Identification of Gear Noise with Single Flank Composite Measurement

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Abstract

This article was written to serve as a guide for the application of single flank composite inspection to the solution of gear noise problems. It includes a discussion of the relationship of transmission error to gear noise, housing dynamics, spectral analysis and how it is used in problem-solving situations. Several case histories are described.

Introduction

Anyone involved in the design, manufacture and use of gears is concerned with three general characteristics relative to their application: noise, accuracy, and strength or surface durability. In this article, we will be dealing with probably the most aggravating of the group, gear noise.

The use and analysis of single flank composite inspection of gears can result in the understanding and control of gear noise problems. This is achieved through the measurement of transmission error, which is the predominant cause of gear noise.

AUTHOR:

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Gear Noise

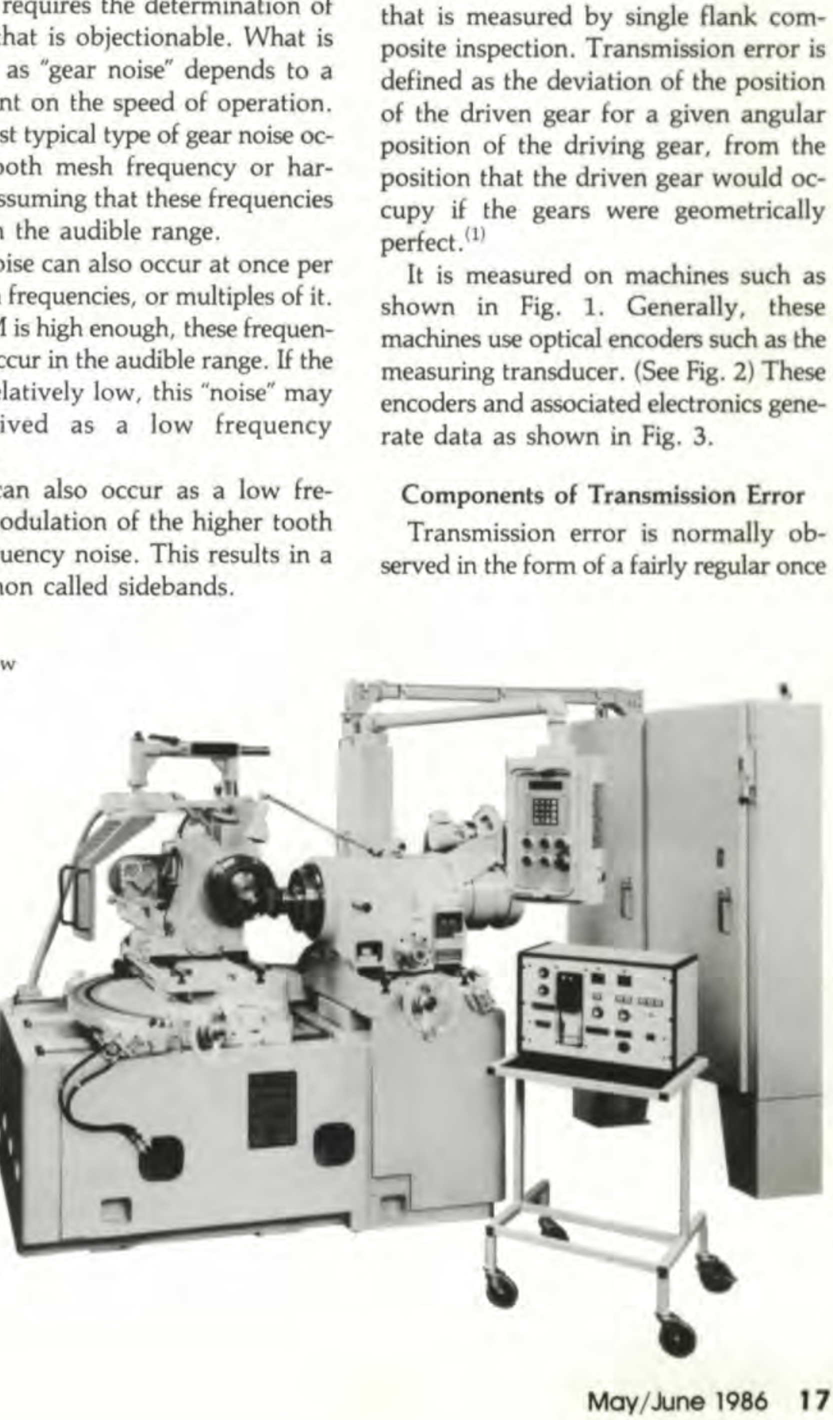
Gear noise comes in many types. The successful solution of gear noise problems first requires the determination of the type that is objectionable. What is perceived as "gear noise" depends to a great extent on the speed of operation.

The most typical type of gear noise occurs at tooth mesh frequency or harmonics, assuming that these frequencies are within the audible range.

Gear noise can also occur at once per revolution frequencies, or multiples of it. If the RPM is high enough, these frequencies will occur in the audible range. If the RPM is relatively low, this "noise" may be perceived as a low frequency vibration.

Noise can also occur as a low frequency modulation of the higher tooth mesh frequency noise. This results in a phenomenon called sidebands.

Fig. 1—below



To properly define the type of noise that is of concern, it is usually necessary to apply some sound analysis measurements to the final application, whether it is a gear box, vehicle or some other structure.

Frequency is the key to understanding the type of gear noise, and ultimately to deciding what corrections must be made to the gears. Frequency analysis is done with equipment such as "tuneable narrow band pass filters" or "real time analyzers". The analysis of data from such instruments will be discussed in detail later.

Transmission Error

Transmission error is the parameter that is measured by single flank composite inspection. Transmission error is defined as the deviation of the position of the driven gear for a given angular position of the driving gear, from the position that the driven gear would occupy if the gears were geometrically perfect.[1]

It is measured on machines such as shown in Fig. 1. Generally, these machines use optical encoders such as the measuring transducer. (See Fig. 2) These encoders and associated electronics generate data as shown in Fig. 3.

Components of Transmission Error

Transmission error is normally observed in the form of a fairly regular once per tooth pattern, superimposed on large waves related to once per revolution type errors. Noise and vibration excitation is generally related to the once per tooth pattern, while accuracy problems are more generally related to the once per revolution type patterns. The curve can be generated by running a pair of work gears together, or by running a work gear with a master gear.

The total transmission error curve is made up of several components:

1. Total composite (F_i')
2. Tooth to tooth composite (f_i')
3. Long term component
4. Short term component (effective profile)

Total Composite (F_i')

The total composite error is read from the "raw" data as the difference between the highest and lowest points on the graph, within one revolution of the largest gear. If the gears are near a 1:1 ratio, several revolutions may be necessary for the errors of both gears to phase together and show the worst case.

This type of error is important for accuracy applications and includes the effect of accumulated pitch variation as well as a portion of profile or involute variation. (See Fig. 4A)

Tooth To Tooth Composite (f_i')

The tooth to tooth composite error is seen as the variation in transmission error at tooth mesh frequency. It is read as the highest to lowest point in any 360°/N (where N = number of teeth) or one angular pitch portion of the total transmission error curve. It results from a portion of the profile error plus the effect of individual pitch variations.

The resulting graph will be the short term component and is most generally related to errors in tooth form occurring at tooth mesh frequencies. (See Fig. 4C) In most gearing, long face helicals excluded, this relates primarily to a portion of the profile mismatch of the gear teeth or involute error. Both the amplitude and shape of this component are of concern relative to noise excitation.

This is the parameter that is most important for analysis of gear noise problems.

Development of the Short Term Component

When describing the relationship of tooth geometry to the transmission error curve, it is best to think in terms of involute spur teeth. However, the same principles apply to helical and bevel teeth when dealing in terms of short face widths.

This, also, is important for accuracy applications when small angle errors are of concern. (See Fig. 4A)

Long Term Component

The long term component results from drawing in the mean, or more properly, the upper envelope curve of the total transmission error. This can also be achieved by recording the output of a low pass filter with a cutoff frequency

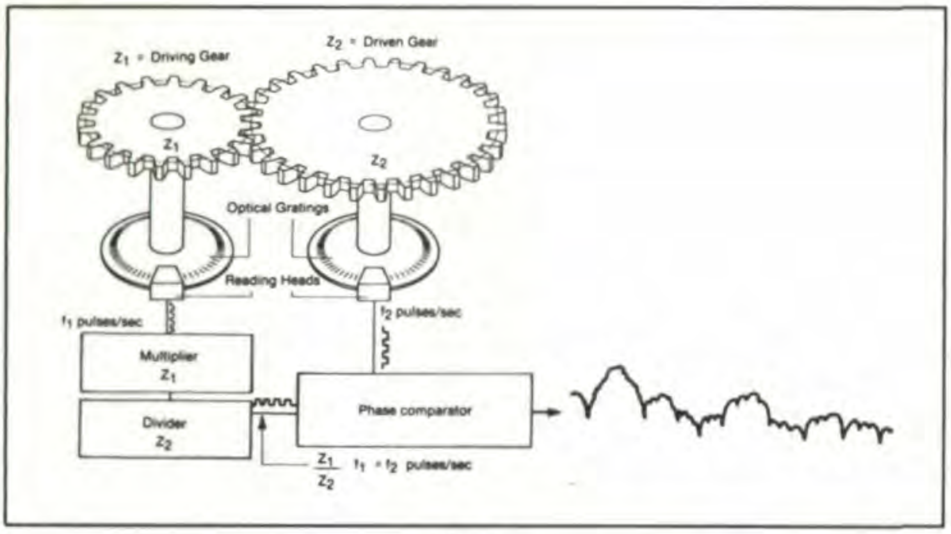


Fig. 2

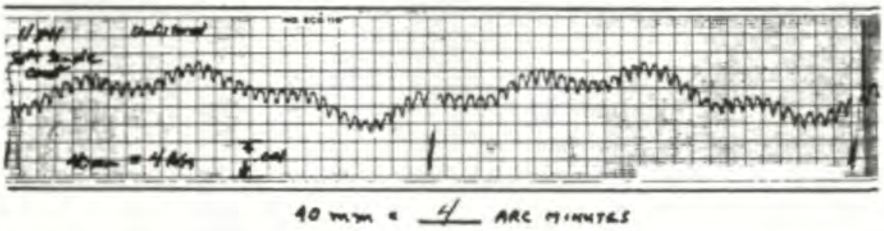


Fig. 3

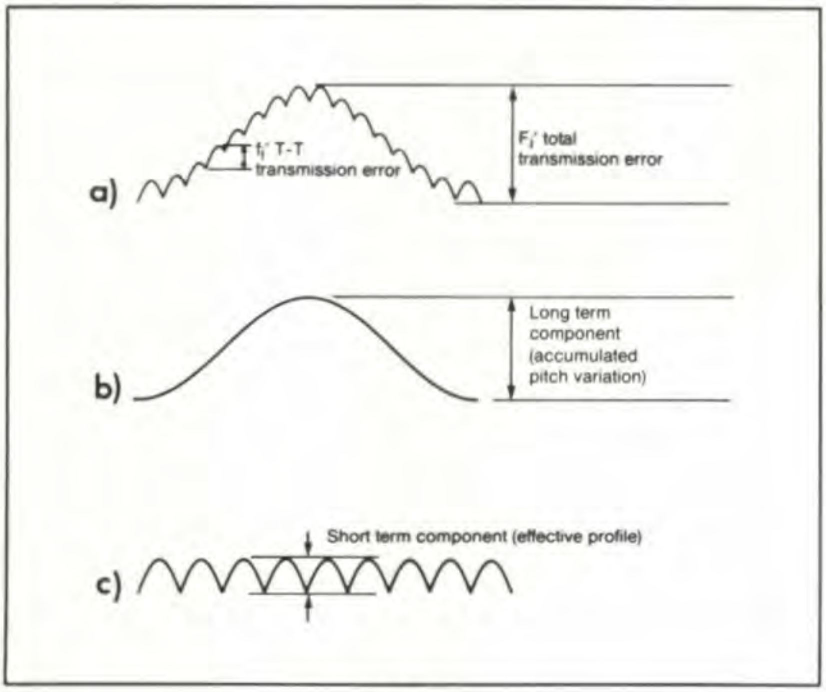


Fig. 4

properly selected to reject mesh and higher frequencies.

The resulting long term component, shown in Fig. 4B, is equivalent to accumulated pitch variation, assuming the gear was run with a perfect master. If a pair of work gears were run together, it would be necessary to use various cutoff frequencies to separate out the long term component of each individual gear.

Short Term Component

The short term component (effective profile) results from subtracting the long term component from total transmission error. This can also be accomplished by using a high pass filter with the cutoff frequency set just below the mesh frequency.

Profile(2)

Fig. 5 shows three typical tooth shapes and their resulting displacement motion curves. Each different tooth shape is shown rolling against a perfect master gear. Visualize the motion curve starting with the test gear tooth in contact near the root of the master tooth. The displacement curve generates as the teeth roll together and the point of contact moves toward the tip of the master tooth. This action repeats for each successive pair of teeth coming in contact or for each pitch. Fig. 5A shows the straight line generated by a perfect involute on the test gear, rolling with the perfect involute on a master gear. Fig. 5B shows the parabolic-like curve generated by a tooth modified with gradual tip and root relief. The zero displacement portion of the curve occurs when the teeth are meshing near the pitch line and the most negative portions occur when the teeth are in contact near tip and root. Fig. 5C shows the ramp shaped curve resulting from a pressure angle modification.

Influential Factors On Effective Profile (Short Term Component)

Helical or Bevels

Helical gears present a more complicated situation. In theory, the line of contact covers the entire face of the tooth as it rolls through mesh with the master so lead would have an influence on the motion curve. (See Fig. 6) However, from a practical standpoint, the tooth will generally have some lengthwise crowning. (See Fig. 7) This results in an instantaneous area of contact that will progress diagonally across the profile of the tooth. The motion curve is then dominated by the profile shape. This would not be true of long face helicals.

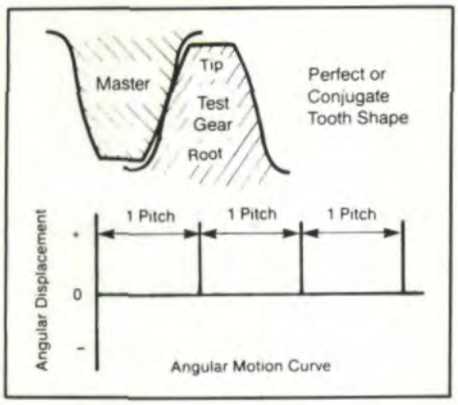


Fig. 5A – Angular Motion Curve

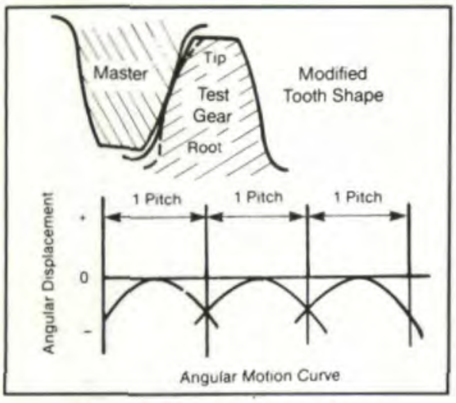


Fig. 5B – Angular Motion Curve

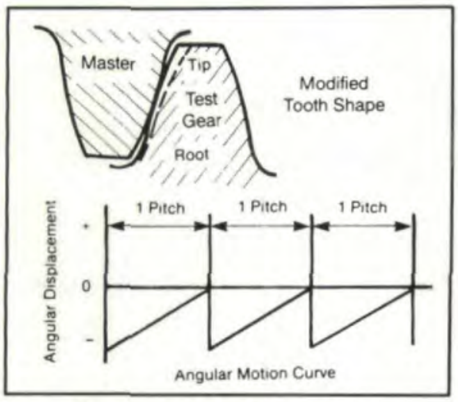


Fig. 5C – Angular Motion Curve

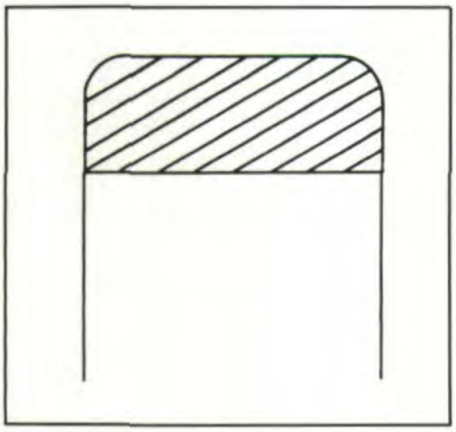


Fig. 6

Lead or Tooth Alignment[3]

Lead or tooth alignment, even on spur gears, is the one element least applicable to single flank measurement. It is impossible to look at a graph from a single flank test and quantify the amount of lead error in a gear. It can, however, have an influence on other elements of transmission error in a gear. Lead variation around a gear (wobble) will modulate the effective profile information. (See Fig. 8) Decreased contact ratio, due to high lead error will increase the effective profile or conjugacy error. However, it would be possible to have a large lead error, resulting in the tooth contact being at one end of the tooth. (See Fig. 9) If the tooth had good profile conjugacy, the resulting single flank graph would be a straight line. In this case, it would falsely indicate a good gear that might fail due

Fig. 7

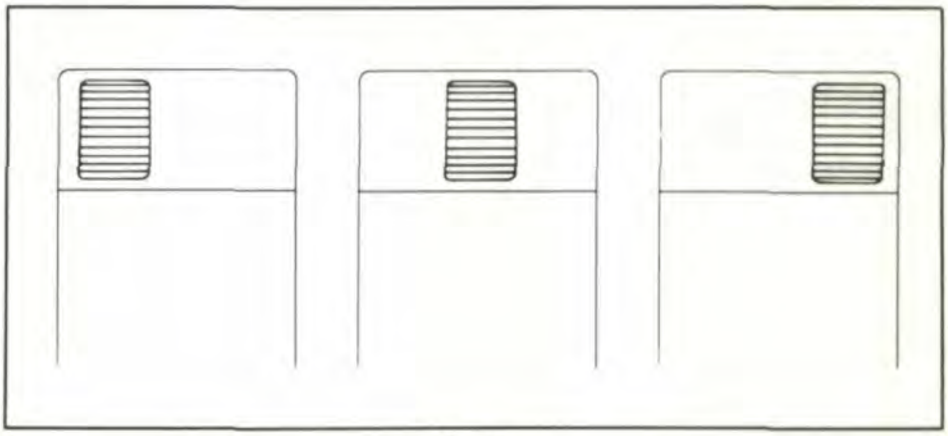


Fig. 8

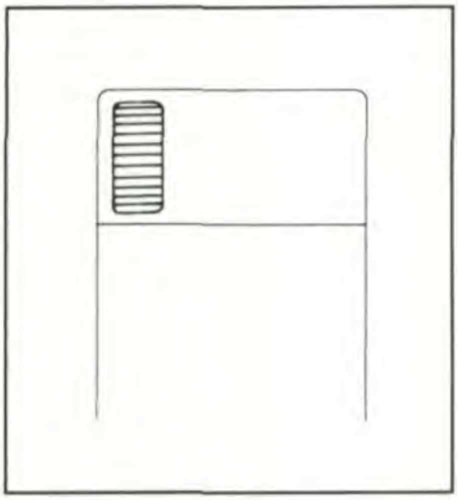


Fig. 9

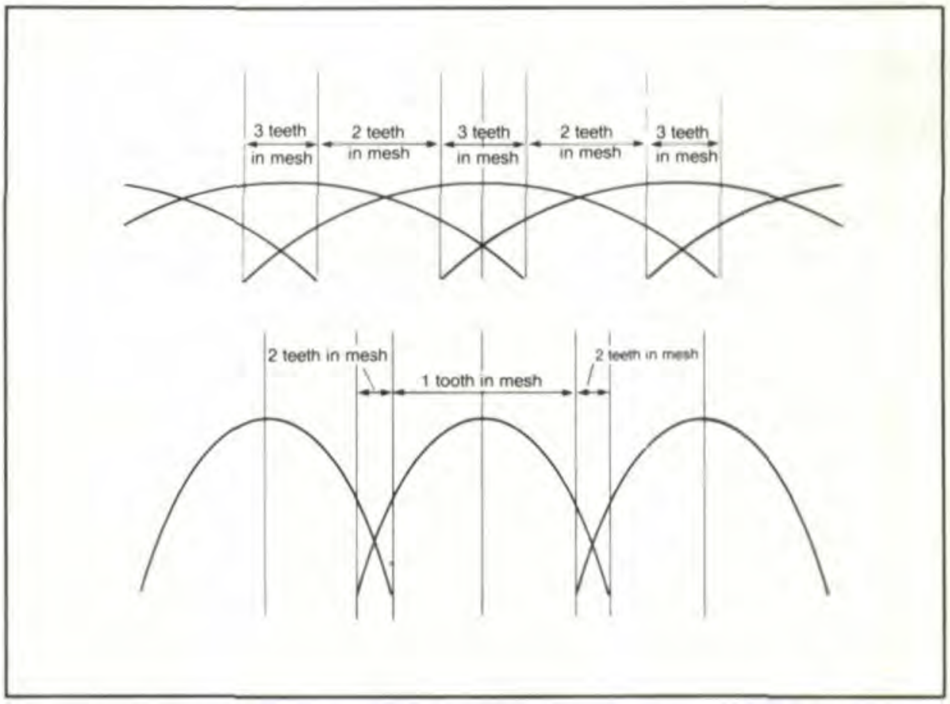


Fig. 10

to strength or surface durability considerations.

Contact Ratio

Consider a spur gear running with a perfect master gear. The gear is designed to have a contact ratio of exactly one and has curvature modification on the profile. The amplitude of the short term component of transmission error would be equal to the involute variation. If the gear was redesigned to increase contact ratio, the amplitude of the short term component would decrease and represent only a portion of the total involute variation.

If we considered a helical gear set, the total contact ratio (transverse plus face) would increase, and the effective profile error (short term component) would decrease even more.

However, there is a big difference between theoretical contact ratio and actual contact ratio. Contact ratio calculations are based upon full length and full profile contact. From a practical standpoint, most gears are designed with profile modifications and lengthwise crowning. This is to allow for housing errors and deflections, as well as tooth deformation under load. If, under load, contact is carried across the whole tooth, the contact ratio calculations are valid. However, in many instances, the contact is still localized even under operating conditions. In this case, the real contact ratio is much lower than the theoretical. In one helical design studied recently, the theoretical contact ratio was 3.4. Based on actual contact area, due to severe localization, the real contact ratio was 1.6. Fig. 10 shows the effect of different contact ratios on the effective profile amplitude shown in the single flank graph. There is a tendency to use too much localization on lightly loaded applications and, therefore, increase the possibilities of noise excitation.

Load Effects

Most single flank inspection is done at relatively low loads in typical test machines. To test at high loads, transducers must be attached to the housing used in the application. This is time consuming and expensive, but can be done in laboratory tests.

However, many applications such as vehicle drive gears usually have noise problems at light drive or float conditions. This correlates well to the lightly loaded single flank testers. In this case, a low effective profile error is a desirable condition. The gears can also be tested

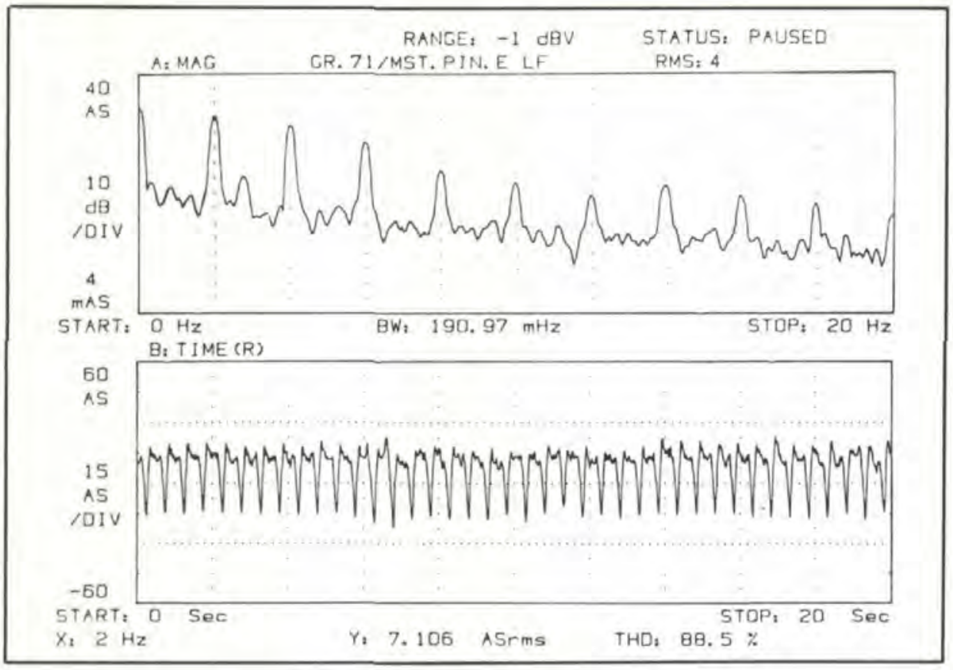


Fig. 11

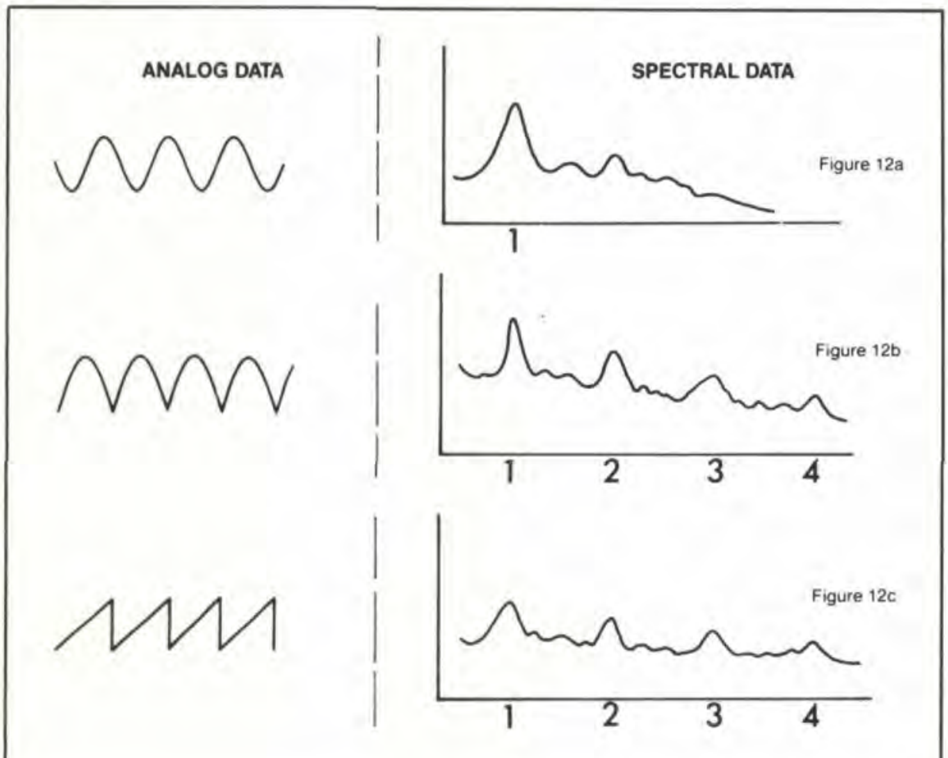


Fig. 12

at different positions to simulate load or thermal deflections of the housing.

If the gears are to be used in a heavily loaded application, then chances are that the tooth shapes have been modified, and that they won't run smoothly in a lightly loaded single flank test. In this case, the job must be investigated experimentally to decide what the desirable transmission error should look like under light loads.

Smith[4] and Mark[5] discuss the effects of tooth deflections, modification and transmission error to a greater extent.

Fig. 11 shows a highly loaded aircraft gear with tip relief running at low loads in a single flank tester. The large negative drop in the curve is the result of tip relief. If the amount of tip relief is correct, the curve would smooth out at operating loads.

Housing Dynamics

Gear noise is really a system problem, not just a gear problem. Most gearing used for power transmission is enclosed in a housing and, therefore, little or no audible sound is actually heard from the gear pair.[6] The minute vibrations created by the gears as they move through mesh are amplified by resonances of structural elements. This amplification occurs when the speed of the gear set is such that the meshing frequency, or a multiple of it, is equal to a natural frequency of the system the gears are mounted in.

Some structural systems are such good amplifiers that it is nearly impossible to make gears good enough to run quietly in them. When this happens, the gearing becomes unnecessarily expensive. At some point, it is more productive to modify the system and make it less critical to gear excitation.

This is done by making a modal study of the structure and then applying corrective measures. Modal studies can be done by actual vibration measurements, or theoretically, by the use of finite element techniques. Corrective measures may include a change in panel stiffness, ribbing in the housing, application of damping techniques or changing the path of transmitted excitation. In the case of vehicle noise, rubber isolaters or tuned absorbers are often used. It may also be possible to change the number of teeth in the gears, or the operating speed, in order to move the mesh frequency away from the resonant frequency. Smith[7] is a source of more information on this subject.

Spectral Analysis

As mentioned earlier, frequency is the key to understanding gear noise. Spectral analysis is really better described as frequency analysis. The analytical system takes a time varying signal and breaks it down into a spectrum of individual frequencies. The amplitude of any frequency of interest is actually the amplitude of a discrete sine wave contained in the more complex time domain waveform. This can be done with analog instrumentation, such as variable narrow band pass filters. Today, it is more often done digitally by the use of "real time analyzers" and a Fast Fourier Transform (FFT) algorithm. Most of these RTA's will display both the time domain data as well as, the frequency domain. (See Fig. 11)

Interpretation of Data
Analog Data
(Time Domain)

Analog data is useful for judging the actual shape or geometry of the tooth form. It is easy to look at the filtered short term component (effective profile error) and decide what corrective actions to take on tooth development. It is also useful, in the unfiltered state, for a quick judgement of accumulated pitch variation of each gear in the pair (long term component).

Spectral Data
(Frequency Domain)

The spectral data, on the other hand, is useful for identifying the characteristics of a transmission error curve that are the source of noise excitation. It takes a complex analog waveform and simplifies the analysis of it.

Relationship to Typical Motion Curve

A look at the relative amplitudes of the various harmonics in a spectrum can sometimes be useful in judging the characteristic wave shape. (See Fig. 12)

A lapped hypoid set of gears will often show an "almost sinusoidal" effective profile curve. The spectrum will show a discrete peak at the 1st harmonic of mesh, with the rest being white noise (all frequencies). (See Fig. 12A)

The most typical effective profile curve, for either spur and helicals or bevels, will be of the form that is nearly parabolic. This will generally show discrete peaks at 1st, 2nd, 3rd and possibly higher harmonics. The second harmonic will be approximately 12 Db lower than the 1st and the 3rd will be approximately 18 Db down. (See Fig. 12B)

The next most typical is the ramp shaped curve. In this case, the second will only be approximately 6 Db down and the 3rd approximately 10 Db down, from the first. This usually results from something like a pressure angle error and is easily recognized when the spectrum has excessive amounts of higher harmonic content. (See Fig. 12C)

These relationships can vary, depending upon how "pure" is the waveform.

Applying Spectral Analysis To Noise Problems

The characteristics of the dominant peaks displayed in any spectrum are very useful to the solution of gear noise problems. Different types of noise problems will relate to different characteristics. There isn't one simple piece of information that will relate to everyone's noise problem.

If one looks at the analog data of the filtered (long term component removed) effective profile error throughout one revolution of the largest gear, it will be observed that the shape and amplitude of each tooth mesh may vary. These geometric deviations, from that of conjugate gear teeth, are made up of two components; mean and random.[8] The mean geometric deviation component for a pinion or gear is defined as the tooth surface formed by taking the average of all tooth surfaces on the pinion or gear under consideration. The random component of the geometric deviation of a tooth surface is defined as the deviation of that tooth surface from the mean tooth surface. Thus, every tooth surface on a pinion or gear has the same mean deviation, but the random deviation generally will differ from one tooth to the next.

In the spectral data, the mean component relates to the mesh frequency and integer harmonics. The random component relates to sideband peaks. (See Fig. 13)

Mean component deviations of normal tooth shapes will create spectral patterns similar to those shown in Fig. 14. Extreme tip and root relief modifications will look like the data in Fig. 11.

Random component deviations can be caused by pinion runout, gear runout, and cyclic distortions of tooth profile forms due to heat treatment. The results of these deviations can be seen on the spectrum as sidebands. Sidebands will occur at the mesh frequency, plus and minus the frequency of the event passing through mesh. They could also occur at plus and minus the event frequency, from other harmonics of mesh. (See Fig. 13)

Other frequencies that are important are sometimes called "ghost" or phantom frequencies. These will be found in the spectrum between integer harmonics of mesh or runout frequencies. They also can occur as unusually high amplitudes of an individual integer harmonic. (See Fig. 13)

Ghost harmonics are generally due to flats or facets on the normal tooth form and are caused by such things as cutter runout or non-uniform motion of an element within the gear train of the machine that generated the tooth form. This could be caused by tooth to tooth transmission error of the "final drive gears" or runout

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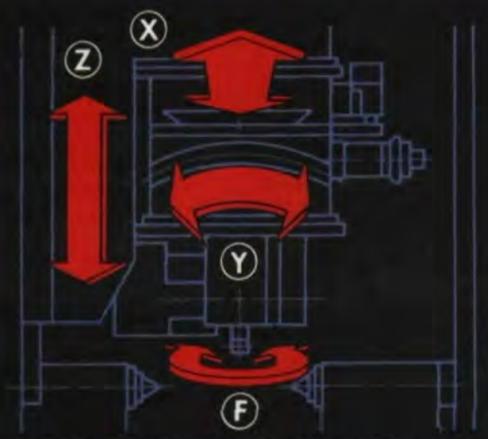

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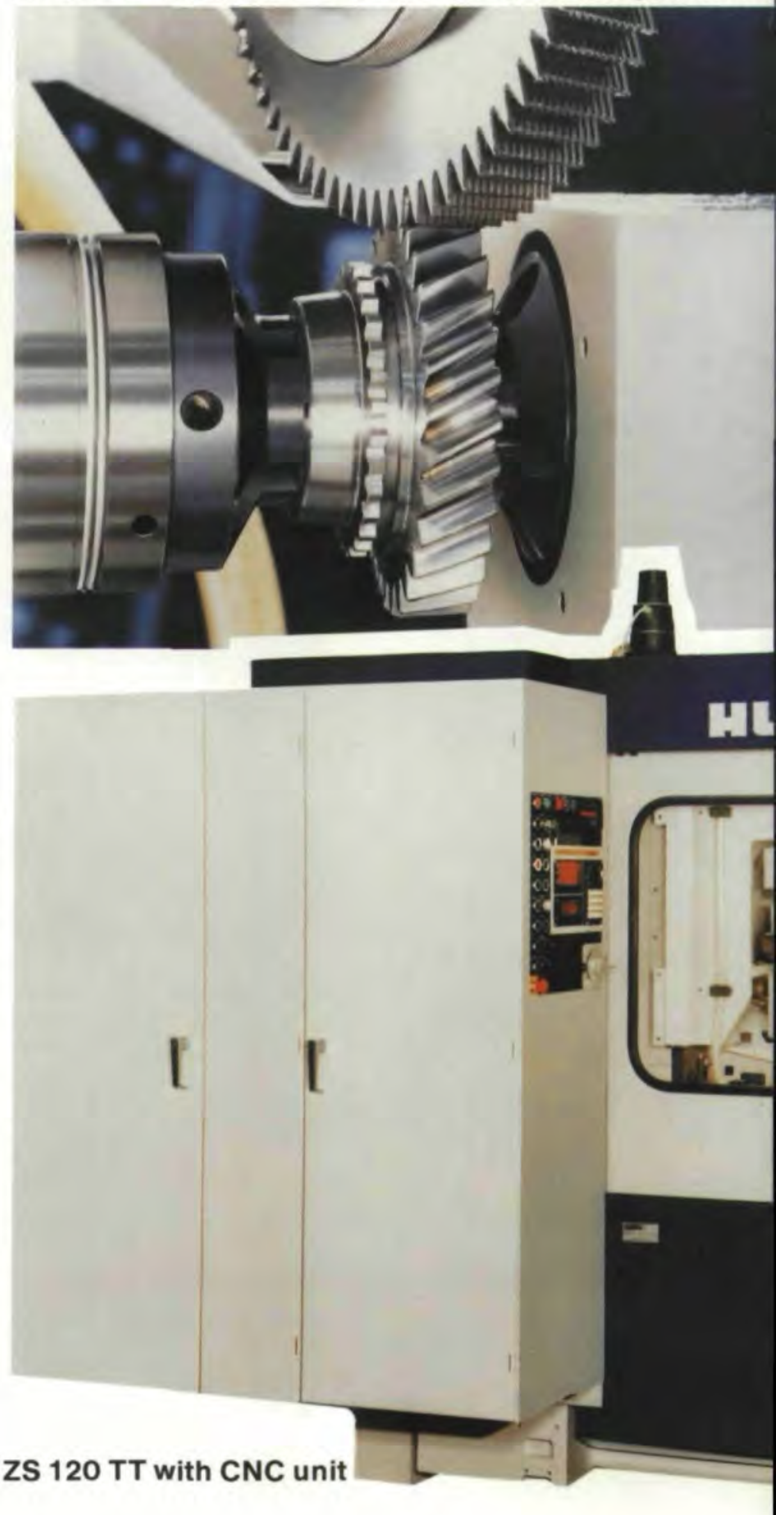

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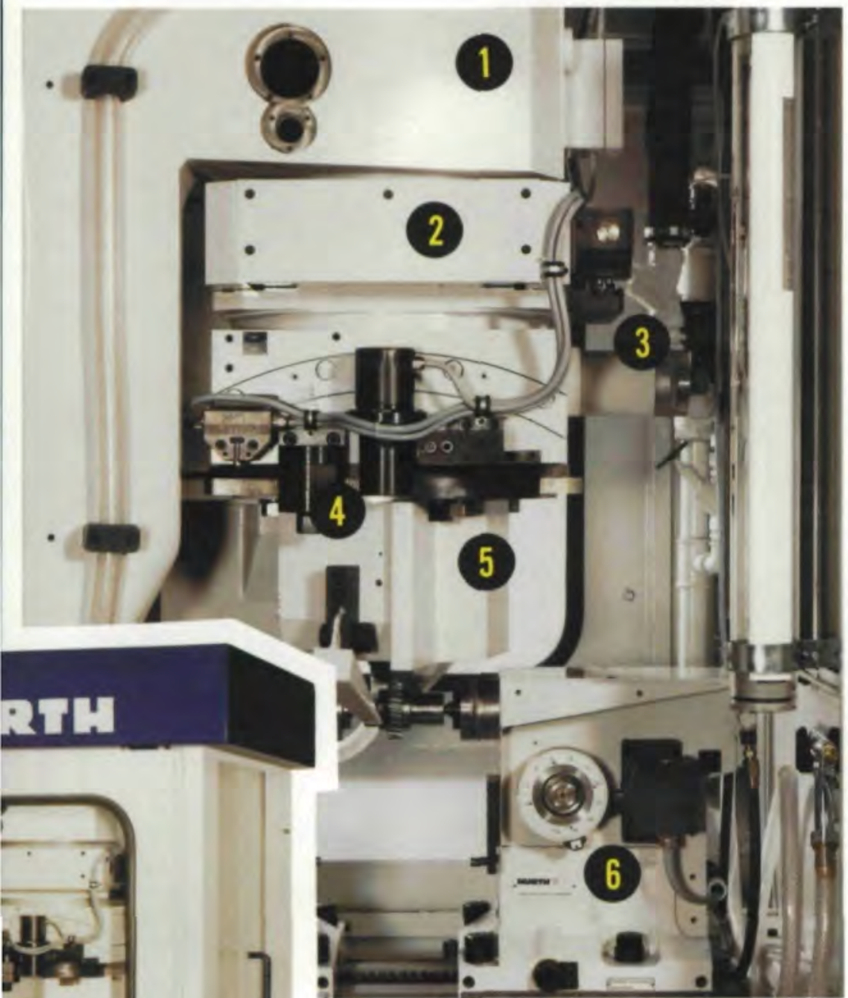

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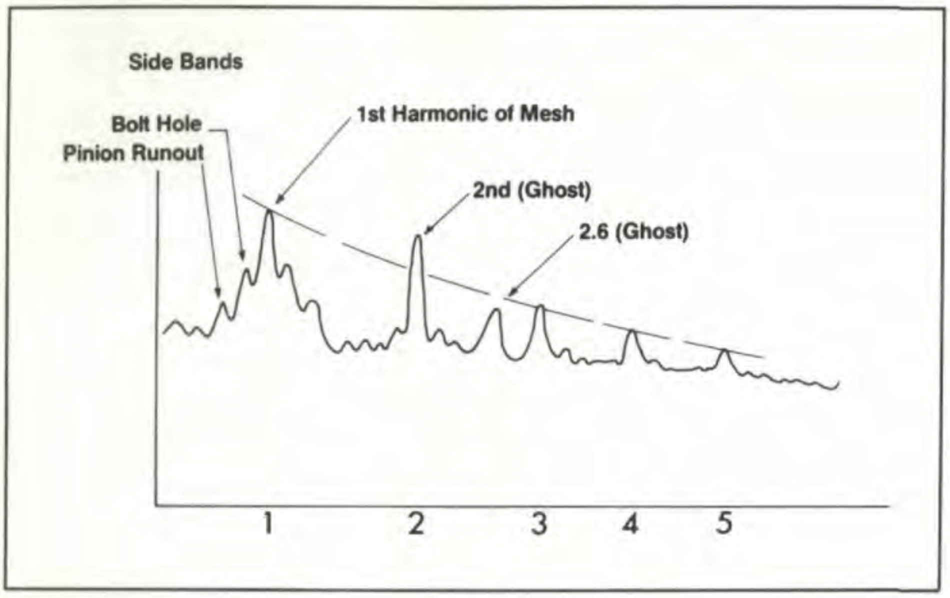


Fig. 13

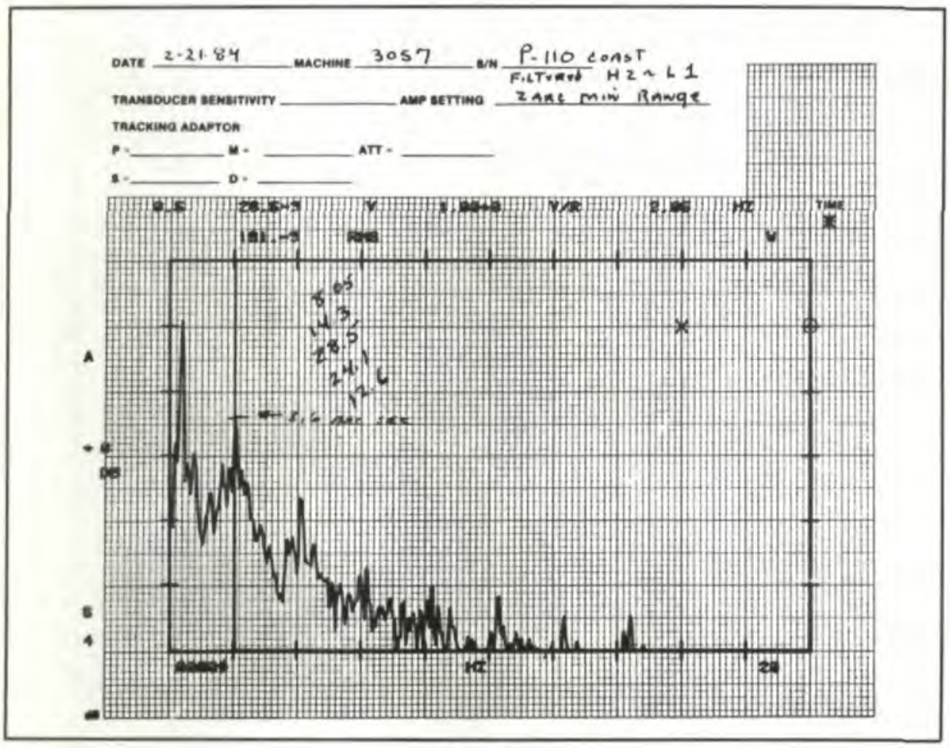


Fig. 14

of other gears or shafts within the drive train.

Case Histories

The following case histories are included to illustrate some of the points discussed above.

1. TYPICAL REAR AXLE NOISE. Fig. 15 shows a set of lapped hypoid rear axle gears. They show the typical parabolic shaped effective profile error, but of an excessive amplitude (.00035"). They were rejected in the vehicle for several "noise periods". This is evident from the content of higher harmonic amplitudes.
2. NOISE-THERMAL DEFLECTION. This case illustrates an unusual situation. They are hypoid gears used in a rear axle, but the housing is made of aluminum. Tests were run on a good set and a reject set. Both sets looked good at the build position. However, it had been previously determined that the pinion moved plus .004" on pinion cone, at the elevated operating temperature. It was shown that the reject pair got worse at the thermally deflected position, but the good set actually improved. (See Fig. 16)
3. NOISE-CONJUGACY VS. ACCURACY. Fig. 17 is used to illustrate the fact that accuracy isn't necessary for noise control. It shows a lapped hypoid pair of gears with a relatively large total transmission error from accumulated pitch and bolt hole distortions, but with a very low tooth to tooth transmission error (less than .0001"). This was a quiet pair in the axle. The other pair was an experimental ground pair with very low total transmission error, but with a very regular high tooth to tooth error (.0002"). This pair was noisy in the vehicle at 1st and 4th harmonics of mesh frequency. It had a small amplitude of waviness superimposed on the tooth to tooth waveform that caused the 4th harmonic ghost noise. In this case, the vehicle was a van type, which is typically sensitive to excitation due to structural dynamics.
4. *PRECISION GROUND HELICAL* GEARS. A first look at the analog graphs (Fig. 18A and 18B) of these two sets might lead to the wrong conclusion. Spectral analysis, however, points out that the first set (Fig. 18A) is very conjugate (hardly any discernable peaks in the data). The spectrum of the second set (Fig. 18B) shows a high 2nd harmonic relative to the 1st. Remember, that these measurements are of angular displacement. If the data were double differentiated, it would represent angular acceleration which is proportional to force and would be more indicative of noise potential. The acceleration goes up by the square of the frequency. This high 2nd harmonic was difficult to identify by interpretation of the involute charts, but is easily discernable by single flank measurement.
5. GROUND AIRCRAFT SPUR GEARS-HIGH RPM. Although the major concern is not noise, because of high altitude operation, the gear producer was experiencing dynamic loading conditions that were excessive. Strain gaged data had shown this. The single flank tests, especially the spectral data, readily showed a high 2nd harmonic of mesh amplitude that coincided with their strain gage

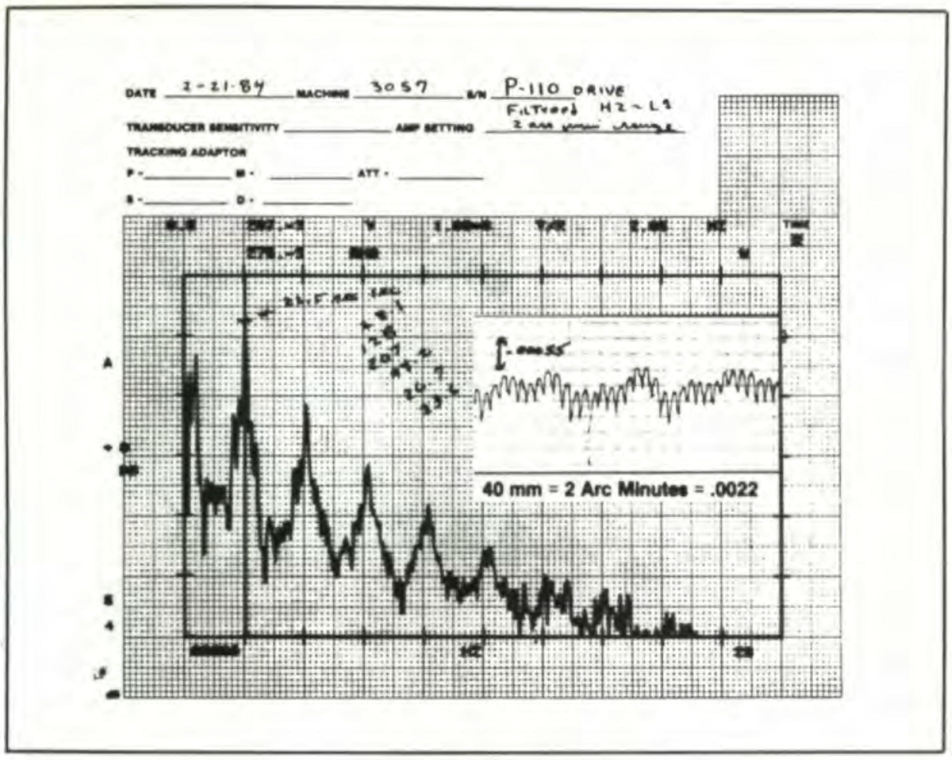
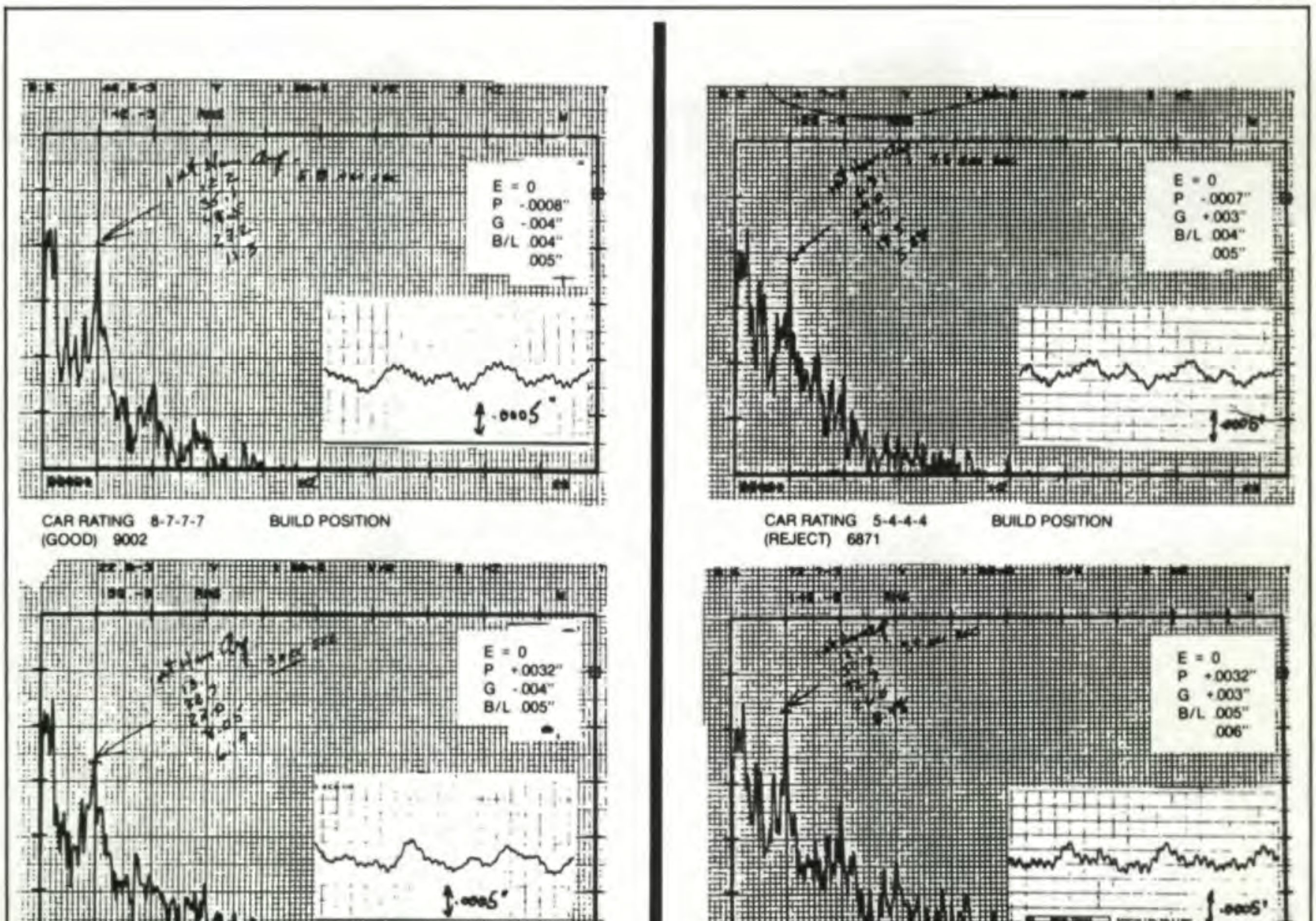


Fig. 15

data. Again, it had been difficult to find the problem with elemental tests such as involute checks (Fig. 19).

6. GHOST NOISE. Fig. 20 shows data taken from an internal helical used in the transaxle of a front wheel drive vehicle. Sound tests of an operating vehicle had detected "ghost" noise at the 1.7th harmonic of gear mesh. The gear had been marked with the number of the gear shaper that produced it. Further tests found other gears from the same shaper as well as others with the same problem. A series of gears were cut, one from each shaper using the same cutting tool and workholding equipment. They were all single flank tested with a master gear and the spectral data was checked for existence of this "ghost" harmonic. The offending machines were identified from the single flank data. A second "ghost"

Figure 16 – below



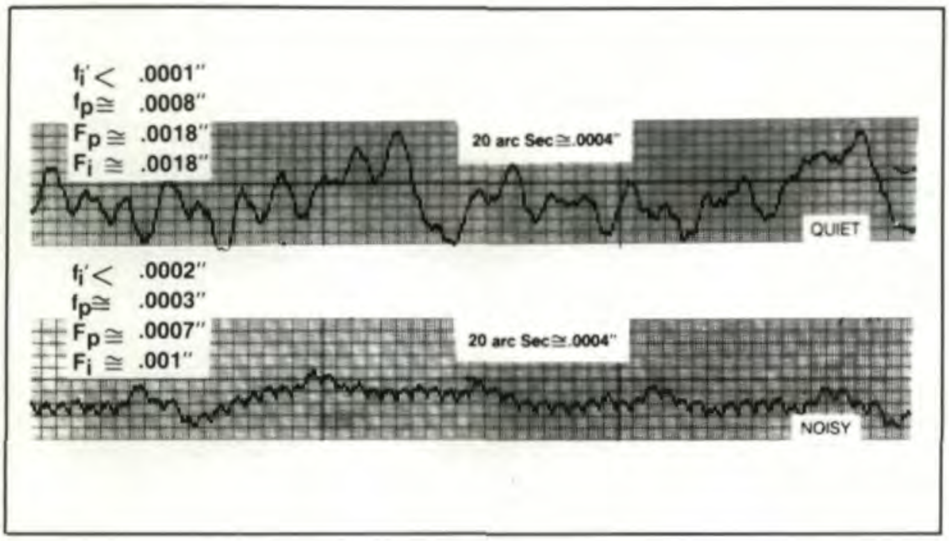


Fig. 17

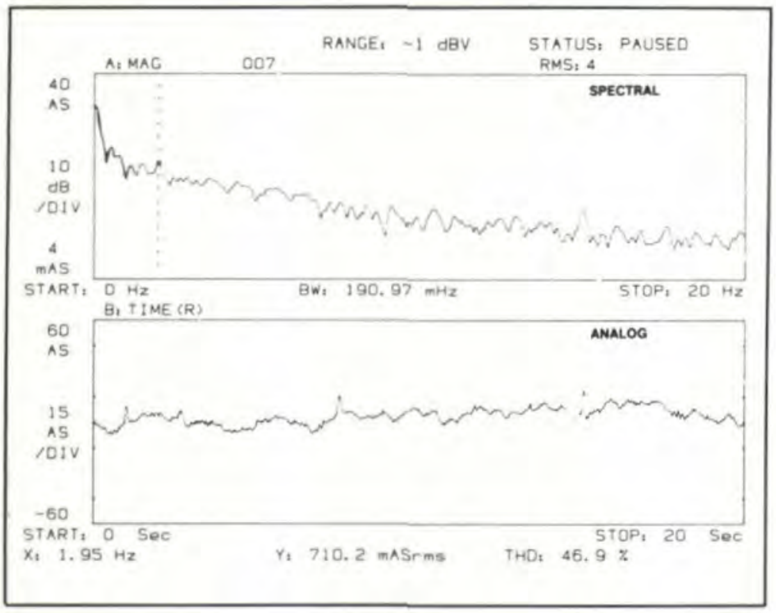


Fig. 18A

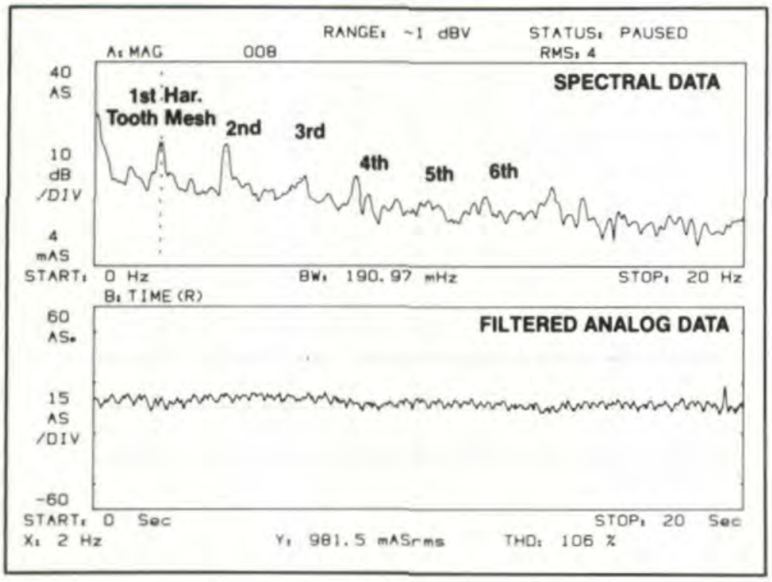


Fig. 18B

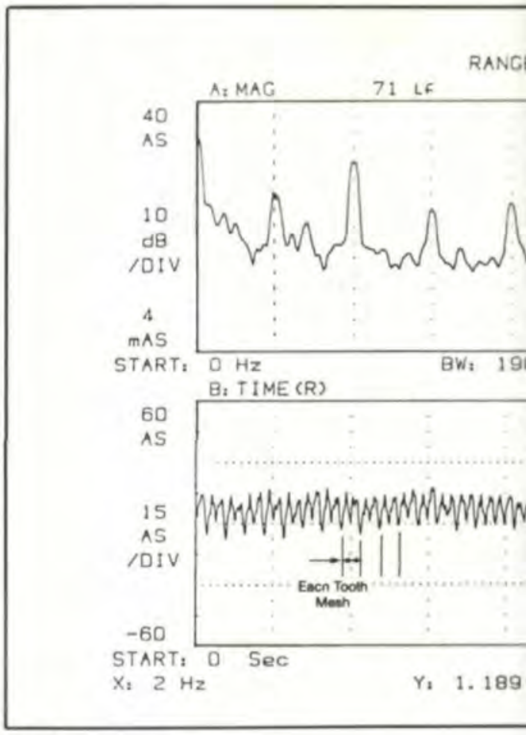
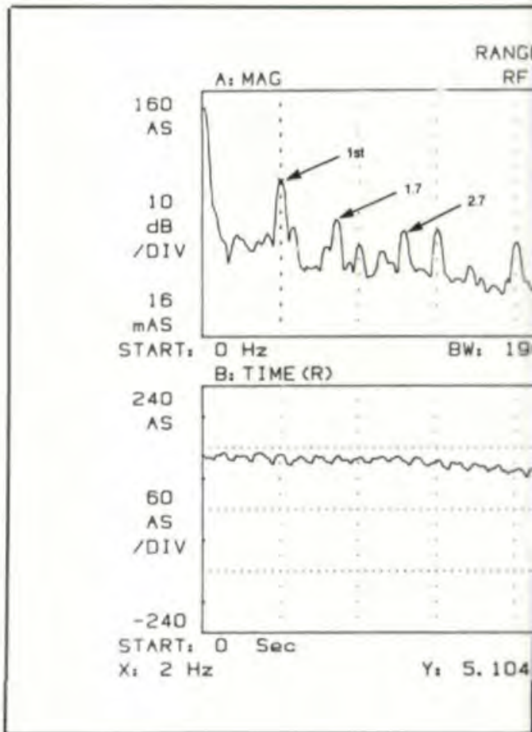


Fig. 19



harmonic was also found at the 2.7th harmonic.

An analysis of the gear train in the shaper showed that the 1.7th harmonic was from the tooth mesh frequency of the table drive worm and wheel, and the 2.7th harmonic was from the tooth mesh frequency of the cutter drive worm and wheel.

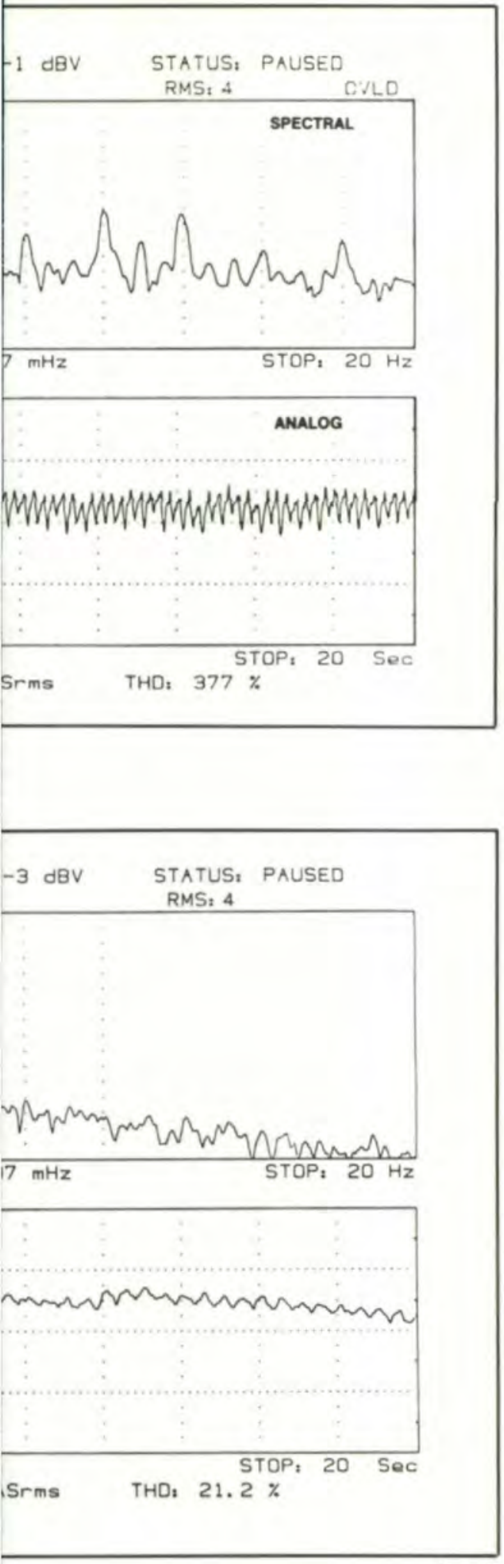


Fig. 20

Conclusions

Gear noise is a very complex subject, as can be seen in this information. This article does not pretend to cover all aspects of it. However, it is hoped that it will help the users of single flank equipment bring gear manufacturing out of the state of being a "black art."

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