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DYNAMIC MOTION ANALYSIS OF A WHEELCHAIR FOR PEOPLE WITH LOCOMOTOR DISABILITIES

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Abstract: *The paper presents the kinematics and dynamics of a vehicle designed for the rehabilitation of people with locomotor disabilities. The transmission that ensures the movement of the vehicle is composed of two kinematic chains, one for traction and the other one that ensures the movement in curves and the orientation of the mechanical system. The two power trains are driven by two geared motors. The mechanical system is steered by two differential groups. The gear transmissions used are composed of worm gears, bevel gears and spur gears. A dynamic analysis of the drive kinematic chain is made in which the main component is a cylindrical gear. The system of equations defining the motion of the gear is solved by the modal superposition method.*

Key words: *wheelchair, differential group, dynamic, traction, orientation.*

1. INTRODUCTION

The paper examines the dynamics of movement of a mechatronic system designed to serve people with locomotor disabilities.

In the paper [1] a wheelchair, capable of going up and down stairs, with a self-adaptive locomotion system is presented.

The wheelchair called invalid chair was invented in 1595 for King Phillip 2nd of Spain, with a continuous evolution into complex mechanical and electromechanical chairs with several degrees of freedom [2, 3, 5, 6, 10, 11, and 12].

Technical devices for locomotor assistance first appeared in the form of prosthetic limbs [7].

Wheelchairs used in locomotor assistance are almost 300 years old, having undergone a particular evolution as mechanical or mechatronic systems with several degrees of freedom [4].

Although science and technology have made major advances over time, there are still innovations in wheelchairs over the last 200-300 years. The folding chair solution appeared in 1933, and electrically powered wheelchairs have been around since 1970 [4].

The electrically powered solution increases the mobility of the chair's wheels, but the special problems imposed by certain obstacles limiting movement have not yet been solved [1].

There are mainly two methods used for chairs when moving up stairs: the IBOT method, which allows stairs to be climbed by changing the angle of movement and the center of gravity of the operator, and the crawler method, whereby locomotion is provided by a crawler system [9].

Design solutions for stair climbing chairs are complex and unsuitable for frequent use [13].

There are several innovative solutions for chairs capable of overcoming obstacles such as: the motorized wheelchair with hybrid locomotion systems (MWHLS), designed for better functional mobility for people with locomotor disabilities. The devices have two locomotion systems: a conventional locomotion system and a tracked locomotion system that can overcome obstacles such as stairs [14].

2. DESIGN OF MECHATRONIC SYSTEM

The solution adopted in the design of this vehicle is a mechatronic one, which incorporates three main components, namely, the mechanical component, the drive component and the command-and-control component.

The mechanical component is shown in Figure 1. It consists of two kinematic chains, one for traction and one for steering. The traction drive is based on two cylindrical gear drives (4-2 and 4'-2') and the steering drive is defined by two differential groups (10-11-12-13 and 10'-11'-12'-13').

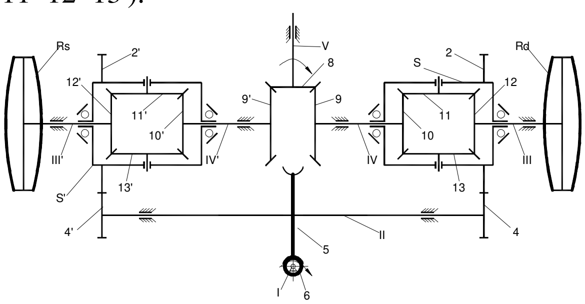

Fig. 1. Kinematics scheme of wheelchair transmission

Based on this kinematic scheme, a virtual 3D model of the whole assembly has been designed, as shown in figures 2, 3..5. The drive is provided by two gearmotors, one for traction and the other for orientation.

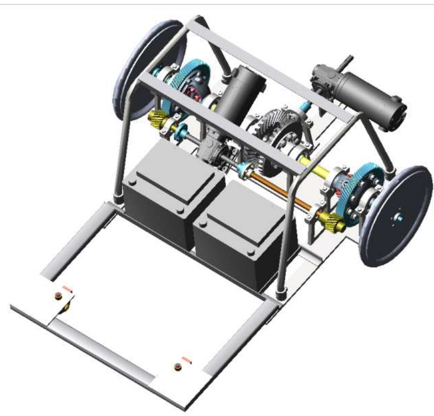

Fig. 2. Assembled 3D model of the mechatronic system

For the steering gear, the input to the two differential gears is provided by a bevel gear system (figure 6).

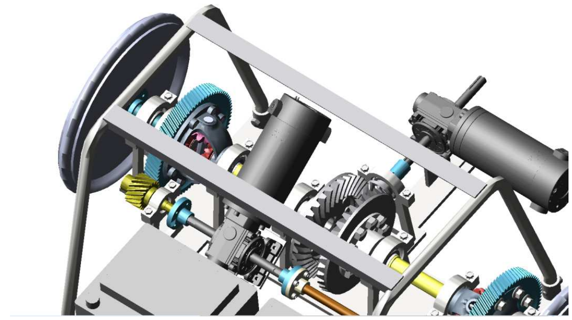

Fig. 3. Assembled 3D model of the mechatronic system (detail)

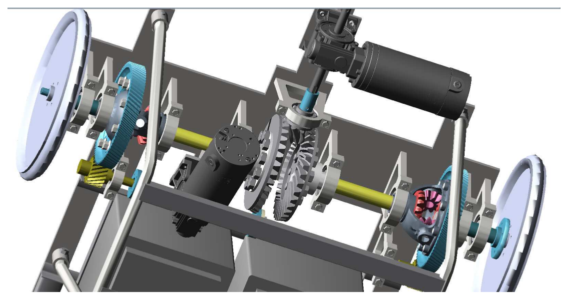

Fig. 4. Assembled 3D model of the mechatronic system (detail)

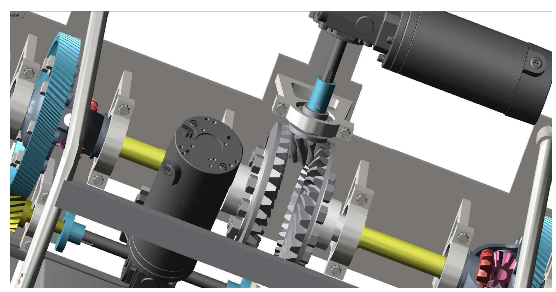

Fig. 5. Assembled 3D model of the mechatronic system (detail)

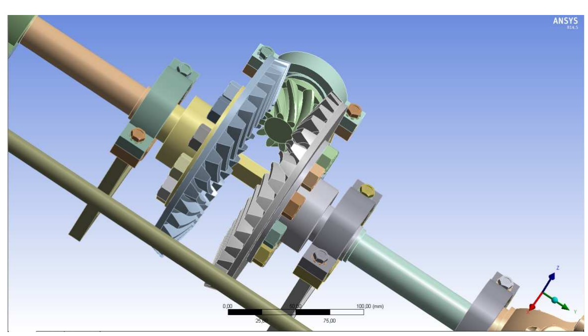

Fig. 6 Transmission for turning

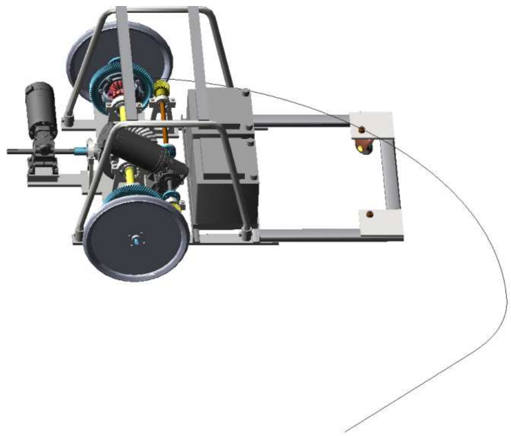

Fig.7. Vehicle movement trajectory

3. KINEMATIC ANALYSIS OF VEHICLE MOTION

Virtual prototyping of the vehicle was carried out using the SolidWorks - Adams software interface.

The kinematic elements were defined by local reference systems and mass and inertial properties, kinematic torques and driving elements. Motion monitoring was performed over a time interval of 8 seconds with straight line and curved displacements.

Thus, figure 7 shows the trajectory of the vehicle motion over an 8 second interval (straight and curved motion). This curve certifies that the vehicle meets the purpose for which it was designed.

Time variation curves have been identified for the angular velocity components in the steering (Fig. 8, 9, 10) and traction gears of the vehicle (Fig. 11, 12) respectively. It can be seen that the angular velocity components vary differently and are influenced by the variation of the angles in the driving torques (Fig. 13).

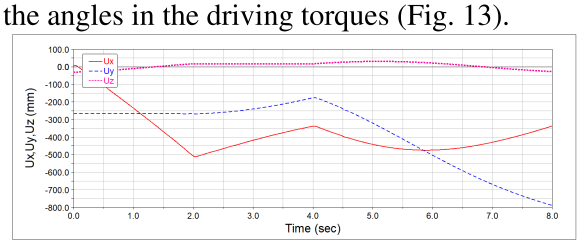

Fig.8. Displacement components along the x,y,z axes

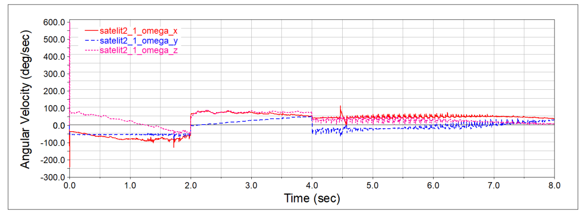

Fig.9. Variation of angular velocity for satellite gear

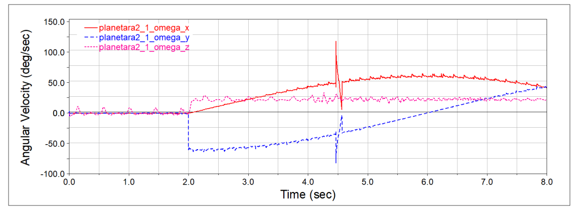

Fig.10. Variation of angular velocity for planetary gear

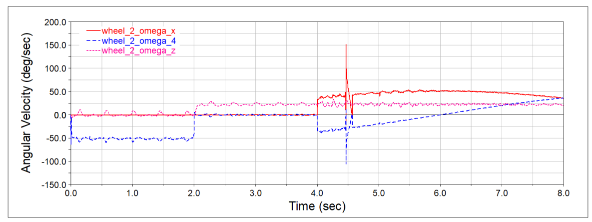

Fig.11. Variation of angular velocity for wheel 2

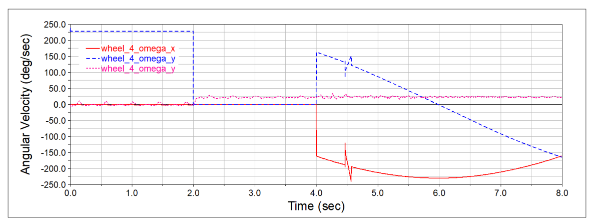

Fig. 12. Variation of angular velocity for wheel 4

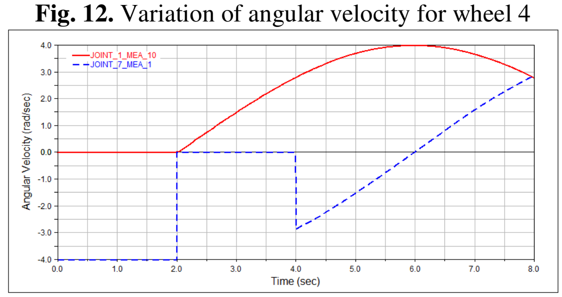

Fig. 13. Angular speed variation diagrams in motor couplings

From the figure, the two sequences of the motor movement can be observed as follows:

Motion 1 (JOINT_1_MEA10);IF (time-2:0 ,0 , 4) - actuates the orientation kinematic chain.

Motion 2 (JOINT_7_MEA1); IF (time-2:4 , 0,IF(time-4:0, 4,4)- actuates the kinematic chain of traction. For the orientation kinematic chain, the motion control is provided by the diagram (JOINT_1_MEA10) and for the traction kinematic chain by the diagram (JOINT_7_MEA1).

Figure 13 shows that in the (0:2)second interval the system moves in a straight line and in the (2:8)second interval it moves in a curve.

4. DYNAMIC GEAR ANALYSIS

For the kinematic scheme in Fig. 1, a dynamic analysis of the fixed axle cylindrical gearing formed by gears 2 and 4 is proposed. The worm gear composed of worm 6 and worm wheel 5 is part of a geared motor driving shaft II. Geometrical gears parameters are presented in Table 1.

Table 1. Design parameters of gears

Gear	*No. of tooth*	*Normal module [mm]*	*Gear*	*No. of tooth*	*Normal module [mm]*
2	34	3	*8*	13	3
4	18	3	*9, 9'*	33	3
5	17	3.5	*10,12*	16	2.5
6	1	3.5	*11,13*	10	2.5

Thus, to develop the dynamic model of the cylindrical gear we consider a system composed of an engine (m), the cylindrical gear (4,2) and the loading system (L).

The shafts S_1 and S_2 considered as torsion springs with stiffnesses k_1 and k_2.

$\theta_m, \theta_1, \theta_2, \theta_L$- generalized coordinates for the motor, gear 1, gear 2 and loading system.

J_m −the equivalent moment of inertia for the engine and the loading system respectively J_L.

J_1, J_2 −the mechanical moments of inertia for wheel 1 and wheel 2;

J_L -equivalent moment of inertia for the load;

$\vec{q}$ - generalized coordinate vector.

$$\vec{q} = [\theta_m \quad \theta_1 \quad \theta_2 \quad \theta_L]^T \tag{1}$$

The equation of motion is:

$$[M]\{\ddot{\theta}\} + [C]\{\dot{\theta}\} + [K]\{\theta\} = \{F\} \tag{2}$$

$[M]$- mass matrix;$[C]$- damping matrix;$[K]$- stiffness matrix.

The solution of the differential equation (2), for which the coefficients we take to be constant, is of the form:

$$\{\theta\} = \{x\}\, sin(\omega_t + \varphi) \tag{3}$$

We derive:

$$\{\dot{\theta}\} = +\{x\}\omega \cdot \cos(\omega_t + \varphi) \tag{4}$$

$$\{\ddot{\theta}\} = -\{x\}\omega^2 \cdot \sin(\omega_t + \varphi) \tag{5}$$

The differential equation, without damping and external loads, will be:

$$[M]\{\ddot{\theta}\} + [K]\{\theta\} = 0 \tag{6}$$

$$([K] - \omega^2[M])\{X\}\, cos(\omega_t + \varphi) = 0 \tag{7}$$

We get:

$$([K] - \omega^2[M])\{X\} = 0 \tag{8}$$

Equation (8) can be written as follows:

$$[K]^{-1}[M]\{x\} = \left(\frac{1}{\omega^2}\right)\{x\} \tag{9}$$

This is an eigenvalue problem for the matrix $[K]^{-1}[M]$.

For a system with n degrees of freedom we have "n" eigenvalues λ and "n" sets of eigenvectors $\{\phi_j\}, j = 1, \overline{n}$.

For every "j" we have:

$$[K]^{-1}[M]\{\phi_j\} = \lambda_j\{\phi_j\} = \left(\frac{1}{\omega^2}\right)\{\phi_j\} \tag{10}$$

ω_j= circular natural frequency;

$$\omega_j = \frac{\pi n_j}{30}\,[[rad/s]] \tag{11}$$

$f_j = \frac{\omega_j}{2\pi}(cycle/sec)$- natural frequency. (12)

$\{\phi_j\}-$ eigenvectors representing the modes of vibration the assumed shape of the structure when vibrating with the frequency f_j,ω_j and $\{\phi_j\}$ satisfy the equation:

$$\omega_j^2[M]\{\phi_j\}=[K]\{\phi_j\} \quad (13)$$

To solve the system of equations of motion (2), the modal superposition method will be used.

This method allows to decouple the equations of motion by a linear transformation of the form:

$$\theta_i(t)=\sum_{j=1}^{n}\phi_{ij}a_j(t), i=\overline{1,n} \quad (14)$$

$\phi_{ij}-$ the coordinate of the modal form representing the position of mass "*i*" in module "*j*";

$a_j(t)-$ generalized coordinate representing the time variation of the response in the mode "j";

$\phi_{ij}-$ depends on position and not time;

$a_j(t)-$ depends on position and not time;

In matrix form:

$$\{\theta\}=[\phi]\cdot\{a(t)\} \quad (15)$$

$[\phi]$ –the modal matrix, in which modal forms are written in columns;

We accept that the vector of modal forms is orthogonal to the matrix of masses, i.e:

$$[\phi]^T[M][\phi]=I \text{ unit matrix} \quad (16)$$

Let's reconsider the equation:

$$[M]\{\ddot{\theta}\}+[K][\theta]=\{F\} \quad (17)$$

We derive the relationship twice:

$$\{\ddot{\theta}\}=[\phi]\{\ddot{a}(t)\} \quad (18)$$

Introduce (18) in (17):

$$[M]\cdot[\phi]\cdot\{\ddot{a}(t)\}+[K]\cdot[\phi]\cdot\{a(t)\}=\{F(t)\} \quad (19)$$

We multiply to the left of relation (19) by $[\phi]^T$ and given relation (16), we obtain:

$$\{\ddot{a}(t)\}+[\phi]^T[K][\phi]\{a(t)\}=[\phi]^T\{F(t)\} \quad (20)$$

The relation (13), can be written as follows:

$$\omega^2[M][\phi]=[K][\phi] \quad (21)$$

The relation (21) multiplies to the left by $\{\emptyset\}^T$ and given (6), we will have:

$$[\omega^2]=[\phi]^T[K][\phi] \quad (22)$$

where:

$$[\omega^2]=diag[\omega_1^2,\omega_2^2,\dots\omega_n^2] \quad (23)$$

We insert (22) into (20) and obtain:

$$\{\ddot{a}(t)\}+[\omega^2]\{a(t)\}=[\phi]^T\{F(t)\} \quad (24)$$

Relation (24) represents the set of decoupled, independent differential equations.

The differential equation, without damping and external loads, is:

$$[M]\{\ddot{\theta}\}+[K]\{\theta\}=\{0\} \quad (25)$$

To obtain a solution different from the banal solution, the condition is:

$$det([K]-\omega^2[M])=0 \quad (26)$$

The dynamic model of the system consisting of gearwheels 4 and 2, which drive the traction kinematic chain is shown in Figure 14.

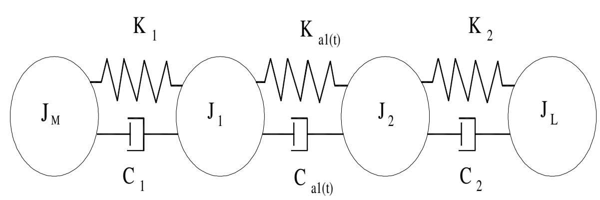

Fig. 14. Dynamic model for traction.

The dynamic model of the gearing (Fig. 15) is based on the gear stiffness and the damping coefficient [15]. If we take into account the presence of gearing and damping factors the dynamic model can be defined by the following system of differential equations, noted (27).

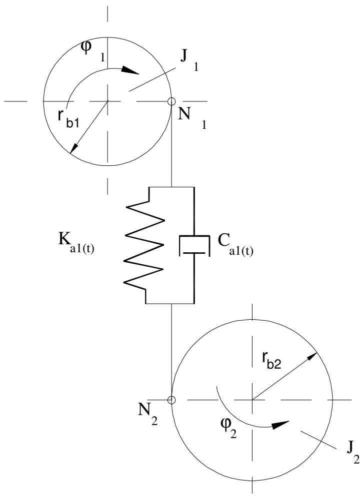

Fig. 15. Dynamic gear model.

$$\begin{aligned} & J_m \cdot \varepsilon_m + C_1 \cdot (\omega_m - \omega_1) + K_1 \cdot (\theta_m - \theta_1) - T_m = 0 \\ & J_1 \cdot \varepsilon_1 + C_1 \cdot (\omega_1 - \omega_m) + C_{a1} \cdot (rb_1 \cdot (rb_1 \cdot \omega_1 - rb_2 \cdot \omega_2)) + \\ & K_1 \cdot (\theta_1 - \theta_m) + K_{a1} \cdot (rb_1 \cdot (rb_1 \cdot \theta_1 - rb_2 \cdot \theta_2)) - T_1 = 0 \\ & J_2 \cdot \varepsilon_2 + C_2 \cdot (\omega_2 - \omega_L) + C_{a1} \cdot (rb_2 \cdot (rb_2 \cdot \omega_2 - rb_1 \cdot \omega_1)) + \\ & K_2 \cdot (\theta_2 - \theta_L) + K_{a1} \cdot (rb_2 \cdot (rb_2 \cdot \theta_2 - rb_1 \cdot \theta_1)) - T_2 = 0 \\ & J_L \cdot \varepsilon_L + C_L \cdot (\omega_L - \omega_2) + K_2 \cdot (\theta_L - \theta_2) - T_L = 0 \end{aligned} \quad (27)$$

$\varepsilon_m, \varepsilon_1, \varepsilon_2, \varepsilon_L$- angular accelerations for the drive shaft, wheel 1, wheel 2 and the loading system (running wheel).

Base circle radii of gear wheels are:

$$rb_1 = \frac{d_1 \cdot cos(\alpha_t)}{2}; rb_2 = \frac{d_2 \cdot cos(\alpha_t)}{2} \quad (28)$$

I_{p1}, I_{p2} - Polar moments of inertia:

Torsional stiffness of shafts:

$$K_1 = \frac{G_1 \cdot I_{pI}}{l_1}; K_2 = \frac{G_2 \cdot I_{pII}}{l_2} \quad (29)$$

Gear stiffness:

$$K_{a1} = \frac{4}{3} \cdot \frac{\sqrt{\frac{i \cdot d_1 \cdot cos(\alpha_t) \cdot tan(\alpha_{tp})}{2 \cdot (1+i) \cdot cos(\beta)}}}{\left(\frac{(1-\nu_1^2)}{E_1} + \frac{(1-\nu_2^2)}{E_2}\right)} \quad (30)$$

The damping coefficient for shaft 1, is:

$$C_1 = 2 \cdot \xi_1 \cdot \sqrt{\frac{K_1}{\frac{1}{J_m} + \frac{1}{J_1}}} \quad (31)$$

The damping coefficient for shaft 2, is:

$$C_2 = 2 \cdot \xi_2 \cdot \sqrt{\frac{K_2}{\frac{1}{J_2} + \frac{1}{J_3}}} \quad (32)$$

The damping coefficient for the gear is:

$$C_{a1} = 2 \cdot \xi \cdot \sqrt{\frac{K_{a1}^2}{\frac{rb_1^2}{J_1^2} + \frac{rb_2^2}{J_2^2}}} \quad (33)$$

$\xi = 0{,}005\ldots0{,}007$ [16]- the damping factor for shafts;

$\xi = 0{,}03\ldots0{,}17$ [17], [18]- damping coefficient for gearing.

The stiffness matrix constructed from the system of differential equations (17) is:

$$K = \begin{pmatrix} K_1 & -K_1 & 0 & 0 \\ -K_1 & K_1 + K_{a1} rb_1^2 & -K_{a1} rb_1 rb_2 & 0 \\ 0 & -K_{a1} rb_1 rb_2 & K_2 + K_{a1} rb_2^2 & -K_2 \\ 0 & 0 & -K_2 & K_2 \end{pmatrix} \quad (34)$$

Mass matrix:

$$M = \begin{pmatrix} J_m & 0 & 0 & 0 \\ 0 & J_1 & 0 & 0 \\ 0 & 0 & J_2 & 0 \\ 0 & 0 & 0 & J_L \end{pmatrix} \quad (35)$$

Matrix of forces:

$$F = [T_m \quad T_2 \quad T_3 \quad T_L]^T \quad (36)$$

The solution of the system of equations (4) is:

$$\{\theta\} = [\phi]\{a(t)\} \quad (37)$$

Generalized coordinate vector:

$$\{a(t)\} = [a_1(t), a_2(t), a_3(t), a_4(t)]^T \quad (38)$$

Corresponding to this vector we have:

$\{\ddot{a}(t)\} = [\ddot{a}_1(t), \ddot{a}_2(t), \ddot{a}_3(t), \ddot{a}_4(t)]^T$-the acceleration vector; (39)

5. NUMERICAL PROCESSING OF MATHEMATICAL MODELS

Input data:

d_I=14[mm]; d_{II}=26[mm]- diameters of shafts I and II; l_1=400[mm]; l_2=525[mm]- shaft lengths;

G_1, G_2 –transverse modulus of elasticity;

E_1, E_2 – longitudinal modulus of elasticity;

$\nu_1 = 0.3$; $\nu_2 = 0.3$; - transverse contraction coefficients of the material;

$i = 1.8$ - transmission ratio; $\beta = 0$- the angle of inclination of the teeth; $\alpha_t = 20^{\circ}$ - pressure angle;

d_1, d_2 – the division diameters of wheels 4 and 2. Moments of inertia have the following values: $J_m = 804.57; J_1 = 92.213; J_2 = 8180.15; J_L = 4 \cdot 10^4$;

Moments of torsion: T_m=75000[N.mm]- driving momentum; T_1=73500[N.mm]- momentum on the driving wheel;

T_2=72397.5[N.mm]- momentum on the driven wheel; T_L=70587.56[N.mm]- resistant moment;

Numerical processing of mathematical models: Determine the natural frequencies
f_1=1652.198669[Hz]; f_2=66.93996166[Hz]; f_3=30.42935750[Hz]; f_4=0.000807684[Hz];

Eigen modes of vibration (modal matrix)

$$\varnothing = \begin{vmatrix} 0.0003476 & -0.2299637 & -0.9988369 & 0.6249324 \\ -0.9997741 & 0.8561547 & -0.02401082 & 0.6249324 \\ 0.0212512 & 0.4543013 & -0.01107430 & 0.3308465 \\ -0.5656e-5 & -0.0878973 & 0.04031868 & 0.3308465 \end{vmatrix} \quad (40)$$

Identify the own forms of vibration as follows

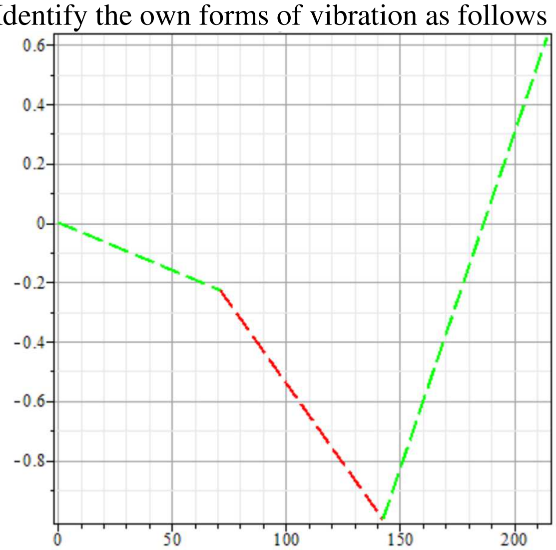

Fig. 16. Vibration mode 1

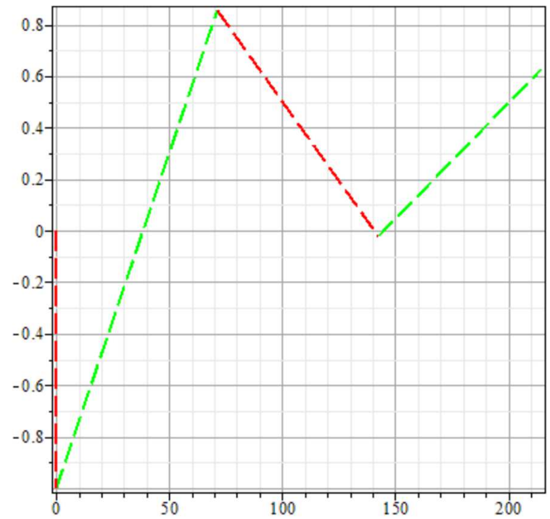

Fig. 17. Vibration mode 2

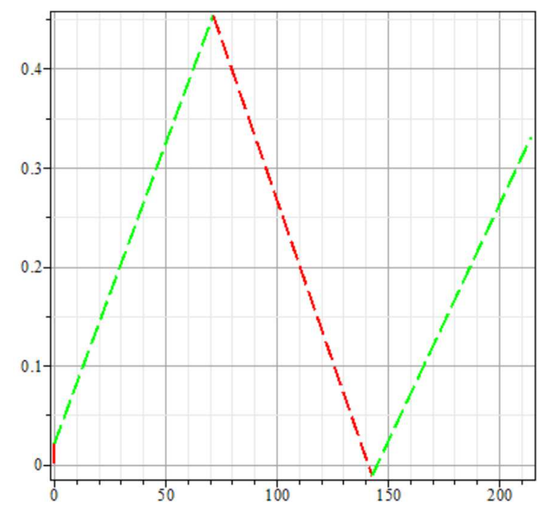

Fig. 18. Vibration mode 3

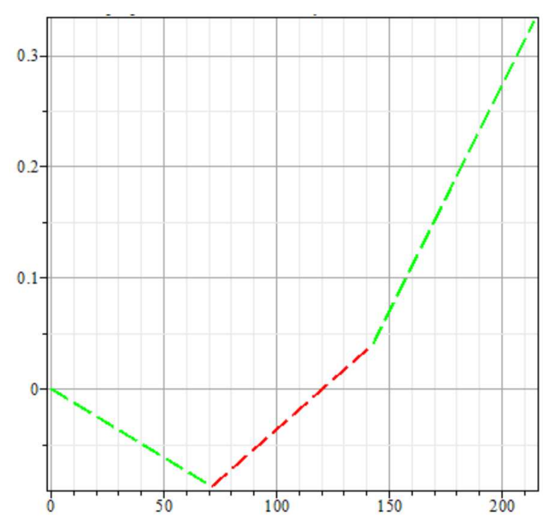

Fig. 19. Vibration mode 4

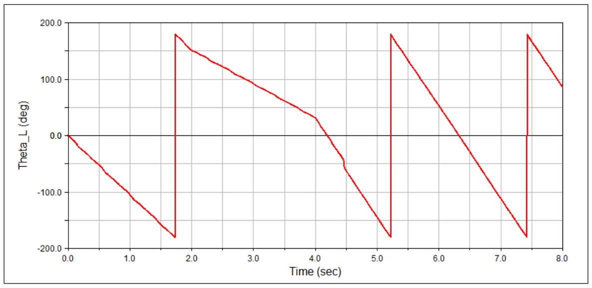

Fig.20 Variation of rotation angle θ_L

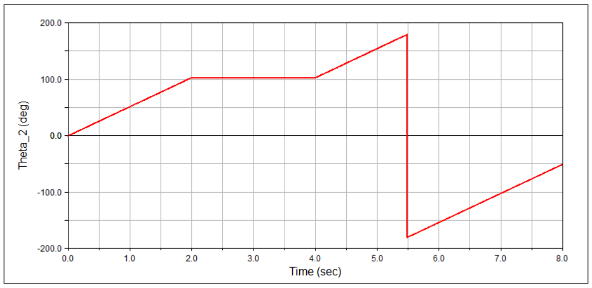

Fig.21 Variation of rotation angle θ_2

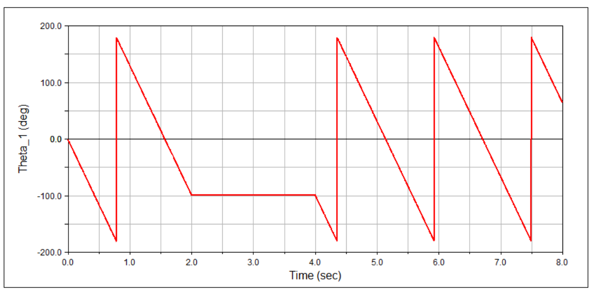

Fig.22 Variation of rotation angle θ_1

The generalized coordinate variation diagrams were obtained by processing mathematical models based on the modal superposition method.

The dynamics of gear motion is defined by systems of differential equations (27).

In the construction of the mathematical model, shaft stiffness, gear stiffness, mechanical moments of inertia, kinematic parameters, loads and generalized coordinates were considered.

Using the modal superposition method the solutions of the systems were determined analytically.

The obtained solutions are expressed graphically by the diagrams in figures (20), (21) and (22).

It can be seen that the variation curves are influenced by the motion laws imposed on the vehicle for the two displacement cases, namely traction and yaw.

Four natural frequencies and correspondingly four modes of vibration have been identified as shown in figures (17), (18), (19) and (20).

6. CONCLUSION

In the first part of the paper the kinematics of vehicle motion for straight and curved motion was studied.

The laws of motion in the driving torques were defined sequentially over an interval of 2 and 4 seconds respectively.

Time variation diagrams were identified for the main kinematic parameters of some components of the vehicle structure (wheelchair).

Thus, time variation curves were obtained for the angular velocity components of the gears in the drive train structure (4,2) and of the wheels (planetary, planetary) in a differential group structure.

For motion monitoring, the contact between the running wheels and the supporting surface and the contact forces in the gears were defined.

In this context small jumps in the variation of kinematic parameters can be neglected.

In the second part, a dynamic model was constructed for the study of the main gear in the kinematic chain structure.

A program has been designed under the Maple programming environment, which allows to solve numerically the system of differential equations, based on the modal superposition method. Thus, the natural frequencies and eigen modes of vibrations have been determined, the equations of motion have been decoupled and the laws of variation in time have been determined for the generalized coordinates.

The dynamic model of the gearing expressed by systems of differential equations (27), is important in that it takes into account the stiffnesses of the components in the systems and the damping factors.

Thus this dynamic model can also be reconsidered for the orientation chain based on two differential mechanisms.

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ANALIZA DINAMICĂ A MIŞCĂRII UNUI SCAUN CU ROTILE DESTINAT PERSOANELOR CU DIZABILITAȚI LOCOMOTORII

Rezumat: În lucrare sunt prezentate elemente cinematica si dinamica mişcării unui vehicul destinat reabilitării persoanelor cu deficiente locomotorii

Transmisia care asigură mişcarea vehiculului se compune din 2 lanțuri cinematice, unul de tracțiune și altul care asigură deplasarea în curbă, respectiv orientarea sistemului mecanic.

Acționarea celor două lanțuri cinematice este asigurată de două moto-reductoare. Orientarea sistemului mecanic este asigurată de două grupuri diferențiale.

Sunt folosite transmisii cu roți dințate compuse din angrenaje melcate, angrenaje conice și angrenaje cilindrice.

Se face analiza dinamică a lanțului cinematic de tracțiune la care componenta principală este un angrenaj cilindric.

Sistemul de ecuații care definește mişcarea angrenajului se rezolvă prin metoda superpoziției modale.

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