Analysis of the effect deformations of individual components of gearbox on her total lifetime

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Abstract. The article deals with the influence of deformations of the individual components of the gearbox as well as their influence on the correctness of the gearing, which is essential for the correct functioning of the gearbox and its overall service life. In the next section the article deals with the effect of the gearbox deformation on the contact of the gears and the effect on the final life of the gearbox. The article it contains processing the computational model of gearbox, strength analysis gearbox, contact analysis of the selected toothed gears, taking into account the deformation of the teeth, shafts and the gearbox. The results of the work is overview size of influence individual components of gearbox on their service life.

1. Introduction
Toothed gears form an important group of transmissions. They are extensively used throughout the machine industry, but also in other mechanical devices where input parameters change to the required working values. The correctness of the well-designed gearbox is achieved in a very precise way, taking into account the effect of the deformations of the individual components of the gearbox resulting from the operating load. Due to the high input performance and the effort to reduce the weight of the individual parts of the gear, the gearboxes become heavy and so their weight is resolved by optimizing cross-sections, resulting in greater flexibility of the cabinet at the same load. It is necessary to highlight the fact, that the overall rigidity and weight of the gearbox also affects the material of cubicle especially for devices designed for example: to speed races, where weight is a key parameter, thus increasing the cabinet flexibility requirement.

2. Input parameters of the gearbox
The subject of the research is a two-stage front planet gearbox comprising three sloping toothed gearing, which is designed for the drive system the crane.

| Parameter                        | size              |
|----------------------------------|-------------------|
| Maximum input performance [kW]  | 20                |
| Maximum input shaft rpm [min⁻¹] | 938               |
| Computational lifetime [hod]     | over 100 000      |
| Operating temperature [°C]       | max. 60           |
| Oil filling                      | synthetic gear oil|
3. Creation computing model of the gearbox by software KISSsoft a KISSsys
The researched two-speed gearbox is modeled in a program that primarily focuses on dimensioning of the machine parts. This is computer software from the company KISSsoft. The software is divided into two parts. The first part consists of the KISSsys program, which serves to calculate the mechanisms kinematics and counts all other unknown variables from the initial input data. The second part consists of the KISSsoft program, which subsequently applies the acquired parameters for calculating the individual elements of the assembly, such as for example strength calculations of the individual parts of the gear unit. In KISSsoft the individual parts of the gearbox acquire its dimensions and mechanical properties. The Creation a model of parts of the gearbox runs on the program KISSsoft in the individual subprogram. After entering all the necessary data, KISSsoft will subsequently execute the selected calculations, which will be saved in the model main assembly in the program KISSsys. The data obtained will be followed by the whole model, where after the calculations have been completed, a technical report containing all the model information, all the solved calculations, the safety results and the lifetime of the individual elements will be created.

3.1. The Geometric and strength calculation of gearings, shafts and bearings
The program works with its own databases of material data, standardized parts and lubricating fluids, which are gradually being expanded and updated. The program KISSsoft contains computational equations compiled from valid norms, in which it allow users to define variables according to the assembled model. The program allows a user to define specific component dimensions, material choice, operating conditions and also a specific method of strength calculation based on the current valid norm. As a result of this calculation, there is an extensive technical report containing all the information about the model and the parameters used, along with the calculations a graphics processing.

4. Strength analysis of the gearbox
The gearbox analysis was done using the ANSYS Workbench numerical calculation software based on the principe Finite Element Method. This a method consists in dividing the solve body into the set of finite elements interconnected at the node points. The finite element method at the individual nodal point resolves the size of the shifts, deformations, and stresses [8].

4.1. Creating a 3D model, FEM mesh and defining a boundary conditions of the gearbox cabinet
The 3D model of the gearbox was created using the software Creo parametric. Transmission housing construction has a welded character and consists of two parts. Belonging of a gearbox also are they covers, which contribute positively to overall stiffness advancing. The model of a gearbox was created simplified form, it means: does not contain welds and holes for screw connections. After this
the assembled model was imported into the ANSYS Workbench computational program in format step. Through the program application, the material properties of the gearbox, namely the isotropic material, the Young's elastic modulus of \( 2 \times 10^5 \) MPa and the poisson number of 0.3 were defined. Using a Boolean operation the gearbox cabinet a assembly counted as a one whole was, for that reason, that it was not considered with a preload in the screw connections. Because is assumed, that a screw connections created a sufficiently rigid joint between the two coupled parts of a gearbox cabinet. By the numerical calculation of the volume bodies the ANSYS Workbench divides a solved volume into final elements. For the calculation to be correct, the network homogeneity must be ensured, that means, the contiguous final elements have common border node points. This homogeneity was created in our case by the aforementioned function Boolean. Setup of network the function on Mechanical, size function selected on Curvature and her a softness (density) on Fine. With this global settings of mesh, a sufficiently dense a mesh, which contains 252 343 finite elements and 403,569 nodal points, was obtained for the calculation [4], [8].

![Figure 2. The depiction the FEM mash.](image)

The gearbox is loaded by the calculated reaction forces in the bearings. For the direction of action of the reaction forces coincided with the KISSsoft coordinate system, a local coordinate system was created. The sizes of the individual load forces are shown in Table 2. In the software ANSYS the utilized function the bearing load - BearingLoad, but which and not include the axial element the acting on the bearing. It was necessary by this bearing loads add the axial forces ceated by program function - Force.

![Figure 3. The definition the load on a gearbox at left and removing degrees of freedom at right.](image)

| Force | 1      | 2      | 3      | 4      | 5      | 6      |
|-------|--------|--------|--------|--------|--------|--------|
| Fx [N]| -4563  | -4638  | 21 883 | 21 808 | -17 144| -17 347|
| Fy [N]| -4831  | 1998   | 2833   | 0      | 0      | 0      |
| Fz [N]| -1393  | -2318  | -4306  | -7133  | 7530   | 7620   |

Table 2. Sizes of reaction forces in bearings.
Anchoring a gearbox is implemented by means of lateral grips, by which displacements were removed from in a Z axis direction, according to the local coordinate system. The lateral grips are coined a four screw openings have, where consequently the degrees of freedom in the X and Y axis direction was been removed. So so loaded and anchored the gearbox cabinet was subjected to strength calculations with the aim of finding locations with a concentration of tension and a maximum size of a displacement. The number of solved a equations in this case is 1 210 707.

4.2. The results of strength analysis and coefficient of safety

The realizated of a FEM analysis revealed the maximum value of the equivalent stress (Von Mises) and its site of action. The concentration of stress is located at the points anchoring of the gearbox to the support frame. Because of the simplified model of the gearbox, is the place of concentration at the edge of the attachment and on the frame of the cabinet (sharp edge). The magnitude of the maximum voltage at this location has been found about as $\sigma_{\text{max}} = 47.79 \text{ MPa}$. 

![Figure 4. Equivalent stress and a place of concentration.](image1.png)

The gearbox cabinet is made of structural steel marked with according norm EN S355J2. It is a weldable non-alloy structural steel, the values of the mechanical properties of the selected material are defined for tensile strength $R_m = 470 – 630 \text{ MPa}$ and the minimum characteristic stress the for the thickness of the material to 40 mm je $R_e = 345 – 355 \text{ MPa}$. The gearbox is loaded statically, allowed static tension $\sigma_{\text{Dt}} = 140 – 210 \text{ MPa}$. The coefficient of the manual welding was selected as $\alpha = 0.7$, then the value of the permitted actual tension in the welding site of value came out 98 MPa. The safety coefficient of the weld with the highest concentration of stress is $k = 2.05$. It is possible to deduce from the above calculation of weld safety under the effect of operating load the conclusion, that the most loaded welded site is suitable on the strength. Due to the influence of the reaction forces in the bearings, it was the gearbox a deformed. In Figure 5 is shown a displacement of the gearbox in face of to the unloaded condition. The dimension of maximum displacement at this place in face of to the unloaded condition is 0.011 mm [4].

![Figure 5. The displacement size delineation of the gearbox.](image2.png)
5. Contact analysis of toothed gears

The toothed gears have the ability to transmit maximum performance then, when the shafts of the toothed gears are perfectly arranged and the load transmitted is evenly distributed across the entire active tooth width. The so how as the load decomposes on the surface the side of the tooth, it greatly affects the life of the gearing. Incorrect load distribution a results inception of a place on the surface of the tooth with high contact stress, resulting in fatigue wear or the fracture fatigue. On the load distribution have on the sides of the toothed has a affect negatively the manufacturing and mounting inaccuracies, the elastic deformations of the individual components of the gear unit, the thermal deformations and the external influences that can cause the toothed engagement to deflect. The during at assembling parts of the gearbox, a check should be made to avoid unexpected a damage the during operation. Inspection of the tooth surface sides contact area should also be performed during operation to detect damage caused, for example, bearings failure. An important consideration of the contact analysis is the so-called the coefficient of unevenness of the tooth load after the width in contact $K_{H\beta}$ [2-5].

5.1. Definition of the coefficient $K_{H\beta}$ without shafts deformation

The software KISSsoft includes several ways of defining a coefficient $K_{H\beta}$. If the value is known, it can be entered manually, which is the first method. Another method of defining the coefficient is, depending of the used the calculating norm, in this case it is the specific norm DIN 3990 method B (method C), which is relate from the norm ISO 6336 method B, consequently it contains the approximate procedure. For the contact analysis, a third gearing unit was selected, which consists of a z4 a pinion gear in middle shaft and a gear a z6 output shaft wheel. When determining the $K_{H\beta}$ the coefficient according to the DIN norm, the dimensions of the pinion shaft are included, but the driven shaft geometry in this case is not included. The specified the pinion shaft values are the distance between the bearings $l$, the distance of the pinion from the center of the distance between the bearings $s$ and the outer diameter of the shaft between the bearings $d$. Another parameter that the norm takes into account is the pinion shaft type. That is defined the based on the norm ISO 6336 (type $a$ as much as $e$). The values given for the calculation of the $K_{H\beta}$ coefficient for the third gear are $l = 243$ mm, $s = 70.5$ mm, $d = 62,256$ mm. With this the values so defined the coefficient is defined $K_{H\beta} = 1,1525$. In this calculation, only the geometry of the pinion shaft is taken into account, the geometry of the output shaft is not included. Due to load, the teeth of the wheels resiliently deform [1-3].

| Table 3. The results of a contact analysis without shafts deformation. |
|---------------------------------------------------------------|
| min       | max      |
|-----------------|----------|
| Transmission error [µm] | -35,849  -31,093 |
| Loss of performance [W] | 53,061   142,002 |
| Temperature of a contact surfaces [°C] | 61,975   110,58 |
| Thickness of the lubrication layer [µm] | 0,046    0,300 |
| Hertz of contact stress [MPa]| 2183,664 |
| Safety against emergence microprinting | 0,164 |
| Safety against abrasion | 12,277 |
| The coefficient of the gear duration under load/without load | 2,753 / 2,660 |
| The gearing efficiency | 98,87 |

Figure 6. Layout of tension across the width of tooth at left of pinion and at right a wheel.
In Figure 6 is shown the distribution of the applied load on the width of the tooth, with the red color showing the location with the highest contact stress. Places with the maximum stress occur at the beginning of the gear engagement and at the center of the gear engagement, while the maximum tension on the pinion occurs at the root of the tooth. The wheel has maximum tension on the head of the tooth, at the beginning and at the center of the tooth width. Such a stress distribution is not ideal because the blue color area, the stress 545 MPa, should be located outside the spacing circle. In this case, the medium contact stress, a green color with a voltage of about 1091 MPa, is located in the head and the root of the tooth from the center to the end of the toothed engagement.

5.2. Definition of the coefficient $K_{H\beta}$ with shafts deformation

The toothed gears transfer performance causing stress and torque load on the shafts what caused their elastic deformations. These deviations may affect the toothed wheel engagement and hence the effect on load distribution on the side of the tooth. A more precise definition of the $K_{H\beta}$ coefficient is the application of the norm ISO 6336-1 Annex E. Annex E of this norm takes into account the load distribution on the tooth width between parallel axis of the cog wheels. This method focuses on the most important deviations, namely bends and twisting of the shafts and the elastic deformation of the wheel teeth. The determination of the $K_{H\beta}$ coefficient was carried out on the basis of the norm ISO 6336-1 Annex E with by considering deflections of a shafts of a pinion, wheels bending, and tooth bending. The shaft deflection of the pinion under the tooth gear has value 0.0852 mm and the output shaft deflection of the under the tooth gear has value 0.0634 mm. The size of twisting angle in the pinion shaft in the place toothing is 0.0016° and the size of twisting angle the output shaft in the place toothing is 0.0126°. This calculation does not take into account manufacturing tolerances. The value of the coefficient of the $K_{H\beta}$ in this case is 1.2155 [1-3].

Table 4. The results of a contact analysis with shafts deformation.

|                        | min     | max     |
|------------------------|---------|---------|
| Transmission error [µm]| -35,847 | -31,072 |
| Loss of performance [W]| 46,805  | 132,954 |
| Temperature of a contact surfaces [°C]| 61,987  | 94,959  |
| Thickness of the lubrication layer [µm]| 0,048   | 0,43    |
| Hertz of contact stress [MPa]| 2066,821 |
| Safety against emergence microprinting| 0,171   |
| Safety against abrasion| 17,764  |
| The coefficient of the gear duration under load/without load| 2,746 / 2,660 |
| The gearing efficiency| 99,03   |

Figure 7. Layout of tension across the width of the tooth at left of the pinion and at right the wheel.

In Figure 7 is shows the distribution of the applied load along the width of the tooth, with a considerable difference in results compared to the result without considering of the shafts deflection. The contact area of the pinion its contents a larger area of pale blue color, meaning a contact voltage of about 517 MPa and a distinctive green color area, representing a voltage of approximately 1033 MPa, mostly at the beginning of the engagement. The range of the maximum stress is again at the bottom at the beginning of the toothed engagement.
5.3. Definition of the coefficient $K_{H\beta}$ with a shafts deformation and the gearbox cabinet deformation

The software KISSsys is in to computing of the $K_{H\beta}$ coefficient is able to incorporate the stiffness of a particular the gearbox cabinet. The stiffness of the cabinet is described by a reduced the stiffness matrix. The stiffness matrix of the cabinet is reduced in this case to six points representing the center of the bearings. Subsequently, these points are loaded by the calculated reaction forces in the bearings and thus of the displacements of the bearings in the direction of the axes $x$, $y$ and $z$ is obtain. The offset values obtained are applied in to the calculation of the $K_{H\beta}$ coefficient, while will be considered with in the calculation of the shaft deformation and of the wheels teeths bending [1-2].

5.3.1. Reduced stiffness matrix of the gearbox cabinet

The software ANSYS Workbench was used to obtain the reduced stiffness matrix of the gearbox cabinet. The same properties of the gearbox were used as in the previous analysis, i.e. the isotropic material, Young's elastic modulus of $2 \times 10^5$ MPa and the Poisson number of 0.3. The model of the gearbox is built-up than simplified, i.e. without screw holes, and withouts internal preload.

![Figure 8. Connecting the master node with a model and main nodes of the gearbox cabinet.](image)

The first step in the static analysis was the creation of six remote points, Remote Point, that corresponds to the center of the bearings. Each remote point is linked to the model of the gearbox cabinet at the node points of the FEM mash. The connection is described by numerical equation. Each Remote Point is defined as the main node, Master Node. For these the main nodes the reduced stiffness matrix is described, namely three the displacements and three a rotation. The static analysis was yielded stiffness in the axisys $x$, $y$, $z$ direction and modal analysis was to obtain residual stiffness by rotation around the $x$, $y$, $z$ axis relative to each of the main nodes. The resulting stiffnesses of the two analyzes are writted in the same file Stiffness_used.txt. The matrix statement in file Stiffness_used.txt is in a columnar shape, which has been transcribed into the shape of the stiffness matrix. The resulting matrix is symmetric in size 36x36, representing three displacements and three rotations for each main node. The last step to determining the $K_{H\beta}$ coefficient is importing the gearbox into the software KISSsys. The position of the housing is set so that the centers of the bearings are in the same position as the main nodes for which the reduced stiffness matrix is formed.

![Figure 9. Import and location of the gearbox cabinet in the software KISSsys.](image)
In the software KISSsys, each main node point it gets loaded with the calculated reaction forces in the bearings. The size and direction of action of the reaction forces is shown in Table 5. Reaction forces are used to calculate the displacements and rotations of nodal points, thereby creating a new the bearings position that impinges on the correctness of the toothed engagement. The results of the displacements of the main node points in the software KISSsys are shown in Table 5.

Table 5. The resultes of the displacements and rotations of the main points.

| Bearing | Displacements in the direction [mm] | Rotations around the axis [°] |
|---------|----------------------------------|-------------------------------|
|         | x      | y      | z      | x     | y     | z     |
| 1       | 0.004011 | 0.001296 | 0.00081 | -1.08*10^-5 | 0     | -1.13*10^-6 |
| 2       | 0.00387 | 0.00107 | 0.00082 | -8.81*10^-6 | 0     | -4.49*10^-6 |
| 3       | 0.00582 | -0.00085 | -0.00271 | -5.53*10^-5 | 0     | -2.69*10^-6 |
| 4       | 0.00172 | 0.000793 | -0.00295 | -4.32*10^-5 | 0     | -2.05*10^-6 |
| 5       | -0.00165 | -2.515*10^-5 | 0.00013 | -6.45*10^-7 | 0     | -1.21*10^-6 |
| 6       | -0.00163 | 9.874*10^-5 | 9.01*10^-5 | -5.99*10^-7 | 0     | 1.41*10^-7 |

From the above results it can be seen that the displacements are quite small, as was also constat by static analysis. To determine the size of the change, the calculation again focuses on changing the size of the $K_{H/β}$ coefficient. The calculation is carried out according to norm ISO 6336-1 Annex E, which considering in the calculates the bending of the teeth of the wheel, shafts deformation and the new bearings positions. The sizes of the pinion deformations and wheel deformations are the same as in the previous calculation.

Table 6. The results of a contact analysis with shafts deformation and the cabinet of the gearbox.

|                  | min     | max     |
|------------------|---------|---------|
| Transmission error [µm] | -36,048 | -31,271 |
| Loss of performance [W]   | 46,123  | 132,926 |
| Temperature of a contact surfaces [°C] | 61,989  | 95,08   |
| Thickness of the lubrication layer [µm] | 0,0484  | 0,483   |
| Hertz of contact stress [MPa] | 2086,018 | 17,703  |
| Safety against emergence microprinting | 0,17     |
| Safety against abrasion | 17,703   |
| The coefficient of the gear duration under load/without load | 2,758 / 2,660 |
| The gearing efficiency | 99,03    |

Figure 10. Layout of tension across the width of a tooth to at left of a pinion and at right a wheel.

The results of the contact analysis do not differ much with the results of the analysis, in which only with the shaft deformation and tooth bends were considered. The coefftient of the tooth load after the width, as it after the analysis without deformations of the gearbox cabinet has changed from value 1,2155 on the value 1,2261. This change in the coeffitient does not have a significant effect on the distribution of the tension over the width of the tooth, which implies that the gearbox cabinet is sufficiently stiff and there is no significant deformation due to the operating loads.
5.4. The optimization of the third gear, the toothed modification

To optimize the toothed engagement, of double-type tooth modification is used in practice. The first type is a modification the evolvent of the teeth, for example a linearly chamfer and a rounding in the area of the head and heel of the tooth, a changing the pressure angle $\alpha$ and their combinations. The advantages of this type are, for example, quiet running, smoother a toothed engagement and higher load stress. The second type is used to improve the load distribution over the width of the tooth, i.e. the $K_{\beta H}$ coefficient. There are various types of these modifications, for example the linear grinding of tooth start and end, rounded grinding of the tooth start and the tooth's end, the modification of angle the inclination tooth, the crowning of the teeth, the eccentric crowning of the teeth, contortion of the teeth and others. In this case, the optimization of the third gear set will be performed only by modification of the gear wheel. Modification of the pinion will not be done due to increased production costs. To improve the $K_{\beta H}$ coefficient, the modification across the width of the teeth was used, namely in the concrete in the crowning teeths to the norm ISO 6336-1 Annex B. Once the calculation has been completed, we have gained a value of the coefficient was $K_{\beta H} = 1,0914$. To remind, the difference of the $K_{\beta H}$ coefficient without modifying the tooth and considering the same deformations was 1,2261 [1-3].

Table 7. The results of contact analysis at contact of the teeth with modification.

|                      | min   | max   |
|----------------------|-------|-------|
| Transmission error [µm] | -36,07 | -31,994 |
| Loss of performance [W] | 42,438 | 130,906 |
| Temperature of a contact surfaces [°C] | 61,929 | 96,616 |
| Thickness of the lubrication layer [µm] | 0,053 | 0,309 |
| Hertz of contact stress [MPa] | 1762,94 (before 2086,018) |
| Safety against emergence microprinting | 0,186 |
| Safety against abrasion | 19,96 |
| The coefficient of the gear duration under load/without load | 2,716 / 2,660 |
| The gearing efficiency | 99,07 |
| The contact safety coefficient of the pinion / the wheel | 1,04 / 1,242 |
| The toothed durability | over 1 000 000 hod. |

Figure 11. Layout of tension across the width of the tooth at left of the pinion and at right a wheel.

Table 7 is shows a clear drop in the Hertz of contact stress with a difference of 323.08 MPa, which this a value achieved only a modification of the toothed wheel was. This difference in contact stress has also caused an increase the safety coefficient in contact. On the Figure 11 is shows the layout of tension across the width of the tooth of the pinion and the wheel, while modified was only the tooths of the wheel. In this case, the stress is distributed over the whole surface evenly, the size of which is about 880 MPa. The maximum stress is predominantly located in the heel of the tooth of the pinion and in the head of the tooth wheel in the beginning and center of the toothed engagement [8].
5.5 Comparison of the resulting coefficients $K_{H\beta}$

For comparison of the calculations, a brief overview of the resulting $K_{H\beta}$ coefficients for selected types of calculations is given. Table 8 describes how the $K_{H\beta}$ coefficient under the influence of deformations of the various parts of the gearbox, as well as the size of contact stress.

Table 8. Overview the results of the individual calculations of the $K_{H\beta}$ coefficient.

| Deformations considered | The calculation norm | Hertz of contact stress | Coefficient $K_{H\beta}$ |
|-------------------------|----------------------|------------------------|--------------------------|
| The bends of teeth      | DIN 3990             | 2183.664 MPa           | 1.1525                   |
| The bends of teeth + deformations of shafts | ISO 6336-1 annex E | 2066.821 MPa           | 1.2155                   |
| The bends of teeth + deformations of shafts + deformation of gearbox cabinet | ISO 6336-1 annex E | 2086.018 MPa           | 1.2261                   |
| The bends of teeth + deformations of shafts + deformation of gearbox cabinet + modification of wheel tooth | ISO 6336-1 annex E modification ISO 6336-1 annex E | 1762.94 MPa           | 1.0914                   |

6. Conclusion

At the conclusion is pointed out that, from the mentioned results of the contact analyzes they have a great influence on the correctness of the toothed engagement the shafts deformation, while increase of a coeffitient with consideration deformations of the gearbox cabinet has changed slightly, i.e. the gearbox has sufficient stiffness at the specified operational loads and material used. It can be said with certainty that the biggest impact on the overall life of the gearbox being examined have a toothed engagement. It appears from this that, the engineering design of the gearbox is a necessary attention and dimensioning of her individual of parts so that, as to minimize the size of the undesirable effects of the various deformations in accordance with the input conditions. An interesting could be the contact analysis of the gearbox, whose stiffness is smaller, so which would be made of more flexible material, the aluminum alloy [6-9].

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