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THE DESIGN END EXECUTION OF THE SPEED INCREASING GEARBOXES

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Abstract: The process of design and execution of a speed increasing gearboxes is a complex process. The correct choice of materials from which its component parts are made is vital for good functioning. Using the specialized software such as KISSsoft AG allows a significant reduction of the design time and a very high accuracy of the calculations. The study aims to present the design of speed multipliers and the influence of the special processes in order to increase fatigue strength of the gears. The first part will focus on the choice of the materials for the main elements of the transmission and we will present the gear sizing using specialized calculation software such as KISSsoft. In the second part, we will focus on the influence of the shot peening and will determine the influence of the pretension on the gears through numerical simulation. Thus, we will demonstrate that through a relatively simple operation of inducing internal tensions at the gears tooth root the fatigue strength increase, which means that the lifetime of the gears increase.
Keywords: process, design, speed increaser, KISSsoft, calculations

1. INTRODUCTION

In this paper we will present the industrial design method of a mechanical transmission, the materials used for the speed increasing gearbox casing and for the pinions and gears end the execution technology for the pinion shaft and the calculation for this pinion using specialized software.
We will also present the influence of the compression stresses introduced at the gears tooth root on the fatigue strength and we will determine this influence through numerical simulation.
The residual stress induced by shoot peening increase the fatigue strength [1], [2,], [3].
In [4], [5], [6], [7], [8], [9] there are recommendations regarding the plastic deformation of surfaces through shoot peening in order to reach maximum fatigue limit [10].
By inducing these compression stresses, fatigue strength increase, figure 1.1 [3].

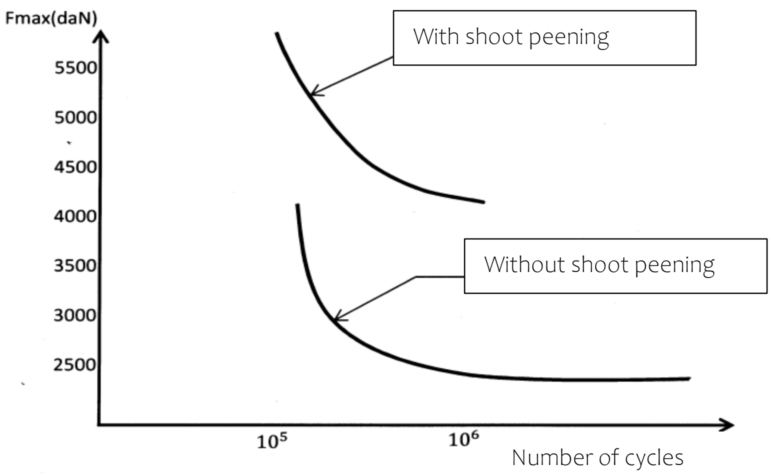

Figure 1.1. Improvement of fatigue strength by shoot peening

Studies on the influence of gear wear in mechanical transmissions were carried out by Solod [11] and Guimiar Renaul [12]. They studied the influence of stress induction through shoot peening in high–alloy material. We will present influence of the shoot peening on non–alloyed materials and how these stresses improve the fatigue behavior of gears.

2. THE CONSTRUCTIVE SOLUTION FOR THE SPEED INCREASING GEARBOX

The speed increasing gearbox is with one stage, the input shaft and the output shaft are placed in the vertical plane with the distance between the axes A = 112.5 mm [10].
The constructive solution chosen for this speed increasing gearbox is presented in figure 2.1 [10].
The gear case consists of three parts: the lower case, the intermediate case and the upper case. Figure 2.2 shows the main dimensions of the welded case [10].

ISSN 1584 – 2665 (printed version); ISSN 2601 – 2332 (online); ISSN-L 1584 – 2665

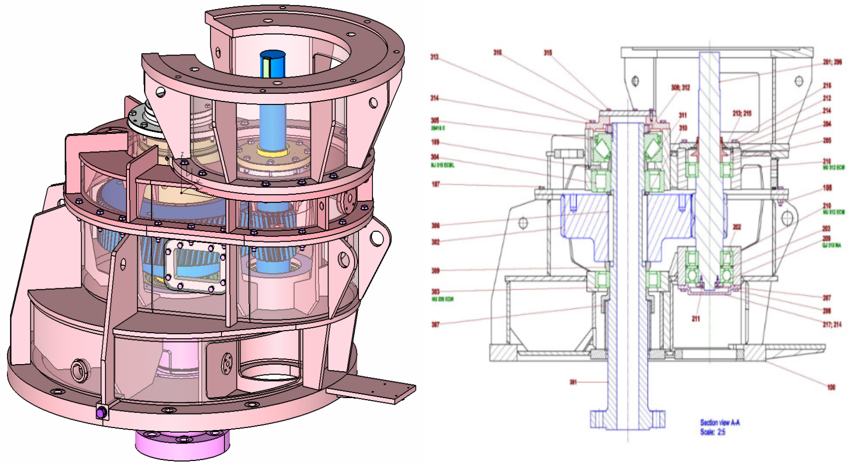

Figure 2.1. Speed increaser tip AV 11,25

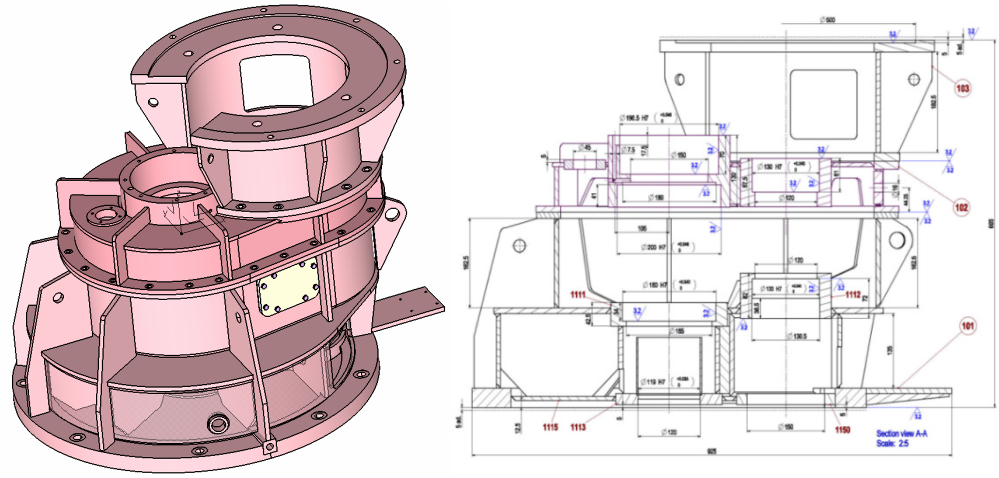

Figure 2.2. The main dimension of the speed increaser gear case type AV

Materials and heat treatments used for construction of the speed increaser gearbox

The gear case is executed from S255 JR steel according to EN 10025–2:2004. The chemical composition and the mechanical characteristics of the material from which the casing is made, is presented in table 2.1 according to EN10025–2:2005.

Table 2.1. Chemical composition S255JR

Name / symbolization material	C % max	Si % max	Mn % max	P % max	S % max	N % max	Cu % max	RP02 [N/mm^2] min	Rm [N/mm^2]	A5 [%] min	Testing strength [KV] 20^{0}C
S255JR	0.17	x	1.4	0.35	0.35	0.012	0.55	225	360–510	26	27

After the welding operation, the stress relieving heat treatment follows. Stress relieving heat treatment aims to reduce the internal stresses that appear, develop and partially remain in metal products as a result of the technological processing [13]. The stress relieving heat treatment was carried out in the S.C. Resita Reductoare & Regenerabile company.

The stress relieving temperature was 550–570**℃**. The heating was increased with 50**℃** per hour until the prescribed temperature was reached. For this multiplier housing, the holding time was 5 h. Cooling was done in the oven at the maximum speed of 50**℃** per hour to the temperature of 120 and then in the atmosphere of the production workshop without air currents [10].

The pinion shafts and the gear shown in figure 2.3 are executed from C45.

The chemical composition of the material from which the pinions and gears are made, is presented in table 2.2 [14].

Table 2.2. Chemical composition C–45

Name symbolization material	C %	Si %	Mn %	P %	S %	Cr %
C45	0.42–0.50	0.17–0.37	0.50–0.80	0.040	0.04	0.30

ISSN 1584 – 2665 (printed version); ISSN 2601 – 2332 (online); ISSN-L 1584 – 2665

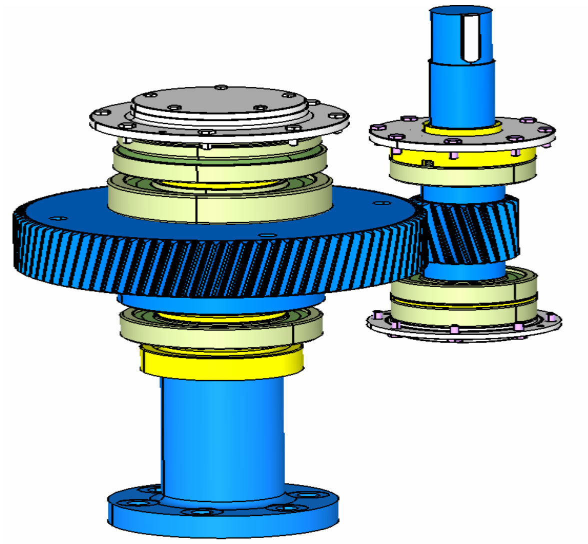

Figure 2.3. The speed multiplier shafts

Geometrical data of the pinions and gears

Figure 2.4 show the main dimensions of the gear [10]. Figure 2.5 show the main dimensions of the output pinion [10].

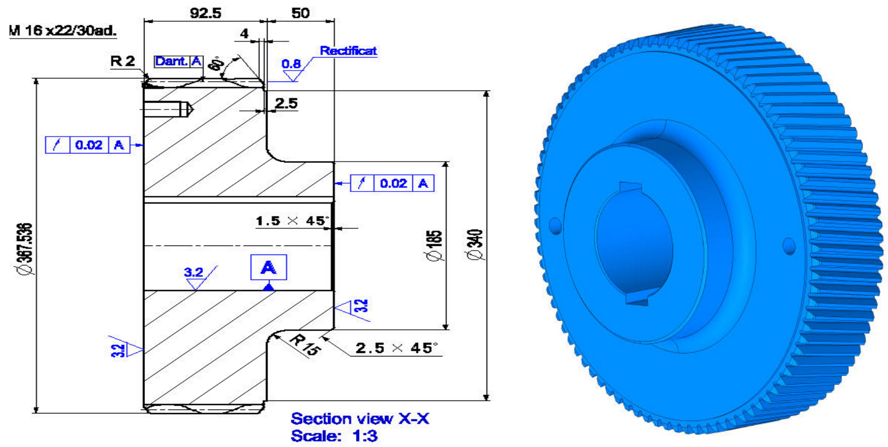

Figure 2.4. Input gearwheel

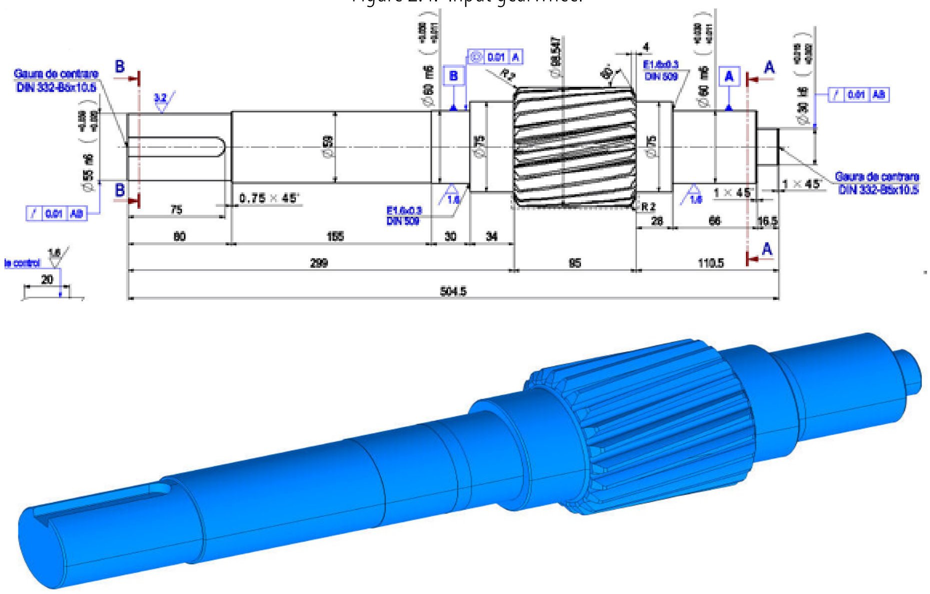

Figure 2.5. Output pinion

The execution of these gears was carried out in accordance with the execution technology of the Reșita Reductoare si Regenerabile company. The technological process is presented in the table 2.3.

ISSN 1584 – 2665 (printed version); ISSN 2601 – 2332 (online); ISSN-L 1584 – 2665

Table 2.3 Pinion technological process

Process	Resource	Processing time/Piece[h]
Mechanical cutting	Cutting machine type HB 550	1.5
Ultrasonic test	Ultrasonic control device type US–USIP	0.33
Lathe turning	Lathe machine type SNA 501x1500	1.25
Gear Cutting	Gear cutting machine type Liebherr L401	2.08
Inductive heat treatment	Induction heat treatment installation	1
External grinding	Grinding machine type RU 450/2500	0.25
Teeth grinding	Grinding machine type Niles	1.2
Keyway execution	Milling machine type Mikromat 900x1400	0.58
Pinion inspection	CNC Klingelnberg machine type PNC 150	0.5
Final control		0.5

The technological process is completed with the control operation on the CNC machine type PNG 150.

Geometric calculation of the gears

The geometric calculation of the tooting was performed using the KiSSsoft software in accordance with the DIN 3960:1987 standard. In table 2.4 the calculation of the teeth is presented.

Table 2.4 Gears geometrical calculation

	Symbol	Gear 2	Gear 1
Centre distance (mm)	[a]	112,5	
Centre distance tolerance	Acording to ISO 286:2010 – js 7		
Normal module (mm)	[m_n]	2	
Pressure angle at normal section (°)	[α_n]	20	
Helix angle at reference circle (°)	[ß]	10	
Number of teeth	[z]	22	88
Face width (mm)	[b]	47,5	45
Hand of gear		left	right
Accuracy grade (according Q–ISO 1328:1995)		6	6
Material		C45 with induction heat treatment	
Surface hardness	–	57 HRC	57 HRC
Fatigue strength tooth root stress (N/mm^2)	[σ_{Flim}]	370	370
Fatigue strength for Herzian pressure (N/mm^2)	[σ_{Hlim}]	1220	1220
Tensile strength (N/mm^2)	[Rm]	700	700
Yield point (N/mm^2)	[Rp]	490	490
Young modulus (N/mm^2)	[E]	206000	206000
Gear ratio	[u]	4,000	
Tooth thickness	[s_n]	1,6790	1,7624
Tip alteration (mm)	[k·mn]	–0,021	–0,021
Reference diameter (mm)	[d]	44,679	178,715
Base diameter (mm)	[db]	41,908	167,633
Tip diameter (mm)	[da]	49.231	183.726
Tip diameter allowance (mm)	[33]	0,000/–0,390	0,000/–0,720
Tip form diameter (mm)	[d_{Fa}]	49.231	183.726
Operating pitch diameter (mm)	[d_w]	45,000	180,000
Root diameter (mm)	[d_f]	39.670	39.670
Nominal circumferential force (N)	[Ft]	14747.5	
Axial force (N)	[Fa]	2600	
Radial force (N)	[Fr]	5450.5	
Required safety	[S_{Fmin}]	1,50	1,50
Safety for tooth root stress	[S_F]	1,41	1,33
Transmittable power (KW)		32.42	30.69
Safety against pitting			
Safety factor for contact stress at operating pitch circle	[S_{Hw}]	0.88	0.88
Required safety	[S_{Hmin}]	1,1	1,1
Transmittable power (KW)		22.20	22.20
Safety for stress at single tooth contact	[$\sigma_{HG}/\ \sigma_H$]	0.88	0.88
Meshing efficiency (%)	[η]	98,926	

ISSN 1584 – 2665 (printed version); ISSN 2601 – 2332 (online); ISSN-L 1584 – 2665

Using the calculation software, the value of the tangential and radial forces in the gear were determined according to the table 2.4 as follows Ft=14747N, Fr=5450 N.

3. DETERMINATION OF THE INFLUENCE OF COMPRESSION STRESSES BY NUMERICAL SIMULATION

The analysis will focus on the study of the influence of shoot peening on the mechanical behavior of the pinion teeth, figure 3.1 executed from OLC 45[10]. The tangential force Ft=14747 N and the radial force Fr=5450 N act on a tooth of the pinion. The static analysis will be performed by finite element simulation [15], using the SolidWorks Simulation module [16]. For comparison, the simulation was also applied to the variant of the tooth without pretension [10].

Figure 3.1 show the geometry of the output pinion shaft, where index 1 marks the analyzed tooth, and index 2 the direction of the tangential force Ft, perpendicular to the side flank of the tooth and tangent to the running diameter equal to 45 mm [10].

The discretization of the geometry was done with 171895 finite elements, figure 3.2.

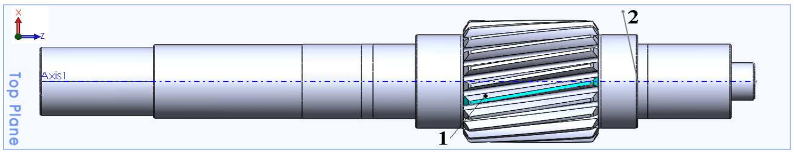

Figure. 3.1. Output pinion shaft geometry

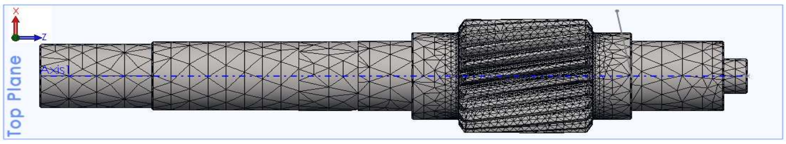

Figure 3.2. Discretization of the output pinion shaft geometry into 171895 finite elements

Figure 3.3 shows the boundary conditions applied to the geometry, for the simulation variant without pretension [10]:

- fixing the geometry through the Fixed Geometry condition next to the two bearings of the shaft, which is equivalent to canceling the degrees of freedom of these entities;
- the tangential force Ft=14747 N, applied perpendicular to the flank of tooth 1, in the direction defined by line 2;
- the radial force Fr=5450 N, applied on the flank of tooth 1, following the radial direction.

Figure 3.4 shows the calculated prestressing force of 4860 N, applied at the base of tooth 1, along the radial direction.

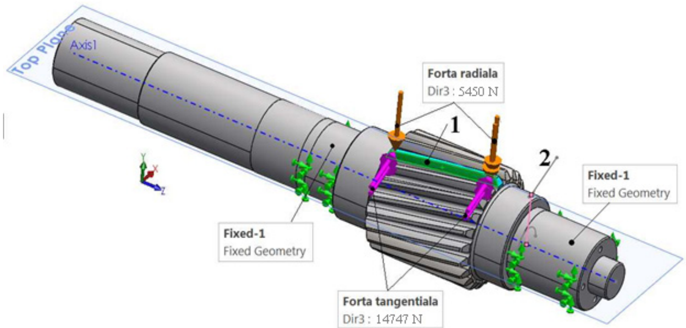

Figure. 3.3. Boundary conditions applied to the output pinion shaft geometry

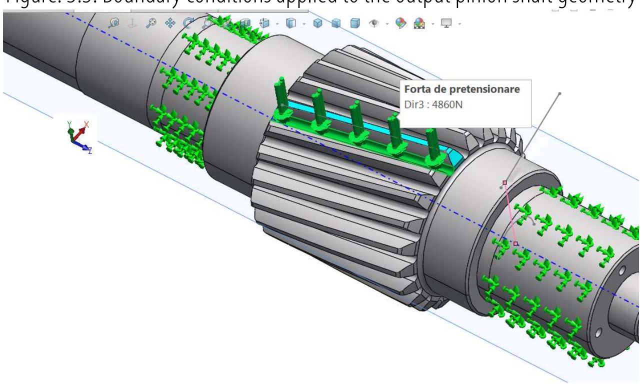

Figure. 3.4 Calculated pretension force of 4860 N

ISSN 1584 – 2665 (printed version); ISSN 2601 – 2332 (online); ISSN-L 1584 – 2665

The numerical results of the simulation are centralized comparatively in table 3.1, where:

- the vonMises stress σ is defined by relation 3.1, σxx, σyy, σzz, σxy, σyz, σzx being the components of the stress matrix

$$\sigma = \sqrt{\frac{1}{2}\left[\left(\sigma_{xx}-\sigma_{yy}\right)^2+\left(\sigma_{yy}-\sigma_{zz}\right)^2+(\sigma_{zz}-\sigma_{xx})^2+6\cdot\left(\sigma_{xy}^2+\sigma_{yz}^2+\sigma_{zx}^2\right)\right]} \qquad (3.1)$$

- the maximum deformation δ is the maximum deformation of the tooth as a result of the applied loads;
- the safety coefficient C_f is defined as the ratio between the yield strength of the material and the maximum vonMises stress σ in the shaft.

Table 3.1 Simulation results

				Ft=14747 N, Fr=5450 N	
Case analysis	Size	Symbol	UM	Value	Figure
Without pretension	vonMises stress	σ	MPa	451	3.6
	Maximum deformation	δ	mm	0,027	3.7
	Safety factor	C_f	–	1,086	
With pretention 45 MPa	vonMises stress	σ	MPa	436,49	3.8
	Maximum deformation	δ	mm	0,025	3.9
	Safety factor	C_f	–	1,123	
With pretention 95 MPa	vonMises stress	σ	MPa	420,44	3.10
	Maximum deformation	δ	mm	0,024	3.11
	Safety factor	C_f	–	1,165	

Figure 3.5 highlights the area of maximum stress at the tooth root. Figures 3.6 ÷ 3.11 present the distribution of the vonMises stress σ respectively the deformation δ for the three analysed cases [10].

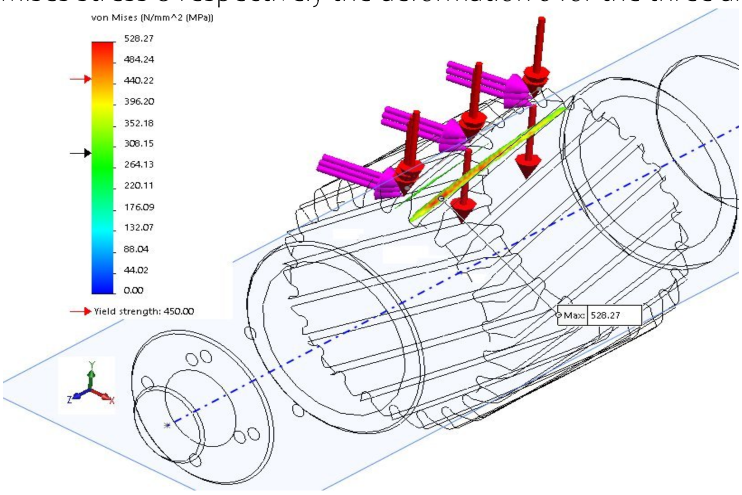

Figure. 3.5 The zone of maximum stress at the base of the tooth

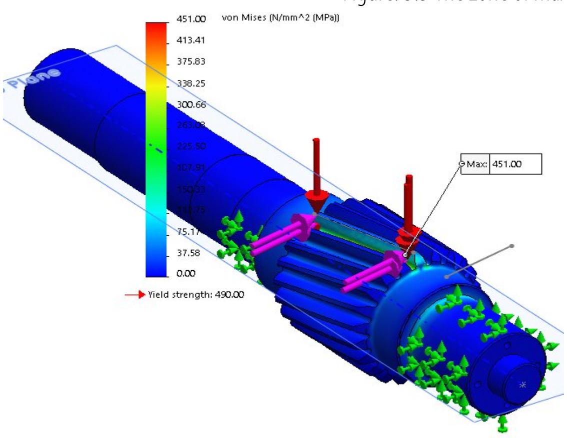

Figure. 3.6 Otput shaft – without pretension – σvonMises=451 MPa
Ft = 14747 N Fr = 5450 N

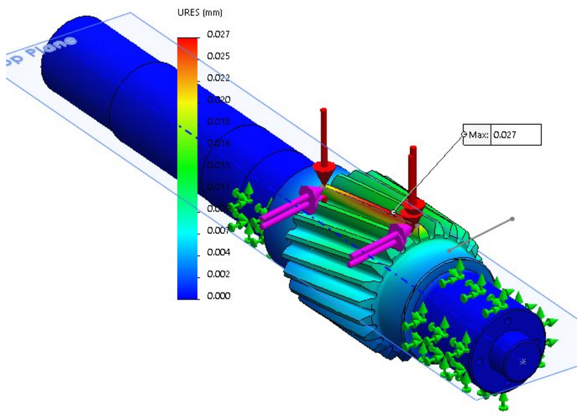

Figure. 3.7 Otput shaft – without pretension – δmax=0,027 mm
Ft = 14747 N Fr = 5450 N

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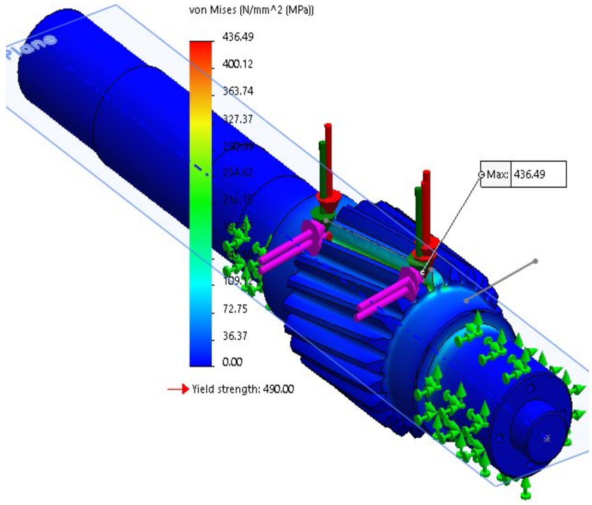

Figure. 3.8. Otput shaft – with pretension 45 MPa – σvonMises=436,49 MPa, Ft = 14747 N Fr = 5450 N

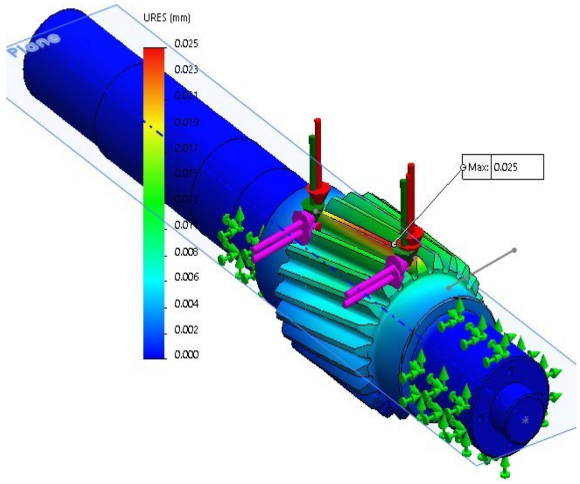

Figure. 3.9. Otput shaft – with pretension 45 MPa – δmax=0,025 mm, Ft = 14747 N Fr = 5450 N

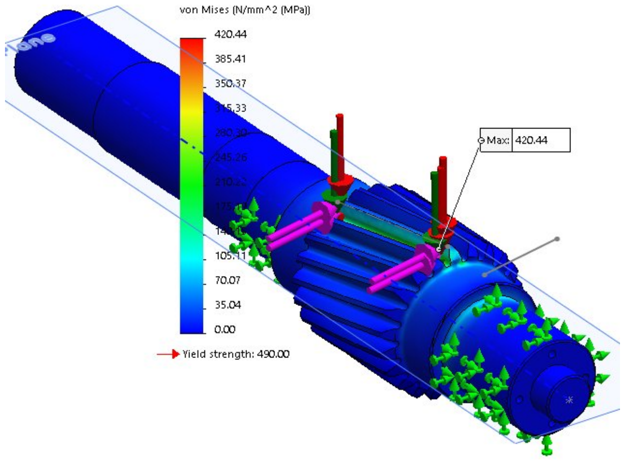

Figure. 3.10. Otput shaft – with pretension 95 MPa – σvonMises= 420,44 MPa, Ft = 14747 N Fr = 5450 N

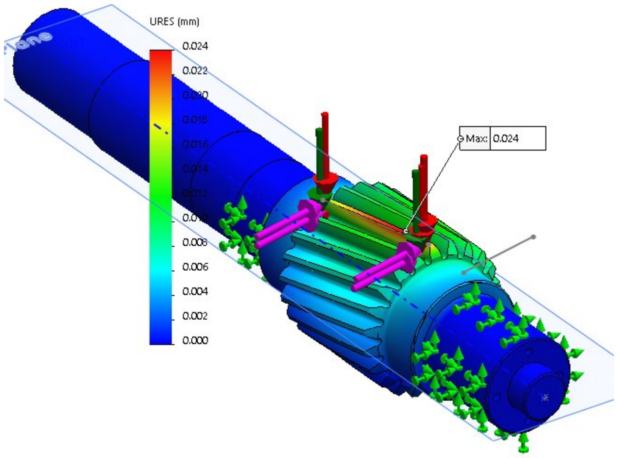

Figure. 3.11. Otput shaft – with pretension 95 MPa – δmax=0,024 mm, Ft = 14747 N Fr = 5450 N

4. CONCLUSIONS

We demonstrated how the behavior of the gears can be improved under fatigue stress by shot peening the teeth.

The analysis carried out highlighted the area at the base of the tooth as the area with maximum tension, regardless of the case analysed: with or without pretension [10].

At the level of deformations, the difference is insignificant for the three analysed cases, the values 0.852 mm, 0.876 mm and 0.904 mm being close.

The vonMises stress registers different values depending on the analysed case: the highest value of 528.27 MPa is recorded for the case without pretension and decreases to the values of 513.76 MPa and 497.71 MPa with the increase of pretension from 45 MPa to 95 MPa; this decrease demonstrates that the roughening has a beneficial effect on the tension at the base of the tooth, in the sense of reducing it simultaneously with the increase of the pretension.

The same beneficial influence also results from the values of the C_f coefficient, whose values increase from 0.852 for the case without pretension to 0.876 and 0.904 respectively.

The pinion was dimensioned so that it could transmit a moment T=329 Nm. This moment corresponds to a tangential load on the teeth Ft=14747N, in which case the service life is unlimited from the point of view of the bending stress at the base of the tooth.

The work focused on the presentation of the design and industrial execution of the component parts of a speed multiplier. The component parts of the mechanical transmission, the materials used and the dimensioning calculation method were presented. Types of thermal treatments applied industrially were presented.

Acknowledgement

This work was supported by the project "The Development of Advanced and Applicative Research Competencies in the Logic of STEAM + Health" / POCU/993/6/13/153310, project co–financed by the European Social Fund through The Romanian Operational Programme Human Capital 2014–2020.

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ISSN 1584 – 2665 (printed version); ISSN 2601 – 2332 (online); ISSN–L 1584 – 2665