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DESIGN, KINEMATIC AND STRUCTURAL ANALYSIS OF A WHEELCHAIR TRANSMISSION

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***Abstract*—** In this paper is presented the design, kinematics and structural analysis of a mechanical transmission intended to wheelchairs for disabled people. The kinematics of proposed transmission is analyzed in order to achieve a proper synthesis of gears. A 3D model of the transmission and wheelchair are designed in Solid Works, and they will be used for the dynamic simulation of the wheelchair robotic system in Adams software, and structural analysis in Ansys. The wheelchair motion simulation is made in Adams in two cases: traction motion and steering motion active. Based on Adams dynamic analysis is performed a structural analysis of designed worm and bevel gears in order to establish maximum value of contact pressure. The obtained results from Adams simulation and structural analysis are presented. Obtained results and conclusions demonstrate the efficiency of wheelchair proposed transmission solution.

***Keywords*—**Wheelchair, dynamics, structural analysis, motion study.

I. INTRODUCTION

THIS paper is structured in five sections. Upon the introduction section is presented a kinematic characterization of a proposed wheelchair transmission. The third part presents the design solution in Solid Works of a wheelchair differential transmission. In the last parts are presented the kinematical characterization of the wheelchair, by motion simulation in Adams and structural analysis in Ansys software of worm and bevel gears of wheelchair transmission.

Wheelchair remains the best transportation means for disabled people. The first mention of such device appears in 1595, for Phillip II of Spain, designed by an unknown inventor [1]-[2]. In last 20 years, the wheelchairs have become complex mechanical and electro-mechanical devices [3]-[6] and robotic systems [7]-[9].

The propulsion system of powered wheelchair typically is made of two motors, one for each drive wheel, and a drive train consisting of gears, belts that couples the motor's shaft to the drive wheel shaft. Speed and torque generated by each motor is controlled by modulating the pulse width (PWM-Pulse Wave Modulation) [10]-[12]. This research brings arguments for a mechanical transmission that achieves the differential movement. The traction and steering are made with different motors. The motion to wheels is made with transmission chains to achieve the proper angular speed difference of wheels. The model will be equipped with a motion controller, using a pulse width modulated DC (direct current) to DC converter (PWM).

II. KINEMATIC STUDY OF WHEELCHAIR TRANSMISSION

Kinematic scheme of wheelchair transmission is presented in Fig.1. Worm pinion 6 is actuated by electric motor M_1, and steering is performed by motor M_2 which rotates bevel pinion 8 (shaft V). Bevel gears, (10, 12) are planetary gears and they are mounted on shafts (IV, III) with pins assembly. Bevel gears, (11, 13) are satellites gears. Satellites gears are mounted on needle bearings to axes fixed on differential casing, and achieve a planetary motion. To calculate the transmission ratio, the principle of motion reversing is applied (called Willis principle). To straight line displacement the traction wheels of the wheelchair spin at the same angular velocity. The motion is transmitted from worm gears 6-5 to the shaft II and by means of final transmission with spur gears 4-2 to wheelchair wheels.

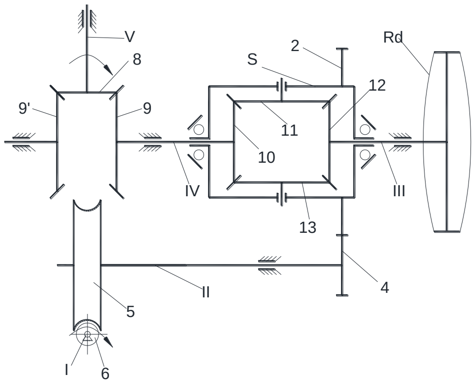


Fig. 1. Kinematic scheme of wheelchair transmission [1].

For steering, the motion is transmitted from motor M_2 to shaft V, trough bevel gears (8, 9), (10, 11), (11, 12) to semi-axis III, respectively to motor wheel Rd. The wheelchair wheels Rd and Rs will spin with the same angular speed and opposite rotation sense. The wheelchair moves in a circle. Casing S, and spur gear (2) are not moving (the motor M_1 is turned off).

The motion transmission chain for straight line displacement and the motion transmission chain for steering, of the wheelchair, are expressed by (1).

$$\begin{aligned}&\boldsymbol{M_1 - I - i_{65}^{w} - II - i_{42}^{c} - III - Rd}\\&\boldsymbol{M_2 - V - i_{89}^{k} - IV - i_{10\,11}^{k} - i_{11\,12}^{k} - III - Rd}\end{aligned} \qquad (1)$$

Where: M_1- traction motor; i_{65}^{w} -worm gear ratio; i_{42}^{c} -final transmission ratio; Rd and Rs, right and left wheel, I, II, III and IV – transmission shafts, M_2- steering motor; i_{89}^{k}-bevel gear ratio; $i_{10\ 11}^{k}$ -bevel gears 10, 11 ratio; $i_{11\ 12}^{k}$ - bevel gears 11, 12 transmission ratio.

Instead of worm gear it could be used a bevel gear, as in Fig. 2, where is illustrated the steering (a) and traction principle (b) of motion transmission [13]-[15].

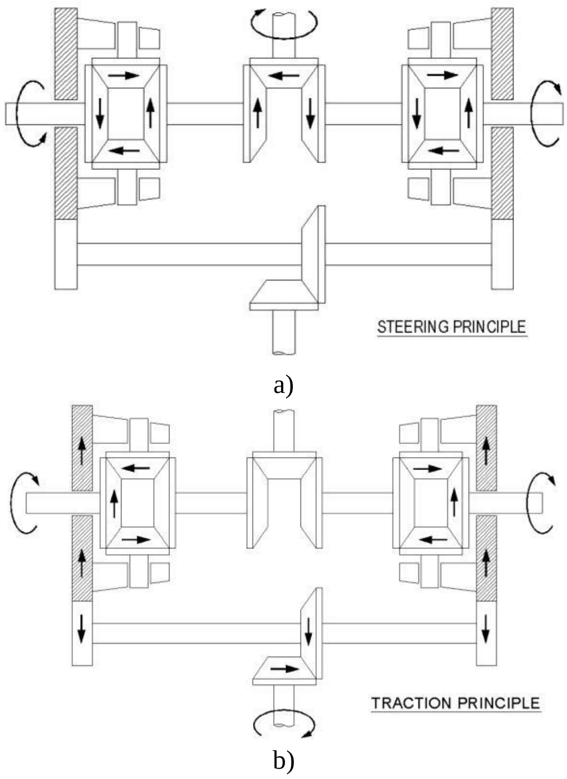


Fig. 2. Steering and traction principle [13].

For wheelchair straight line displacement when both drive wheels encounter the same resistance to ground, is valid (1). For steering, the governing equations of the wheels angular velocity are deducted in the following. In case of steering motion, the angular velocity of differential casing S is:

$$\omega_s = \frac{\omega_{10} + \omega_{12}}{2} \qquad (2)$$

ω_{10}, ω_{12} -absolute angular velocity of bevel planetary gears 10 and 12, considered in relation to differential casing;

ω_S - is absolute angular velocity of differential casing upon wheelchair frame.

From (2) is concluded that the angular velocity of the central gear (planetary) is twice the angular velocity of the differential box. If the steering motor is turned off, then $\omega_{10} = 0$, from (2) it is obtained:

$$\omega_{12} = 2\omega_s = 2\omega_2 \qquad (3)$$

Taking into account the gears ratio, is obtained (4):

$$\omega_s = \frac{\omega_{M_1}}{i_{65}^{w} \cdot i_{42}^{c}} \qquad (4)$$

Where: ω_S is the motion received from the traction motor. In (4) notations used are:

ω_{M_1} -is traction motor angular velocity; i_{65}^{w} -worm gear ratio; i_{42}^{c} -spur gears 4-2 transmission ratio.

In case of steering motion of the wheelchair, is obtained (5):

$$\omega_{10} = \frac{n_{M_2}}{i_{89}^{k}} \; ; \; \omega_{10}' = \frac{n_{M_2}}{i_{89'}^{k}} \, . \qquad (5)$$

ω_{10} is the motion received from steering motor M_2.

Assume that the right wheel Rd is outside the curve, in this case the pairs of satellite gears (11, 13) will spin. It is distinguished: the absolute motion of planetary gears, transportation movement of differential casing and relative movement of satellites gears towards differential casing S. In this context, the distribution of velocities in absolute motion of a planetary gear, is:

$$\omega_{10} \cdot R_W = \omega_S \cdot R_W + \omega_{11} \cdot r_W \qquad (6)$$

If is considered (2), is obtained the angular velocities for satellite gear (11) and planetary gears (10) and (12):

$$\begin{aligned}&\omega_{11} = (\omega_{10} - \omega_s)\frac{R_W}{r_W}, \; \omega_{10} = \omega_s + \omega_{11}\frac{r_W}{R_W},\\&\omega_{12} = \omega_s - \omega_{11}\frac{r_W}{R_W}\end{aligned} \qquad (7)$$

Where: R_W -is the rolling circle radius for planetary gears 10 and 12; r_W - rolling circle radius for satellites gears 11 and 13.

Relations deducted above are helpful to the kinematic synthesis of wheelchair transmission.

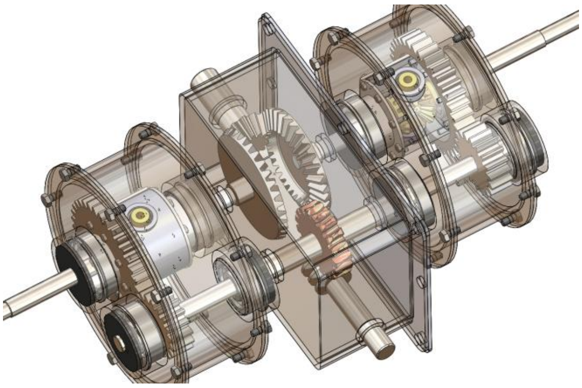
They are used to establish gears ratio, upon is designed the 3D model.

III. SOLID WORKS DESIGN OF WHEELCHAIR TRANSMISSION

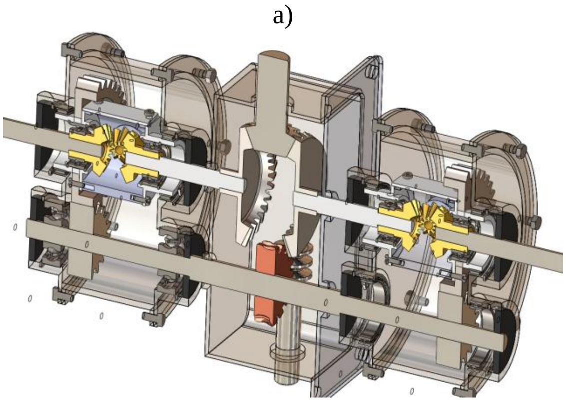
Gears and parts modeling are made with Gear Trax and Solid Works. The assembly models of transmissions, is shown in Fig. 3. Gears dimensions and ratios are presented in Table I.

TABLE I
GEARS RATIOS AND DIMENSIONS

Gear pair	Gears ratio[-]	Gears module [mm]	Gears rolling diameter [mm]
$Z_2=34$ $Z_4=18$	1,88	3	$dw_2=54,84$ $dw_4=101,5$
$Z_8=13$ $Z_9=33$	2,538	3	$dw_8=39,0$ $dw_9=99,8$
$Z_6=1$ $Z_5=17$	17	3,5	$dw_6=26,25$ $dw_5=58,8$
$Z_{10}=16$ $Z_{11}=10$	1,6	2,5	$dw_{10}=40$ $dw_{11}=25$
$Z_{12}=16$ $Z_{13}=10$	1,6	2,5	$dw_{12}=40$ $dw_{13}=25$



a)



b)

Fig. 3. Transmission assembly (with section)

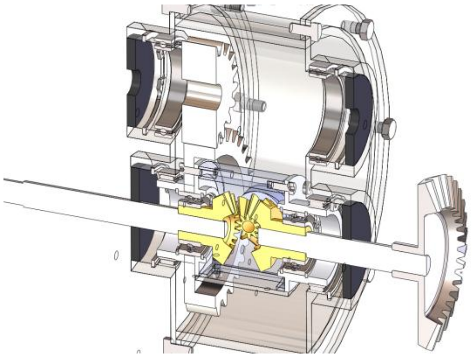


Fig. 4. Detail of a transmission (section).

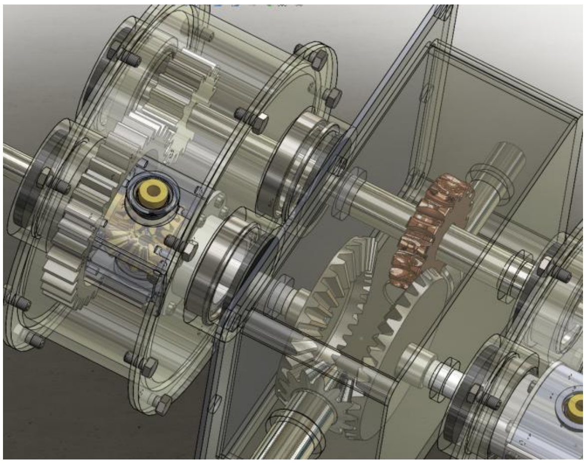


Fig. 5. Detail of a differential transmission.

Designed transmission are mounted on the wheelchair frame, the assembly view is presented in Fig. 6.

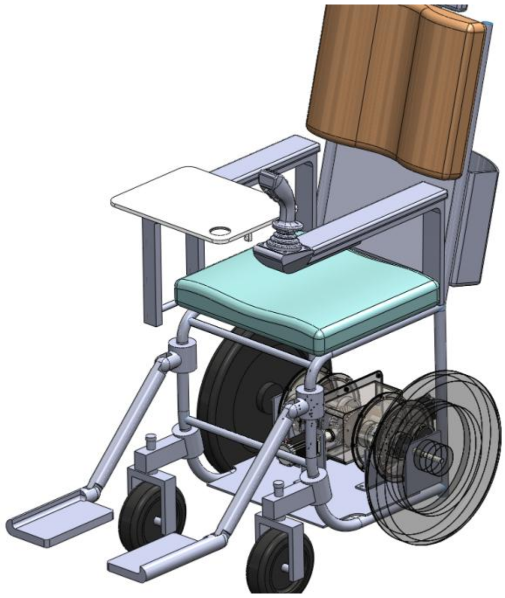


Fig. 6. Wheelchair assembly.

The obtained 3D virtual prototype is important because it will be used to simulate in Adams the wheelchair motion trajectory and study of motion dynamics. Also the virtual prototype is useful in further design studies, to optimize the construction (minimize weight, ergonomics studies).

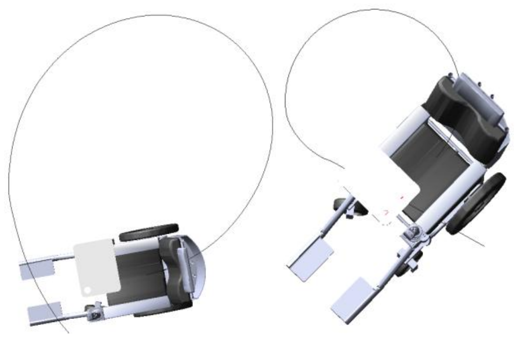
IV. Wheelchair motion simulation in Adams

The virtual prototype of the wheelchair, presented in Fig. 6, is transferred into Adams multi-body model, using the transfer interface embedded in Solid Works.

It is achieved robotic system workspace analysis in Adams, in first case for a combined trajectory, composed by a straight line motion and steering. Functions used in Adams to define straight line motion and steering motion are given by (8).

IF (time-2 : 30, 0, IF (time-4:0 , 32, 32)) –traction motion
IF(time-2 : 0 ,0 , 3.2) – for steering motion (8)

In the second case of simulation, is active only the steering motion, with value $\omega_2 = 3,2$ (rad / sec) (applied to shaft V). The simulation is made using WSTIFF solver with SI2 integration [16]. The wheelchair motions trajectory obtained in both cases are presented in Fig. 7.



a) b)

Fig. 7. Wheelchair motion trajectory computed in Adams.

The traction torque obtained by motion simulation is presented in Fig. 8, and steering torque in Fig. 9.

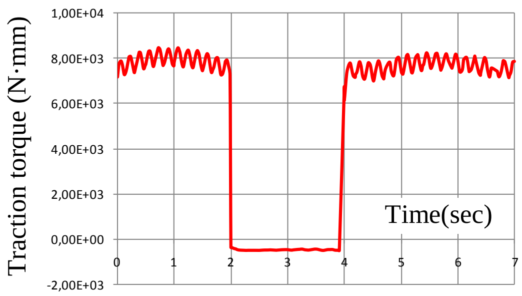


Fig. 8. Wheelchair computed traction torque in Adams.

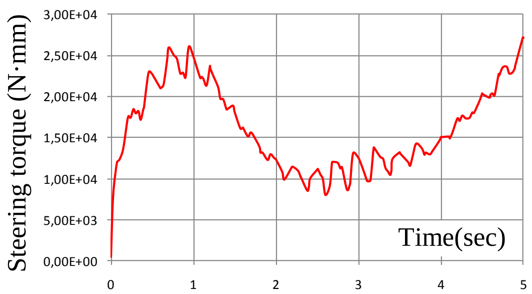


Fig. 9. Wheelchair computed steering torque in Adams.

V. Structural analysis of transmission gears

Structural analysis of transmission gears is made in Ansys to establish stress distribution and maximum values of stress and elastic deformations. The gears model is imported in Ansys database, are specified the material properties, boundary conditions, and loads. As loads are used the maximum values of shaft torques calculated in Adams dynamic simulation.

A. Worm gear structural analysis

Worm gear is imported in Ansys, are defined the proper joints, are specified the contact parameters as bonded, and is made the model meshing, as in Fig. 10.

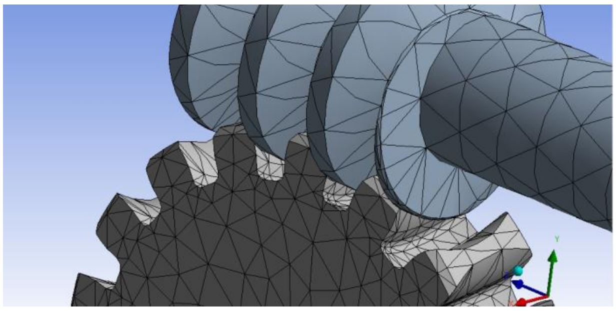


Fig. 10. Mesh model of worm gear.

The joint moment is specified as 9 (Nm), upon the Adams dynamic simulation results. Obtained equivalent von Misses stress distribution, is presented in Fig. 11. The maximum value is by 35 (MPa).

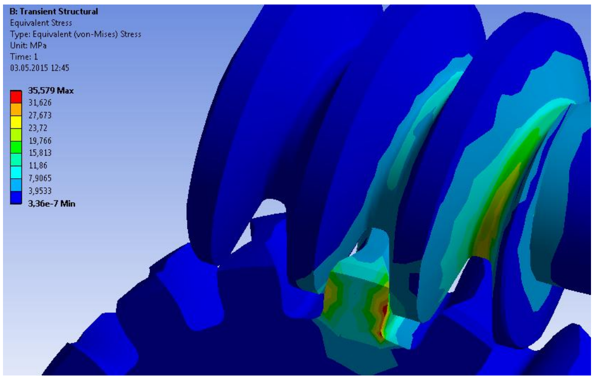


Fig. 11. Worm gear equivalent stress distribution.

Elastic deformation distribution of worm gear is shown in Fig. 12. Maximum value is by 0.0031 (mm).

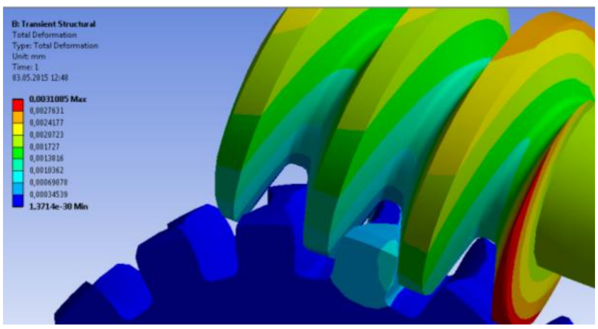


Fig. 12. Worm gear elastic deformation distribution.

B. Bevel gears structural analysis

The same procedure is followed for bevel gears 8-9 and differential bevel gears: 10-11and 12-13.

The mesh model of bevel gears 8-9 is shown in Fig.13. The steering torque, by 3.2 (Nm) calculated in Adams is used to define the rotational joint moment load. The model is solved, and the equivalent stress distribution is presented in Fig. 14. The maximum value is 23 (MPa). The maximum value for the elastic deformation of bevel gear 8, is 0.003 (mm), Fig.15.

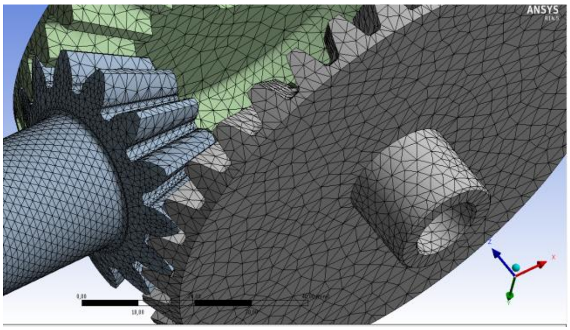


Fig. 13. Bevel gears mesh model.

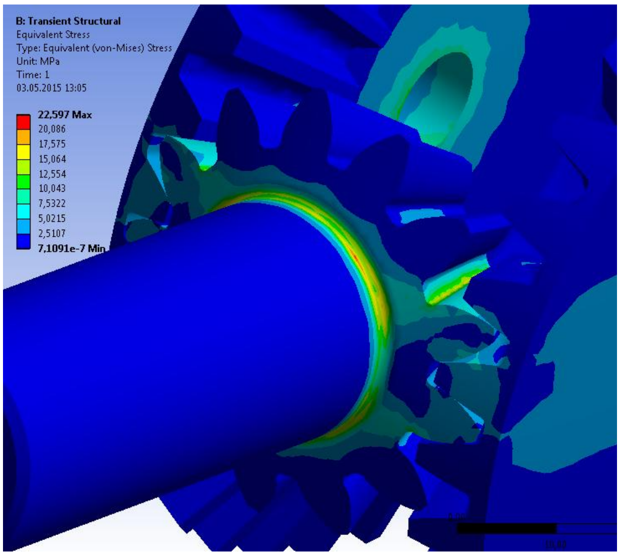


Fig. 14. Bevel gears equivalent stress distribution.

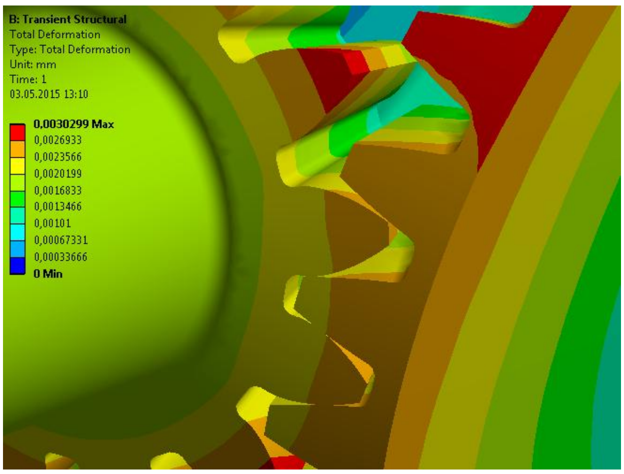


Fig. 15. Bevel gears elastic deformations distribution.

The mesh model of differential gears is shown in Fig.16.

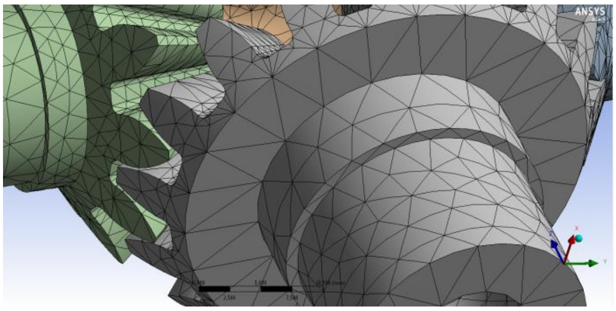


Fig. 16. Differential gears mesh model.

As load in Ansys simulation is used the joint load calculated in Adams simulation, by 3 (Nm), applied to gear 10. The model is solved and is obtained the results for equivalent stress distribution and elastic deformation, presented in Fig. 17 and 18. The maximum value of equivalent von Misses stress for this gear pair is 17.4 (MPa), and for elastic deformation is 0.0017 (mm).

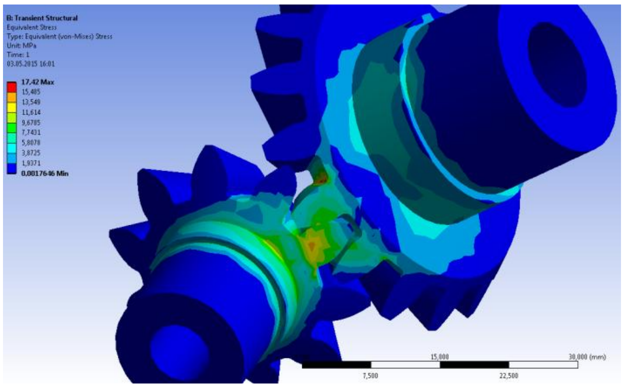


Fig. 17. Differential gears equivalent von Misses stress distribution.

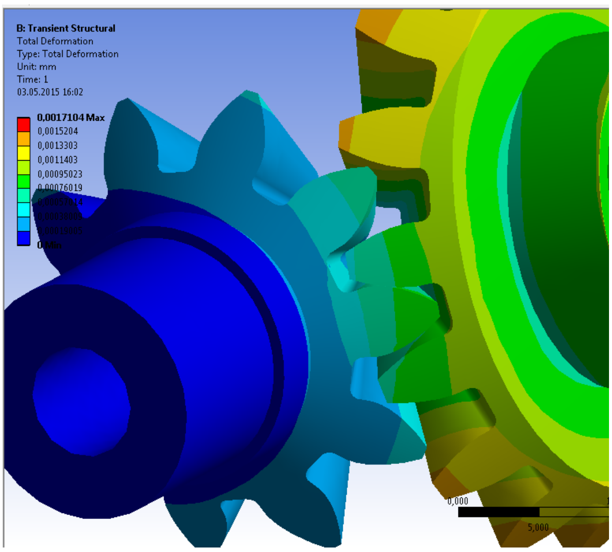


Fig. 18. Differential gears elastic displacement distribution.

VI. CONCLUSION

This paper proposes to develop a kinematic and structural analysis model of a robotic wheelchair. Proposed transmission uses differential gears for steering. The virtual model of the wheelchair is designed in Solid Works and upon this model it will be developed the dynamic model in Adams. Proposed solution uses two kinematic chains, one for straight line motion and second for steering motion. The solution uses two motors: straight line motion motor and steering motor. The motors can operate independently or in correlation, in this case the wheelchair can be drive with different steering radius. Based on the virtual model it is made a dynamic simulation in Adams which reveals motion trajectories of the wheelchair, kinematic and dynamic parameters of the system: running speed, propulsion and steering torque. From Adams simulation is determined that the propulsion necessary torque, when the wheelchair carries a 70 (kg) human, is 9 (Nm) and the steering torque is by 3,2 (Nm). Furthermore, based on Adams dynamic simulation is made in Ansys a transient structural analysis, which reveals the gears transmission solicitations. As a conclusion, in situation when the wheelchair is moving on a flat surface, the necessary torque for traction and steering are relatively low. Much greater values are expected in case of a hill climbing.

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