Noise Reduction in Plastic & Powder Metal Gear Sets

Robert E. Smith & Irving Laskin

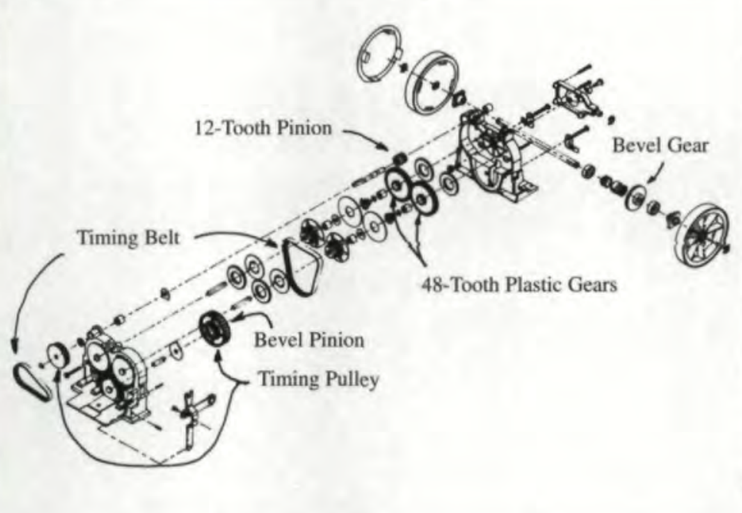


Fig. 1 — Drive system.

TABLE 1—MULTIPLES OF MOTOR SHAFT SPEED

Feature	Order of Rotation	
Motor	1st	
12T Pulley		
22T Pulley		
Main Timing Belt	12th	
	24th	2nd Harmonic
12-Bladed Cooling Fan	12th	
	24th	2nd Harmonic
	36th	3rd Harmonic
12T Pinion		
48T Gear	6.55	
	13.1	2nd Harmonic
	19.6	3rd Harmonic
	26.2	4th Harmonic
11-Bladed Vac Fan	11th	
	22nd	2nd Harmonic
	33rd	3rd Harmonic
Secondary Belt		
and Pulley	3.41	
	6.82	2nd Harmonic
	10.22	3rd Harmonic
Commutator Segments	22nd	
	44th	2nd Harmonic

Introduction

The data discussed in this article was taken from an upright vacuum cleaner. This was a prototype cleaner that was self-propelled by a geared transmission. It was the first time that the manufacturer had used a geared transmission in this application.

The transmission (Fig. 1) is driven by a timing belt takeoff from the fan motor shaft. It contains a 12-tooth powder metal pinion driving a pair of 48-tooth plastic gears. Because of the molding and sintering processes, these gears had been specified as AGMA Q6 quality. Through a clutching arrangement, one or the other of the plastic gears is driving the machine in the forward or reverse direction. Pushing or pulling on the upright handle actuates a rocker arm that engages the appropriate clutch. Another timing belt drive connects the 48-tooth gears to a final set of bevel gears. The large bevel gear is on the final drive axle.

When operating the vacuum cleaner in the self-propelled mode, a nearly pure tone whine was heard while the machine was moving in each direction. It was of a fairly high pitch and was objectionable when compared to previous non-self-propelled machines. The vacuum fan and beater bar also made considerable noise and were responsible for most of the overall dbA sound pressure level. However, because of the whine characteristic of the gear noise, it was objectionable even though it was lower than the overall sound level.

Identification of Noise Sources

Spectral Analysis. Because of the complexity of the drive assembly, it was necessary to use spectral analysis to identify the offending components (Ref. 1). This was done by a FFT real time analyzer. This takes a complex sound waveform and breaks it down into its various spectral frequencies.

Relationship to Transmission Error. Moving elements in a complex train such as this one cause airborne noise and force variations that are applied to the structure. Often the structure will act as a mechanical amplifier of the noise at frequencies that coincide with its resonant frequencies.

The gears and other elements act as exciters. The excitation from the gears and timing pulleys comes from what is known as "transmission error" (Refs. 2–3). Transmission error is a non-uniform motion that is the result of runout and errors in geometry of the gear or pulley teeth.

Comparison of Noise Frequencies. Gears will generate noise excitation at their mesh frequency and its multiples. Runout can also generate noise. This usually shows up at a once-per-revolution frequency or at once-per-revolution sideband of mesh frequency. In order to identify the offending source, one must know the operating speed of the machine and the discrete frequencies of the noise. In this case, the speed varied with load and was different for each test. Therefore, it was easier to create a table of "rotational orders" that could be used for any operating speed (See Table 1).

Measured Sound Data. Fig. 2 shows spectral data of a noise test of the prototype machine. It was taken with a sound pressure level (SPL) meter that was set for "A" weighting. This attenuates the higher and lower frequencies to approximate the response of the human ear. The mesh frequency of the 12 x 48-tooth gear set was 1,350 Hz. A marker is set at that frequency and at all harmonics of it. It can be seen in the figure that noise exists at mesh frequency and several harmonics. The second harmonic of mesh is the highest peak. This could be due to the "A" weighting mentioned above or to waviness in the tooth form. A peak can also be seen at 2,275 Hz. That is from the 11-bladed fan.

Measurement & Modification of Gears

It was decided to remove the gears from the machine and do further diagnostic testing. This consisted of transmission error testing and involute inspection. In gears of this type, noise at mesh and harmonic frequencies is most apt to be caused by lack of conjugacy or mismatch of involute profiles (Refs. 2–3). This will show up in both of the tests mentioned above.

Transmission Error Tests. Transmission error testing is usually done with high resolution optical encoders and instrumentation that measures small deviations in the smoothness of rotational motion. Typically it is done at relatively light loads. In this case, tests were run at both light and at operating loads. There was no discernable difference in the tooth-to-tooth transmission error at either load condition. This showed that tooth deflection under load was not a problem and that tip relief on the teeth was not necessary. Fig. 3 shows the results of a transmission error test. The first half of Fig. 3 shows the total transmission error curve for three revolutions of the 48-tooth gear. One can see a sine wave from each revolution of the 48-tooth gear, a sine wave from each revolution of the 12-tooth pinion and, finally, the very fine waves superimposed on the larger ones. These fine waves are from each tooth mesh and are the ones of concern as far as gear noise. The second half of Fig. 3 shows these fine waves, but magnified after removal of the long term components from the gear and pinion runout. This last chart shows 12 tooth meshes or one pinion revolution.

This tooth-to-tooth transmission error has a sawtooth characteristic, as well as a double bump in each tooth mesh The sawtooth characteristic will generate noise at all harmonics of tooth mesh, and the double bump will accentuate the

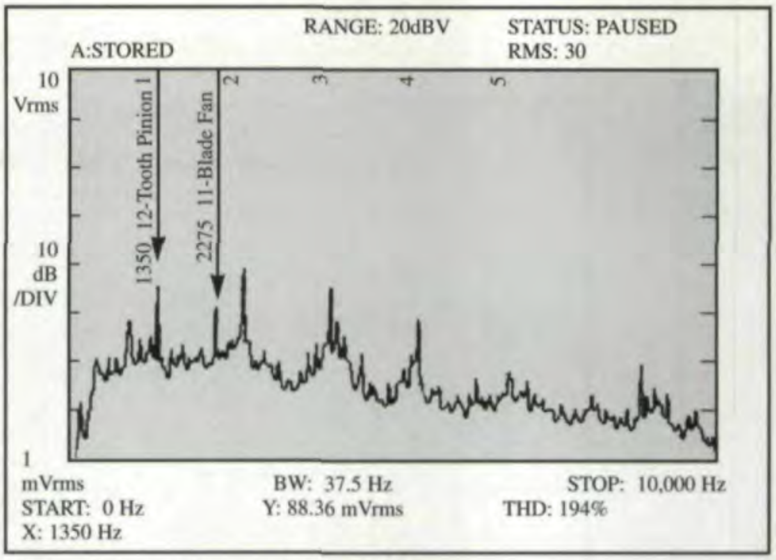


Fig. 2 — Noise spectrum—prototype machine.

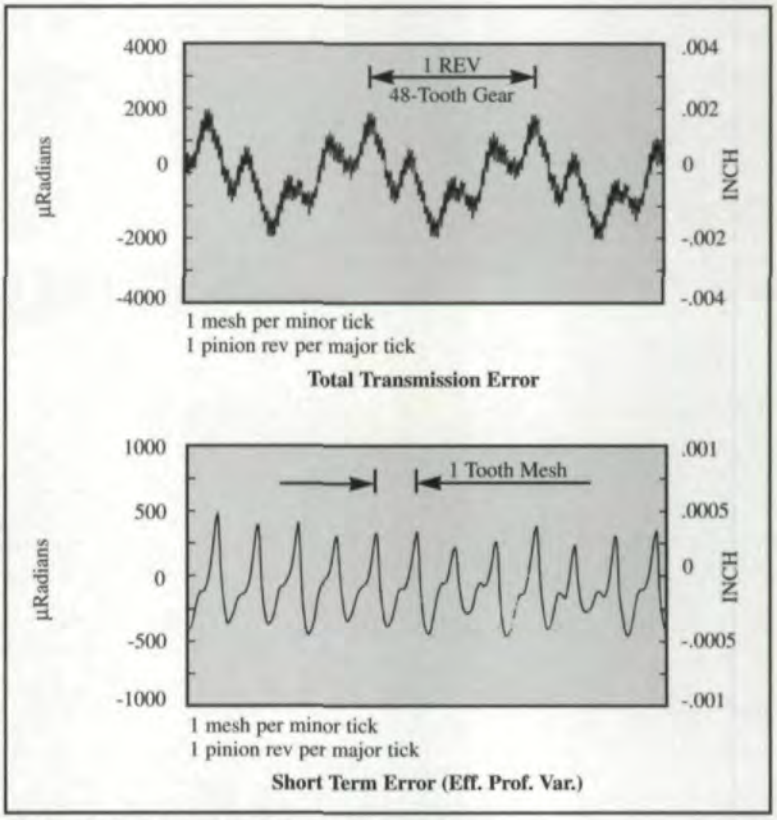


Fig. 3 — Transmission error test results—noisy set.

Robert E. Smith

is the principal in Robert E. Smith & Co., Inc., gear consultants, in Rochester, NY. He is also one of Gear Technology's *technical editors.*

Irving Laskin

is an independent gear consultant in Sharon, MA.

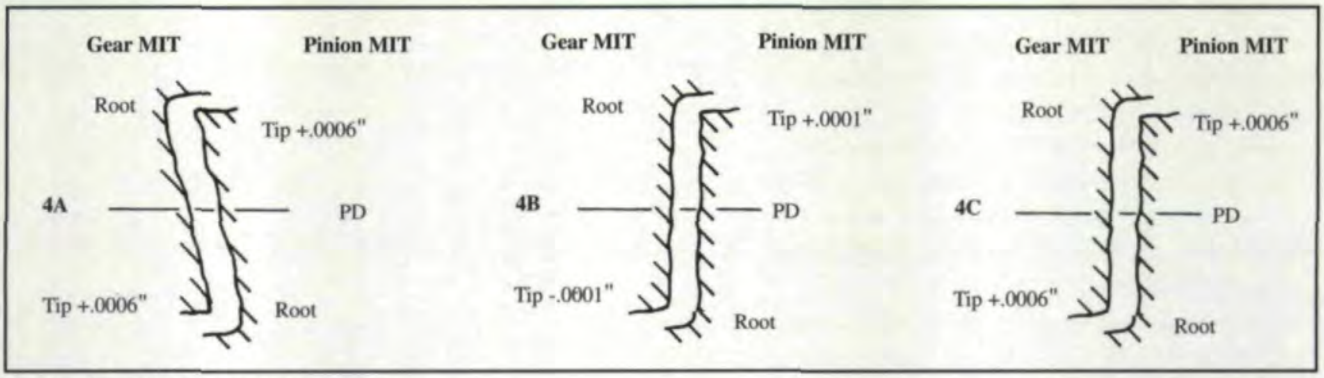
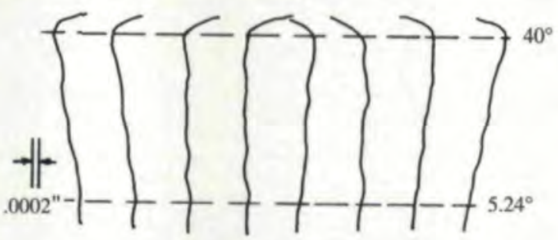


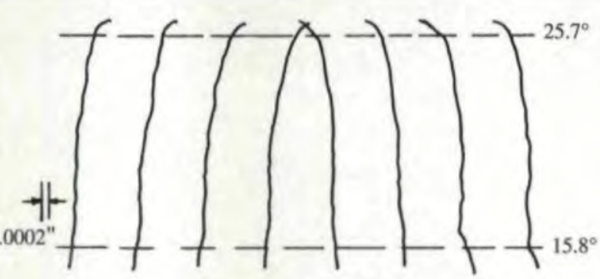
Fig. 4 — Mean involute traces (MIT).



Results — 1/10,000"

	Left Flank				Right Flank			
	4	3	2	1	1	2	3	4
fHα	9.4	9.5	1.7	1.8	3	-0.7	5.5	10.2
fHαm (MIS)	5.6				4.5			

Fig. 5 — Involute tests of 12-tooth pinion from noisy gear set.



Results — 1/10,000"

	Left Flank				Right Flank			
	4	3	2	1	1	2	3	4
fHα	-6	-7.3	-7.5	-6.6	-4.8	-6.7	-5.6	-4.8
fHαm (MIS)	-6.8				-5.5			

Fig. 6 — Involute tests of 48-tooth gear from noisy gear set.

second harmonic of mesh. In addition, the peak-to-peak amplitude of each tooth mesh is in the order of .0007" to .0009", which is a very significant amount. It indicates that the mating teeth are not very conjugate.

Involute Tests

Evaluation Method. The traditional method for evaluating these traces is to fit them within a tolerance zone. The AGMA tolerance zone is a "K" chart. The width of the "K" zone is determined by the tolerance allowed in the different quality grades. On the other hand the ISO tolerance zone is rectangular. These zones are sufficiently wide to allow for involute variation that is the result of runout in addition to the variation in tool quality. This is all right for the determining a quality rating, but it is insufficient for controlling gear noise.

Conjugacy. In order for a pair of gears to be smooth and quiet (minimal tooth-to-tooth transmission error), they have to be conjugate. This means that they must have the same operating pressure angle (Ref. 4).

Mean Involute Slope (MIS) and Mean Involute Trace (MIT). The best indicator from involute traces of conjugacy is the MIS (known as fHαm in the ISO system). The AGMA "K" chart system does not address this characteristic.

The MIS is determined by fitting a line to the individual involute traces. CNC involute checking machines usually do this by a least-squares-best-fit of a straight line. When doing it manually, one can fit a line by eye between two control diameters. The slope of this line for four teeth approximately 90° apart is averaged to determine the MIS.

If the four involute traces are averaged together, a Mean Involute Trace (MIT) is established. Fig. 4 shows MIT diagrams of several examples. Each one is the average of four involute traces taken approximately 90° apart on each mating member. This is a very useful bit of information that would be easy for CNC inspection machines to do, but so far, none of them do.

The gear trace is inverted in relation to the pinion trace in order to show how they visually match each other. The first example shows a combined mismatch of .0012" (not conjugate). The second example shows a pair that is conjugate and of the correct pressure angle. The third example shows a pair that is conjugate, although of a pressure angle different than specified.

For example, a pinion and gear could both be classed as AGMA Q8. The pinion teeth could have a MIS of +.0006". The gear teeth could have a MIS of -.0006". They would not be conjugate and would therefore be noisy (Fig. 4a). Another pair, also AMGA Q8, could have an individual tooth involute variation of .0006", a MIS of ±.0001, and be nearly conjugate, as well as quiet. Both sets, however, are no better than AGMA Q8. This example is shown in Fig. 4b.

In another example, the pinion and gear could both have a MIS of +.0006" (still only a Q8) and also be quiet. These are shown in Fig. 4c.

Involute Measurements. Further diagnostic tests were done using elemental involute measurements. Four teeth on each part, approximately 90° apart, were measured. The results of the 12-tooth, p/m pinion are shown in Fig. 5. Fig. 6 shows the results of the 48-tooth plastic gears.

It should be noted again that the slope of each involute trace varies as the result of runout in the part. This slope, between the EAP and SAP, is noted for each trace (fHα). Another important number on these charts is the value for (fHαm), the MIS. (These symbols are from ISO. There are no equivalent AGMA symbols).

It can be seen that these parts were not very conjugate, even though they might be classified as AGMA Q8 gears. The 12-tooth pinion has a MIS (fHαm) of about +.0005", and the 48-tooth gear had a MIS (fHαm) of approximately -.0005".

Putting this deviation into terms of pressure angle, it was like running a 19.5° pinion with a 20.5° gear.

This conversion was made with the following equation:

$$\phi_A = \phi - (57.3°)\ \frac{2\ (\text{MIS})}{d_B\ (\varepsilon_2 - \varepsilon_1)\ \tan\phi}$$

Where:

ϕ_A = actual pressure angle (degrees)

ϕ = nominal (or setup) pressure angle (degrees)

MIS = mean involute slope

d_B = nominal (or setup) base circle diameter

ε_2, ε_1 = upper, lower roll angles over which the MIS is measured

For the pinion:

$\phi = 20°$

MIS = + .0005"

d_B = .4698"

ε_2 = 40.00° at .573" diameter

ε_1 = 5.24° at .4718" diameter

and from the equation:

ϕ_A = 19.45°

For the gear equation:

$\phi = 20°$

MIS = - .0005"

d_B = 1.8794"

ε_2 = 25.72° at 2.060" diameter

ε_1 = 15.53° at 1.9472" diameter

and from the equation:

ϕ_A = 20.47°

Experimental Results. In order to prove that MIS was a good measure of conjugacy and quietness, even for low quality gears, some pinions were reworked to match the -.0005" tip MIS of the

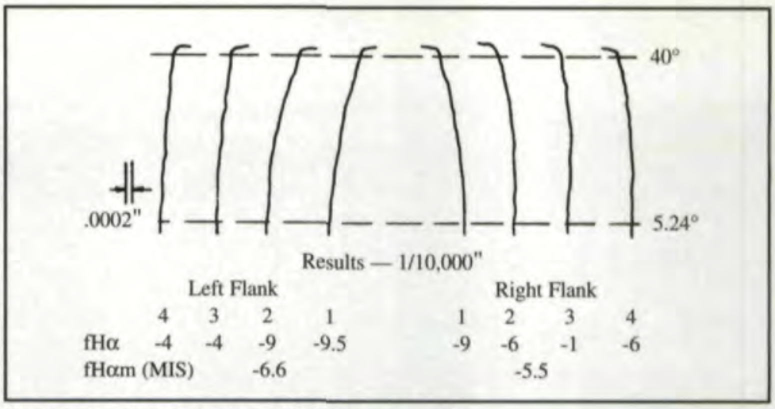


Fig. 7 — Involute tests of modified 12-tooth profile.

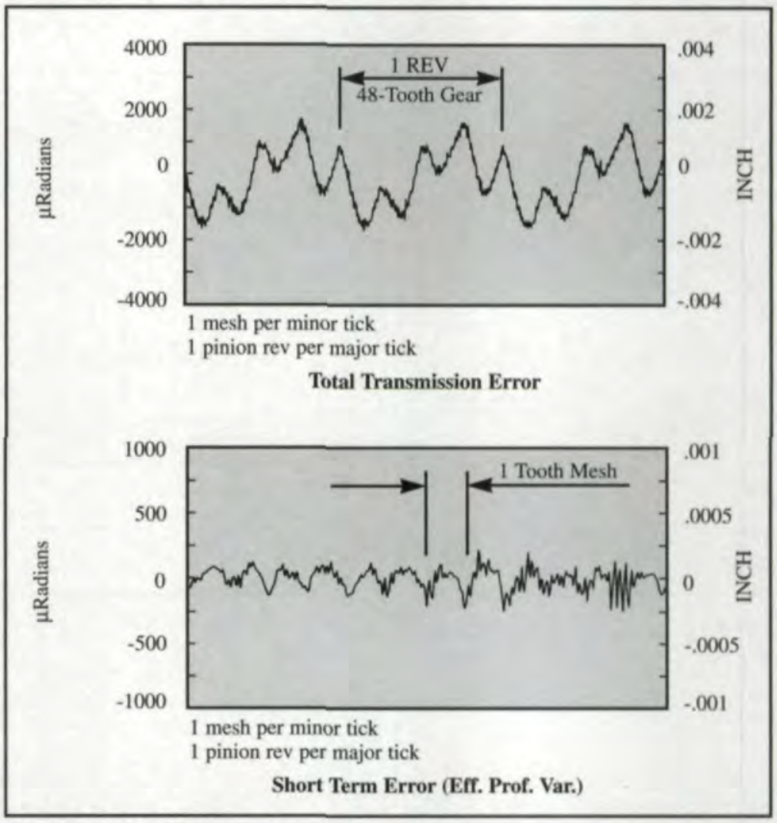


Fig. 8 — Transmission error test results—quiet set.

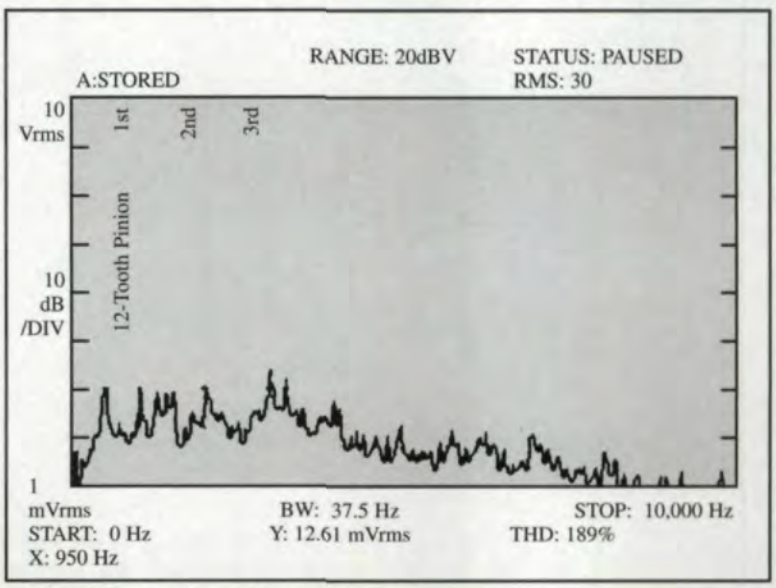


Fig. 9 — Noise spectrum of improved gear set.

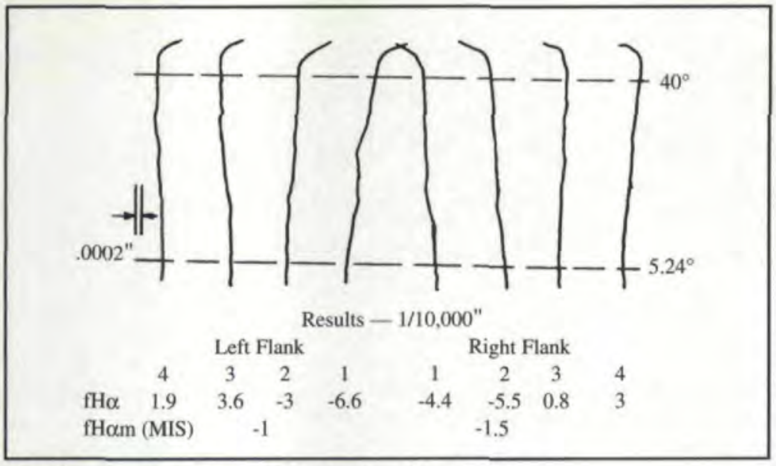


Fig. 10 — Involute tests—final production, 12-tooth pinion.

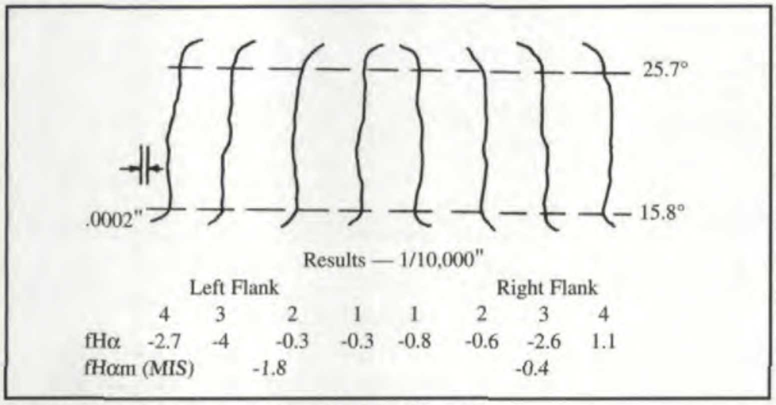


Fig. 11 — Involute tests—final production, 48-tooth gear.

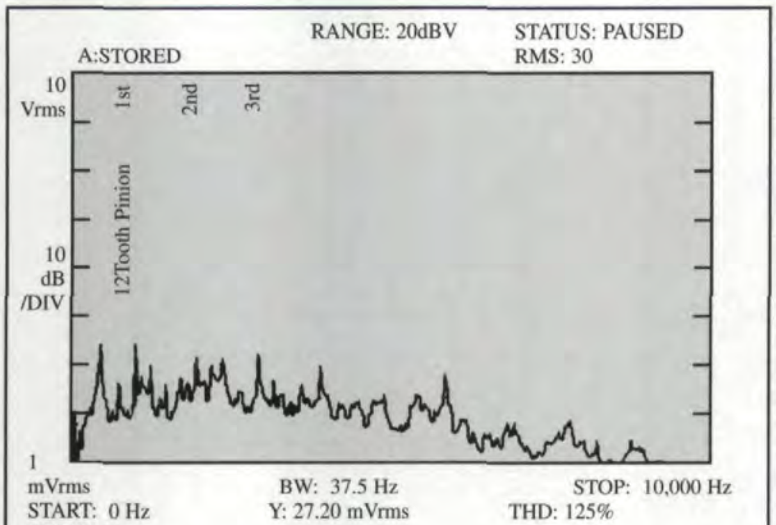


Fig. 12 — Noise spectrum of final production gear set.

48-tooth plastic gears. The pinions were made to have a -.0005" to -.0006" tip MIS also.

The involute results are shown in Fig. 7. They now have basically the same pressure angle as the 48-tooth gears.

Fig. 8 shows the improved results on the transmission error test. The spectrum chart in Fig. 9 shows the improved noise test. It is difficult to see peaks in the spectrum that relate to the gear mesh and harmonic frequencies, and the subjective whine was gone.

New Tolerance Specifications

The makers of the gears felt that they couldn't make gears to a quality level better than AGMA Q6. However, meeting a specification of MIS is more of a development problem than a quality problem. This involves making the tools right in the first place. In this case, the tool is the cavity.

The specification was changed to AGMA Q7 to keep the total composite variation down and avoid a tight mesh condition. To control noise, a requirement of a MIS (fHαm) within ± .0002" for each member of the pair was also specified. The ± .0002" limit on MIS was based on prior experience with many other applications.

Manufacturing Process in Relation to Profile

P/M Pinion. With steel powder metal parts, the profile shape will be determined primarily by the shape of the tool. There isn't much net change in the shape of the part as it is removed from the cavity and is sintered. There usually is a very slight increase in size, which is largely offset by the slight shrinkage in sintering, as the part expands when leaving the cavity. Therefore, obtaining the proper tooth shape depends on getting the cavity developed right in the first place. It isn't possible to play with other process variables such as temperature to change part geometry.

Molded Plastic Gear. Molded plastic gears are another story. The cavity size has to be determined by knowing accurately the shrink rate of the material being used. Typical gear materials might have a shrink rate that varies from .005" per inch to .030" per inch, depending on material as well as fillers, such as glass fibers. This has to be taken into account when designing the size of the cavity. Other process variables, such as temperature and pressure, will also affect the shrink rate and therefore the resulting part size and profile shape.

Redesign of Mold Cavity (Plastic Gears)

Measurement of Parts from First Cavity. Molded plastic parts made of unreinforced material generally shrink at a nearly uniform rate. In this case, the material was an acetal (Delrin 500). The original cavity was designed for a specific shrink rate. The resulting parts were measured for various diameters such as outside, root, rim and hub diameters. The teeth were also measured for involute by adjusting the base diameter until the MIS was near zero. This showed that the base diameter shrunk at nearly the same rate as all the other diameters. The mold cavity was also measured for these various diameters. From these measurements, it was determined that the original parts shrunk more than expected. This resulted in the gears having a higher pressure angle than desired.

Cavity Redesign. A new cavity was designed and made to the new actual shrink rate for the material and process being used. This resulted in parts that were closer to the desired specification.

Evaluation of Final Production Parts

Final Involute Tests. Involute results of the new parts are shown in Figs 10–11. The MIS for both mating parts are within .0002", meaning that they are nearly conjugate.

Final Noise Test. Noise tests of the final parts show little indication of gear noise in the spectrum. Gear-related peaks are no worse than peaks from other sources, such as the timing pulley and the fan. Fig. 12 shows these results.

Process Control of Molded Gear Profile

Relation of Profile Variation to Shrink Variation. A study was conducted to establish the shrink rate vs. process variables such as temperature and pressure. The purpose was to find a method of controlling the involute by adjusting a process variable. Changing temperature is not as desirable as changing the pressure, because it has a greater effect on material properties such as strength.

Gears were molded at various molding pressures from 7,500 to 13,000 psi. Gears from each pressure level were checked for MIS and outside diameter. The results were plotted and are shown in Fig. 13. This shows a reasonably linear relationship between pressure and mean involute slope. It also shows a good relation between outside diameter and MIS. Therefore, molding pressure became a good process variable for control of the desired parameter.

Control of Shrink (Involute) by Control of Outside Diameter. The discussion in the section above showed that all diameters, including the base diameter, shrink at a nearly uniform rate. Therefore a decision was made to use the outside diameter measurement as a control of mean involute slope (See Fig. 14). As long as the same mold cavity dimensions are used, this relationship will hold true. A fixture was made that could be used for quick measurements of the OD of the gear teeth. This has now been used successfully in production for over two years.

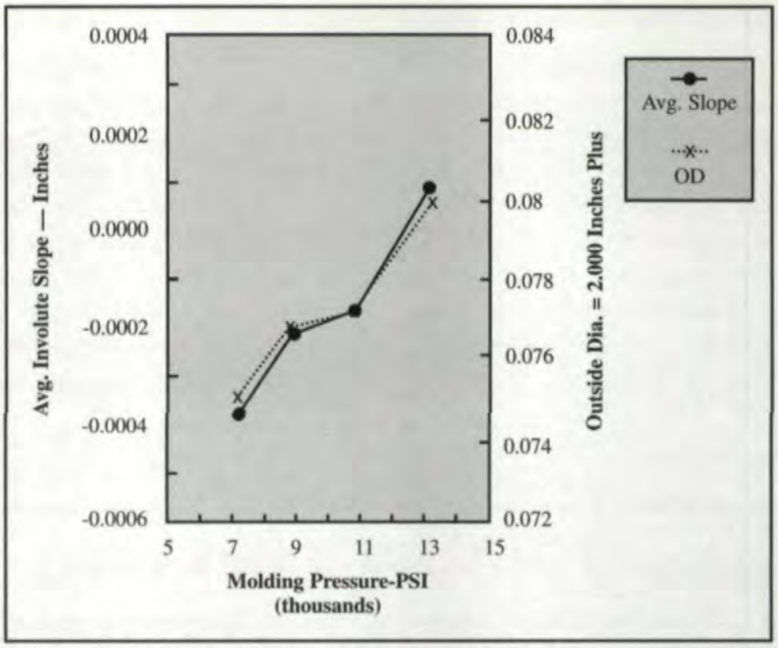


Fig. 13 — PSI vs. average involute slope and outside diameter.

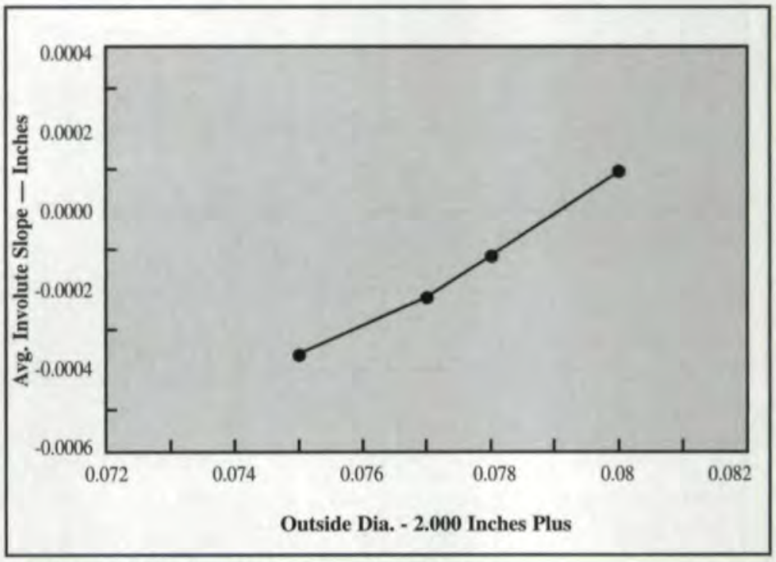


Fig. 14 — Outside diameter vs. average involute slope.

References:

1. Smith, R. E. "The Relationship of Measured Gear Noise to Measured Gear Transmission Errors," AGMA Fall Technical Meeting, 87FTM6, Cincinnati, OH, October 6–7, 1987.
2. Smith, R. E. "Identification of Gear Noise With Single Flank Composite Inspection," AGMA Fall Technical Meeting, 85 RTM12, San Francisco, CA, 1985.
3. Smith, J. D. *Gears and Their Vibration*. Marcel Dekker, 1983.
4. Maag Gear Company. *Maag Gear Book*. Maag Gear Company, Ltd., Zurich, Switzerland, 1990, pp. 318–320.

Acknowledgements:

Gratitude is expressed to the Kirby Company of Cleveland, OH, for the use of data collected on one of their projects.

Originally presented at the AGMA Fall Technical Meeting, 1992. AGMA Paper 92FTM12. Reprinted with permission.

Tell Us What You Think...

If you found this article of interest and/or useful, please **circle 207.**

For more information about R.E. Smith & Co., please **circle 208.**

For more information about Irving Laskin consulting, please **circle 209.**