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OPTIMISATION OF THE CENTRE DIFFERENTIAL FOR A HEAVY TRUCK

Summary. There are currently placed ever increasing demands on machine parameters. This are reflected in the growth dynamism load and continuously improve the speed of its parts. Therefore, the optimization of machine parameters is necessary to focus on increasing needs for performance, accuracy, durability and reliability of the machinery itself. The paper discusses optimizing center differential heavy truck wheel without 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 so as to reduce tensions. The basis for this optimization is the results of task solution by finite element method.
Keywords: gear transmission, stress, FEM.

OPTYMALIZACJA MIĘDZYOSIOWEGO MECHANIZMU RÓŻNICOWEGO SAMOCHODU CIĘŻAROWEGO

Streszczenie. Obecnie stawia się wciąż rosnące wymagania parametrom maszyn, co przejawia się wzrostem obciążenia dynamicznego i ciągłym zwiększaniem prędkości poszczególnych elementów, dlatego też podczas optymalizacji paramentów maszyn konieczne jest uwzględnianie rosnących wymagań dotyczących mocy, dokładności, żywotności i niezawodności samych maszyn. Artykuł dotyczy optymalizacji międzyosiowego mechanizmu różnicowego ciężkiego samochodu ciężarowego bez reduktora obrotów. Przez optymalizację w tym przypadku rozumie się modyfikację skrzyni mechanizmu różnicowego przeznaczonego dla tzw. lekkiego typu przekładni, tak aby zredukowała obciążenia z nowej tzw. przekładni ciężkiej, a więc poprawiła krytyczne miejsca ze zwiększonym naprężeniem oraz projekt modyfikacji skrzyni w celu zmniejszenia tych naprężeń. Podstawą przedstawionej optymalizacji były wyniki uzyskane podczas rozwiązywania zagadnienia za pomocą metody elementów skończonych.
Słowa kluczowe: przekładnia zębata, naprężenie, MES.

1. INTRODUCTION

The paper should have the optimalisation center differential for heavy truck with 8x8 wheel drive. Compared with the previous type of gear used is in the output increased torque

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output. This torque causes in the gearing of center differential more onerous force and the differential gearbox is more stressed. Proposed modifications of this component are based on the finite element analysis.

2. CONSTRUCTION OF CENTRE DIFFERENTIAL

The basic internal center differential is mounted in the carrier pipe and is part of the distribution of the drive torque to the axles. Differential divides torque from the drive on the two rear axles. Unlike classical solutions axle with bevel differential is this cylindrical type. It is designed as a planetary gearbox with spur gears with two internal ring gear (position No. 17 and No. 19 on the Fig. 1), ring gear (position No.16 on the Fig.1), eight planetary gears (position No. 2 on the Fig. 1). The planetary gears are images with the internal gears (4 planetary gears are images to the internal ring gear of third axle and 4 planetary gears are images to the internal ring gear of fourth axle.) and are also in meshing with each other. Planetary gears are rotational mounted through the sliding sleeve (position No. 26 on the Fig. 1) on pins (position No. 3 on the Fig. 1). Pins are pressed into the gearbox (position No.1 on the Fig. 1). Driving torque enters to the gearbox through involute castellated shaft. Torque is split over planet carriers (position No. 22 and No. 23 on the Fig. 1) transmitted to the two rear axles. In blocking planetary gears is identical rotation speed rear axle. The planetary gears will be roll over only in the adjust difference influence slipping of gears.

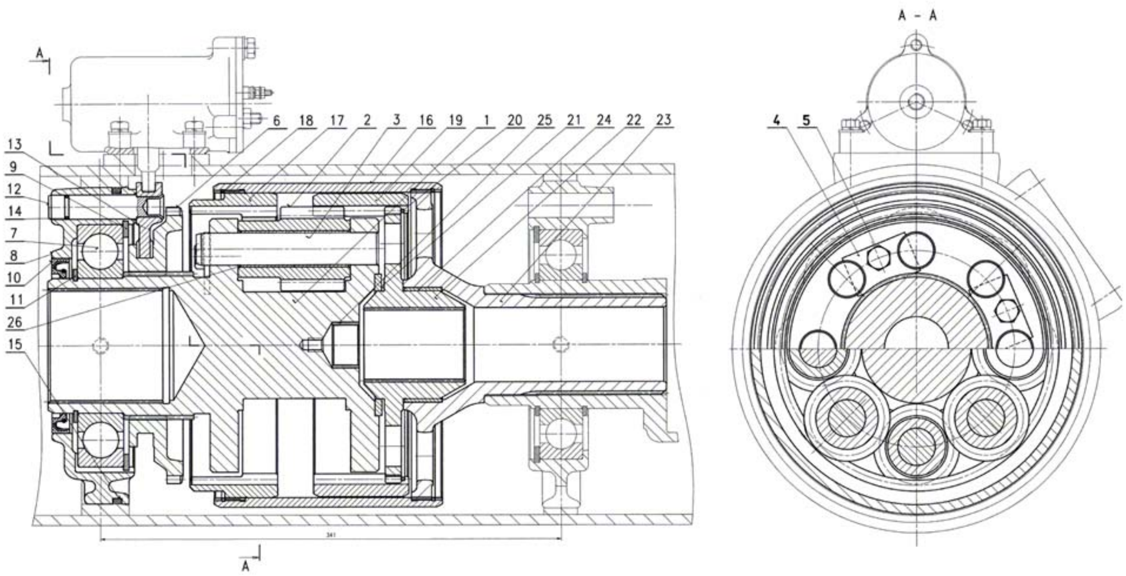

Fig. 1. Basic center differential
Rys. 1. Tylny międzyosiowy mechanizm różnicowy z kołem koronowym

3. LOAD OF CENTRE DIFFERENTIAL

To determine the load center differential cabinets need to know the size ballast torque input into the gearbox.

Calculation onerous torque M_k is for chosen to type the vehicle. Load is considered for off-state torque to the front axle and the distribution of torque to the rear axle adhesion limits under the permissible load axle. This is a condition where the center differential is locked - satellites are blocked. When calculating 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 forces in the gear center differential and strength of pins (Fig. 2).

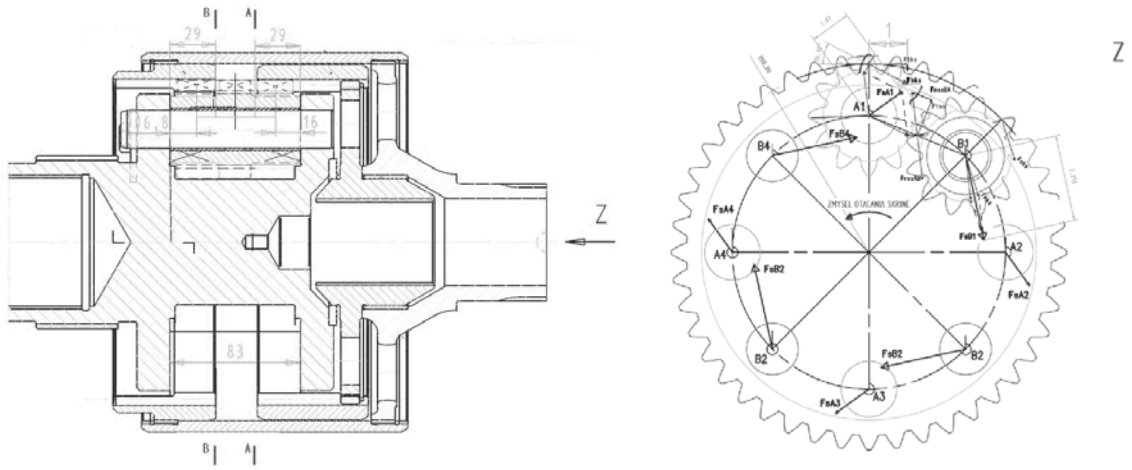

Fig. 2. Forces in the center differential
Rys. 2. Siły w międzyosiowym mechanizmie różnicowym

Results of the load of gearbox are processed in Table 1.

Table 1

Results of the load of gearbox

The 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	-
efficiency of gears	η_P	0,85	-
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	-
pressure angle	α	28	[°]
pitch pressure angle	α_W	26,98	[°]
pitch diameter	D_W	196,18	[mm]
load of gearbox			
circumferential force OBVODOVÁ	F_t	46618	[N]
force in line of contact	F_n	52311	[N]
the resulting force on the pin in the plane A	F_{sA}	49921	[N]
the resulting force on the pin in the plane B	F_{sB}	104140	[N]

4. STRENGTH CALCULATION OF THE DIFFERENTIAL GEARBOX BY FEM

Strength calculation for gearbox finite element method (FEM), is solved by SolidWorks 2009 in supplementary module for FEM calculations SolidWorks Simulation, as well as all model kit parts are modeled in this program.

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, 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 Fig. 3 is the result of task solution by FEM. The greatest tension in the gearbox is in place notch, maximum voltage value 890 MPa. When the strength of the material is 800 MPa static safety factor 0.89, ie static security condition is not satisfied.

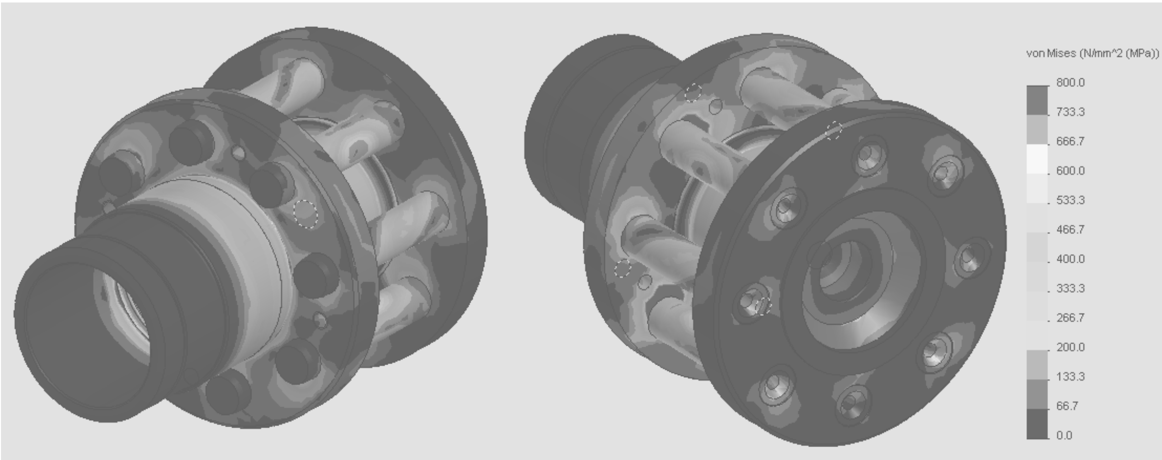

Fig. 3. Result of FEM analysis
Rys. 3. Wynik obliczeń MES

In Fig. 4 shows the locations where it is necessary to perform geometric optimization to meet the conditions of static security. Figure 5 is suggested solution at the point A.

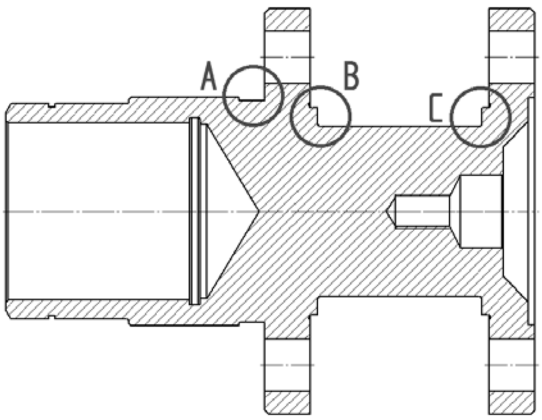

Fig. 4. Places for geometric optimization
Rys. 4. Miejsca optymalizacji geometrycznej

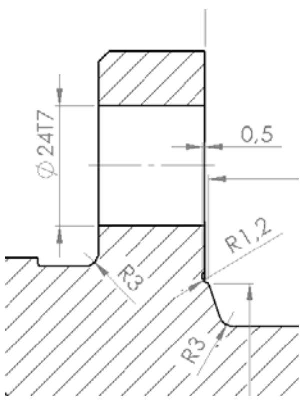

Fig. 5. The proposed adjustment in the A
Rys. 5. Projekt modyfikacji w miejscu A

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 these sites may be infringed material. Proposal of these changes in Figure 6 chamfered corners 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 Technology recess, which separates areas with different surface roughness and should retain space for free rotation of the satellites.

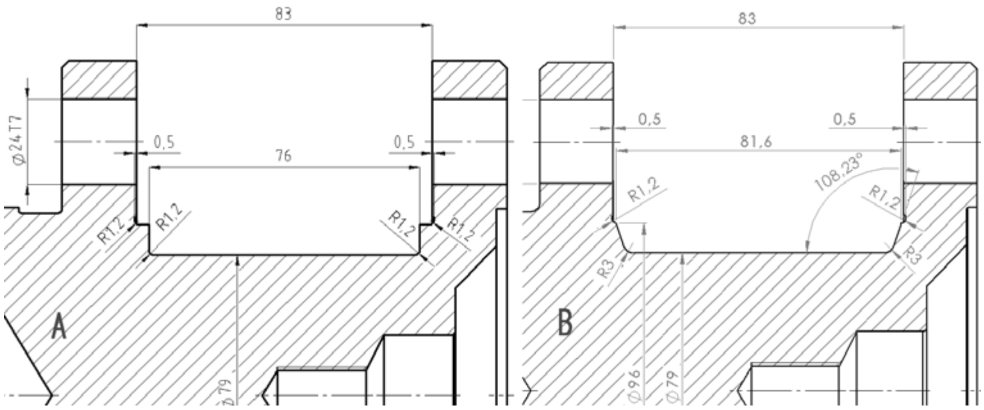

Fig. 6. The proposed adjustment of housing: A - original shape, B - the proposed shape
Rys. 6. Projekt modyfikacji skrzyni: A - kształt pierwotny B - proponowany kształt

At the point of curvature at the head of the gearbox fell under the FEM analysis after adjusting voltage gearbox value of 890 MPa to 582.7 MPa, which is a voltage drop of about 34%. The place arrangement between the face of the gearbox (at the point C according to Figure 4) is the maximum measured voltage value 438 MPa (fig.7). In FEM analysis on the original gearbox was in this place stress 723.5 MPa. This means that at this point it is after adjusting the stress drop by 40%.

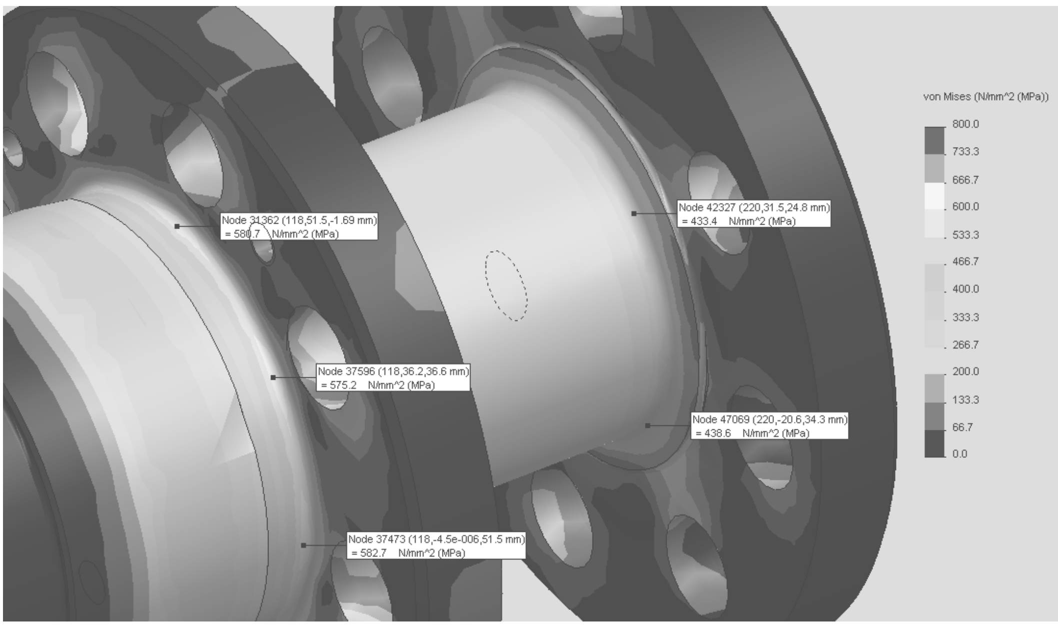

Fig. 7. Stress in selected nodes after geometry optimization
Rys. 7. Napięcie w wybranych węzłach po optymalizacji geometrycznej

Static safety factor is calculated after geometry optimization is 1,37. 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 on the value 1,37. It follows that the adjusted differential gearbox may also be used for larger loads than the loads calculated in this work.

5. CONCLUSION

After reading the results of strength FEM original differential gearbox, cause increased stress, a value exceeding the specified yield strength of the material gearbox in place of the notch at the head of the gearbox. Strength were performed FEM calculations on simplified models for which it was established as the voltage decreases on torsional impact of increasing the radius of curvature. On this basis, propose the necessary adjustments to the differential gearbox. Of subsequent FEM analysis were by evaluated stress in adjusted locations. It showed a drop in stress at the point of the notch at the front is by 34%. Static safety coefficient increased from value 0,89 to the value 1,37. In place of the front, in the space satellites value of stress is decreased by 40%. This adjusted gearbox is useful for larger loads.

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