APPLICATION OF FEM METHOD IN REDUCTION OF GEAR TRANSMISSION HOUSING VIBRATION

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Abstract

The present paper describes the studies of vibroactivity of FEM models of gear transmission housings. The FEM model of the gear transmission housing installed on a laboratory stand was modified in order to reduce its vibration level. Two different variants of the housing ribbing are presented herein. The models were subjected to load exerted by dynamic forces in bearing nodes, determined by means of a dynamic model of a gear transmission in a powertrain. The measure of vibroactivity v^2_{avg} *as defined in the publication, was used for the quantitative comparisons. The studies performed proved that the models, depending on the location of ribbing, exhibit various vibroactivity rates, usually lower than those achieved in the basic, unribbed models. That was proven by analyses carried out for various load rates applied to the transmission. Furthermore, the paper presents the results of studies of sensitivity of FEM models of gear transmission housings to excitation by dynamic forces in the meshing, depending on the precision of gears. The studies confirmed an increase in vibroactivity of toothed gear housings along with an increase of gear inaccuracies.*

Keywords: *Gear Transmission, FEM, Vibration*

1. Introduction

The importance of the reduction of noise generated by means of transport has grown in the recent years. A crucial reason for the actions undertaken, by both scientific and industrial research institutions, is to minimise the environmental impacts on human body. One of the possible ways to reduce the environmental noise is to decrease vibration level in the very source of the sound emitted. Currently, research into that area is conducted mainly with the aid of numerical models, and only the measurements and verification analyses are performed on test stands.

The noise of a gear transmission is emitted mainly by its housing. In order to reduce its level, research is made into such selection of engineering features of the housing, that the structural vibration is held as low as possible. In such a case, research using the FEM and BEM methods is conducted [1, 3-9].

Dynamic forces evoked in the meshing between gears, acting on the housing through the bearing nodes, are an important factor causing vibration of the housing. The intensity of such forces is strongly correlated with the following parameters: engineering of the toothed gears and bearings, manufacturing precision of meshing and other components of the transmission. The influence of such factors may be analysed on the basis of numerical calculations with the aid of dynamic models of gear transmission in a powertrain, which are discussed in various papers [2, 5].

The present paper presents the results of further research into the selection of engineering features of gear transmission housings with the aim to reduce their vibroactivity. The effects of load and rotational velocity of the gears, and manufacturing precision class of toothed gears on the vibroactivity of various models and their housings, were analysed.

2. Numerical model of gear transmission housing

Vibroactivity of a gear transmission was studied with the aid of FEM on a housing installed on the test stand for power circulating FZG in the Laboratory for Transmission Systems of the Transport Department of the Silesian University of Technology. Fig. 1 presents the gear tested on the stand.

Fig. 1. View of the gear tested on the FZG power circulating stand

The study was aimed at comparing the vibroactivity of the housings with and without additional ribbing (Fig. 2). Based on CAD models of the gear transmission housing, its FEM models were made. Various possibilities of placement of the additional ribbing and its influence on the reduction of vibroactivity were studied. Sample engineering solutions for the ribbing are presented in Fig. 1, where the mass of those housings is also provided.

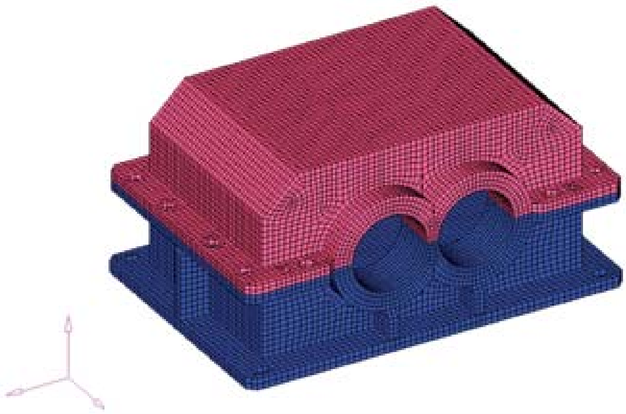
The model presented in Fig. 1b is an example of a structural solution for strengthening the lid of the upper gear, often used in practice, in which the ribbing is placed laterally and along the centreline of the toothed gears. In case of the model presented in Fig. 1c, the results of the modal analysis, made previously for the housing model without ribbing, were taken into consideration while designing location of the ribbing. The analysis allowed main resonance frequencies of the housing and shape of vibration in resonance state to be determined, and further allowed to decide such placement of ribbing that would allow significant reduction of resonance vibration. The proposed ribbing solution may contribute to the reduction of vibroactivity of the transmission housing in specified ranges of rotational speed. The method for selection of stiffing and shape of the ribbing has been presented in paper [6-8].

In the studies, reactions of forces excitating the vibration in bearing nodes were implemented, which were determined with the aid of the model of gear transmission in a powertrain. The model is being developed under other research projects on toothed gears, conducted by the Department of Transport of the Silesian University of Technology, described, among others, in paper [2].

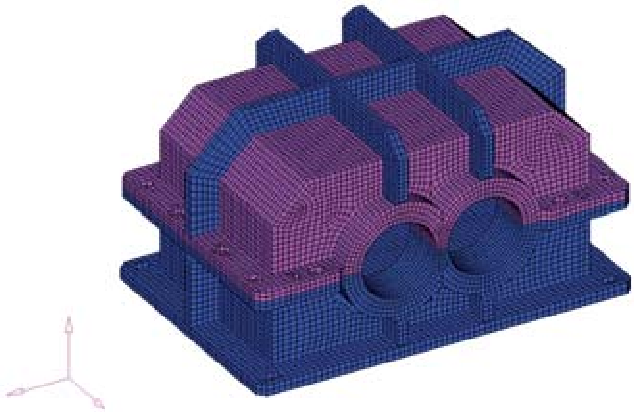
While determining the forces causing vibration in the bearing nodes in the dynamic model of a gear transmission in powertrain, the following gear parameters were assumed:

- number of pinion teeth $z_1=19$,
- number of gear teeth $z_2=30$,
- module pitch $m=3.5$ mm,
- pressure angle $\alpha_{on}=20°$,
- helix angle $\beta=15°$,
- distance between the center of two gear $a=91.5$ mm,
- face width $b=56$ mm,
- manufacturing precision class of toothed gears 6, 7 and 8.

a) housing mass 32,8 kg



b) housing mass 35,51 kg



c) housing mass 34,4 kg

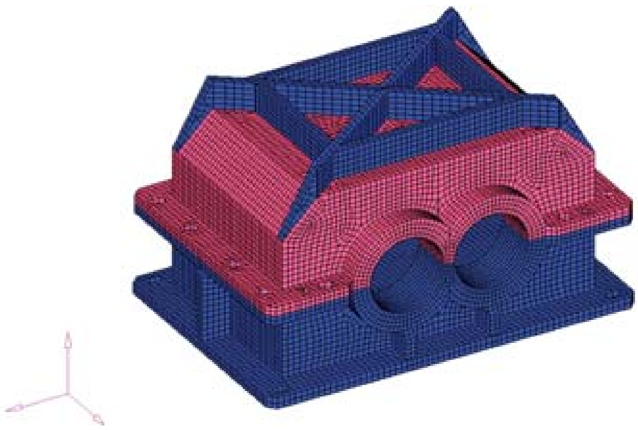


Fig. 2. FEM models of the toothed gear housing: before modification (a), the 1st ribbing variant (b), the 2nd ribbing variant (c)

The tests were performed for various rotational velocity, for the shaft of transmission gear n= 900, 1200, 1500, 1800, 2100 and 2400 rpm respectively, and two load rates applied to a pair of toothed gears in the transmission, which were Q_1 =1 MPa and Q_2 =3 MPa, respectively.

3. Research results and analysis

Based on the simulation carried out with the aid of FEM, vibration velocity runs were obtained for the transmission housing in node points of the model. While comparing the vibroactivity of various FEM models, further in this paper, the measure of vibroactivity was assumed as an average value of the square of vibration normal velocity of the node points of the upper part of the housing lid $v^2_{avg.}$ [1], according to the dependency (1):

$$v_{avg}^2 = \frac{1}{n}\sum_{i=1}^{n}\left(\frac{1}{k}\sum_{j=1}^{k}(v_{ij}(t_i))^2\right) \tag{1}$$

gdzie:
$v_{ij}(t)$ - vibration velocity at the time of t_i, of the j^{th} node point of the FEM model,
n - number of samples of the analysis time,
k - number of node points in the FEM model.

Such measure was successfully applied in the studies referred to in [6-8], for the assessment of vibroactivity of the FEM models of the gear housings.

Figures 3 and 4 present the determined measure of vibroactivity of the FEM models, calculated for the signals of the normal vibration velocities in the upper lid of the housing, at various rotational velocity of shafts and various unit load rates of the gears of Q_1 =1 MPa and Q_2 =3 MPa. For the tests, it was assumed that the gears are manufactured in 6^{th} precision class.

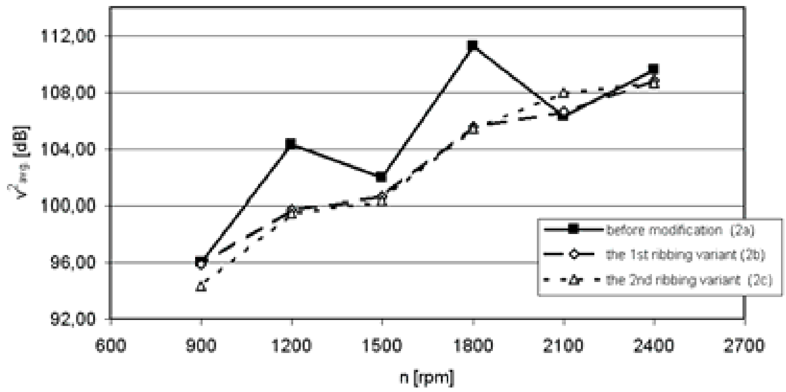


Fig. 3. The change of the $v^2_{avg.}$ depending on the rotational velocity of the gear and ribbing solution variant of the toothed gear housing was determined at the load rate Q_1=1 MPa

According to characteristics derived in the paper, the ribbed housings showed lower vibroactivity as compared to those without ribbing. The greatest differences in vibroactivity between the ribbed and unribbed housings are observed at lower load rates applied to the transmission. In that case, the ribbed housings were also characterised by similar levels of vibration emission. In tests performed with greater load applied to the gear transmission, Q = 3 MPa, variation of the v^2_{avg} measure is observed, depending on the shape of ribbing and rotational velocity, whereby the housing model in variant 2 (Fig. 2c) showed lower vibroactivity.

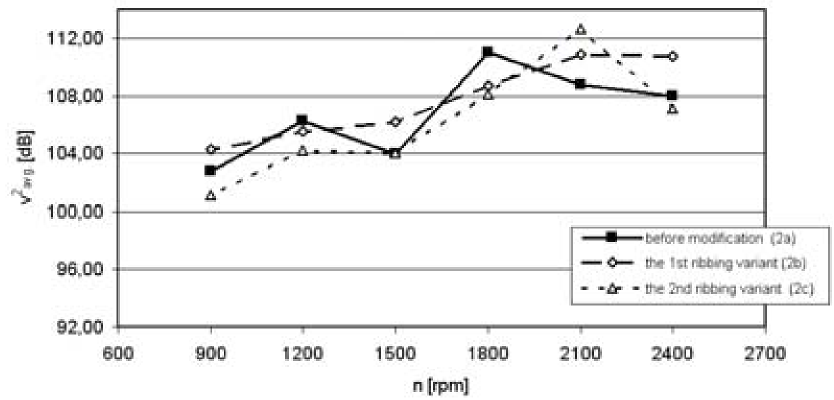


Fig. 4. The change of the $v^2_{avg.}$ depending on the rotational velocity of the gear and ribbing solution variant of the toothed gear housing was determined at the load rate Q_2=3 MPa

While analysing the effect of manufacturing precision of toothed gears, housings with and without ribbing were tested in variant 2 (Fig. 2 c), for a gear operating under the unit load rate of Q_2 = 3 MPa and gear shaft velocity n = 1800 rpm. In Fig. 5, the results of vibroactivity measure $v^2_{avg.}$ are presented, depending on the manufacturing precision of gear teeth.

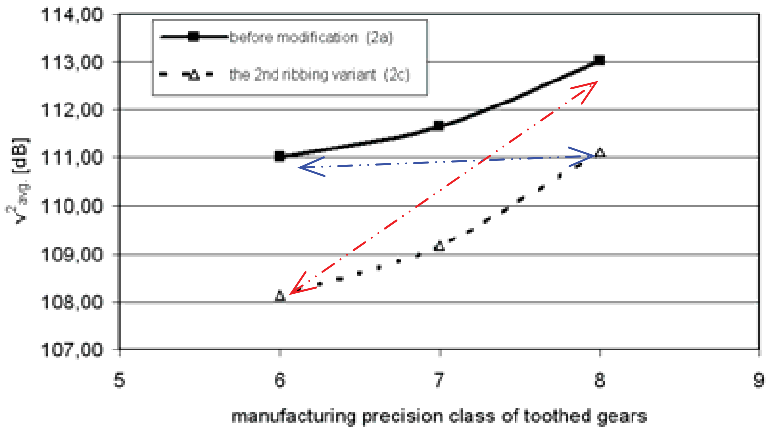


Fig. 5. Comparison of the vibroactivity measure $v^2_{avg.}$ depending on the manufacturing precision class of the toothed gears for the housings with and without ribbing in variant 2 (Fig. 2 c) under the load rate of Q_2 = 3 MPa

The values of the vibroactivity measure determined in that case indicate sensitivity of the gear housings to variations in the forces excitating vibration resulting from the manufacturing inaccuracy of toothed gears. In both models, with and without ribbing (Fig. 2 c), an increase of vibration rates was observed, along with the reduction of the quality (manufacturing precision) of the toothed gears. Variations of that measure in the models of the ribbed and unribbed housings (Fig. 2c) range from 3 dB for the 6th manufacturing precision class, to 2 dB for the 8th class, respectively. It can also be noted that the use of appropriately shaped ribbing of the housing, together with higher manufacturing precision of toothed gears classes from 8 to 6, allows to reduce the vibroactivity of the gear transmission housing by almost 5 dB (the red arrow in Fig. 5). Furthermore, it can be concluded that when a transmission of low vibroactivity rate of gears (Fig. 2 c) is used with the manufacturing inaccuracy class 8, the rate of the vibroactivity measure is close to the respective value calculated in case of the housing without ribbing (Fig. 2a), assuming that the toothed gears are manufactured in 6th precision class (the blue arrow in Fig. 5).

4. Summary and conclusions

The research and development activities carried out these days allow the use of simulation methods for designing machinery of low vibroactivity. Application of the FEM models of gear transmissions allows the housing vibration to be structurally analysed and to seek optimum solutions for the location of ribbing due to the reduction of vibroactivity. Such approach reduces also the cost of testing prototypes, as each of the proposed structural solutions can be analysed by means of virtual models, without creating the real objects.

Results of the research presented herein allow a conclusion to be drawn that the proposed solutions for ribbing of housings add to the reduction of vibroactivity. It was further acknowledged that, along with the increase of manufacturing inaccuracy of the meshing, the vibration level of the housing increases, both the ribbed (Fig. 2 a) and unribbed (Fig. 2c) ones.

While analysing the results of calculations of vibration velocity in the virtual models of gear transmission housings, their vibroactivity can be compared in a specified range of rotational velocity and such location and shape of ribbing can be selected that ensure the lowest vibroactivity level.

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