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LONGITUDINAL - ANGULAR OSCILLATION OF WHEELED VEHICLES WITH NON-LINEAR POWER CHARACTERISTICS OF ABSORBER SYSTEM

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Resume. *An in parameterized form of analytical dependences that describe the parameters defining a longitudinally - angular oscillation of the sprung wheeled vehicles. It is shown that for wheeled vehicles for special purposes for which the base is selected chassis production car, more ergonomically satisfying the conditions of suspension system with progressive law change resilient shock absorbers.*

Keywords: *system pidresoryuvannya, longitudinally - angular oscillation amplitude, frequency.*

Problem setting. There are much more stringent requirements on the performance of special purpose wheeled vehicles (SPWV) carrying small and medium load and operated at considerable speeds and in difficult driving conditions over rugged terrain. These requirements refer not only to the engine, transmission and other components or systems, but first and foremost to their suspension. The latter should provide adequate smoothness and protect people, goods and equipment from overloads (excessive fluctuations [1]). Suspension system of vehicles with linear or close to it regenerative power law not only fails to protect against significant overloads (including instant), but also leads to a significant fatigue of the driver or people during long-term transportation.

Analysis of the known research results. Experimental and certain theoretical studies indicate that characteristics of elastic force applied to the sprung mass, must meet certain conditions – to be small for small deformations of shock absorbers and grow rapidly with significant deformations. Suspension with non-linear relationship between renewable power and deformation satisfies these requirements [2] (progressive power characteristics of absorber systems (AS). However, the study of nonlinear dynamics of SPWV with nonlinear power characteristic has not been completed because of purely mathematical problems related to solving nonlinear differential equations that describe the dynamics of the process.

Some studies [2-5] relating to the vertical and cross – angular oscillations of vehicles within AS elastic properties progressive law change, point to a fundamental difference compared to the oscillations of its sprung part (SP) for linear characteristics of the AS. This primarily concerns the natural oscillation frequency dependence on the amplitude, and therefore on the conditions of existence of resonance vibrations, resonance amplitude depends on nonlinear characteristics of power and so on. It is a challenging task to get generalized results using a numerical analysis of relevant mathematical models of nonlinear dynamics of SPWV sprung mass (SM). The main objective is the development of methods for analyzing the impact of non-linear power characteristics of the AS on the dynamics of the SM, which would be the base for creating new or improving existing suspensions. Such approach proves to be the most reasonable and resource (material) saving.

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The aim is to develop analytical methods of study longitudinally angular oscillations of the SM of SPWV with nonlinear power AS to provide practical advice on the choice of main power parameters necessary to meet ergonomic requirements of operating the vehicle on a wide range of amplitudes of longitudinally angular oscillation of SM (dynamic load on the driver and people at transportation).

The implementation of work. A flat system, which is shown in Figure 1, is taken for a calculation model. It consists of a not sprung part and a sprung part that interact with each other through a system of suspension (elastic shock absorbers and damper devices)

Key assumptions and limitations of calculation model:

– suspension system is characterized by elastic forces and resistance, described by dependencies [2-5] $F_{ipr} = c_i \Delta_i^{\nu+1}$, $R_{op} = \alpha_i \dot{\Delta}_i^{s+1}$, where c_i, α_i, s – constants, Δ_i and $\dot{\Delta}_i$ deformation of elastic shock absorbers and its speed ($i=1$– for the front suspension, and $i=2$– for rear suspension), respectively;

– the maximum elastic force of the AS during the relative longitudinal – angular oscillation of SM is of much greater value than maximum resistance force of damper devices;

– the SM gravity center in relation to the vehicle base is defined by a, b, and c parameters (Fig. 1 a);

– the horizontal movement of the SM gravity center is an insignificant value, which is neglected;

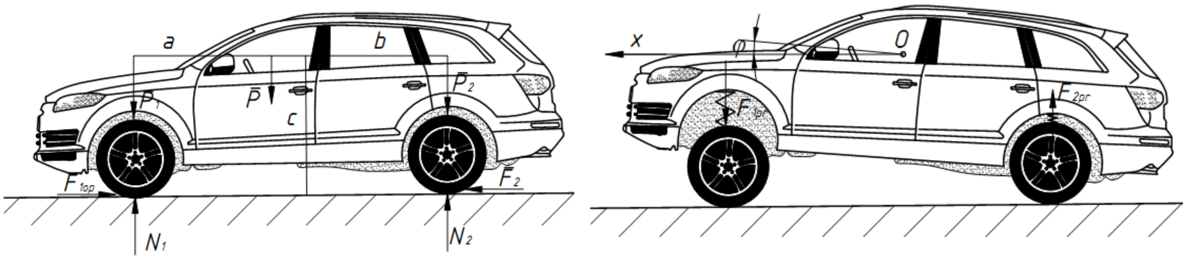
– deformation of the tire while driving SPWV on rugged terrain is a small quantity compared to the elastic deformation of the shock absorbers, and is neglected;

– SP during the motion of SPWV makes small oscillations around its gravity center (p. O) and its position is uniquely determined by an angle of rotation $\varphi(t)$;

– the work considers SPWV movement along the path with constant speed $\vec{F}_2 = \vec{F}_{1pr}$ (where $\vec{F}_{1pr}$ - the resistance, $\vec{F}_2$ - the driving force).

The impact of road irregularities on oscillations of the SM is not considered. It is believed that by hitting an abruptness SP gets the initial disturbance (initial vibration amplitude).

Considering these assumptions, in case of longitudinally angular oscillation the following conditions should be true: $c_2 = a/bc_1$, $(c_1 + c_2)\Delta_{st}^{\nu+1} = \vec{P}$, where $\vec{P}$ – weight of SPWV SM, and Δ_{st} – static deformation of elastic shock absorbers.



a) b)

Figure. 1. The calculated model: a) general view; b) distribution of external forces affecting a vehicle

The problem is getting the functional relationships that describe the basic parameters of SP longitudinally angular fluctuations while being the basis for determining the strength characteristics of the AS.

Methods of solving. To solve this problem use longitudinal differential equation - angular oscillation of SPWV SM

$$I_o\ddot{\varphi}=-\alpha\big(F_{1pr}+R_{1op}\big)-b\big(F_{2pr}+R_{1op}\big) \qquad (1)$$

where I_o – the moment of SP inertia in relation to the horizontal axis, which passes through the gravity center and is perpendicular to the velocity vector of the vehicle portable movement, ie, $I_o = P/(3g)(\alpha^2 + b^2 + c^2/4$,

$\varphi(t)$ – deviations in random periods of time from equilibrium position of SP. In the case of small oscillations of SM regarding reference system relative to the beginning position of SP static equilibrium, elastic forces value and the resistance can be written as

$$\begin{gathered} F_{1pr} = c_1(\varphi\alpha - \Delta_{st})^{v+1}, F_{2pr} = c_2(\varphi b + \Delta_{st})^{v+1}, \\ R_{1op} = \alpha_1\alpha^{s+2}\big(\dot{\varphi}(t)\big)^{s+1}, R_{2op} = \alpha_2 b^{s+2}\big(\dot{\varphi}(t)\big)^{s+1} \end{gathered} \qquad (2)$$

In view of (2) the differential equation (1) can be written as

$$I_o\ddot{\varphi} + (c_1\alpha^{v+2} + c_2 b^{v+2})\varphi^{v+1} = (v+1)\Delta_{st}(c_1\alpha^{v+1} - c_2 b^{v+1})\varphi^{v} - [\alpha_1\alpha^{s+2} + \alpha_2 b^{s+2}]\dot{\varphi}^{s+1}. \qquad (3)$$

It is not possible to find its exact solution. However, the above imposed restrictions on SPWV internal power factors can be used as perturbation methods general ideas [7, 8]. The effectiveness of their use largely depends on the possibility of finding a solution to an undisturbed analogue of equation (3), in other words

$$I_o\ddot{\varphi}_0 + (c_1\alpha^{v+2} + c_2 b^{v+2})\varphi_0^{v+1} = 0. \qquad (4)$$

Equation (3) and (4) describe SM oscillatory process if the parameter $v+1$ is defined by correlation $v + 1 = (2m + 1)/(2n + 1), (m, n = 0, 1, 2, ..)$. Aperiodic solution of (4) in the given case is expressed through periodic Ateb-function [9] as [10, 11]

$$\varphi_0(t) = \alpha_\varphi c\alpha\big(v + 1, 1, \omega(\alpha_\varphi)t + \theta\big) \qquad (5)$$

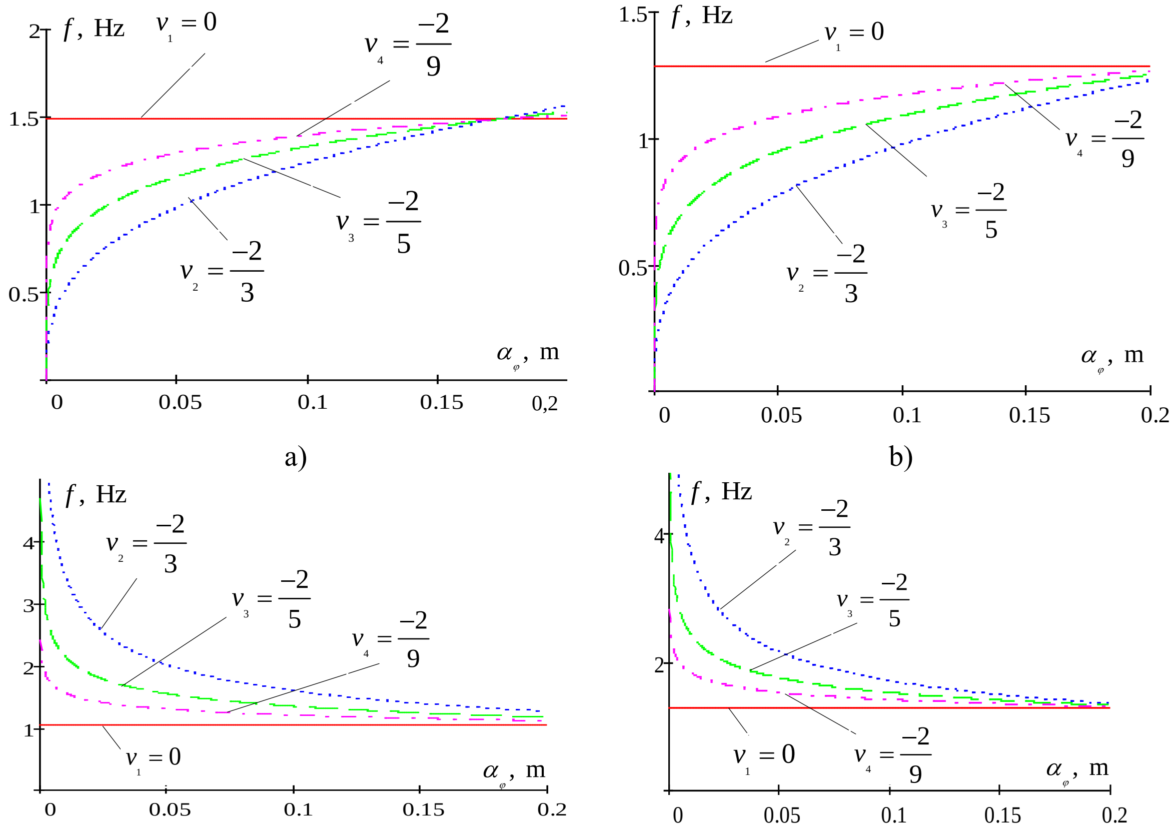
where α_φ, $\omega(\alpha_\varphi) = \sqrt{(c_1\alpha^{v+2} + c_2 b^{v+2})(v + 2)/(2I_0)}\,\alpha_\varphi^{\frac{v}{2}}$ – amplitude and frequency of longitudinally angular SM oscillation respectively, and $\omega(\alpha_\varphi)t+\theta$ – their phase. The frequency of natural oscillations can be replaced by more convenient dependence based on the following considerations: If the "rigidity" of elastic dampers of the vehicle AS are subject to correlation $c_2 = \kappa c_1$ ($\kappa = a/b$ is a known constant), it is more appropriate to use the concept of static deformation of elastic shock absorbers – Δ_{st}. In this case $c_1 = P/\big((1 + \kappa)\Delta_{st}^{v+1}\big)$, and the frequency of natural oscillations enters values

$$\omega(\alpha_\varphi) = \sqrt{P(\alpha^{v+2} + \kappa b^{v+2})(v + 2)/(2(1 + \kappa)I_0\Delta_{st}^{v+1})\alpha_\varphi^{\frac{v}{2}}} \qquad (6)$$

If we consider that the used periodical Ateb – functions $2\Pi = 2\sqrt{\pi}\Gamma\big(1/(v + +2)\big)\Gamma^{-1}\big(1/2 + 1/(v + 2)\big)$ are periodic in phase, the natural frequency f (in hertz) is defined by dependence

$$f = \frac{1}{2\Pi}\sqrt{3g(v+2)(\alpha^{v+2}+\kappa b^{v+2})/(2(1+\kappa)(\alpha^2+b^2+c^2/4)\Delta_{st}^{v+1})}\,\alpha_{\varphi}^{\frac{v}{2}}$$

Fig. 2 shows the dependence of different values of AS power characteristics on the natural frequency f of oscillation of amplitude having the following parameters: a = c = 1 m; b = 1,1 m; k = 1,2; Δ_{sr} = 0.2 m; Δ_{st} = 0.15 m.



a) b)

c) d)

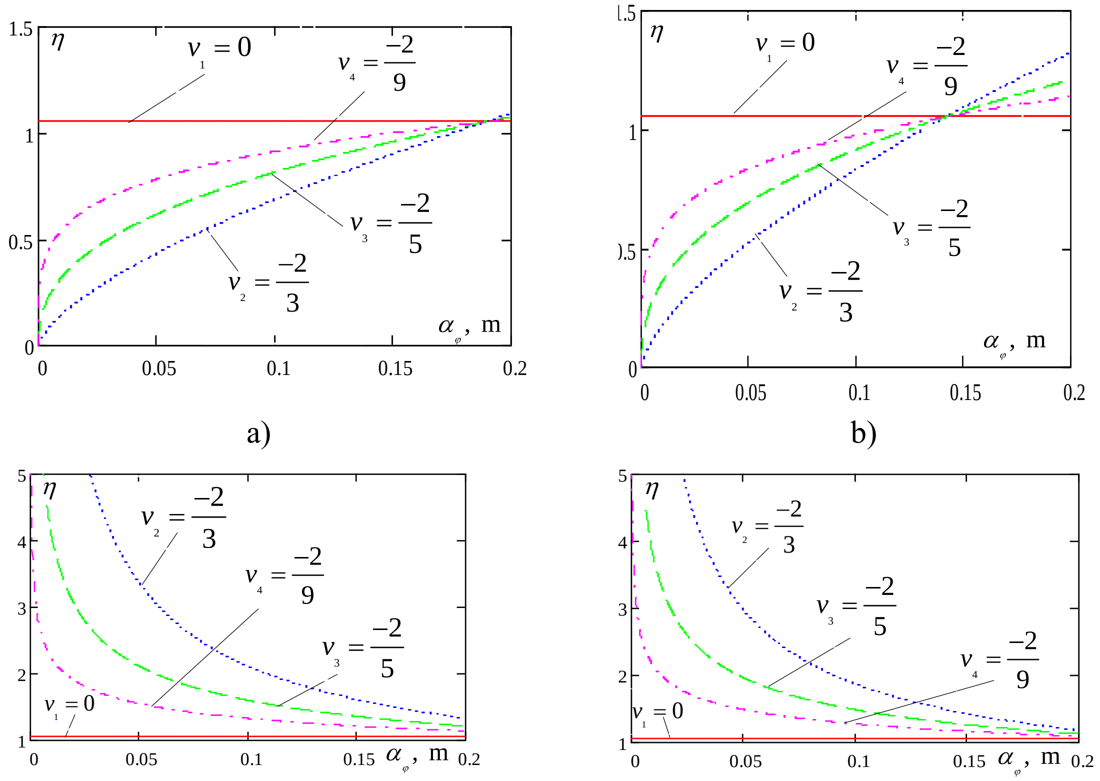
Figure 2. The dependencies of frequency versus amplitude of longitudinal - angular oscillations of absorber system with progressive (a, b) and regressive (c, d) power laws of elastic force

Presented correlations and built on their basis graphic dependencies show one of the fundamental differences of oscillations of vehicle SM with non-linear characteristics of its AS compared to its linear analogue, which mean that the frequency of natural oscillations of vehicle SM depends on the amplitude. In addition, for the AS with progressive characteristic of its elastic shock absorbers the greater amplitude longitudinal angular oscillation value corresponds to higher value of natural frequency, whereas for regressive characteristics it is vice versa: more amplitude value corresponds to less natural frequency value. In addition, the AS progressive characteristic better satisfies SPWV ergonomic operating conditions. As for the dynamic overload acting on a person in the longitudinally - angular oscillation that is positioned at a distance d from the sprung part gravity center, this characteristic of vehicle safety design dynamics is determined in accordance with dependency $w = d\sqrt{\ddot{\varphi}_0^2 + \dot{\varphi}_0^4}$. Considering the relationship (5) and (6) and limiting in dependency to speed the point of SPWV SM maximum tangent value, we obtain $\bar{w} = 2d\alpha_\varphi\omega^2\left(\alpha_\varphi\right)/\left(\nu+2\right)$. Given that for AS with a linear restoring force characteristics ($\nu = 0$) the maximum value of the abovementioned variable equals to

$\bar{w}_l = 3gd\alpha_\varphi\left(a^2+\kappa b^2\right)/\left(\left(1+\kappa\right)\left(a^2+b^2+c^2/4\right)\Delta_{st}\right)$, for the comparative assessment of the impact of AS non-linear characteristics on overload under considered oscillations, we obtain dependence

$$\eta = \frac{\bar{w}}{\bar{w}_l} = \frac{\left(a^{\nu+2}+\kappa b^{\nu+2}\right)}{\left(a^2+\kappa b^2\right)}\left(\frac{\alpha_\varphi}{\Delta_{st}}\right)^{\nu}, \qquad (7)$$

and built graphical dependencies (Fig. 3).



a) b)

c) d)

Figure. 3. The dependence of coefficient η versus amplitude of longitudinal - angular oscillations of absorber system with progressive (a, b) and regressive (c, d) power laws

The presented information implies that the overloading of vehicles, in the case of the progressive power characteristics of the AS its overload is less with its linear analogue at large static deformations and is more at large amplitude vibrations and small static deformations of AS; in the case of regressive power characteristic the dynamic overload is greater than the linear analog of the AS power characteristics.

As for the influence of the resistance on the SP dynamics, as in the case of linear resistance forces, it is manifested in amplitude attenuation of longitudinal – angular oscillations. Amplitude attenuation rate can be calculated using Van der Pol method [8] for the systems of that type. According to the basic idea of Van der Pol method, the first asymptotic approximation that describes longitudinal angular fluctuations of PS considering the resistance can be written as

$$\varphi_0(t)=\alpha_\varphi(t)ca\left(\nu+1,1,\omega\left(\alpha_\varphi\right)t+\theta(t)\right), \tag{7}$$

where $\alpha_\varphi(t)$ and $\theta(t)$ - unknown variables in time function, which change laws are determined by the right-hand side of the equation (5). Differentiating dependence (7) by the time, we get $\frac{d\varphi_0}{dt}=-2/(\nu+2)\omega\left(\alpha_\varphi\right)sa\left(1,\nu+1,\omega\left(\alpha_\varphi\right)t+\theta\right)$. Similarly, we find the first derivative by the time of perturbed motion

$$\begin{gathered}\frac{d\varphi}{dt}=-\frac{2\alpha_\varphi}{\nu+2}\omega\left(\alpha_\varphi\right)sa\left(1,\nu+1,\omega\left(\alpha_\varphi\right)t+\theta\right)+\frac{d\alpha_\varphi}{dt}ca\left(\nu+1,1,\omega t+\theta\right)-\\-\frac{2}{\nu+2}\frac{d\theta}{dt}\omega(a)sa\left(1,\nu+1,\omega t+\theta\right)\ .\end{gathered} \tag{8}$$

By another differentiation of the presented dependence, taking into account the fact that $\frac{d\alpha_\varphi}{dt}ca\left(\nu+1,1\omega t+\theta\right)=\frac{2}{\nu+2}\frac{d\theta}{dt}\omega(a)sa\left(1,\nu+1,\omega t+\theta\right)=0$ we get

$$\begin{gathered}\frac{d^2\varphi(t)}{dt^2}=-\frac{2\alpha_\varphi\omega^2\left(\alpha_\varphi(t)\right)}{\nu+2}ca^{\nu+1}\left(\nu+1,1,\omega\left(\alpha_\varphi(t)\right)t+\theta(t)\right)-\frac{2}{\nu+2}\times\\\times\frac{d\alpha_\varphi}{dt}\left[\omega\left(\alpha_\varphi(t)\right)+a\frac{d\omega\left(\alpha_\varphi\right)}{d\alpha_\varphi}\right]sa\left(1,\nu+1,\omega\left(\alpha_\varphi(t)\right)t+\theta(t)\right)-\\-\frac{2\omega\left(\alpha_\varphi(t)\right)}{\nu+2}\frac{d\theta}{dt}aca\left(\nu+1,1,\omega\left(\alpha_\varphi(t)\right)t+\theta(t)\right).\end{gathered} \tag{9}$$

Obviously, the correlation $\omega\left(\alpha_\varphi\right)+\alpha_\varphi\frac{d\omega\left(\alpha_\varphi\right)}{d\alpha_\varphi}$ can be replaced by a simpler one, namely $(\nu+2)\omega\left(\alpha_\varphi\right)/2$.

Thus, out of the differential equation (3) we can obtain correlation, which describes longitudinal – angular oscillations of SP that is a typical differential equation of the first order, which connects the amplitude and phase of longitudinal - angular perturbed motion oscillations

$$\frac{d\alpha_\varphi}{dt}sa\left(1,\nu+1,\omega\left(\alpha_\varphi\right)t+\theta\right)+\frac{d\varphi}{dt}\frac{2}{\nu+2}\alpha_\varphi ca^{\nu+1}\left(\nu+1,1,\omega\left(\alpha_\varphi(t)\right)t+\theta(t)\right)=\bar{f}\left(\alpha_\varphi,\bar{\theta}\right), \tag{10}$$

where $\bar{\theta}=\omega\left(\alpha_\varphi\right)t+\theta$, $\bar{f}\left(\alpha_\varphi,\bar{\theta}\right)$ corresponds to the right side of equation (3), provided that the function and its first derivative is determined according to the undisturbed case.

Given the preliminary calculations, we can find relationships, describing the laws of change of SP amplitude and frequency of vibrations taking the following form

$$\frac{d\alpha_\varphi}{dt}=-\frac{1}{\omega\left(\alpha_\varphi\right)}sa\left(1,\nu+1,\bar{\theta}\right)\bar{f}\left(\alpha_\varphi,\dot{\bar{\theta}}\right)$$

$$\frac{d\theta}{dt} = -\frac{(\nu+2)}{2\alpha_{\varphi}\omega(\alpha_{\varphi})}\varepsilon c a(\nu+1,1\bar{\theta})\,\bar{f}(\alpha_{\varphi},\dot{\bar{\theta}}) \ . \tag{11}$$

The resulting system of differential equations can be considerably simplified on the basis of these considerations: the value of changes in the amplitude and frequency of vibrations in one period is small because the maximum value of the resistance is much less than the maximum regenerative power shock. This is the basis for averaging [9] in phase of the right sides of the equation (11)

$$\frac{da}{dt} = -\frac{(\alpha_{1}a^{s+2}+\alpha_{2}b^{s+2})\alpha_{\varphi}}{2\Pi I_{0}}\left(\frac{2\alpha_{\varphi}\omega(\alpha_{\varphi})}{(\nu+2)}\right)^{s-1}\Gamma\left(\frac{1}{\nu+2}\right)\Gamma\left(\frac{s+2}{2}\right)\Gamma^{-1}\left(\frac{1}{\nu+2}+\frac{s+2}{2}\right), \tag{12}$$

As expected, the resistance causes a decrease in amplitude over time, and amplitude attenuation rate is higher for larger values of the parameter s. In addition, at first glance it does not affect the frequency of vertical vibrations. However, the natural frequency depends on the amplitude and, therefore, given the above, the frequency of damped oscillations depends on the resistance.

Conclusions. Results obtained using the developed technique indicate that:

– for a progressive power law absorber system higher amplitude longitudinal angular oscillation value corresponds to higher natural frequency value, whereas for regressive power law the opposite is true: increasing value of the amplitude value leads to decreasing value of the natural frequency oscillations;

– SPWV ergonomic operating requirements are better met by progressive characteristics of AS static deformation 0.2 m and $\nu < 2/3$ with transverse angular oscillations amplitude $0,25 < \alpha_{\varphi} < 0,16$; AS static deformation 0.15 m and transverse angular oscillations amplitude $0,05 < \alpha_{\varphi} < 0,2$; AS static deformation 0.1 m.

The results can be the basis for the study of more complex cases of SPWV movement, such as movement along an ordered or disordered system of irregularities, SP resonance vibrations etc.

References

1. Rotenberh R.V. Podveska avtomobylya. M.: Mashynostroenye. 1972 392 p. [In Russian].
2. Kuz'o I.V. Vplyv parametriv pidvisky na neliniyni kolyvannya transportnykh zasobiv I.V. Kuz'o, B.I. Sokil, V.M. Palyukh Visnyk NU "LP" "Dynamika, mitsnist 'ta proektuvannya mashyn i pryladiv". 2007. №588 P. 49-52. [In Ukrainian].
3. Hrubel 'M.H. Vplyv vidnovlyuval'noyi syly pruzhnoyi pidvisky na kolyvannya, ta stiykist 'rukhu kolisnykh transportnykh zasobiv M.H. Hrubel', R.A. Nanivs'kyy, M.B. Sokil Perspektyvy rozvytku ozbroyennya i viys'kovoyi tekhniky: tezy dopovidey Mizhnarodnoyi NTK. L'viv: ASV, 2014. P. 35. [In Ukrainian].
4. Hrubel 'M.H. Vertykal'ni kolyvannya pidresorenoyi chastyny kolisnykh transportnykh zasobiv pid diyeyu vypadkovykh zburen ', M.H. Hrubel ', O.P. Krasyuk, R.A. Nanivs'kyy, M.B. Sokil, Naukovi notatky. – Luts'k, 2014. – №. 46. – P. 112 – 116. [In Ukrainian].
5. Dushchenko V.V. Matematycheskoe modelyrovanye kolebanyy podresorenoho korpusa mnohoopornoy kolesnoy mashyny, V.V. Dushchenko, Y.Y. Yakymenko, Mekhanika ta mashynobuduvannya. 2004. № 2. P. 139-147. [In Russian].
6. Ayzerman M.A. Klassycheskaya mekhanyka, M.A. Ayzerman. M .: Nauka, 1980 p. [In Russian].
7. Koul Dzh. Metody vozmushchenyy v prykladnoy matematyke, Dzhulyan Koul [per. s anhl. A.Y. Derzhavynoy y V.N. Dyesperova, pod red. O.S. Ryzhova], M.: Myr, 1972. 276 p. [In Russian].
8. Boholyubov N.N. Asymptotycheskye metody v teoryy nelyneynykh kolebanyy, N.N. Boholyubov, Yu.A. Mytropol's'kyy. M. : Nauka, 1974 504 p. [In Russian].
9. Senyk P. M. Obernennya nepovnoyi Beta-funktsiyi, P.M. Senyk, Ukr. mat. zhurn. K., 1969. 21, no 3. P. 325 – 333. [In Ukrainian].

10. Senyk P.M. Pro zastosuvannya u-metodyky dlya odnoho klasu kolyvnykh system / Senyk P. M., Sokil B.I., Dop. AN URSR. – 1977, no 1. P. 12 – 16. [In Ukrainian].

11. Senyk P.M. Ob opredelenyy parametrov nelyneynoy kolebatel'noy systemy po amplytudno-chastotnoy kharakterystyke, Senyk P. M., Sokyl B. Y., Mat. metody y fyz.-mekh. polya. 1977.no7. – P. 94 – 99. [In Russian].

Список використаної літератури

1. Ротенберг, Р.В. Подвеска автомобиля [Текст] / Р.В. Ротенберг. – М. : Машиностроение. 1972. – 392 с.

2. Кузьо, І.В. Вплив параметрів підвіски на нелінійні коливання транспортних засобів [Текст] / І.В. Кузьо, Б.І. Сокіл, В.М. Палюх // Вісник НУ “ЛП” “Динаміка, міцність та проектування машин і приладів”. – 2007.– № 588 – С. 49 – 52.

3. Грубель, М.Г. Вплив відновлювальної сили пружної підвіски на коливання та стійкість руху колісних транспортних засобів [Текст] / М.Г. Грубель, Р.А. Нанівський, М.Б. Сокіл // Перспективи розвитку озброєння і військової техніки СВ: тези доповідей Міжнародної НТК. – Львів: АСВ, 2014. – С. 35.

4. Вертикальні коливання підресореної частини колісних транспортних засобів під дією випадкових збурень [Текст] / М.Г. Грубель, О.П. Красюк, Р.А. Нанівський, М.Б. Сокіл // Наукові нотатки. – Луцьк, 2014. – Вип. 46. – С. 112 – 116.

5. Дущенко, В.В. Математическое моделирование колебаний подресореного корпуса многоопорной колесной машины [Текст] / В.В. Дущенко, И.И. Якименко // Механіка та машинобудування. – 2004. – № 2. – С. 139 – 147.

6. Айзерман, М.А. Классическая механика [Текст] / М.А. Айзерман. – М.: Наука,1980 с.

7. Коул, Дж. Методы возмущений в прикладной математике [Текст] / Джулиан Коул; пер. с англ. А.И. Державиной и В.Н. Диесперова; под ред. О.С. Рыжова. – М. : Мир, 1972. – 276 с.

8. Боголюбов, Н.Н. Асимптотические методы в теории нелинейных колебаний [Текст] / Н.Н. Боголюбов, Ю.А. Митропольський. – М.: Наука, 1974. – 504 с.

9. Сеник, П.М. Обернення неповної Beta-функції [Текст] / П.М. Сеник // Укр. мат. журн. – К., 1969. – 21, № 3. – С. 325 – 333.

10. Сеник, П.М. Про застосування u-методики для одного класу коливних систем [Текст] / П.М. Сеник, Б.І. Сокіл. // Доп. АН УРСР. – 1977. – № 1. – С. 12 – 16.

11. Сеник, П.М. Об определении параметров нелинейной колебательной системы по амплитудно-частотной характеристике [Текст] / П.М. Сеник , Б.И. Сокил // Мат. методы и физ.-мех. поля. – 1977. – № 7. – С. 94 – 99.

ПОЗДОВЖНЬО-КУТОВІ КОЛИВАННЯ КОЛІСНИХ ТРАНСПОРТНИХ ЗАСОБІВ ІЗ НЕЛІНІЙНОЮ СИЛОВОЮ ХАРАКТЕРИСТИКОЮ СИСТЕМИ ПІДРЕСОРЮВАННЯ

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Резюме. *Отримано у параметризованій формі аналітичні залежності, які описують визначальні параметри поздовжньо-кутових коливань підресореної частини колісних транспортних засобів. Показано, що для колісних транспортних засобів спеціального призначення, за базу яких вибрано шасі серійних автомобілів, більшою мірою ергономічним умовам задовольняє система підвіски з прогресивним законом зміни пружних амортизаторів.*

Ключові слова: *система підресорювання, поздовжньо–кутові коливання, амплітуда, частота.*

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