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OPTIMIZATION OF GEAR-MESH SINGLE-STAGE GEARBOX FOR REGIONAL TRAINS BY FEM

Summary. This article describes the methodology, calculations and results of gear-mesh optimization by finite element method for single-stage gearbox – type AGV-250. Gearbox is meant for regional trains. Results of the calculations are used for examination of changes in the gear-mesh of the tooth flank under load, particularly in term of improved uniform load distribution through a width of the tooth. The aim of the methodology is to design optimized tooth profile with application of longitudinal, transverse and angular modifications of tooth flank on the basis of analysis of calculation model by FEM. The article compares several variants of modifications according to distribution of contact pressure (stress).

Keywords: gear-mesh; FEM; contact pressure; modifications; gear; gearbox.

OPTYMALIZACJA ZĄ POMOCĄ MES PRZYPORU ZAZĘBIENIA PRZEKŁADNI JEDNOSTOPNIOWEJ STOSOWANEJ W POCIĄGACH PODMIEJSCKICH

Streszczenie. Artykuł opisuje metodykę, obliczenia oraz rezultaty optymalizacji przyporu zazębienia za pomocą metody elementów skończonych dla przekładni jednostopniowej – typ AGV-250. Przekładnia ta jest przeznaczona dla pociągów podmiejskich. Wyniki obliczeń deformacji elementów przekładni zostały zastosowane podczas modyfikacji boku zębów, aby osiągnąć lepszy przypór zazębienia. Uzyskano to z wykorzystaniem optymalizacji rozłożenia nacisków stykowych na szerokości i wysokości zęba. Celem metodyki jest zaproponowanie optymalnego kształtu powierzchni zęba za pomocą modyfikacji podłużnej, poprzecznej lub kątowej zgodnie z obliczeniami MES. W artykule porównano kilka wariantów modyfikacji, opierając się na rozłożeniu ciśnienia stykowego na powierzchni zęba.

Słowa kluczowe: przypór zazębienia, MES, ciśnienie stykowe, koło zębate, przekładnia.

1. INTRODUCTION

This article describes the problem of position and shape changes of gears in the gear-mesh under load and methodology, which handles compensation for these negative phenomena. Application of the finite element method (FEM) for the calculation of gearing

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with respect to real shapes of individual components of the transmission system can accelerate the gear-mesh optimization process [1].

The results of the calculations are used for examination of changes in the gear-mesh of tooth flank under load, particularly in term of improved uniform load distribution through a width of tooth. The aim of methodology is to design optimized tooth profile with application of transverse, longitudinal and angular modifications on the basis of analysis calculation of model by FEM.

The modifications are implemented to the single-stage gearbox AGV-250 from Wikov MGI [2]. The gearbox is shown in the Fig. 1. Gearbox is through output shaft of wheel-set, which is in contact through two wheels with rails and through elastic elements, attached to the bogie frame. Gearbox is on other side attached through suspensions, which include two silentblocks, into the bogie frame. The input shaft is connected through coupling with the motor. FEM results from seven variants of modifications are compared on the basis of contact pressure (stress).

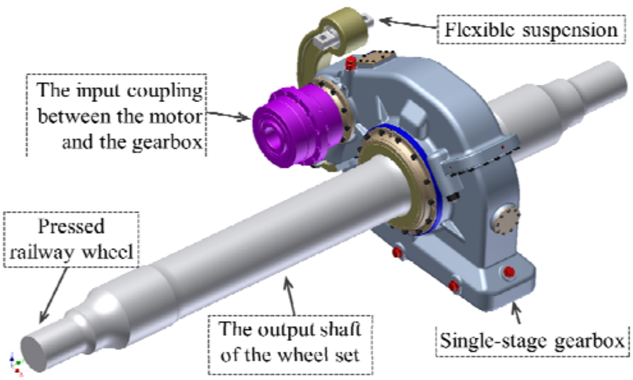

Fig. 1. Single-stage gearbox AGV-250 with coupling and wheel-set output shaft
Rys. 1. Przekładnia jednostopniowa AGW-250 ze sprzęgłem oraz wałem zestawu kołowego

2. THE REASONS OF THE INEQUALITY OF GEAR-MESH UNDER LOAD

The final gear-mesh conditions are influenced by several factors. The most important factors are deviations from the ideal shape of gears, clearance, assembly variations and transmission system parts deformation. The final position of gear-mesh is heavily affected by loaded geometry of gearbox, shafts, bearings and gears. These components have fixed relations among each other and are shown in the Fig. 2. These reasons (deviations) are included in the final modified shapes of the tooth.

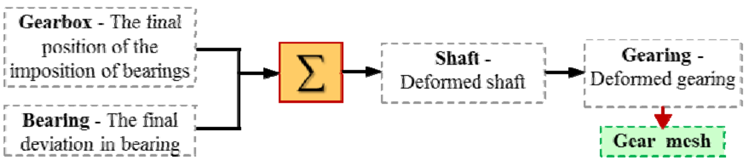

Fig. 2. Relations affecting the gear-mesh [1]
Rys. 2. Stosunki oddziałujące na zazębienie [1]

Based on calculations by FEM for individual elements according Fig. 2, these conclusions can be made. An essential element that influences the design process is the shaft.

Another important part is the gearbox, which optimized construction, must have sufficient rigidity for the position of the bearings. The part which affects the calculation on the lower level is bearing. Separate problem is the stiffness of own teeth.

3. FEM MODELS AND CALCULATIOS OF INDIVIDUAL GEARBOX ELEMENTS

Description of individual elements of the gearbox is divided into three parts.

3.1. FEM model and calculation of gearbox

Material of gearbox is ductile cast iron EN-GJS-500-7. After loading of CAD model into FEM connection between upper and lower part of the gearbox by the contact „Surface to surface" was made. Both parts of gearbox are bolted together. Transfer reaction forces from the shafts are applied at the center point of bearing and the transfer of the gearbox is simulated as type „Coupling". In the Fig. 3 is shown completely meshed FEM model (on the left) and deformation from the load reaction forces (on the right) [2].

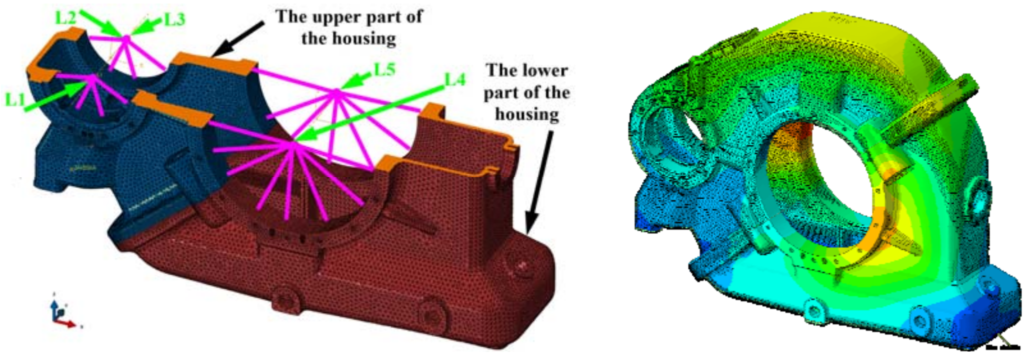

Fig. 3. FEM model of gearbox with simulation of bolts in dividing plane and deformation of gearbox
Rys. 3. MES model skrzyni z symulacją śruby w płaszczyźnie dzielącej oraz deformacja skrzyni

The result of the calculation is the spatial layout of points which characterizes the entry position of the points of the shaft ($f_{xU\_deform\_i}$, $f_{zU\_deform\_i}$).

3.2. FEM model and calculation of shafts

The output shaft has locked rotation around own axis Y. On the input shaft is applied load torque (M_1 = 1960 N·m). The calculation is performed in two steps. At first, are activated tensile forces into the bolts between the output gear and the hub, and then is applied the load torque on the input shaft to M_1.

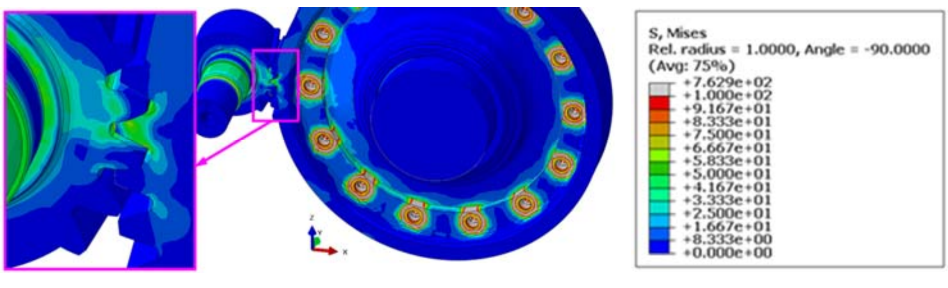

Fig. 4. FEM model of shafts with results of Mises stress [Nmm^{-2}] under load
Rys. 4. MES model wału z wypadkową naprężeń Misesa [Nmm^{-2}] podczas obciążenia

3.3. Summation of all parameters affecting the gear-mesh

In this point the result is a displacement and rotation of gears. Calculation starts by displacement of points in the shaft fits. Deviations of the support and the shaft in directions X and Z are calculated according to formulas (1) and (2).

$$f_{xL\_i} = f_{xU\_prod\_i} + f_{xU\_deform\_i} + \left(f_{L\_clearance\_i} + f_{L\_deform\_i}\right) \cdot \cos\varepsilon_i \tag{1}$$

$$f_{zL\_i} = f_{zU\_prod\_i} + f_{zU\_deform\_i} + \left(f_{L\_clearance\_i} + f_{L\_deform\_i}\right) \cdot \cos\varepsilon_i \tag{2}$$

where: $f_{xU\_prod\_i}$ – production deviation from the ideal position of support in the dir. X [mm];
$f_{xU\_deform\_i}$ – deviation result from the deformation of gearbox in the dir. X [mm];
$f_{L\_clearance\_i}$ – radial clearance in loaded bearing [mm];
$f_{L\_deform\_i}$ – radial deviation resulted from the deformation of the bearing „i" [mm];
ε – angle of the direction of load bearing in the plane XY [°].

In the Fig. 5 are shown representations of the new shaft positions. In the Table 1 are shown the resulting values of new midpoints position of gear according to Fig. 5.

Table 1

Resulting values of new midpoint position of gears

	Total Displacement [mm]		Total Rotation [°]	
	δ_{xL}	δ_{zL}	δ_z – Plane XY	δ_x – Plane ZY
O_{12}	0,02622381	-0,04098320	0,00655374	0,01529100
O_{34}	0,02077178	-0,03239230	0,00939368	0,01688300

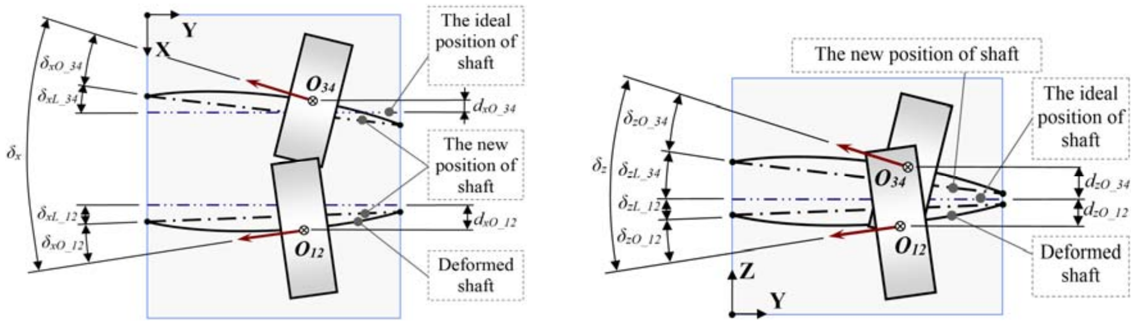

Fig. 5. Representation of total displacements and rotation of gears in the plane XZ and plane ZY
Rys. 5. Przedstawienie ogólnej deformacji nachylenia kół zębatych w płaszczyźnie XZ oraz płaszczyźnie ZY

3.4. FEM model and calculation of gear-mesh

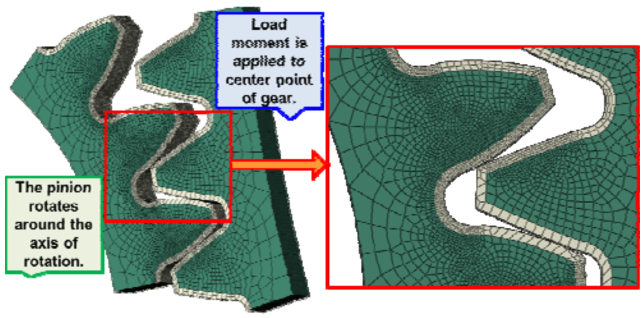

Fig. 6. The complete FEM model of gearing for simulation of modifications [3]
Rys. 6. Kompleksowy MES model przyporu zazębienia dla symulacji modyfikacji [3]

The solution of the contact problem is created in the gearing (3D model - gear and pinion). The contact bond among tooth flank is added to the model and meshed. The gears are connected with the center point of rotation with help of interaction type „Coupling". The subsequent rolling of the gear (pinion) and the implementation of all moments are applied through these points. Simulation of gear-mesh is performed quasi-statically.

4. THE VARIANTS OF MODIFICATION USED FOR FEM CALCULATION

In the Table 2 are shown descriptions of modifications (seven variants) used for the FEM calculation of gear-mesh.

Table 2

Description of modifications used for the FEM calculation

Description of modifications (variants)	Indications
The ideal gear-mesh (without modification)	Ideal
Gear-mesh with displacement and rotation (without modif.)	Def
Transverse modification (c_{aa} = 0,015 mm)	Trans_0.015
Transverse (0,015mm) + longitudinal (0,009mm = c_b → pinion) modif.	Trans_Long
Longitudinal (angular) modifications ($\Delta\beta$ → 0,015 mm = c_b)	Long_0.015
Longitudinal (angular) modifications ($\Delta\beta$ → 0,009 mm = c_b)	Long _0.009
Longitudinal (angular) modifications ($\Delta\beta$ → 0,007 mm = c_b)	Long _0.007

4.1. Comparative graph and results of selected variants

From the program KISSsoft is maximum contact stress (pressure) for pinion σ_{H_p} = 1029,26 $N{\cdot}mm^{-2}$ and for gear σ_{H_g} = 1025,37 $N{\cdot}mm^{-2}$ (according DIN 3990). In the Table 3 are shown average and maximum values of contact pressure. Contact pressures on gears for seven variants of FEM calculations are shown in Fig. 7 and Fig. 8.

Table 3

Average and maximum values of contact pressure for each variant

Variants	Max. contact stress [$N{\cdot}mm^{-2}$]	Average contact stress [$N{\cdot}mm^{-2}$]
Ideal	1038	731
Def	858	760
Trans_0.015	993	808
Trans_Long	865	801
Long_0.015	1136	725
Long _0.009	959	694
Long _0.007	899	694

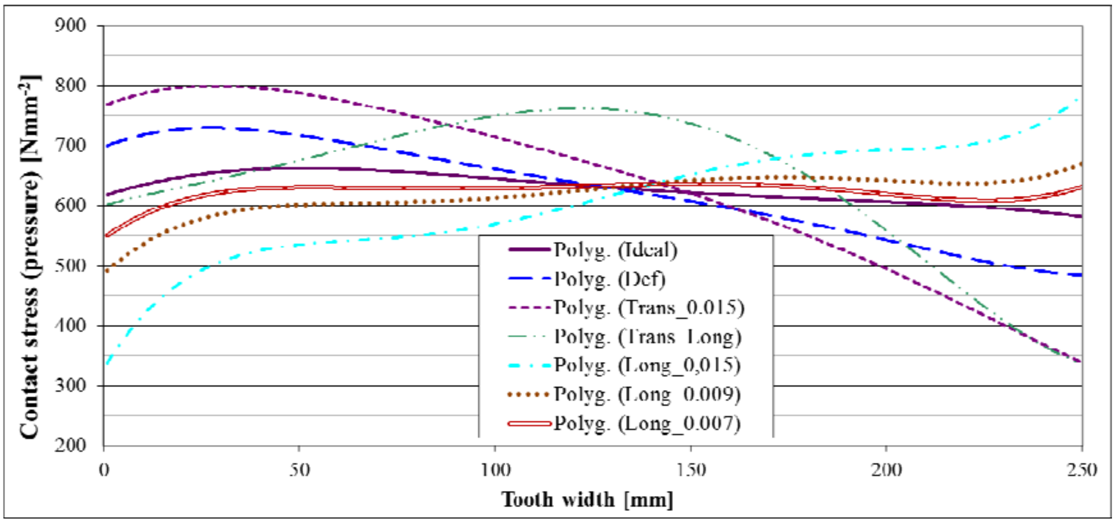

Fig. 7. Comparison of the average contact pressures on the pinion tooth flank along tooth width
Rys. 7. Porównanie naprężeń stykowych na boku zęba koła napędowego na szerokości zęba

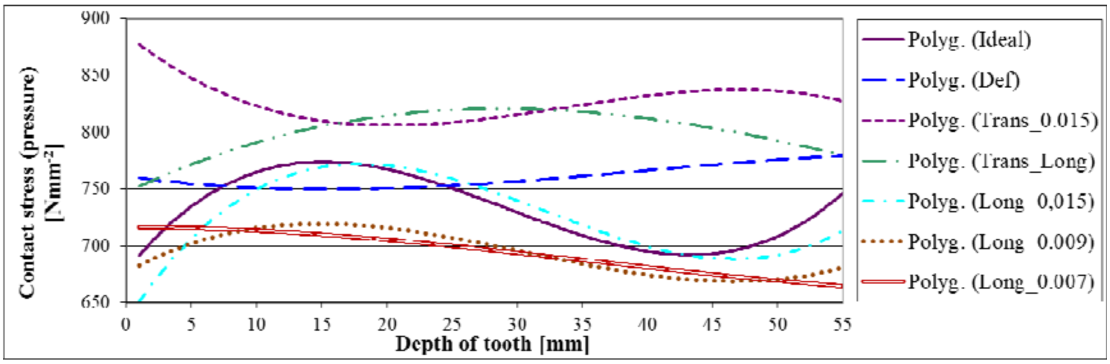

Fig. 8. Comparison of the average contact pressures on the pinion tooth flank along tooth depth
Rys. 8. Porównanie naprężeń stykowych na boku zęba koła napędowego na wysokości zęba

5. CONCLUSION

Results from the application of the methodology: The calculations of the modifications with help of the finite element method confirmed the applicability in the design of longitudinal (angular) and transverse modifications.

Already the first step where are calculated deformation of the gearbox and shafts, provided important results for the design of angular modification. By summing the calculated deviations of deformations, manufactured deviations from the ideal shape of the bearing and clearances is calculated final position from which can be easily calculated the angle modification. This has a decisive influence on the life of gears.

There were no problems during creating FEM models for the contact problem. During debugging and the FEM calculations a problem with hardware computing power has occured.

In the Fig. 7 and Fig. 8 are shown (Hertz) contact stress (pressure) on tooth flank without modification and deformation of shaft, gearbox (ideal); with respect of deformation of shaft and gearbox (def); and for five variants of modifications (with respect of deformation of shaft, gearbox, etc.).

The best solution was achieved with usage of longitudinal modification c_{bb} = 7 μm (Long_0.007). FEM results with 7 μm modification was compared in program KISSsoft and they corresponded.

Basic technical data about gearbox AGW-250: gear ratio 3,857; nominal power 340 kW; maximum input torque 3 528 Nm; maximum input shaft speed 4 200 min^{-1}; operating temperature -30° / +45° [2].

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