Запропоновано метод оцінки керованості автомобіля у сталому режимі руху за допомогою коефіцієнта динамічності. Визначено, що зміна тиску повітря у шинах, з використанням розробленого алгоритму, дозволяє підвищити стійкість автомобіля проти рискання і, відповідно, безпеку дорожнього руху. Отримані результати можуть бути використані як у процесі експлуатації, так і при проектуванні нових конструкцій машин

Ключові слова: керованість, стійкість, коефіцієнт динамічності, сталий рух, автомобіль

Предложен метод оценки управляемости автомобиля в установившемся режиме движения с помощью коэффициента динамичности. Определено, что изменение давления воздуха в шинах, с использованием разработанного алгоритма, позволяет повысить устойчивость автомобиля против рыскания и, соответственно, безопасность дорожного движения. Полученные результаты могут быть использованы как в процессе эксплуатации, так и при проектировании новых конструкций машин

Ключевые слова: управляемость, устойчивость, коэффициент динамичности, установившееся движение, автомобиль

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A METHOD OF EVALUATING VEHICLE CONTROLLABILITY ACCORDING TO THE DYNAMIC FACTOR

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1. Introduction

Handling and driving stability are the most important performance characteristics that influence road traffic safety. When driving in a steady mode, control is one of the features of a more general (complex) property – stability of the vehicle. Manageability at a steady motion allows the driver to promptly and accurately stabilize the azimuth angle and the vehicle turning radius.

The stability and manageability properties should be formed both at the stage of designing new vehicles and during operation of existing vehicles. This issue is especially urgent for trucks with understeer. Development and implementation of specific scientific and technical recommendations based on design and operational parameters of vehicles will increase the efficiency of their operation and road safety in general.

2. Analysis of previous studies and statement of the problem

The problem of ensuring stability of vehicles has been extensively considered in numerous studies published in Ukraine [1] and abroad [2, 3]. These studies, however, express different opinions on the relationship between the most important performance properties – the stability and controllability of a vehicle, indicating absence of a common approach to their establishment and evaluation.

A steady rectilinear motion of a wheeled car takes place at regular periodic vibrations of the guide wheels [4], which relates to the need to ensure the course stability by correcting the yaw rate [5]. However, the published studies insufficiently specify the criteria for handling the rectilinear movement of the vehicle, i. e. the course control criteria.

The authors of [6, 7] describe a wheeled car in the road plane as a resilient oscillating system producing forced oscillations. The equation for these oscillations is the following [6]:

$$\frac{d^2\psi}{dt^2}+2\cdot n\cdot\frac{d\psi}{dt}+\omega_1{}^2\cdot\psi=h\cdot\sin pt, \qquad (1)$$

where ψ is a course angle of the vehicle at any given time; $\frac{d\psi}{dt}=\omega_z$ is an angular velocity of the vehicle in the road plane; $\frac{d^2\psi}{dt^2}=\omega_z$ is an angular acceleration of the vehicle in the road plane; h is the ratio between the amplitude of the driving torque M_{torq} of the wheeled vehicle and the vehicle's moment of inertia in the road plane in relation to the vertical axis stretching through the centre of elasticity I_{Z_M}

$$h = \frac{M_{torq}}{I_{Z_M}}, \quad (2)$$

where ω_1 is a circular natural frequency of the elastic system;

$$\omega_1 = \sqrt{\frac{C_{angle}}{I_{Z_M}}}, \quad (3)$$

where p is an angular frequency of the forced oscillation (in this problem – the frequency of the driver's actions or those of the automatic control device on the steering); $2n$ is a relation of the damping coefficient α_{damp} to the moment of inertia I_{Z_M}.

Forced oscillations occur with a frequency of a disturbing force (in this problem – the torque); and if the circular natural frequency is bigger than the frequency of the disturbing force, there is no phase shift between the oscillations and the force (torque). If the frequency of the disturbing force exceeds the natural frequency, there is a phase shift π (180°) [6]. This method can be used to determine the natural frequencies of single vehicles, but it does not provide an assessment of dynamic oscillations in transient modes of movement.

It is known that the biggest instability of vehicles is revealed during braking [8] and in motion within road trains [9]. When the frequencies of natural and forced oscillations coincide, there is a sharp increase in the amplitude of the forced oscillations, known as resonance. In the absence of damping ($\alpha_{damp}=0$ and $2n=0$), the amplitude A of the forced oscillations increases to infinity. In the vicinity of the resonance point, the value of the forced oscillation amplitude is determined by a dynamic factor F_{dyn} [6].

$$A = A_0 \cdot \left| \frac{1}{1 - \frac{F^2}{\omega_1^2}} \right| = A_0 \cdot \omega_{damp}, \quad (4)$$

where A_0 is a static turn of the system at the amplitude value of torque equal to M_{torq}.

If there is damping (viscous friction forces of resistance), the value of the amplitude of the oscillations is determined by the following correlation:

$$A = A_0 \left| \frac{1}{\sqrt{\left(1 - \frac{p^2}{\omega_1^2}\right)^2 + 4\frac{n^2 \cdot p^2}{\omega_1^4}}} \right|. \quad (5)$$

This approach is used to determine the amplitude of the vehicle's oscillations, but it does not evaluate the impact of structural and operational factors and the technical condition of the vehicle on its directional control.

Thus, adaptation of frequency methods for evaluating resistance to the vehicles' yaw is an expedient issue that requires further research.

3. Research aim and objectives

The aim of the study is to improve the stability and manageability of vehicles at a steady motion by adjusting the air pressure in the tires, which helps reduce the dynamic factor of the vehicle's yaw.

To achieve this aim, it is necessary to solve the following tasks:

– to determine the dynamic factor of the vehicle's yaw;

– to analyse the dependence of the dynamism on the design and operational parameters of the vehicle.

4. Determination of the dynamic factor at the vehicle yaw

To ensure the stability and controllability of a vehicle at a steady rectilinear motion, it is necessary to satisfy the condition $v_{poss}^{max} < v_{nat}$, at which the maximum frequency v_{poss}^{max} of the disturbing torque moment M_{torq} should not exceed the natural frequencies of the vehicle oscillations in the road plane v_{nat}.

Under the condition of $v_{poss}^{max} < v_{nat}$, when v_{poss}^{max} increases from 0 to the value of v_{nat}, the amplitude A increases (5). After the transition from the circular frequencies p and ω to the oscillation frequencies, expressions (4) and (5) will look as follows:

$$A = A_0 \cdot \left| \frac{1}{1 - \frac{v_{poss}^2}{v_{nat}^2}} \right|, \quad (6)$$

$$A = A_0 \left| \frac{1}{\sqrt{\left(1 - \frac{v_{poss}^2}{v_{nat}^2}\right)^2 + \frac{n^2}{4\pi^2}\frac{v_{poss}^2}{v_{nat}^4}}} \right|. \quad (7)$$

5. Analysis of the dynamic factor dependence on the design and operational parameters of the vehicles

As v_{poss} increases, there is an increase in the dynamic factor [6], which is determined by the following dependence:

$$F_{dyn} = \left| \left[\left(1 - \frac{v_{poss}^2}{v_{nat}^2}\right)^2 + \frac{n^2}{4\pi^2}\frac{v_{poss}^2}{v_{nat}^4} \right]^{-\frac{1}{2}} \right|. \quad (8)$$

Since the action of the torque moment M_{torq} is disturbing, the increase in the amplitude A of forced oscillations results in an imbalance between the control action and the vehicle's reaction to it. Therefore, when $v_{poss} < v_{nat}$, the closer the value of the dynamic factor F_{dyn} to 1, the better the controllability and, hence, the stability of the steady rectilinear movement of the vehicle.

Let us consider the worst case in which in expression (8) $n = 0$. This expression will be reduced to the following:

$$F_{dyn} = \left| \frac{1}{1 - \frac{v_{poss}^2}{v_{nat}^2}} \right|. \quad (9)$$

In [10], a formula is suggested to determine the frequencies of natural oscillations of a vehicle in the road plane:

$$\nu_{nat} = \frac{L}{2\pi}\sqrt{\frac{C_{y_1}/m_a}{\left(1+\frac{C_{y_1}}{C_{y_2}}\right)\left[i_Z^2 + a^2\left(\frac{\frac{C_{y_1}}{C_{y_2}}-\frac{b}{a}}{1+\frac{C_{y_1}}{C_{y_2}}}\right)^2\right]}}, \tag{10}$$

where C_{y_1} and C_{y_2} are the total lateral stiffness indices of the wheels of the front and rear axes of the vehicle; m_a is the mass of the automobile; L is the longitudinal wheel base of the vehicle; a and b denote the distance from the front and rear axes to the projections of the vehicle centre of gravity in relation to the horizontal surface; i_z is the vehicle inertia radius in relation to the vertical axis.

Let us estimate the possibility of obtaining the maximum frequency of natural oscillations of the vehicle in the road plane. The analysis of equation (10) shows that an increase of ν_{poss} is possible either at a higher C_{y_1} or as a result of the equation:

$$\frac{C_{y_1}}{C_{y_2}} - \frac{b}{a} = 0. \tag{11}$$

We assume that the ratio of the total lateral stiffness of the front and rear wheels of the vehicle is equal to the ratio of the coefficients of resistance to lateral withdrawal, i. e.:

$$\frac{C_{y_1}}{C_{y_2}} \cong \frac{C_{y_1}}{C_{y_2}}, \tag{12}$$

condition (11) expresses the condition for a neutral steering of the vehicle, when the neutral steering of the vehicle is $\delta_1 = \delta_2$.

Thus, the neutral steering of the vehicle allows getting the maximum value of the frequency of natural oscillations ν_{max} in the road plane.

Equation (10) also shows that at both

$$\frac{C_{y_1}}{C_{y_2}} - \frac{b}{a} > 0, \tag{13}$$

and

$$\frac{C_{y_1}}{C_{y_2}} - \frac{b}{a} < 0, \tag{14}$$

the natural frequency is less than under condition (11). This means that vehicles with understeer and oversteer have their natural frequency lower than automobiles with neutral steering. It is obvious that the operation of two-axial vehicles with single wheels on the driving axes at changing their centre of mass position, it is possible to satisfy condition (11) by adjusting the internal air pressure in the tires of the front and rear wheels.

Equation (10) with regard to the correlation $2n = \frac{\gamma}{I_{kz}}$ and equation (11) will look as follows:

$$\nu_{nat} = \frac{1}{2\pi}\sqrt{\frac{a}{L}\frac{2C_{y_1}/m_a}{\frac{a}{L}\left(1-\frac{a}{L}\right)+\frac{B^2}{6L^2}}}. \tag{15}$$

In Fig. 1, there are graphs of the dependence $\nu_{nat}\left(\frac{a}{L}\right)$ at m_a=8,100 kg; L =4.2 m; B =2.0 m; C_{y_1} =1.475×10[5] N/m (curve 1) and C_{y_2} =2.264×10[5] N/m (curve 2).

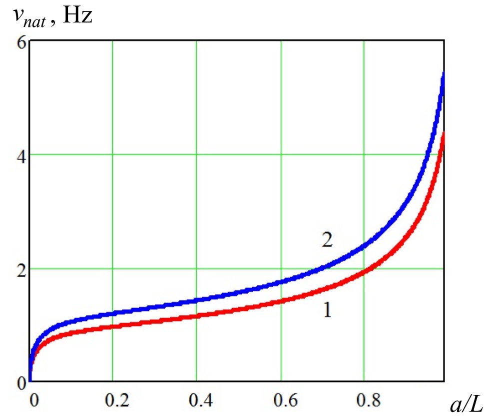

Fig. 1. The dependence $\nu_{nat}\left(\frac{a}{L}\right)$ for vehicles with a neutral steering: m_a=8,100 kg; L=4.2 m; B=2.0 m; C_{y_1}=1.475×10[5] N/m (curve 1) and C_{y_2} =2.264×10[5] N/m (curve 2)

The analysis of the graphs given in Fig. 1 shows that in vehicles with a neutral steering an increase in the parameter a/L (a displacement of the centre of mass to the front axis) increases the natural frequency of oscillations in the road plane.

When expression (10) is added to equation (9), we obtain the following:

$$F_{dyn} = \left|\frac{1}{1-\nu_{poss}^2\cdot\frac{4\pi^2 m_a}{L^2 C_{y_1}}\left(1+\frac{C_{y_1}}{C_{y_2}}\right)\left[i_Z^2 + a^2\left(\frac{\frac{C_{y_1}}{C_{y_2}}-\frac{b}{a}}{1+\frac{C_{y_1}}{C_{y_2}}}\right)^2\right]}\right|. \tag{16}$$

The analysis of expression (16) shows that with a higher ν_{poss} there is an increase of the F_{dyn}. If $\nu_{poss} = 0$, the value of F_{dyn}=1. With an increase in the vehicle base L, there is a lower dynamic factor F_{dyn}, whereas with a bigger overall weight, the F_{dyn} increases. The F_{dyn} also increases with a bigger radius of inertia of the automobile i_Z in relation to the vertical axis and with a lower C_{y1}. In Fig. 2, there are graphs of the dependence $F_{dyn}\left(\frac{C_{y_1}}{C_{y_2}}\right)$ for the vehicle Ural-4320 at different values of C_{y1} (the benchmark data for the plotting of the graphs are shown in Table 1). The analysis of these graphs shows the presence of a minimum value of F_{dyn}.

On achieving the neutral steering of the vehicle (C_{y1}/C_{y2}= =b/a), the right side of expression (16) becomes reduced. Therefore, expression (16) becomes as follows:

$$F_{dyn}^{**} = \left|\frac{1}{1-\nu_{poss}^2\cdot\frac{4\pi^2 m}{L\cdot a\cdot C_{y_1}} i_Z^2}\right|. \tag{17}$$

We assume that $\nu_{poss}=\nu^{max}_{poss}$=0.7 Hz. Equation (17) becomes as follows:

$$F_{dyn}^{**}=\left|\frac{1}{1-19.34\cdot\frac{m_a}{L\cdot a\cdot C_{y_1}}i_Z^2}\right|. \qquad (18)$$

Table 1

The geometric parameters and the weight of the vehicle Ural-4320

The condition of the vehicle	The specifications of the vehicle					
	m, kg	a, m	h, m	L, m	B, m	i_z, m
Equipped	8,100	2.390	1.270	4.225	2.0	1.598
Fully loaded	13,025	2.970	1.230	4.225	20	1.482

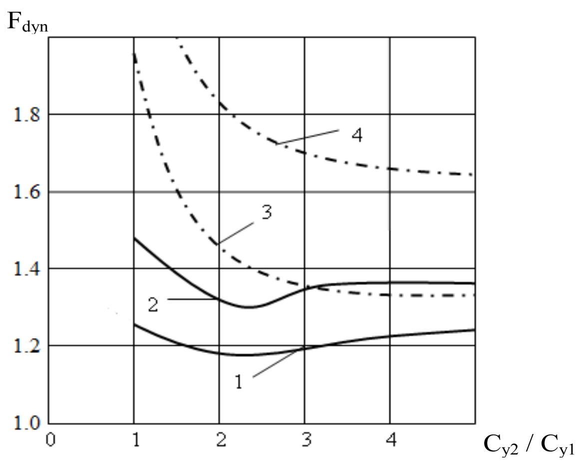

Fig. 2. The dependence of the dynamic factor on the relationship C_{y2}/C_{y1}: 1 and 2 denote an equipped vehicle; 3 and 4 denote a fully loaded vehicle; 1 and 3 refer to $C_{y1}=2.264\cdot10^5$ N/m; 2 and 4 refer to $C_{y1}=1.475\cdot10^5$ N/m

In Fig. 3 there are the graphs of the dependence $F_{dyn}^{**}\left(C_{y_1}\right)$ for an equipped Ural-4320 and a fully loaded Ural-4320.

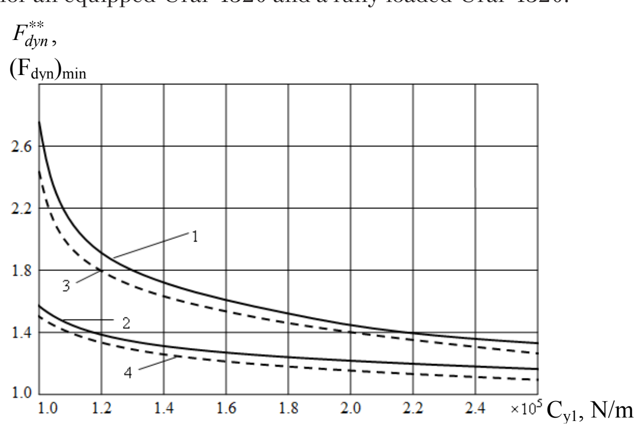

Fig. 3. The dependence of the dynamic factor on the total lateral rigidity of the front wheel tires: 1 and 2 – $F_{dyn}^{**}\left(C_{y_1}\right)$; 3 and 4 – $(F_{dyn})_{min}(C_{y1})$; 1 and 3 – a loaded vehicle Ural-4320; 2 and 4 – an equipped vehicle Ural-4320

The analysis of the graphs in Fig. 3 reveals that in a fully loaded vehicle Ural-4320 the dynamic factor is higher than in an equipped vehicle.

When the C_{y1} value increases at the described loads of the vehicle Ural-4320, the difference of the values F_{dyn}^{**} decreases.

The right side of equation (16) is minimal if

$$\left(\frac{C_{y_1}}{C_{y_2}}\right)^*=\frac{L}{\sqrt{i_Z^2+a^2}}-1. \qquad (19)$$

When equation (19) is integrated into equation (16), the result is the following:

$$F_{dyn}=\left|\frac{1}{1-v_{poss}^2\cdot\frac{4\pi^2 m_a}{L^2\cdot C_{y_1}}\cdot F\left(\frac{C_{y_1}}{C_{y_2}}\right)}\right|. \qquad (20)$$

The minimal value $(F_{dyn})_{min}$ can be determined by integrating the expression

$$\left[F\left(\frac{C_{y_1}}{C_{y_2}}\right)\right]_{min}=L\cdot\frac{i_Z^2+\left(a-\sqrt{i_Z^2+a^2}\right)^2}{\sqrt{i_Z^2+a^2}}, \qquad (21)$$

in (20)

$$(F_{dyn})_{min}=\left|\frac{1}{1-v_{poss}^2\cdot\frac{4\pi^2 m_a}{L\cdot C_{y_1}}\cdot\frac{i_Z^2+\left(a-\sqrt{i_Z^2+a^2}\right)^2}{\sqrt{i_Z^2+a^2}}}\right|. \qquad (22)$$

Fig. 3 also contains graphs of the dependence $(F_{dyn})_{min}\left(\frac{C_{y_1}}{C_{y_2}}\right)$ for the vehicle Ural-4320 when it is equipped and fully loaded. For the Ural-4320, the ratio of the lateral rigidity of the tires $\left(\frac{C_{y_1}}{C_{y_2}}\right)^*$ is 0.47 for an equipped vehicle, and 0.22 for a loaded vehicle.

For the vehicle Ural-4320, the neutral-steering ratio of the total lateral stiffness of the front wheels and the balancing wheel suspension is:

– $\frac{C_{y_1}}{C_{y_2}}$ = 0.531 for an equipped vehicle;

– $\frac{C_{y_1}}{C_{y_2}}$ = 0.757 for a fully loaded vehicle.

The results are a basis for further research on the yaw stability of multi-axial vehicles and multi-unit automobile trains.

6. Conclusion

1. The dynamic factor of the vehicle yaw $(F_{dyn})_{min}$ only slightly differs from F_{dyn}^{**}. However, the correlations C_{y1}/C_{y2} differ greatly. For an equipped vehicle, C_{y1}/C_{y2}= =0.531 at a neutral steering. With $F_{dyn}=(F_{dyn})_{min}$, the value $(C_{y1}/C_{y2})^*$ is equal to 0.47. For a fully loaded vehicle, these values are 0.414 and 0.22, respectively. Therefore, for the

loaded truck Ural-4320 having single tires on all wheels, it can be recommended to ensure that $F_{dyn}. = F^{*}_{dyn}$ at a neutral steering of the vehicle. At equal lateral rigidities in all six wheels in this case C_{y1}/C_{y2}=1/2, which is close to 0.531.

2. The design and operational parameters have a significant influence on the vehicle's dynamic factor. With a fully loaded vehicle Ural-4320, C_{y1} should be increased relatively to C_{y2} by adjusting the internal air pressure in the tires. For trucks Ural-4320 with dual tires on the equalizer trolley and with equal lateral rigidity of the tires, the ratio C_{y1}/C_{y2}=1/4 allows having $(F_{dyn})_{min}$ when the vehicle is loaded. If the vehicle is loaded, it is necessary to reduce the air pressure in the tires of the wheels of the equalizer trolley with respect to the air pressure in the tires of the front wheels. This helps bring the ratio C_{y1}/C_{y2} to the value of 0.47, which can allow obtaining $(F_{dyn})_{min}$. For two-axial vehicles to have a neutral steering, it is necessary to control and adjust the ratio of the air pressure in the tires of the front and rear wheels.

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