EVALUATION OF TRANSMISSION ERROR

TOMAS MAREK

Faculty of Mechanical Engineering

TUL - Technical University of Liberec, Czech Republic

DOI: 10.17973/MMSJ.2022_10_2022124

e-mail: tomas.marek3@skoda-auto.cz

tomas.marek01@gmail.com

This paper addresses the issue of transmission error (TE) evaluation. The motivation for writing this paper is the research on TE measurement and calculation, which focuses on finding a correlation between TE and the noise performance of spur gears. In the context of the study a research of an accurate description of the procedure for evaluating the TE waveform was carried out, but only incomplete descriptions were found. Therefore, the author tries to describe this problema according to the available sources and his own experience.

The beginning of this article includes a description of what transmission error is and how it can be determined. The main part of this article, however, concerns developing a method for evaluating transmission error and introducing some new ideas on how to evaluate this magnitude more accurately.

KEYWORDS

Transmission error, gear meshing, incremental rotary encoder, phase demodulation, closed loop test stand

1 INTRODUCTION

Competition and rivalry in the automotive industry are becoming more intense over time. Quality and comfort are often the deciding factors in successful or unsuccessful sales. With the development of cars, for example, those with silent electric drive, the noise of gears, i.e., the meshing of gears, will increasingly rise above the level of other noises.

Acoustic problems caused by gearing have been solved mainly by precision manufacturing. However, today's advanced manufacturing, computing, and measuring technology brings a possibility of more detailed research of the gear meshing parameters. This research can have a major positive influence on acoustic performance. One of the meshing parameters (see chapter 2) is the transmission error.

There are, however, relatively few sources that address the issue of transmission error and its evaluation. One of the reasons might be, that the transmission error is in the order of micrometers and fractions of micrometers. New technologies of computing and measurement systems offer a possibility of more accurate simulation and measuring of transmission error.

The study aims to describe the evaluation of transmission error and to introduce some new ideas on how to evaluate it more accurately.

2 GEAR MESHING PARAMETERS

The main source of noise and vibration in the gearbox is the gearing. Vibration and noise from the gearing are mainly influenced by so-colled *gear meshing parameters*:

- Gear contact ratio and meshing stiffness variation
- Transmission error
- Gradient of change in the applied engagement force
- Edge tooth-bearing

3 TRANSMISSION ERROR

Transmission error (TE) is the difference between the actual and expected rotation of the driven wheel (see Fig. 1).

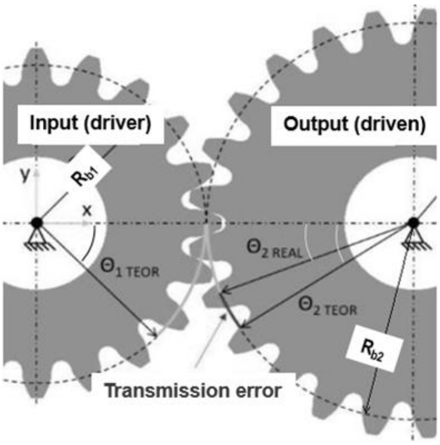


Figure 1. Transmission error

Θ_{1TEOR}, Θ_{2TEOR} … theoretical rotation angle

R_{b1}, R_{b2} … base circle radius

According to Fig. 1, TE can be calculated using the equation:

$$TE = \Theta_{1TEOR} R_{b1} - \Theta_{2TEOR} R_{b2} \tag{1}$$

The transmission error is the largest deviation on one tooth pitch, evaluated as a peak-to-peak (P2P) value.

The total transmission error is the largest deviation per revolution (see Fig. 2).

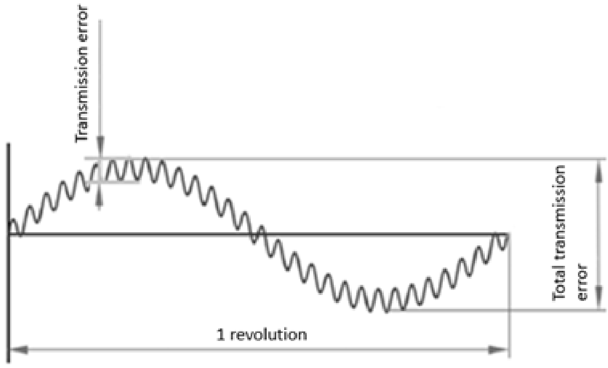


Figure 2. Transmission error on a tooth pitch and a total transmission error per revolution

Transmission error (TE) is the difference between the actual and expected amount of rotation of the driven wheel. Most scientific studies deal with transmission errors on tooth pitch, since the noise from the gearing is manifested especially on the tooth pitch (and its harmonic frequencies, i.e., multiples) [Trochta 2019; Åkerblom 2008, Tuma 2014].

There is considerable inconsistency in the literature regarding the way TE is expressed. Some authors prefer to express the difference in wheel rotation in radians, while others prefer to express the angular difference in degrees [Chung, Steyer, Abe, Clapper, Shah 1999; Tharmakulasingam 2009]. In [Åkerblom 2008], TE was specified as the second derivative in time with the unit [*rad/sec²*]. The current trend is to convert TE to a linear displacement in [*μm*] or [*in*] on the base circle.

A gearing with ideally accurately manufactured rigid teeth on ideally rigid wheels on perfectly parallel ideally rigid shafts, housed in ideally rigid bearings and ideally rigid housings with a perfectly uniform drive, has a zero transmission error. The driven wheel, regardless of load, will rotate at the same angular speed as the wheel which drives it. However, once we establish a realistic stiffness even with ideally manufactured teeth, the angular velocity of the driven wheel will begin to differ slightly from the driving wheel. It starts to oscillate around the ideal.

Causes of transmission error during rotation and at the tooth pitch can be both manufacturing (geometric deviations, surface quality) and operational (deformation of parts, etc.). Both gears are involved in the error as they are meshing together [Hiroaki, Nader 2012].

4 MEASURING AND COMPUTING OF TRANSMISSION ERROR

The measuring of TE, is based on sensing the rotation behavior of the meshing gearings. It is in principle possible in three ways [Trochta 2019]:

- tangentially mounted accelerometers or torsional accelerometers
- laser vibration sensors based on the Doppler effect
- incremental rotatry encoder

The most common way of measuring TE is the last mentioned, using incremental rotary encoders. TE is measured under load and during rotation. It can be measured on a closed loop test stand (the methodology of this TE measurement is discussed in the literature [Tuma 2014; Tuma 2003]) or on a open loop test stand. Each method has its advantages and disadvantages, and none is primarily preferred.

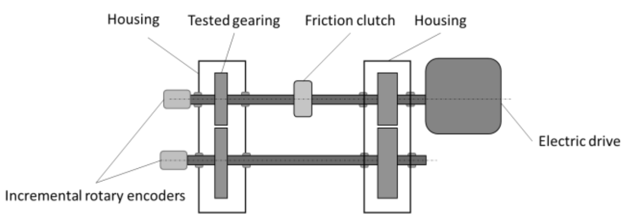


Figure 3. Scheme of the closed loop test stand

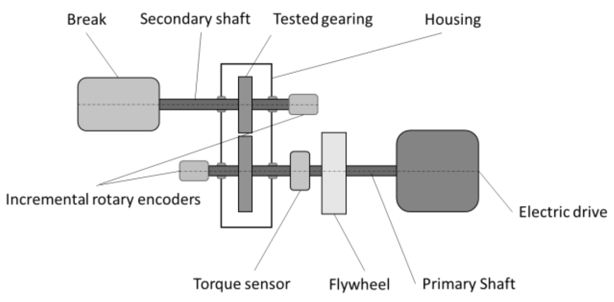


Figure 4. Scheme of the open loop test stand

For the computation of the TE, it is best to use an MBS Program (*Multi-Body System*), such as Adams. MBS provides the possibility of simulating systems of bodies in which there is a mutual force interaction. These interactions are solved using dynamic equilibrium equations. The individual bodies have a finite element interpretation.

The basic difference between FEM (*Finite Element Method)* and MBS, is that FEM allows the calculation of nonlinear bodies. Therefore, it allows plastic deformation, hyperelasticity, destruction, etc. In contrast, MBS requires that the bodies are linear and elastic. In simulations of common engineering applications, this simplification is sufficient and advantageous. Computing systems of bodies in a FEM interpretation would be very time-consuming and it is currently no longer used for such applications.

5 PROBLEMATIC OF EVALUATION THE TE-MAGNITUDE

In the calculation and measurement, we initially obtain the waveform of TE. The TE waveform is not ideally sinusoidal. On the opposite, it can fluctuate quite a lot, as the TE can be slightly different on each tooth pitch. The fluctuations may be due to the variation of the individual meshing teeth, dynamic phenomena, etc. (Fig. 5). In addition, the TE is modulated on another carrier curve (of considerably lower frequency than the tooth meshing frequency), which may be for example due to the non-uniformity of the drive or braking over a considerably longer period than the time required to cross one tooth gap, or to radial tooth throw, for example. Thorough system validation, high measurement accuracy and correct calculation settings are essential to distinguish between measurement/calculation error and transmission error.

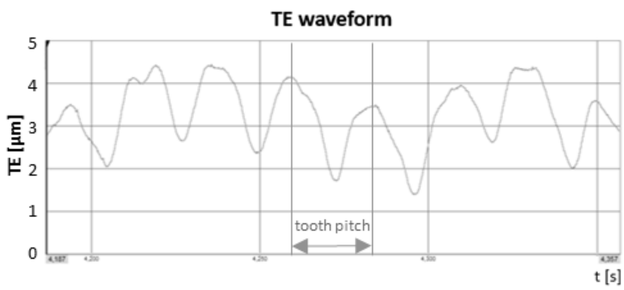


Figure 5. Example of a measured TE waveform

It is necessary to choose a way to get one number, one value, from this TE waveform, which represents the examined gearing, so that different gearings compare with each other according to this value.

5.1 *Current status of evaluation of the transmission error*

Normally, the evaluation of the waveform of TE using FFT (Fast Fourier Transformation) appears in the literature. Perhaps the most detailed evaluation description is in the literature [Tuma 2014]. This procedure is divides the experimentally measured TE waveform into individual revolutions from the first to the n-th revolution of the drive wheel. However, this number of revolutions is here not defined in any way. From this n-waveform, an average waveform is constructed, which is then converted into the frequency domain using an FFT. TE is on the tooth pitch. Thus, we find the amplitude at the tooth meshing frequency (f_z), which we calculate using the formula below. For the magnitude of TE, we multiply this amplitude by two, since TE is a peak-to-peak value (total amplitude).

$$f_z = (n \cdot z_1)/60 \qquad (2)$$

f_z.................. tooth meshing frequency .... [Hz]

n.................. revolutions ....[1/min]

z_1 .... a number of teeth of the driver wheel .... [-]

The above procedure is not ideal, because the TE value after the FFT representing the magnitude of the amplitude of the sinusoidal waveform on the 1st harmonic is already based on the averaged waveform. Thus, the result obtained are distorted.

6 NEW RECOMMENDATIONS FOR TRANSMISSION ERROR EVALUATION

The most correct would be to evaluate each tooth pitch separately by reading the P2P value and creating an average TE P2P. It is not feasible in a reasonable evaluation time, so the entire time signal is evaluated using an FFT. The value at the 1st harmonic tooth meshing frequency is the amplitude of the sinusoidal waveform replacing the original signal. Twice this value replaces the original P2P value, and we declare it to be $\mathbf{TE_{FFT\_H1}}$ (i.e., the transmission error obtained by the FFT at the 1st harmonic tooth meshing frequency). It is essential to observe, that the length of the time record should be just enough to allow the first tooth of the driving gear to roll off sequentially with all the teeth of the driven gear - let us call it one cycle. The value of $\mathbf{TE_{FFT\_H1}}$ is then not so distorted. This solution also has its limitations, e.g., the length of the time record coming from the number of revolutions for completing a full cycle – i.e. each tooth of the driving wheel meshing with every tooth of the driven wheel. Let's touch on some of the challenges arising from this evaluation method.

In Fig. 6, we see a cut-out TE waveforms of two gearings with the same basic geometrical parameters. From these waveforms, we perform an FFT and find the value of $\mathbf{TE_{FFT\_H1}}$ at the 1st harmonic tooth meshing frequency for each of the gearings (Fig. 7).

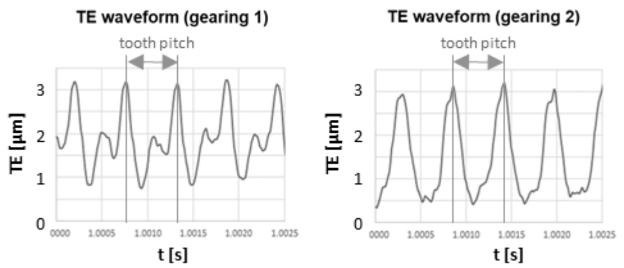
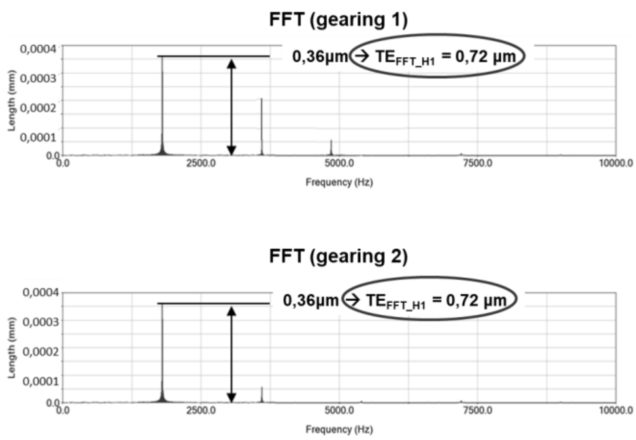


Figure 6. Transmission error waveform of two different gearings



Figures 7. FFT of TE waveform of two different gearings

Let's illustrate with a concrete example. It can theoretically happen that two different gears having different waveforms of transmission error can have equally large values of $\mathbf{TE_{FFT\_H1}}$ (0.72µm in this case).

It leads us to the new idea that the value $\mathbf{TE_{FFT\_H1}}$ alone may not be sufficient to describe the gear. Gear 2 has a fairly smooth TE waveform within a pitch, while gear 1 oscillates within one tooth pitch. In vibration and acoustic performance, this oscillation could play a role. In other words, the speed of change of the TE could also play a role. It tells us to investigate not only the magnitude of the TE, but also the shape of the waveform.

It is, therefore, certain that the description of the gearing by a single parameter is inadequate. To capture the curve shape factor, one can, for example, evaluate other harmonics in the FFT, or one has to look for another parameter describing the curve shape.

The TE waveform can be of any shape. One way to reflect the different shapes of the TE time waveforms is to evaluate even more harmonic frequencies in the FFT.

But to easily compare the waveforms with each other, it is better to look for a parameter representing the shape of the TE curve. As mentioned above, the speed of change of TE could affect the acoustic manifestation and vibration. Thus, the TE waveform within one cycle is twice deviates. We get the TE acceleration waveform. On this curve, we identify the largest value of acceleration and the largest value of deceleration of TE at the tooth pitch (example shown in Fig. 8) and call them $\mathbf{+a_{TE}}$ and $\mathbf{-a_{TE}}$. They are parameters representing the shape of the TE curve.

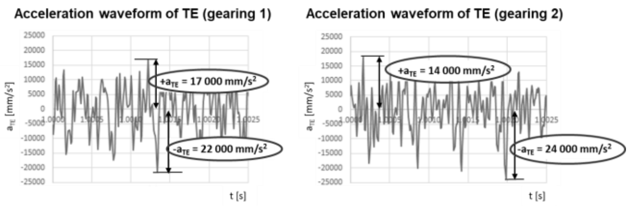


Figure 8. TE acceleration waveforms and identification of maximum TE acceleration on tooth pitch

7 CONCLUSIONS

The study deals with the evaluation of one of the tooth meshing parameters. This so-called transmission error, which could have a significant impact on the noise emission of gears. There are relatively few studies that address the TE-evaluation. This article briefly describes what TE is and how can one obtain it. The main section first describes the evaluation of the TE waveform, which is also in the science literature. Then it gives new suggestions on the evaluation method.

The main idea of the new concept of TE waveform evaluation has two parts.

- **TE magnitude evaluation:** this section contains the definition of the interval from which the FFT evaluation must be performed, the definition of the parameter $\mathbf{TE_{FFT\_H1}}$, and also describes some challenges associated with this evaluation method.
- **TE shape evaluation:** a completely new parameter is introduced to describe the shape of the TE waveform. It is the maximum acceleration $\mathbf{+a_{TE}}$ and deceleration $\mathbf{-a_{TE}}$ of the transmission error. It is a matter of further investigation whether + and - should be evaluated separately, or whether only the maximum value of the TE acceleration can be determined for a given gear, regardless of its sign. It is not yet certain whether the energy flow from wheel 1 to wheel 2 (+ acceleration) has the same acoustic effect as the energy flow from wheel 2 to wheel 1 (- acceleration). Therefore, for the time being, we will continue to look at each direction of acceleration separately.

REFERENCES

[Åkerblom 2008] Åkerblom M., Gearbox noise correlation with transmission error and influence of bearing preload, Disertation, KTH Royal institute of Technology, ISSN 1400-1179 Stockholm 2008

[Chung, Steyer, Abe, Clapper, Shah 1999] Chung C. H., Steyer G., Abe T., Clapper M. and Shah C., Gear Noise Reduction Through Transmission Error Control and Gear Blank Dynamic Tuning, presented at the SAE Sound & Vibration Conference, Traverse City, SAE paper no. 1999-01-1766, May 1999

[Hiroaki, Nader 2012] Hiroaki E., Nader S., Gearbox simulation models with gear and bearing faults. in: mechanical engineering, 2012, InTech, DOI: 10.5772/37687. ISBN

[Tharmakulasingam 2009] Tharmakulasingam R., Transmission Error in Spur Gears: Static and Dynamic Finite-Element Modeling and Design Optimization, Brunel University 2009

[Trochta 2019] Trochta, M. Transmission Error of Involute Gearing, its Measuring and Relation to Noise and Vibration of Gearboxes, (in Czech: Chyba prevodu evolventniho ozubeni, jeji mereni a jeji vztah k hluku a vibracim prevodovek), Disertation, VSB Ostrava 2019

[Tuma 2003] Tuma J., Gearbox Transmission Error Measurements Based on Phase Demodulation of Encoder, MECCA 2003

[Tuma 2014] Tuma J., Vehicle gearbox noise and vibration: measurement, signal analysis, signal processing and noise reduction measures, Chichster 2014, West Sussex: Wiley. ISBN 978-1-118-35941-9

CONTACTS:

Ing. Tomas Marek
Faculty of Mechanical Engineering
TUL – Technical University of Liberec
Department of Vehicles and Engines
Studentska 2, Liberec, 461 17, Czech Republic
Email: tomas.marek3@skoda-auto.cz; tomas.marek01@gmail.com