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Optimization of the Transmission Ratio by Fuel Consumption

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Abstract
Introduction. The issues of ensuring optimal fuel consumption modes are the subject of constant research in the field of vehicle operation. Works in the field of reducing fuel consumption in most cases are devoted to the study of the design of power units, transmission or aerodynamic features of the body of cars. At the same time, the issues of determining the optimal laws of control of unsteady movement from the point of view of the synthesis of algorithms for controlling the movement of a car have not been considered. To optimize the transmission ratio of cars with unsteady movement, the authors of the work carried out analytical calculations to simplify the synthesis of motion control algorithms, which in turn allowed reducing fuel consumption. The aim of the work was to determine the optimal transmission ratio, which allowed solving the problem of synthesizing vehicle control to reduce fuel consumption.
Materials and Methods. The synthesis of algorithms for controlling the movement of a car considered in the work was based on the application of the needle variation of L.S. Pontryagin to invariant features of the real movement. An analytical method was used to estimate energy efficiency of vehicle performance, which was based on determination of optimal transmission ratio of motor vehicles taking into account minimum fuel consumption. The presented method took into account the amount of torque transmitted from the engine crankshaft to the transmission elements, which, depending on the engine power, was realized in the form of traction force on the wheels of the car.
Results. The law of optimal change in the transmission gear ratio during acceleration of the car in a minimum time was built. The problem of determining the optimal transmission ratio of the vehicle in the case of driving the vehicle at a constant speed and constant fuel supply and in the case of accelerating the vehicle to a given speed at a constant fuel supply, when the condition $\varepsilon = \text{const}$ was met, was solved. The result of the considered case of applying the optimal law of change in the transmission gear ratio was the minimization of fuel consumption under restrictions on acceleration (traction force) and speed of the car.
Discussion and Conclusion. The use by the authors of the analytical method for determining the transmission ratio of a car, as well as the use of this method in practical calculations for a car with given characteristics, showed the possibility of solving the problem of synthesizing vehicle control using a mathematical apparatus. This was confirmed by the built graphical dependence based on the results of the calculations. The considered cases of movement made it possible to determine the analytical dependencies of the optimal transmission ratio and the speed of the car. The initial data obtained by analytical relationships are applicable for cars with a mixed control mode.

Keywords: transmission ratio, fuel consumption, energy efficiency

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Научная статья

Оптимизация передаточного числа трансмиссии по расходу топлива

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Аннотация

Введение. Вопросы обеспечения оптимальных режимов расхода топлива являются предметом постоянного исследования в области эксплуатации автотранспортных средств. Работы в области снижения расхода топлива в большинстве случаев посвящены исследованию конструкции силовых установок, трансмиссии или аэродинамических особенностей кузова автомобилей. Вместе с тем вопросы определения оптимальных законов управления неустановившимся движением с точки зрения синтеза алгоритмов управления движением автомобиля не рассматривались. Для оптимизации передаточного числа автомобилей при неустановившимся движении авторами работы проведены аналитические расчеты, позволяющие упростить синтез алгоритмов управления движением, что позволяет снизить расход топлива. Целью работы являлось определение оптимального передаточного числа, позволяющего решить задачу синтеза управления транспортным средством для снижения расхода топлива.

Материалы и методы. Рассматриваемый в работе синтез алгоритмов управления движением автомобиля основан на применении к инвариантным признакам действительного движения игольчатой вариации Л.С. Понтрягина. Использован аналитический метод оценки энергетической эффективности эксплуатационных характеристик автомобиля, в основе которого лежит определение оптимального передаточного числа трансмиссии автотранспортных средств с учетом наименьшего расхода топлива. Представленный метод учитывает величину крутящего момента, передаваемого от коленчатого вала двигателя на элементы трансмиссии, который, в зависимости от мощности двигателя, реализуется в виде силы тяги на колесах автомобиля.

Результаты исследования. Построен закон оптимального изменения передаточного числа трансмиссии при разгоне автомобиля за минимальное время. Решена задача определения оптимального передаточного числа трансмиссии автомобиля в случае движения автомобиля с постоянной скоростью и постоянной подачей топлива, а также при разгоне автомобиля до заданной скорости при постоянной подаче топлива, когда соблюдается условие ε = const. Результатом рассмотренного случая применения оптимального закона изменения передаточного числа трансмиссии является минимизация расхода топлива при ограничениях на ускорение (силу тяги) и скорость движения автомобиля.

Обсуждение и заключения. Применение авторами аналитического метода для определения передаточного числа трансмиссии автомобиля, а также использование данного метода в практических расчетах для автомобиля с заданными характеристиками показывает возможность решения задачи синтеза управления транспортным средством с применением математического аппарата. Это подтверждается построенной графической зависимостью по результатам проведенных расчётов. Рассмотренные случаи движения позволили определить аналитические зависимости оптимального передаточного числа и скорости автомобиля. Исходные данные, полученные по аналитическим зависимостям, применимы для автомобилей со смешанным режимом управления.

Ключевые слова: передаточное число трансмиссии, расход топлива, энергетическая эффективность

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Introduction. Vehicle control mode is a combination of alternating acceleration, braking, as well as the need for various maneuvering when driving motor transport, taking into account its movement in urban areas. Such control leads to increased fuel consumption. Reducing fuel consumption in such conditions is usually associated with the need to increase the time for maneuvers [1]. This makes it necessary to search for optimal laws for controlling the unsteady movement of the car. In [2], energy characteristics of vehicle movement are determined primarily by the amount of torque transmitted from the engine crankshaft and the thrust force realized on the propellers, which depend on the engine power and the value of the gear ratios. The need to determine the optimal values of the transmission ratios of the vehicle is due to a decrease in the values of fuel characteristics [3], as well as a decrease in the amount of energy that is spent when maneuvering the car in case of unsteady movement. Gear ratios are determined in accordance with the specified characteristic of the engine and depend on the speed of movement of the vehicle, which has variable values [4].

When considering the approaches used to solve optimization problems, their diversity should be mentioned. Most of these approaches use methods of optimal control and calculus of variations. The authors in [5] used the maximum principle of L.S. Pontryagin with the definition of analytical dependencies for hybrid cars. A similar approach is considered in [6]. Based on this approach, it is proposed to construct the law of optimal change in the transmission ratio during acceleration of the car in the shortest time, as well as to determine fuel efficiency in the driving cycle [7]. The analysis of the conducted research has shown a significant theoretical groundwork in the field of determining the optimal gear ratio under various operating modes of the car. At the same time, it should be said that the optimal law of changing the transmission ratio should be determined taking into account the minimization of fuel consumption with restrictions on acceleration (traction) and the speed of the car. The authors of the presented work propose to apply an optimization method to determine the optimal laws of control of unsteady motion (from the point of view of the synthesis of control algorithms), which is based on the application of L.S. Pontryagin's needle variation to invariant signs of actual motion. The proposed approach has not been considered by the authors of previous studies.

The aim of the work is to determine the optimal gear ratio, which allows solving the problem of synthesizing vehicle control to optimize fuel consumption.

Materials and Methods. When calculating the instantaneous fuel consumption of a car, we used the following dependence:

$$G_T = q_N K_\omega K_N N_e, \tag{1}$$

where q_N — indicator of specific fuel consumption corresponding to the maximum value $N_e = N_{max}$; K_ω — value of the coefficient determined taking into account the speed mode of the engine; K_N — value of the coefficient determined taking into account the degree of engine load; N_e — engine power at the current time.

Coefficients K_ω and K_N were determined taking into account empirical dependencies, K_ω was determined taking into account the speed mode of the engine. Let us define it by formula:

$$K_\omega = a_\omega - b_\omega \cdot \frac{\omega_e}{\omega_N} + c_\omega \cdot \left(\frac{\omega_e}{\omega_N}\right)^2, \tag{2}$$

where $a_\omega = 1.27$; $b_\omega = 0.94$; $c_\omega = 0.67$; ω_e — current value of angular velocity of the crankshaft; ω_N — angular velocity of rotation of the crankshaft corresponding to the maximum engine power N_{max}.

Value ω_e is related to the gear ratio of transmission i and speed of the car $\dot{x}$ by dependence:

$$\omega_e = (i \cdot \dot{x}) / r_к . \tag{3}$$

Then formula (2) will take the form:

$$K_\omega = a_\omega - \frac{b_\omega}{\omega_N r_к}(i \cdot \dot{x}) + \frac{c_\omega}{(\omega_N r_к)^2}(i \cdot \dot{x})^2 . \tag{4}$$

Coefficient K_N depends on the degree of engine loading ε. Value ε was determined by the ratio of engine power at the current time, regardless of the mode of movement of the vehicle. We took into account: current value of the angular velocity of the crankshaft ω_e, engine power N_{ec} at the time of full fuel supply and the same value of the angular velocity of the crankshaft ω_e:

$$\varepsilon = \frac{N_e}{N_{ec}}. \tag{5}$$

Value of the current power of the engine N_e was determined by the differential equation of the movement of the car along the x axis, having the form [5]:

$$\frac{\delta}{g}\ddot{x} = \frac{N_e \eta_T}{mg\dot{x}} - \psi - k\dot{x}^2. \tag{6}$$

Here δ — value of the coefficient determined taking into account the power input, taking into account the increment of the kinetic energy of the rotating masses of the engine, transmission and wheels during acceleration of the car; η_T — efficiency of transmission; m — gross weight of the car; $\psi = f\cos\alpha + \sin\alpha$ — coefficient of resistance to movement; k — coefficient of air shape, which determines the strength of air resistance.

Coefficient δ in the presented work was determined by the following dependence $\delta = 1 + \gamma i^2$, in which value γ depended on the design parameters of the car and varied within $\gamma = 0.001–0.003$. Then, from equation (6), an expression can be obtained for the current value of engine power:

$$N_e = \frac{mg\dot{x}}{\eta_T}\left[\frac{1+\varepsilon i^2}{g}\ddot{x} + \psi(t) + k\dot{x}^2\right]. \tag{7}$$

Value of engine power N_{ec} at the time of maximum supply of the fuel mixture was determined using a well-known dependence (Leiderman formula) for the external characteristics of the engine, having the form:

$$N_{ec} = N_{max}\frac{\omega_e}{\omega_N}\left[A + B\frac{\omega_e}{\omega_N} - C\left(\frac{\omega_e}{\omega_N}\right)^2\right]. \tag{8}$$

Empirical coefficients A, B, C, depending on the engine parameters, can be determined, for example, by the dependencies given in [5]. After transformations taking into account (3), expression (8) takes the form

$$N_{ec} = F(ai\dot{x} + bi^2\dot{x}^2 - ci^3\dot{x}^3), \tag{9}$$

where coefficients a, b and c were determined by formulas:

$$F = \frac{N_{max}}{\omega_N r_k};\ a = A;\ b = \frac{B}{\omega_N r_k};\ c = \frac{C}{(\omega_N r_k)^2}.$$

Empirical dependence for coefficient K_N has the form:

$$K_N = a_N + b_N\cdot\varepsilon - c_N\cdot\varepsilon^2 - d_N\cdot\varepsilon^3, \tag{10}$$

where constants $a_N = 3.27$; $b_N = -8.22$; $c_N = -9.13$; $d_N = 3.18$ for gasoline engines; $a_N = 1.20$; $b_N = 0.14$; $c_N = 1.80$; $d_N = -1.46$ for diesel engines.

Degree of engine loading ε depending on (10), taking into account (5), (7) and (9), was determined by formula:

$$\varepsilon = \frac{\dfrac{mg\omega_N r_k}{N_{max}\eta_T}\left[\dfrac{1+\gamma i^2}{g}\ddot{x} + \psi(t) + k\dot{x}^2\right]}{ai + bi^2\dot{x} - ci^3\dot{x}^2}. \tag{11}$$

As an objective function, we used the amount of fuel consumption during time $t_к$, during which the car was moving along trajectory $x(t) \le x_к$ with variable speed $\dot{x} = f(t)$. The optimization problem was solved in accordance with the following algorithm: a law was established according to which the change in the transmission ratio was determined $i(\dot{x})$, and the corresponding trajectory $x(t)$ taking into account that the target functional took a minimum value:

$$J = \int_0^{t_K} G_T\dot{x}dt \to min, \tag{12}$$

gear ratio, current speed and power satisfied the constraints:

$$i \ge i_0,\ \dot{x} \le \dot{x}_{max},\ N_e \le N_{max}, \tag{13}$$

where i_0 — transmission ratio of the main gear.

Results. The calculation results were used for the case of a continuous change in the transmission ratio. Then, neglecting the inertia of rotating masses (at $\gamma = 0$), optimal value i_{opt}, if it existed, was found from condition:

$$\frac{\partial(G_T\dot{x})}{\partial i} = \left(\frac{\partial K_\omega}{\partial i} + \frac{\partial K_N}{\partial i}\frac{K_\omega}{K_N}\right)\dot{x} + K_\omega\frac{\partial\dot{x}}{\partial i} = 0. \tag{14}$$

Resulting expression (15) was used as the results of the study for two limiting cases.

The first case was the movement of the car with constant speed $\dot{x} = const$ and constant fuel supply, when ε and K_N = const. Therefore, the second term in equation (14) was zero, which allowed us to obtain a simple formula for determining the optimal transmission ratio:

$$i_{opt} = \frac{b_\omega\omega_N r_к}{2c_\omega\dot{x}}. \tag{15}$$

When minimizing the acceleration time to a given speed, the optimal transmission ratio was determined by formula [5]:

$$i_{opt}^p = \frac{B\omega_N r_к}{3C\dot{x}}\left(1 + \sqrt{1 + \frac{3AC}{B^2}}\right). \tag{16}$$

The second case was the acceleration of the car to a set speed with constant fuel supply, when the condition ε = const was met. In this case, the value of coefficient K_N = const and the optimal transmission ratio were determined by the formula from the solution of equation:

$$\frac{\partial K_\omega}{\partial i}\dot{x}+K_\omega\frac{\partial \dot{x}}{\partial i}=0.$$

After the transformations, taking into account expression (4) for K_ω we get:

$$\left[-\frac{b_\omega}{\omega_N r_\kappa}\dot{x}+2\frac{c_\omega}{(\omega_N r_\kappa)^2}i\cdot\dot{x}^2\right]\dot{x}+\left[a_\omega-\frac{b_\omega}{\omega_N r_\kappa}(i\cdot\dot{x})+\frac{c_\omega}{(\omega_N r_\kappa)^2}(i\cdot\dot{x})^2\right]\frac{\partial\dot{x}}{\partial i}=0. \quad (17)$$

Derivative $\frac{\partial \dot{x}}{\partial i}$ in equation (17) was determined from the condition ε = const using expression (11). At the same time, we neglected, as before, the inertia of rotating masses (γ = 0) and the force of air resistance $\left(k\dot{x}^2\approx 0\right)$. Then we established the relationship between the speed of the car and the transmission ratio (at ε = const) from solving the following equation:

$$ci^3\dot{x}^2-bi^2\dot{x}-ai+\frac{z(t)}{\varepsilon d}=0,$$

where the notation was introduced:

$$d=\frac{N_{max}\eta_T}{mg\omega_N r_k},\; z(t)=\left[\frac{\ddot{x}}{g}+\psi(t)\right].$$

The resulting equation with respect to velocity had a solution if its discriminant was greater than or equal to zero.

$$D=b^2i^4+4aci^4-\frac{4ci^3}{\varepsilon d}z(t)\ge 0.$$

Then the condition must be met for the transmission ratio:

$$i_*\ge\frac{z(t)}{\varepsilon d(B^2/4C+A)}. \quad (18)$$

The speed value corresponding to this condition was determined by formula:

$$\dot{x}_*=\frac{B\omega_N r_k}{2Ci_*}\left\{1+\sqrt{1+\frac{4C}{B^2}\left[A-\frac{z(t)}{\varepsilon d i_*}\right]}\right\}. \quad (19)$$

When accelerating the car, the maximum (minimum) value of the transmission ratio i_{min} was determined from condition (18), which allowed us to obtain a fairly simple expression for derivative $\frac{\partial \dot{x}}{\partial i}$ extend it to the entire trajectory of movement:

$$\frac{\partial\dot{x}}{\partial i}=-\frac{b}{2ci_{min}^2}=-\frac{B}{2Ci_{min}^2}\omega_N r_k. \quad (20)$$

Substituting the obtained expressions into equation (17), after the transformations, we obtained the equation for the optimal value of the transmission ratio:

$$\left[-\frac{b_\omega}{\omega_N r_\kappa}\dot{x}+2\frac{c_\omega}{(\omega_N r_\kappa)^2}i\cdot\dot{x}^2\right]\dot{x}+\left[a_\omega-\frac{b_\omega}{\omega_N r_\kappa}(i\cdot\dot{x})+\frac{c_\omega}{(\omega_N r_\kappa)^2}(i\cdot\dot{x})^2\right]\frac{\partial\dot{x}}{\partial i}=0,$$

$$\left[-\frac{Bb_\omega}{2Ci_*}\beta+\frac{c_\omega B^2}{2C^2}\beta\right]\frac{B\omega_N r_k}{2Ci_*}\beta-\left[a_\omega-\frac{b_\omega B}{2C}\beta+c_\omega\left(\frac{B}{2C}\beta\right)^2\right]\frac{B}{2Ci_{min}^2}\omega_N r_k=0,$$

$$\left[-\frac{b_\omega}{i_*}+\frac{2c_\omega B}{C}\right]\frac{\omega_N r_k}{i_*}-\left[\frac{2Ca_\omega}{B\beta^2}-\frac{b_\omega}{\beta}+c_\omega\left(\frac{B}{2C}\right)\right]\frac{\omega_N r_k}{i_{min}^2}=0,$$

$$\left[-\frac{b_\omega}{i_*}+\frac{2c_\omega B}{C}\right]\frac{1}{i_*}\omega_N r_k-\frac{R}{i_{min}^2}\omega_N r_k=0,$$

$$-b_\omega+\frac{2c_\omega B}{C}i_*\omega_N r_k-\frac{Ri_*^2}{i_{min}^2}\omega_N r_k=0,$$

$$\frac{Ri_*^2}{i_{min}^2} - \frac{2c_\omega B}{C} i_* \omega_N r_k + b_\omega \omega_N r_k = 0 ,$$

$$i_{*opt} = i_{min}^2 \left[\frac{c_\omega B}{RC} \mp \sqrt{\left(\frac{c_\omega B}{RC}\right)^2 \omega_N r_k - \frac{b_\omega}{i_{min}^2 R} \omega_N r_k} \right],$$

$$R = \left[\frac{2Ca_\omega}{B\beta^2} - \frac{b_\omega}{\beta} \omega_N r_k + c_\omega \left(\frac{B}{2C} \right) \omega_N r_k \right],$$

$$i_{min} = \frac{z(t)}{\varepsilon d(B^2 / 4C + A)} \omega_N r_k .$$

For example, the calculation of fuel consumption for a car with the following characteristics was carried out: total weight of the car $m = 1800$ kg; coefficients of external characteristics of the engine $A = 0.64$; $B = 1.36$; $C = 1.0$; maximum engine power $N_{max} = 100$ kW; angular rotation speed of the crankshaft corresponding to the maximum engine power $\omega_N = 576$ s^{-1}; wheel radius $r_k = 0.34$ m; efficiency of transmission $\eta_{mp} = 0.85$; transmission ratio of the main gear $i_0 = 3.4$ transmission ratios of the gear box $i_5 = 1.0$; $i_4 = 1.5$; $i_3 = 2.2$; $i_2 = 3.2$; $i_1 = 4.8$; coefficient of rotating masses $\delta = 1.0$; coefficient of resistance to movement $\psi = 0.12$.

The given example of calculating fuel consumption for a car with specified characteristics clearly showed the solution to the problem of synthesizing vehicle control. Acceleration graphs of a vehicle with a step-by-step gearbox, illustrating the solution for the optimal law of gear ratio change, are shown in Figure 1. For comparison, a graph for acceleration of a vehicle with a step-by-step gearbox is presented, which was built according to the obtained analytical dependence.

At the first stage of acceleration to speed $\dot{x}_{\min}$ the curves coincide. In this case $\dot{x}_{\min} = 6.14$ m/s or 22.1 km/h.

$$V_i(t) = \frac{V_{i1} \dfrac{V_{0i} - V_{i2}}{V_{0i} - V_{i1}} exp\left[t\sqrt{b_i^2 + 4a_i c_i} \right] - V_{i2}}{\dfrac{V_{0i} - V_{i2}}{V_{0i} - V_{i1}} exp\left[t\sqrt{b_i^2 + 4a_i c_i} \right]}, \tag{21}$$

$$a_i = \frac{A\phi_{\partial max} - \psi}{\delta} G; \; b_i = \frac{\beta_i \phi_{\partial max}}{\delta} G. \tag{22}$$

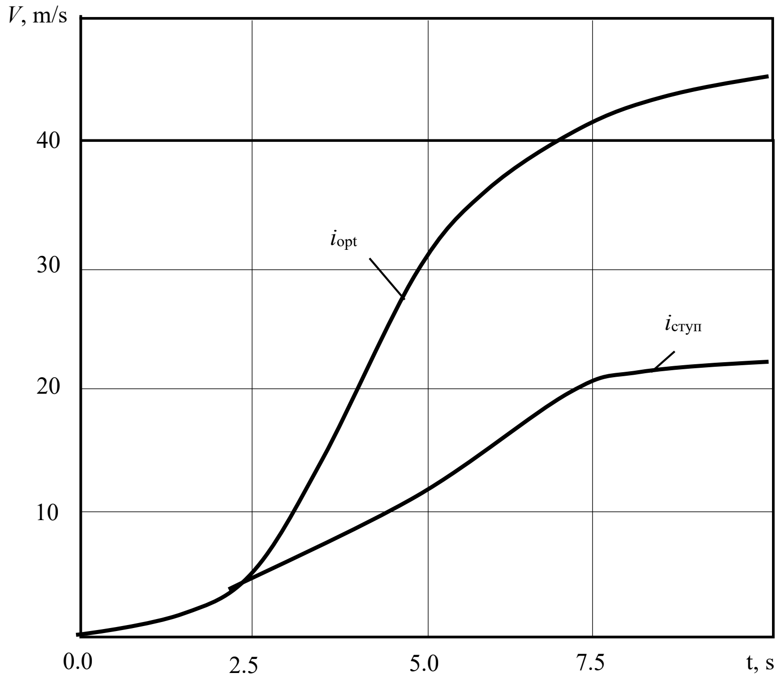

Fig. 1. Acceleration graphs of a car with a step-by-step gearbox
(dry rolled road α = 100; f = 0.025; ψ = 0.118):
i_{opt} — optimal transmission ratio;
i_{cm} — transmission

$$c_i = \frac{\gamma_i \phi_{дmax}}{\delta} G; \; V_{i1,2} = \frac{b_i \pm \sqrt{b_i^2 + 4a_i c_i}}{2c_i}, \tag{23}$$

where $\varphi_{дmax}$ — engine power factor; V_{0i} — initial speed of the machine in i-th gear; $i = \overline{1.3}$ — index indicating the gear number; $V_i(t)$ — speed of the car in i-th gear.

Discussion and Conclusion. The use of an analytical method by the authors in determining the transmission ratio, as well as its use in practical calculations for a car with specific characteristics, demonstrates the potential for solving the problem of vehicle control synthesis using a mathematical approach.

The goal set by the authors of this work — to determine the optimal gear ratio that allows them to solve the problem of vehicle control synthesis in order to reduce fuel consumption — has been achieved. As a result of the research conducted, analytical dependencies were obtained for two limiting cases of car motion.

The calculations carried out by the authors for the first case showed the optimal value of the transmission ratio, which was determined in accordance with formulas (15) and (16). The value of the transmission ratio is inversely proportional to the speed of the car. The speed value determined for the optimal transmission ratio will be less than the gear ratio determined by acceleration time. The formation of an empirical dependence in the second case with variable fuel supply $K_N = f(t)$ and the optimal transmission ratio were determined as a result of the numerical solution of equation (14).

The significance of the research is to simplify the solution to the problem of synthesizing control of unsteady vehicle movement, which is clearly demonstrated by the obtained graphical dependencies. A similar outcome can be achieved by applying the combined maximum principle.

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