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Influence of elements dynamic cohesiveness in power shafting on torsional vibrations spreading and dynamic equality of reducible model

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1. Introduction

Complex of transmission or powertrain elements is one of the most important parts of any traction and transport vehicles. In operation powertrain elements load has dynamic character. Both internal and external disturbances form these loads. Fluctuations of tractive resistance and engine torque, disturbances from the frame vibrations on the suspension, and for tracked vehicle – from caterpillar rewinding irregularities are considered as main among external. Kinematic and force disturbances from gears meshing, shafts misalignment, universal-joint rotation irregularities, deformations and displacements of case details are main among the internal disturbances [1-4].

Irregularity of acting of the external loads produces torsional and bending vibrations in the powertrain shafting. Its role in the fatigue damages accumulation in material is considerable. Accordingly to data from domestic and foreign researchers [1-3, 5-8] more that 50% of damages and faults in vehicles powertrains are produced by vibrations exactly.

It is known [1, 4, 9] that in some systems where rotary motions are, vibrations with certain frequency pass through whole shafting practically without damping but in other systems vibrations are damped at areas next to source.

Professor Mandelshtam L.I. did research of process of vibrations spreading in cohesiveness dynamic systems. It is shown in his work [10] that degree of dynamic cohesiveness influences on process of vibrations spreading. Process of vibrations spreading was researched on the example of motion of two pendulums connected by elastic linkage (Fig. 1). Due this research the fact was established that if partial frequency of vibrations of one element of this system (first pendulum) is equal to partial frequency of vibrations of other element (second pendulum), vibrations fast and practically without damping transfer from one element to other.

Some points from this works can be used to research character of torsional vibrations spreading in systems with rotary motions, particularly in machines powertrains. Differ from system presented on Fig. 1 shafting of this powertrain is *n*-mass system performing constrained torsional vibrations.

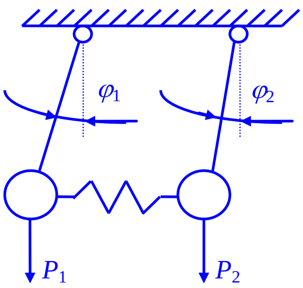

Fig. 1 Pendulums with elastic linkage

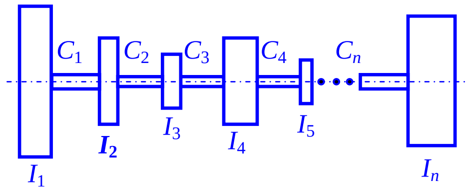

Fig. 2 *n*-mass shafting system

During power shaftings design information about how dynamic parameters of its elements influence on character of torsional vibrations spreading is necessary. In particular, does dynamic cohesiveness of its elements influence on this process or not.

Models are created by researchers should include all moving mass of system if it is possible. But influence of some masses with very small inertia moments on research results is negligible. And this masses are reduced thus model can be simplified. In this case researchers need information of what masses can be reduced, how much model can be simplified and what parameter should be used as criteria for model reduce limits.

2. Determination of degree of system elements dynamic cohesiveness

For simplicity we'll consider two-element rotary system with torsional vibrations [8] that include 3 masses connected by two elastic linkages (Fig. 3). Let's define element as model area that connects two vibrating masses.

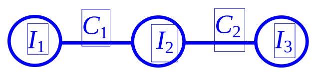

Fig. 3 Two-element model

Differential equations of free vibrations of elements of this model can be written in next form:

$$\left.\begin{aligned} & I_1\ddot{\varphi}_1 + C_1(\varphi_1 - \varphi_2) = 0, \\ & I_2\ddot{\varphi}_2 + C_2(\varphi_2 - \varphi_3) - C_1(\varphi_1 - \varphi_2) = 0, \\ & I_3\ddot{\varphi}_3 - C_2(\varphi_2 - \varphi_3) = 0, \end{aligned}\right\} \tag{1}$$

where I_i is inertia moments of masses; C_i is torsional stiffness of its linkages; $\varphi_i, \dot{\varphi}_i, \ddot{\varphi}_i$ are angular displacements, speeds and accelerations of masses respectively.

New variables are defined below:

$$\varphi_1 - \varphi_2 = q_1, \quad \varphi_2 - \varphi_3 = q_2, \qquad (2)$$
$$\ddot{\varphi}_1 - \ddot{\varphi}_2 = \ddot{q}_1, \quad \ddot{\varphi}_2 - \ddot{\varphi}_3 = \ddot{q}_2,$$

where q_i and $\ddot{q}_i$ are relative angular displacements and accelerations of elements respectively.

After some modifications we will get system of two equations:

$$\left.\begin{aligned} \ddot{q}_1 + \omega_{11}^2 q_1 - \omega_{12}^2 q_2 = 0, \\ \ddot{q}_2 + \omega_{22}^2 q_2 - \omega_{21}^2 q_1 = 0, \end{aligned}\right\} \qquad (3)$$

where ω_{11}, ω_{22} are partial frequencies of elements, and

$$\omega_{11}^2 = \frac{C_1(I_1 + I_2)}{I_1 I_2}, \quad \omega_{22}^2 = \frac{C_2(I_2 + I_3)}{I_2 I_3}, \quad \omega_{12}^2 = \frac{C_2}{I_2},$$
$$\omega_{21}^2 = \frac{C_1}{I_2}.$$

Frequency or characteristic equation of system (3) is written as:

$$p^4 + a_2 p^2 + a_4 = 0, \qquad (4)$$

where $a_2 = \omega_{11}^2 + \omega_{22}^2$, $a_4 = \omega_{11}^2\omega_{22}^2 - \omega_{12}^2\omega_{21}^2$, $p = j\Omega$,

$$\Omega_{1,2} = \sqrt{\frac{\omega_{11}^2 + \omega_{22}^2}{2} \mp \sqrt{\frac{\left(\omega_{11}^2 + \omega_{22}^2\right) - 4\left(\omega_{11}^2\omega_{22}^2 - \omega_{12}^2\omega_{21}^2\right)}{4}}},$$

$\Omega_{1,2}$ is natural vibrations frequencies.

In accordance with [1], maximal interference influence of elements vibrations, or in other words maximal dynamic cohesiveness, takes place when elements partial frequencies are equal, that is when $\omega_{11} = \omega_{22}$, and degree of system elements dynamic cohesiveness is defined by equation:

$$\gamma = \frac{\omega_{12}^2\omega_{21}^2}{\omega_{11}^2\omega_{22}^2}. \qquad (5)$$

Parameter γ is named coefficient of dynamic cohesiveness of elements vibrations. It is got from considering of characteristic Eq. (4) and it shows degree of difference from zero of absolute term a_4. When

$$a_4 = \omega_{11}^2\omega_{22}^2 - \omega_{12}^2\omega_{21}^2 = 0, \text{ so } \omega_{11}^2\omega_{22}^2 = \omega_{12}^2\omega_{21}^2$$

and coefficient $\gamma = 1$. The more diffrence from zero value of absolute term, the closer to zero value of coefficient γ becomes.

Let's consider two extreme cases:

1) $\gamma << 1$, that is $\gamma = \dfrac{\omega_{12}^2\omega_{21}^2}{\omega_{11}^2\omega_{22}^2} \approx 0.$

It is possible when numerator of this formula tends to zero. Substituting $\omega_{12}^2\omega_{21}^2 = 0$ in expression of frequencies of natural vibrations we get:

$$\Omega_{1,2} = \sqrt{\frac{\omega_{11}^2 + \omega_{22}^2}{2} \mp \frac{\omega_{11}^2 - \omega_{22}^2}{2}}.$$

Then $\Omega_1 = \omega_{22}$, $\Omega_2 = \omega_{11}$, that is system natural frequencies are equal to partial. It indicates that dynamic cohesiveness of elements is small.

2) $\gamma \approx 1$. It is possible when $\omega_{12}^2\omega_{21}^2 \approx$ $\approx \omega_{11}^2\omega_{22}^2$.

Then

$$\Omega_{1,2} = \sqrt{\frac{\omega_{11}^2 + \omega_{22}^2}{2} + \frac{\omega_{11}^2 + \omega_{22}^2}{2}} \text{ so } \Omega_1 = 0,$$
$$\Omega_2 = \sqrt{\omega_{11}^2 + \omega_{22}^2}.$$

What is the physical meaning of this coefficient? As mentioned above, when $\gamma << 1$ dynamic cohesiveness of elements is small. It means that during natural vibrations each element will oscillates with frequency which is close to partial. That is with frequency of system vibrations that could be equal to frequency of vibration of system alone. Meanwhile influence of other elements on frequency of vibration of every isolated element is negligibly small. The further value of coefficient γ from zero, the more diffrence is between natural and partial vibrations frequencies of elements and natural frequencies become inherent to whole system, but not to its local partial elements.

To obtain analytical expression defining absolute term of frequency equation of high order is very hard. Let's consider it on example of analysis of system of equations for four-element system. As opposed to system at Fig. 3 this system includes 5 masses and 4 linkages.

Equations of motions of this system are presented in form:

$$\left.\begin{aligned} \ddot{q}_1 + \omega_{11}^2 q_1 - \omega_{12}^2 q_2 = 0 \\ \ddot{q}_2 + \omega_{22}^2 q_2 - \omega_{21}^2 q_1 - \omega_{23}^2 q_3 = 0 \\ \ddot{q}_3 + \omega_{33}^2 q_3 - \omega_{32}^2 q_2 - \omega_{34}^2 q_4 = 0 \\ \ddot{q}_4 + \omega_{44}^2 q_4 - \omega_{43}^2 q_3 = 0 \end{aligned}\right\}, \qquad (6)$$

where $\omega_{11}^2 = \dfrac{C_1(I_1 + I_2)}{I_1 I_2}$, $\omega_{22}^2 = \dfrac{C_2(I_2 + I_3)}{I_2 I_3}$,

$\omega_{33}^2 = \dfrac{C_3(I_3 + I_4)}{I_3 I_4}$, $\omega_{44}^2 = \dfrac{C_4(I_4 + I_5)}{I_4 I_5}$, $\omega_{12}^2 = \dfrac{C_2}{I_2}$,

$\omega_{21}^2 = \dfrac{C_1}{I_2}$, $\omega_{23}^2 = \dfrac{C_3}{I_3}$, $\omega_{32}^2 = \dfrac{C_2}{I_3}$, $\omega_{34}^2 = \dfrac{C_4}{I_4}$, $\omega_{43}^2 = \dfrac{C_3}{I_4}$.

Characteristic equation for this system is:

$$p^8 + a_2 p^6 + a_4 p^4 + a_6 p^2 + a_8 = 0. \qquad (7)$$

On introducing of new variables (as it was made in two-element system) for decreasing of order of equations system (6) and on its modification we get matrix of coefficients of its equations:

$$\begin{vmatrix} \omega_{11}^2 & -\omega_{12}^2 & 0 & 0 \\ -\omega_{21}^2 & \omega_{22}^2 & -\omega_{23}^2 & 0 \\ 0 & -\omega_{23}^2 & \omega_{33}^2 & -\omega_{34}^2 \\ 0 & 0 & -\omega_{43}^2 & \omega_{44}^2 \end{vmatrix}. \qquad (8)$$

On system expanding we get value of its determiner in analytical form:

$$a_8 = \omega_{11}^2\omega_{22}^2\omega_{33}^2\omega_{44}^2 - \omega_{11}^2\omega_{22}^2\omega_{34}^2\omega_{43}^2 - \\ -\omega_{11}^2\omega_{44}^2\omega_{23}^2\omega_{32}^2 - \omega_{33}^2\omega_{44}^2\omega_{12}^2\omega_{21}^2 + \\ +\omega_{12}^2\omega_{21}^2\omega_{34}^2\omega_{43}^2. \quad (9)$$

So coefficient of dynamic cohesiveness for four-element system is:

$$\gamma = (\omega_{11}^2\omega_{22}^2\omega_{34}^2\omega_{43}^2 + \omega_{11}^2\omega_{44}^2\omega_{23}^2\omega_{32}^2 + \\ + \omega_{33}^2\omega_{44}^2\omega_{12}^2\omega_{21}^2 - \omega_{12}^2\omega_{21}^2\omega_{34}^2\omega_{43}^2)/\omega_{11}^2\omega_{22}^2\omega_{33}^2\omega_{44}^2. \quad (10)$$

Thus with increasing of number of system masses inconvenience of expression of absolute term (and coefficient γ) becomes more significant. So less labor-intensive method of estimation of coefficient γ value is proposed by autors.

It is known that determinator of matrix with structure mentioned above is equal to product of roots of characteristic equation:

$$Q_n = \Omega_1\Omega_2\Omega_3\ldots\Omega_n\,. \quad (11)$$

Therefore if values of inertia moments of system masses and its linkage stiffness are known it is possible to estimate its natural frequencies and then to estimate determinator Q_n. Knowledge of parameters of masses and linkages also provides estimation of values of partial frequncies of every element and then to get its product:

$$S_n = \omega_{11}^2\omega_{22}^2\omega_{33}^2\ldots\omega_{nn}^2\,. \quad (12)$$

Now it is possible to estimate value of coefficient γ by means of next expression:

$$\gamma = \frac{Q_n - S_n}{S_n}\,. \quad (13)$$

Expression (13) provides for determination of value of coefficient γ for random multimass chain system with torsional vibrations. It should be noted that values Q_n and S_n must be estimated with equal degree of precision.

3. Using of γ-coefficient as criteria for model reducing limit

Equality to zero of one of Eq. (4) roots means that dynamic cohesiveness of elements is very significant. Thus one of elements can be replaced with solid body so number of system elements can be reduced by one. Therefore coefficient γ of elements dynamic coehesiveness obtained from free term of a frequency equation can be used for theoretical basis of simplification or reducing of multimass model. Reduced model must be dynamic equvivalent to initial model in range of its mases operation frequencies. Below we consider example of using of coefficient γ on the stage of reducing of tractors T-5 powertrain shafting dynamic model from 9 to 5 masses. This tractor was produced by Volgorad tractor factory. Values of inertia moments of masses, torsional stiffness of linkages, natural and partial frequencies of masses vibrations at every stage of reducing are given in Table 1.

Table 1

Reducing model parameters changes

Quantity of masses	Number of mass or linkage									Value γ
	1	2	3	4	5	6	7	8	9	
9	Masses inertia moments, kg·m^2									0.9999
	$7.13\cdot10^{-2}$	$2.61\cdot10^{-2}$	$1.27\cdot10^{-3}$	$2.43\cdot10^{-4}$	$4.20\cdot10^{-5}$	$1.00\cdot10^{-5}$	$6.34\cdot10^{-3}$	$3.38\cdot10^{-1}$	$6.34\cdot10^{-3}$	
	Linkages torsional stiffness, N·m/rad									
	1150	18617	2810	255	714	2240	293	271		
	Partial frequencies, Hz									
	245	3930	3710	2670	9400	$1.5\cdot10^{5}$	215	207		
	Natural frequencies, Hz									
	5.7	33.1	37.4	44.7	480.8	683.9	736.6	2757		
8	Masses inertia moments. kg·m^2									0.9996
	$7.13\cdot10^{-2}$	$2.61\cdot10^{-2}$	$1.27\cdot10^{-3}$	$2.43\cdot10^{-4}$	$4.20\cdot10^{-5}$	$6.35\cdot10^{-3}$	$3.38\cdot10^{-1}$	$6.34\cdot10^{-3}$		
	Linkages torsional stiffness. N·m/rad									
	1150	18617	2810	255	542	293	271			
	Partial frequencies, Hz									
	245	3930	3710	2670	3600	217	209			
7	Masses inertia moments. kg·m^2									0.9995
	$7.13\cdot10^{-2}$	$2.74\cdot10^{-2}$	$2.43\cdot10^{-4}$	$4.20\cdot10^{-5}$	$6.35\cdot10^{-3}$	$3.38\cdot10^{-1}$	$6.34\cdot10^{-3}$			
	Linkages torsional stiffness. N·m/rad									
	1150	2460	255	542	293	271				
	Partial frequencies, Hz									
	241	3200	2670	3600	217	209				

Quantity of masses	Number of mass or linkage									Value γ
	1	2	3	4	5	6	7	8	9	
6	Masses inertia moments. kg·m^2									0.9970
	$7.13 \cdot 10^{-2}$	$2.74 \cdot 10^{-2}$	$2.43 \cdot 10^{-4}$	$6.39 \cdot 10^{-3}$	$3.38 \cdot 10^{-1}$	$6.34 \cdot 10^{-3}$				
	Linkages torsional stiffness. N·m/rad									
	1150	2460	174	292	271					
	Partial frequencies, Hz									
	241	3200	862	216	209					
5	Masses inertia moments. kg·m^2									0.9327
	$7.13 \cdot 10^{-2}$	$2.76 \cdot 10^{-2}$	$6.39 \cdot 10^{-3}$	$3.38 \cdot 10^{-1}$	$6.34 \cdot 10^{-3}$					
	Linkages torsional stiffness. N·m/rad									
	1150	163	292	271						
	Partial frequencies, Hz									
	240	177	216	209						
	Natural frequencies, Hz									
	5.7	33.1	37.4	44.7						

Data from Table 1 show that coefficient γ of 9-mass system is almost equal to one (0.9999), so it is possible to decrease number of model elements by one. The sixth element of model has the greatest partial frequency thus its reducing is performed by means of Rivins method. For 8-mass model γ is 0.9996. Now we have to reduce second element. For 7-mass model γ is 0.9995 and forth element is to be reduced. For 6-mass model γ is 0.997 so second element is reduced again. For 5-mass model γ is 0.9327 and it noticeably differs to 1. Now all partial frequencies are lesser than 1000 Hz. Consecuently reducing must be stoped now.

Comparison of natural frequencies values of reducing 9-mass system and reduced 5-mass system shows that main (first) model natural frequencies haven't changed during reducing. It means that reduced model is equvivalent to reducing model in researched frequency range.

4. Influence of degree of elements dynamic cohesiveness on character of torsional vibration spreading

To find out how degree of elements dynamic cohesiveness influences on torsional vibrations spreading computational research was made [9, 11]. For simplicity, 3-mass model which includes 4 masses and 3 linkages was used.

Values of inertia moments of masses and coefficient of dynamic cohesiveness of vibrations γ are given in Table 2. It is believed that stiffnesses of all system linkages are equal and its value is 100000 N·m/rad.

Table 2

Influence of inertia moments of masses on degree of dynamic cohesiveness of vibrations of system elements

Mass No	Inertia moments of masses, kg·m^2				Value γ
	I_1	I_2	I_3	I_4	
1	0.001	1	1	1	0.2630
2	0.01	1	1	1	0.2650
3	0.1	1	1	1	0.3075
4	1	1	1	1	0.5261
5	10	1	1	1	0.7057
6	100	1	1	1	0.7454
7	1000	1	1	1	0.7512
8	10000	1	1	1	0.7510
9	0.001	1	0.001	1	0.9980
10	0.001	1	0.01	1	0.9803
11	0.001	1	0.1	1	0.8267

As shown in Table 2, change of inertia moment of first mass of the system from 0.001 to 10000 kg·m^2 (variants 1-8) provokes change of coefficient γ only in range 0.263 - 0.7512, that is due change of inertia moment of one mass of system from lowest to highest limit (for automotive powertrains) strong dynamic cohesiveness isn't reached.

But changing of one more mass inertia moment provides for to get values γ which are close to one (variants 9 and 10).

During the research of vibrations on first mass was generated by unit torque with frequency 1 Hz and process of vibration energy spreading from one element to another was analyzed. During first second of the process average value of potential energy of spin of every area at the period of 0.1 second was estimated and compared.

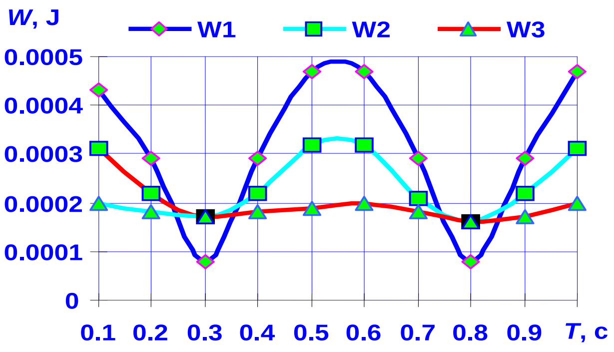

Fig. 3 Low dynamic cohesiveness

Analyze of results shows that when dynamic cohesiveness of system elements vibrations is low (variants 1, 4, 7) there isn't regularity in process of vibration energy spreading from one element to another (Fig. 3), but when dynamic cohesiveness is high (Fig. 4, variant 9) values of potential energy of second and third elements turn out to be equal in every moment of time. By symbols W1, W2 and W3 values of potential energy of spin of proper elements are presented.

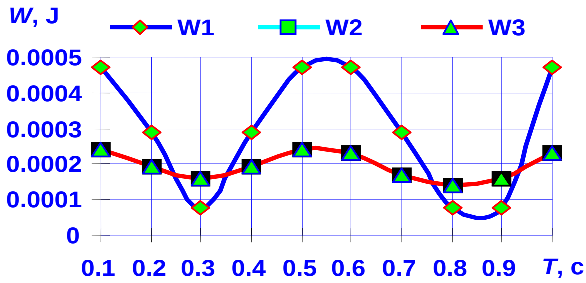

Fig. 4 High dynamic cohesiveness

As can be seen from considering of expression for coefficient γ definition, its value becomes greater when difference between products of natural and partial frequencies of system elements increases. Values of natural and partial frequencies of system from Table 2 are presented in Table 3.

Data from Table 2 testify that only one variant (number 9) from all considered is characterized by that when dynamic cohesiveness is high all elements of system have equal partial frequencies. To investigate how this fact influences on process of vibration energy transfer, research of some variants of model with various combinations of elastic-inertial parameters was done. There are equal partial frequencies of vibrations of all elements when these combinations are used. Parameters mentioned above are presented in Table 4.

Table 3

Natural and partial frequencies of system variants

Variant number	Natural frequencies, Hz			Partial frequencies, Hz		
	1	2	3	1	2	3
1	50	87	1592	10000	447	447
2	50	87	505	3180	447	447
3	49	86	167	1050	447	447
4	38	71	92	447	447	447
5	25	63	90	332	447	447
6	22	62	90	318	447	447
7	22	62	90	316	447	447
8	22	62	90	316	447	447
9	50	1592	2251	10000	10000	10000
10	50	713	1592	10000	3180	3180
11	50	230	1592	10000	1050	1050

Table 4

Variants with equal partial frequency

Variant number	Linkage stiffness, N·m /rad			Inertia moments of masses, kg·m^2				Partial frequencies, Hz			Value γ
	1	2	3	1	2	3	4	1	2	3	
1	10^2	10^2	10^2	1	10^{-3}	1	10^{-3}	316	316	316	0.9982
2	10	10	10	1	10^{-3}	1	10^{-3}	100	100	100	0.9980
3	10^3	10^3	10^3	1	10^{-3}	1	10^{-3}	10^4	10^4	10^4	0.9980
4	10^4	10^4	10^4	1	10^{-3}	1	10^{-3}	10^4	10^4	10^4	0.9980

For example character of change of potential energy of areas for variant 4 is shown on Fig. 5.

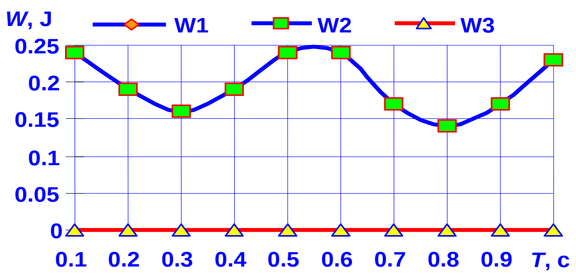

Fig. 5 Character of change for variant 4

For all of four areas considered potential energy of spin of first and second areas of system is equal all time, that is point that to transfer equal values of vibration energy from area to area, high dynamic cohesiveness of system elements vibrations and equal values of its vibrations are necessary, was proved.

Does frequency of driving signal influence on character of process of vibration energy transfer? To get answer on this question research for variants of system with equal partial frequencies of elements and generation of vibrations with frequencies 0,1 Hz, 5 Hz и 10 Hz was done. For example result with frequency 0.1 Hz is presented on Fig. 6.

Analysis of Fig. 6 shows that change of frequency of driving signal doesn't influence on character of vibration energy transfer between second and third areas of system – curves W2 and W3 coincide. If frequency of driving signal increases to 10, values of average potential energy of areas spin become equal all time.

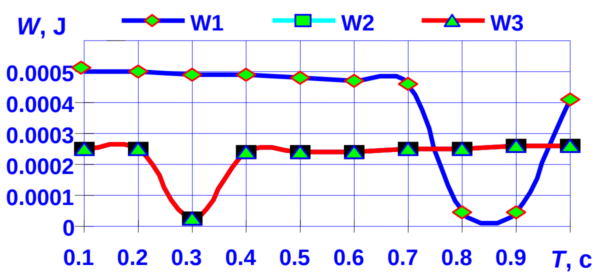

Fig. 6 Vibrations with frequency 0.1 Hz

Analysis of results of whole complex of experiments done provides for range of regularities to be made. Character of spreading of torsional vibrations in systems with rotary motion submits to range mentioned above. This range of regularities is presented in conclusions.

5. Conclusions

1. Coefficient of dynamic cohesiveness of powertrain shafting model elements can be used just as well as criteria for model reducing limit.

2. Character of transfer of torsional vibration energy from one area to another in systems with rotary motion is defined by combination of elastic-inertial parameters of its elements and order of its connections. Energy of vibrations from area to area transfers without losses if further conditions are fulfilled:

a) dynamic cohesiveness of system elements vibrations is strong;

b) partial frequencies of all system elements are equal;

c) transfer of energy performs from area which begins with mass with high moment of inertia and ends with mass with low moment of inertia, to area begins with mass with low moment of inertia and ends with mass with high moment of inertia, meanwhile value of greater moment must be greater than lesser moment at least on one order.

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ELEMENTŲ DINAMINIO SUKIBIMO JĖGINĖSE TRANSMISIJOSE ĮTAKA SUKAMŲJŲ VIRPESIŲ SKLIDIMUI IR REDUKUOJAMO MODELIO DINAMINIAM TAPATUMUI

Reziumė

Šiame straipsnyje pristatyti sukamųjų virpesių sklidimo krumpliaratinių pavarų velenuose tyrimai. Ištirta sistemos elementų virpesių dinaminio sukibimo įtaka šiam sklidimui. Aprašyti virpesių energijos perdavimo be energijos nuostolių atvejai.

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INFLUENCE OF ELEMENTS DYNAMIC COHESIVENESS IN POWER SHAFTING ON TORSIONAL VIBRATIONS SPREADING AND DYNAMIC EQUALITY OF REDUCIBLE MODEL

Summary

This paper presents investigation of torsional vibrations spreading in powertrain shafting. Influence of dynamic cohesiveness of system elements vibration on this spreading was researched. Cases of vibrations energy transfer without energy losses was described.

Keywords: Torsional vibrations, dynamic cohesiveness, shafting, automotive drivetrain, powertrain.

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