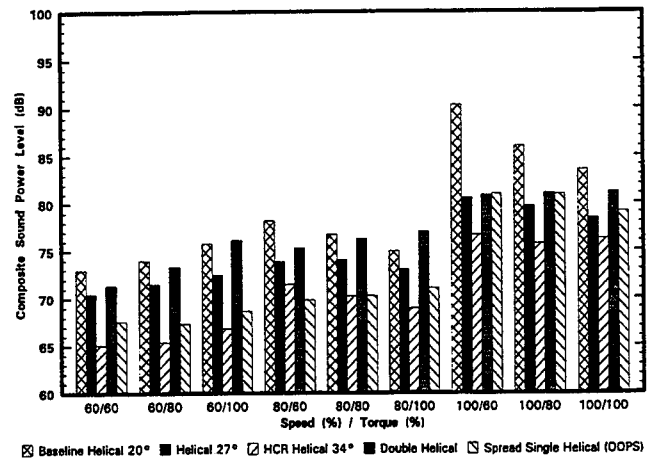


Fig. 8 - Helical gear composite noise levels

reacting the thrust) thus this result is especially interesting since it suggests that it may be possible to trade off helix angle against increasing profile contact ratio to improve the noise level without the weight penalty associated with accomplishing the same reduction with helix angle alone.

A surprising result, the double helical gear set was noisier (by 4 dB on average) than its single helical (OOPS gear) counterpart. The OOPS gear set is essentially a single helical set with a gap in the middle of the tooth face. Its effective face contact ratio is similar to that of the high contact ratio helical gears (config. 6).

The double helical phenomena appears to be related to axial shuttling which occurs as the double helical pinion moves to balance out the net thrust loading. The shuttling is due to the presence of small mismatches in the relative positions of the teeth on each helix. No matter how accurate the gear is, some mismatch will always be present, thus this is an unavoidable phenomena.

While the thrust balancing characteristic of a double helical gear is a valuable design feature since it greatly simplifies the bearing system, a price is paid in terms of noise and vibration as the gear set shuttles back and forth.

Since the per helix face contact ratio, face width, profile contact ratio, etc. are identical for the OOPS and the double helical gear sets, the only operational difference is the lack of axial shuttling. The double helical set will be in a

constant equilibrium seeking state because of the theoretically zero net thrust load while the OOPS gear set will run in a fixed axial position due to the net thrust load.

This test provides some insight into the magnitude of the noise penalty which is paid when double rather than equivalent single helical gears are used. Since these test gears are all very accurate (typical for helicopter gears), it should be obvious that a larger penalty would be paid if gears of lesser quality were to be used because the lower the gear quality is, the more shuttling would be likely to occur.

4.3 Sample, Build, and Specimen Variations

We took at least five sets of noise scans at each operating condition. Our goal was to obtain confidence limits within 1 dB for each value of harmonic sound power level. This goal was met on about 60 percent of the test sets.

During other testing, the authors have noted significant variations in the measured (and perceived) noise level of the same gear system before and after disassembly. In some cases, this variation was of considerable magnitude. To investigate this phenomena, the first set of baseline spur gears (config. 1), was assembled, tested, disassembled, re-assembled and then tested again. This process was repeated until the gears had been tested three times.

The largest minimum to maximum build variation was 7.8 dB (at the high speed, low torque condition) while the minimum build variation was 0.7 dB (at the medium speed condition). The average build variation was about 3 dB. While no real pattern is apparent, it appears that the variation decreased slightly with increasing load.

Since we tested two samples of each of the eight gear designs, we can compare the "build" variation to the variation between "identical" parts. For the eight gear designs, the average part-to-part variation in the composite noise levels was 2.8 dB. One would expect the variation between samples of the same part to equal or exceed the variation from rebuilding the same parts. The "build" test was performed at the beginning of the test program. Increased experience may have reduced the variation for later tests.

The factors considered above point out the difficulty in defining a noise reduction effort in that the variations due to unintended effects are often of the same order of magnitude of the changes which may be attributed to gear configuration or treatment. Such differences should exceed the variations due to sample and build effects and those observed among different specimens of the same part before they can be considered significant of themselves.

CONCLUSIONS

Nine different spur and helical gear designs were tested in the NASA gear-noise rig to compare the noise radiated from the gearbox top for the various gear designs. Sound power measurements were made under controlled conditions for a matrix of operating conditions. The following conclusions were made:

1. The most significant factor for noise reduction, within a gear designer's control, was found to be the total contact ratio. Gear noise may be reduced by increasing either the profile or face contact ratio.
2. The non-involute tooth form spur gears were found to have a 3-4 dB noise penalty compared to their conventional involute counterparts.
3. The high-contact-ratio spur gears (with a 58 percent increase in profile contact ratio) showed an average noise reduction of about 2 dB over standard gears.
4. The noise level of double helical gears averaged about 4 dB higher than otherwise similar single helical gears.
5. In noise reduction tests, variation due to unintended effects such as testing different part specimens or even reassembly with the same parts may be of the same order of magnitude as the effect of deliberate design changes.

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