Meshing Stiffness—A Parameter Affecting the Emission of Gearboxes

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Abstract: Emission is one of the key problems in the automotive industry, which engineers try to eliminate by lowering emissions to the minimum. Transmission emission plays an important part here. The basic characteristics of gears include their shape, load capacity, and emissions. The most significant source of noise and vibration in the gearbox is the step change in the meshing stiffness of the gearing, which depends on the path of meshing at the entry and exit of the meshed teeth. Ensuring a permanent multi-pair mesh is a way to mitigate these step changes as much as possible. This leads to the design and implementation of gears in an integer contact ratio. In addition to this, the article deals with the impact of individual parameters on the stiffness of the gearing, which is a source of noise and vibration. The meshing stiffness of the gearing was determined on the deformation basis of the gearing, as solved by the Finite Element Method.

Keywords: emissions; gearboxes; meshing stiffness; FEM

1. Introduction

The negative impact of traffic noise (and protection against it) is becoming more serious worldwide. Traffic noise is a disturbing phenomenon. A direct connection between the total helplessness of the inhabitants and the noise conditions of their dwellings has been statistically proven. Noise emissions from a car affect not only the surrounding environment, but also the people inside the car [1]. According to the law of the National Council of the Slovak Republic, the permitted noise level of cars is 74 dB. Designers must ensure that, while maintaining the car’s increased performance with reduced emissions and fuel consumption, the noise level falls below the given limit [2]. At the same time, it is necessary to abide by the noise limits according to the EEC (European Economic Commission) regulation no. 51, which focuses on the homologation conditions for granting approval for external noise.

Increased legislative pressure puts the ecological aspect of automobile noise at the forefront. This leads to the requirement of identifying and quantifying noise sources. A significant share of automobile noise is created by the vehicles’ transmissions. Designers are trying to eliminate and reduce noise to a minimum, as the lower the noise, the more pleasant the car ride is. In some positive cases, such an effort also results in an increase in engine power and a reduction in fuel consumption, but the most important factor is the reduction in noise emissions.

The gearbox in automobiles is used to transform the power generated from the engine. It can be regarded as a self-excited vibration source [3], which then affects the performance of the entire vehicle. Generally, the gearbox consists of parts like gears, shafts, bearings, synchronizers and housings, which help the gears play a significant part in the process of transmitting power. Gear transmission
error is the main source of noise and vibration in gear transmissions, which has been proven by several researchers [4–6]. Transmission error is defined as the deviation between the theoretical position and the actual position of the output shaft, given the gear drive is perfectly accurate and stiff [7]. However, manufacturing errors and teeth deformations make transmission errors in gear transmissions unavoidable. Current research mainly aims at devising ways to measure transmission error for single-gear drives [8,9]. With the ongoing development of the Finite Element Method and the Boundary Element Method, the technology of vibration and noise prediction of gear transmission has been rapidly improving. Fang Yuan and Zhang Qiang based their research on these design parameters using CAE (Computer-Aided Engineering) software to predict the vibration and noise of the transmission [10]. With the aid of Finite Element Analysis, A. Kumar and H. Jaiswal came to the conclusion that the transmission’s mechanical properties are directly related to the natural frequency and vibration [11]. Then, they continued their research work on optimizing the number of connecting bolts using Response Surface Methodology and Finite Element Analysis. RSM was used for parametric optimization, and FEA was used for calculating the modal frequency and mode shapes [12].

In general, the gearbox is an acoustically closed system from which noise is spread mainly by vibrations of the housing surface or connected units, including the basic structure. The most important cause of noise is considered to be transmission error, which is related to meshing stiffness [13] and gear kinematic accuracy. These studies have addressed the issue of meshing stiffness [14–16]. This article deals with the impact of individual parameters on the stiffness of gearing as a source of noise and vibration in gearing. The meshing stiffness of the gearing was determined on the deformation basis of the gearing, as solved by the Finite Element Method.

2. Materials and Methods

The most commonly used transmissions in the automotive industry include helical gears and spur gears. The influence of parameters on the stiffness of gearing was analyzed in order to reduce noise in gearboxes with straight teeth and helical teeth.

2.1. Automobile Transmission Noise Sources

The gearbox is an acoustically closed system from which noise is propagated mainly through vibrations of the housing surface or connected units, including the basic structure. The most important cause of noise is considered to be transmission error, which is related to the kinematic accuracy of the gear and gear stiffness.

Vibrations from the gears transmitted to the gearbox housing are the most significant sources of noise [17]. From a physical point of view, the cause of vibrations is a dynamic force that can change its amplitude, direction, or field of action. In the case of involute gearing, the most significant is the amplitude change. This change can cause variable stiffness of the gearing and impact the entry of the gear mesh due to deformations, deviations of the pitch, and profile of the teeth from the theoretical ones.

Vibrations in the meshing of the gears are influenced by many other factors, including vibrations transmitted to the gearing from the drive or driven unit, and oscillations of the shafts and bearings. All these phenomena are involved in increasing the amplitude in the gearing. The total energy of the radiated noise continues to increase.

A specific source of noise is the occurrence of shocks due to the axial and lateral (tooth) clearance of helical gears. This occurs mainly with lightly loaded gears (for example, at the idle speed of an internal combustion engine) or, conversely, with very loaded gears at low speed. Irregular operation of the drive unit contributes to this and generates torsional vibration (change in angular acceleration during one revolution). This type of noise is referred to as rattling and knocking (ringing, rattling, etc.). The engines of modern cars therefore have a dual-mass flywheel and clutches with shock absorbers.

Other phenomena causing noise during gear meshing include air pocketing (related to air pockets in the lubricant) and lubricant entrainment (due to small clearances, excess lubricant is not pushed out of the mesh and stresses the gearing by additional dynamic forces that cause an increase in vibrations).
The gearbox contains components that can be a source of noise and vibration themselves, or they can excite or transmit and amplify vibrations. These include, for example, auxiliary elements such as steering elements, bearings, and shafts. The most significant of these elements are bearings, which are the second-most important source of noise after gearing. Vibrations are caused by the rolling elements of the bearing rolling along the inner and outer paths [18]. Their frequency is given by surface irregularities (pitting) or irregularities of functional surfaces, which are caused by wear or faulty production (deformation during clamping).

However, the dominant share of noise in gearboxes is almost always due to the excitation of vibrations in the gear mesh. Figure 1 shows an example of the overall evaluation of the noise of an automobile transmission, where the share of the separated gear mesh noise (labeled as N and 3) is in the region of a maximum of 40% of the transmission noise, which contributes 53% to the total noise. The remaining 47% is background noise (labeled Bgr), which can mainly include bearing noise.

![Figure 1. Example of car transmission noise evaluation.](image)

2.2. Definition of Meshing Stiffness

Deformations of teeth are generally quantified by tooth stiffness, which is defined as the ratio of load to deformation. We define tooth stiffness as the force per width unit that is required to deform by 1 μm. During the meshing cycle, the stiffness of the teeth changes, mainly due to a change in the bending arm.

For spur gears, the contact ratio CR is in the range 1 < CR < 2. In the two-pair engagement section, two pairs of teeth, I and II, engage at the same time, which corresponds to a parallel model of two springs (Figure 2b).

In general, the resulting stiffness \( c \) is defined:

\[
c = \frac{w}{\delta} = \sum_p c_p
\]

where \( c \) is meshing stiffness (N/mm.μm), \( p \) characterizes the order of a pair of teeth in contact, \( w \) is total width load of the gear (N/mm), and \( \delta \) is resulting deformation (μm).
where \( \xi \) is the coordinate of the path of contact and \( \delta \) is the coordinate of the path of contact.

\( \gamma \) is the coordinate of the path of contact

\( \delta \) is the coordinate of the path of contact

\( \xi \) is the coordinate of the path of contact

\( \xi_0 \) is the coordinate of the path of contact

\( p_{tb} \) is the pitch of the base circle.

Typical value of engagement stiffness \( c_\gamma = 20 \text{ N/mm} \mu \text{m} \) [19].

For spur gearing, the stiffness of the gearing was a clear function of the position of the contact point on the line of action. The meshing stiffness of the individual teeth pairs of helical gearing \( c_p \) varies not only depending on the position of the contact line in the field of view, but also along this contact line.

It is relatively easy to determine the stiffness \( c_p \) along the length of the contact line of one pair of teeth of helical gearing. This varies depending on the coordinate of the path contact \( \xi \), as shown in Figure 3a, i.e., for one pair of teeth in contact. The teeth enter the meshing at point E. To a point at a distance equal to \( \varepsilon b p_{tb} \) from point E, the contact length increases. Then, up to point A, the contact length is constant. Then the contact length is shortened. In the section of constant contact length, the
course of meshing stiffness of helical gearing is similar to that of the meshing stiffness of the spur gear. In sections with a change in contact length, the stiffness of helical gearing changes faster. The result here is, even if there are more pairs of teeth in contact (with meshing stiffness of helical gearing for one pair of teeth—$c_p$), there is a periodically recurring step change in the meshing stiffness of helical gearing—$c$ (Figure 3b).

![Figure 3](image_url)

**Figure 3.** The course of meshing stiffness of helical gearing. (a) Single-pair contact (meshing stiffness $c_p$); (b) resulting meshing stiffness.

The meshing stiffness of the helical gearing is thus given by the stiffness of the individual pairs of teeth in contact (Figure 3b). The course of the meshing stiffness of the helical gearing $c$ ($\xi$) depends on the contact ratio of the profile $\varepsilon_\alpha$ and contact ratio from the step $\varepsilon_\beta$. The helix angle is $\beta_b$ and $b$ is face width.

2.3. Determination of Stiffness

Due to the complex shape of the teeth, the theoretical determination of deformation and stiffness is difficult. Many theses are devoted to this issue, some dealing with the influence of tooth processing on the meshing stiffness. Calculating the stiffness of a spur gear with straight teeth by the method of layers (beams placed on a flexible base) was proposed in the literature [20]. Nevertheless, the question of determining the exact stiffness and deformation of the teeth still remains unsolved at the required level; therefore, experimentally determined values are usually used.

Previous experimental procedures dealing with evaluating the stiffness of gears are based on the static measurement of the deformation of a gear loaded at a constant force or by seismic measurement of offsets at slow rotation. Other experimental procedures were based on the measurement of optical phenomena. The method of planar photoelasticimetry in particular had wide practical use [21–24].

Recently, with the ever-evolving computational technology that performs extensive calculations, it is easier to find modern numerical methods for determining the meshing stiffness of a wide range of gears in the available literature. These methods include the Finite Element Method, which is one of the numerical methods of mathematics widely used to solve problems of elasticity and strength, dynamics of flexible bodies, heat transfer, and many other engineering problems [25–27].

Stiffness results are based on gear deformation. Gear deformation was determined using the Finite Element Method. To verify the results of deformation and gear stiffness solved by the Finite Element Method, tooth deformation was experimentally determined by static measurement of tooth deformation at the point of load with a constant force (Figure 4).
3. Results and Discussion

The stiffness (as well as the deformation) of the gearing is not constant for all teeth of the examined gears. It depends on the shape of the teeth, i.e., on the basic parameters of the examined gearing, such as number of teeth, gear modulus, angle of engagement, gear width, and correction and modification of gearing. It also depends on the load, shape, and construction of the gear wheel body itself.

3.1. Influence of Number of Teeth on Meshing Stiffness

The influence of the number of teeth was analyzed for spur gears with the number of teeth $z_i = 17, 19, 21, 27, 35, \text{ and } 61$, for the module $m = 1 \text{ mm}$, where the width of the gears was $b = 20 \text{ mm}$ and load force was $F_{tb} = 1000 \text{ N}$. Figure 5 shows the course of the teeth stiffness depending on the number of teeth for one pair of teeth in contact. As the number of teeth on the gear increased, the stiffness of one pair of teeth in contact increased as well.

![Figure 5. Dependence of single-pair stiffness of teeth on the number of teeth.](image)
Figure 6 shows the course of the meshing stiffness of the spur gearing. It can be stated that with the increasing number of teeth, the meshing stiffness increases. This is the result of the shape of the lateral curve of the tooth. The higher the number of teeth, the larger the radius of curvature on the active tooth side.

3.2. Influence of Contact Ratio on Meshing Stiffness

The meshing stiffness also depends on the contact ratio of the gearing. The effect of this contact ratio $CR$ on the meshing stiffness of spur gearing is shown in Figure 7, where $c$ is the meshing stiffness of the spur gearing, $c'$ is the maximum stiffness at the contact of one pair of teeth, $\xi$ is the coordinate of the contact path, $CR$ is contact ratio, and $p_{bt}$ is pitch of the base circle. It can be said that as the contact ratio of spur gearing increases, so does the meshing stiffness of this gearing.

In helical gears, the value of two factors, namely the contact ratio of the profile (in the front plane) $\varepsilon_{\alpha}$ and the contact ratio from the step $\varepsilon_{\beta}$, has an effect on the reduction in noise emissions and vibration (Figure 8).
We assume that this is due to the smoothing of the meshing stiffness curve. Figure 9 shows the course of the meshing stiffness for the helical gear set, examined for specific gear parameters. The number of gear teeth was the same \( z_1 = z_2 = 46 \), the modulus in the normal plane \( m_n = 2.5 \) mm, the pressure angle \( \alpha_n = 20^\circ \), the contact ratio of the profile \( \varepsilon_\alpha = 1.54 \), and the contact ratio from step \( \varepsilon_\beta \) varied. The change in the meshing stiffness curve for integer contact ratio values was smaller during teeth contact.

3.3. Influence of Gear Body Design on Meshing Stiffness

The development of modern machines and their means of production is characterized by constantly increasing performance parameters with decreasing equipment weight. This is also prevalent in the design of the body shape of larger gears. Lightening the gear body has an effect on the deformation of teeth in contact and meshing stiffness.

The influence of rim thickness \( v \) and web thickness \( t \) on the meshing stiffness (Figure 10) on spur gearing was investigated on spur gear set with number of teeth \( z_1 = z_2 = 32 \), where the modulus in the normal plane \( m_n = 5 \) mm, mesh angle was \( \alpha_n = 20^\circ \), and face width was \( b = 15 \) mm. As the thickness of the rim increased, so too did the meshing stiffness; furthermore, as the thickness of the web varied.

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![Figure 10](image1.png)

**Figure 10.** Influence of rim thickness $v$ and web thickness $t$ on meshing stiffness.

![Figure 11](image2.png)

**Figure 11.** Influence of rim thickness on meshing stiffness.

The stiffness of teeth was not constant, even over the face width of the toothing (Figure 12). If the face width was the same for both the driving and the driven gear wheel, the stiffness of the teeth was smaller at these edges due to the free end of the tooth lacking a supporting effect (Figure 12a). If the face width was not the same for the driving and driven gears, on a wheel with a larger face width where the line of contact does not end at the edge of the face width (Figure 12c), there will be a fairly sharp increase in teeth stiffness at the edges of the contact due to the support effect. This is due to the low deformation of the tooth at the edges of the face width. The location of the web will affect the course of the stiffness of the tooth along the face width. At the location of the web, the stiffness of the tooth increased slightly locally (Figure 12b).
Figure 12. Stiffness of teeth—course along the face width of the toothing. (a) The face width was the same for both the driving and the driven gear wheel; (b) at the location of the web, the stiffness of the tooth increased slightly locally; (c) the face width was not the same for the driving and driven gears, stiffness of tooth on a wheel with a larger face.

4. Conclusions

There are many factors that cause noise and vibration in gearboxes, which must be taken into account during design, manufacture, installation, and service. Detailed analyses of the gearbox manufacturers have shown that the noise level of the gear unit is reduced only by a very small degree by improving the accuracy of the gears. Changes in the shape of the tooth, its geometry, and changes in production technology can achieve a significant reduction in noise in the transmission mechanism.
A significant source of noise and vibration in the gearbox is the step change in meshing stiffness during gear contact. Based on the analysis of the influence of individual parameters on the meshing stiffness of the gearing, it can be concluded that the course of the meshing stiffness in gearing increases with an increasing number of teeth. It is affected by the shape of the lateral curve of the tooth. The higher the number of teeth, the larger the radius of curvature on the active tooth side, and thus the lesser the bending stress on the tooth. The meshing stiffness also depends on the contact ratio of the gearing. The change in the meshing stiffness curve for integer contact ratio values is smaller during teeth contact. Such gears set with integer contact ratio values are characterized by lower noise. The design of the shape of the gear body has an effect on the change in the course of the meshing stiffness. As the thickness of the body rim increases, the value of the meshing stiffness increases. Very important is the wrack thickness value, which is equal to 3.5 times the modulus. When selecting a rim thickness greater than this value, the value of the meshing stiffness changes to a lesser extent. As the thickness of the web of the wheel body increases, the meshing stiffness increases. Stiffness of gear teeth is not constant, even along the face width. The results of meshing stiffness were determined on the basis of tooth deformation, as solved by the Finite Element Method.

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