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TOOTH CONTACT ANALYSIS OF A DESIGNED PLANETARY GEAR DRIVE FOR THE VEHICLE INDUSTRY

Abstract: *A planetary gear drive consists of a sun gear, planet pinions and an internal gear. We designed a complex gear system which is usable in the field of the vehicle industry into the automatized robots. The system was designed by GearTeq software which is connected with the SolidWorks designer software. After the assembly and the motion simulations tooth contact analysis (TCA) was made to analyse the normal stresses and the normal deformations on the connecting surface of the planet pinions and the internal gear by different load moments.*
Key words: *Planetary gear drive, CAD, TCA, normal stress, normal deformation, analysis.*

Analiza kontakta zuba projektovanog planetarnog zupčanika za industriju vozila. *Pogon planetarnog zupčanika sastoji se od sunčanog zupčanika, planetarnih zupčanika i unutrašnjeg zupčanika. U automatizovane robote dizajnirali smo složen sistem zupčanika koji je upotrebljiv u oblasti industrije vozila. Sistem je dizajnirao GearTek softver koji je povezan sa SolidVorks dizajnerskim softverom. Nakon montaže i simulacije kretanja izvršena je analiza kontakta zuba (TCA) za analizu normalnih napona i normalnih deformacija na spojnoj površini planetarnih zupčanika i unutrašnjeg zupčanika po različitim momentima opterećenja.*
Ključne reči: *Planetarni zupčanik, CAD, TCA, normalno naprezanje, normalna deformacija, analiza.*

1. INTRODUCTION

The planetary gear drives have two gear systems. The axis of the first system is fixed where the planet gears can rotate around it. The planet carrier can also rotate around it. The axes of the second system are assembled into the planet carrier and their teeth can connect with the first system. These planet pinions can rotate around their axes and the fixed axes of the first system [1, 3-5, 7, 9]. The overall mechanism show a similar motion as the Earth moves around the Sun (two rotation motions around two axes). The sun gear is the central gear which has a fix axes. The planet gears can do two rotation motions parallely. The internal gear is fixed. The planet gears are rotated by the sun gear and they are connected with the internal gear (Figure 1) [1, 3-5, 7, 9].

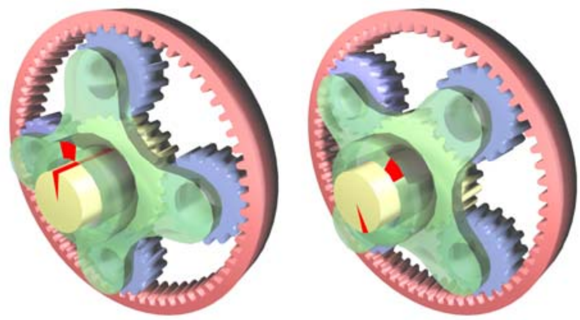


Fig. 1. The theorem of the planetary gear drive [1]

Considering the function of the gear system the sun gear can be pinion or gear. The planet pinions can also be pinions or gears. The yellow and green axes are not connected. The connection between them depends on the gear ratio (Figure 1) [1, 3-5, 7, 9].

2. THE GEOMETRIC DESIGN OF A GEAR SYSTEM

The geometric design process [3, 6-9] was created by the GearTeq software [2] with which different type of gear pairs can be designed (Figure 2). After knowing of the output geometric parameters the CAD models can be created by SolidWorks software (Figure 3).

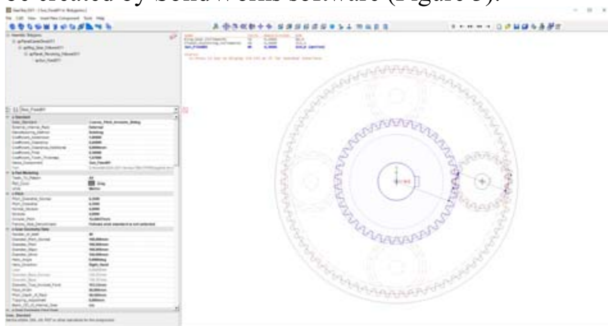


Fig. 2. Geometric design by GearTeq software

Fig. 3. The geometric establishment of the designed planetary gear drive

The calculated geometric parameters can be seen on Table 1, 2 and 3. After the assembly and the motion simulations the TCA can be determined.

SYMBOL	VALUE	UNIT	TERM
	Coarse_Pitch_Involute_20deg		Standard
Pdn	6,35		Normal Diametral Pitch
Pd	6,35		Diametral Pitch
	4		Normal Modular Pitch
m	4		Modular Pitch
øn	20	deg	Normal Pressure Angle
ø	20	deg	Pressure Angle
	0	deg	Helix Angle
	Gear Data		Ring_Gear_Follower01
Np	76		Number of Teeth
Dp	304	mm	Pitch Diameter
Dpn	304	mm	Pitch Diameter, Normal
do	314	mm	Major Diameter
dr	296	mm	Minor Diameter
a	4	mm	Addendum
b	5	mm	Dedendum
x	0		Addendum Modification Coefficient
	0	mm	Addendum Modification
db	285,667	mm	Base Diameter
dbn	285,667	mm	Base Diameter, Normal
TIF	312,317	mm	True Involute Form Diameter
ht	9	mm	Whole Depth
p	12,566	mm	Circular Pitch
pn	12,566	mm	Circular Pitch, Normal
	1,2	mm	Fillet Radius
B	0,4	mm	Backlash
	6,6832	mm	Space Width
	6,8352	mm	Space Width Maximum
t	5,8832	mm	Tooth Thickness
tn	5,8832	mm	Tooth Thickness, Normal
t	5,7312	mm	Tooth Thickness Minimum
F	50	mm	Face Width
			Chordal Tooth Thickness
	0,869		Chordal Tooth Height
	312,317		Chordal Tooth Reference Circle
	2,6396		Chordal Tooth Thickness
	2,4835		Chordal Tooth Thickness Minimum
			Size Between Pins
dw	5,225	mm	Pin Diameter
M	302,141	mm	Measurement Between Pins
	301,753	mm	Measurement Between Pins-Minimum
			Span Over Teeth
k	0		Number of Teeth to Span Over
	-1,271	mm	Span Measurement
	-1,128	mm	Span Measurement Minimum
			Master Gear Test
	0		Master Pitch Diameter
	0	mm	Test Radius (Max. Act.)
	0	mm	Test Radius (Min. Act.)
	AGMA-Q7		AGMA Quality Class
	0,1524	mm	Max Runout
	0,0381	mm	Pitch Variation
	0,0508	mm	Profile Tolerance
	0	mm	Tooth Alignment Tolerance
	0,05842	mm	Tooth to Tooth Composite Tolerance
	0,21336	mm	Total Composite Tolerance
	0,152	mm	Tooth Thickness Tolerance
	0	mm	Hob Protuberance
	26,14	deg	Roll Angle at Major Diameter
	25,32	deg	Roll Angle at TIF Diameter

Table 1. Geometric parameters of the internal gear

SYMBOL	VALUE	UNIT	TERM
	Coarse_Pitch_Involute_20deg		Standard
Pdn	6,35		Normal Diametral Pitch
Pd	6,35		Diametral Pitch
	4		Normal Modular Pitch
m	4		Modular Pitch
øn	20	deg	Normal Pressure Angle
ø	20	deg	Pressure Angle
	0	deg	Helix Angle
mg	0,237		Ratio, 1:x
C	116	mm	Center Distance
	0	mm	Center Distance Extension
	0	mm	Center Distance Backlash
MA	13,223	mm	Approach Length
MR	9,032	mm	Recess Length
mp	1,885		Contact Ratio

	Not Hunting		Hunting Determination
	4560		Hunting Mesh Cycle
	1, 2		Hunting Common Factors
	6.7cpm		Hunting Tooth Frequency
	253,3		Pinion RPM
	Gear Data		Ring_Gear_Follower01
Np	76		Number of Teeth
Dp	304	mm	Pitch Diameter
Dpn	304	mm	Pitch Diameter, Normal
do	314	mm	Major Diameter
dr	296	mm	Minor Diameter
a	4	mm	Addendum
b	5	mm	Dedendum
x	0		Addendum Modification Coefficient
	0	mm	Addendum Modification
db	285,667	mm	Base Diameter
dbn	285,667	mm	Base Diameter, Normal
TIF	312,317	mm	True Involute Form Diameter
ht	9	mm	Whole Depth
p	12,566	mm	Circular Pitch
pn	12,566	mm	Circular Pitch, Normal
	1,2	mm	Fillet Radius
B	0,4	mm	Backlash
	6,6832	mm	Space Width
	6,8352	mm	Space Width Maximum
t	5,8832	mm	Tooth Thickness
tn	5,8832	mm	Tooth Thickness, Normal
t	5,7312	mm	Tooth Thickness Minimum
F	50	mm	Face Width
			Chordal Tooth Thickness
	0,869		Chordal Tooth Height
	312,317		Chordal Tooth Reference Circle
	2,6396		Chordal Tooth Thickness
	2,4835		Chordal Tooth Thickness Minimum
			Size Between Pins
dw	5,225	mm	Pin Diameter
M	302,141	mm	Measurement Between Pins
	301,753	mm	Measurement Between Pins-Minimum
			Span Over Teeth
k	0		Number of Teeth to Span Over
	-1,271	mm	Span Measurement
	-1,128	mm	Span Measurement Minimum
			Master Gear Test
	0		Master Pitch Diameter
	0	mm	Test Radius (Max. Act.)
	0	mm	Test Radius (Min. Act.)
	AGMA-Q7		AGMA Quality Class
	0,1524	mm	Max Runout
	0,0381	mm	Pitch Variation
	0,0508	mm	Profile Tolerance
	0	mm	Tooth Alignment Tolerance
	0,05842	mm	Tooth to Tooth Composite Tolerance
	0,21336	mm	Total Composite Tolerance
	0,152	mm	Tooth Thickness Tolerance
	0	mm	Hob Protuberance
	26,14	deg	Roll Angle at Major Diameter
	25,32	deg	Roll Angle at TIF Diameter
	Pinion Data		Planet_Revolving_Follower01
Np	18		Number of Teeth
Dp	72	mm	Pitch Diameter
Dpn	72	mm	Pitch Diameter, Normal
do	80	mm	Major Diameter
dr	62	mm	Minor Diameter
a	4	mm	Addendum
b	5	mm	Dedendum
x	0		Addendum Modification Coefficient
	0	mm	Addendum Modification
db	67,658	mm	Base Diameter
dbn	67,658	mm	Base Diameter, Normal
TIF	67,658	mm	True Involute Form Diameter
ht	9	mm	Whole Depth
p	12,566	mm	Circular Pitch
pn	12,566	mm	Circular Pitch, Normal
	1,2	mm	Fillet Radius
B	0,4	mm	Backlash
t	5,8832	mm	Tooth Thickness
tn	5,8832	mm	Tooth Thickness, Normal
F	50	mm	Face Width
			Chordal Tooth Thickness
	6,299		Chordal Tooth Height
	67,658		Chordal Tooth Reference Circle
	6,5266		Chordal Tooth Thickness
	6,3844		Chordal Tooth Thickness Minimum
			Size Over Pins
dw	6,967	mm	Pin Diameter

M	80,811	mm	Measurement Over Pins
	80,449	mm	Measurement Over Pins-Minimum
			Span Over Teeth
k	0		Number of Teeth to Span Over
	-5,272	mm	Span Measurement
	-5,415	mm	Span Measurement Minimum
			Master Gear Test
	0		Master Pitch Diameter
	0	mm	Test Radius (Max. Act.)
	0	mm	Test Radius (Min. Act.)
	AGMA-Q7		AGMA Quality Class
	0,10922	mm	Max Runout
	0,03048	mm	Pitch Variation
	0,04064	mm	Profile Tolerance
	0	mm	Tooth Alignment Tolerance
	0,0635	mm	Tooth to Tooth Composite Tolerance
	0,17272	mm	Total Composite Tolerance
	0,152	mm	Tooth Thickness Tolerance
	0	mm	Hob Protuberance
	36,15	deg	Roll Angle at Major Diameter
	0	deg	Roll Angle at TIF Diameter

Table 2. Geometric parameters of the planet pinions

SYMBOL	VALUE	UNIT	TERM
	Coarse_Pitch_Involute_20deg		Standard
Pdn	6,35		Normal Diametral Pitch
Pd	6,35		Diametral Pitch
	4		Normal Modular Pitch
m	4		Modular Pitch
øn	20	deg	Normal Pressure Angle
ø	20	deg	Pressure Angle
	0	deg	Helix Angle
mg	0,45		Ratio, 1:x
C	116	mm	Center Distance
	0	mm	Center Distance Extension
	0	mm	Center Distance Backlash
MA	9,032	mm	Approach Length
MR	10,117	mm	Recess Length
mp	1,622		Contact Ratio
	Not Hunting		Hunting Determination
	2052		Hunting Mesh Cycle
	1, 2		Hunting Common Factors
	5.7cpm		Hunting Tooth Frequency
	114		Pinion RPM
	Gear Data		Sun_Fixed01
Np	40		Number of Teeth
Dp	160	mm	Pitch Diameter
Dpn	160	mm	Pitch Diameter, Normal
do	168	mm	Major Diameter
dr	150	mm	Minor Diameter
a	4	mm	Addendum
b	5	mm	Dedendum
x	0		Addendum Modification Coefficient
	0	mm	Addendum Modification
db	150,351	mm	Base Diameter
dbn	150,351	mm	Base Diameter, Normal
TIF	153,335	mm	True Involute Form Diameter
ht	9	mm	Whole Depth
p	12,566	mm	Circular Pitch
pn	12,566	mm	Circular Pitch, Normal
	1,2	mm	Fillet Radius
B	0,4	mm	Backlash
t	5,8832	mm	Tooth Thickness
tn	5,8832	mm	Tooth Thickness, Normal
t	5,7312	mm	Tooth Thickness Minimum
F	50	mm	Face Width
			Chordal Tooth Thickness
	7,389		Chordal Tooth Height
	153,335		Chordal Tooth Reference Circle
	7,5198		Chordal Tooth Thickness
	7,3743		Chordal Tooth Thickness Minimum
			Size Over Pins
dw	6,967	mm	Pin Diameter
M	168,891	mm	Measurement Over Pins
	168,503	mm	Measurement Over Pins-Minimum
			Span Over Teeth
k	0		Number of Teeth to Span Over
	-4,039	mm	Span Measurement
	-4,182	mm	Span Measurement Minimum
			Master Gear Test
	0		Master Pitch Diameter
	0	mm	Test Radius (Max. Act.)
	0	mm	Test Radius (Min. Act.)
	AGMA-Q7		AGMA Quality Class
	0,13208	mm	Max Runout
	0,03556	mm	Pitch Variation
	0,04572	mm	Profile Tolerance
	0	mm	Tooth Alignment Tolerance
	0,05842	mm	Tooth to Tooth Composite Tolerance
	0,18796	mm	Total Composite Tolerance
	0,152	mm	Tooth Thickness Tolerance
	0	mm	Hob Protuberance
	28,56	deg	Roll Angle at Major Diameter
	11,47	deg	Roll Angle at TIF Diameter
	Pinion Data		Planet_Revolving_Follower01
Np	18		Number of Teeth
Dp	72	mm	Pitch Diameter
Dpn	72	mm	Pitch Diameter, Normal
do	80	mm	Major Diameter
dr	62	mm	Minor Diameter
a	4	mm	Addendum
b	5	mm	Dedendum
x	0		Addendum Modification Coefficient
	0	mm	Addendum Modification
db	67,658	mm	Base Diameter
dbn	67,658	mm	Base Diameter, Normal
TIF	67,658	mm	True Involute Form Diameter
ht	9	mm	Whole Depth
p	12,566	mm	Circular Pitch
pn	12,566	mm	Circular Pitch, Normal
	1,2	mm	Fillet Radius
B	0,4	mm	Backlash
t	5,8832	mm	Tooth Thickness
tn	5,8832	mm	Tooth Thickness, Normal
F	50	mm	Face Width
			Chordal Tooth Thickness
	6,299		Chordal Tooth Height
	67,658		Chordal Tooth Reference Circle
	6,5266		Chordal Tooth Thickness
	6,3844		Chordal Tooth Thickness Minimum
			Size Over Pins
dw	6,967	mm	Pin Diameter
M	80,811	mm	Measurement Over Pins
	80,449	mm	Measurement Over Pins-Minimum
			Span Over Teeth
k	0		Number of Teeth to Span Over
	-5,272	mm	Span Measurement
	-5,415	mm	Span Measurement Minimum
			Master Gear Test
	0		Master Pitch Diameter
	0	mm	Test Radius (Max. Act.)
	0	mm	Test Radius (Min. Act.)
	AGMA-Q7		AGMA Quality Class
	0,10922	mm	Max Runout
	0,03048	mm	Pitch Variation
	0,04064	mm	Profile Tolerance
	0	mm	Tooth Alignment Tolerance
	0,0635	mm	Tooth to Tooth Composite Tolerance
	0,17272	mm	Total Composite Tolerance
	0,152	mm	Tooth Thickness Tolerance
	0	mm	Hob Protuberance
	36,15	deg	Roll Angle at Major Diameter
	0	deg	Roll Angle at TIF Diameter

Table 3. Geometric parameters of the sun gear

3. TOOTH CONTACT ANALYSIS

The aim of the TCA is to determine and analyse the mechanical parameters into the tooth connection zone by different loads [3, 4]. In our establishment, the sun gear is the pinion that is why it was loaded by different moments. The gear materials are steel (E=210 GPa, ν=0.3, isotropic elasticity).

Coordinate systems are defined into the rotation axes of the gears and the contact zones between the teeth.

The mesh method is tetrahedrons. Body of influence sizing type is defined into the contact zone to enhance the accuracy of the calculation process. The element size is 0.4 mm into the contact zone.

3.1. TCA between the sun gear and the planet pinion

The sun gear is loaded by different moments (40 – 80 Nm, step: 10 Nm). The effect of the load moment is analyzed on the tooth surface of the planet pinion. The mesh distribution can be seen on Figure 4.

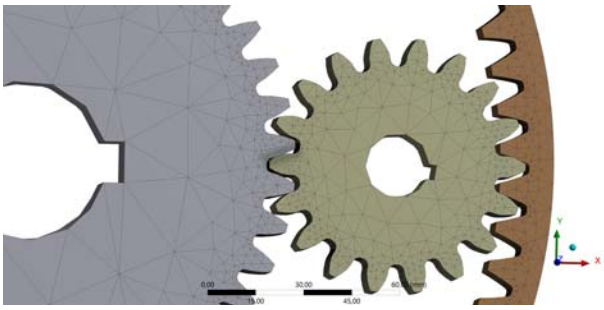
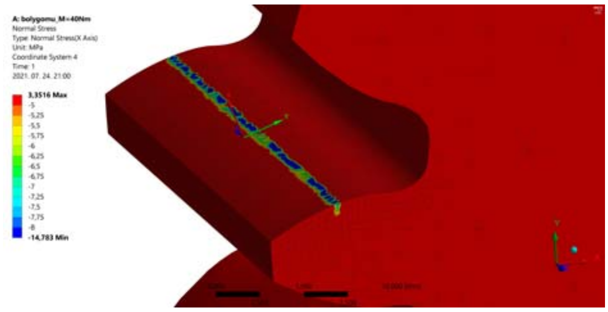
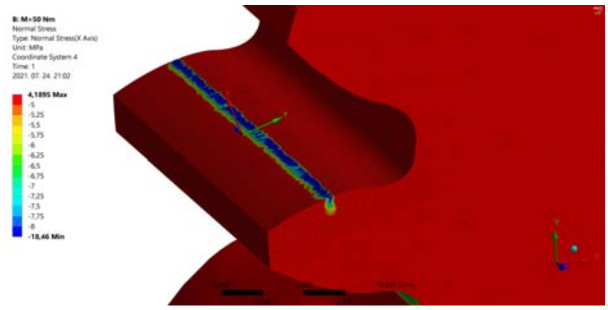


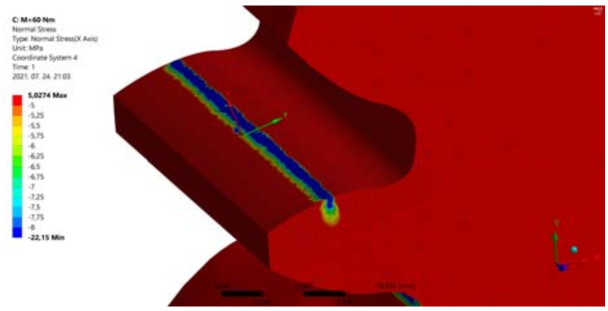
Fig. 4. The mesh for connection analysis between the sun gear and the planet pinion



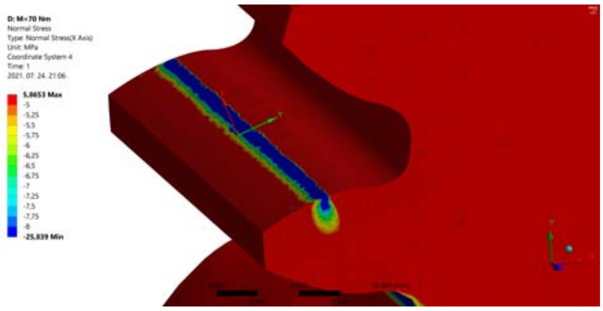
a) M=40 Nm



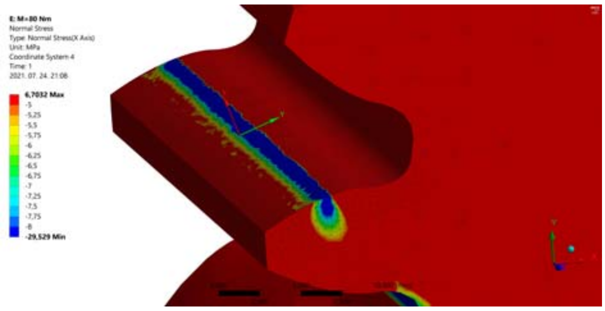
b) M=50 Nm



c) M=60 Nm



d) M=70 Nm



e) M=80 Nm

Fig. 5. The distribution of the normal stress on the surface of the planet pinion

The results of the normal stress on the tooth surfaces of the planet pinions can be seen on Figure 5.

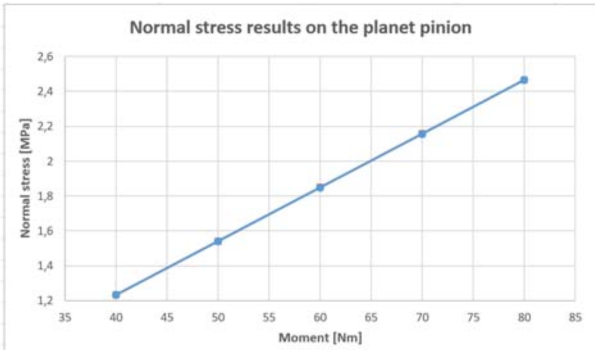
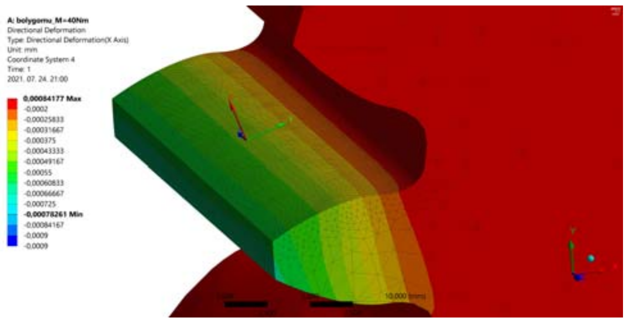


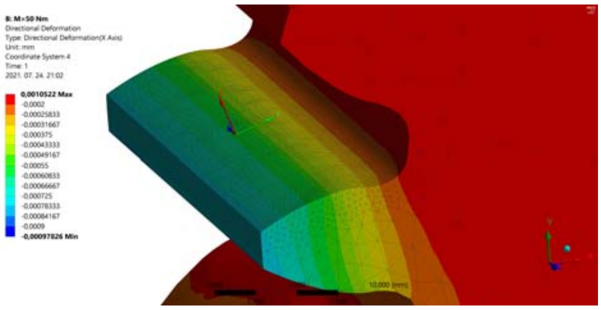
Fig. 6. The results of the normal stress in the funtion of the moment on the surface of the planet pinion

The results of the average normal stresses in the function of the moment can be seen on Figure 6. The more the load moment, the more the normal stress on the tooth surface of the planet pinion.

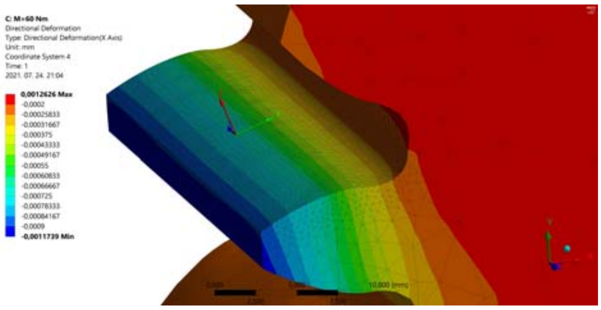
The results of the normal deformations into the ‘x‘ direction on the tooth surfaces of the planet pinion can be seen on Figure 7.



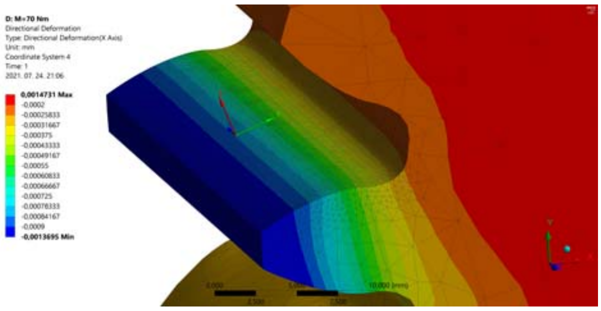
a) M=40 Nm



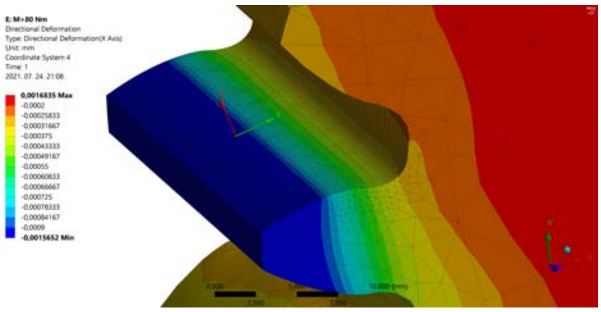
b) M=50 Nm



c) M=60 Nm



d) M=70 Nm



e) M=80 Nm

Fig. 7. The distribution of the normal deformation on the surface of the planet pinion

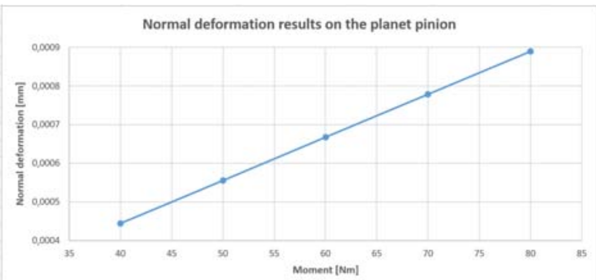


Fig. 8. The results of the normal deformation in the funtion of the moment on the surface of the planet pinion

The results of the average normal deformations in the function of the moment can be seen on Figure 8. The more the load moment, the more the normal deformation on the tooth surface of the planet pinion.

3.2. TCA between the planet pinion and the internal gear

Considering the gear ratio between the sun gear and the planet pinions, the moments have to be recalculated for the planet pinions since these gears are connected with the internal gear. The calculated moments can be seen on Table 4.

Sun gear	Planet pinions
50 Nm	18 Nm
60 Nm	22.5 Nm
70 Nm	27 Nm
80 Nm	31.5 Nm
90 Nm	36 Nm

Table 4. The moments on the sun gear and the planet pinions accordingly the gear ratio

The effect of the load moment is analysed on the surface of the internal gear. The meshing strategy is similar than the previous case (Figure 9).

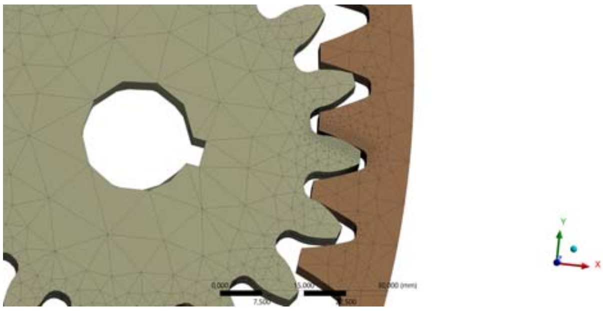
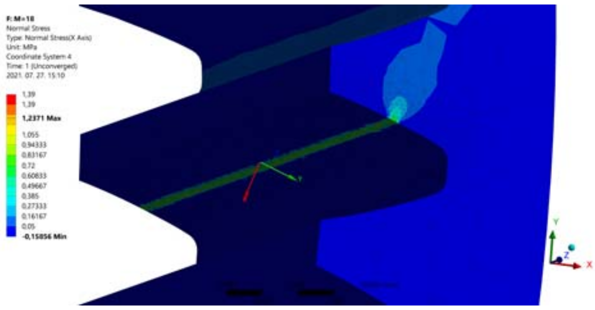
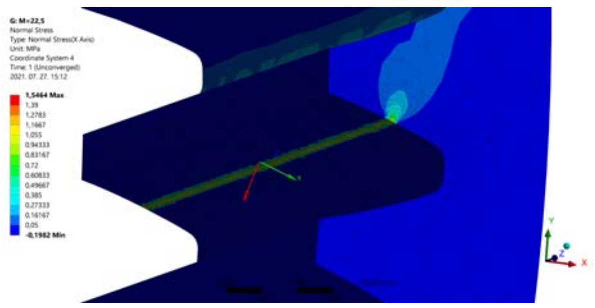


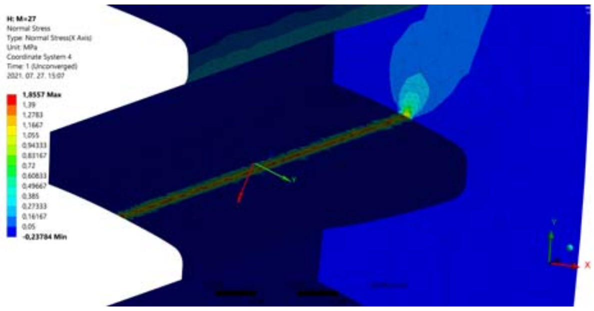
Fig. 9. The mesh for connection analysis between the planet pinion and the internal gear



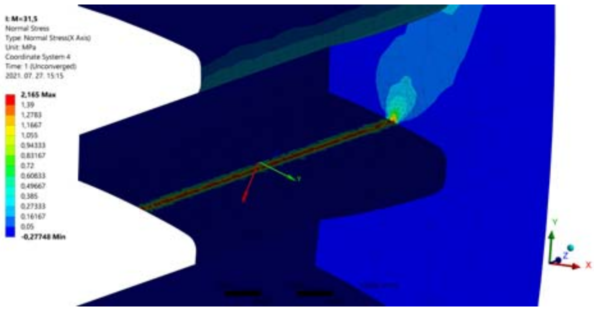
a) M=18 Nm



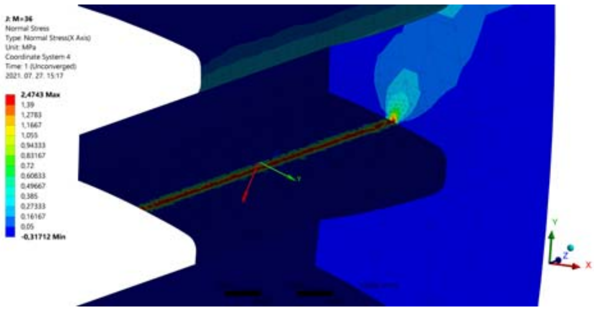
b) M=22.5 Nm



c) M=27 Nm



d) M=31.5 Nm



e) M=36 Nm

Fig. 10. The distribution of the normal stress on the surface of the internal gear

The results of the normal stress on the tooth surfaces of the internal gear can be seen on Figure 10.

The results of the average normal stresses on the tooth surface of the internal gear in the function of the moment can be seen on Figure 11. We got lower stress values since the load moments were lower due to the gear ratio. It is also true the stress is higher if we increase the moment.

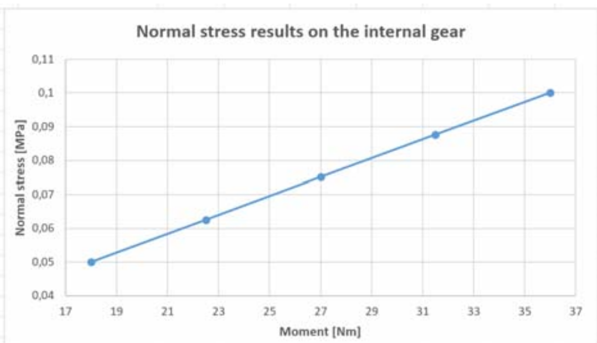
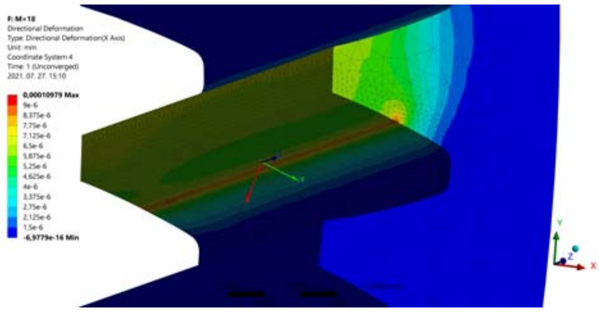
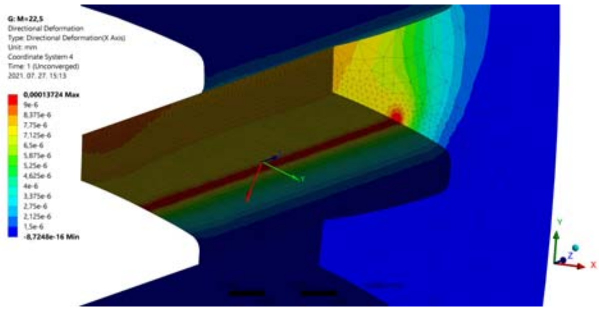


Fig. 11. The results of the normal stress in the funtion of the moment on the surface of the planet pinion

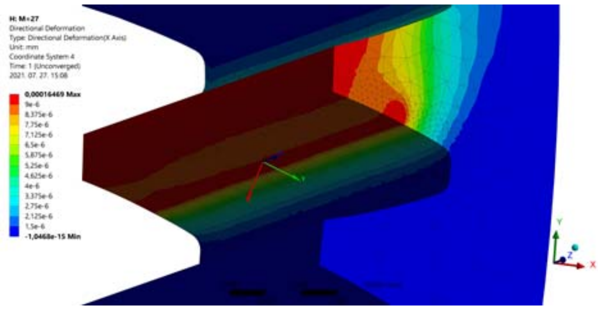
The results of the normal deformations (‘x’ directional) on the tooth surfaces of the internal gear can be seen on Figure 12.



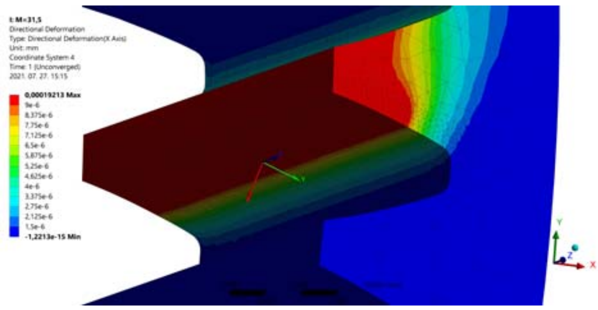
a) M=18 Nm



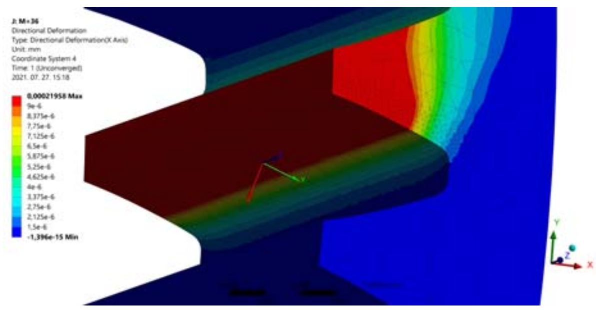
b) M=22.5 Nm



c) M=27 N



d) M=31.5 Nm



e) M=36 Nm

Fig. 12. The distribution of the normal deformation on the surface of the planet pinion

The results of the average normal deformations in the function of the moment can be seen on Figure 12. We got much lower results than in case of the previous analysis. The reason is the gear ratio, the lower moment and the mass. It is also true that increasing the load moment on the planet pinion the normal deformation is also increasing on the tooth surface of the internal gear.

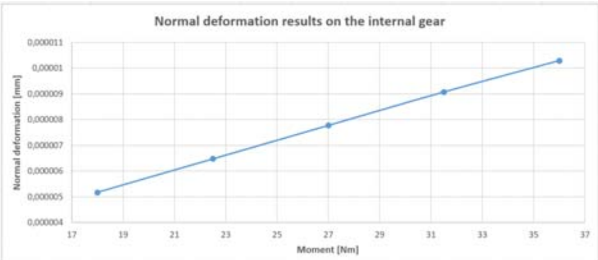


Fig. 13. The results of the normal deformation in the funtion of the moment on the surface of the internal gear

4. CONCLUSION

The vehicle industry is a big filed in the countries that contains two huge fields: vehicle design and vehicle manufacturing. There are more and more vehicles on the roads, consequently the development and the research on this field is actual.

In this study, we designed a complex planetary gear box which is usable in the robotic systems for the vehicle manufacturing.

The geometric parameters was calculated by the help with the GearTeq software. After that, the results could be imported into the SolidWorks three dimensional designer software where the assembly and the motion analysis could be done.

The aim of the TCA is to analyze the mechanical parameters into the tooth connection zone of the gear pairs by different loads. In our case, the load was the moment on the pinions. Firstly, we analyzed the TCA parameters between the sun gear and the planet pinion. Four planet pinions were used around the perimeter of the sun gear. Secondly, we analyzed the same parameters between the planet pinion and the internal gear. In this case, the moments had to be recalculated accordingly the gear ratio from the sun gear, which is the pinion, to the planet pinions, which are intermediary gears. We made diagrams from the results and evaluated the overall analysis. This analysis process is necessary to control the correctness and the function of such gear systems before the real installation into the machines [8].

5. REFERENCES

[1] https://hu.wikipedia.org/wiki/Bolyg%C3%B3m%C5%B1

[2] Bodzás, S., Békési, Zs., Ketész, J., Szorcsik, T.: The CAD modelling possibilities of the GearTeq 2021 software in the mechanical engineering practice, International Journal of Engineering and Management Sciences, Debrecen, 2021, (under review)

[3] Litvin, F. L., Fuentes, A. A.: Gear Geometry and Applied Theory, Cambridge University Press, 2004, ISBN 978 0 521 81517 8

[4] Litvin, F. L., Fuentes. A. A., Vecchiato, D., Gonzalez-Perez, I.: New Design and Improvement of Planetary Gear Trains, NASA Center for Aerospace Information, 2004, p. 32, https://ntrs.nasa.gov/api/citations/20040086788/downloads/20040086788.pdf

[5] Goldfarb, V., Trubachev, E., Barmina, N.: Advanced Gear Engineering, Springer, 2018, p. 197., ISBN 978-3-319-60398-8

[6] Rackov, M., Knežević, I., Čavić, M., Penčić, M., Čavić, D., Kuzmanović, S. Design Solutions Overview of Universal Motor Gear Drives with Helical Gears, 10th International Symposium on Graphic Engineering and Design (GRID 2020), Novi Sad, Serbia, 12-14 November 2020, pp. 597–608, https://doi.org/10.24867/GRID-2020-p68

[7] Radzevich, S. P.: Dudley's Handbook of Practical Gear Design and Manufacture, Third edition, CRC Press, 2016, p. 656, ISBN 9781498753104

[8] Golebski, R., Boral, P.: Study of Machining of Gears with Regular and Modified Outline Using CNC Machine Tools, Materials 14 (11): 2913, 2021, DOI:10.3390/ma14112913

[9] Terplán, Z., Apró, F., Antal, M., Döbröczöni, Á.: Fogaskerék-bolygóművek, Műszaki könyvkiadó, Budapest, 1979, p. 258

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