GRANT journal
ISSN 1805-062X, 1805-0638 (online), ETTN 072-11-00002-09-4 Vol. 9, Issue 1
EUROPEAN GRANT PROJECTS | RESULTS | RESEARCH & DEVELOPMENT | SCIENCE

Gear stiffness and its effect on noise emissions of automotive transmissions

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Grant: APVV-16-0259
Grant name: Research and development of combustion technology based on controlled homogenous charge compression ignition in order to reduce nitrogen oxide emissions of motor vehicles.
Field specialization: JR - Other machinery industry

Grant: VEGA 1/0473/17
Grant name: Research and development of technology for homogeneous charge self-ignition using compression in order to increase engine efficiency and to reduce vehicle emissions.
Field specialization: JR - Other machinery industry

Grant: KEGA 006TUKE-4/2020
Grant name: Implementation of Knowledge from Research Focused on Reduction of Motor Vehicle Emissions into the Educational Process.
Field specialization: JR - Other machinery industry

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 aquisition 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 intetested 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

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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 (change of angular accelerations during one rotation). Consequently, the engines of modern automobiles have a dual-mode flywheel and clutches with an intervention absorber. Among other elements causing noise during cogging intervention belongs so-called Air Pocketing (related with the air pockets in lubricant) a Lubricant Entrainment (due to the small will-power, the excess lubricant is not squeezed from the intervention and stresses the cogging with added dynamic forces which cause the increase in vibrations). The gearbox includes components which can be the sources of noise and vibrations themselves, or they create the vibrations as well as 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.

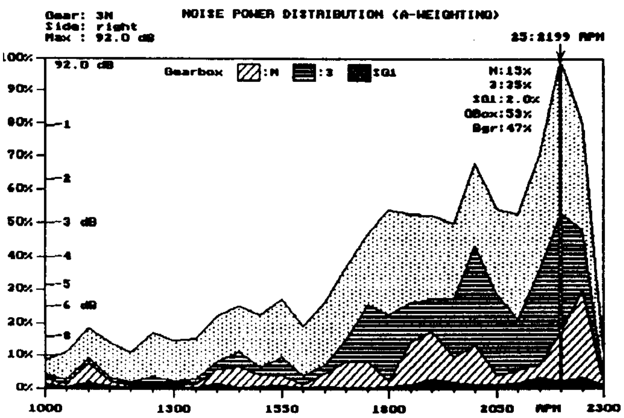


Fig. 1 Example of evaluation of noise gearbox of car [6]

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 ε_α a ε_β.

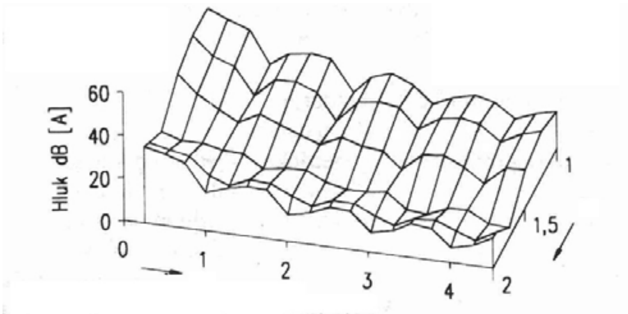


Fig.2. Influence ε_α and ε_β for trefoil of noise

ε_α - *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.
ε_β - *coefficient step in gear mesh.* It is the ratio of the face width of the teeth to the axial pitch.

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 ε_α and ε_β, which include:

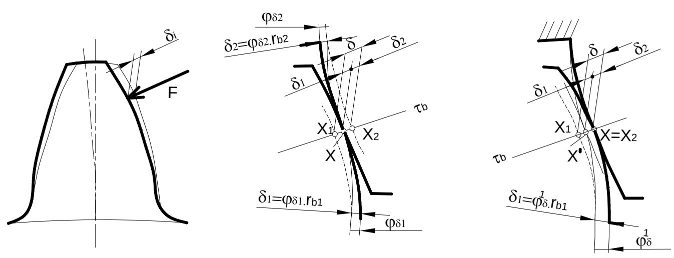
- minimum vibration is achieved at any size with ε_α integer value ε_β,
- vibration excitation in increasing integer values of ε_β steadily decreasing,
- for any ε_β vibration excitation decreases with increasing value of ε_α values from 1 to the value of 1.7 ÷ 1.8, where there is a local minimum,
- for integer values of ε_β 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 δ_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.



a) b) c)

Fig. 3 a) The deformation of the tooth, b) - c) deformation of a one pair of spur gear teeth

It is necessary to determine the deformation meshing teeth in fact, ie 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 τ_B. 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 δ_1 and δ_2. Figure 3 - c) shows a pair of teeth than in non-loaded aspect are contact at point X' on the line of contact τ_B. 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).

$$\delta = \delta_1 + \delta_2 = \left|\varphi_{\delta 1} \cdot r_{b1}\right| + \left|\varphi_{\delta 2} \cdot r_{b2}\right| = \left|\varphi^1_{\delta} \cdot r_{b1}\right| \qquad (1)$$

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):

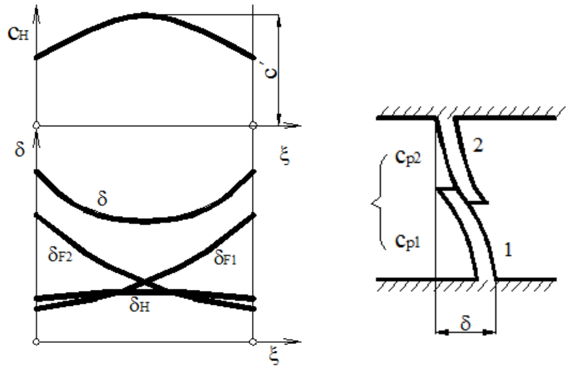
$$c = \frac{w}{\delta} = \sum_p c_p \text{ , p = I, II [N/mm.µm]} \qquad (2)$$

where
w - load across the width of the teeth [N/mm],
where $w = w_I + w_{II}$
w_I - load across the width of the first pair of teeth,
w_{II} - load across the width of the second pair of teeth,
δ - resulting deformation [µm].
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:

$$\frac{1}{c_p} = \frac{1}{c_1} + \frac{1}{c_2} \qquad (3)$$

where
c_p - resulting stiffness of a pair of teeth [N/mm.µm],
$c_{1/2}$ - stiffness of each tooth to which it applies
$c_{1/2} = w/\delta_{1/2}$ [N/mm.µm].

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.µm for single-pair engagement for spur gears.



a) b)
Fig. 4 The course of single-pair stiffness and deformation of gearing

The stiffness is individual pairs of teeth in the mesh by changing the length of the engaging line.

Since the involute tooth of spur gear has a complex shape, the theoretical determination of the deformation the tooth is a difficult. The existing experimental techniques are based on static deflection measurements gearing loaded of constant force or seismic measurement deviations at slow rotation. Recently we can meet with modern numerical methods, such as finite element method (FEM), which can serve as one of the methods for the determination of deformation gearing [11 - 13]. As the basis for calculating the stiffness of gearing results serve deformation analysis examined gearing solved by FEM.

Create a geometric model of the gear is considered the first step to deal with tooth deformation FEM. Universal user to create geometry computer model does not exist. In this case, the geometric model has been created a combined method. The final shape of 2D was by created in program AutoCAD. 3D model of examined the spur gear with straight teeth was by created in program COSMOS/M as editing from the 2D model. To determine the computer model for studying deformation of the teeth using FEM was necessary to determine the material constants, define the type of finite element, and selecting appropriate boundary conditions (geometry and power).

We will focus on the value of the total deformation in the direction of action forces. To determine the deformation of gearing under load is necessary to know the apportionment of the load on each gearing pairs with two pairs meshing. At the beginning was considered with the simplest, ideal load apportionment. The load for the two pairs meshing is divided by half for each couple of teeth in the meshing.

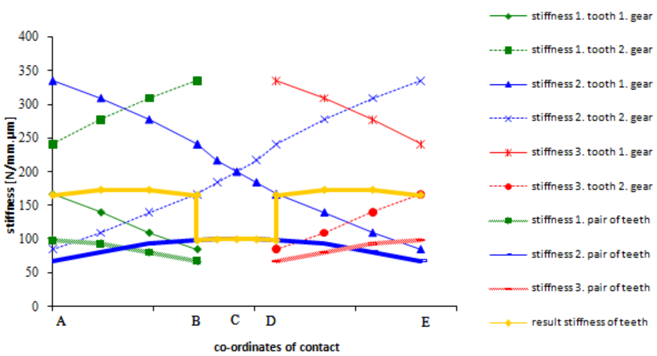


Fig. 5 Course of tooth stiffness

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 $m=3,75[mm]$, the force $F_N=1000[N]$ and width of gearing $b_{1,2=}10[mm]$, 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 endpoints lonely 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 endpoints lonely 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.

References

1. Femandez del Rincon, A. et al., A model for the study of meshing stiffness in spur gear transmissions, Mechanism and Machine Theory. 61 (2013), p.30-58.
2. Kender, Š. et al., Advantages of using composite materials in automotive manufacture process. Trans&Motauto World. - Sofia (Bulharsko) : Scientific Technical Union of Mechanical Engineering, Roč. 5, č. 1 (2020),p. 3-5.
3. Czech, P., Intelligent approach to valve clearance diagnostic in cars, Communications in Computer and Information Science, 395 (2013), p. 384-391.
4. Bocko, J., Frankovský, P., Kostelníková, A., Ostertagová, E., Ostertag, O. Structural design and photoelasticimetric verification of landing gear of ultralight aircraft, Metalurgija. 2 (2010) 145-150.
5. Wojnar, G., Detecting local defects in toothed gears, Transactions of the University of Košice, 3 (2012) p.135-138.
6. Moravec, V. et al., Čelní ozubená kola v převodovkách automobilů. VŠB-TU Ostrava, 2009, 220 p.
7. Babjak, Š. et al.: Lean creative automotive product design (LCAPD). A Jövő Járműve. Vol. 5, no. 3/4 (2012), p. 70-78.
8. Dúbravčík, M., Babjak, Š., Kender, Š., Product Design Techniques in Automotive Production. American International Journal of Contemporary Research. Vol. 2, no. 5 (2012), p. 43-54.
9. Noga, S., Markowski, T., Bogacz, R., Method of determining thenormal modes of toothed gears with complex geometry, Scientific Journal of Silesian University of Technology. Series Transport. 89 (2015) p. 119-127.
10. Baworski, A., Garbala, K., Czech, P., Witaszek,K. Estimation of the ability to use a mass of air from amoving vehicle in wind turbine propulsion. Scientific Journal of Silesian University of Technology. Series Transport. 88 (2015) 5-17.
11. Tristan M. Ericson, Robert G. Parker, Experimental measurement of the effects of torque on the dynamic behavior and system parameters of planetary gears, Mechanism and Machine Theory, 74 (2014) 370-389.
12. Czech, P., Folęga, P., Wojnar, G.: Defining the change of meshing rigidity caused by a crack in the gear tooth's foot. In: International Journal of Engineering, Science and Technology. Vol. 2, No. 1, (2010), pp. 49-56.
13. Sánchez, M. B. et al., Approximate equations for the meshing stiffness and the load sharing ratio of spur gears including hertzian effects. Mechanism and Machine Theory, Vol. 109, (2017), p.231-249.