Розглядається задача параметричного синтезу електронного блоку керування гальмовою системою автомобіля, яка полягає в процесі вибору значень варійованих параметрів електронного блоку, що забезпечують максимальний запас стійкості і максимальну швидкодію замкнутої системи автоматичного управління заданим напрямом руху автомобіля в режимі гальмування. Це важливо, тому що сучасні швидкісні автомобілі що експлуатуються в умовах неякісних дорожніх покриттів автомобільних доріг, часто стають об'єктами дорожньо-транспортних пригод, незважаючи на оснащення їх антиблокувальними і противобуксувальними системами. У зв'язку з цим конструкторські розробки новітніх зразків автомобільної техніки спрямовані на створення систем курсової стійкості, підвищеної керованості автомобіля при втраті зчеплення коліс з дорожнім покриттям. Поставлена задача вирішується за допомогою методу «заморожених коефіцієнтів», в основі якого покладено ітераційний процес послідовного вибору чисельних значень варійованих параметрів електронного блоку. Зокрема встановлено, що характеристичне рівняння замкнутої системи підтримки курсової стійкості містить три варійованих параметра, а сама система є двоканальною.

Перший канал управління по куту відхилення корпусу автомобіля містить варійовані параметри k_{ψ} і $k_{\dot{\psi}}$, другий канал управління по бічному зносу містить варійований параметр k_y. Показано, що 5÷6 кроків ітераційного процесу достатньо для отримання оптимальних значень варійованих параметрів, що забезпечують максимальний запас стійкості і максимальну швидкодію замкнутої системи курсової стійкості автомобіля. В роботі доказана ефективність розроблених алгоритмів параметричного синтезу електронного блоку керування автомобілем, простота їх реалізації, а також висока якість процесів стабілізації корпусу автомобіля. Оптимальні значення варійованих параметрів складають $k_{\psi}^{*} = 399$ В, $k_{\dot{\psi}}^{*} = 13,8$ В·с, $k_y^{*} = 143$ В·м$^{-1}$, при цьому запас стійкості системи складає $\alpha^{*} = -8,8$ с$^{-1}$

Ключові слова: керованість автомобіля, система курсової стійкості, канал кутового відхилення корпуса, канал бічного зсуву центру мас

UDC 629.3.017.5

DOI: 10.15587/1729-4061.2019.188185

PARAMETRIC SYNTHESIS OF THE ELECTRONIC CONTROL UNIT OF THE COURSE STABILITY SYSTEM OF THE CAR

Ye. Aleksandrov
Doctor of Technical Sciences, Professor
Department of Automobiles
named after A. B. Gredeskul
Kharkiv National Automobile and Highway University
Yaroslava Mudroho str., 25, Kharkiv, Ukraine, 61002
E-mail: aleksandrov.ye.ye@gmail.com

T. Aleksandrova
Doctor of Technical Sciences, Associate Professor*
E-mail: aleksandrova.t.ye@gmail.com

Y. Morhun
Postgraduate Student*
E-mail: yarki95@gmail.com

*Department of System Analysis and Information-Analytical Technologies
National Technical University
"Kharkiv Polytechnic Institute"
Kyrpychova str., 2, Kharkiv, Ukraine, 61002

Received date 04.09.2019
Accepted date 11.11.2019
Published date 27.12.2019

1. Introduction

Currently, the intensity of road traffic is growing steadily, which requires improved traffic safety. The fulfillment of this requirement can be carried out by improving the quality of roads and organizing traffic on them, as well as by improving the safety of the vehicles themselves. The main role is given to the active safety of vehicles, since the solution to this problem ensures the prevention of traffic accidents. Recent advances in electronics and microprocessors have contributed to improving the reliability and safety of cars. The performance of the vehicle's brake system directly affects its dirigibility and stability. That is why the control and analysis of the technical condition of this car system is given priority. The design developments of the latest models of automotive equipment are aimed at creating course stability systems, increased vehicle dirigibility in case of loss of wheel adhesion with the road surface, which determines the relevance of scientific research in this direction.

2. Literature review and problem statement

To assess the dirigibility of the car in the steady-state driving mode, a dynamic coefficient is introduced in [1] and concluded that "the closer the value of this coefficient to unity, the better the dirigibility and, accordingly, the stability of the rectilinear movement of the car". However, it should be noted that to maintain the required value of the coefficient of dynamism, the authors propose to adjust the ratio of air pressure in the tires of the front and rear wheels of the car, and this is very difficult to implement in practice. In [2], it is proposed to increase the dirigibility and stability of the car during braking by a targeted distribution of braking moments between the wheels of the front and rear axles. The paper shows that the distribution of drive forces can have a significant impact on the stability characteristics of the vehicle. However, this approach is relevant only for vehicles with four-wheel drive, and also, as in [1], the technical implementation of the proposals received in [2] cause significant difficulties. In [3, 4], the authors set out the calculation methods and the results of experimental studies of an automated vehicle braking control system that provides stable movement of the car body during braking. The results of experimental studies have shown the high efficiency of the system under conditions of vehicle braking on both dry asphalt and rolled snow surfaces. An analysis of the accuracy of calculating the coefficient of utilization of traction force by an automated system with and without taking into account the rolling resistance of the wheels of a vehicle showed the possibility of using the proposed calculation methods in the practice of automotive technical expertise in the investigation of traffic accidents involving vehicles equipped with systems such as ABS. However, it should be borne in mind that the automatic control of the vehicle's brake system consists of three parallel automatic control systems. This is not only the ABS anti-lock braking system, which protects the wheels from blocking when the brake pedal is pressed sharply, but also the TRC traction control system, which protects the drive wheels from towing when the fuel supply pedal is pressed too hard, and the VSC system of course stability, which improves dirigibility a car with a loss of traction [5]. In [6], various systems for measuring the angular deviation of the body of a wheeled or tracked vehicle from a given direction of its movement are considered. The simplest method is using a ratio linking the angular velocities of the drive wheels of the right and left sides with the angular velocity of rotation of the vehicle body, however this is possible only in the absence of skid phenomena and slipping of the drive wheels characteristic of the braking conditions of vehicles. Analyzing the literature, it is possible to conclude that almost all of them are associated with the analysis of systems that provide dirigibility and stability of the car during braking. And the development of any automated system is a sequence of solutions, interspersed between the problems of analysis and synthesis, including the tasks of parametric synthesis. As a result of solving these problems, the values of the varied design parameters of the system are determined that provide the specified requirements for the controlled processes. Publications related to the synthesis of brake systems of a car, based on the achievements of modern control theory, are practically absent in the world scientific and technical literature, which makes it necessary to conduct research in this direction.

3. The aim and objectives of research

The aim of research is finding the values of the variable parameters of the electronic control unit of the brake system of the car, providing high quality processes for controlling the car braking, first of all, providing the required safety margin and performance.

To achieve the aim, the following objectives are set:

– to build a mathematical model of the disturbed movement of the closed-loop course stability control system of the car;

– to carry out parametric synthesis of the closed-loop VSC system using the "frozen coefficients" method and to analyze the stabilization processes of the vehicle body during braking.

4. The mathematical model of the disturbed movement of the closed-loop course stability system of the car

The automatic control of the car brake system consists of the following parallel-acting automatic control systems: firstly, the ABS anti-lock braking system; secondly, the TRC traction control system; thirdly, the system of maintaining vehicle stability control VSC [5].

Both the ABS system and the TRC system constantly maintain stability in the direction of travel of the car. However, the accuracy of maintaining the car's directional stability with the help of these two systems may be insufficient in case of loss of traction by two front or two rear wheels and a car skid characteristic for this situation. In this case, the VSC system comes into operation, the working diagram of which is shown in Fig. 1. The MBC is controlled by the driver using the brake pedal BP.

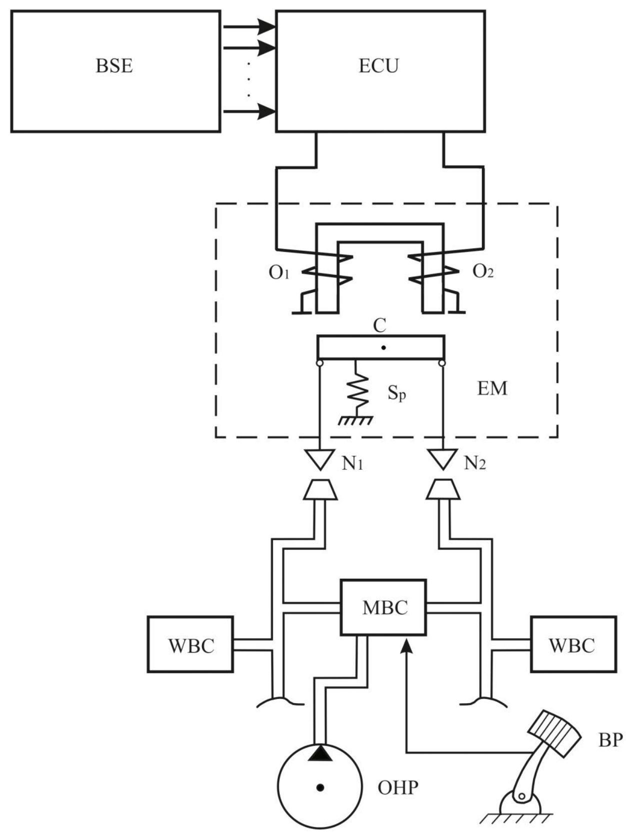

Fig. 1. The working scheme of the course stability system: BSE – block of sensitive elements; ECU – electronic control unit; EM – electromagnet; O_1 and O_2 – control windings; C – electromagnet rocker; Sp – fixing spring; N_1 and N_2 – locking needles; OHP – hydraulic pump, MBC – main brake cylinder; WBC – working brake cylinders of the wheels of the left and right sides of the car; BP – break pedal

If the car during braking makes a movement in the strictly required direction, then under the action of a fixing spring the beam of the electromagnet is in the neutral position. And the brake fluid pressure created by the head cylinder is the same in the working cylinders of the wheels of the starboard and starboard sides of the car.

When the car is skidded, the corresponding BSE sensors react to it, and the ECU generates a non-zero control signal, which leads to the deviation of the magnet arm from the neutral position and to the corresponding movement of the locking needles N_1 in N_2. In this case, the brake fluid pressure in the working cylinders of the running side of the car increases, and in the working cylinders of the lagging side decreases, which leads to the return of the car body to a predetermined position [6].

The simplest way to measure the angular deviation of the body of a wheeled or tracked vehicle from a given direction of its movement is use the kinematic relation

$$\dot{\psi}(t)=\frac{r}{B}\left[\omega_{2r}(t)-\omega_{2l}(t)\right], \tag{1}$$

linking the angular velocities of the driving wheels of the right and left sides бортов $\omega_{2r}(t)$ and бортов $\omega_{2l}(t)$ with the angular velocity of rotation of the vehicle body $\dot{\psi}(t)$, in which the radius of the drive wheel is indicated by r and the track gauge by B [6]. However, relation (1) is valid only in the absence of skid phenomena and skidding of the driving wheels characteristic of the braking conditions of vehicles. More accurate information about the angular deviation of the vehicle body from a given direction can be obtained using a gyro-stabilized platform (GSP) with accelerometers installed on it, the sensitivity axes of which coincide with the main central axes of inertia of the body. Using this combination of sensors, it is possible to obtain information not only about the angular deviation of the body from a given direction, but also the lateral demolition of the center of mass in case of loss of wheel adhesion with the road. However, the GSP is very sensitive to various kinds of disturbances, especially to shock and jolts characteristic of driving on the road, not to mention its very high cost. In this regard, in [6], the authors proposed to choose a strapdown inertial system (SIS) as the sensitive elements of the course stability system of the car. SIS contains three gyroscopic angular velocity sensors (GAVS), the sensitivity axes of which coincide with the main central axes of inertia of the vehicle body. And also, an on-board digital computer (BDC), which implements the SIS algorithm, with the help of which it is not the measurement of the angle $\psi(t)$, but its calculation through the parameters of Rodrigues-Hamilton [7, 8]. In addition to the GAVS, the course stability system contains an accelerometer, the sensitivity axis of which coincides with its own transverse axis of the body and with the help of which it measures the lateral drift of the vehicle's center of mass $y(t)$. The scheme of the analog system of the course stability system of the car is presented in Fig. 2 [6].

In the scheme shown in Fig. 2, the BDC plays the auxiliary role of a computing device that implements the SIS algorithm for calculating the mismatch angle $\psi(t)$. Analogue VSC of a car implements a control algorithm as follows

$$U(t)=-k_{\psi}\psi(t)-k_{\dot{\psi}}\dot{\psi}(t)+k_y y(t), \tag{2}$$

where k_{ψ}, $k_{\dot{\psi}}$ and k_y – the variable ECU parameters.

In addition to the angular velocity sensors and the accelerometer, the DSE also contains a Doppler sensor for the current vehicle velocity $v(t)$.

The control signal (2) generated by the ECU is supplied to one of the windings 01 or 02 of the EM electromagnet. The magnitude of the electric current $i(t)$ flowing through any of the windings 01 or 01 is determined by the equation

$$L_o\frac{di(t)}{dt}+r_o i(t)=u(t), \tag{3}$$

where r_o and L_o are the ohmic resistance and inductance of the windings 01 and 02, respectively.

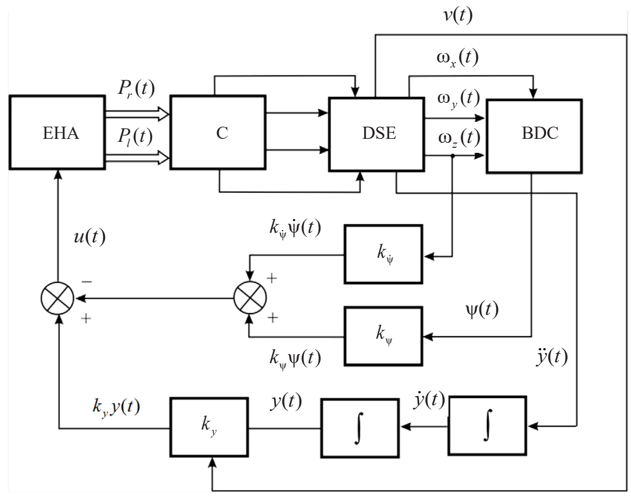

Fig. 2. Scheme of analogue VSC

When the current $i(t)$ flows through any of the windings 01 or 02, the rocker arm R of the electromagnet EM turns in one direction or another by an angle $\gamma(t)$ in accordance with the equation

$$I_r\frac{d^2\gamma(t)}{dt^2}+f_r\frac{d\gamma(t)}{dt}+c_r\gamma(t)=k_e i(t), \tag{4}$$

where I_r – the moment of rocker arm inertia relative to the axis of rotation; f_r – the coefficient of hydraulic friction in the axis of rotation of the rocker arm; c_r – the stiffness coefficient of the fixing spring Sp; k_e – the proportionality coefficient.

The pressure difference of the working fluid in the brake lines of the right and left sides of the car $\Delta p(t)$ is

$$\Delta p(t)=k_h\,\gamma(t).$$

The disturbing movement of the car as an adjustable object is recorded [6]:

$$I_c\frac{d^2\psi(t)}{dt^2}=k_c\Delta p(t)+M_d(t); \tag{5}$$

$$\frac{dy(t)}{dt}=-v(t)\psi(y), \tag{6}$$

where I_c – the moment of car inertia relative to its own vertical axis of inertia; k_c – the proportionality coefficient; $M_d(t)$ – the disturbing moment; $v(t)$ is the current velocity of the center of mass of the vehicle.

Equations (2)–(6) are a mathematical model of the disturbed movement of a closed system of course stability system of the car.

Let's introduce the notation

$$T_o = \frac{L_o}{r_o};\ T_r^2 = \frac{I_r}{c_r};\ T_{2kg} = \frac{f_r}{c_r};\ T_d^2 = \frac{I_c}{k_c k_p};$$

$$k_o = \frac{1}{r_o};\ k_m = \frac{k_e k_o}{c_r T_m^2}.$$

As a result, the mathematical model of a closed system in operator form takes the following form:

$$(T_o s+1) i(t) = k_o\left[-k_\psi \psi(t) - k_\psi s\psi(t) + k_y y(t)\right];$$

$$\left(T_{1r}^2 s^2 + T_{2r} s + 1\right)\gamma(t) = \frac{k_e}{c_r} i(t);$$

$$T_m^2 s^2\, \psi(t) = \gamma(t) + \frac{1}{k_c k_h} M_d(t);$$

$$s\, y(t) = -v(t)\, \psi(t). \quad (7)$$

The system of differential equations (7) is solvable with respect to the highest derivatives

$$s\, i(t) = -\frac{1}{T_o} i(t) - \frac{k_o}{T_o} k_\psi\, \psi(t) -$$
$$-\frac{k_o}{T_o} k_\psi\, s\,\psi(t) + \frac{k_o}{T_o} k_y\, y(t);$$

$$s^2\gamma(t) = -\frac{1}{T_{1r}^2}\gamma(t) - \frac{T_{2r}}{T_{1r}^2} s\,\gamma(t) + \frac{k_e}{T_{1r}^2 c_r}\, i(t);$$

$$s^2\psi(t) = -\frac{1}{T_m^2}\gamma(t) + \frac{1}{T_m^2 k_o k_h} M_d(t);$$

$$s\, y(t) = -v(t)\psi(t). \quad (8)$$

Let's introduce the state vector of a closed system of the 6th order

$$X(t) = \begin{bmatrix} x_1(t) \\ x_2(t) \\ x_3(t) \\ x_4(t) \\ x_5(t) \\ x_6(t) \end{bmatrix} - \begin{bmatrix} i(t) \\ \gamma(t) \\ s\,\gamma(t) \\ \psi(t) \\ s\,\psi(t) \\ y(t) \end{bmatrix}.$$

Then let's write the system of differential equations (8) in the normal Cauchy form:

$$s x_1(t) = \frac{1}{T_o} x_1(t) - \frac{k_o}{T_o} k_\psi x_4(t) -$$
$$-\frac{k_o}{T_o} k_\psi x_5(t) + \frac{k_o}{T_o} k_y x_6(t);$$

$$s x_2(t) = x_3(t);$$

$$s x_3(t) = \frac{k_e}{T_{1r}^2 c_r} x_1(t) - \frac{1}{T_{1r}^2} x_2(t) - \frac{T_{2r}}{T_{1r}^2} x_3(t);$$

$$s x_4(t) = x_5(t);$$

$$s x_5(t) = \frac{1}{T_m^2} x_2(t) + \frac{1}{T_m^2 k_c k_h} M_d(t);$$

$$s x_6(t) = -v(t) x_4(t). \quad (9)$$

System (9) is written in vector-matrix form:

$$s\, X(t) = A(k) X(t) + F(t), \quad (10)$$

where k – the 3-dimensional vector of variable system parameters; A(k) – the proper matrix of the system; $F(t)$ – the 6-dimensional vector of external disturbances:

$$A(k) = \begin{bmatrix} -\frac{1}{T_o} & 0 & 0 & -\frac{k_o}{T_o}k_\psi & -\frac{k_o}{T_o}k_\psi & \frac{k_o}{T_o}k_y \\ 0 & 0 & 1 & 0 & 0 & 0 \\ \frac{k_e}{T_{1r}^2 c_r} & -\frac{1}{T_{1r}^2} & -\frac{T_{2r}}{T_{1r}^2} & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & \frac{1}{T_m^2} & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & -v(t) & 0 & 0 \end{bmatrix};$$

$$k = \begin{bmatrix} k_\psi \\ k_\psi \\ k_y \end{bmatrix};\ F(t) = \begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \\ \frac{M_d(t)}{T_m^2 k_c k_h} \\ 0 \end{bmatrix}.$$

The characteristic equation of a linear non-stationary system (10) is written

$$R(s) = \det\left[A(k) - Es\right] = 0,$$

or

$$R(s) = T_{1r}^2 T_o s^6 + \left(T_{2r}^2 T_o + T_{1r}^2\right) s^5 + \left(T_o + T_{2r}\right) s^4 +$$
$$+ s^3 + k_m k_\psi s^2 + k_m k_\psi s + k_m v(t) k_y = 0. \quad (11)$$

The closed-loop course stability control system under consideration is non-stationary, since in the last equation of system (9), the coefficient $v(t)$ representing the current vehicle velocity during braking depends on time. The free term of the characteristic equation also contains coefficient $v(t)$. In the process of parametric synthesis, let's use the most common method for studying non-stationary dynamic systems, called the "method of frozen coefficients" [9]. In accordance with this method, the operating time of the system T is divided into N intervals of duration $\frac{T}{N}$ each. During each of the intervals, the value of the time-varying coefficient is considered constant, equal to its value at the beginning of the interval. If to assume that the deceleration of the car during braking is constant and equal to – w, then the change in its current velocity is determined by the dependence

$$v(t) = v_0 - wt, \quad (12)$$

where v_0 – the car velocity at the time of the braking start (Fig. 3).

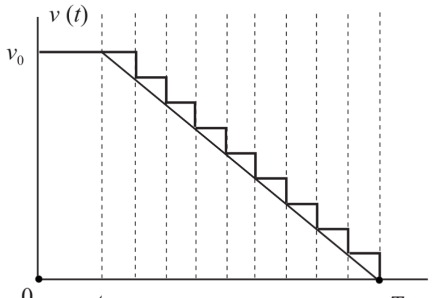

Fig. 3. To the method of "frozen coefficients"

The method of "frozen coefficients" involves the approximation of a linear function (12) by the piecewise constant function shown in Fig. 3. In accordance with this method, the non-stationary system (9) is stable on the braking interval (t_0, t_0+T) if each of the stationary subsystems is stable in the sections $i=\overline{1,N}$. For each interval $i=\overline{1,N}$, the characteristic equation of a closed system takes the following form

$$R(s)=T_{1r}^2T_os^6+\left(T_{2r}T_o+T_{1r}^2\right)s^5+\left(T_o+T_{2r}\right)s^4+$$
$$+s^3+k_mk_{\dot\psi}s^2+k_mk_\psi s+k_mv_ik_y=0; \qquad (13)$$

$$i=\overrightarrow{1,N}.$$

The method of "frozen coefficients" does not have a rigorous theoretical justification, but with its help many practical problems of the analysis and synthesis of non-stationary dynamic systems are solved.

5. Parametric synthesis of a closed-loop VSC system

The characteristic equation (13) of the closed-loop VSC system contains three variable parameters k_ψ, $k_{\dot\psi}$ and k_y, and the VSC system itself is two-channel. The first control channel for the angle of deviation of the car body contains variable parameters k_ψ and $k_{\dot\psi}$, the second control channel for lateral skid contains a variable parameter k_y (Fig. 2).

The iterative process described below is proposed to select the numerical values of these parameters. At the initial step of the iterative process, it is assumed that k_y [0]=0 and the characteristic equation (13) is written as

$$A(s)=T_{1r}^2T_os^5+\left(T_{2r}T_o+T_{1r}^2\right)s^4+\left(T_o+T_{2r}\right)s^3+$$
$$+s^2+k_mk_{\dot\psi}[1]s+k_mk_\psi[1]=0, \qquad (14)$$

where $k_{\dot\psi}[1], k_\psi[1]$ – the first approximations of the varied parameters k_ψ and $k_{\dot\psi}$.

In the characteristic equation (14) let's make the change [10]

$$s=\alpha+j\omega, \qquad (15)$$

select the real and imaginary parts in the obtained ratio, equate them to zero, and from the obtained equations let's obtain

$$k_\psi[1]=\frac{1}{k_m}\begin{Bmatrix}4T_{1r}^2T_o\left(\alpha^5+\alpha^3\omega^2-\alpha\ \omega^4\right)+\\+\left(T_{2r}T_o+T_{1r}^2\right)\left(3\alpha^4+2\alpha^2\omega^2-\omega^4\right)+\\+\left(T_{2r}+T_o\right)\left(2\alpha^3+5\alpha\omega^2\right)+\alpha^2+\omega^2\end{Bmatrix}; \qquad (16)$$

$$k_{\dot\psi}[1]=\frac{1}{k_m}\begin{Bmatrix}-T_{1r}^2\ \ {}_o\left(5\alpha^4-10\alpha^2\omega^2+\omega^4\right)-\\-4\left(T_{2r}T_o+T_{1r}^2\right)\left(\alpha^3-\alpha\omega^2\right)-\\-\left(T_{2r}+T_o\right)\left(3\alpha^2-\omega^2\right)-2\alpha\}.\end{Bmatrix}.$$

Using relations (16) in the plane of variable parameters (k_ψ [1] and $k_{\dot\psi}$ [1]), let's draw lines of equal degree of stability when ω changes from zero to infinity for various values of α<0. For α=0, the constructed line is the boundary of the stability region of the closed system. With an increase in the absolute value of α, lines of equal degree of stability contract to a point *a*, which corresponds to the maximum margin of stability and the maximum performance of the closed system (Fig. 4). With the values of the parameters of the mathematical model equal

$k_т=1{,}9\ V^{-1}$; $T_o=0{,}1{\cdot}10^{-3}$ s;

$T_{1r}^2=10^{-4}\,cs^2$; $T_{2r}=0{,}55\cdot10^{-2}$ s

the maximum stability margin of the channel of angular deviation is $\alpha_\psi[1]=-13{,}8\ s^{-1}$, and the values of the first approximations of the varied parameters k_ψ [1] and $k_{\dot\psi}$ [1] corresponding to point *a* are

$$k_\psi^*[1]=90{,}49\ \text{V};$$

$$k_{\dot\psi}^*[1]=13{,}40\ \text{V}\cdot\text{s}.$$

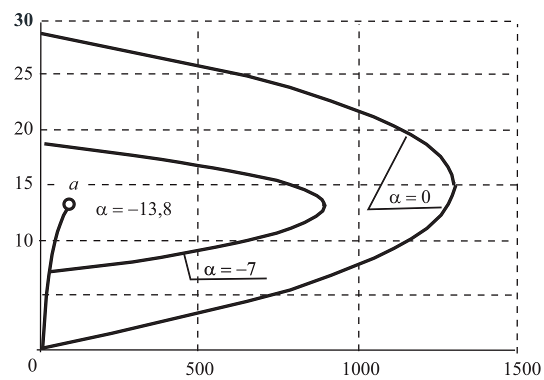

Fig. 4. Lines of equal degree of stability in the plane of variable parameters k_ψ [1], $k_{\dot\psi}$ [1]

Taking into account the obtained values $k_\psi^*[1]$ and $k_{\dot\psi}^*[1]$ of the first approximations of the varied parameters of the angular deviation channel, let's solve the characteristic equation (3) with respect to the parameter k_y

$$k_y[1]=\frac{1}{k_mv_o}\begin{Bmatrix}-k_mk_\psi^*[1]s-k_mk_{\dot\psi}^*[1]s^2-s^3-\\-\left(T_o+T_{2r}\right)s^4-\\-\left(T_{2r}T_o+T_{1r}^2\right)s^5-T_{1r}^2T_os^6\end{Bmatrix} \qquad (17)$$

and in relation (17) let's make the change (15). As a result, let's obtain:

$$\operatorname{Re} k_y[1] = \frac{1}{k_m v_o}\begin{Bmatrix} -k_m k_\psi^*[1]\alpha - k_m k_{\dot\psi}^*[1]\left(\alpha^2-\omega^3\right) - \\ -\left(\alpha^3 - 3\alpha\omega^2\right) - \\ -\left(T_o + T_{2r}\right)\left(\alpha^4 - 6\alpha^2\omega^2 + \omega^4\right) - \\ -\left(T_{2r}T_o + T_{1r}^2\right)\left(\alpha^5 - 10\alpha^3\omega^2 + 5\alpha\omega^4\right) - \\ -T_{1r}^2 T_o\left(\alpha^6 - 15\alpha^4\omega^2 + 15\alpha^2\omega^4 - \omega^6\right) \end{Bmatrix}, \tag{18}$$

$$\operatorname{Im} k_y[1] = \frac{1}{k_m v_o}\begin{Bmatrix} -k_m k_\psi^*[1]\omega - k_m k_{\dot\psi}^*[1]2\alpha\omega - \\ -\left(3\alpha\omega^2 - \omega^3\right) - \\ -\left(T_o + T_{2r}\right)\cdot 4\alpha\omega\left(\alpha 2^4 - \omega^2\right) - \\ -\left(T_{2r}T_o + T_{1r}^2\right)\begin{pmatrix} 5\omega\alpha^4 - \\ -10\alpha^2\omega^3 + \omega^5 \end{pmatrix} - \\ -T_{1r}^2 T_o \cdot 2\alpha\omega\left(3\alpha^4 - 10\alpha^2\omega^2 + 3\omega^4\right) \end{Bmatrix}. \tag{19}$$

In the complex plane ($\operatorname{Re} k_y$ [1], $\operatorname{Im} k_y$[1]) let's draw lines of equal degree of stability for various values of $\alpha<0$, shown in Fig. 5. These lines limit the sections of the real axis, which are the stability regions of the closed system along the lateral displacement channel. With an increase in the absolute value of α, the segments of the real axis bounded by lines of an equal degree of stability contract to point b, which determines the value of the first approximation of the variable parameter $k_y^*[1]$, which for the accepted values of the parameters of the mathematical model is $k_y^*[1] = 9.35\ \text{V}\cdot\text{m}^{-1}$ at $\alpha_y[1] = -4.5\ \text{s}^{-1}$.

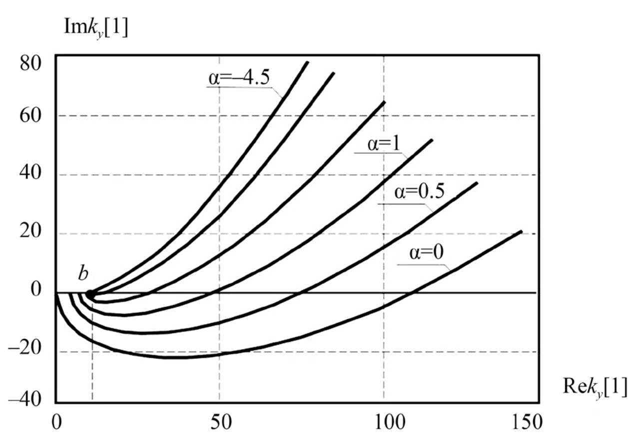

Fig. 5. Lines of equal degree of stability in the plane of the complex parameter k_y [1]

It should be noted that the stability margins of the closed-loop control system along the channels of angular deviation and lateral displacement α_ψ [1] and α_y [1] are different. This suggests that the dynamic processes along the channels of the system at a first approximation of the varied parameters are of different tempo, and the margin of stability of the system as a whole is determined by the stability of the channel with a minimum margin [11]. Thus, the values of the varied parameters obtained at the first step of the iterative process provide a margin of stability of the system as a whole $\alpha[1]=\alpha_y[1]=$ $=-4.5\ \text{s}^{-1}$ [11].

At the next step of the iterative process in characteristic equation (12), let's assume that $k_\psi = k_\psi[2]$; $k_{\dot\psi} = k_{\dot\psi}[2]$; $k_y = k_y^*[1]$, and replace (14) in it, select the real and imaginary parts, and equate them to zero. Then, let's select the values $k_\psi[2]$, $k_{\dot\psi}[2]$ in the obtained relations and determine the point of maximum stability margin in the ($k_\psi[2]$, $k_{\dot\psi}[2]$) plane, and then using the relations (18) and (19) let's obtain the following approximation of the variable parameter $k^*[2]$. Usually, 5–6 steps of the described iterative process are enough to obtain optimal values of the variable parameters k_ψ^*, $k_{\dot\psi}^*$ and k_y^*, providing the maximum stability margin and the maximum performance of the closed-loop course stability system, and the components is $k_\psi^* = 399\ \text{V}$; $k_{\dot\psi}^* = 13.8\ \text{V}\cdot\text{s}$; $k_y^* = 143\ \text{V}\cdot\text{m}^{-1}$.

An analysis of relations (18) and (19) shows that the values of the variable parameter k_y^* depend on the current car velocity during braking and increase with decreasing velocity $v(t)$. The value $k_y^* = 143\ \text{V}\cdot\text{m}^{-1}$ is calculated at the maximum braking start velocity $v_{0\max}$=20 m·s^{-1}. It is advisable to change the coefficient k_y^* in time depending on the current velocity of the car, measured by the Doppler velocity sensor, in accordance with the formula [12]

$$k_y^*(t) = \frac{k_y^* v_{0\max}}{v(t)}. \tag{20}$$

For small values of the current velocity of movement $v(t)$, the variable coefficient $k_y^*(t)$ becomes unacceptably large; therefore, it is advisable to choose the static characteristic of the Doppler velocity sensor in the form shown in Fig. 6.

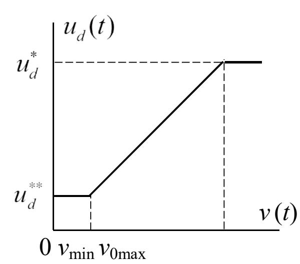

Fig. 6. Static characteristic of the Doppler velocity sensor

Fig. 7 shows the stabilization processes of the car body during braking with constant acceleration w, when the velocity of the center of mass changes in accordance with formula (12), for different values of v_0 and w.

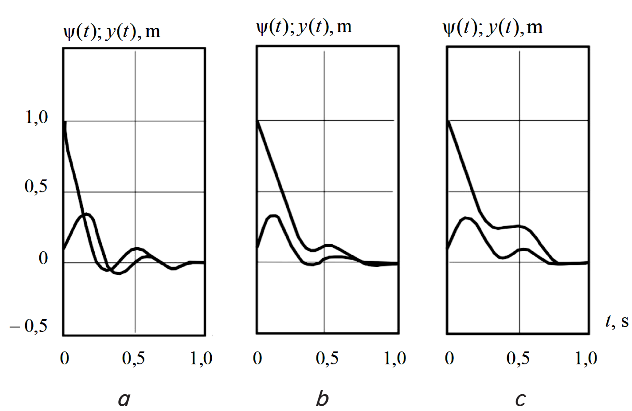

Fig. 7. Transient stabilization processes of the oidy: a – v_0=20 m·s^{-1}; w=4 m·s^{-2}; b – v_0=15 m·s^{-1}; w=3 m·s^{-2}; c – v_0=10 m·s^{-1}; w=2 m·s^{-2}

Analysis of the processes shown in Fig. 7 leads to the following conclusions:

– with a decrease in the initial braking velocity v_0, the time of transient processes decreases – if for v_0=20 m·s^{-1} the regulation time is 1 s, for v_0=15 m·s^{-1} the regulation time is 0.85 s, then for v_0=10 m·s^{-1} regulation time does not exceed 0.75 s;

– with decreasing values of v_0, the oscillation index of the closed-loop course stability system decreases, and the transition process along the coordinate of lateral drift $y(t)$ approaches aperiodic.

6. Discussion of the research results of parametric synthesis of the electronic control unit of the brake system of the car

The values of the variable parameters of the electronic unit are obtained, which ensure the maximum safety margin and maximum performance, which are the main quality indicators of a closed two-channel control system, which is the aim of research. The high quality of the synthesized system is ensured by an adequate mathematical model of the stabilization object and the closed system as a whole, strict use of the selected algorithm for the parametric synthesis of the system and the subsequent analysis of the stabilization processes of the car body. The formulated problem is solved by the method of successive approximations using the method of "frozen coefficients". The method is based on an iterative process of sequential selection of numerical values of the variable parameters of the electronic unit. The method of "frozen coefficients" does not have a rigorous theoretical justification, but with its help many practical problems of the analysis and synthesis of non-stationary dynamic systems are solved. In the future, the possibility of solving this problem using the algorithmic method of synthesis of an automatic control system is considered. The method is based on the direct calculation of the integral quadratic quality functional with its subsequent minimization using the Optimization Toolbox or Minimize software packages. The method also offers a targeted choice of weight coefficients of the additive quality functional, reflecting a set of tendencies towards a stabilized object [13].

7. Conclusions

1. A mathematical model of the disturbed motion of a closed-loop course stability system of a sixth-order automobile is obtained, which most fully describes the physical processes that occur in the course stability system during car braking.

2. Parametric synthesis of the closed-loop VSC system is carried out using the "frozen coefficients" method. Stability domains are constructed in the plane of variable parameters. As a result, the optimal values of the varied parameters are obtained, providing the maximum safety margin and maximum performance of the closed-loop course stability system. They amounted to $k_\psi^* = 399\,\mathrm{V}$; $k_{\dot\psi}^* = 13.8\,\mathrm{V}\cdot\mathrm{s}$; $k_y^* = 143\,\mathrm{V}\cdot\mathrm{m}^{-1}$. In this case, the margin of stability of the system is $\alpha^* = -8.8\,\mathrm{s}^{-1}$.

3. An analysis of the stabilization processes of the vehicle body during braking made it possible to conclude that, with a decrease in the initial braking velocity v_0, the transient time decreases. Also, the oscillation index of the closed-loop course stability system decreases, and the transient in the lateral drift coordinate $y(t)$ approaches aperiodic.

References

1. Turenko, A., Podrygalo, M., Klets, D., Hatsko, V., Barun, M. (2016). A method of evaluating vehicle controllability according to the dynamic factor. Eastern-European Journal of Enterprise Technologies, 3 (7 (81)), 29–33. doi: https://doi.org/10.15587/1729-4061.2016.72117
2. Reński, A. (2016). Analysis of the Influence of the Drive Force Distribution Between Axles on an Automobile Stability in Its Curvilinear Motion. CONAT 2016 International Congress of Automotive and Transport Engineering, 55–63. doi: https://doi.org/10.1007/978-3-319-45447-4_6
3. Zalohin, M. Y., Liubarskyi, B. A., Schuklinov, S. N., Mychalevych, M. G., Leontiev, D. V. (2018). Study of Proportional Pressure Modulator on the Basis of Electromagnetic-Type Linear Motor. Science & Technique, 17 (5), 440–446. doi: https://doi.org/10.21122/2227-1031-2018-17-5-440-446
4. Leontiev, D. N., Nikitchenko, I. N., Ryzhyh, L. A., Lomaka, S. I., Voronkov, O. I., Hritsuk, I. V. et. al. (2019). About Application the Tyre-Road Adhesion Determination of a Vehicle Equipped with an Automated System of Brake Proportioning. Science & Technique, 18 (5), 401–408. doi: https://doi.org/10.21122/2227-1031-2019-18-5-401-408
5. Bazhynov, O. V., Smyrnov, O. P., Serikov, S. A. et. al. (2018). Hibrydni avtomobili. Kharkiv: KROK, 327.
6. Aleksandrov, E. E., Volkov, V. P., Podrigalo, M. A. et. al. (2007). Povyshenie ustoychivosti i upravlyaemosti kolesnyh mashin v tormoznyh rezhimah. Kharkiv: NTU «KhPI», 320.
7. Branets, V. N., Shmyglevskiy, I. P. (1992). Vvedenie v teoriyu besplatformennyh navigatsionnyh sistem. Moscow: Nauka, 280.
8. Panov, A. P. (1995). Matematicheskie osnovy teorii inertsial'noy orientatsii. Kyiv: Naukova dumka, 279.
9. Igdalov, I. M., Kuchma, L. D., Polyakov, N. V., Sheptun, Yu. D. (2010). Dinamicheskoe proektirovanie raket. Dnepropetrovsk: DNU, 264.
10. Orurk, I. A. (1965). Novye metody sinteza lineynyh i nekotoryh nelineynyh sistem. Moscow-Leningrad: Nauka, 207.
11. Aleksandrov, Y. Y., Aleksandrova, T. Y. (2017). The Method of Main Coordinate in the Theory of Parametric Synthesis of the Linear Stabilized Systems. Journal of Automation and Information Sciences, 49 (3), 34–45. doi: https://doi.org/10.1615/jautomatinfscien.v49.i3.50
12. Aleksandrov, Ye. Ye. (2010). Osnovy avtomobilnoi avtomatyky. Kharkiv: KhNADU, 172.
13. Aleksandrova, T. Y., Aleksandrov, Y. Y. (2015). Parametric Synthesis of Digital Stabilization System of Tank Gun. Journal of Automation and Information Sciences, 47 (11), 1–17. doi: https://doi.org/10.1615/jautomatinfscien.v47.i11.10