

Tooth contact analysis of x-zero helical gears by the modification of the tooth trace

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Abstract

The helical gear pairs are widely used in different mechanical constructs. The main advantage is the good efficiency and the low noise during the connection.

We have worked out a computer software with which the designing process of the helical gears could be eased. Based on the calculated gear parameters the CAD models of the elements could be prepared. Knowing of the geometries the assembly could be created based on the fixed freedom degrees and the elementary centre distance. We have designed five types of gear pairs. The differences are only the different tooth traces. All of other parameters have not been modified.

The aim of this research is the function between the tool trace and the mechanical parameters. Using of the TCA the mechanical parameters of the connecting elements on the contact zone could be analysed by different loads.

We have done this analysis which means we have analyzed the effect of the modified parameter for the normal stress, normal deformation and the normal elastic strain on the contact zone. Based on the results we have prepared diagrams for the analysis of the correlations between the changing parameters.

Before the real production the TCA is suggested because of the determination of the estimated mechanical parameters. If these parameters would not give required results it could be possible to return to the geometric designing process and modify the geometric parameters.

Keywords: Helical gear, tooth trace, TCA, CAD, normal

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1. Introduction

The helical gears are used very frequently in many mechanical constructs [6, 8, 9, 10, 11, 12, 13]. The one of the main applications is on the gear transmissions of vehicles (Figure 1).



Fig. 1. A Volkswagen gear transmission [14]

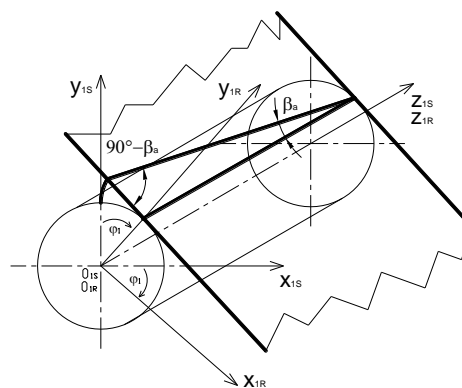


Fig 2. The derivation of the involute surface [1, 6]

The teeth are picked up the loads at the same time and ceased at the same time. The connection is continuously. The force effects from the friction are worked into two directions on the same tooth. Based on this the helical gear is run more

silently [2, 3, 5, 6, 11].

The disadvantage is the axial force which could be compensated by Herringbone gear [6].

Based on Figure 2. a tangential plane is turned around the basic cylinder on the K_{IR} rotation coordinate system. The basic cylinder is situated on the K_{IS} standing coordination system. This is the derivation of the involute helical surface (Figure 2.). [1, 6] An arbitrary oblique line of this tangential plane is describe an involute helical surface. Every planar section of this is verdant involute. The tangential plane of the basic cylinder is the connection plane (Figure 3) [3, 5, 6].

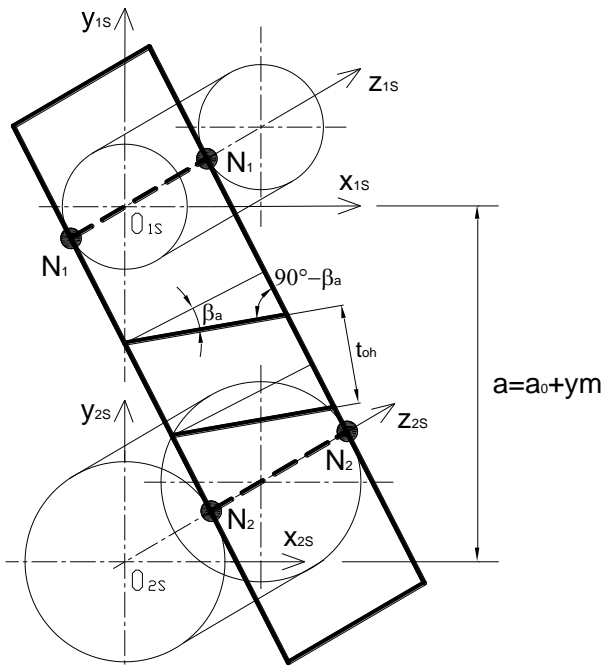


Fig. 3. The basic cylinders of the involute helical surfaces [1, 6]

The generator of the tooth surface is closed β_a angle with the axis of the basic cylinder. The $(90^\circ - \beta_a)$ is the helix angle of the involute helical surface [3, 5, 6, 7].

2. Designing of the x-zero helical gear pairs

The basic property of the x-zero gear drives is the connection is done on the pitch circle diameter, that is why the pitch circle diameter and the rolling circle diameter are the same [3, 5, 6]:

$$a_o = \frac{d_1 + d_2}{2} \quad (1)$$

$$d_1 = d_{w1} \quad (2)$$

$$d_2 = d_{w2} \quad (3)$$

Knowing of the geometric parameters from the references we have worked out a computer software because of the facilitation of the designing process [1].

We have designed five types of helical gears based on

the modification of the tooth trace (Table 1). All of other parameters has been the same, only the tooth trace has been modified on the pitch cylinder (β_0) (Figure 4):

$$\beta_a = a \sin(\cos \alpha_0 \cdot \sin \beta_0) \quad (4)$$

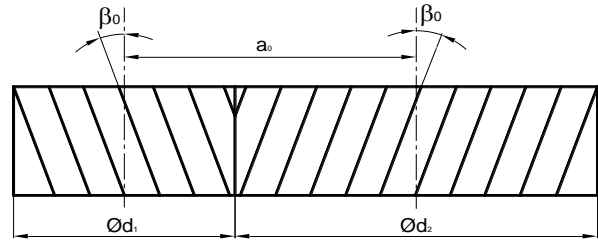
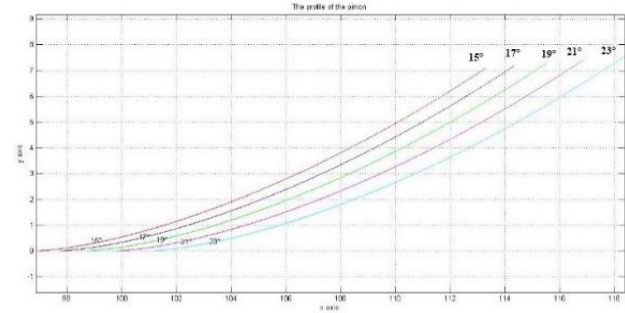


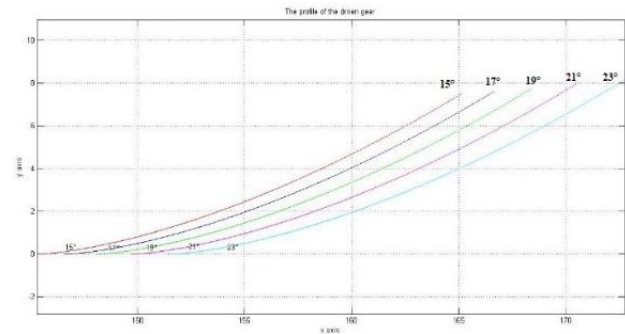
Fig. 4. The position of the β_0 angle

The involute profile angle on the pitch circle:

$$\alpha_{oh} = a \cos(\tan \beta_a / \tan \beta_0) \quad (5)$$



a) The profiles of the pinion



b) The profiles of the driven gear

Fig. 5. The calculated gear profiles in the function of the tooth trace

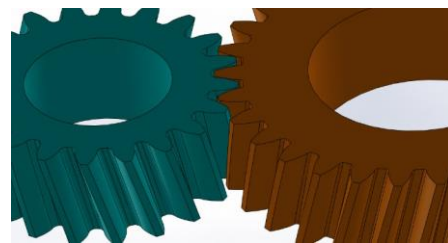


Fig. 6. The CAD models of the designed gear pairs

Table 1. The parameters of the designed helical gear pairs

	Gear drive I.	Gear drive II.	Gear drive III.	Gear drive IV.	Gear drive V.
Axial module (m) [mm]	10				
Number of teeth of the pinion (z_1)	20				
Number of teeth of the driven gear (z_2)	30				
Tooth trace (β_0) [°]	15	17	19	21	23
Profile angle (α_0) [°]	20				
Axial pitch (t_{0h}) [mm]	32.524	32.851	33.226	33.651	34.129
Addendum (h_a) [mm]	10				
Bottom clearance (c) [mm]	2.5				
Dedendum (h_f) [mm]	12.5				
Whole depth (h) [mm]	22.5				
Pitch circle diameter of the pinion (d_1) [mm]	207.055	209.138	211.524	214.229	217.272
Pitch circle diameter of the driven gear (d_2) [mm]	310.582	313.707	317.286	321.343	325.908
Elementary centre distance (a_0) [mm]	258.819	261.422	264.405	267.786	271.590
Tip circle diameter of the pinion (d_{a1}) [mm]	227.055	229.138	231.524	234.229	237.272
Tip circle diameter of the driven gear (d_{a2}) [mm]	330.582	333.707	337.286	341.343	345.908
Root circle diameter of the pinion (d_{r1}) [mm]	182.055	184.138	186.524	189.229	192.272
Root circle diameter of the driven gear (d_{r2}) [mm]	285.582	288.707	292.286	296.343	300.908
Backlash (j_s) [mm]	1.626	1.642	1.661	1.682	1.706
Working depth (h_w) [mm]	20				
Tooth thickness (S_{ax}) [mm]	15.449	15.604	15.782	15.984	16.211
Basic circle diameter of the pinion (d_{a1}) [mm]	193.756	195.460	197.403	199.596	202.050
Basic circle diameter of the driven gear (d_{a2}) [mm]	290.634	293.190	296.105	299.394	303.076
Transmission ratio (i)	1.5				

The calculated profiles of the pinion and the driven gear could be seen in the function of the modification of the tooth trace on Figure 5. The involute profile is changed in the function of the modification of the tooth trace.

Knowing of the calculated parameters of the designed gear pairs the CAD (Computer Aided Designing) models of the elements could be prepared by Solidworks software (Figure 6).

Our developed program can export the profile curves of the elements in *txt* format. Based on this points interpolated B-spline [4] could be fitted for the points by Solidworks software. After that the CAD models could be built up [1].

3. Tooth contact analysis (TCA)

The aim of the TCA is the analysis of the established mechanical parameters of the gear pairs before the production. If the analyzed mechanical parameters are not appropriate there are possibilities for the modification of the ge-

ometric parameters of the gears for the better working conditions [5, 7, 8].

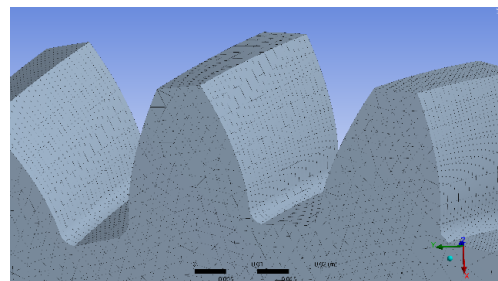


Fig.7. The adoption of the FEM mesh

Face sizing type meshing has been used on the contact zone around a sphere having 35 mm. The type of the plane figure has been a triangle. The applied density of the meshing has been 2 mm (Figure 7). The meshing has been handed out equally along the tooth length.

The applied material has been structural steel (Table 2).

Table 2. The parameters of the applied materials

Density	7850 kg/m ³
Yield limit	250 MPa
Ultimate strength	460 MPa

3.1 Defining of the load and boundary conditions

During the analysis, 3 coordinate systems have been defined: two standing coordinate systems on the middle rotation axes of the gear pairs (K_{1S}, K_{2S}), one coordinate system

on the contact zone of the gears.

All freedom degrees of the driven gear have been fixed. Five freedom degrees of the pinion have been fixed, only the rotation around the axis of rotation has been permitted.

The pinion has been loaded by 500 Nm moment. The contact tooth surfaces of the gear pairs have been analyzed.

3.2 Analysis of the normal stresses on the tooth surfaces

The normal stress is defined perpendicularly for the tooth surfaces on the coordinate system on the contact zone of the gears. These direction is the most determinative in aspect of tooth deformation.

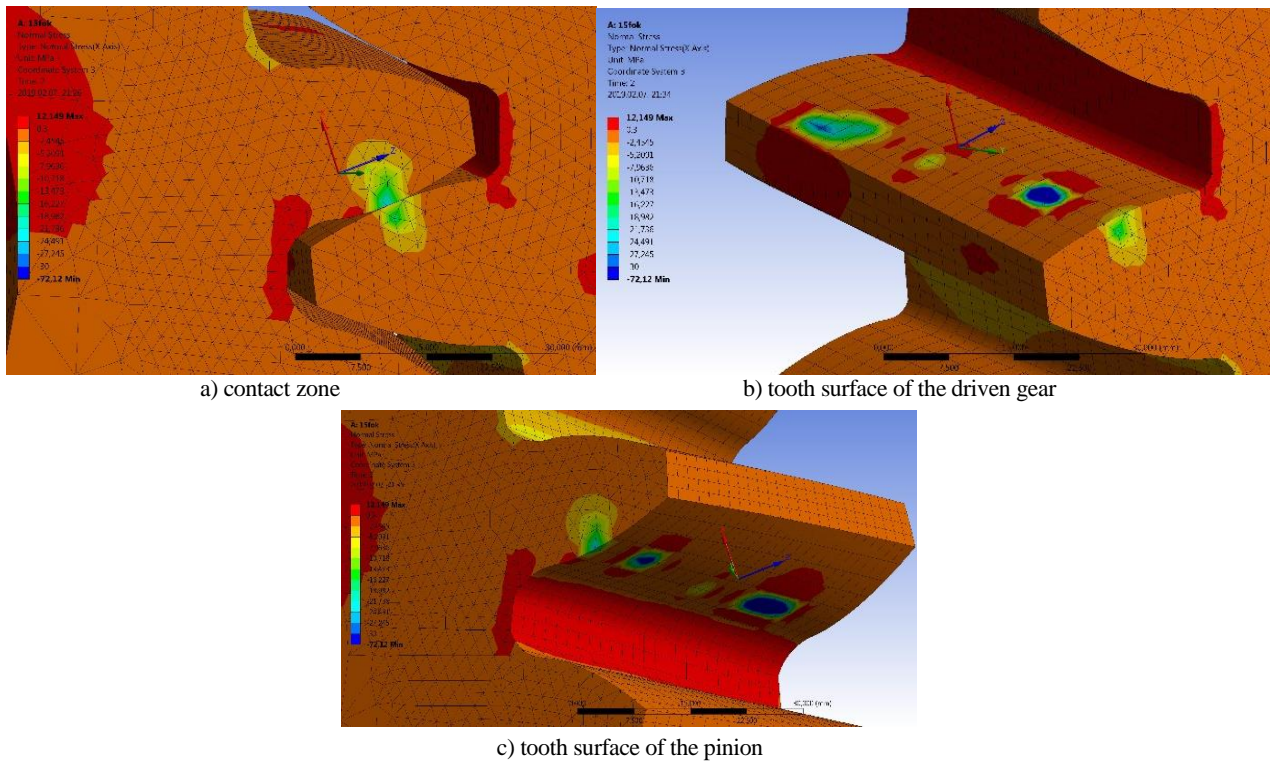


Fig. 8. Normal stress distribution (m=10 mm, z₁=20, z₂=30, β₀=15°)

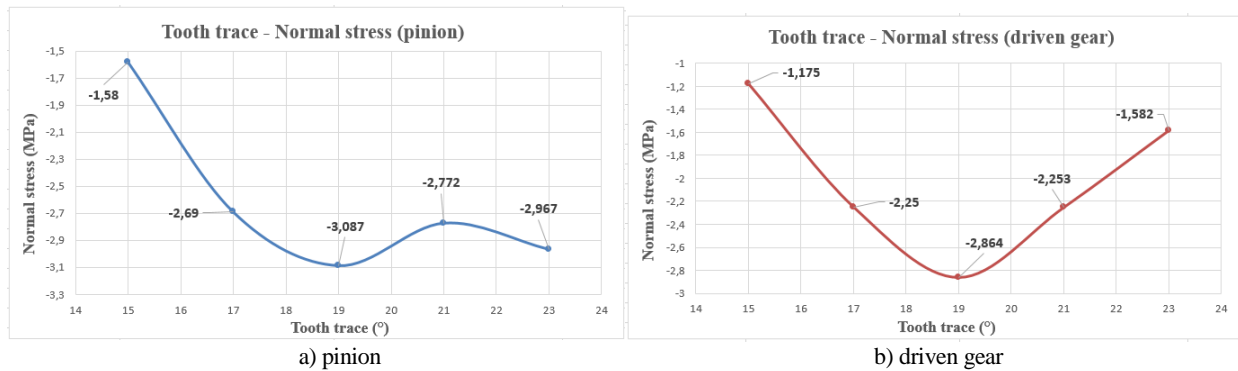


Fig. 9. The normal stress results in the function of the tooth trace

3.3 Analysis of the normal elastic strain on the tooth surfaces

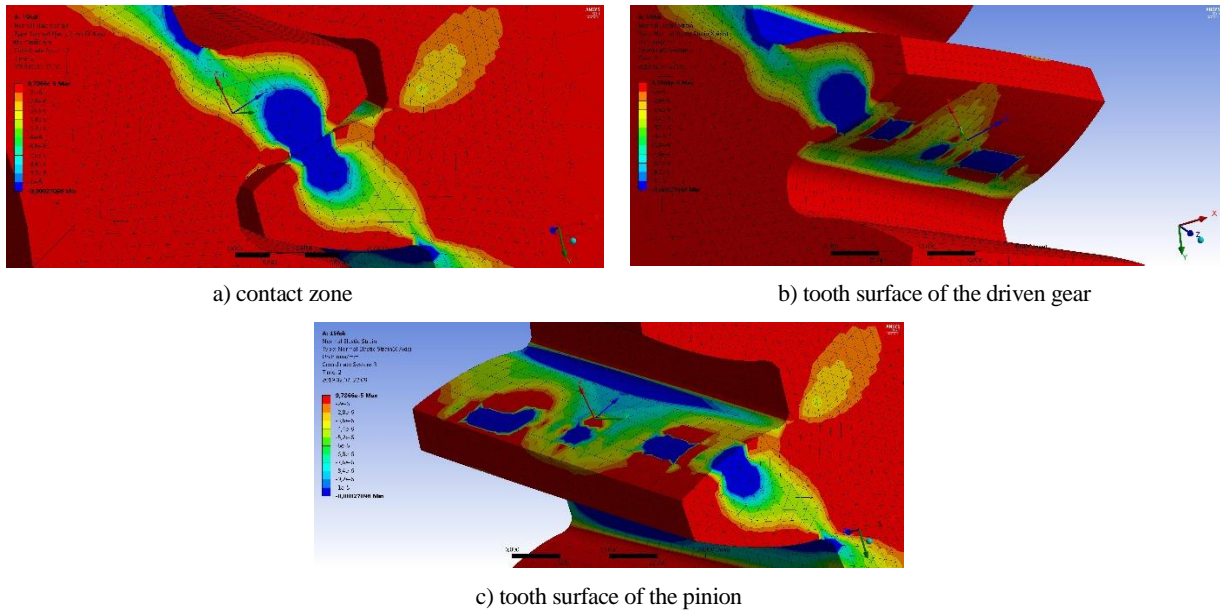


Fig. 10. Normal elastic strain distribution
($m=10\text{ mm}$, $z_1=20$, $z_2=30$, $\beta_0=15^\circ$)

The normal elastic strain is determined perpendicularly for the tooth contact surface. This direction is the most determinable in the tooth connection.

3.4 Analysis of the normal deformation on the tooth surfaces

The deformation could be determinable to many directions. In case of the tooth connection the main direction is the normal direction which is perpendicular for the contact surfaces. It could be determined by the help of the coordinate system on the contact zone of the gears (Figure 12.a).

The normal stress results in the function of the tooth trace could be seen on Figure 9.

A little fluctuation could be seen on the pinion side (Figure 9.a). The lowest normal stress in absolute value could be achieved in case of Gear drive I.

The normal stress is increased until $\beta_0=19^\circ$ tooth trace. After that the normal stress is started decreasing (Figure 9.b) on the driven gear side. The lowest normal stress in absolute value could be achieved in case of Gear drive I.

According to the normal stress distribution the Gear drive I has better stress results.

The normal elastic strain results in the function of the tooth trace could be seen on Figure 11.

The lowest result could be found in case of $\beta_0=15^\circ$ in absolute value on the pinion (Figure 11.a). The highest result could be found in case of $\beta_0=23^\circ$ in absolute value.

The lowest result could be found in case of $\beta_0=15^\circ$ in absolute value on the driven gear (Figure 11.b). The highest result could be found in case of $\beta_0=19^\circ$ in absolute value.

According to the normal elastic strain distribution the Gear drive I has better results beside the constancy of the other gear parameters (Figure 11).

The normal deformation results in the function of the tooth trace could be seen on Figure 13.

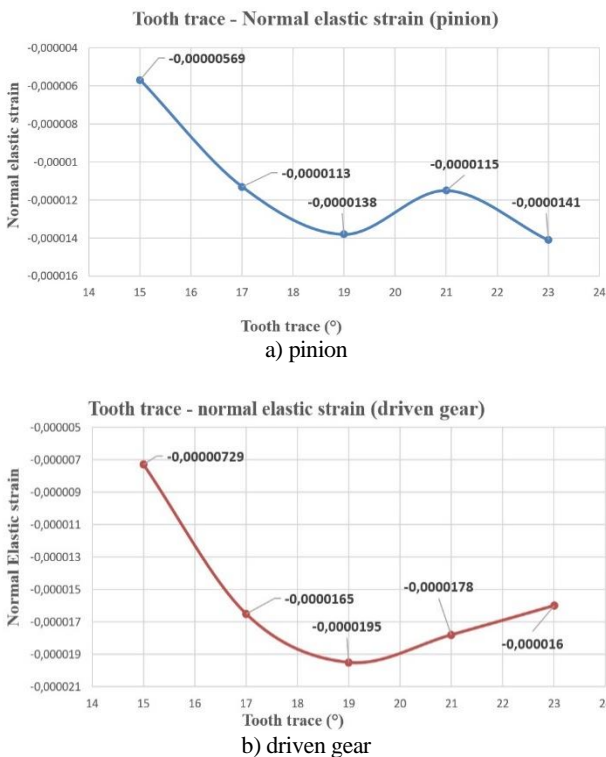
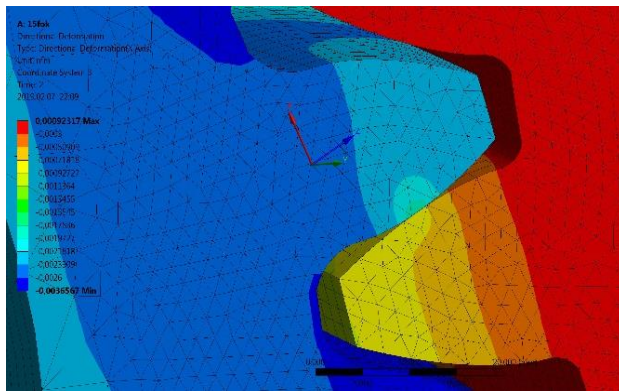


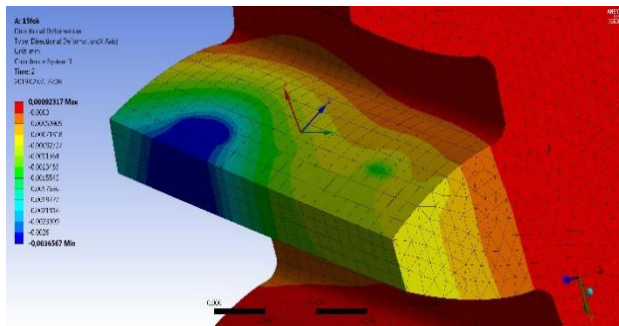
Fig. 11. The normal stress results in the function of the tooth trace

The lowest result could be found in case of $\beta_0=21^\circ$ in absolute value on the pinion (Figure 13.a). The highest result could be appeared in case of $\beta_0=17^\circ$ in absolute value. This value is outstanding high than the other deformation values.

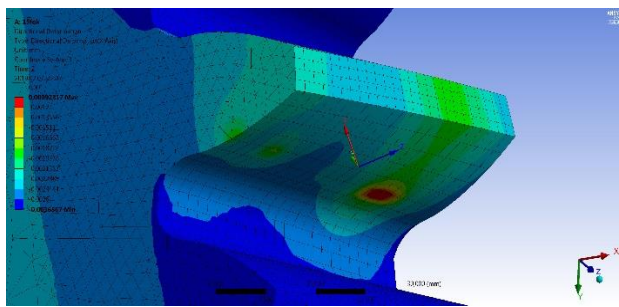
The lowest result could be found in case of $\beta_0=15^\circ$ in absolute value on the driven gear (Figure 13.b). The highest result could be appeared in case of $\beta_0=19^\circ$ in absolute value.



a) contact zone

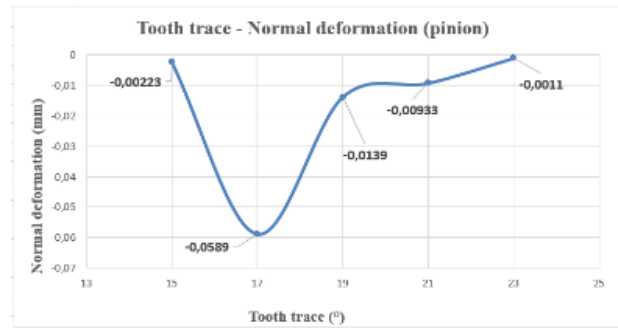


b) tooth surface of the driven gear

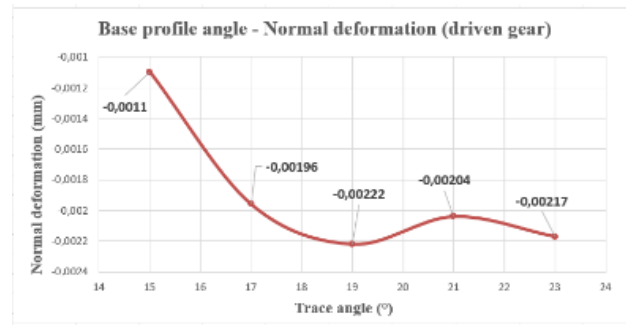


c) tooth surface of the pinion

Fig. 12. Normal deformation distribution ($m=10$ mm, $z_1=20$, $z_2=30$, $\beta_0=15^\circ$)



a) pinion



b) driven gear

Fig. 13. The normal deformation results in the function of the tooth trace

4. Conclusions

The helical gear pairs are used in many mechanical constructions for instance vehicle's gear transmissions or working machines. The TCA is important because of the achievement of better mechanical conditions during the working. The modification of the geometric parameters of the gear pairs is important due to the optimized TCA's results.

Five types of α -zero helical gear pairs have been designed. Only the tooth trace has been modified during the designing process. We have worked out a computer program with which the designing process could be easier. After the designing process the CAD models and the assembly construction could be created for the TCA.

TCA has been done for the analysis of the normal stresses, normal deformations and normal elastic strains. Our purpose is the comparative analysis of the TCA's results in the function of the modification of the tooth trace on the tooth contact zone. After that the correlations between the modified gear parameters and the TCA results could be determinable.

Based on our results we could achieve better TCA' results if we apply lower tooth trace beside the constancy of other gear parameters.

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Nomenclature

β_0	: the tooth trace on the pitch cylinder ($^\circ$)
β_a	: the tooth trace on the basic cylinder ($^\circ$)
$K_S(x_S, y_S, z_S)$: the stationary coordinate system of the helical gear
$K_R(x_R, y_R, z_R)$: the stationary coordinate system of the helical gear
φ_1	: angular displacement ($^\circ$)
a	: centre distance (mm)
a_0	: elementary centre distance (mm)
y	: specific increase of centre distance
m	: axial module (mm)
α_0	: profile angle ($^\circ$)
t_{0h}	: axial pitch (mm)
h_a	: addendum [mm]
h_f	: dedendum [mm]
h	: whole depth [mm]
$d_{1,2}$: pitch circle diameters (mm)
$d_{a1,2}$: tip circle diameters (mm)
$d_{f1,2}$: root circle diameters (mm)
$d_{w1,2}$: rolling circle diameters (mm)
j_s	: backlash (mm)
h_w	: working length (mm)
S_{ax}	: tooth thickness (mm)
$d_{a1,2}$: basic circle diameters (mm)
i	: transmission ratio
$z_{1,2}$: number of teeth

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