CASE STUDY OF ASSEMBLY ERRORS INFLUENCE ON STRESS DISTRIBUTION IN SPUR GEAR TRAIN

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ABSTRACT
This paper presents the results of complex FEA simulation research, which purpose was to evaluate influence of gearbox assembly errors on stress distribution in gear meshing zone. Review of the most common assembly errors and reasons of their appearance was described. Description of assumed simulation parameters like: geometry of tooth, number of teeth, gear ratio, centre distance and material of helical spur gear, loads and modelling methods of assembly errors were presented. In the end of paper, results of FEA simulation and their interpretation were described.

Keywords: assembly errors, FEA simulation, tooth geometry, stress distribution.

INTRODUCTION
Nowadays, the use of spur gears in power and torque transmission systems is widespread in mechanical industry. The reason for this are numerous advantages of gear train, among which the most important is high efficiency, tight structure and stable speed ratio [Palermo et. al. 2013; Zhou et. al., 2014]. However, even though that use of gear train is quite common these days, there are many issues that must be taken under consideration during gear designing process. Among them the most important is to assure the proper meshing contact of gears. That is very complex dynamic problem that can be significantly affected by many factors [Liasa et. al., 2013; Tsay, 1998]. One of these factors are assembly errors which may contribute to substantial decrease in gear train durability and significant drop of its performance, due to occurrence of kinematic transmission errors.

Assembly errors in gear transmission are caused by numerous inaccuracies that occurred in production process. These errors results in undesirable displacements which should be understood as a deviations in terms of the relative position and orientation between theoretical assembly and real assembly that actually happened. As it is stated in [ Palermo et. al. 2013; Houser], existence of assembly errors in gear transmission lead to the various types of teeth mesh misalignment which can be divided into the following categories:

- radial misalignment (known also as parallel misalignment in the offline plane of action,
- parallel misalignment in the plane of action,
- angular misalignment in the plane of action,
- angular misalignment in the offline plane of action.
According to the state of art in the field of assembly errors influence on gear transmission [ISO 6336-1, Palermo et al. 2013; Houser; Li, 2015], occurrence of angular misalignment in the plane of action (POA) is highly unwanted. It is found that this misalignment has great effect on contact stress distribution along tooth’s surface and the value of bending stress generated in tooth’s root [Li, 2015; ISO 6336-2, 3; Chen et al., 2002]. This is caused by change in teeth contact pattern from the uniform contact into side contact. In consequence mesh stiffness become significantly smaller, causing an appearance of kinematic transmission errors. Similar effect can be seen in case of parallel misalignment along the line of action (LOA), as it acts along the normal direction to the teeth surfaces. For this reason it also has great influence on teeth surfaces contact pattern and mesh stiffness. Occurrence of radial misalignment contribute to change in center distance. Because this misalignment acts along the offline line of action (OLOA), it is found to have substantial influence on value of tooth root banding stress, and therefore it affect significantly mesh stiffness. Effect of angular misalignment in the offline plane of action (OPOA) and axial offset is described in [Palermo et al. 2013; Li, 2007]. According to what is stated in this papers, both of this misalignments have limited influence on contact stress distribution and tooth root bending stress. For this reason this kinds of misalignments can be neglected, as their effect on mesh stiffness is limited.

Aim of presented research was to introduce the finite element (FE) modelling technique to evaluate influence of assembly precision on stress value and their distribution in elements of gear transmission. In order to achieve this goal, numerous simulations using finite element method were developed and conducted. In following sections, assumed 3D model of gear transmission and value of both torque load and assembly error for this studies were described in following sections.

THE SUBJECT OF RESEARCH

The tested object of these research was the single-stage reducing spur gear transmission that was a part of main drivetrain of a mixed-traffic, electrical EU-07 type locomotive. Essential parameters of this transmission are presented in Tab. 1, and view of meshed pinion with gear is shown in Fig. 2.
### Table 1 - Pinion and gear parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Gear</th>
<th>Pinion</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outside circle dia. [mm]</td>
<td>972</td>
<td>240</td>
</tr>
<tr>
<td>Pitch circle dia. [mm]</td>
<td>948</td>
<td>216</td>
</tr>
<tr>
<td>Root circle dia. [mm]</td>
<td>918</td>
<td>918</td>
</tr>
<tr>
<td>Width [mm]</td>
<td>121</td>
<td>127</td>
</tr>
<tr>
<td>Module [mm]</td>
<td>12</td>
<td></td>
</tr>
<tr>
<td>Pressure angle [°]</td>
<td>79</td>
<td>18</td>
</tr>
<tr>
<td>Number of teeth</td>
<td>1:4,39</td>
<td></td>
</tr>
<tr>
<td>Material</td>
<td>42CrV6</td>
<td></td>
</tr>
<tr>
<td>Yield strength [MPa]</td>
<td>950</td>
<td></td>
</tr>
<tr>
<td>Tensile strength [MPa]</td>
<td>1040</td>
<td></td>
</tr>
<tr>
<td>Young Modulus [MPa]</td>
<td>206000</td>
<td></td>
</tr>
<tr>
<td>Poisson coef.</td>
<td>0,3</td>
<td></td>
</tr>
</tbody>
</table>

Tested gear was assumed to experience two main types of external loads. First type of load was connected with transmitted torque, which value was calculated basing on known locomotive engine power, speeds and wheel diameter. Output torque value for tested gear transmission was determined as 42 kNm. As all conducted simulations were static type, dynamic effects were neglected.

Second type of load was connected with assembly errors existence. This errors were assumed to be angular misalignments in both POA and OPOA. Occurrence of these errors was the consequence of angular rotation of pinion caused by inaccurate bearing positioning. Value of pinion rotation were equal to 0,4° and 0,8°. These values were calculated as angle resulting from pinion bearings displacement distance which was equal to respectively 0,1 mm and 0,2 mm. Pinion rotation with bearing displacement was illustrated in Fig. 3.

#### FE MODELING

The geometric 3D model of the tested gear transmission was made and placed in three dimensional Cartesian coordinate system. Main axis of bigger wheel was fixed in space. Wheel was constrained as axially fixed and only rotation of wheel around its axis was
allowed. Cylindrical faces of pinion shaft, which supported bearings, were constrained with displacements in perpendicular planes, as described in previous chapter. Loading torque of value equal to 42,000 Nm was applied to the wheel. Pinion shaft rotation around its axis was fixed, in order to balance torque load and ensure convergence of simulation.

Using ANSYS Workbench environment, contact between gears was considered as frictional with coefficient of friction equal to 0.4. Contact problem formulation was described as augmented Lagrange formulation. Because of geometrical non-linearity resulting from contact formulation, solver used for this problem was selected as iterative, rather than direct type.

Geometrical model was simplified in order to optimize computing and obtain shorter processing time. The most important model geometry simplification was to remove teeth that were outside of meshing zone. Another important simplifications included removing shaft features, like reliefs, grooves, chamfers and radii. The model of tested gear transmission with the simplifications of gears geometry was presented in Fig. 4. In Fig. 5 concentration of finite element mesh in meshing zone was presented.

**RESULTS AND DISCUSSION**

Conducted analyses resulted in numerous stress distributions, representing strain of gear transmission in regard to applied assembly errors. These results especially included stress distribution within meshing zone and teeth cross-sections. In order to describe strain of gears and teeth, Huber-Mises-Hencky (HMH) strain hypothesis was used in stress value calculations. Therefore, all presented stress distribution graphs show distribution of HMH reduced stress in MPa.

The first executed simulation assumed desirable situation with no occurrence of assembly errors in tested gear transmission and application only of external load in the form of torque of value equal to 42 kNm. This assumption allowed to determine the correctness of the developed gear transmission model as well as to obtain reference data that would help in evaluation of influence of particular assembly error on generated stress and strain of meshed gears. In Fig. 6 the result of this simulation was presented. According to obtained data, maximum noticed value of reduced stress was 585 MPa. What is more, this value occurred in the contact zone of meshed teeth as expected. In the teeth cores and outside of the meshing zone stress value did not exceeded above 200 MPa. This kind of stress distribution confirmed correctness of developed gear transmission model and assumed boundary conditions.
The next two conducted simulations assumed existence of angular misalignment in the plane of action. As it was stated in the previous chapter, occurrence of this misalignment was caused by rotation of the pinion shaft, whose value was equal to 0.4° and 0.8°.

In Fig. 7 and Fig. 8 the obtained results are presented. In compare to case without assembly errors presence, it was observed that stress value and its distribution in contact zone of meshed teeth changed. On the basis of the simulation results analysis it was concluded, that increase of pinion shaft angular misalignment resulted in increase of the stress concentration on one side of the tooth combined with increase of this stress value. The maximum value of HMH stress that occurred in the contact zone of meshed teeth was equal to respectively 1351 MPa (in case of misalignment equal to 0.4°) and 2026 MPa (in case of misalignment equal to 0.8°).
Next conducted simulations assumed presence of angular misalignment in the offline plane of action. Similarly to the previous case, value of this misalignment was equal respectively 0,4° and 0,8°. On the basis of the conducted simulations results, that were presented in Fig. 9 and Fig. 10, it was concluded that the occurrence of the assumed in this case misalignment, had considerably bigger impact on the stress that was generated within the pinion shaft, rather than stress that was generated in the contact zone of the meshed teeth. Analysing the stress distribution in the contact zone it was found to be slightly shifted to the side of the tooth. Compared with reference case slight increase of stress in mesh zone was also noted, as its maximum value was equal respectively 948 MPa (in case of misalignment equal to 0,4°) and 1111 MPa (in case of misalignment equal to 0,8°).
DISCUSSION AND CONCLUSIONS

In this paper an influence of assembly errors on stress distribution in spur gears was examined. For this purpose the multibody model of gear transmission was developed to conduct tooth load, surface contact stress and root bending stress calculations of a pair of meshed spur gears. The model of tested transmission was developed with taking into account the instantaneous operating conditions, defined as relative positioning and instantaneous transmitted load.

The results of the conducted simulations had allowed to formulate the following conclusions:

- The presence of the angular misalignment in the plane of action of the meshing zone was considerably more unfavorable for stress distribution and their value in compare with the occurrence of angular misalignment in the offline plane of action.

- For the simulations where occurrence of the angular misalignment in the POA was assumed, significant change in stress distribution in contact zone combined with increase of this stress was noted. In this case stress was concentrated on the side of the tooth. What is more, the value of this stress increased significantly and was equal to respectively 1351 MPa (in case of misalignment equal to 0,4°) and 2026 MPa (in case of misalignment equal to 0,8°). This results allowed to conclude that even small increase of angular misalignment in the POA is highly unwanted, as it may result in significant change in stress distribution and increase in value of generated stress.

- For the simulations where occurrence of the angular misalignment in the OPOA was assumed, slight change in stress distribution in contact zone combined was slight increase of this stress was noted. In this case, stress that was generated within contact zone achieved value of respectively 948 MPa (in case of misalignment equal to 0,4°) and 1111 MPa (in case of misalignment equal to 0,8°). However, it was observed that presence of this kind of misalignment has much bigger effect on stress generated within shaft, therefore its occurrence is highly unwanted for shaft durability.
REFERENCES


