Gear stiffness and its effect on noise emissions of automotive transmissions

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Abstract Problems with the negative impact of traffic noise and protection against it are becoming more serious worldwide. Traffic noise is not only a disturbing phenomenon, but there is a statistically proven direct connection between the total helplessness of the inhabitants and the noise conditions of their dwellings. In the automotive industry the lower the noise, the more pleasant the car ride. This is one of fact leads to the need to identify noise sources and quantify them. The periodic change in the stiffness of the gearing during engagement has a significant effect on the noise in the gears. The paper is devoted to the stiffness of gearing, its characteristics and the possibilities of its determination as a parameter influencing the noise emissions of cars.

Key words Automotive transmissions, noise emission, deformation, stiffness of gearing

1. INTRODUCTION

The development of engine plants in past focused on the acquisition of the highest capacity and durability. Engines and machines with gear transmission are very popular and draw sufficient attention. Lowering the weight of the construction machines and engine plants as well as increasing their efficiency and productivity, are all part of the compelling task the construction, technology and research workers must accomplish. These intensity factors have often a significant influence on the increment of vibrations and noise in the monitored engine plants [1]. The society becomes gradually more interested in these noise and vibration emissions produced by the gearing mechanisms. The issue of lowering noise emission in a gearbox is interconnected with the sources of noise, together with measurement and evaluation of vibrodiagnostic performance. There are many influences that cause vibrations in the gearbox and that have to be taken into account already in the phase of design, manufacture, installation and operation. Detailed analysis of gearboxes manufacturers have shown that improving of the gear accuracy can not reduce the transmission unit noise to the desired level. Only fundamental changes to the shape of the tooth and changes in production technology can achieve stronger noise reduction of gear mechanism [2]. Current products constructed with the usage of computer programs for the firmness check of suggested solutions (FEM) together with the rich experience of construction workers, reach optimal parameters from the perspective of rigidity, material utilization and longetivity.

An intense pressure is carried upon the completion of a legislative law from the aspect of making noise and vibration in the present day. Mainly the automobile industry must follow these noise limits according to the regulation EEC (European Economic Comittee) n.51 [3, 4].

Performance of vibrations in the gearboxes proved to be a significant noise source. Only a minimal decrease in the level of noise can be achieved by using some simple modifications (increment of accuracy and usage of coverings).

2. NOISE OF GEARBOX IN PASSENGER CARS

Gear box is an acustic enclosed system, from which the noise travels mainly through vibrations of the closet surface or plugged aggregates inclusive of the base construction. One of the essential causes of noise is so-called transmission error, which is related to kinematic accuracy and stiffness of the teeth [5].

The vibrations from the cogged gear, transmitted to the case of gearbox, are the most important source of noise. From physical point of view, the cause of vibrations is the dynamic force which can change its amplitude, direction or origin. In the evolvent gearing
is the most critical change of amplitude. Its main cause is alterative stiffness of teeth and shock when teeth enter the image, due to the deformation, deviation of gaps and tooth profile from the theoretical ones. Many other effects, i.e. vibrations transmitted into the gearing from the driving or powered aggregate, oscillation of the shafts and bearings, influence the vibrations in cogged wheels in a mesh. All of these elements play a role in the enlargement of amplitude in gearing. The total energy of the radiated noise further increases.

A specific source of noise comes from the creation of shock influenced by the axial and side (cogged) will-power of the cogged wheels with slanted cogging. Arises mainly in little loaded gearwheels (eg idling internal combustion engine), or vice versa when heavily loaded gear wheels too slowly. This contributes irregular running the drive unit and there is a torsional vibration when heavily loaded gearwheels (eg idling i nternal combustion engine), or vice versa or by the deformation from a grip.

imperfection of the functional planes, which are created by abrasion course. Their frequency is given by pitting as well as by essential source of noise in cogging. Vibrations are created by primary components are the bearings, which are the second most components, bearings and shafts all belong to the gearbox. The transmit and enlarge them. Added components, such as driving components, bearings and shafts all belong to the gearbox. The primary components are the bearings, which are the second most essential source of noise in cogging. Vibrations are created by generating of rolling components in a bearing on its inner and outer course. Their frequency is given by pitting as well as by imperfection of the functional planes, which are created by abrasion or by the deformation from a grip.

A dominant contribution of noise in a gearbox; however, comes from the creation of vibrations during intervention of the cogged wheels. On Fig. 1, a total evaluation of gearbox noise in an automobile, where the contribution of separated noise of the gearbox intervention (defined as N and 3) is at maximum 40 % of the gearbox noise with a contribution of 53% to the total noise, is displayed. The remaining 47% constitute for the background noise (defined as Bgr), inside which mainly the noise created by bearing is incorporated.

For a detailed analysis of the causes of vibration revealed significant impact on the gear mesh of the intensity of vibration. In [6] was expressed that dependence on uncorrected teeth of spur gear (Fig.2). Local minimums of noise occurring at integer values of coefficients and gear mesh $\varepsilon_\alpha$ a $\varepsilon_\beta$.

Based on experiments and experience of renowned manufacturers gears showed that reducing noise respectively vibration resistance increases. Significant improvement in quality gears is achieved under certain principles in coefficient $\varepsilon_\alpha$ and $\varepsilon_\beta$, which include:

- minimum vibration is achieved at any size with $\varepsilon_\alpha$, integer value $\varepsilon_\beta$,
- vibration excitation in increasing integer values of $\varepsilon_\beta$ steadily decreasing,
- for any $\varepsilon_\beta$ vibration excitation decreases with increasing value of $\varepsilon_\alpha$ values from 1 to the value of $1.7 \div 1.8$, where there is a local minimum,
- for integer values of $\varepsilon_\beta$ is always at an absolute minimum value $\varepsilon_\alpha = 2$.

The noise in gear transmissions particularly affects periodic change of stiffness teeth during meshing caused by changing the number of pairs of teeth, which are simultaneously in meshing.

3. TEETH STIFFNESS AND THEIR IMPACT FOR GEARBOX NOISE

Deformations of teeth are generally quantified by tooth stiffness, which is defined as the ratio of load to deformation. Knowledge of the deformation properties of gearing is very important. Consider first the one tooth (Fig. 3-a). Action of the resulting normal force $F$ is deformed tooth. This is shown in the figure by a thin line. The resultant deformation in the direction of action of the normal force $F_i$ (i = 1, 2 - index that distinguishes whether it is a tooth of the pinion - drive wheel or driven wheel tooth) consists of a deflection bending, shear, deformation in the area of constraint and the touch deformation.

![Fig. 2. Influence $\varepsilon_\alpha$ and $\varepsilon_\beta$ for trefoil of noise](image)

$\varepsilon_\alpha$ - coefficient profile of gear mesh, profile mesh path corresponds to the line of contact and is the ratio of the gear mesh path in front of the plane to pitch in front of a plane at the base circle.

$\varepsilon_\beta$ - coefficient step in gear mesh. It is the ratio of the face width of the teeth to the axial pitch.

![Fig. 3 a) The deformation of the tooth, b) - c) deformation of a one pair of spur gear teeth](image)
It is necessary to determine the deformation meshing teeth in fact, i.e., deformation of teeth pair that you can imagine and illustrate two ways (Fig. 3-b-c)). Figure 3 - b) shows a pair of teeth than in non-loaded aspect are contact at point \( X \) on the line of contact \( \tau \). The profiles meshing teeth are deformed after loading. The deformed teeth profiles intersect the line of contact at points \( X_1 \) and \( X_2 \). The total deformation of the pair of teeth we can then determined as the sum of the deformation of both teeth \( \delta_1 \) and \( \delta_2 \). Figure 3 - c) shows a pair of teeth than in non-loaded aspect are contact at point \( X' \) on the line of contact \( \tau \). The deformed teeth profiles intersect the line of contact at points \( X_1' \) and \( X_2' \), because one of wheel gears is was fixated (restrained). The total deformation of one pair of meshing teeth determine by equation (1).

\[
\delta = \delta_1 + \delta_2 = \left| \frac{\phi_{s1} \cdot r_{s1}}{r_{s2}} + \frac{\phi_{s2} \cdot r_{s2}}{r_{s1}} \right| = \left| \frac{\phi_1}{r_{s1}} \cdot \frac{\phi_2}{r_{s2}} \right|
\]  

(1)

Gear teeth are deformed due to the load. Deformation of teeth is usually expressed quantitatively by stiffness gearing. Periodic changes in the stiffness tooth mesh, caused by changes in the number of pairs of teeth, which are also mesh in a significant noise impact on teeth [7, 8]. One of the ways to specify the tooth stiffness is calculated using the total deformation gearing.

In general the resulting stiffness \( c \) defined by equation (2):

\[
c = \frac{w}{\delta} = \sum_{p} c_p, \quad p = I, II \quad [N/mm, \mu m]
\]  

(2)

where

- \( w \) - load across the width of the teeth [N/mm],
- \( w_I = w_{1I} + w_{II} \) - load across the width of the first pair of teeth,
- \( w_{II} \) - load across the width of the second pair of teeth,
- \( \delta \) - resulting deformation [\( \mu m \)].

The resulting stiffness the teeth equal to the sum of partial stiffness of pairs of teeth, which are mesh. The stiffness of each pair of teeth is calculated according to equation (3) to final stiffness of a pair of teeth:

\[
\frac{1}{c_p} = \frac{1}{c_1} + \frac{1}{c_2}
\]  

(3)

where

- \( c_p \) - resulting stiffness of a pair of teeth [N/mm, \( \mu m \)],
- \( c_{1/2} \) - stiffness of each tooth to which it applies

\[
c_{1/2} = \frac{w}{\delta_{1/2}} \quad [N/mm, \mu m].
\]

The stiffness of the individual tooth tooth pairs varies along the path of engagement [9, 10]. The course of stiffness as well as the corresponding deformation of one pair of straight gears is shown in Fig. 4-a). The maximum single-pair gear stiffness is denoted \( c' \) in the middle of the engagement length, and its typical value is \( c' = 14 \) to 16 N/mm, \( \mu m \) for single-pair engagement for spur gears.

To determine the resulting deformation of the teeth is necessary to determine the deformation of individual pairs. In Figure 5 shows the
progress of the overall deformation of teeth solved by FEM for the spur gears with number of teeth \( z_1,2=24 \), the module of teeth \( m=3,75\,[\text{mm}] \), the force \( F_N=1000\,[\text{N}] \) and width of gearing \( b_{1,2}=10\,[\text{mm}] \), which in the meshing reaching gear ratio 1 and for the ideal division of load. Deformation of pairs of teeth over the meshing along the length of meshing line is changes. Maximum value of deformation shall in this case the endpoints lonely meshing (if we consider the image-pair) and the minimum value shall also meshing in two pairs of endpointslonely meshing. The points B and D, it is the solitary meshing points leads to a step change deformation teeth and it will input the next couple teeth to meshing. In Figure 5 shows the course of total stiffness of the teeth, tooth pair stiffness and total stiffness of gear teeth for the spur gears, in the teeth, which in the meshing reaching gear ratio 1. The stiffness is individual pairs of teeth in the mesh by changing the length of the engaging line. The minimum value shall end in the engaging points and lines shall at maximum point lone mesh, the so-called pitch point C. The resulting stiffness teeth after track mesh changes periodically with a period equal to the basic pitch frontal. The endpoints solitary mesh leads to sudden changes in stiffness resulting teeth. This is due to a step change in deformation resulting from the entry into another pair of teeth in the mesh his cause’s vibrations that cause noise gearbox.

4. CONCLUSIONS

There are many influences that cause vibrations in the gearbox and that have to be taken into account already in the phase of design, manufacture, installation and operation. Detailed analysis of gearboxes manufacturers have shown that improving of the gear accuracy can not reduce the transmission unit noise to the desired level. Only fundamental changes to the shape of the tooth and changes in production technology can achieve stronger noise reduction of gear mechanism.

The internal dynamics of the teeth is one of the most common gearing problems. This is reflected vibrations all parts of gears, their noise and increased stress teeth. One of the factors that aggravate environmental is a noise. The noise in gear transmissions particularly affects periodic change of stiffness teeth during meshing caused by changing the number of pairs of teeth, which are simultaneously in meshing. Using FEM we can with sufficient accuracy to solve the direct deformation of teeth of spur gear. Deformation of pairs of teeth over the meshing along the length of meshing line is changes. Maximum value of deformation shall in this case the endpoints lonely meshing (if we consider the image-pair) and the minimum value shall also meshing in two pairs of endpointslonly meshing. The stiffness is individual pairs of teeth in the mesh by changing the length of the engaging line. The minimum value shall end in the engaging points and lines shall at maximum point lone mesh, the so-called pitch point C. The resulting stiffness teeth after track mesh changes periodically with a period equal to the basic pitch frontal. The endpoints solitary mesh leads to sudden changes in stiffness resulting teeth. This is due to a step change in deformation resulting from the entry into another pair of teeth in the mesh his cause’s vibrations that cause noise gearbox.

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