ECONTECHMOD. AN INTERNATIONAL QUARTERLY JOURNAL – 2018, Vol. 07, No. 2, 59-69

Optimal laws of gear shift in automotive transmissions

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Received July 07.2017: accepted Marc 10.2018

Abstract. In the article the influence of the moment of beginning of shifting gears on the efficiency of acceleration of the vehicle has been evaluated - on the fuel consumption at a given level of dynamism of the car. Different programs of switching gear at different values of the length of the friction trailing have been studied. Arguments are made in favor of the fact that the gearshift processes should not be too fast and that the work of friction trailing is not decisive in view of the need to increase the energy efficiency of the car. It is emphasized that between the laws of gear shifting, which are optimal considering the fuel economy, which, having in mind the dynamism, there are no fundamental differences. On the basis of the information obtained, it is concluded that the real variety of transmissions is unreasonably excessive and even harmful, since optimality is usually monotonous.

Key word: automobile, transmission, gear shift, optimal laws, transmission excellence, control algorithms, optimality criterions, time acceleration, fuel consumption.

INTRODUCTION

The diversity of existing designs of automotive transmissions and their laws and algorithms are extremely broad [1-12]. But this, of course, is not a sign of the remarkable achievements of engineering thought and technical science, since any variety normally reduces itself to the search for anything perfect or optimal.

It is believed that the automatic gearbox was invented due to the intellectual efforts of the agents of the American concern General Motors. The first car with an automatic transmission - Oldsmobile Custom & Cruiser - descended from the conveyor in 1939. Since then, apparently, there were grounds for measuring the principles, laws, algorithms of expedient or optimal control of the structure and parameters of automotive transmission.

Typical automatic transmission with a hydrodynamic (predominantly complex) transformer (or hydraulic coupler) usually "proposes" for the choice of the driver himself the following modes of operation: P ("Park") - parking (wheeled running gears are locked, but the lock is done internally in gearboxes and not connected with the parking brake); N ("Neutral") - an idle run; R ("Reverse") – back run; D ("Drive") - movement (forward, of course; all forward programs can be used, all except for upgrades); L ("Low") - lower gear, slow down (in case of difficult road conditions) . The choice among these modes is exercised by the driver with the lever-selector of the mode ranges. But the selection of transmission and the actual process of switching gears from one to another carries out a system-automatic machine (in our time - with the participation of the on-board computer). The system also imposes restrictions on passive safety: for example, it will not be possible to start the engine when the selector lever is not in the "P" and "N" positions. It also makes it impossible for the self-propelled movement of a car while parked on a non-horizontally or locally unequal platform (it is possible to remove the key from the ignition lock only if the lever-selector is switched to the "P" position). At the same time, the "P" mode does not replace the handbrake.

But so many special modes of auto-transmissions, which the driver has to pick up independently, is rather a sign of its excessive complexity. Intectualization of electronic control systems has created the basis for the provision of automatic transmissions of special properties. In particular, there appeared so-called adaptive automatic gearboxes. The adaptability is that the on-board computer monitors the driver's way of driving the car and adjusts to it. The algorithm of computer operation at times even assumes control over the degree of wear of frictions.

But what is so useful in adaptability? If the machine is perfect, why should it be adjusted to the driver? If the driver is smarter than an automatic machine, then what is this machine for?

RECENT RESEARCHES AND PUBLICATIONS ANALYSIS

There are automatic gearshift control systems - AutoStick (Steptronic, Tiptronic) - that give the driver the option of the command itself to choose gears, but they are entirely responsible for the process of shifing. In this case, the lever-selector has additional mode settings. But the independence of the driver is sometimes illusory, because the Autostick mode is not less automatic: the computer system still "will not allow" to directly affect the unit; the driver only sends his wishes to the computer, and it analyzes their adequacy and makes decisions about switching (to move, for example, from a third transmission or to turn on the transmission, at which modes of the engine will leave the set of permissible ones will not succeed). In everything else, the transmission is like an usual one, mechanic. Of course, the driver at any moment can move the selector lever to the "D" position, refusing to act as if directly controlling the transmission.

But what for the developer to invest in the intelligence and means in automation, and then also in the means of simulating non-automatic ("manual")? From what such a miracle (rather than ignorance), there is such a subjective, unreasonable demand in the market?

It is also introduced, in paralle, several transmission control algorithms for the drivers' choice - energy-saving, sports, winter. The energy-saving mode seems to be designed to ensure the smooth movement of the car with minimal fuel consumption (which, of course, interpretation and energy-saving meters are usually questionable). The sports algorithm is configured to realize the maximum engine power and, accordingly, the maximum acceleration of the car. There is also a kickdown mode, when in the case of a sharp press to the edge of the accelerator pedal, the system switches the transmission to a lower gear - one or one through one. Reverse switching again to a higher gear can occur if the engine reaches the maximum frequency of operating cycles. The winter algorithm foresees the possibility of a smooth start grip of a car on a slippery surface (usually a start should take place on the second or third gear).

But is there any certainty that these algorithms are really needed and that the driver is capable of handling them rationally? Is this variation not controlled by the automatic system? And can not the problem of choosing an algorithm be solved on the basis of a compromise, at least partial? Or maybe the motivation of polyalgorythmicity is fictitious? It is on this that we will have to focus further.

The automatic transmission is less energy-efficient (its performance index is generally lower) compared to the mechanical transmission, but due to the optimal combination with the engine and the implementation of optimal blocking laws for the hydrodynamic transformer, as well as the laws of gear shifting in its mechanical part, it is potentially capable of still sometimes provide higher fuel economy and dynamism of the car. And yet: the property of an automatic transmission to absorb shock loads contributes to an increase in engine resource and undercarriage.

But this does not mean that the classic automatic transmission does not have an equivalent alternative. Rather, on the contrary, it outlines the possible directions for the improvement of mechanical energy transformers. So, to improve the traditional mechanical transmission becomes possible, introducing the principles of optimal combination of modes, optimal control laws, means of increasing the elasticity - all that the automatic transmission is attractive. It is interesting that in the transmission of the Mercedes AMG Speedshift MCT 7, the hydraulic transformer was replaced by a "fluid" friction clutch - this made it possible to significantly increase the efficiency of the transmission and combine it with high-speed engines.

Variator (belt, chain, torque ...) is a transmission that has an infinite number of gears (levels). It can implement any transfer ratio from its operating range and change it smoothly so that the rotating torque and speed of the output shaft will change accordingly, even if the engine shaft rotation frequency is constant, corresponding, for example, to nominal power . The variator is not capable of properly displaying its positive properties without a perfect control system. It was with the development of microelectronics that the variator attracted persistent attention. The on-board computer is instructed to coordinate the work of the engine and the transmission, achieving at each moment the optimal transfer ratio. It is also instructed to inculcate even absurd modes. For example, here and there they succeed in simulating the step-by-step gearbox in order to please the capricious driver who is poorly aware of what is good and what is not, and he insistently wants to shift the transmission on his own; in the memory of the control unit, there several transfer relations are recorded that the driver chooses (the Tiptronic principle is a marketing trick that actually only worsens the energy-efficient and dynamic transmission properties). There is also an analogue of the kickdown in the variator: the stroke on the accelerator pedal to the edge (if desired, to accelerate the car) generates a sharp increase in the torque at the output of the variator.

The traditional mechanical transmission has in its turn a friction clutch, a mechanical gearbox, in which the driver at his discretion by means of the lever chooses the transmission and makes the switching from the transmission to the transmission through the clutch and the mechanism of switching with the synchronizers (once a long time without them) . The skillful driver changes the transmission in a synchronized box for 0.5 ... 0.6 seconds. The first mechanical boxes "contained" two transmissions - one for starting the car, the second for movement. Nowadays - dozens of gears, as if a means to approach the variator. But the primitive increase in the number of gears in a mechanical transmission is a deadlocked path, because the transmission loses to a certain extent and, in a certain sense, controllability.

Of course, the first thing that naturally comes to mind - the traditional mechanical transmission to provide a machine that would itself select and switch gears. Such an operation is called automation or robotization. Accordingly - automated (automatic) or robotizated (robotic) – so will have to call such improved transmission. It is also referred to as the Manual Transmission Automatically Shifted or MTA (Automated Manual Transmission) abbreviation.

The path of robotization can be different. It is worth mentioning that among the first not always successful steps in the way of automation was the automation of only coupling. As an example, Toyota's FreeTronic (TFT) is very unreliable, it's purely mechanical, but with automatic coupling. In the mechanical transmission of the Mercedes-Benz A-Class car, for example, an automatic (electro-hydraulic) coupling device was also installed. The gearshift is placed entirely on the driver, and in order to help him, the electro-gliding machine must track the current position of the gear lever control (selector) and carry out the necessary manipulations with the clutch. The electronic control system must take into account the signals of the engine and ABS sensors and prevent the jerks when switching gears and prevent the engine from dampening. If we limit ourselves to automating only the coupler, then we will deal with a semi-automatic transmission. In the case of partial or complete automation, the coupling control pedal disappears, as in the case of an automatic gearbox - the coupling is controlled by the machine.

As an example of deeper robotization we can present the adaptive robotized gearbox 2-Tronic, developed by the

French group PSA Peugeot Citroen in conjunction with Siemens and Bosch for the (Peugeot 207) car: the transmission has ripened on the basis of a five-stage purely mechanical, which was made already two decades ago ; two electromechanical drives were attached to it, one of which makes gear shifting, and the second switches on and off the coupling, as well as an on-board computer that manages these processes based on information on the values of different modes of operation, first of all - about the position of accelerator and the speed of movement of a car. Simplified, with a single clutch, robot (we mean, for example, the transmissions of cars Toyota, Opel, Alfa Romeo, Peugeot, Suzuki) makes a gear shift in 1 ... 2 seconds. It remains possible for manual gearshifting. The 2-Tronic gearbox, for example, provides the ability to use three modes: the first one - fully automated; the second is the so-called semi-mechanical, which can be used without leaving the fully automatic mode, in the case of urgent need to switch to the lower gear (in the process of overtaking, let's say, when the situation has run out and the car has returned to normal mode of movement, the gearbox after a while unwittingly restores its automatism); the third - quite manual (though, if the driver will prove the speed of the engine shaft to the maximum allowable value, without switching the transmission to a higher gear, then it will still make it for him).

Recognized for high-quality robotics, for example, the six-speed manual transmission of the BMW M-series, which is called SMG-Sequential M Gear-box (sequential transmission from the transmission to the transmission, from Latin *sequentia* – following up). The switching of the coupler and the gearshift is placed on an electronically controlled hydraulic system. The speed of switching of the gears is very high, during acceleration of the car it takes 0.08 seconds.

The computer-aided intellectualization of the automaton-robot reveals the way to a higher level of perfection of the robotizated transmission in comparison with the parametric same traditional mechanical transmission. The high-quality electronic control of actuators allows even the refusal of so-called synchronizers. Due to this AMT becomes in a linear dimension (in length) more compact and also capable of sending big rotary moments, it will need less oil, and even the weight will decrease. Robotized gearboxes may have either an electric or hydraulic coupling and a gear changeover mechanism. In the case of an electric drive, servodevices are the executive bodies (electric motors), and in the case of a hydraulic drive - hydraulic cylinders. It so happened that in the case of a hydraulic drive (which happens more often), the gearbox is sequential (it is a matter of sequential gear shifting in manual mode).

Generally speaking, potentially there are ways to technically improve a robotic transmission to such an extent that it can compete both in the dynamic transmission of energy and in energy-saving with any conventional mechanical and with any automatic (either traditional or variational). In this case, the usual problem of overheating of the clutch will remain in the past.

A fundamentally deeper refinement overcomes a robotic transmission, if it is to be introduced into one more coupler. The most famous among them is the transmission with the so-called DSG transmission box (Direct Shift Gearbox, sometimes - Dual Clutch Transmission), which was used on cars Volkswagen and Audi. It came to a wide world from motorsport (this is a technology from Formula 1). Transmission DS (G) was massively used on Volkswagen's Golf R32 cars in 2002/03. It is tried to be applied even on tractor technology [12].

So, one clutch should serve uneven gears, and the other - even. If, for example, the car's motion occurs on the third transmission with the closed first coupler (second off), then if necessary, switches on the pre-configured by computer machine the fourth gear (with an increase in speed) or a pre-configured - the second (in case of speed reduction), the first coupling turns off (and goes into the stand-by mode), and the second synchronously turns on. The gearshift takes fractions of a second. For example, gearboxes with two DSG, S-Tronic couplings are switched from transmission to transmission for 0.2 ... 0.4 s, SMG and DCT M Drivelogic boxes of sports cars BMW - for 0.1 s.

In this case, it can be said that the transmission contains a pre-selective gearbox: after switching on any transmission, you can pre-select the next one and activate it at the right moment without interrupting the energy flow. Transmissions can be switched without loss of power. Potentially, the DSG transmission, compared with all of the rest, can provide the car with the highest both dynamism and energy efficiency. Of course, in the DS (G) -ransmission from an odd one, let us say, to the odd one can be switched only through an intermediate even one. Instead, the automatic transmission with planetary gears rows friction clutches can provide a random switching sequence, for example, the Mercedes 7G-Tronic can jump in one step into four transmissions "down". But it is unlikely that this can be considered an advantage of an automatic transmission.

In the DCT M Drivelogic dual-clutch transmission from BMW, the Drivelogic features a control system that allows for the use of eleven gearshift programs. Six programs are implemented in manual switching mode, and five are programmed for automated gearshifting. So there is an opportunity to adapt the conditions of the change of gear according to the style of driving. But this kind of programmatic diversity and adaptability are evidence of imperfections (let's mention the adaptability and combination of energy-saving, sports, winter control algorithms).

The more powerful the engine with which the DSG transmission works, the greater the energy losses in the vicinity of the so-called kiss point, when both couplings are simultaneously intensively trailing. This is why the new seven-speed gearbox DSG with two dry clutches, developed jointly by Volkswagen and the Luk companies, should not send a torque greater than 250 Nm. Instead, a longer six-speed DSG gearbox with two fluid clutches can work with a larger engine (clutches in a fluid crankcase, frictions). But the energy loss during the switchover is inevitable, they accompany switching in any other transmission.

It is believed that in order to completely eliminate a person from the control circuit AMT should algorithmically cooperate with some hundreds of sensors that would have to supply an automatic system with an adequate amount of information of the proper quality. Such a large number of constituents, of course, negatively affects the potential reliability of the control system. Inadequate work of one sensor causes distortion of control algorithms: energy saving, dynamic and any other algorithm cease to be such, but rather transform into non-working, emergency... Although there are quite effective computer tools for a detailed study of the energy performance of a car and an engine [13, 14], it is also possible to assume that the optimality of the laws of switching gear is substantially simpler to rely on such a large amount of instrumental and computer information [15-18].

OBJECTIVES

The purpose of the research: to find out how much the existing diversity of the laws of mechanical gearbox transmissions control is motivated; determine how deeply different criteria of optimality of the laws of switching the steps (levels) contradict each other; to assess whether there is a sense in parallel to foresee several transmission control algorithms.

PRESENTATION OF THE MAIN MATERIAL

Task and scientific hypothesis. The attitude to different transmissions can be substantiated if it is possible to answer impartially such questions.

1. What should be the ideal gear shifting time? Is transiency really relevant to the gear shift (remember, in the Dynamic mode, some transmissions change the transmission for 0.08 s and this is considered to be a huge success)?

2. What parameters determine the expediency of using one or another transmission at one time or another? Should the number of sensors, that supply the information to the micro-roprocessor control system of the transmission, be almost one hundred?

3. What distinguishes the special energy-saving and dynamic (sports) transmission laws? Are they essential to each other and must necessarily be programmed in a perfect step-up transmission?

4. Is the kickdown on the driver's share of the accelerator pedal on the floor? Is this a truly special transmission control mode that would have to oblige the gearbox to accelerate the acceleration of the vehicle by switching to a lower gear or through one down.

5. Has a mechanical shift with manual switching the objective chances to hold on the market as an attractive product? At the very least, it is appropriate to say about these chances if optimal transmission control laws were easy to reproduce (at least with an acceptable accuracy) without automation.

The scientific hypothesis can be put forward in the form of a statement that for all of the listed questions there are reasons to give a warning or a completely negative answer.

About Optimal Laws Of Gear Shift. For the implementation of the procedure for the synthesis of optimal gearshift laws, the known information is, as is known, the parameters of the gearbox and fuel characteristics of the engine $Q_t = Q_t(M_e, \omega_e)$ [19-22], where Q_t - the speed of fuel consumption, M_e - the torque, ω_e - the speed of rotation of the engine shaft; $M_e \omega_e = N_e$ - engine power performance. The special features of the process of synthesis of optimal transmission laws without interruption of the power flow should be monitored on an example of a simple mechanical gearbox with friction controls in case of realization by a given car, let us notice again - a simple program of motion of a vehicle.

Let us reproduce in a certain interval $[t_A, t_B]$ of time t a fragment AB of the program $v = V(t)$ of motion of a car with a constant acceleration (fig. 1: v - speed). The assumption of the linearity of the program $v = V(t)$ is quite acceptable [19, 20], since the process of gear shifting is short and hence the vehicle's motion is considered during a very small time interval. This fragment of the program of motion of the vehicle can be fully or partly implemented, forcing the engine to work on the sets of modes represented by some curves of the dependence $N_e = N_{ek}(\omega_e)$ or $N_e = N_{e(k+1)}(\omega_e)$ of power N_e of the engine on the speed ω_e of rotation of its shaft on certain *k*-th and *(k+1)*-th gears (Fig. 2). The lines $N_e = N_e^+(\omega_e)$, $N_e = N_e^-(\omega_e)$, $\omega_e = \omega_e^+ = \text{const}$, $\omega_e = \omega_e^- = \text{const}$ reflect in the coordinate system $\omega_e O N_e$ of the so-called external modes of operation of the engine and encircle a set of possible modes of operation of the engine. As a matter of fact, let this set of external include the modes and operation $R_k^{t_A}$ and $R_{k+1}^{t_B}$ of the engine, corresponding to the *A* beginning and *B* the end of the given program of motion of the vehicle (corresponding to the momentum t_A and t_B, (see Fig. 1).

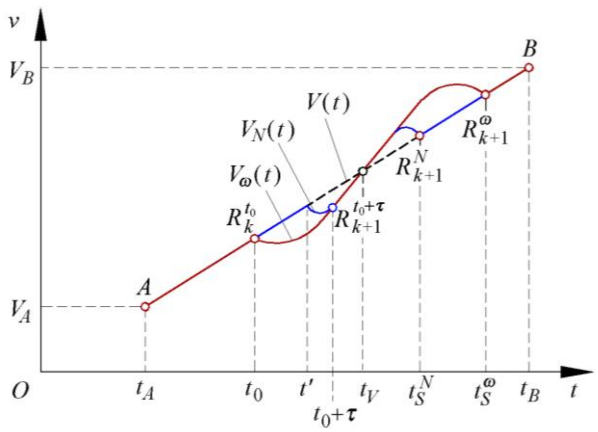


Fig. 1. Fragments of the program of movement of the car

In this case, the chosen vehicle program $v = V(t)$ can be implemented, using necessarily two transmissions (two levels) of the transmission; the gearshift from *k*-th to *(k+1)*-th should occur when the engine reaches on *k*-th gear some mode $R_k^{t_0}$ on the line $N_{ek}(\omega_e)$, see Fig. 2.

Of course, the driver chooses program, guided by the conditions and circumstances of the traffic. And, of course, the machine must not adjust the driver's choice. But the deviation $V_* = V_\omega(t)$ or $V_* = V_N(t)$ from the given program $V(t)$ of motion (see Figure 1) is possible, however, due to different technical constraints. But it is possible to put forward a logical requirement that for a moment $t_* = t_S^*$ ($t_S^* = t_S^\omega$ or $t_S^* = t_S^N$, see fig. 1) after switching gears the consequences of the deviation of the traffic program could not be found. And this means that conditions must be fulfilled at this moment

$$S = \int_{t_0}^{t_S^*} V_*(t)dt = \int_{t_0}^{t_S^*} V(t)dt$$

and

$$v(t_S^*) = V_*(t_S^*) = V(t_S^*)$$

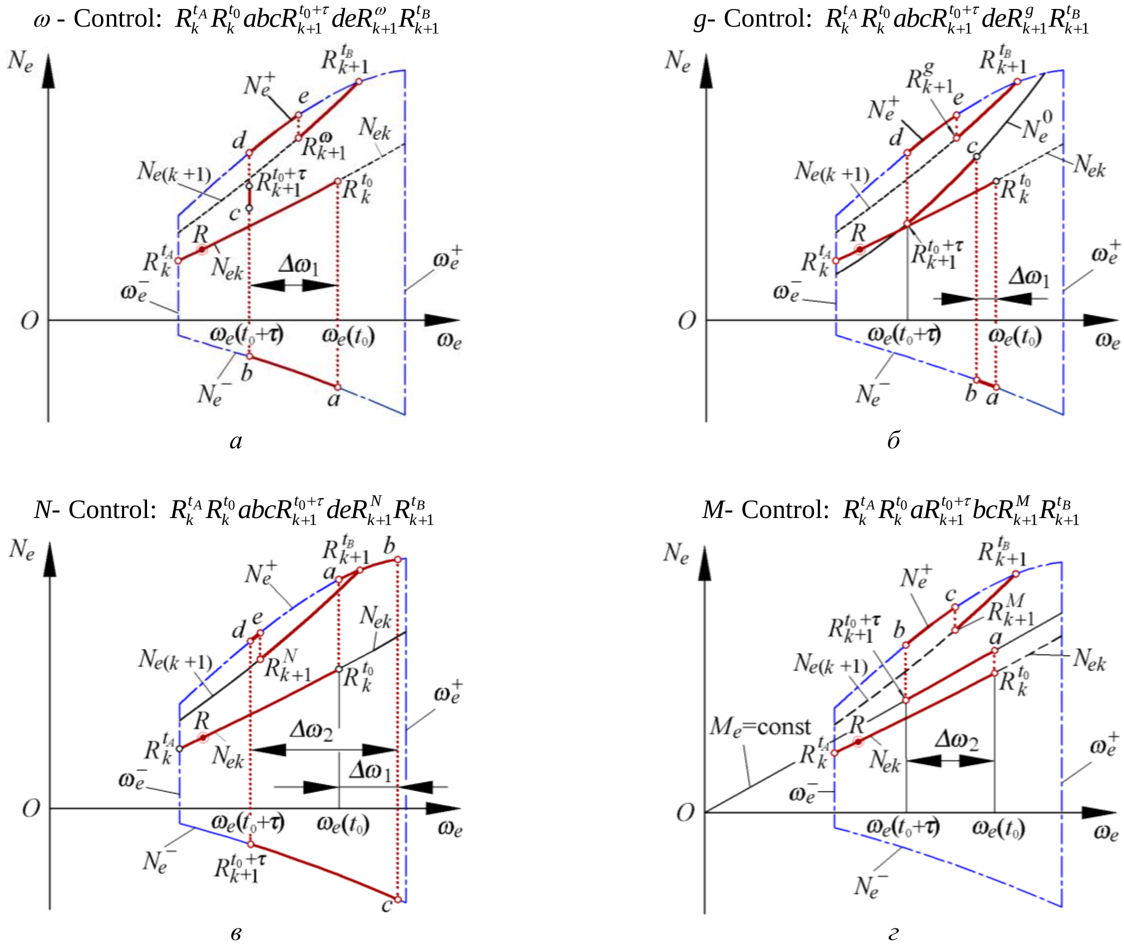


Fig. 2. Combination of operating modes of the engine and a step-by-step transmission of the car

Parameters characterizing the process of gear shifting in the step-by-step transmission of the car. Let the current mode R of the car engine (see Fig. 2), moving to the right along the curve $N_e = N_{ek}(\omega_e)$ (a certain k-th gear is turned on), will turn into a mode $R_k^{t_0}$, and at this moment should begin shifting gears to *(k+1)*-th. The upper symbol in the designation $R_k^{t_0}$ indicates precisely at the moment of the program of movement of the car, which coincides with the beginning of the gear shifting process; therefore, the symbol t_0 in the designation $R_k^{t_0}$ indicates that the moment of the beginning of the re-transmission of the transmissions from k-th to *(k+1)*-th coincides with the moment t_0, indicated on the program of motion $v = V(t)$, see. Fig. 1. Actually, at the moment t_0 frictions Φ_k and Φ_{k+1} switching devices start to operate (Fig. 3; I_e moment of inertia of the transmission masses associated with the primary shaft of the gearbox; I_a - the instantaneous moment of inertia of the transmission mass between the gearbox and the main transmission); and so that after a certain fixed time τ of full *(k+1)*-th gear transfer, the current mode of the engine coincides with the mode $R_{k+1}^{t_0+\tau}$, which belongs to $N_e = N_{e(k+1)}(\omega_e)$ curve. The lower index in the designation $R_{k+1}^{t_0+\tau}$ indicates the number of gear after switching transmission, and the upper symbol - $t = t_0 + \tau$ - the moment of switching completion; τ - duration of the gearshift process (in the sense of purely switching-trailing of frictions).

The process of changing the mode of operation of the engine due to switching over the transmission time τ with frictions Φ_k and Φ_{k+1} depends on the method of engine control. As soon as the engine's operating state reaches the switching position $R_{k+1}^{t_0+\tau}$, the implementation of the program of motion of the car will be complete so that the current mode R will move along the curve $N_{e(k+1)}(\omega_e)$, until a completely definite position $R_{k+1}^{t_B}$ which corresponds to the moment of completion of the controlled part of the program of movement of the car.

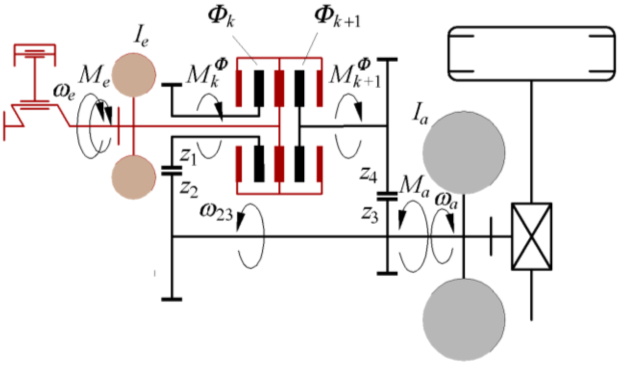


Fig. 3. The scheme of combination in the transmission of a car of two levels, one of which – the direct transmission

Of course, in the moment t_0 of the switching gear start, another arbitrary moment t_i of the reproduction process of the given program of the car's motion can be taken arbitrarily, and instead of the quantity $t_0+\tau$ corresponding to the moment of completion of the work of the frictions, it is possible to give preference to some other acceptable value $t_i+\tau_j$. Therefore, it seems quite natural the task of finding the optimal values of the parameters t_i and $t_i+\tau_j$, as well as the optimal (or at least rational) method of engine and friction control.

Work of frictions. Frictions Φ_k and Φ_{k+1} each time together must provide a transition $R_k^{t_i}$ - $R_{k+1}^{t_i+\tau_j}$ of the mode R of operation of the engine from the curve $N_{ek}(\omega_e)$ to the curve $N_{e(k+1)}(\omega_e)$ without any violation, as noted, the given by the driver of the program of motion of the car.

But even this particular task can be accomplished, embodying a variety of programs for trailing friction elements. In particular, it may be required that the gearshift be carried out either in a predetermined time interval, subject to a particular requirement, or at the shortest time (dynamic switching with limited friction resources), or with the least energy dissipation (energy saving switching), or else.

In accordance with the scheme given in Fig. 3,

$$M_e - I_e\frac{d\omega_e}{dt} = M_k^\Phi + M_{k+1}^\Phi = M_{k(k+1)}^\Phi, \qquad (1)$$

$$\frac{M_k^\Phi}{u_k} + M_{k+1}^\Phi = M_a, \qquad (2)$$

where M_k^Φ and M_{k+1}^Φ - rotating moments created by frictions Φ_k and Φ_{k+1}; $M_k^\Phi\omega_e \ge 0$, $M_{k+1}^\Phi\omega_e \ge 0$; u_k - transfer ratio (k-th gear). Since the program of the car's motion is given, then it is also known the quantity at every moment of time (see (2))

$$\frac{M_k^\Phi(t)}{u_k} + M_{k+1}^\Phi(t) \equiv M_a(t). \qquad (3)$$

Consider the algorithm for synthesizing the optimal control laws of the step-by-step mechanical transmission of a vehicle, not taking into account the energy loss on friction in toothed catching and bearing shafts. In addition, let's put into (1):

$$M_k^\Phi + M_{k+1}^\Phi = M_{k(k+1)}^\Phi(t) = at+b, \qquad (4)$$

where the coefficients a and b are determined by the values of the predefined parameters t_i, $t_i+\tau_j$ and the coordinates of the points

$$\left(M_{ek}^{t_i} - I_e\frac{d\omega_{ek}^{t_i}}{dt}, \omega_{ek}^{t_i}\right), \left(M_{e(k+1)}^{t_i+\tau_j} - I_e\frac{d\omega_{e(k+1)}^{t_i+\tau_j}}{dt}, \omega_{e(k+1)}^{t_i+\tau_j}\right)$$

respectively ($M_{ek}^{t_i}$, $\omega_{ek}^{t_i}$, - the coordinates of the point $R_k^{t_i}$; $M_{e(k+1)}^{t_i+\tau_j}$, $\omega_{e(k+1)}^{t_i+\Delta t_j}$ - the coordinates of the point $R_{k+1}^{t_i+\tau_j}$). Expression (4) peculiar reflects the program of compatible work of two frictions in the process of switching gear from k-th to ($k+1$)-th.

Thus, the relations (3) and (4) together clearly define the programs $M_k^\Phi = M_k^\Phi(t)$ and $M_{k+1}^\Phi = M_{k+1}^\Phi(t)$ of the work of frictions Φ_k and Φ_{k+1}; (through parameters a, b, u_k, time t, and function $M_a = M_a(t)$):

$$M_k^\Phi = \frac{M_a(t) - at - b}{1-u_k}u_k, \quad M_{k+1}^\Phi = \frac{at+b-M_a(t)u_k}{1-u_k}.$$

The function $M_a = M_a(t)$ is known, since it is uniquely determined through the program of the car movement (except, of course, that time interval when there are technical restrictions):

$$M_a(t) = \frac{r_\text{к} u_0}{\eta_\text{тр}}\left(G_\text{а}\psi + k_\text{п}Fv^2(t) + m_\text{а}\delta\frac{dv(t)}{dt}\right),$$

where u_0 - transfer ratio of the main transmission; $\eta_\text{тр}$ - coefficient of efficiency of the transmission; $r_\text{к}$ - wheel radius (we believe that the radius of rolling and the dynamic radius are one and the same concept); $G_\text{а}$ - weight of the car; ψ - the total coefficient of resistance of the road; $k_\text{п}$ - coefficient of air resistance (aerodynamic resistance); F - area of wind resistance (sail area); $m_\text{а}$ - mass of the car; δ - coefficient of account of the inertia of the rotating masses of the car, which is determined by the formula

$$\delta = 1 + \frac{I_a u_0^2 \eta_\text{тр}}{m_\text{а} r_\text{к}^2} + \Sigma I_\text{к}\frac{1}{m_\text{а} r_\text{к}^2},$$

where $\Sigma I_\text{к}$ - the total moment of inertia of the wheels.

It should be noted that compliance with a given program of motion throughout the switching time is possible only under exceptional conditions. The speed of the sliding (trailing) of frictions and (fig. 3) is determined by the formulas respectively

$$\omega_{sk} = \omega_e - \omega_1 = \omega_e - \frac{z_4}{z_3}\frac{z_2}{z_1}\omega_4 = \omega_e - \frac{z_4}{z_3}\frac{z_2}{z_1}\omega_a$$

$$\omega_{s(k+1)} = \omega_e - \omega_4 = \omega_e - \omega_a,$$

where ω_1, ω_4 - the speed of rotation of gears with the number of teeth; z_1, z_4; z_2 and z_3 - the number of teeth on the gears of the intermediate shaft (which speed is - ω_{23}).

Let us assume that $M_a > 0$ (that is, the deceleration of the car by the engine does not matter). Therefore, the conditions must also be met

$$\omega_e > 0,\ \omega_a > 0,\ \omega_{s(k+1)} = \omega_e - \omega_a \geq 0 \qquad (5)$$

(in case $\omega_{s(k+1)} = \omega_e - \omega_a < 0$ the condition $\omega_{sk} = \omega_e - \omega_a / u_k < 0$ ($u_k < 1$), would be true, which would mean braking by the engine). Quantity

$$\omega_{sk} = \omega_e - \omega_a / u_k \ (u_k < 1)$$

in general, it can be both positive and negative. Therefore, the transmission of torque in both frictions is possible when

$$M_k^{\Phi}\omega_{sk} = M_k^{\Phi}\left(\omega_e - \frac{\omega_a}{u_k}\right) \geq 0,$$

$$M_{k+1}^{\Phi}\omega_{s(k+1)} = M_{k+1}^{\Phi}(\omega_e - \omega_a) \geq 0,$$

preventing the ability to receive energy from the outside (frictions can only dissipate energy).

Ways to control the engine. In the process of trailing frictions, the engine operation mode R can move in the coordinate system $\omega_e O N_e$ along the lines $R_k^{t_A}$ $R_k^{t_0}$ a b c $R_{k+1}^{t_0+\tau}$ d e R_{k+1}^{ω} $R_{k+1}^{t_B}$ (Fig. 2, a); $R_k^{t_A}$ $R_k^{t_0}$ a - b c $R_{k+1}^{t_0+\tau}$ d e R_{k+1}^{g} $R_{k+1}^{t_B}$ - (Fig. 2, b); or lines $R_k^{t_A}$ $R_k^{t_0}$ a - b c $R_{k+1}^{t_0+\tau}$ d e R_{k+1}^{N} $R_{k+1}^{t_B}$ - (Fig. 2, c). In this case, in first two modes of control, the engine is "obliged" to implement at the the beginning of the work of frictions (at any moment t_0), the forced modes (areas *ab*), then unforsed (areas c $R_{k+1}^{t_0+\tau}$), and from some moment $t_0+\tau$ again forced (areas *de*). For the third mode, forced modes are implemented throughout the entire control. However, the implementation of the final stage of control "requires" from the engine to provide: in the case of the first mode of control - the constant angular velocity (vertical section c $R_{k+1}^{t_0+\tau}$); in the case of the se cond - the section c $R_{k+1}^{t_0+\tau}$ belongs to the line of minimum specific fuel consumption $N_e = N_e^0(\omega_e)$. In these three control options, the total duration of friction work will be considered the same - τ . Therefore, at the point of the mode $R_{k+1}^{t_0+\tau}$ the trailing of frictions stops and switching gears can be considered conditionally not completed. The first way to control the engine, let us call it ω -control (by the sign $\omega_e = \text{const}$), the second - g -control (on the basis of compliance N_e^0 modes with the minimum value of specific fuel consumption $g_e = Q_t / (M_e \omega_e)$), and the third - the N -control (or dynamic control - on the basis of the full independence of the external modes of the engine $N_e = N_e^+(\omega_e)$ and $N_e = N_e^-(\omega_e)$). In fig. 2, d shows another version of the engine control - $R_k^{t_A}$ $R_k^{t_0}$ a $R_{k+1}^{t_0+\tau}$ - b c R_{k+1}^{M} $R_{k+1}^{t_B}$. In this case, the engine immediately, from the moment of the start of the friction "is obliged" to implement a constant torque. This way of controlling the engine t_0 and τ for these parameters does not involve external modes, and therefore does not belong to the forced ones. We will call it (by the sign $M_e = \text{const}$) *M*-control.

Note, that the duration $\Delta\tau_1$ (see Fig. 5) of the passage of the sections *ab* (see Figures 2, a and b) and the section $cR_{k+1}^{t_0+\tau}$ (see Figure 2, c) are small compared with the duration τ of the whole process of frictions' work. The interval of time $\Delta\tau_1$ corresponds to the areas of falling of angular speed of the engine $\Delta\omega_1$ (see Fig. 2, a and b).

Fig. 5 shows that for ω -; g - and M - control throughout the switching time for frictions units Φ_k and Φ_{k+1}, accordingly, the conditions are fulfilled: $\omega_{sk} = \omega_e - \omega_a / u_k < 0$, and $\omega_{s(k+1)} = \omega_e - \omega_a > 0$. In particular, when ω -control, the program $v = V(t)$ of the car's motion in the time interval from t_0 to $t_0 + \tau$ changes (see Fig. 1 - the curve $V_{\omega}(t)$). Having in mind the strategy of inviolability of the chosen program of movement, it is necessary to take measures to "return" to the curve $V(t)$. To do this, after completing the friction trailing from the moment $t_0+\tau$ ($R_{k+1}^{t_0+\tau}$), the engine operation mode must instantly go to the line $N_e = N_e^+(\omega_e)$ and belong to it for a moment t_S^{ω}, when the effects of such control will be eliminated (point R_{k+1}^{ω}).

Unlike the ω -control, during *N*-control there is a slight deviation from the program $V(t)$ (curve $V_N(t)$ in Fig. 1). This is explained by the fact that at the beginning of the switching the friction teailing speed Φ_k is initially positive (to a moment t'), but only subsequently negative (see Fig. 5, c). Therefore, the start of the offset from the program $V(t)$ will occur later, and the duration of stay on line modes $N_e = N_e^+(\omega_e)$ is completed significantly earlier (point R_{k+1}^{N}). That is, for points R_{k+1}^{N} and R_{k+1}^{ω}, respectively, we have $t_S^N < t_S^{\omega}$. Consequently, the use of dynamic controls does not perceptibly distort the motion program, unlike others.

ω - Control

g- Control

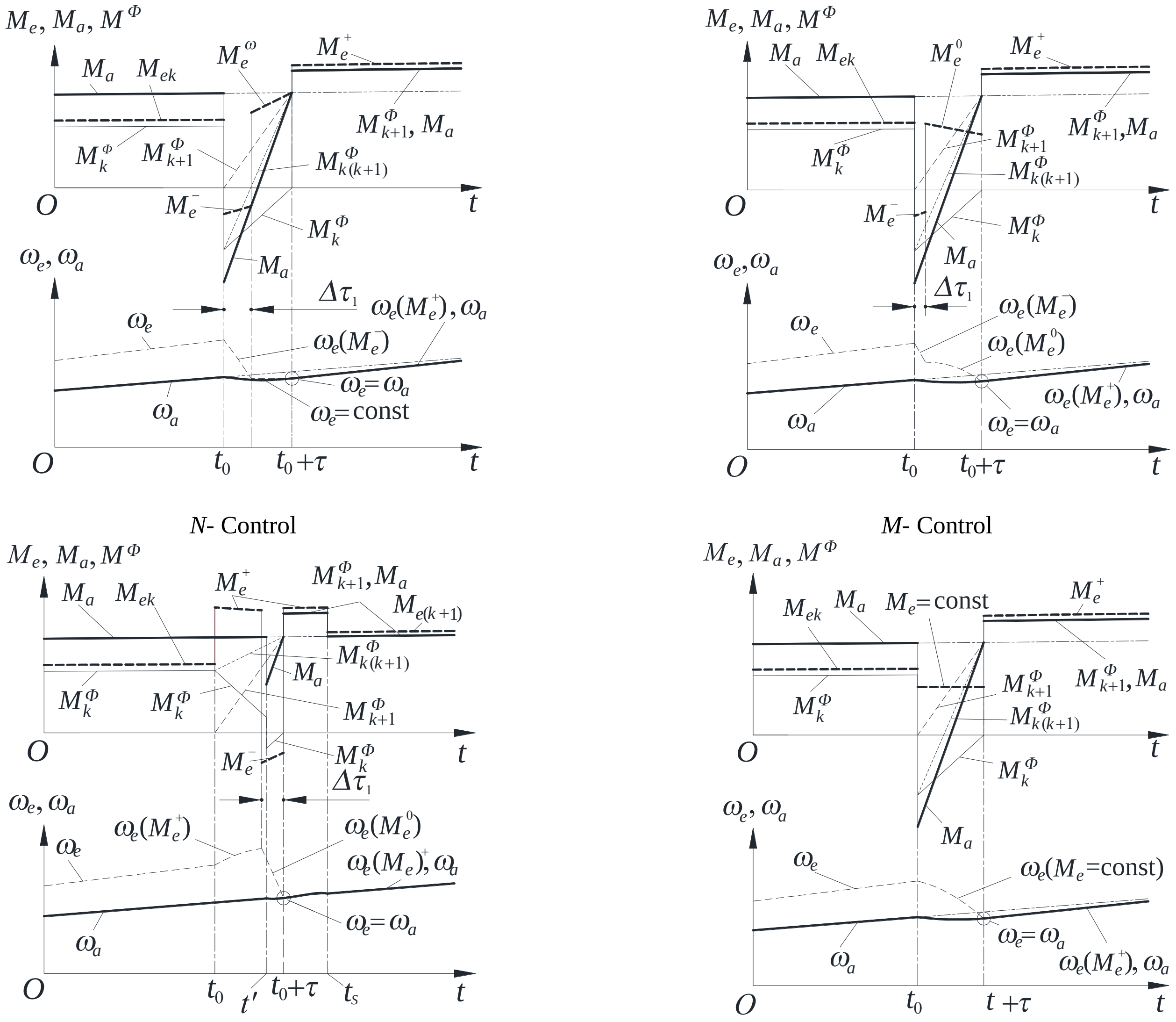


Fig. 5. Fragments of programs for changing the modes of operation of the engine and frictions

Efficiency of gear switching laws. To substantiate the expediency of choosing one or another method of engine control in the process of gear shifting in the case of set moments of its start $t_i = t_0$ and the duration of work of frictions $\tau_j = \tau$, becomes possible if guided by fuel consumption $Q_{AB}(t_i, \tau_j)$ during the process of reproduction of the given fragment of the program $v = V(t)$ of motion of the vehicle. To determine fuel consumption, we use the following formulas:

$$Q_{AB}^{\omega}(t_0,\tau) = \int_{t_A}^{t_0} Q_t(M_{ek}(t), \omega_{ek}(t))dt + \int_{t_0}^{t_0+\Delta\tau_1} Q_t(M_e^-(\omega_e(t)), \omega_e(t))dt +$$

$$+ \int_{t_0+\Delta\tau_1}^{t_0+\tau} Q_t(M_{k(k+1)}^{\Phi}(t), \omega_{k+1}^{t_0+\tau} = \text{const})dt + \int_{t_0+\tau}^{t_S} Q_t(M_{e(k+1)}^{+}(t), \omega_{e(k+1)}(t))dt + \int_{t_S}^{t_B} Q_t(M_{e(k+1)}(t), \omega_{e(k+1)}(t))dt\,, \qquad (6)$$

$$Q_{AB}^{g}(t_0,\tau) = \int_{t_A}^{t_0} Q_t(M_{ek}(t), \omega_{ek}(t))dt + \int_{t_0}^{t_0+\Delta\tau_1} Q_t(M_e^-(\omega_e(t)), \omega_e(t))dt +$$

$$+ \int_{t_0+\Delta\tau_1}^{t_0+\tau} Q_t(M_{ek(k+1)}^{g}(t), \omega_e(t))dt + \int_{t_0+\tau}^{t_S} Q_t(M_{e(k+1)}^{+}(t), \omega_{e(k+1)}(t))dt + \int_{t_S}^{t_B} Q_t(M_{e(k+1)}(t), \omega_{e(k+1)}(t))dt\,, \qquad (7)$$

$$Q_{AB}^{N}(t_0,\tau) = \int_{t_A}^{t_0} Q_t(M_{ek}(t), \omega_{ek}(t))dt + \int_{t_0}^{t_0+\Delta\tau_1} Q_t(M_e^+(\omega_e(t)), \omega_e(t))dt +$$

$$+ \int_{t_0+\Delta\tau_1}^{t_0+\tau} Q_t(M_e^-(\omega_e(t)), \omega_e(t))dt + \int_{t_0+\tau}^{t_S} Q_t(M_{e(k+1)}^{+}(t), \omega_{e(k+1)}(t))dt + \int_{t_S}^{t_B} Q_t(M_{e(k+1)}(t), \omega_{e(k+1)}(t))dt\,, \qquad (8)$$

$$Q_{AB}^{M}(t_0,\tau)=\int_{t_A}^{t_0} Q_t(M_{ek}(t),\omega_{ek}(t))dt+$$

$$+\int_{t_0}^{t_0+\tau} Q_t(M_{ek(k+1)}(t)=\mathrm{const},\omega_e(t))dt+\int_{t_0+\tau}^{t_S} Q_t(M^{+}_{e(k+1)}(t),\omega_{e(k+1)}(t))dt+\int_{t_S}^{t_B} Q_t(M_{e(k+1)}(t),\omega_{e(k+1)}(t))dt\,. \qquad (11)$$

The formula (6) is valid in the case of ω-control of the thermal engine (Fig. 4a), in the case of *g*-control, the fuel consumption of the engine is calculated by the formula (7) similar to (6), but with the other limits of integration (the value of the quantity $\Delta\tau_1$ in Fig. 4, a and b are generally not the same), calculation of fuel consumption in the case of *N* - control is carried out by the formula (8). When the fuel supply to the cylinders is stopped at the brake operating modes of the engine, in the expressions (6) and (7) the second conjunctions should be zero:

$$\int_{t_0}^{t_0+\Delta\tau_1} Q_t(M_e^{-}(\omega_e(t)),\omega_e(t))dt=0\,; \qquad (9)$$

and in the expression (8) the third conjunction is zero:

$$\int_{t_0+\Delta\tau_1}^{t_0+\tau} Q_t(M_e^{-}(\omega_e(t)),\omega_e(t))dt=0\,. \qquad (10)$$

The relations (9) and (10) are practically always valid when it comes to a diesel engine. In the case of *M*-control, fuel consumption is determined by the formula (11), which consists of four conjunctions (Fig. 4, d).

Universal fuel characteristic of a car. Choosing one of the ways to control the engine - ω -, *g*-, *N*- or *M*-control - and using equation (6), (7), (8) or (11) for any value t_i of the moment t of the start and any value τ_j of the duration τ of the friction trailing we can calculate the corresponding values of the absolute fuel consumption in the case of the implementation of the fragment *AB* of the car's movement program $v=V(t)$. On the basis of these calculations in the coordinate system tOQ_{AB} , it is possible to construct a so-called fuel characteristic of a compatible engine operation, gear box (transmission) and frictions (fig. 6).

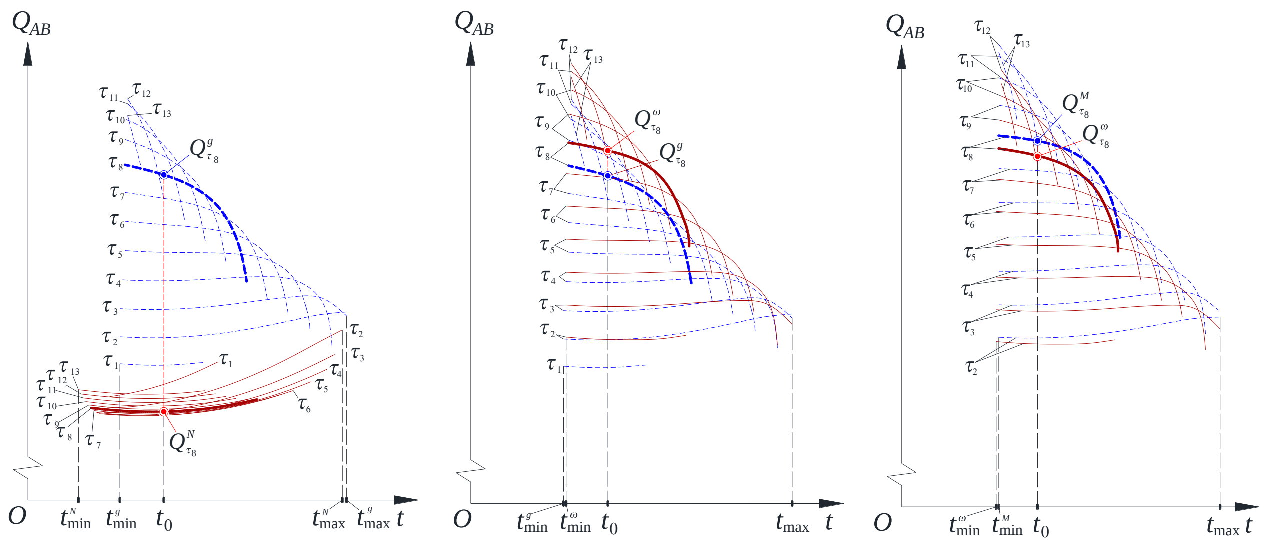
The fuel characteristic for a given pair of gears is represented by the pairs of families of curves $Q_{AB}^{N}(t_i,\tau_j=\mathrm{const})$ and $Q_{AB}^{g}(t_i,\tau_j=\mathrm{const})$ - in Fig. 6, *a*; $Q_{AB}^{g}(t_i,\tau_j=\mathrm{const})$ and $Q_{AB}^{\omega}(t_i,\tau_j=\mathrm{const})$ - in Fig. 6, *b*; and $Q_{AB}^{\omega}(t_i,\tau_j=\mathrm{const})$ and $Q_{AB}^{M}(t_i,\tau_j=\mathrm{const})$ - in Fig. 6, *c* (upper indices N , g , ω , M respectively mark the programs N -, g -, ω ,- M -control of the engine; $\tau_i<\tau_{i+1}$). These series of curves belong to completely different types of surfaces, which identify fuel consumption in the process of different ways of engine control (by parameters t_i,τ_j , Q_{AB} there are specific numbers, which are not reflected in the figures for their simplification).

Regarding the selected modes (programs, algorithms) for controlling the thermal engine at some given t_0 for many values of the duration τ of the friction trailing (for example, for $\tau=\tau_8$, see Fig. 6) inequality is realized $Q_{\tau 8}^{N}<Q_{\tau 8}^{g}<Q_{\tau 8}^{\omega}<Q_{\tau 8}^{M}$. In particular, it's easy to figure out that in the case $\tau=\tau_8$

$$\frac{Q_{\tau 8}^{g}}{Q_{\tau 8}^{N}}=2{,}47\,,\quad \frac{Q_{\tau 8}^{\omega}}{Q_{\tau 8}^{N}}=2{,}63\,,\quad \frac{Q_{\tau 8}^{M}}{Q_{\tau 8}^{N}}=2{,}72$$

there is a convincing advantage of *N*-control (dynamic modes). It is evident that the optimal controls, due to the specific fuel consumption (optimal in terms of the efficiency of the engine, which is the same) modes – *g*-control for fuel economy (!) are significantly inferior to the so-called dynamical (*N*-control of engine); but in comparison with other (ω - or with *M*-control) *g*-control has not so much significant advantage, as it might seem.

Consequently, the forced methods of driving the engine have an indisputable advantage over unforced ones. But, analyzing fuel consumption characteristics, it is easy to see that the surfaces Q_{AB}^{M} and Q_{AB}^{ω} are interconnected (see Fig. 6, c). A line of interconnection also have the surfaces Q_{AB}^{ω} Q_{AB}^{g} (see Fig. 6, b). This indicates that there are such combinations of allowable values of parameters t_i and τ_j for which the fuel consumption is either greater or lesser, and the same, such that the grounds for refusing to substantiate g – or ω -, ω or M way of control. Consequently, an unconditional general conclusion about the appropriateness of the application *g*, or only *M*-, or only ω -control of the heat engine in the process of switching gears can not be done. It turns out that with increasing the value of the τ_j parameter (the duration of the friction trailing), the range of the selection of the allowable switching moment t_i for all, without exception methods of driving the engine, first increases, and then falls.



a *б* *в*

Fig. 6. Fuel characteristics of a car

In general, it can be noted that against the backdrop of other types of control, the choice of the moment of start and the duration of switching in the process of *N*-control does not have a significant effect on fuel consumption. This conclusion should be taken as positive. But still, looking at the Fig. 6, *a*, we can observe that undoubtedly there are optimal values of the quantities $t_0 = t_0^*$ and $\tau = \tau^*$ - the coordinates of the minimum point on the surface $Q_{AB} = Q_{AB}^N(t,\tau)$. Therefore, the trailing of frictions does not necessarily have to be as small as possible.

Adhering to the principle of (global, but not always local) inviolability of the driver-selected driver's program, there is no reason to distinguish between dynamic and energy-saving transmission laws. Indeed, whatever the driver chooses, an optimal transmission control system is required to behave in such a way as to minimize fuel consumption, without in any way having to worry about how the driver works (without disturbing him). And therefore, nothing prevents the driver from choosing an extremely dynamic vehicle driving program if necessary, and in this case, the transmission control system will not have the opportunity to save fuel. From these considerations it follows that the problem of laying a step-by-step control system of various control algorithms (in particular, dynamic and energy-saving) is completely fictitious. It is easy to understand that the criteria for energy saving and dynamism do not contradict each other.

CONCLUSIONS

Summing up the implementation of the "principal" tasks declared here, it can be argued that the proposed scientific hypothesis in general was true:

1. There is a certain optimum friction shaft trailing (or any other gearshifting device), and therefore there is no reason to advertise the seams of the switching system as a sign of the excellence of the transmission. Because of this, the prospects of using, for example, cam clutches can be considered very shaky (especially out of sport). Switching gears with cam clutches without the use of a coupler (as on a motorcycle) for some 0.15 ... 0.20 s is unlikely to be advantageous even when it comes to dynamism of acceleration - there are more weighty factors (and, in general, this issue requires a more detailed study). But the short-live of these couplings exacerbates, among other things, even the problem of environmental friendliness.

2. The optimal gear shift laws are identified on the basis of not so much information that, therefore, one could beware of the excessive complexity of the microprocessor control system of the transmission.

3. There is no reason to oppose the energy-saving and dynamic (sports) transmission laws. Consequently, the motivation of poly-algorithmic control in perfect transmissions is actually supposed.

4. The "kickdown" response per driver share "accelerator pedal on the floor" does not belong to the special modes of control, it is rather just one of the modes of dynamic transmission control. And since dynamism is not an alternative to energy saving, then there is no reason to notice the need for "kickdown" at all.

5. Of course, one should not ignore the fact that from time to time (including now) the primitive mechanical with manual transmission in one way or another restore their position in the market. And the reason is not so much in the tastes of aggressive drivers (autosportsmen, for example, are committed to manual transmissions), but that there is no harmony between the promised advertising properties, reliability, price and cost of operation. On the other hand, even if the driver liked manual control or comfort based on the low cost of a primitive transmission, this is by no means the basis for "adjusting" such preferences: if manual control is far from optimal, then for "preferences" of an individual driver indirectly pays the whole society through the resource and environmental problems of inefficient car driving. The cost will no longer be particularly disturbing if the production of a robotic transmission will have an appropriate level of massive involvement. In general, automation should become so perfect as to completely eliminate the driver from the control of the transmission, turning the ergative control system with "hints" on automatism into a truly automatic [15].

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