TECHNICAL UNIVERSITY OF CLUJ-NAPOCA

ACTA TECHNICA NAPOCENSIS

Series: Applied Mathematics and Mechanics
Vol. 54, Issue I, 2011

MECHANISMS POSITIONED IN A NEIGHBOURHOOD OF THEIR SINGULAR POSITIONS - VELOCITY AMPLIFICATION IN THE ELECTROMAGNETIC DAMPING. PART 2. NUMERICAL VERIFICATIONS

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Abstract: *In the paper, studies on the electromagnetic damping are presented. A continuous mechanical system and its vibrations are considered. To amplify the damping, a double-bar mechanism is introduced. Its position is set to a neighbourhood of its singular (kinematically) position. To improve its properties, tests are performed on a related numerical model. Two structurally different segments have to be considered in the model. The elastic part is composed of finite elements. The mechanism is modelled as a multibody system. Constraint equations joint the sub-models. The free mode vibrations are estimated. Its energy is converted into the electrical current (by use of a DC generator) and dissipated. As the fundamental task is to keep its work configurations in the singularity neighbourhood, an additional automatic control of the arms length is considered. Results are presented to confirm the effectiveness of the damping method.* ***Key words:*** *electromagnetic damping; velocity amplification; singular position; multibody modelling; finite elements; constraint equations.*

1. INTRODUCTION

In the paper, vibrations of a continuous mechanical system are studied. An effective damping method is searched. At present, the focus is set on electromagnetic damping elements and on velocity amplification methods, as well. According to the task, three sub segments can be distinguished in the considered system. To illustrate it, its general sketch is presented in Fig. 1.

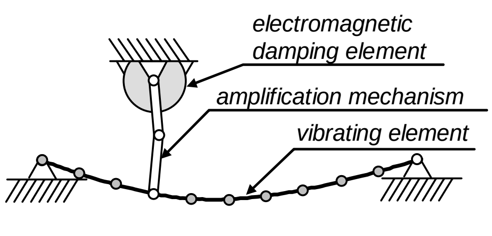

Fig. 1. Main elements of the considered system

As mechanical vibrations are commonly observed in their everyday experiences, industrial significance of the problem is substantial. Except of the some exceptional cases (when vibrations are generated intentionally), if they are present, they are non-required. Their negative outcomes are commonly cited. The uncontrolled motion of the effectors is the main unrequited outcome. The phenomenon complexity has to be underlined, as few sources can be responsible for such disturbances. The manual (or automatic) control of the system can be noised, as vibrations indicate a noise signal introduced in the control loop [1]. In-between the rest of the outcomes, significant acoustic noises have to be pointed. Motions of the surrounding elements can be evoked, when synchronized in frequencies. Some long-term effects are commonly cited. Harmful influence on the human health [2-4], and the fatigue effects on the mechanical elements are the examples. Summarising it, a search for an effective damping method is the essential topics in the most of the industrial design processes.

Within the presently used applications, the methods based on the viscous dampers look as the dominant method. However, even if successfully applied in the medium size applications (as vehicles for example), they are difficult to operate in the small size applications. There, the electromagnetic effects look as the challenging ones [5-7]. Such devices are favoured by their simpler construction, lower dimensions, easier semi active control and luck of the potential oil impurities.

In the case of the low frequency vibrations, the structural damping is ineffective. Velocities of the related motions are relatively low, and low damping forces are present. Even when some external damping is present, such frequencies are damped badly. To accelerate the damping, velocity amplification can be useful. The one considered below focuses on a double-bar mechanism introduced between the vibrating element and the damper (Fig. 1). The effective amplification is obtained as the mechanism configuration is close to its singular position (position of its kinematical singularity). According to the singularity, each motion of the vibrating element (even insignificant) effects in a significant motion at the mechanism joints. With the velocity amplification, the damping forces increase, simultaneously. However, as a neighbourhood of the mechanism singular position is considered, non-linear relations are necessary to describe the mechanism vibrations (variations of the inertial effects have to be considered in the mechanism). Even if small range vibrations are considered the non-linear effects are significant. At the time characteristics of the system vibrations, the non-symmetrical forms are present (see Fig. 11a).

Let us to remain that the part one of the paper was devoted to present the background of the used methodology. With the method presented there, model the mechanical structure of the considered system is prepared now. However, as some additional elements have to be presented, in the present part, six sections are proposed. Initially, a model of a standard DC-motor / DC-generator is considered. Details of their dynamics equations are presented. They refer to the classical modelling method recalled from [10, 11]. The model is based on the Kirchhoff's low of circuit's voltages. To dissipate the electrical power, the circuit is closed via a single resistance. Next, base it on the flux of the stator's magnetic filed, and the actual armature current, the mechanical torque is calculated for the motors shaft.

Section three is introduced to present the control algorithm. Again, as the focus is set on the amplification effectiveness, relatively elemental, no sophisticated algorithm is proposed at the present state of the research.

Numerical data of the considered system are presented in section four. As the focuses are set on the proposed amplification method, detail modelling is not the critical point of the presentation. Simplifications are justifies. Low discretization of the continuous element is accepted. Lowly sophisticated mechanism is employed. Then, numerical results are described in section five. Conclusions and perspectives are presented in section six.

2. DC MOTOR DYNAMICS

In the introduced model, the considered DC motor (Fig. 2), is composed of two elements. A statically fixed stator (1) and a rotor (armature) (4) connected to the arm of the mechanism have to be considered. To create a magnetic torque, the stator is equipped with a field winding (3). It is supposed that its magnetic field is distributed homogenously. To distribute the filed, two ferromagnetic poles (2) are present in the stator. An additional winding (5) is associated with the armature, and the armature winding is connected to the output brushes (6). The necessary connection is done via a commutator (7). During the armature's rotation, its consecutive segments slide on the brush surfaces.

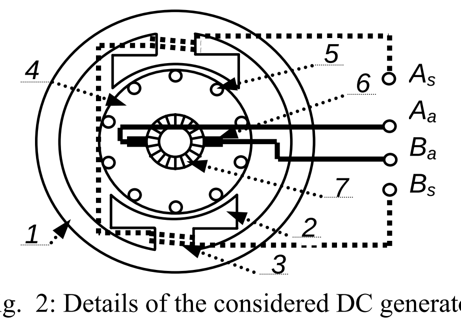

Fig. 2: Details of the considered DC generator

As the armature rotates within the stator contour, the armature winding moves in the stator generated magnetic field and an electromotive force, E_t, is induced in the winding. Starting from the winding, an electrical circuit is composed. As some additional elements, external resistance, the armature and brush resistances and the armature inductance are considered in the circuit. Since additional sources of the electromotive force are absent, the armature's electromotive force balances with the circuit losses. Moreover, when the armature current is present, mechanical interaction between the current and the magnetic field (i.e. mechanical torque) is created. Thus the Kirchhoff's voltage low, accompanied with the torque equation, can leads to the following circuit dynamics equations [10,11]:

$$E_t = c_{el} \cdot \Phi \cdot \omega_t = k_\Phi \cdot \omega_t \; ; \qquad (1)$$

$$L_t \cdot \frac{dI_t}{dt} + (R_t + R) \cdot I_t + k_\Phi \cdot \omega_t = 0 \; ; \qquad (2)$$

$$t_{el} = c_{el} \cdot \Phi \cdot I_t = k_\Phi \cdot I_t \; . \qquad (3)$$

where: ω_t – armature rotational velocity; L_t – total inductance of the armature; R_t – total resistance of the armature, brushes, and the contact area; R – external resistance; c_{el}, k_Φ – constants dependent on the engine design.

6. CONTROL ALGORITHM

As it is pointed in the introduction, the velocity amplification is based on the mechanism singular position. Amplification effectiveness is stronger when the mechanism motion is closer to the neighbourhood of its singularity position. To increase the effectiveness, an automatic control of the length of the arm of the mechanism should be a profitable idea. Refined algorithm is necessary, as a hard limit is present in the end-effector displacement (above of the limit, none of the displacements is physically possible). To avoid the impact phenomena, the limit may not to be touched. If a move above of the limit is expected, the mechanism length has to be modified previously. But, the amplification effectiveness is reduced when the length is unnecessary long.

Going within the details of the described idea, when significant vibrations are considered, positions at the singularity neighbourhood are satisfied for the whole period of a single oscillation (Fig. 3a). The neighbourhood is reached when the biggest deviations are present. The considered velocity amplification increases during this phase.

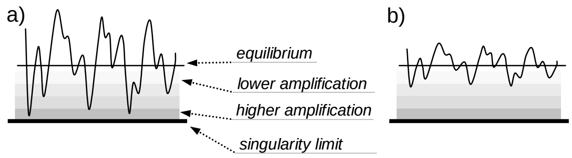

Fig. 3: Exemplary vibration characteristics: higher amplification (a); lower amplification (a);

As the range of the considered vibrations decreases during the motion (damping), duration of the effective phase reduces (Fig. 3b). Motions closer to the equilibrium position become dominant, and the mechanism is far from its amplification effectively. Mechanism modification is required to fit the singular configuration to the actual work condition. To obtain it, a control algorithm is proposed to control the work conditions. The length of the mechanism final arm has to be modified.

As it was pointed at the beginning of the section, the considered problem is not a straightforward, especially when a multi mode vibration is present in the element. In these cases, the potentially lowest value is difficult to predict, especially when the detail modal identification of the actual vibration is impossible. With the presented problems kept in background, a set of tests performed by the proposed algorithm are presented in Fig. 4.

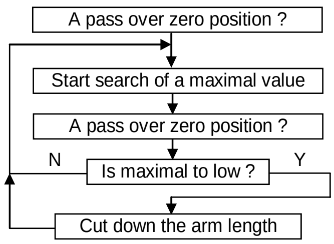

Fig. 4: Draft of the used algorithm of control of the length of the mechanism arms

7. CONSIDERED SYSTEM

To verify effectiveness of the proposed amplification, numerical tests are performed on a numerical model of a potentially possible system. A multibody part and of a finite elements structure are considered in the model. Considered multibody system is planar (fig. 5). It is composed of two bodies (arms). Its joints are rotational. The first arm is fixed to the reference. Its fixing point is located *40 cm* above of the forth node of the elastic structure. All the arms are made of light pipes. The first is *20 cm* long. The length of the second is modifiable. Its initial length equals *22.4 cm*. Identical properties are proposed for the arms. Their masses equal *1 kg*. The moments of inertia equal $0.05\ kg{\cdot}m^2$. The mass centres are shift *10 cm* from the fixing points. The inertia products equal zero. According to the used length, the system configuration is close to its singular position (the second arm is almost aligned with the initial). In the neighbourhood of the singular position, according to the low value of the determinant of the Jacobian matrix, the joint velocities have to be large, even if the vertical component of the velocity is small for the endpoint of the mechanism.

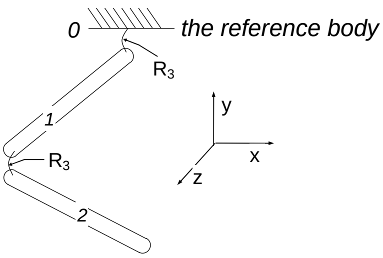

Fig. 5: The considered the multibody model

To introduce the vibrating system, the end of the second arm is joined rotationally to a beam model composed of finite elements. The length of the beam is set to *1m*. A set of ten finite elements is used to model the beam (fig. 6). Relatively low discretisation is introduced, as the detail imaging of the modes in not a critical point of the presented considerations. Surface of its cross sections equals $0.001\ m^2$, its geometrical moment of inertia equals $10^{-8}\ m^4$ and density of its material is $7.8{\cdot}10^3\ kg/m^3$. According to the illustrative reasons, relatively low Young modulus is supposed. It equals $2.1{\cdot}10^9\ Pa$, only. The end of the multibody arm is joined to the forth node of the beam (Fig. 1).

To obtain a symbolic form of its dynamics equations of the multibody structure, ROBOTRAN [8, 9] program is used. A vibrating system is modelled by use of beam finite elements. Its dynamics is described according to equations presented in first part of the paper. The constraint equations are introduced as described in the first part, too. As the damping element, a DC generator is introduced. Its rotor is fixed to the first joint of the multibody structure. Its electrical circuit is closed with an external resistance of 10Ω.

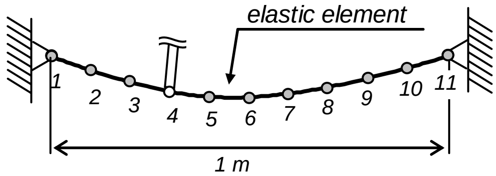

Fig. 6: The considered beam and its nodes numbering

The run of the program is done in MATLAB [12]. Numerical integrations are performed. Their initial positions correspond to a deformed beam position. The shape, and the amplitude, of the deformation correspond to an equilibrium position, obtained when a single force is set to the beam. At the initial moment, the beam is liberated from the force.

8. NUMERICAL TESTS

Initially, the inertia free mechanism is tested, i.e. masses and moments are zero for both arms of the mechanism. Moreover, low structural damping is considered in the beam. The beam is deformed. Its initial deformation is presented in Fig. 7.

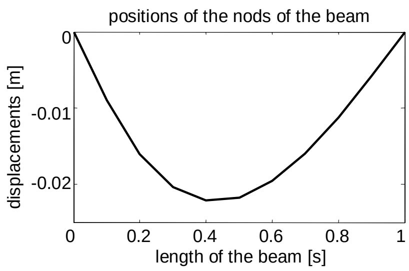

Fig. 7: Initial deformation of the beam

When the electrical circuit is disconnected (i.e. external damping is omitted), the time evolutions of the beam vibration (precisely, displacements of the joint between the beam and the mechanism) are presented in Fig. 8a.

In the obtained results, few modes of vibrations can be detected. Moreover, the low damping is confirmed. Corresponding deformations of the multibody joint (precisely the first joint of the mechanism) are presented in Fig. 8b. The nonlinear work of the used mechanism is confirmed, as the initial vertically "symmetrical" behaviour in Fig. 8a is converted next into a "non symmetrical" one, presented in Fig. 8b.

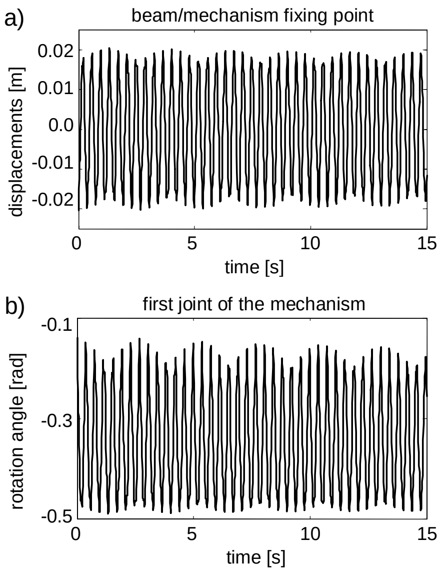

Fig. 8: Interaction between the beam and an inertia free mechanism with a disconnected electrical circuit: vibrations at the fixing point between the mechanism and the beam (a); displacements at the first joint of the mechanism (axis of the DC-generator) (b)

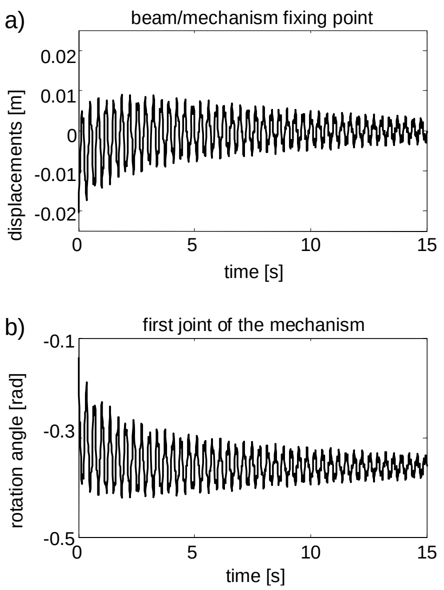

Fig. 9: Interaction between the beam and an inertia free mechanism with the circuit closed by the resistance: vibrations at the fixing point between the mechanism and the beam (a); displacements at the first joint of the mechanism (axis of the DC-generator) (b)

In the next test, a resistance is used to close the circuit. Identical initial conditions (presented previously in Fig. 7) are considered. Obtained results are presented in Fig. 9. The beam vibrations are presented in Fig. 9a. Better damping is obtained. The multibody joint deformations are presented in Fig. 9b. Related circuit current is presented in Fig. 14a.

In the third test, a control of the arms length is considered. Again, the used initial deformation is identical with the one presented previously. Obtained results are shown in Fig. 10. Time evolution of the beam vibration is presented in Fig. 10a. Significantly better damping is obtained now. Related multibody joint deformations are presented in Fig. 10b. Obtained time evolution of the arm length is shown in Fig. 15a. As imposed in the control algorithm, the arm is shortened when the range of the motion is lower then *0.16 rad*, in the motored joint of the multibody system.

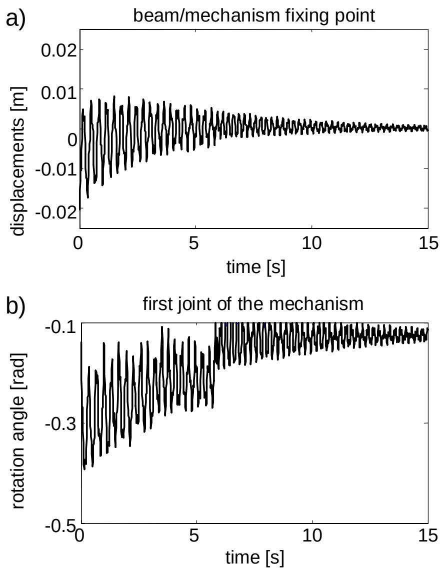

Fig. 10: Interaction between the beam and an inertia free mechanism with the circuit closed by a resistance and the arms control: vibrations at the fixing point between the mechanism and the beam (a); displacements at the first joint of the mechanism (axis of the DC-generator) (b)

The influence of the mechanism inertia is verified in the next tests. When the electrical circuit is disconnected (a system without any external damping), time evolutions of the system motion are presented in Fig. 11. The previously used initial conditions are employed in the actual tests (see Fig. 7). Time evolution of the beam vibration (i.e. displacements of the connecting joint between the beam and the mechanism) is presented in Fig. 11a. Again, the low damping is confirmed. Related multibody joint deformations are presented in Fig. 11b. The nonlinearity of the mechanism is re-confirmed. Vertically non-symmetrical behaviour can be observed in Fig. 11a. The turning point is narrower in the upper part of the characteristic. The variability of the mass matrix could be responsible for it. Moreover as the system total inertia increases (in compare to the inertia free mechanism) the period of vibrations increases in the present results, too.

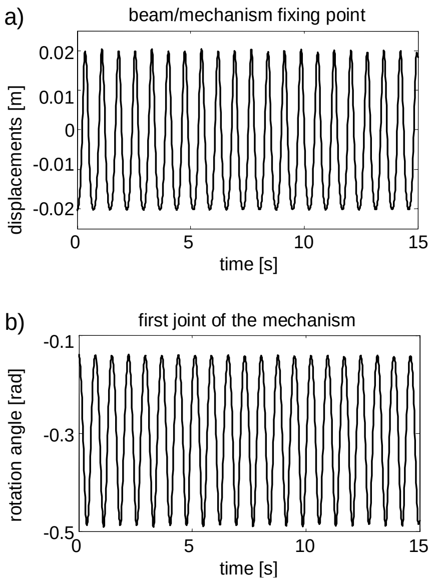

Fig. 11. Interaction between the beam and the inertial mechanism with a disconnected electrical circuit: vibrations at the fixing point between the mechanism and the beam (a); displacements at the first joint of the mechanism (axis of the DC-generator) (b)

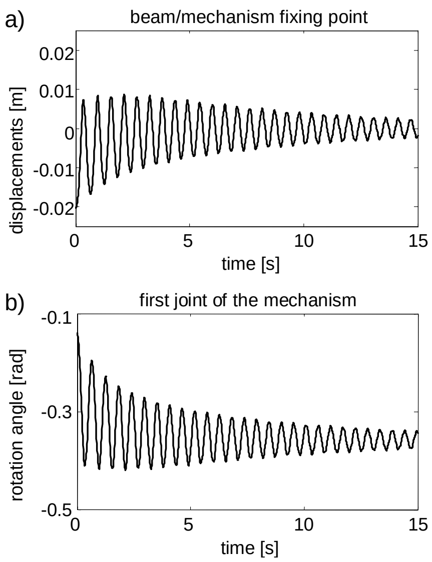

Fig. 12: Interaction between the beam and the inertial mechanism with the circuit closed by the resistance: vibrations at the fixing point between the mechanism and the beam (a); displacements at the first joint of the mechanism (axis of the DC-generator) (b)

In the next test, the electrical circuit is closed by the resistance. Obtained results are presented in Fig. 12. The presently used initial position is identical with the one used previously. For the beam vibration, its time evolution is presented in Fig. 12a. Better damping is obtained now. Corresponding joint deformations (the first joint of the mechanism) are presented in Fig. 12b. Finally, its time evolution of the current is shown in Fig. 14b.

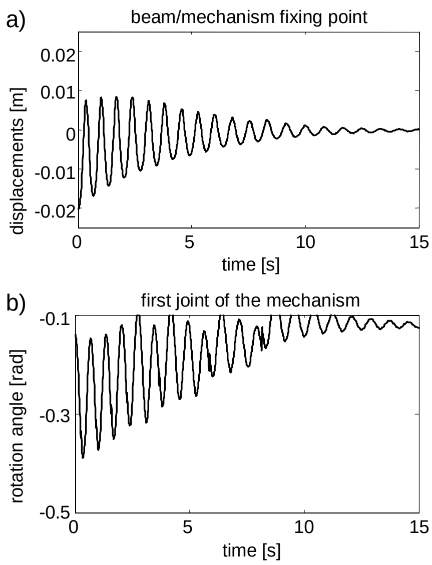

Fig. 13. Interaction between the beam and the inertial mechanism with the circuit closed by a resistance and the arm control: vibrations at the fixing point between the mechanism and the beam (a); displacements at the first joint of the mechanism (axis of the DC-generator) (b)

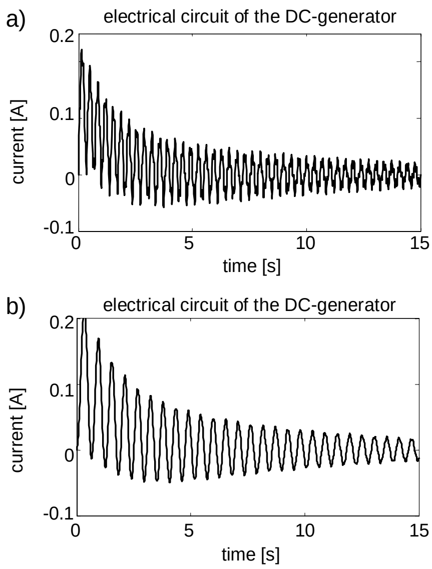

Fig. 14: Current at the electrical circuit: the inertia free mechanism (a); the inertial mechanism (b)

In the final test, the control of the arms length is tested. Obtained results are presented in Fig. 13. Again, its initial deformation is identical with the one used previously. The time evolution of the beam vibrations are presented in Fig. 13a. Significantly better damping is obtained. Corresponding joint deformations (the first joint of the mechanism) are presented in Fig. 13b. Next, a time evolution of the arm length is shown in Fig. 15b. In the present test, the arm is shortened when the joint rotation is lower then *0.12 rad*.

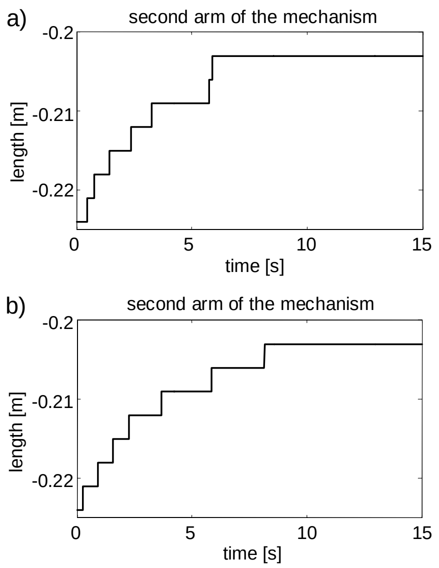

Fig. 15: Length of the arm of the mechanism: the inertia free mechanism (a); the inertial mechanism (b)

9. CONCLUSIONS AND PERSPECTIVES

In the paper, a damping method base on the electromagnetic effects is presented. The performed tests confirmed an effective damping present in the considered system. To increase the damping effectives, velocity amplification is introduced. Proposed amplification is possible as a transmission mechanism is set between the vibrating element and the DC generator. Moreover, to obtain the required amplification effect, the mechanism position is close to its singularity configuration. Benefits of the proposed amplification method are confirmed by the performed numerical tests.

On the base of the obtained results an observation has to be pointed. When the inertial transmission system is introduced, the total inertia of the system varies, and some significant nonlinear effects can be observed in the obtained vibrations. The nonlinear vibrations are present in the system.

It is useful to be underlined, that some additional amplification is obtained, when a length control is imposed on the arms of the mechanism. It correlates the mechanism singular position to the actual ranges of the element deformations cased by its actual vibrations. The idea of proposed amplification system is not limited to be used to damp vibrations of beam structures, only. It can be fruitfully employed in more complex applications, too.

The proposed modelling method combines the multibody modelling and the finite elements modelling. The introduced methodology is found as an effective modelling method. Calculation times are relatively short. Long integrations as well as optimizations are possible. The model based on the constraint equations is an interesting an effective alternative to the classical elastic contacts.

The presented results indicate that some future investigations are necessity. As an example, the influence of the mechanism inertia has to be detailed. Parameters of the control algorithm could be estimated more precisely and more robust. Alternative singular configurations as well as alternative mechanisms should be tested. The influences of the joint frictions and backslashes have to be introduced. Alternative electrical generators (the AC-generators for example) have to be tested, as well as alternative components of the electrical circuits, too.

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LES MECANISMES SITUES SUR LE VOISINAGE DE LA POSITION SINGULIERE – L'AMPLIFICATION DE LA VITESSE DANS L'AMORTISSEMENT ELECTROMAGNETIQUE DES VIBRATIONS

Cet article présente les études sur l'amortissement électromagnétique. Un système mécanique continu et ses vibrations sont considérés. Pour amplifier l'amortissement, un mécanisme de double-bar est introduit. Sa position est définie sur un voisinage de sa position singulière (cinématique). Ensuite, des tests sont effectués sur le modèle numérique. Deux segments structurellement différents sont considérés. La partie élastique est composée des éléments finis. Le mécanisme est modélisé comme un système multicorps. Les équations de contrainte relient des sous-modèles. Les libres modes de vibrations sont estimés pour les modèles non amortis et amortis. Son énergie est transformée en courant électrique (par l'utilisation d'un générateur de courant continu) et dissipée. Comme la tâche fondamentale est de garder ses configurations de travail dans le voisinage de la singularité, on a ajouté le système de contrôle automatique de la longueur des bras du mécanisme. Les résultats des tests numériques ont le but de confirmer l'efficacité de la méthode d'amortissement.

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