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MODELING THE TRANSITIONAL PROCESSES OF MOTOR-VEHICLE ACCELERATION

KEYWORDS: slipping clutch, partial velocity characteristics of automobile engine, position of fuel-feeding pedal

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ABSTRACT The transitional processes of motor-vehicle acceleration are modelled in this article with reading the clutch slippery and the change of fuel-feeding pedal position by transition from one to another partial automobile engine characteristic up to the reach of a fixed regime of the engine. The composed systems of differential equations are solved by means of the programme product Matlab.

Introduction

The modern automotive flow is represented by various types of motor-vehicles, as they all are with continually higher dynamics properties. The dynamics properties of a motor-vehicle depend basically on the change of acceleration.

A model for studying the linear acceleration of the mass-centre of a motor-vehicle found in rectilinear motion without reading the transitional processes occurring during the motor-vehicle acceleration is created [1].

The goal of this research work is the formation of a mechanic-mathematical model of the transitions processes occurring during the motor-vehicle acceleration with reading the change of fuel-feeding pedal position by transition from one to another partial automobile engine characteristic when gears shifting and clutch slippery are available.

A Mechanic-mathematical Model

In Fig.1 is represented a model of a mechanical system “engine/clutch/working machine” with generalized coordinates as following:

- φ_d - an angle of crankshaft rotation;
- φ_c - an angle of clutch shaft rotation.

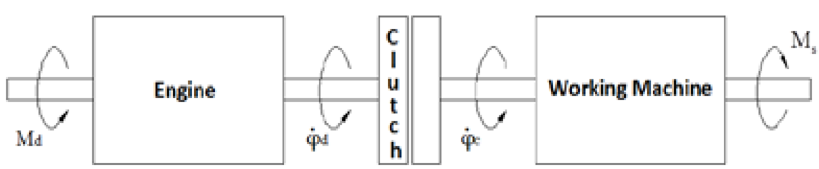

Fig.1 A mechanical system “engine/clutch/working machine”

The considered mechanical system is presented by a system of integral-differential equations

$$(1)\quad \begin{aligned} & I_d \cdot \ddot{\varphi}_d = M_i(\dot{\varphi}_d) - M_M(\dot{\varphi}_d) - M_{tr}(t) \\ & I_{a_j} \cdot \ddot{\varphi}_c = M_{tr}(t) - M_s(\dot{\varphi}_c) \\ & A_b = \int_0^t M_{tr}(t) \cdot (\dot{\varphi}_d - \dot{\varphi}_c) \cdot dt \end{aligned} ,$$

where: $\dot{\varphi}_d$ is the angular velocity of crankshaft; $\dot{\varphi}_c$ is the angular velocity of clutch shaft; I_d is the mass inertia moment of the motor-vehicle engine reduced to the crankshaft; I_{a_j} is the mass inertia moment of the motor-vehicle of the j -gear reduced to the clutch shaft.

During the clutch slippery I_{a_j} will be of the form $\left(I_{a_j} - I_d\right)$; $M_i\left(\dot{\varphi}_d\right)$is an indicator torque of the engine; $M_M\left(\dot{\varphi}_d\right)$ is the moment of mechanical losses in the engine; $M_s\left(\dot{\varphi}_c\right)$ is a resistance moment caused by the external load upon the clutch shaft; $M_{tr}(t) = \psi . t$ is a moment of friction of the clutch; ψ is a coefficient reading the rate of the increase of clutch moment of friction.

The work of clutch slippery A_b can be determined from the empirical dependence

(2) $$A_b = \frac{0{,}5 \cdot I_{a_j} \cdot M_{d_{max}} \cdot \dot{\varphi}_d^2}{M_{d_{max}} - M_s\left(\dot{\varphi}_c\right)}.$$

The effective torque of the engine is determined from the difference

(3) $$M_d = M_i\left(\dot{\varphi}_d\right) - M_M\left(\dot{\varphi}_d\right).$$

The indicator torque and the moment of mechanical losses are presented by the dependences

(4) $$M_i\left(\dot{\varphi}_d\right) = a_1 + a_2 \cdot \dot{\varphi}_d + a_3 \cdot \dot{\varphi}_d^2,$$

(5) $$M_M\left(\dot{\varphi}_d\right) = a_M + b_M \cdot \dot{\varphi}_d + c_M \cdot \dot{\varphi}_d^2,$$

where: $a_1, a_2, a_3,\ \ end\ \ a_M, b_M, c_M$ are approximation coefficients for the indicator characteristic and the characteristic of the mechanical losses, respectively [3].

The effective torque of the engine is presented with two sections in the system of equations. At the first section the motion of fuel-feeding pedal is simulated through the passage from one to another engine partial characteristic (from point B to point C; from point D to point E; etc. in Fig.2). The equation of this section has the form

(6) $$M_i\left(\dot{\varphi}_d\right) - M_M\left(\dot{\varphi}_d\right) = b \cdot \dot{\varphi},$$

where b is a linear coefficient showing the rate of fuel-feeding pedal pressing. This simulation is presented in details in Ref. [1].

The second section is responsible for the engine work at static characteristic presented with equations (4) and (5) (from point C to point D; from point E to point F; etc. in Fig.2).

The system of integral-differential equations (1) will change its form according to the considered stage of process of motor-vehicle acceleration. The mechanic-mathematical model is composed at the admission that the wheels motion is a pure rolling.

During the process of engine acceleration in a regime of idle running the system will be of the form

(7) $$\begin{aligned} I_d \cdot \ddot{\varphi}_d &= M_i\left(\dot{\varphi}_d\right) - M_M\left(\dot{\varphi}_d\right) \\ I_{a_j} \cdot \ddot{\varphi}_c &= M_M\left(\dot{\varphi}_c\right) \end{aligned},$$

where: $M_M\left(\dot{\varphi}_c\right)$ are the mechanical losses of the clutch during the engine in regime of working at idle running, which are slightingly small, and they are assumed to be equal to zero.

The change of the indicator moment and the moment of mechanical losses in the engine in regime of working at idle running is got from the system solution (7) by means of Matlab, and is shown with the section A-B in Fig. 2.

The start of motor-vehicle at engaging first gear and with clutch slippery is presented with the following differential equations:

(8) $$\begin{aligned} I_d \cdot \ddot{\varphi}_d &= M_i\left(\dot{\varphi}_d\right) - M_M\left(\dot{\varphi}_d\right) - M_{tr}(t) \\ \left(I_{a1} - I_d\right) \cdot \ddot{\varphi}_c &= M_{tr}(t) - M_s\left(\dot{\varphi}_c\right) \\ A_b &= \int_0^t M_{tr}(t) \cdot \left(\dot{\varphi}_d - \dot{\varphi}_c\right) \cdot dt \end{aligned},$$

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where: I_{a1} is a reduced mass inertia moment at first gear.

The simulation using system (8) continues up to the fulfilment of condition of equality for the angular velocities of engine crankshaft and clutch shaft, i. e. $\dot{\varphi}_d = \dot{\varphi}_c$.

The acceleration of motor-vehicle with engaging gear and completely engaged clutch is described by the following equation

$$\ddot{\varphi}_d = \ddot{\varphi}_c = \frac{1}{I_{aj}} \cdot \left[M_i(\dot{\varphi}_d) - M_M(\dot{\varphi}_d) - M_s(\dot{\varphi}_d) \right], \tag{9}$$

where: $M_s(\dot{\varphi}_d)$ is the resistance moment caused by the external load supplied upon the engine shaft.

The resistance caused by the external load is presented by the known dependence [2].

$$M_s = \frac{(f \cdot \cos\alpha + \sin\alpha) \cdot m_a \cdot g \cdot r + k \cdot F \cdot V^2 \cdot r}{i_{tr} \cdot \eta_{tr}}, \tag{10}$$

where: f is a coefficient of resistance at rolling; α is an angle of road longitudinal slope; m_a is the motor-vehicle mass; g is gravitational acceleration; r is the radius of motor-vehicle wheel; k is a coefficient of streamlining; F is the front area of the motor-vehicle; V is the relative velocity between air and motor-vehicle (it is equal to the motor-vehicle velocity at wind absence); i_{tr_j} is the gear ratio of the transmission of the j-gear; η_{tr} is the transmission efficiency.

Equation (9) is solved when a fixed regime of engine work is reached. In practice a fixed regime of engine work is not always reached because it is necessary to be passed in working at the regulating characteristic. The next gear shifting is most frequently executed as a decision of the motor-vehicle driver, and as a result a passage to a lower partial characteristic is done.

The gear shifting from j to $j+1$ is described with the system of equations

$$\begin{aligned} I_d \cdot \ddot{\varphi}_d &= M_i(\dot{\varphi}_d) - M_M(\dot{\varphi}_d) \\ (I_{aj} - I_d) \cdot \ddot{\varphi}_c &= -M_s(\dot{\varphi}_c) \end{aligned} . \tag{11}$$

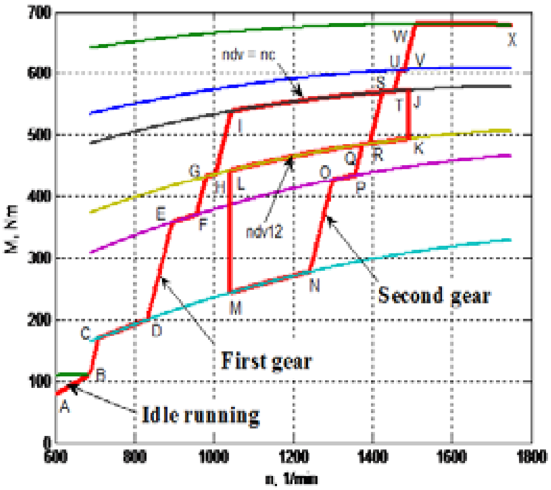

Fig. 2. A change of the effective engine moment during the motor-vehicle acceleration up to a fixed regime at second gear.

B. Results from the Numerical Solution

The systems of equations are solved by means of the programme product Matlab at the following data:

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$I_d = 3{,}1\left[kg \cdot m^2\right]$; $I_{aI} = 6{,}05\left[kg \cdot m^2\right]$; $I_{aII} = 14{,}95\left[kg \cdot m^2\right]$; $M_{d_{max}} = 700\left[\mathrm{N \cdot m}\right]$; $i_{tr_I} = 38{,}11$; $i_{tr_{II}} = 19{,}01$; $\eta_{tr} = 0{,}85$; $f = 0{,}02$; $\alpha = 0^\circ$; $m_a = 22500\left[kg\right]$; $F = 7{,}9\left[m^2\right]$; $r = 0{,}505\left[m\right]$; $k = 0{,}05\ \left[N \cdot s^2 / m^4\right]$.

The change of the effective engine moment as a function of the rotation frequency of crankshaft is presented in Fig. 2, where the nature of moment change during motor-vehicle acceleration at first gear from point B to point I (the system (8) is solved), and from point I to point J (the system (9) is solved) is evident. The second gear shifting at turned off clutch is according the J-K-L-M characteristics (the system (11) is solved). The motor-vehicle acceleration with engaging second gear is according the N-O-P-Q-R-S-T-U-V-W-X characteristic.

By means of this characteristic a greater accuracy is achieved in the determination of the motor-vehicle acceleration during its starting from an immobile position, during gear siftings, and when a fixed regime of second gear is reached. This accuracy is of a great significance for the determination of motor-vehicle kinematics parametres during turn at juncture.

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