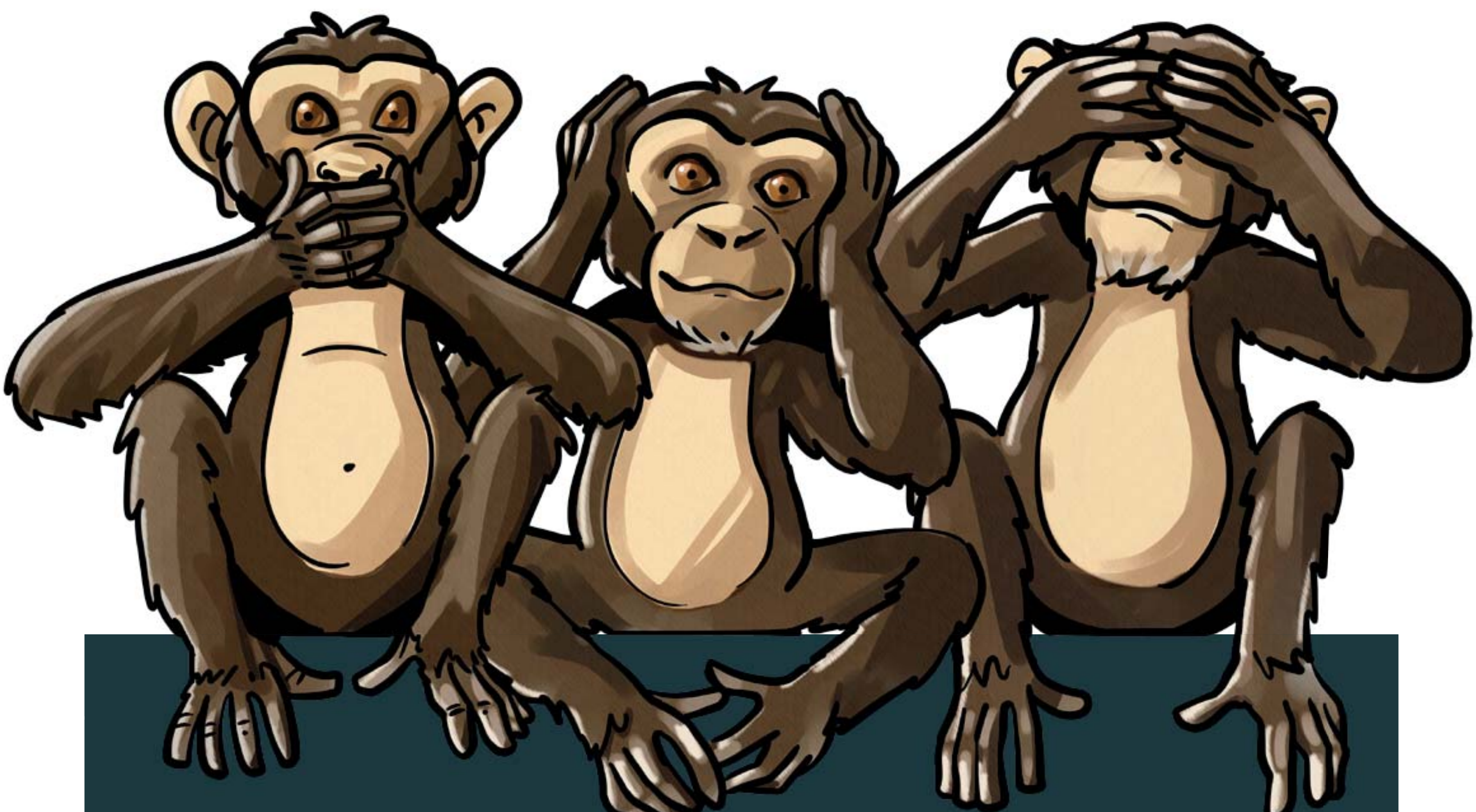
NOISE GENERATION IN GEARINGS: SEE NO EVIL, HEAR NO EVIL, SPEAK NO EVIL ...



You may recall the story of the three monkeys, who see no evil, hear no evil, or speak no evil. One is often reminded of this when searching for the cause of noise in gearings. When it comes to evaluating the quality of a toothed gear pair, there's no shortage of criteria. Many manufacturers have invested their know-how in proprietary evaluation methods to assess one geometric feature or the other. But what geometric features are suitable for obtaining a correlation?

The more precise the teeth of a gear are, the better it must be: that's what every quality manager hopes. Yet the saying "the more precise the better" hasn’t seemed relevant for quite some time. Ever since a transmission's noise behavior became the primary focus of vehicle acoustics engineers, it is clear that quality and noise characteristics do not go hand in hand. Thus for many production managers, a daily battle ensues on the manufacturing floor against the end-of-line test stand, which in many respects resembles Don Quixote's duel against the windmill. It is even more exasperating when a transmission of merely average toothed gear quality in the vehicle delivers thoroughly acceptable noise behavior while high-quality toothed gears prove to be annoying sources of noise. Why is this so?

Gearing noise can only occur as a result of tooth contact. Because tooth contact is determined by the shape of the tooth flanks, one might imagine that one could predict the noise behavior of a toothed gear pair based on the geometry of the tooth flanks. Experience teaches us, however, that conventional toothed gear measurement is only a suitable instrument for controlling production and is no good at predicting noise behavior.

Yet in many places, tolerances for the geometry of the tooth flanks and gear body are continually being narrowed – in the hope that a more precise geometry will lead to improvement.

This article will examine the subject of noise analysis with a view to understanding it better. No one disputes the fact that tooth contact is always the source of noise. We must therefore find out what causes rotary transmission errors in a toothed gear pair and look for ways to objectively describe the properties of the actual rotary transmission while at the same time pinpointing the causes of faults, however small they may be.

Gear Design: Transmission Error Analysis …

The ease-off is always the centerpiece of a bevel gear set design. For the design engineer, this representation of the transmission error-free minimum distance between two intermating tooth flanks is as descriptive as it is meaningful (see Figure 1).

By examining the ease-off, one can immediately identify the flank profile modifications and obtain an initial impression of the sensitivity to production-related or load-related displacements between the ring gear and pinion. But the ease off is also suitable for determining a computational contact pattern and the theoretical transmission error.

The transmission error of a bevel gear pair can easily be calculated based on the ease off. For every angular position of the gear and pinion, there is a specific distance of the flanks to one another. If

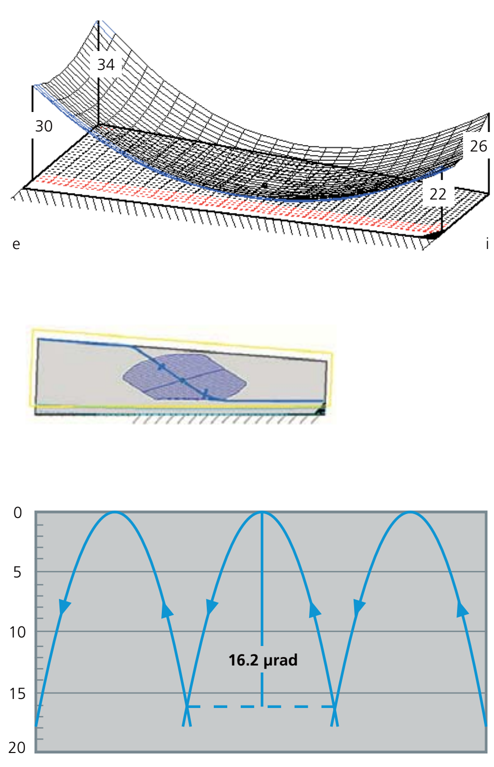


Fig. 1: Ease-off, contact pattern, and transmission error

Hear No Evil …

Although everything can be seen on a measuring machine, there is no indication of conspicuous noise behavior.

we turn the ring gear until it comes into contact with the pinion flank, the intermating flanks will come into contact at precisely one point. This point contact is the consequence of crowning on the tooth flanks. If we had an ease-off with a contact distance of zero, the flanks would contact each other in a line. This is called a conjugate gearing.

The line on which all contact points are located is called the path of contact. Because each point on the contact path also has an associated pinion angle, the start of engagement and end of engagement of the respective tooth pair can be determined by means of the pitch.

If we project the contact path onto the ring gear flank in the ease-off, the distance of a point on the contact path in the ease-off is the transmission error for this angular position (see Figure 2). For conventionally designed gearings, we obtain a parabolic transmission error for a tooth pair. The transmission error curve for all teeth is obtained by copying the transmission error curve for one tooth pair and shifting it one pitch. The transmission error curve for the next pair of teeth intersects the current transmission error curve at the point where the individual engagement starts and ends. We thus obtain the enveloping curve shown in Figure 2 as a no-load transmission error of a bevel gear pair.

… and Order Analysis

Depending on the design of the tooth flanks, this transmission error is more or less tall, pointed, wide, or short. Of course, such attributes are not suitable for a quantifiable representation of a transmission error curve. For many years, order analysis has been established as a suitable tool for describing periodic functions. Order analysis is based on the Fourier transform. In his work, the French mathematician Joseph Fourier postulated that every periodic function could be bro-

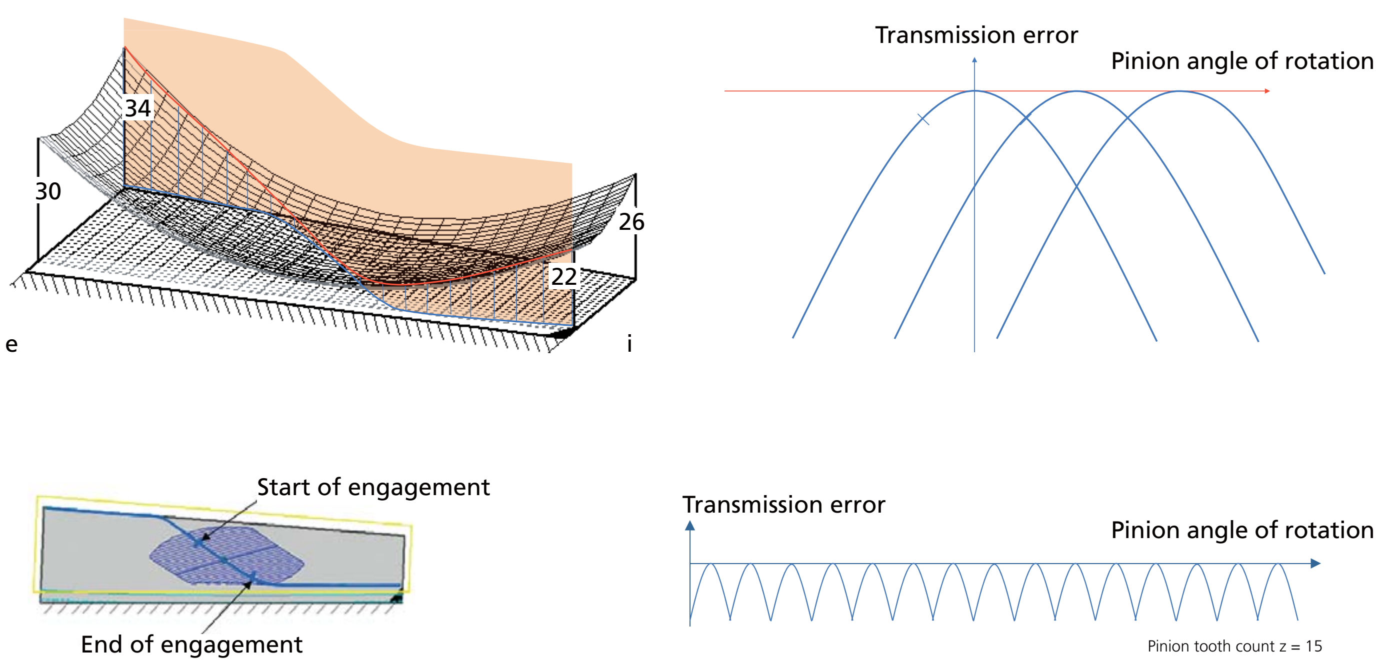


Fig. 2: Contact path, individual tooth contact, multiple tooth contact, and transmission error

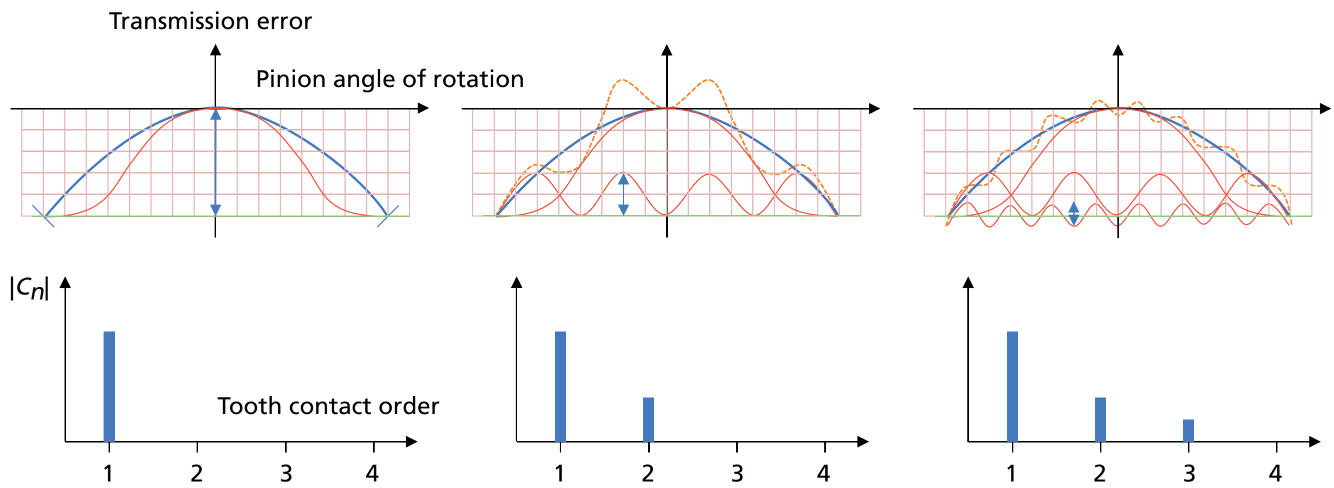


Fig. 3: Order analysis of a transmission error curve

ken down into harmonic sine oscillations of a different phase, amplitude, and frequency (see Figure 3).

The blue curve is the transmission error curve for a tooth pair. The first step is to look for a sine curve with a wavelength that corresponds to the length of the tooth mesh. The amplitude and phase should be chosen to ensure that the sine curve approaches the blue curve as closely as possible. The amplitude of this sine curve is called the first order. Each subsequent step in the analysis looks for a sine curve with half the wavelength of the curve in the previous step, such that the summation of all previous sine curves approaches the blue transmission error curve as closely as possible. Each amplitude here is called an order. The cumulative curve appears as a dotted red line in Figure 3. As mentioned above, Fourier advanced the theory that every periodic curve (and the transmission error is such a curve) can be precisely described as an infinite number of sine curves with increasingly shorter wavelengths.

This principle was adopted to describe the transmission error curve by the magnitude of orders, where the nth order corresponds to the amplitude of the sine function of wavelength 1/n. Thus we have found a suitable mathematical, number-based tool for describing the transmission error curve.

Among vehicle acoustics engineers, however, another reference has become the accepted way to describe the transmission error of a gear pair: Instead of the tooth contact order, an entire revolution of the pinion is used as a reference. If we have a pinion with 15 teeth, for instance, the first tooth contact order becomes the 15th order of the pinion when using this reference.

It is important to know that with this description of the transmission error curve, only integer multiples of the number of teeth can occur in the order spectrum. Because a Fourier analysis has as its basis a sine function with a half wavelength from order to order, an order that is not divisible by the number of teeth will never occur.

Manufacturing Errors: Inaccuracies …

But we are still in the wonderful world of gear design and have no idea of manufacturing inaccuracies, which significantly

See No Evil …

In a single flank test, we obtain an order analysis with tooth contact orders and ghost orders. We can essentially "hear" the gearing, but there's no one to tell us what the cause is.

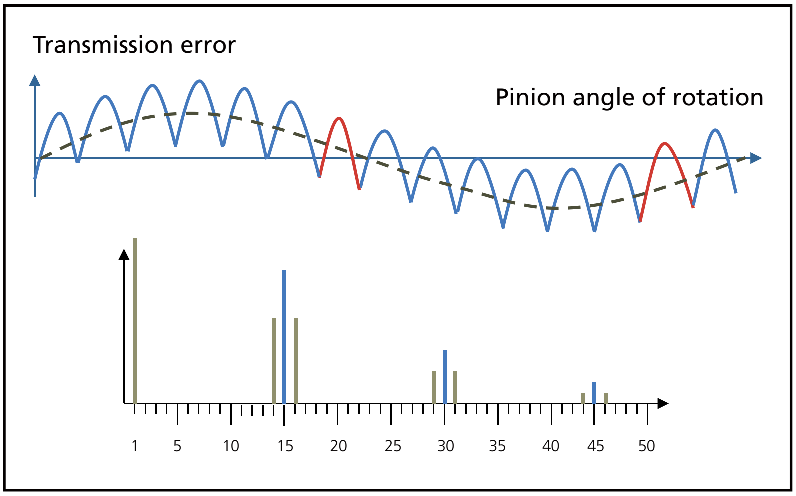


Fig. 4: Radial runout

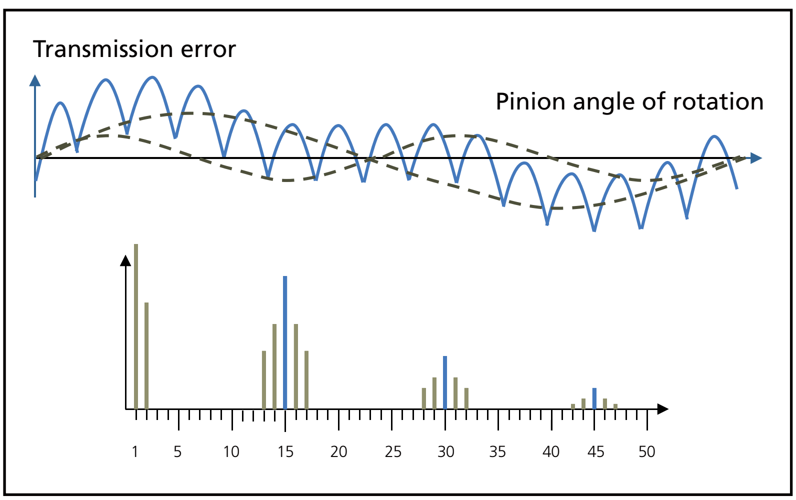


Fig. 5: Radial runout and double strike

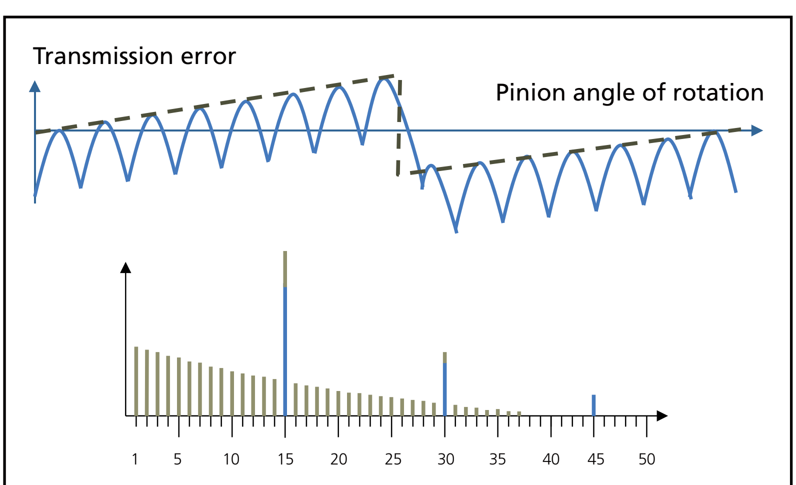


Fig. 6: Pitch error

change our transmission error curve from tooth to tooth. What is the impact of manufacturing errors? What happens, for example, in the event of radial runout of the gearing resulting from an eccentric clamping in the machine tool?

Figure 4 shows the transmission error curve for a component with radial runout. The individual tooth meshes are no longer lined up straight as in Figure 2; instead, they are "strung out" along a sine wave. Certain tooth pairs generate a shorter, less pronounced transmission error; some have a longer contact with a larger error, as illustrated by the curves in red. This is termed amplitude modulation and frequency modulation; in an order analysis, this can be seen as sidebands. The eccentricity in itself appears in the first order of the pinion.

If the radial runout is not a simple sine wave, additional sidebands occur alongside the tooth contact orders (see Figure 5). A so-called "double strike" is present here, in addition to the radial runout.

Another effect is manifested for indexing errors. In this case progressively diminishing orders are produced instead of sidebands (see Figure 6). Such pitch errors can occur in ground bevel gear teeth or profile-ground cylindrical gears. Due to the wear of the grinding wheel that machines one tooth gap after another, the teeth become continually thicker. A large indexing error can occur as a result during the transition from the first to the last tooth gap. This is evidenced in the order spectrum by a multitude of orders rather than by sidebands.

... and Ghost Orders

Every order in the spectrum that cannot be divided by the number of teeth is called a ghost order – for lack of a better term. Ghost orders are given particular attention during noise analysis because they cannot be influenced by an alternate

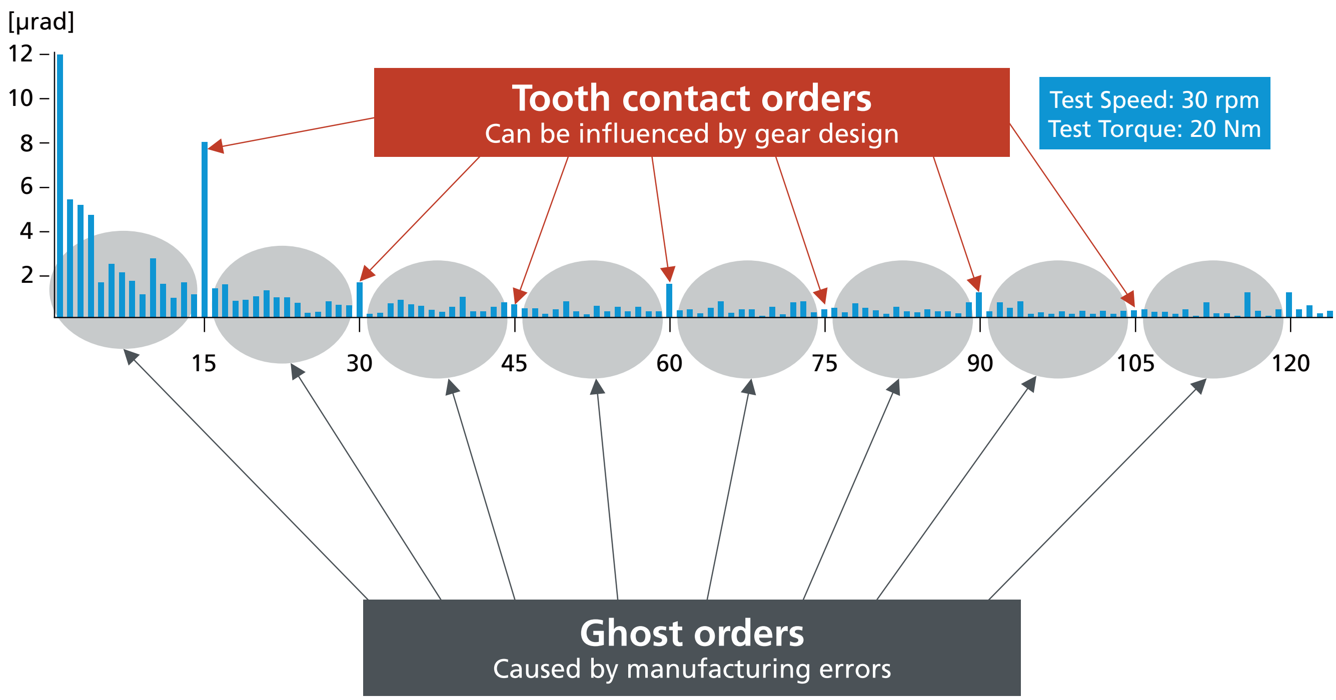


Fig. 7: Orders for tooth contact and ghost orders

design of the gear geometry; rather, they seem to be conjured up as if by an apparition, the result of a variety of manufacturing inaccuracies.

Figure 7 shows a measured transmission error curve for a gearing with 15 teeth at the pinion. Some of the tooth contact orders are clearly identifiable as the 15th, 30th, 45th, 60th, 85th, 90th, and 105th order. But in between there are many other ghost orders, which, due to their unclear origin, are difficult to control in terms of optimizing gearing noise.

The Third Monkey Speaks – but We Must Understand Him

The image of the three monkeys is reasonable accurate for describing the dilemma represented by measuring technology, running tests, and sources of noise. To make the third monkey talk, we must find the geometric feature responsible for generating the noise perceived as so unpleasant by the human ear.

Measuring Principle for Cylindrical Gears

If we consider the profile measurements of a cylindrical gear, for example, we can identify a number of effects: There are geometric errors such as flank angle errors, imperfect crowning, or an incorrect relief. Surface defects on the tooth flank become visible only when the macro-geometric errors are eliminated.

Figure 8 shows profile measurements from which geometric errors have already been extracted. If we superpose all the profile measurements, as shown in the section on the right, even the trained eye can no

Speak No Evil …

How can we get the third monkey – who can see and hear – to speak?

longer detect anything. One might assume that a tooth flank's roughness would be visible here. But this assumption is precisely what sets us on the wrong path.

In a straight gearing with an unmodified tooth trace, no-load intermating tooth flanks contact each other on a straight line parallel to the tooth root. For every rotational angle of the component, a different contact line is produced. If we now add the pitch measurement, we can assign a rotational angle of the toothed gear for all teeth at each point on the profile measurement. And if we arrange all profile measurements along the rotational angle and pitch measurement, we obtain the curve shown in Figure 9.

What seemed before to be an uneven surface roughness now reveals itself as a regular structure on the tooth surface. The blue line is produced by the radial runout of the gearing. As we have learned the hard way through experience, however, this is not the reason behind loud gear noise. A waviness on the tooth flanks is clearly visible in Figure 9 – and as it turns out, unpleasant noise characteristics in toothed gears are caused by such waviness.

If we apply the order analysis method presented above to describe the sequencing of all profile measurements, we obtain key information to explain the noise: Slowly but surely, our third monkey is beginning to speak!

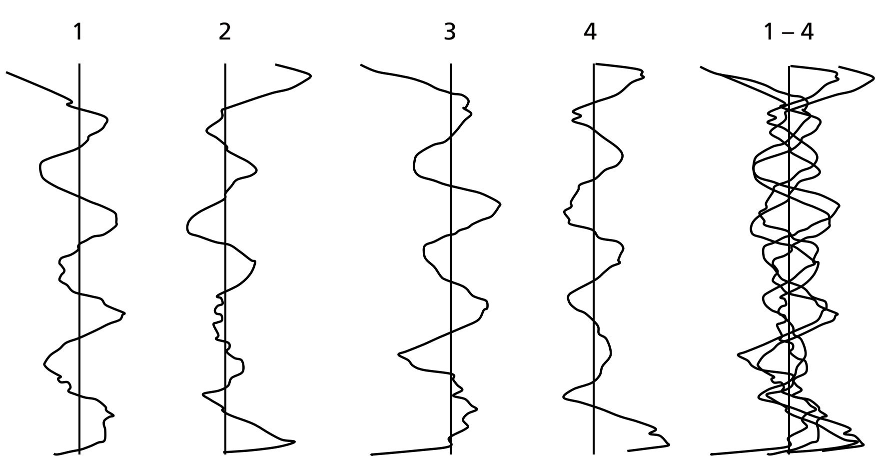
Figure 10 shows the order analysis of the sequence of profile measurements from which the macro-geometric errors have been extracted. The first order – caused by the radial runout of the gearing – is clearly identifiable. It is followed by the very small orders two and three. The fourth order in turn exhibits a higher amplitude. This is worth a closer look. Figure 10 then shows the complete sequence below the order spectrum and below it the sequence minus the first, second, and third order. An unusual quadruple strike is superposed on the waviness.

The reason behind this effect occurs long before the tooth flanks are machined: The forged piece from which the turned part for this toothed gear was manufactured originates from a rectangular cast section that was forged into a circular shape. During heat treatment, the material remembered its original rectangular shape as it were, giving rise to hardening distortions that caused this quadruple strike.

If we eliminate orders 1 to 4 from the sequence, we obtain the bottom graph in Figure 10. Here we can easily identify a conspicuous waviness with the 28th order. In fact, this was also the origin of the noise problem that caused a disturbance in the gearing running test. Because this component has 33 teeth, this is a true ghost order, whose cause we must now identify. An optimization of the gear geometry will not help to resolve the phenomenon, since this can only influence orders that are divisible by the number of teeth.

Consequently, this effect must come from faults that occur while the teeth are being machined. In many cases, the machine tool is the problem. It must also be noted that the amplitude of this wavi-

Fig. 8: Profile measurements



ness is very small; we see an amplitude of less than one micrometer. This is caused by periodic irregularities in the machine tool. For example, a spindle bearing may have lost its original perfect concentricity due to wear. Sometimes imperfectly calibrated electronic control parameters for a direct drive are key factors. If the direct drive has a number of pole pairs that corresponds to the order sought, this is usually the root of the problem.

In addition to machine-related causes, there are also other influences stemming from the machine tool's surroundings. For example, machines located on the same manufacturing floor can excite a floor slab into vibration – and this vibration can be transmitted under certain circumstances to the toothed gear processing machine. We must always keep in mind, however, that we are looking for amplitudes of less than 1 µm.

The next anomaly in Figure 10 is the 33rd order. This is not surprising, as the gear has 33 teeth. If we want to minimize this order, we must make use of gear optimization.

Measuring Principle for Bevel Gears

For cylindrical gears, the orders of a gearing can be understood through intelligent sequencing of the profile measurements. This is just as possible for bevel gears if the contact path is measured and evaluated accordingly, instead of the profile measurement.

Now on to Practical Experience: The Third Measuring Sheet

Although this new method for evaluating measuring results correlates conclusively with noise behavior, it has one critical drawback: The cylindrical gear profile or bevel gear contact path must be measured on each tooth. This time-consuming procedure is difficult to justify for the application of waviness analysis in day-to-day operations.

For quality control during regular production, adequate results can be obtained without performing the measurement on each tooth. The evaluation is then simply based on this measured tooth. This of course does not allow us to find the causes of the noise excitation, but it does allow us to determine whether the toothed gear will be noisy or not.

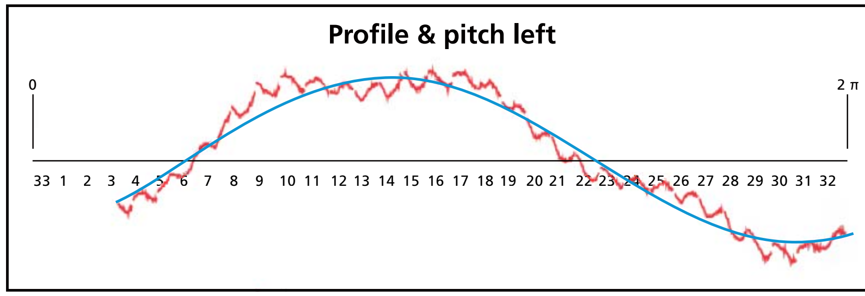


Fig. 9: Sequencing of profile measurements taking pitch into account

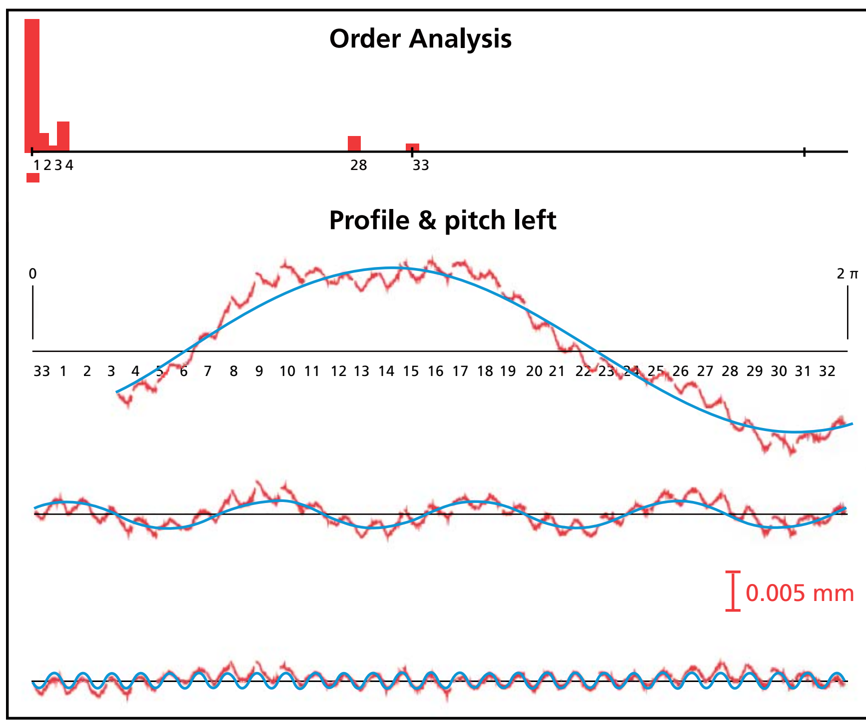


Fig. 10: Order analysis of sequencing

New method for evaluating measuring results correlates conclusively with noise behavior.

In the so-called third measuring sheet (see Figure 11), the order analysis for the gear measurement is displayed for each individual tooth. Despite the limitations of this highly simplified measurement, this method is widely used in the field. Because it does not cost any additional measuring time yet still provides key information from the waviness analysis, this method is used by some large-scale manufacturers of toothed gears in daily operations.

The third measuring sheet shows the profile measurement displayed in red in Figure 11 without macro-geometric errors – and for every profile measurement, the order analysis is shown next to it as a blue bar. It must be noted that the orders shown here do not allow any conclusions to be drawn regarding the cause of the waviness. Rather, they indicate the existence of waviness. The user can then choose to superpose a tolerance band on the curve of orders. If one of the orders exceeds the tolerance band, the corresponding bar is highlighted in red.

Summary

Like the analogy to the monkeys, crucial capabilities for evaluating noise generation in gearings are often lacking in gear measurements and running tests:

Toothed gear measurements are extremely well-suited for the geometric inspection of gearings. The measured deviations make it possible to reliably control production within narrow tolerances. But conventional toothed gear measurements are ill-suited for predicting noise characteristics – a monkey who sees but hears no evil …

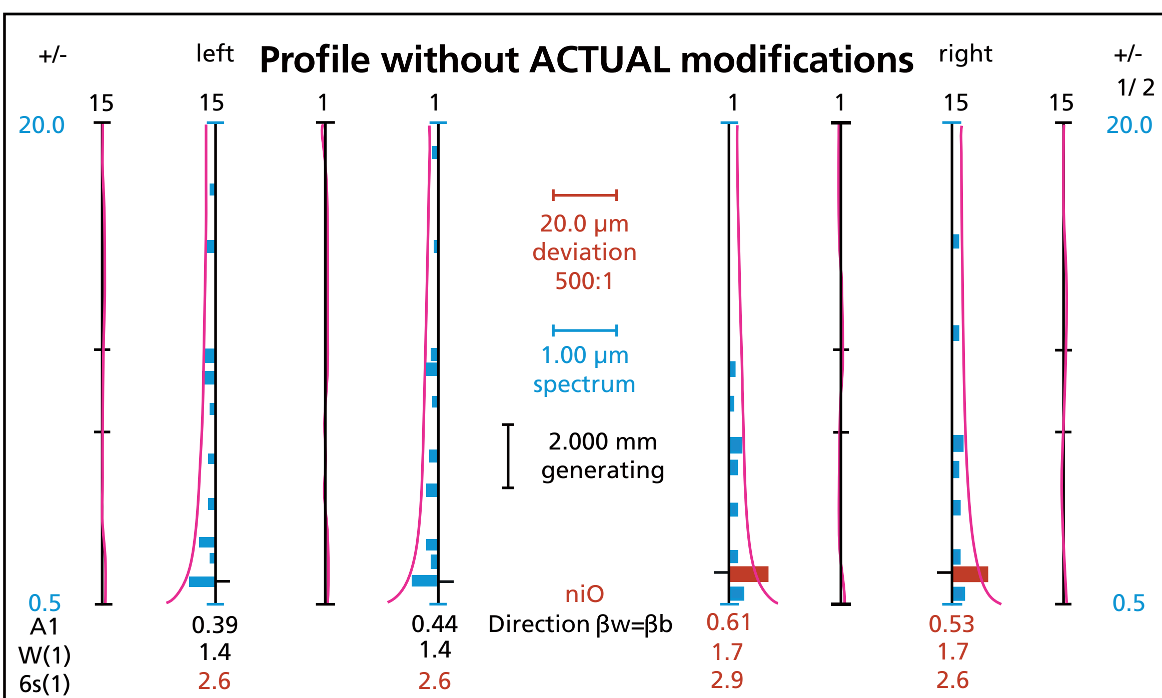


Fig. 11: Third measuring sheet

The running test is quite a different story. It is extremely well-suited for evaluating noise in a gear pair. Exactly where that noise is coming from cannot be determined, however. It isn't even possible to attribute the noise behavior or parts thereof to one of the two intermating toothed gears – a monkey who hears all but sees no evil …

How can we successfully find geometric features that cause the noise and can also be measured during toothed gear testing? There is a way to give the third monkey a voice. Fourier order analysis provides the basic mathematical framework for this. It allows us to find waviness in the sub-micrometer range and attribute these to various causes. Of course, our monkey still cannot speak accent-free "toothed gear English" but with a little effort, we can understand him quite easily ... ◆

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