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Modification of the Centre Differential Gearbox

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Abstract The paper deals with the optimization of the centre differential heavy truck without a wheel reductions. Under optimization in this case means customizing cabinet differential type designed for easy gear to withstand a heavy load of new gear. Critical structural adjustment of high-voltage areas must be such as to reduce tensions. The basis for this optimization is the results of task solution by finite element method (FEM).

Keywords: *centre differential, stress, FEM*

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1. Introduction

There are currently placed ever increasing demands on machine parameters. These are reflected in the growth dynamism load and the continuous improvement of the speed of its parts [1,2]. Therefore at the optimization of machine parameters it is necessary to focus on increasing needs for performance, accuracy, durability and reliability of the machinery itself [3]. The paper discusses optimizing center differential for heavy truck with 8x8 without wheel reductions. Compared with the previous type of gear in the output is increased the torque output. This torque causes in the gearing of centre differential more onerous force and so the differential gearbox is more stressed. Proposed modifications of this component are based on the finite element analysis.

2. Centre Differential for Heavy Truck

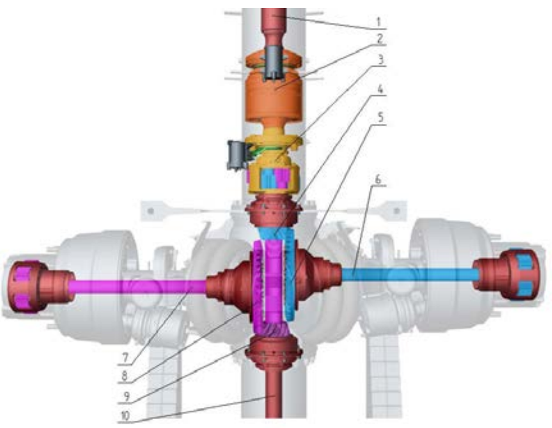

Figure 1. Distribution of the drive torque to the axles

The unique constructions concept solution has a heavy truck because the distribution of the drive torque is placed in the central distribution tube, as shown in Figure 1.

The solved center differential has position No.2 in the Figure 1, position No.1 is the drive shaft, position No.3 is the axial axle differential, positions No.4 and No.9 are bevel pinions, positions No.5 and No.8 are a bevel gears, positions No.6 and No.7 are half axle shafts and position No.10 is drive shaft of rear axle.

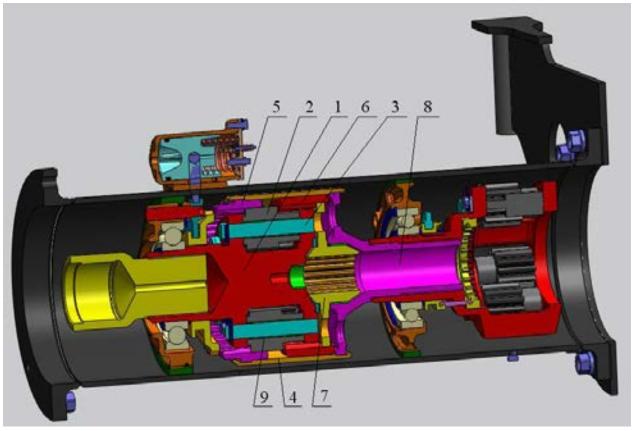

Figure 2. The basic center differential in the rear-axle tube

The basic internal centre differential is mounted in the carrier pipe and is part of the distribution of the drive torque to the axles. The differential divides the torque from the drive on the two rear axles. Unlike the classical solutions of the axles with bevel differential this is cylindrical type. It is designed as a planetary gearbox with spur gears with two internal ring gears (position No.5 and No.6 on Figure 2), ring gear (position No.4 on Figure 2) and eight planetary gears (position No.2 on Figure 2). The planetary gears are images with the internal gears (4 planetary gears are images to the internal ring gear of the third axle and 4 planetary gears are images to the internal ring gear of the fourth axle) and are also in meshing with each other. The planetary gears are rotationally mounted through the sliding sleeve (position No.9 on Figure 2) on pins (position No.3 on Figure 2). The pins are pressed into the gearbox (position No.1 on Figure 2). The driving torque enters the gearbox through involute castellated shaft. The torque is split over planet carriers (position No.7 and No.8 on Figure 2) transmitted to the two rear axles.

To determine the load centre differential cabinets it is necessary to know the size ballast torque input into the gearbox.

The calculation of onerous torque is for chosen type of the vehicle. The load is considered for off-state torque to the front axle and the distribution of torque to the rear axle with adhesion limits under the permissible load axle. This is a condition where the centre differential is blocked – the satellites are blocked. The calculation of the input torque of the differential gearbox was based on the maximum car engine torque.

On the basis of the calculated input torque, were subsequently calculated the forces in the gear centre differential and the strength of pins. Results of the load of gearbox are processed in Table 1.

Table 1. Results of the load of gearbox

Input data		
Maximum engine torque	M_{kMOT}	= 2100 Nm
Engine efficiency	η_M	= 0.92
Maximum gear ratio	i_p	= 7.655
Gear ratio of additional gearing	i_{RT}	= 2.91
Input torque on the gearbox	M_{kMM}	= 36582 Nm
Distribution of the axle	n	= 2
Torque on the ring gear	M_{kK}	= 18291 Nm
Number of planetary gears on the ring gear	i_S	= 4
Load of gearbox		
Circumferential force of gearing	F_t	= 46618 N
Force in line of contact of gearing	F_n	= 52311 N

3. Strength Calculation of the Differential

Strength calculation for gearbox by finite element method (FEM), is solved by SolidWorks in supplementary module for FEM calculations SolidWorks Simulation, as well as all model kit parts are modelled in this program.

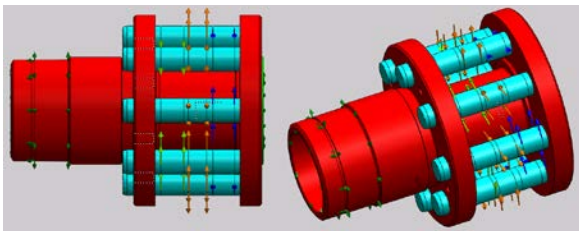

Figure 3. The simplified model

Model of arrangement for FEM stress analysis is composed of the differential gearbox, eight pins satellites and support ring. The geometric model of the gearbox than the actual gearbox is simplified (Figure 3), without involute splines of the drive shaft and gear sleeve. Pin model is compared to the actual pin is simplified. The computational model is no pin lock plate groove and without groove for lubrication. On the surface of the pin are plotted coverage gears the input forces. Diameter pin is modeled with maximum interference.

To the resulting computational model was necessary to define material properties, boundary conditions and power conditions, and define the type and size of the finite element. It is a linear static stress analysis of elastic stress.

In Figure 4 is the result of task solution by FEM. The greatest tension in the gearbox is in place notch, maximum voltage value 890MPa. When the strength of the material is 800MPa static safety factor 0.89, ie static security condition is not satisfied.

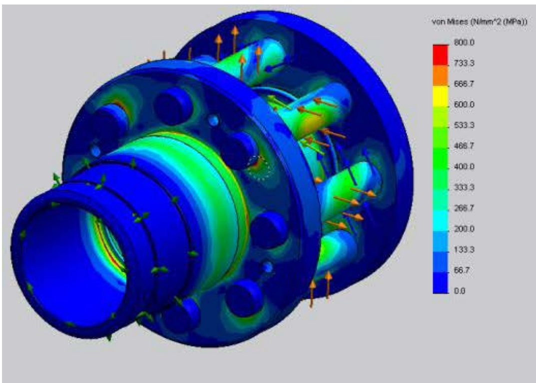

Figure 4. Result of FEM analysis

In Figure 5 are shown the locations where it is necessary to perform geometric optimization to meet the conditions of static security.

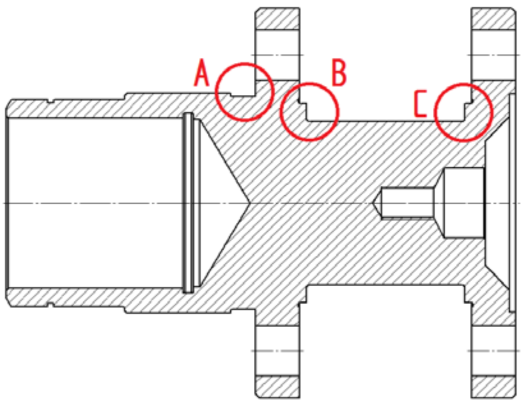

Figure 5. Designed of geometric optimization

Figure 6 is the designed solution at the point A, from Figure 5. In areas B and C are proposed adjustments due to the removal of structural strength horns that there have been proposed due to concerns that in these sites the material may be infringed.

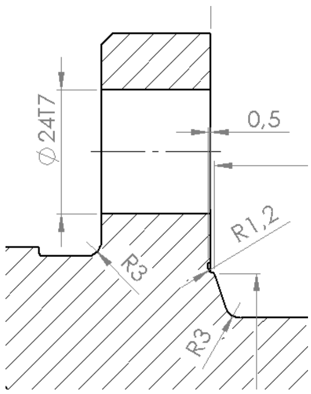

Figure 6. Proposed adjustment in the point A

Proposal of these changes is shown in Figure 7 where are chamfered the corners of structural strength at an angle of 105° and (corner radius R = 3 mm) transition and the resulting tapered cylindrical surface. After adjustment must be maintained the technology recess, which separates areas with different surface roughness and should be retained the space for free rotation of the satellites.

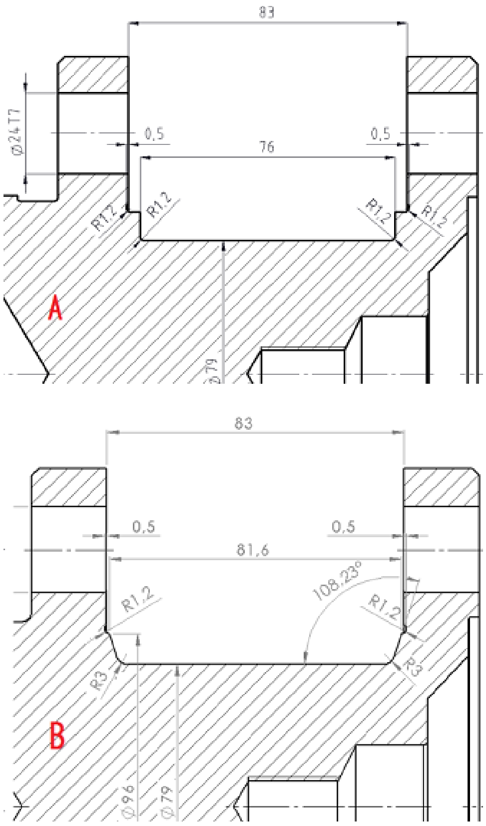

Figure 7. Designed geometrical adjustment of housing: A - original shape, B - designed shape

At the point of curvature at the head of the gearbox fell under the FEM analysis after adjusting of voltage gearbox the value of 890 MPa to 582.7 MPa, which is a voltage drop of about 34 % (Figure 8).

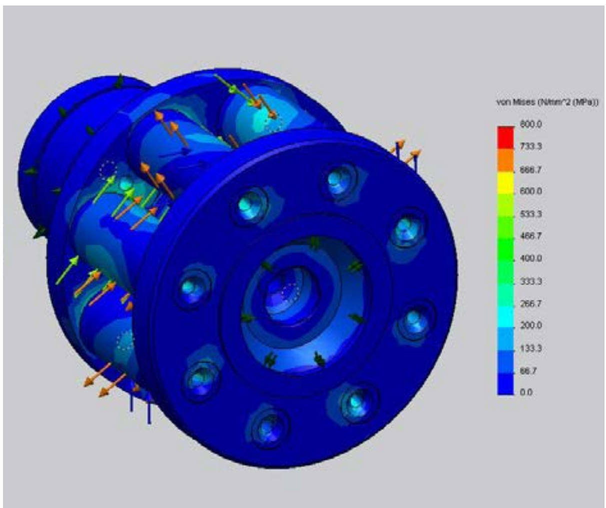

Figure 8. Result of FEM analysis after geometrical modification

At the place of arrangement between the faces of the gearbox (at the point C according to Figure 5) the maximum measured voltage value is 438 MPa (Figure 9). In FEM analysis on the original gearbox in this place the stress was 723.5 MPa. This means that at this point after adjusting the stress drop is by 40 %.

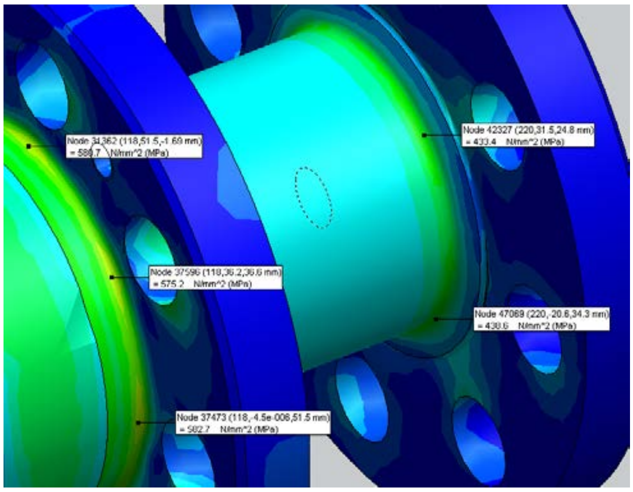

Figure 9. Stress in selected nodes after geometry optimization

Figure 10 shows the original geometry design of the lock plates. When designing fillet radius at the head (in place A from Figure 5) of the body is necessary to consider the position and shape of an lock plate.

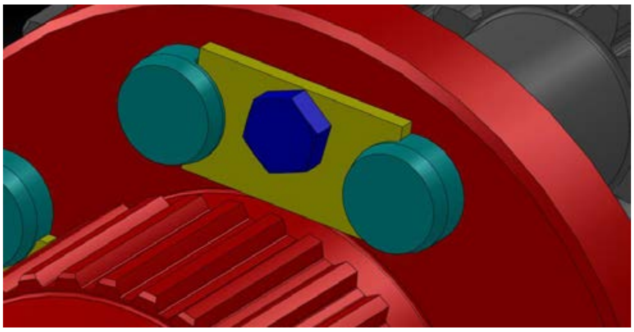

Figure 10. Original shape of lock plate

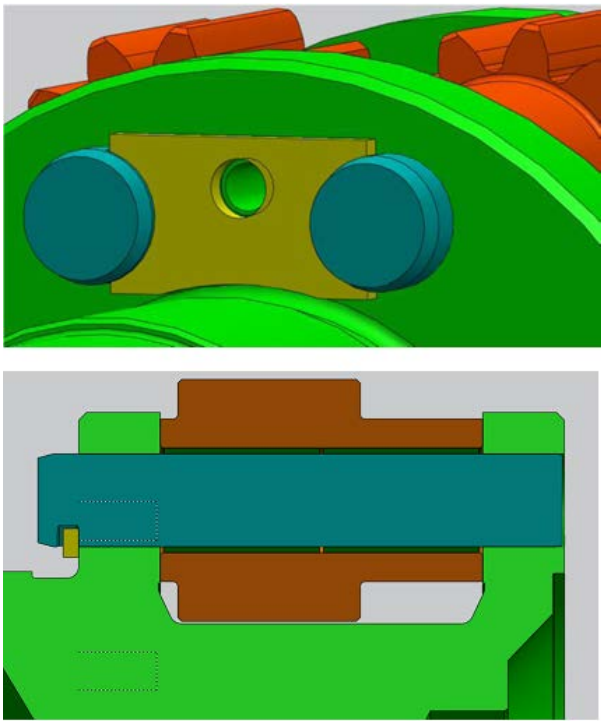

Figure 11. Designed geometrical modification of lock plate

If you change a fillet radius at the transition at the head, it must also change the geometric shape of plate for locking (Figure 11).

The static safety factor calculated after geometry optimization is 1.37. The static condition is met and the safety component complies. Compared with the previous calculation of the factor of safety, the value of the modified differential gearbox increased from value 0.89 to the value 1.37. It follows that the adjusted differential gearbox may be also used for larger loads than the loads calculated in this work.

4. Conclusion

After reading the results of strength FEM of original differential gearbox, there was found increased stress with a value exceeding the specified yield strength of the material gearbox in place of the notch at the head of the gearbox. FEM strength calculations were performed on simplified models for which it was established how the voltage decreases on torsional impact of increasing the radius of curvature. On this basis were proposed the necessary adjustments to the differential gearbox. By subsequent FEM analysis where evaluated the stresses in adjusted locations. It showed a drop in stress at the point of the notch at the front by 34%. Static safety coefficient increased from value 0.89 to the value 1.37. In the place of the front, in the space satellites the value of stress is decreased by 40 %. This adjusted gearbox is useful for larger loads.

Acknowledgement

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