Design and Motion Study of a Wheelchair for Disabled People

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Abstract— **This paper intends to develop a kinematic and dynamic analysis for a wheelchair through a differential transmission model used for traction and steering. Wheelchairs enable disabled people to perform many activities of daily living thus improving their quality of life. Proposed solution uses for traction one DC motor with steps adjustable angular speed, and for steering one smaller motor. Is described the kinematic scheme of the proposed transmission and kinematic analysis. Upon this scheme is developed a CAD model of the transmission, mounted on a wheelchair. Simulations in MSC.Adams, are made in order to verify the functionality of the proposed solution. The obtained results validate proposed transmission model and enable success implementation of this design to a wheelchair experimental model.**

Index Terms—**Wheelchair, design, kinematics, dynamics, differential transmission.**

I. Introduction

THERE are many examples of assistive devices for people with manipulative and locomotive disabilities. These devices enable disabled people to perform many activities of daily living thus improving their quality of life. Disabled people are increasingly able to lead an independent life and play a more productive role in society. In the case of disabled children, such assistive devices have been shown to be critical to their cognitive, physical and social development [1, 9].

Wheelchair is still the best transportation means for disabled people, since its invention in 1595 (called an invalids chair) for Phillip II of Spain by an unknown inventor. They have since evolved into complex multi-degree-of-freedom mechanical and electro-mechanical devices and robotic systems [2, 3, 5, 6, 10, 11 and 12].

Despite rapid scientific and technological progress, there has been very little innovation in wheelchair design over the last 200-300 years. The folding wheelchair came in 1933, and powered wheelchairs were developed in the early 1970s [4, 7]. New materials such as plastics, fiber-reinforced composites and aluminum alloys have found their application into the design and manufacture of lighter, stronger and more reliable wheelchairs [9]. The wheelchair industry has also benefitted from the development of lighter, efficient, durable and reliable motors, better amplifiers and controllers and most important of all superior batteries.

Since the user is in physical contact with the chair for extended periods of time, the contact surfaces especially the seat, requires a certain degree of customization to ensure comfort. Commercially available standup wheelchairs afford better seating and reaching, relief from pressure sores, and better health. They also allow users to operate equipment designed to be operated by standing people and improve the quality of social interaction with non disabled standing people [18].

Conventional wheelchairs are difficult to maneuver in constrained spaces, because they only have two degrees of freedom (forward/backward movement and steering). However, the Alexis Omnidirectional Wheelchair [10], TRANSROVR and the European TIDE Initiative OMNI Wheelchair [11] can move omni-directionally by adapting non-conventional wheels developed for use by robotic vehicles for this application [9].

Many electric wheelchairs use some form of direct drive, involving belts, chains and/or gears. On a smooth level surface, relatively little torque is required to propel the wheelchair and occupant at constants speeds of up to 5 or 6 mph. If this were the only requirement, small motors of a few hundred watts capacity would serve. But wheelchairs must overcome obstacles, usually at low speed, climb substantial grades and accelerate at reasonable rate. All of these conditions demand high torque, most often at low speed. Thus, most powered wheelchairs are equipped with motors (and associated electronic controls), that are much larger than necessary for constant level propulsion and with poor efficiency most of the time since high torque is obtained only at low speed. To overcome this, a new infinitely variable automatic transmission, that automatically changes its speed ratio, in response to load torque being transmitted, is presented in [13].

This research brings arguments for a mechanical transmission that achieves the differential movement, on which the traction and the steering components are controlled separately by motors with suitable synthesis to achieve the proper angular speed difference of wheels.

II. Kinematics of Wheelchair Transmission

Kinematic scheme of wheelchair transmission is presented in Fig.1. Shaft I is actuated by straight line displacement electric motor M_1, steering is performed by steering motor M_2 placed on shaft V.

Bevel gears, (10, 12) and (10’, 12’) are planetary gears and they are mounted on shafts (IV, III), respectively (IV’, III’), with pins assembly. Bevel gears, (11, 13), and (11’,

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13') are satellites gears of differential mechanism. 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) [1]. To straight line displacement the traction wheels of the wheelchair rotate 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 gears 4-2 to wheels.

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. For left wheel Rs the transmission flow go through bevel gears (8, 9'), (10', 11'), (11', 12') to semi-axis III'. Those two wheels Rd and Rs will rotate with the same angular speed and opposite rotation sense. Casing S, S' and spur gear (2, 2') do not rotate (the motor M_1 is turned off).

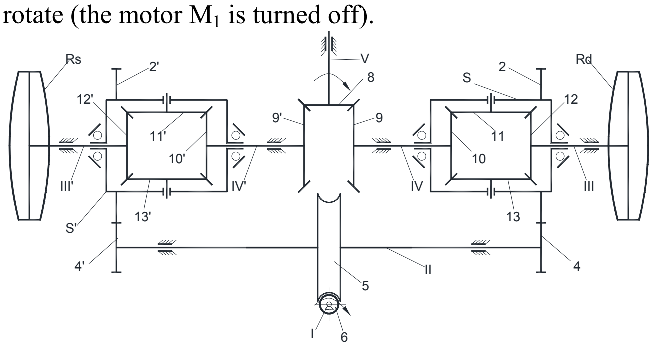

Fig.1. Kinematics scheme of wheelchair transmission.

A. Transmission ratios

The motion transmission chain for straight line displacement of the wheelchair is expressed by Eq. (1), and for steering the motion transmission chain is expressed by Eq. (2).

$$\begin{aligned} &M_1 - I - i_{65}^{w} - II - i_{42}^{c} - III - Rd \\ &M_1 - I - i_{65}^{w} - II - i_{4'2'}^{c} - III - Rs \end{aligned} \tag{1}$$

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 and III – shafts.

$$\begin{aligned} &M_2 - V - i_{89}^{k} - IV - i_{10\,11}^{k} - i_{11\,12}^{k} - III - Rd \\ &M_2 - V - i_{89'}^{k} - IV' - i_{10'\,11'}^{k} - i_{11'\,12'}^{k} - III' - Rs \end{aligned} \tag{2}$$

Where: M_2- steering motor; i_{89}^{k} $i_{89'}^{k}$ -bevel gear ratio; $i_{10\,11}^{k}$ -bevel gears 10, 11 ratio; $i_{11\,12}^{k}$ -bevel gears 11, 12 ratio; Rd and Rs, right and left wheel, III, IV and V – shafts.

For wheelchair straight line displacement when both drive wheels encounter the same resistance to ground, is valid Eq.(1). For steering, the governing equations of the wheels angular velocity are deducted in the following. Used notations are described below:

ω_{10}, ω_{12} -absolute angular velocity of bevel planetary gears 10 and 11, considered in relation to differential casing;

$\omega_{10}^{S} = \omega_{10} - \omega_S$ -relative angular velocity of gear 10, towards differential casing S;

$\omega_{12}^{S} = \omega_{12} - \omega_S$ -relative angular velocity of gear 12, towards differential casing S;

ω_{11}, ω_{13} - is relative angular velocity of satellite gears 11 and 13, towards differential casing S.

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

R_W - the rolling circle radius for planetary gears 10 and 12.

r_W - rolling circle radius for satellites gears 11 and 13.

According to the principle of gearing, the tangential velocities are equal in the gearing point:

$$-\left(\omega_{10} - \omega_s\right) R_W = \omega_{11} r_W = \left(\omega_{12} - \omega_S\right) R_W = \omega_{13} r_W \tag{3}$$

We obtain:

$$\omega_s = \frac{\omega_{10} + \omega_{12}}{2} \tag{4}$$

Namely, 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 Eq. (4) we obtain:

$$\omega_{12} = 2\omega_s = 2\omega_2 \tag{5}$$

In the same way: $\omega_{10}^{'} = 0$, we obtain:

$$\omega_{12'} = 2\omega_{S'} = 2\omega_{2'} \tag{6}$$

Taking into account the gears ratio, we write the equation, to calculate the motion received from the traction motor:

$$\omega_s = \frac{\omega_{M_1}}{i_{65}^{w} \cdot i_{42}^{c}} \left[rad / s \right] \tag{7}$$

Where:

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 curve displacement of wheelchair, we have the equation, for the motion received from steering motor M_2:

$$\omega_{10} = \frac{n_{M_2}}{i_{89}^{k}} \; ; \; \omega_{10}' = \frac{n_{M_2}}{i_{89'}^{k}} \tag{8}$$

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

$$\omega_{10} \cdot R_W = \omega_S \cdot R_W + \omega_{11} \cdot r_W \tag{9}$$

If we take into account the Eq. (4), we obtain the angular velocities for satellite gear (11) and planetary gears (10) and (12):

$$\begin{aligned} &\omega_{11} = \left(\omega_{10} - \omega_s\right) \frac{R_W}{r_W}, \\ &\omega_{10} = \omega_s + \omega_{11} \frac{r_W}{R_W}, \\ &\omega_{12} = \omega_s - \omega_{11} \frac{r_W}{R_W} \end{aligned} \tag{10}$$

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III. DESIGN OF WHEELCHAIR TRANSMISSION CAD MODEL

Based on the kinematic scheme of the transmission (Fig.1) it is performed the design calculations of gears. Input data, respectively powers and angular speeds on shafts are known, kinematic and dynamics parameters being establish. We calculate geometric elements of gears, upon we design the 3D models of gears, with GearTrax and Solid Works software. Also we performed transmission shafts and bearings calculation. Gears ratios and dimensions are given in Table I.

TABLE I
GEARS RATIOS AND DIMENSIONS

Gear pair	Z_2=34 Z_4=18	Z_8=13 Z_9=33	Z_6=1 Z_5=17	Z_{10}=16 Z_{11}=10	Z_{12}=16 Z_{13}=10
Gears ratio[-]	1,88	2,538	17	1,6	1,6
Gears module [mm]	3	3	3,5	2,5	2,5
Gears rolling diameter [mm]	d_{w2}=54,84 d_{w4}=101,5	d_{w8}=39,0 d_{w9}=99,8	d_{w6}=26,25 d_{w5}=58,8	d_{w10}=40 d_{w11}=25	d_{w12}=40 d_{w13}=25

Calculated data allowed us to realize the 3D model of designed transmission on wheelchair, which is presented in Fig.2. Differential transmissions have been mounted on the wheelchair structure, the assembly view being presented in Fig. 3. The weight of the entire assembly (Fig. 4) is 45 kg.

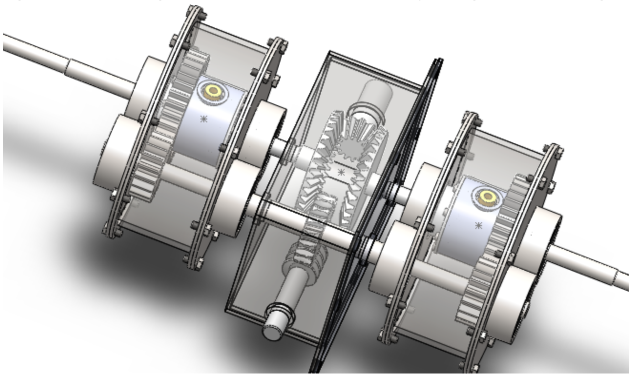

Fig.2. CAD model of differential transmission.

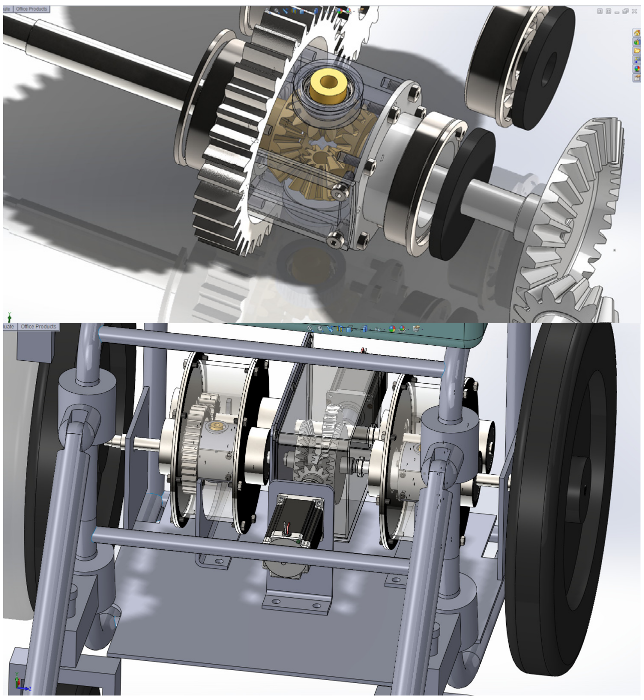

Fig.3. Wheelchair transmission assembly in Solid Works.

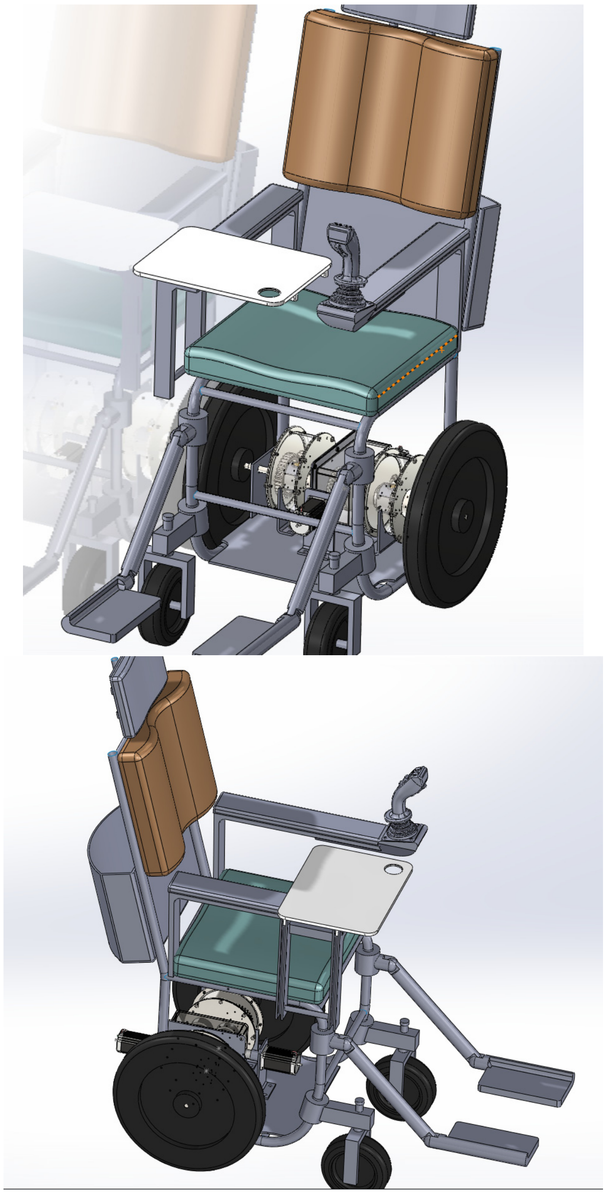

Fig.4. Wheelchair assembly model in Solid Works.

IV. MULTI-BODY DYNAMICS MODEL OF WHEELCHAIR

The virtual prototype of the wheelchair, as modeled in Fig. 4, is transferred into Adams multi-body model, using the transfer interface embedded in Solid Works. Trough a suitable modeling we perform the dynamic simulation of wheelchair assembly, see Fig. 5. Steps taken to completely define the dynamic model consist in: specify of kinematic elements mass properties, define of kinematics joints, specify of friction models (friction to joints, and wheels-ground contact), specify of human load by 70 kg and define of gear type connections (by defining the contact between gears solids).

The contact forces in the gear set are described by a contact mechanics model which is determined by parameters such as stiffness, force exponent, damping and friction coefficients, penetration depth. MSC.Adams has the capability of rigid-elastic model computation, which is suitable for the multi-body model of the gearbox. The rigid-elastic model is a compromise between the rigid body and elastic body, in which the shafts and gears body are taken as rigid, but the contact surfaces of the meshing gears are deformable bodies.

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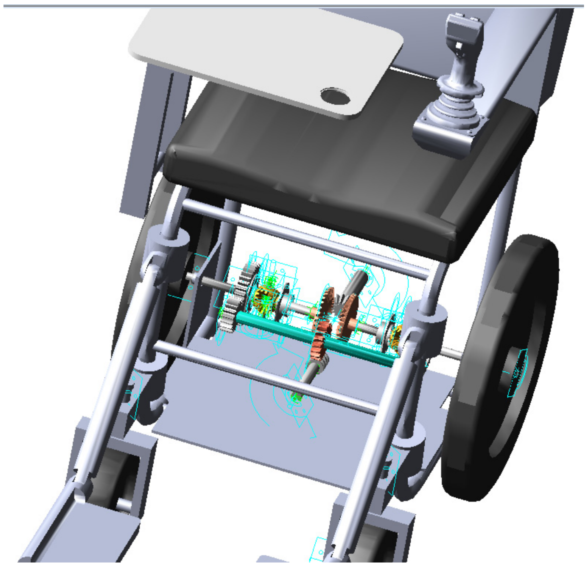

Fig.5. Dynamic model of wheelchair in MSC.Adams.

A. Contact parameters

Considering the computing efficiency and accuracy, is adopted the impact method to define gears contact. Necessary parameters for this method are shown in Fig. 7. The contact force computed by this method is composed of two components, the elastic force caused by the deforming components and the damping force caused by the relative deforming velocity.

The contact between ground and wheels is specified according to the literature: Ψ_a - 0.5…0.6 for old asphalt road, old concrete [1]. It should be noted that the front wheels are self-directed, being modeled by rotation joints with friction.

B. Establish of the parameters for a gearing pair contact model

The contact force model is shown in Fig.6. The contact force in Adams [8] can be expressed as:

$$F=\begin{cases} K\left(x_0-x\right)^e+CS\dot{x} & x<x_0 \\ 0 & x\geq x_0 \end{cases} \tag{11}$$

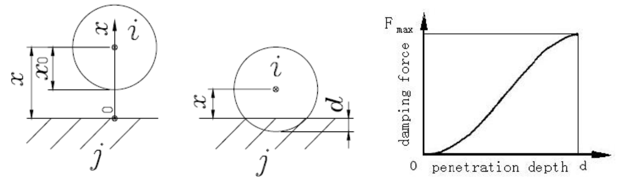

Fig.6. Contact force model in Adams [8].

In Eq.(11), x_0-x is the deformation in the process of contact-collision. Eq.(11) shows that the contact does not occur while $x \geq x_0$ and the contact force is zero. Contact occurs while $x<x_0$ and the contact force is related to the parameters such as stiffness *K*, the deformation x_0-x, contact force exponent *e*, damping coefficient *C* and the penetration depth *d* which is the maximum value of x_0-x. *S* is a step function defined as (12):

$$S=\begin{cases} 0 & x<x_0 \\ \left(3-2\Delta d\right)\Delta d^2 & ,x_0-d<x<x_0 \quad \cup \Delta d=x_0-x \\ 1 & x\leq x_0-d \end{cases} \tag{12}$$

In Eq. (12), Δd = x_0-x, is the deformation of the body. Eq.(12) also shows that the contact force defined in Adams is composed of two parts. An elastic component $K(x_0$-$x)e$, acts like a nonlinear spring. The other is the damping force *CS(dx/dt)*, which is a function of the contact-collision velocity. By the definition of the step function in Eq.(12) we know that the damping force is defined as a cubic function of penetration depth. To avoid the function discontinuity caused by the dramatic variation of the damping force while contact-collision occurs, as shown in Fig.6, the damping force is set to zero when the penetration depth of the two contacted bodies is zero, and approaches a maximum value F_{max} when the specified penetration depth *d* is reached.

a) Stiffness K: according to the Hertzian elastic contact theory [8], the stiffness of two contacted bodies could be described by a pair of ideal contacted cylindrical bodies, from the equivalent radius of engaged helical gear pair. Consequently, the stiffness could be expressed as:

$$K=\frac{4}{3}R^{\frac{1}{2}}E^*=\frac{4}{3}\left[\frac{id_1\cos\alpha_t\tan\alpha_t^{'}}{2\left(1+i\right)\cos\beta}\right]^{\frac{1}{2}}E^*$$

$$\frac{1}{E^*}=\frac{1-\upsilon_1^2}{E_1}+\frac{1-\upsilon_2^2}{E_2}$$

$$\beta_b=a\tan\left(\tan\beta\cos\alpha_t\right) \tag{13}$$

Where: d_1-diameter of standard pitch circle; *i*-gear ratio; E_1, E_2- Young's modulus of two contacting bodies; E* - equivalent Young's modulus; $\alpha_t^{'},\alpha_t$ - transverse pressure angle at engaged, standard pitch circle; υ_1,υ_2 -Poisson ration of the pinion and gear; β,β_b -helical angle at the pitch, base circle.

The materials for the pinions and gears of the transmission are alloy steel and the values for the Poisson ratio and the Young's modulus are: $\upsilon=0.3$ and $E=2.1\cdot10^5\,N/mm^2$. Through calculations, the stiffness values for the gear pairs in the drive line are inserted in the model, see Fig 7.

b) Force exponent: Considering numerical convergence and computation speed, a force exponent *e*=1.5 is determined [8].

c) Damping coefficient C, generally takes values 0.1%~1% of the stiffness K. For this simulation the damping coefficient is set to C=1000Ns/mm.

d) Penetration depth: The relationship between damping force and penetration depth is shown in Fig.6. In common cases, a reasonable value for penetration depth is 0.01 mm. We used *d*=0.1 mm, considering the numerical convergence in Adams.

e) Dynamic and static friction coefficient and viscous velocity. The materials for the engaged gears are alloy steel and the meshing pairs are lubricated. Typical values found in mechanical design handbooks are: static friction coefficient $\mu_s=0.1$; static transition velocity $v_s=1mm/s$, dynamic friction coefficient $\mu_d=0.08$ and friction transition velocity $v_d=10mm/s$.

In Fig. 7 are presented the specified parameters used to define contact and friction model in case of gear pair Z_{12} and Z_{13}.

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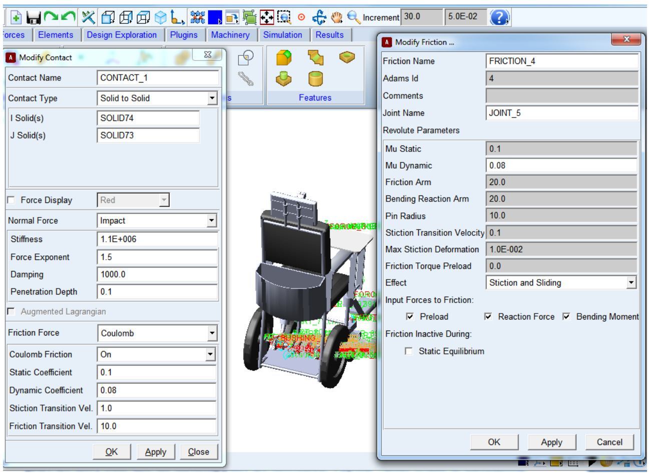

Fig.7. Wheelchair contact and friction parameters.

It is achieved robotic system workspace analysis in Adams, in two assumptions: a combined trajectory composed of a straight line motion with steering and only steering motion active for displacement in a circle. In the first case there are active both motions, straight line and steering. Functions used in Adams to define straight line motion and steering motion are given by Eq. 14.

IF(time-2: 35.9 , 0, IF(time-4: 0 , 35.9, 35.9)) - for traction motion (shaft I)

IF(time-2: 0 ,0 , 3.5) – for steering motion (shaft V) (14)

In the second case of wheelchair simulation, is active only the steering motion, with value $\omega_2 = 4\ \mathrm{rad/sec}$ (applied to shaft V). The simulation was achieved using WSTIFF solver with SI2 integration. The wheelchair motions trajectory obtained in both cases are presented in Fig. 8.

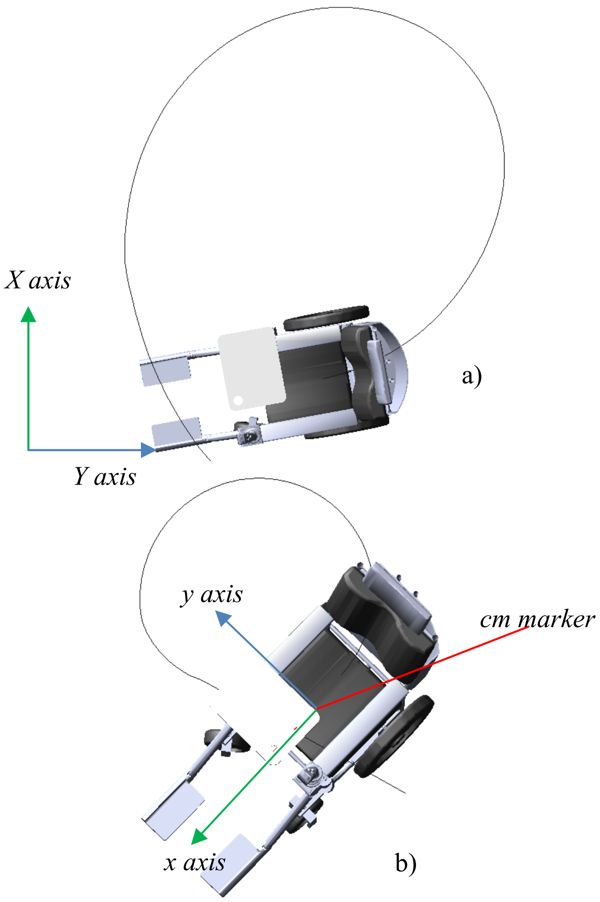

Fig.8. Wheelchair trajectory: combined motion (a) and steering motion (b).

In case of combined traction and steering motion simulation, is obtained the right (Fig. 9, a) and left wheel angular velocity (Fig. 9, b), (respectively, planetary gear 12 and 12'), presented in Fig. 9.

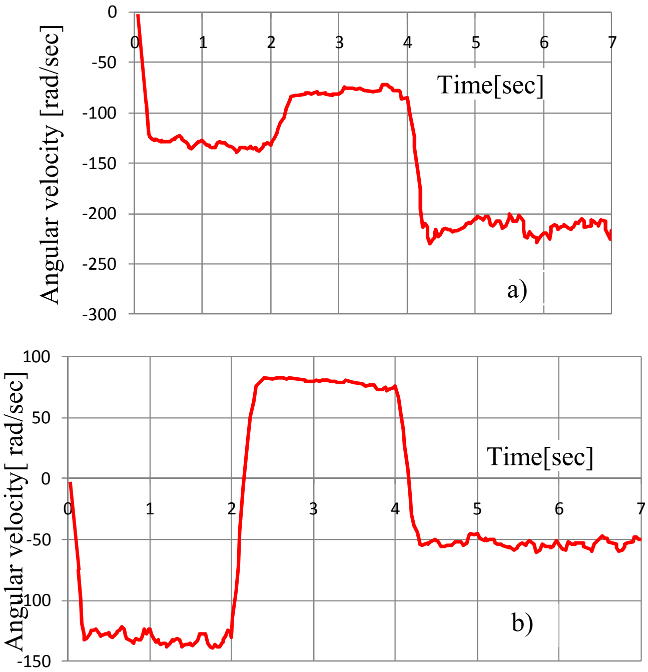

Fig.9. Right and left wheel angular velocity, in case of combined motion.

In case of combined motion trajectory, the wheelchair trajectory is presented in Fig. 8, a. The obtained torque variation, for the traction motor is presented in Fig. 10, a, and for steering motor the torque variation is shown in Fig. 10, b.

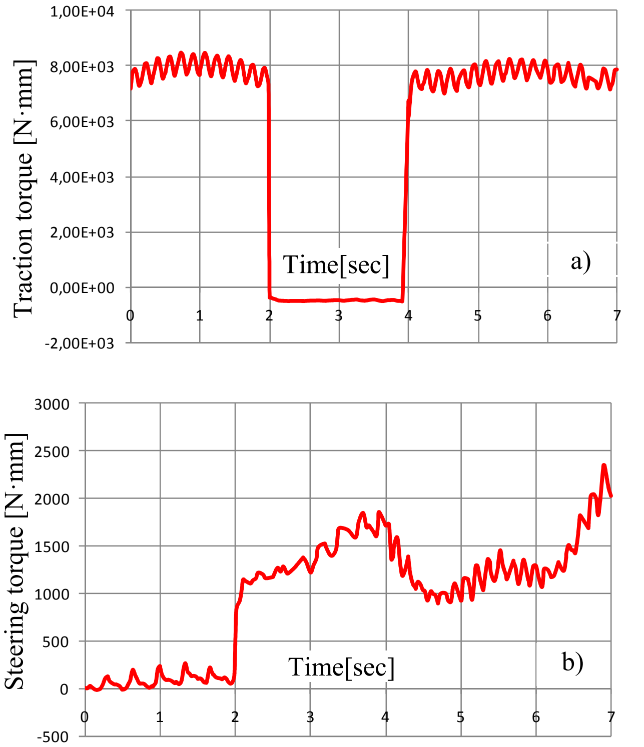

Fig.10. Traction and steering torque variation, in case of combined motion.

In case of steering motion simulation, is obtained the right (Fig. 11, a) and left wheel (Fig. 11, b) angular velocity

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(respectively, planetary gear 12 and 12'), presented in Fig.11. Is observed that for steering motion (when is rotate only motor M_2) the wheelchair wheels rotate with the same angular velocity but in opposite directions.

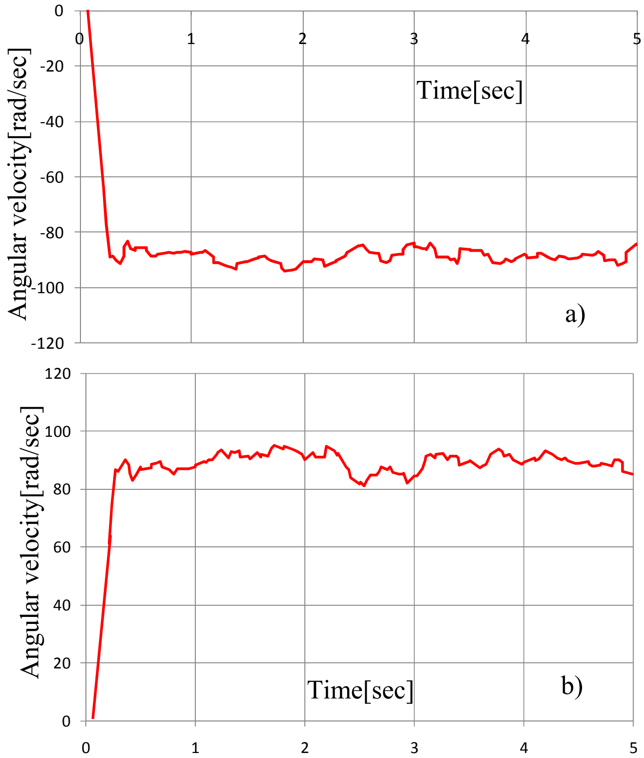

Fig.11.Right and left wheel angular velocity, in case of steering simulation.

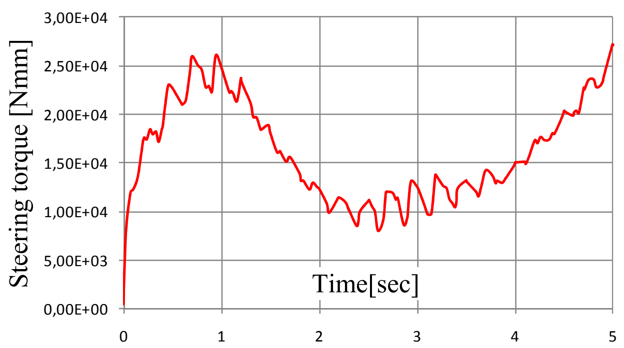

Fig.12. Steering torque variation, in case of steering simulation.

The wheelchair steering torque magnitude in case of steering motion simulation is shown in Fig.12.

V. Conclusion

In this paper is achieved the design solution of a robotic wheelchair. For straight line motion and steering the wheelchair uses a transmission with differential gearboxes. Based on the CAD modeling, it is developed a dynamic simulation in Adams, in order to validate the wheelchair prototype. The simulation is achieved in the case of a combined straight line and steering motion and only steering motion. In both cases are presented the wheelchair motion trajectory, gears angular velocity, wheelchair traction and steering motors computed torque. Is observed that the traction torque of the wheelchair, carrying a 70 kg human, is 8 Nm and the steering torque is by 30 Nm. Dynamic simulation performed in Adams demonstrates the efficiency of the proposed wheelchair transmission. Obtained torque values for traction and steering are suitable for this system. In order to decrease these values one solution is to use traction wheels with increased diameter.

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