VEHICLES

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THE CONCEPT OF THE CLUTCH CONTROL LAW OF A CAR

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Abstract. Problem. *The combination of comfortable automatic control of the car's transmission and at the same time preservation of high indicators of energy efficiency and cost of a design is possible by use of automatic mechanical transmission. The automatic clutch control system plays a significant role in providing comfort in such transmissions. The laws of controlling it are not perfect today.* ***Goal.*** *The aim of the work is to create a clear concept of the law of clutch control, which is easy to implement in a microcontroller and is well adapted to adapt to different driving conditions.* ***Methodology.*** *Graphically, the concept of the perspective law of clutch control is formed by two Bezier curves. One of the curves acts as a guide, and the other forms the surface of the law.* ***Results.*** *On the basis of the Bézier curves of the third degree the concept of the law is formed and the connection of the reference points of the Bézier curves with the physical parameters of the working process of vehicle movement is substantiated.* ***Originality.*** *The formation of the Bézier curve, which is decisive for the concept of the law, is formed on the basis of a typical working process of synchronization of the angular velocity of the clutch discs during the movement of the vehicle. In contrast to the laws of clutch control considered in the scientific literature, the proposed concept provides for clutch control outside the site of the synchronization process and ensures the avoidance of jerks during further acceleration.* ***Practical value.*** *The proposed algorithm provides full engagement of the clutch only after full synchronization of the clutch discs. The formation of a special form of the law in the form of a curve tangent to the abscissa axis reduces the jerks when closing the clutch discs.*

Key words: *clutch control law, Bezier curve, jerk effect.*

Introduction

With the development of automatic control systems for transmission units, the clutch control laws have been evolving in the same way as their hardware. During the existence of electronic units with unconditional logic, simple control laws were used [1]. There is the law of control of the bite point as a function of the angle of rotation of the accelerator pedal and as a function of the speed of the engine crankshaft among them. The latter is more adjusted to adapt to the pace of movement of the accelerator pedal because it responds directly to the operation of the engine, and not to the position of the accelerator pedal. As far as there is a delay corresponding to the constant time of the engine between them. During the creation of systems with unconditional control logic [1] various manual mode switches were provided to ensure the adaptation of the control law to the appropriate road conditions.

Currently, these laws have been modernized into one with the possibility of a gradual change in the law of control of the bite point:

$$M_C = f(n_{cr};\alpha), \qquad (1)$$

where M_C – bite point; N·m;

α – the angle of the accelerator pedal, deg.;

n_{cr} – the crankshaft speed of the internal combustion engine, min^{-1}.

Thus, the law (1) is the simplest modern law of clutch control and is a field of characteristics. The field of characteristics is formed and bounded by the second-order polynomials that determine the dependence of the crankshaft speed.

Analysis of publications

In the papers [2, 3], the clutch control is proposed to be performed both by the body of the clutch control device and by fuel supply to the engine based on the lack of direct connection between the accelerator pedal position and the amount of fuel supplied to the internal combustion engine. This approach to fuel management provides both environmental performance and the operation of related vehicle systems [4, 5] (ASR, cruise control or adaptive cruise control). Accordingly, the law of control of the clutch and the engine is as follows (2):

$$\begin{bmatrix} T_e \\ T_c \end{bmatrix} = \begin{bmatrix} 1 + s \cdot J_e \cdot G(s) & 1 \\ 1 & 1 \end{bmatrix} \cdot \begin{bmatrix} v_e \\ v_{sl} \end{bmatrix}, \qquad (2)$$

where T_e – engine torque, N·m;

T_c – bite point, N·m;

s – wheel slippage;

J_e – the moment of inertia of the engine;

v_e – engine speed;

v_{sl} – slip speed;

$G(s)$ – function, which is defined by the formula (3):

$$G(s) = \frac{s \cdot J_v + b_w}{\begin{bmatrix} (s \cdot (J_c + J_t + J_w \cdot r^2) + \\ + r^2 \cdot b_w) \cdot (s \cdot J_v + b_w) - \\ - b_w^2 \cdot r^2 \end{bmatrix}}, \qquad (3)$$

where J_v – equivalent moment of inertia of a vehicle;

J_c – the moment of inertia of a clutch;

J_t – the moment of inertia of the transmission elements;

J_w – the moment of inertia of the drive wheels;

r – transmission gear ratio;

b_w – rigidity of driving wheels.

In a modified form, this law is given in the paper [3] where the authors propose to use the PI-regulator in some parts of the law of clutch control. To implement such a law, it is necessary to set a large enough number of parameters related to the vehicle, which is not convenient. Parameters J_v and b_w can create special complexity. If for a passenger car the change in the weight of the car in operation is not decisive because it is up to 20…30 % of the maximum weight of the vehicle [6], then for the bus this indicator reaches 50 % [7], and for the truck this indicator reaches the critical points of 200…250 % [8] and for some road trains it reaches 300 % [9]. In addition, the movement of the truck can be carried out with different gears. Usually trucks start with the second gear at low load and with the first gear at maximum loads or difficult traffic conditions. Thus, all this causes a significant change in the J_v parameter that can be predicted, but only after the start of the vehicle [10]. Naturally, the first movement after loading may not be as provided by the control law. Although the stiffness of the tires does not change as significantly as the weight of the vehicle, but it is not constant and varies according to the temperature, the degree of wear, the pressure, the design and the manufacturer. Therefore, this introduces standard deviations. In addition, the paper [11] states that the control law based on the feedback on the speed of the clutch or other transmission shafts causes significant jumps in torque and acceleration of the vehicle after closing the clutch discs. These jumps are characterized by such a parameter as the first derivative of the acceleration of the vehicle by time [12, 13, 14]. Thus, the paper [11] proposes to use a law based on the control of torque in the transmission to reduce the jumps.

The works of Belarusian researchers [15, 16, 17] are aimed at creating a system with the control law, which is based on a feedback on the angular acceleration of the driven parts of the clutch with the division of the control algorithm into two possible accelerations supported by the algorithm during starting. Switching between the branches of the algorithm occurs depending on exceeding the limit moving speed of the fuel pedal. The feature of the system proposed in above-noted works is the independent control of the internal combustion engine, which is aimed at preventing the engine from stalling and maintaining the appropriate speed of the crankshaft in the process of starting.

The paper [18] proposes to present the law of clutch control in the form of a function (4):

$$\begin{gathered} S = F(\alpha_e, \dot{\alpha}_e, \omega_e, \dot{\omega}_e, \mathrm{t}) = \\ = \mathrm{A}(\dot{\alpha}_\mathrm{e}) \cdot \left[F_1(\alpha_e) + \mathrm{F}_2(\omega_e)\right] + \\ + B(\mathrm{M}_\mathrm{c}, k, t, \dot{\alpha}_e) \end{gathered}, \qquad (4)$$

where S – the characteristic which is regulated by the control law;

$\mathrm{A}(\dot{\alpha}_\mathrm{e})$ and $B(M_c, k, t, \dot{\alpha}_e)$ – indexes that exclude the “jerk” effect of a clutch pedal.

Analysis of publications

Based on a review of the existing laws of clutch control, it can be confirmed that they do not take into account the real characteristics of the actuators; some of them adapt hardly to the changes in traffic and are not simple ones to be implemented in the microcontroller. Therefore, the aim of the work is to create a visual law that is easy to implement in a microcontroller and that has the ability to adapt to changing traffic conditions.

The concept of the law of clutch control

To ensure the coherence with the ideology of control of conventional automatic transmissions, the starting should begin immediately after releasing the brake pedal with the additional and parking brakes deactivated, as well as with the transmission engaged. Thus starting from a place is possible without adjustment of this process by the driver at the minimum steady angular velocity of a cranked shaft of the engine. During such a starting, the vehicle will move out of location and will drive at a constant minimum speed which corresponds to the engaged transmission. Further acceleration is possible if the accelerator pedal is affected and the engine torque is increased when the clutch is fully engaged. It is essentially that such a starting is possible providing the appropriate road conditions, which are characterized by the coefficient of road resistance ψ. Thus, at this stage of starting, the control law has the general form (5):

$$S = f(Z, \alpha_e, \omega_e, \omega_c), \qquad (5)$$

or, with the driver's influence on the accelerator pedal in the process of controlling (6):

$$[S, E] = f(Z, \alpha_e, \omega_e, \omega_c), \qquad (6)$$

where S – the position of the rod of the actuator of the clutch control, %;

E – the degree of fuel distribution, %;

Z – complex signal of brakes engagement.

The position of the rod of the actuator of a clutch control is controlled by a PID controller [19] in proportion to the signal that simulates the desired position of the rod of the actuator of a clutch control S.

As it is noted by many researchers [2, 15, 20], the prevention of the "jerk" effect during the clutch locking can be achieved with a fairly smooth, and almost tangent acquisition of the same values of the angular velocity of the crankshaft and driven clutch discs. This workflow, with the clutch control made by the driver, is shown in Fig. 1.

With the usual display of the process, its analysis is quite difficult. For example, it is necessary to monitor the infeasibility of stumbling of the clutch discs separately until the angular velocity of the driven discs reaches the operating range of the internal combustion engine (otherwise it will stop the engine).

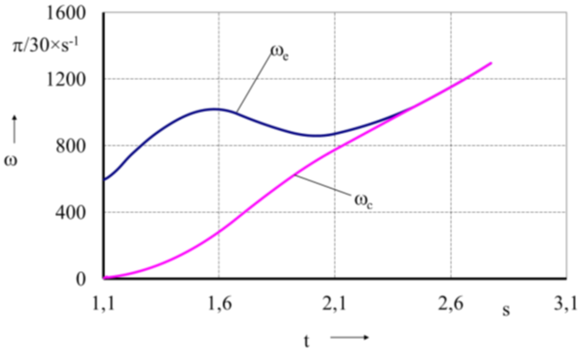

Fig. 1. A typical process of changing the angular velocitys during the clutch skidding with a smooth closing of the clutch discs

It is necessary to monitor that during acceleration without pressing the accelerator pedal the angular velocity of the crankshaft does not become less than a constant minimum speed. In addition, the process of forming the tangent nature of the change in angular velocitys during the closing of the clutch discs is quite difficult to organize in coordinates using time.

Let us make the following transformations for more convenient representation of the process. Let us recreate the working process of towing the clutch in the coordinates $\Delta\omega_e = f(\omega_c)$. As $\Delta\omega$ let us take $\omega_e - \omega_c$ as a subtraction. The result of the transformation can be interpreted as the law of clutch control in a static setting (excluding the time of the process of starting). In this case, the dynamics of the towing process does not matter (Fig. 2).

Analyzing the obtained dependence, we can say that under any circumstances, with the value of $30 \cdot \omega_c / \pi < 500\,\text{min}^{-1}$ the clutch closing should not be carried out. The maximum value of the function $\Delta\omega$ should depend on the desired dynamics of the movement. And the point of contact along the axis ω_c is responsible for the duration of the clutch towing and the ability to enter a zone of maximum torque during the clutch towing in the process of starting. The point of traversal of the curve with the y-axis is responsible for the possibility of starting without affecting the accelerator pedal. As we see, the working process of starting, which is reproduced in this form, can be analyzed very well, has its characteristic features and there are good opportunities for its adaptation.

Reproduction of this curve by polynomial correspondence has a rather complex form, and the most important information is very difficult to transform for the adaptation of this law to different conditions.

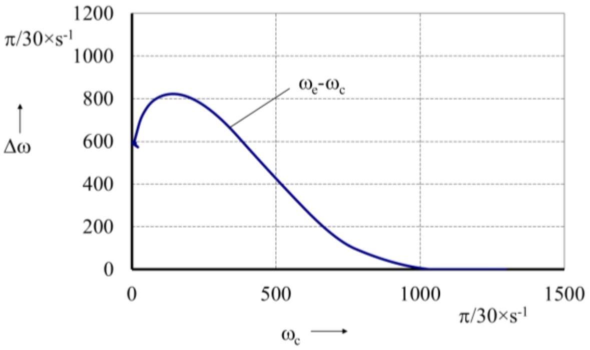

Fig. 2. Clutch towing working process with the time excluded

To reproduce this dependence qualitatively and, that is the most important, to ensure a high-quality touch to the abscissa axis, it is necessary to use a polynomial of the fifth or sixth degree. The determinant for the possibility of adaptation is the preservation of the tangent to the abscissa axis, rather than the accuracy of the reproduction of the curve, so it is suggested to replace the proposed function with the Bézier curve [21, 22, 23]. Bézier curves were developed by a French mathematician and were used for the computer modeling of "Renault" car bodies. Parametric polynomial Bézier curves are the sum of Berstein polynomials which are the basic functions of the Bézier curve. With the availability of a cell array $P=\{P_0, P_1, \ldots, P_m\}$, the parametric Bézier curve of m degree is determined by the following formula (7):

$$\mathrm{R}(\mathrm{t}) = \sum_{i=0}^{m} B_i^m(t) \cdot P_{i,} \quad {}_{t \in [0 \ldots 1]}, \tag{7}$$

where $B_i^m(t)$ – the basic function of the Bézier curve which is determined by the formula (8);

t – parameter that varies in the range from 0 to 1;

$P_i = \begin{pmatrix} x_i \\ y_i \end{pmatrix}$ – coordinates of the i-th point.

$$B_i^m(t) = C_m^i \cdot t^i \cdot (1-t)^{m-i}, \tag{8}$$

where C_m^i – parameter, which is determined by the formula (9):

$$C_m^i = \frac{m!}{i! (m-i)!}. \tag{9}$$

The important properties of the Bézier curve for us are:

– the degree of the polynomial $\mathrm{R}(\mathrm{t})$ that defines the Bézier curve as "one less than the number of reference points". For example, for four reference points, the Bézier curve will be of the third degree;

– the extreme points of the Bézier curve coincide with the start and end points of the P array;

– since $R(0) = m \cdot (P_1 - P_0)$ and $R(1) = m \cdot (P_m - P_{m-1})$, the vectors of the tangents at the ends of the Bézier curve completely coincide with the extreme links of the reference polyline in the direction, and their length affects the encroaching speed of the Bézier curve to the tangent in direct proportion;

– the Bézier curve has a derivative at each point because it is a smooth curve;

– polynomial (7) unambiguously describes the Bézier curve because there are no free parameters in it.

To reproduce a typical working process during the starting of a vehicle, let us perform an approximation of the representation of the process shown in Figure 2 using the Bézier curve. For this purpose, let us note the characteristic points of the curve.

These are the extreme points of the curve, the point P_0 with the coordinates (0, 600) and P_3 with the coordinates (1000, 0). Also, to ensure a guaranteed tangent to the axis ω_c, we have a point P_2 with the coordinates (x_2, 0). The x_2 coordinate value determines how quickly the derivative to the left of the point P_3 will change. The only free point left for the approximation of the curve is the point P_1 with the coordinates (x_1, y_1).

Thus, only three numbers (x_1, x_2, y_1) determine the quality of the curve approximation that characterizes the desired process of synchronization of the angular velocitys of the leading and driven parts of the clutch.

This approach is very convenient to use for the adaptive changes in the electronic control unit because it is much more convenient to control such coordinates than the coefficients of the polynomial which often acquire very inconvenient values for processing in ECU (Fig. 3).

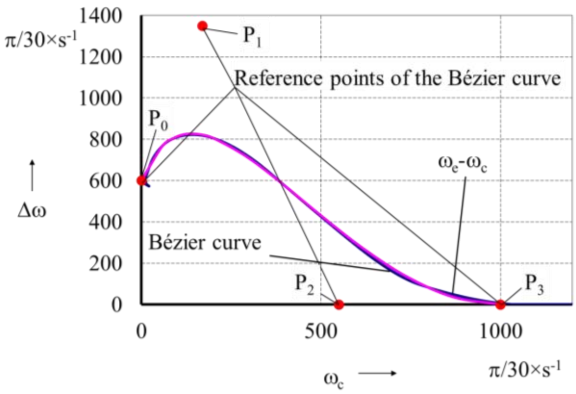

Fig. 3. Approximation of the curve from Fig. 2 using the Bézier curve

The disadvantage of the Bézier curve is its parametric form and the impossibility of creating a direct correspondence to determine the function by its argument. In a vector form, the Bézier curve, which is defined by four points, can be written as (10):

$$\begin{aligned}R(t) &= (1-t)^3 \cdot P_0 + \\ &+ 3 \cdot t \cdot (1-t)^2 \cdot P_1 + \\ &+ 3 \cdot t^2 \cdot (1-t) \cdot P_2 + t^3 \cdot P_3\end{aligned}, \qquad (10)$$

where $t \in [0 \ldots 1]$.

To determine the coordinates in a scalar form, let us write the equations (11) and (12), taking into account the marking of the coordinates in relation to the control law. Correspondingly, instead of the X coordinate along the abscissa axis, let us input the coordinate ω_c, and instead of the Y coordinate along the y axis, let us input the coordinate $\Delta\omega$. So, in order to determine $\Delta\omega(t)$ let us write:

$$\begin{aligned}\Delta\omega(t) &= (1-t)^3 \cdot \Delta\omega_0 + \\ &+ 3 \cdot t \cdot (1-t)^2 \cdot \Delta\omega_1 + \\ &+ 3 \cdot t^2 \cdot (1-t) \cdot \Delta\omega_2 + t^3 \cdot \Delta\omega_3\end{aligned}, \qquad (11)$$

and in order to determine $\omega_c(t)$ let us write:

$$\begin{aligned}\omega_c(t) &= (1-t)^3 \cdot \omega_{c0} + \\ &+ 3 \cdot t \cdot (1-t)^2 \cdot \omega_{c1} + \\ &+ 3 \cdot t^2 \cdot (1-t) \cdot \omega_{c2} + t^3 \cdot \omega_{c3}\end{aligned}. \qquad (12)$$

To change such parameters of the working process as ω_e and ω_c in the form of $\Delta\omega = \omega_e - \omega_c$ according to the correspondence shown in Fig. 2 the clutch control actuator must affect the clutch. So throughout the whole working process there must be some appropriate part of the torque that is transmitted with the transmission. It is proposed to determine this part as a function of clutch control, which as well as the function $\Delta\omega = f(\omega_c)$ is represented by the Bézier curve. To control the clutch, let us present the approximate Bézier curve in the coordinates $S = f(\Delta\omega)$. Since the value $\Delta\omega$ referred to in (11) is predetermined on the basis of the driver's control actions, and the value $\Delta\omega$ in a function $S = f(\Delta\omega)$ is associated with the actual process of changing angular velocitys, in order to avoid confusion in the definitions, let us denote the difference in angular velocitys that change in real time as a relative speed $\omega_r = \omega_e - \omega_c$. In this way, the clutch control function will be associated with the approximate curve of the working process, and will be denoted as $S = f(|\omega_r|)$. The relative angular velocity module ω_r provides a mirror image of the control parameter S(t) and prevents the parameter ω_r from acquiring significant negative values. For example, let us introduce the clutch control curve in the beginning of the control process in Figure 4. Let us write down the equation for determining the curve shown in Figure 4 at once in a scalar form in the corresponding coordinates. So, in order to determine $\Delta\omega(t)$ let us write (13):

$$\begin{aligned}\omega_r(t) &= (1-t)^3 \cdot \omega_{r0} + \\ &+ 3 \cdot t \cdot (1-t)^2 \cdot \omega_{r1} + \\ &+ 3 \cdot t^2 \cdot (1-t) \cdot \omega_{r2} + t^3 \cdot \omega_{r3}\end{aligned}, \qquad (13)$$

and in order to determine $S(t)$ let us write (14):

$$\begin{aligned}S(t) &= (1-t)^3 \cdot S_0 + \\ &+ 3 \cdot t \cdot (1-t)^2 \cdot S_1 + \\ &+ 3 \cdot t^2 \cdot (1-t) \cdot S_2 + t^3 \cdot S_3\end{aligned}. \qquad (14)$$

In the formulas (13) and (14) and in fig. 4 the coordinates of the point $S\omega_{r0}$ are (0, 0), the coordinates of the point $S\omega_{r1}$ are (60, 0), the coordinates of the point $S\omega_{r2}$ are (480, 0), and the coordinates of the point $S\omega_{r3}$ are (600, 1).

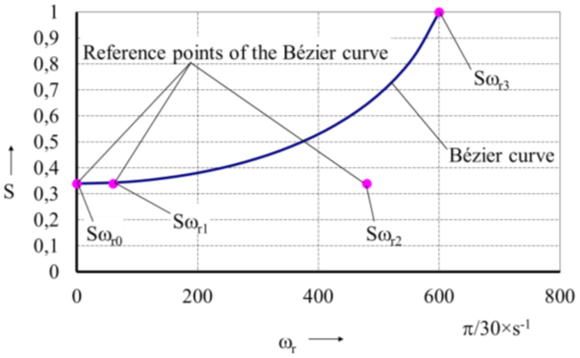

Fig. 4. The form of correspondence $S = f(\omega_r)$ for $\omega_c = 0$

Moreover, the values of the ordinates of the points $S\omega_{r0}$, $S\omega_{r1}$ and $S\omega_{r2}$ is always equal to zero, the value of the ordinate of the point $S\omega_{r3}$ is always equal to one and corresponds to the entirely left clutch pedal (the entirely engaged clutch). The value of the abscissa of the points $S\omega_{r1}$ and $S\omega_{r2}$ is calculated by the correspondences (15) and (16):

$$\omega_{r1} = k_{s\omega1} \cdot \omega_{r3}, \qquad (15)$$

$$\omega_{r2} = k_{s\omega2} \cdot \omega_{r3}, \qquad (16)$$

where $k_{s\omega1}$ and $k_{s\omega2}$ – the coefficients.

In addition, the ordinate of the point $S\omega_{r3}$ is obtained from the equation (11).

Correspondingly:

$$\omega_{r3} = \Delta\omega(\omega_c). \qquad (17)$$

To calculate the value of the function $\Delta\omega = f(\omega_c)$ which is used in the formula (17) and the function $S = f(\omega_r)$, the algorithm for finding the parameter t is used. It corresponds to the value of the input variable ω_c or ω_r from the application of the cycle. The required initial value is calculated according to the determined value t, by the formulas (11) and (13), respectively.

Considering the formulas (11)–(17) wholistically we obtain the form of the control law in three-dimensional representation (Fig. 5).

In the time of starting, the working process, which characterizes the change of its three parameters at once, can be reproduced in the coordinates $S - \omega_r - \omega_c$ in which the graph is plotted in the form of a spatial line in Figure 5.

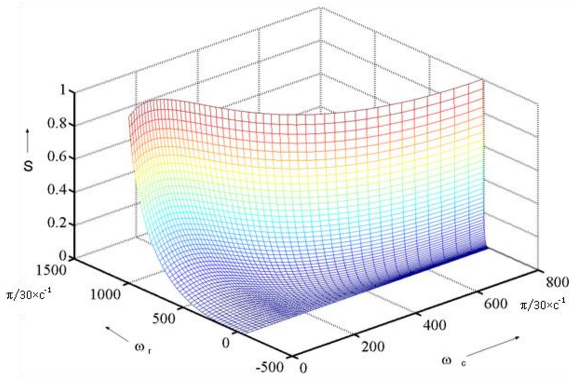

Fig. 5. The form of a typical control law in three-dimensional representation

Each point of such a line reproduces the desired control effect in accordance with the real values ω_r and ω_c. Fig. 6 shows an example of several lines that reproduce the course of the working process, during the starting of the vehicle at different positions of the accelerator pedal.

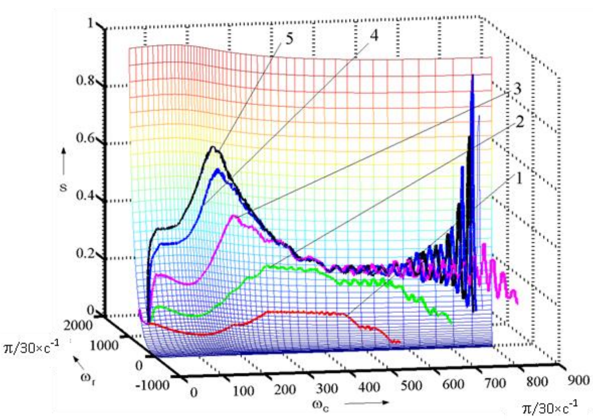

Fig. 6. The law of clutch control with the plotted lines of the working process of starting of a vehicle: 1 – accelerator pedal position of 0 %; 2 – accelerator pedal position of 10 %; 3 – accelerator pedal position of 20 %; 4 – accelerator pedal position of 30 %; 5 – accelerator pedal position of 50 %

As can be seen from the figures, all points of the course line of the working process belong to the surface of the control law. Therefore, each ratio of the parameters ω_r and ω_c corresponds to the appropriate degree of clutch engagement. The degree of increase of the function S, during the working process depends on the curvature of the Bézier curve, which, in turn, depends on the coefficients that determine the direction of tangents at the endpoints of the curve ($k_{s\omega1}$ and $k_{s\omega2}$).

The control of the shape of the surface of the control law is carried out by defining several characteristic points. There are the maximum point P_1 for which both coordinates are the directive ones, the point P_2 for which only the coordinate (ω_{c2}) is the directive one, the point P_3 for which only the coordinate (ω_{c3}) is the directive one, and the point $S\omega_{r0}$ for which only the coordinate (ω_{r0}) is the directive one. The last coordinate changes only at the stage of final full engagement of the clutch.

Other points and coordinates do not change because they have certain correspondences. For example, the point P_0 with the coordinates $(\Delta\omega_0, \omega_{c0})$ is responsible for engaging the clutch during the starting without affecting the accelerator pedal. It is important that its abscissa is always equal to zero $\omega_{c0} = 0$, and its ordinate $\Delta\omega_0$ must be such that the clutch control function $S(t)$ provides the angular velocity of the engine crankshaft within a stable section of the control branch of the engine speed characteristic.

To meet this condition the appropriate values $\Delta\omega_0$ and the coefficients $k_{s\omega 1}$ and $k_{s\omega 2}$ are required. The coordinates $\Delta\omega_2$ and $\Delta\omega_3$ of the points P_2 and P_3 accordingly will also be equal to zero because the second point ensures a guaranteed end of the process of the clutch towing with zero difference $\omega_r = \omega_e - \omega_c = 0$, and the first point guarantees the determinative tangent curve of the control law to the abscissa axis at the point P_3. Let us give the equation of connection for each characteristic point with its defining coordinates in accordance with their functional purposes. For example, the coordinate $\Delta\omega_1$ is related to the degree of pressure on the accelerator pedal. Therefore, the functional connection $\Delta\omega_1 = f(\alpha)$ will look like this:

$$\Delta\omega_1 = k_{\Delta\omega 1} \cdot \alpha_e + b_{\Delta\omega 1}, \quad (18)$$

where $k_{\Delta\omega 1}$ and $b_{\Delta\omega 1}$ are the coefficients of the correspondence $\Delta\omega_1 = f(\alpha_e)$.

According to the functional purpose of the characteristic points of the control law: the coordinate $\Delta\omega_1$ of the point P_1 determines the possible level of increase in the angular velocity of the engine crankshaft during the starting; the coordinate ω_{c1} of the point P_1 together with the coordinate ω_{c3} determines the beginning and duration of the engine load during the starting, and is characterized by a decrease in the angular velocity of the crankshaft. It is important that the reduction of the angular velocity of the crankshaft of the engine is not critical in terms of its stop. The degree of angular velocity decline must be coordinated with the entry of the angular velocity into the operating range of the angular velocitys of the engine; the coordinate ω_{c2} of the point P_2 determines the smoothness of the approach of the control law curve to the point of tangency P_3. The coordinate ω_{c3} of the point P_3 together with the coordinate ω_{c1} determines the duration of the towing process and the level of acceleration that is achieved in the process of towing the clutch; the coordinate ω_{r0} of the point $S\omega_{r0}$ provides the full clutch engagement after alignment of the angular velocitys of the crankshaft and the driven clutch parts, so it changes only at the stage of the final full engagement of the clutch. The coordinate ω_{c3} of the point P_3 is related to the increase in acceleration which is achieved during the towing of the clutch. The functional correspondence $\omega_{c3} = f(j_{\alpha\%})$ is indirectly related to the degree of pressure on the accelerator pedal, so it can be written in the form of $\omega_{c3} = f(\alpha_e)$ and calculated by the formula (19):

$$\omega_{c3} = k_{\omega_{c3}} \cdot \alpha_e + b_{\omega_{c3}}, \quad (19)$$

where $k_{\omega_{c3}}$ and $b_{\omega_{c3}}$ are the coefficients of the correspondence $\omega_{c3} = f(\alpha)$.

After the relative angular velocity ω_r becomes negative, the coordinate ω_{r0} ceases to be zero and shifts to the negative area. Together with the mirror nature of the function $S(t)$, this leads to an overall increase in the control parameter $S(t)$ and the engagement of the clutch (Fig. 7).

The increase in the negative value of ω_{r0} occurs according to the following algorithm (Fig. 8).

The initial value of ω_{r0} is zero. Then, according to the algorithm, the condition $\omega_r < 0$ is checked. Under this condition it becomes clear that the synchronization has taken place and the closing of the clutch discs is possible.

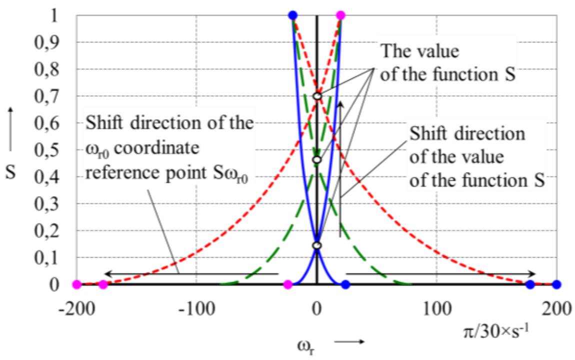

Fig. 7. The behavior of the function $S(t)$ when the coordinate ω_{r0} is shifted

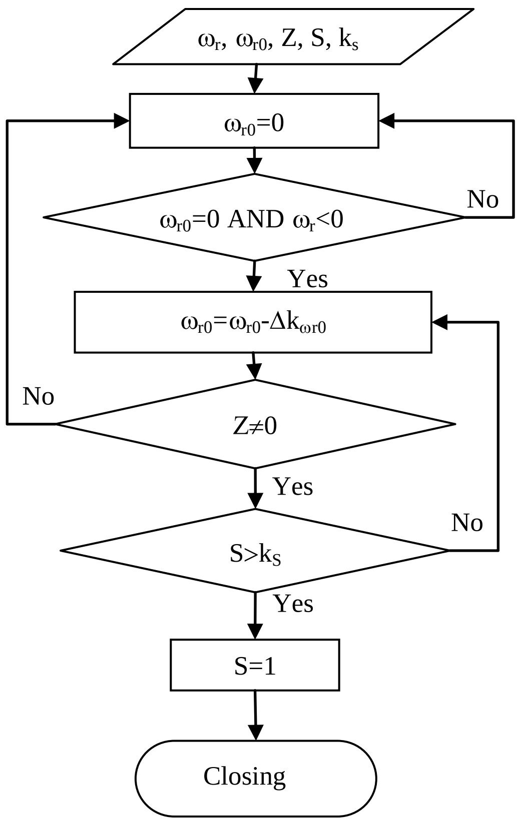

Fig. 8. Flow chart of the algorithm of ensuring of the full engagement of the clutch

When the conditions $\omega_{r0} = 0 \wedge \omega_r < 0$ are met, the cyclic subtraction of the constant subtractor $k_{\omega r0}$ from ω_{r0} begins. The constant subtractor $k_{\omega r0}$ together with the setting of the timer of the microcontroller ensures the clutch engagement within 2 sec. In the middle of this cycle, there is a control of the complex clutch disengagement signal Z which can be used to exit the cycle and set the initial value ω_{r0} in case of abrupt release of the accelerator pedal or activation of any brake. The second exit from the cycle is provided on condition that the control signal $S(t)$ reaches the appropriate level k_S, after which the command to engage the clutch fully is given, and the algorithm ends.

This exit from the cycle is due to the fact that during the real working process, the synchronization of angular velocities can occur at different levels of $S(t)$, which can lead to a faster activation of the clutch than within 2 sec. Therefore, the clutch can be turned on and the algorithm will end without waiting for the specified time. Accordingly, the thrust bearing will be shorter under the load.

Conclusions

The presented concept provides a flexible tool for creating a clutch control law with simple means to adapt to the driving conditions of the vehicle.

The proposed algorithm provides the full clutch engagement only after the full synchronization of the clutch discs.

The formation of a special form of the law in the form of a curve which is tangent to the abscissa axis reduces the jerks during closing the clutch discs.

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Концепція закону керування зчепленням

Анотація. Проблема. *Поєднання комфортного автоматичного керування трансмісією автомобіля й одночасно збереження високих показників енергоефективності та вартості конструкції можливе завдяки використанню автоматичної механічної трансмісії. Автоматична система керування зчепленням здійснює функцію забезпечення комфорту в таких трансмісіях. Закони контролю над ним на сьогодні не є досконалими.* ***Мета.*** *Метою роботи є створення чіткої концепції закону керування зчепленням, яку можна впровадити в мікроконтролер і пристосувати для адаптації до різноманітних умов руху.* ***Методологія.*** *Графічно концепція перспективного закону керування зчепленням формується двома кривими Безьє. Одна них здійснює функцію напрямляльної, а інша утворює поверхню закону.* ***Результати.*** *На основі кривих Безьє третього ступеня формується концепція закону та обґрунтовується зв'язок опорних точок кривих Безьє з фізичними параметрами робочого процесу руху автомобіля.* ***Оригінальність.*** *Формування кривої Безьє, що є визначальним для концепції закону, формується на основі типового робочого процесу синхронізації кутової швидкості дисків зчеплення під час руху транспортного засобу. Згаданий робочий процес наведено в координатах, які не мають часу його перебігу. На відміну від законів керування зчепленням, розглянутих у науковій літературі, запропонована концепція передбачає керування зчепленням поза ділянкою процесу синхронізації та забезпечує уникнення ривків під час подальшого прискорення.* ***Практичне значення.*** *Запропонований алгоритм забезпечує повне вмикання зчеплення лише після повної синхронізації дисків зчеплення. Формування особливої форми закону у вигляді кривої, дотичної до осі абсцис, зменшує ривки під час замикання дисків зчеплення.*
Ключові слова: *закон керування зчепленням, крива Безьє, ефект ривка.*

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