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Reduction of energy losses on car movement while using a combined electromechanical drive of leading wheels

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Abstract: Fluctuations in the torque of an internal combustion engine (ICE) lead to additional energy losses, as it causes fluctuations in the speed and kinetic energy of the car. These losses increase as the frequency of oscillations of the torque of the internal combustion engine approaches the frequency of free (natural oscillations) of the running gear of the car in the longitudinal direction. If there is an elastic connection between the traction force and the movement of the car, the movement of the latter can be represented as complex. At the same time, the portable movement is uniform, and the relative movement is oscillatory. This article presents the results of the study of these losses for cars with mechanical and combined electromechanical drive wheels. Analytical expressions are obtained, which allow to take into account additional energy losses taking into account the tangential rigidity of the tire and the rigidity of the suspension in the longitudinal direction. When using a combined electromechanical drive of the drive wheels as well as in the case of a mechanical transmission of a car, resonance is dangerous. But with the increase in the share of torque k_{em} on the wheel generated by the electric motor, the relative additional energy losses for the movement of the car are reduced.

Keywords: torque, internal combustion engine, fluctuations, energy consumption, speed, combined electromechanical drive wheels

1. Introduction

Fluctuations in the torque of an Internal Combustion Engine (ICE) lead to additional energy losses, as it causes fluctuations in the speed and kinetic energy of the car.

The work of [1,2,3] is devoted to the influence of the irregularity of the torque of the internal combustion engine on the additional energy losses during steady-state car movement.

In the work [4], the authors note that the achievement of energy efficiency, road safety, environmental and economic safety indicators becomes the main organizational goal of the functioning of the motor transport system and is considered as an ongoing process for managing the technical level of vehicles. The works of many authors [5,6,7] is devoted to the study of energy-transforming properties of automobiles.

The stability of the characteristics of the vehicle's movement essentially depends on the work of the suspension of the car [8,9,10,11]. The suspension system acts as a bridge between the driver and the passengers of the vehicle and the road it is moving. It has two main functions [8]. One of them is to isolate the vehicle body from external inputs, which are mainly derived from non-ideal road surface. The other is to maintain a reliable contact between the road and the tires to ensure the stability of the trajectory. The best way in variable operating conditions is provided by adaptive active suspension systems [9]. The analysis of the influence of the traction properties of tires on the dynamic characteristics of vehicles the work [12] is devoted. Many authors emphasize the need to control the air pressure in car tires [12, 13].

The amplitude of the oscillations Apk of the traction force depends on the amplitude of the oscillations of the indicator torque Mi of the internal combustion engine [14].

2. Methodology

In the works [2, 3] it was determined that the additional energy losses due to fluctuations in torque and, accordingly, fluctuations in the tractive force on the wheels can be determined from the following relationship

$$\Delta W_s = \frac{A_p}{\pi} S, \tag{1}$$

here A_p - is the amplitude of oscillations of the traction force (when simulating these oscillations by the harmonic law [2]); S - the path traveled by the car (controlled mileage).

Analysis of equation (1) shows that additional energy losses due to vehicle movement caused by fluctuations in traction force on the wheels when using a combined electromechanical drive. Received [1] an expression to determine additional energy loss

$$\Delta W_s = \frac{k_1}{2\pi} \cdot \Sigma P_r \cdot S \cdot \left(1 - \frac{M_{em} \cdot n_1}{r_d \cdot \Sigma P_r}\right), \tag{2}$$

here k_1 - the coefficient of non-uniformity of torque [2, 3],

$$k_1 = 0.08 + \frac{14.44}{i_c}; \tag{3}$$

i_c - the number of cylinders of the internal combustion engine; $\sum P_r$ - the total force of resistance to movement of the vehicle; M_{em} - torque from the electric motor, supplied to the wheel of the car; n_1 - the number of driving wheels of the car; r_d - the dynamic radius of the drive wheels.

In the works [3], a relative indicator of additional energy loss of the car with the combined electromechanical drive of the wheels and variations in the torque of the internal combustion engine was proposed

$$\eta_{\Delta W} = \frac{\Delta W_s}{S \Sigma P_r} = \frac{\Delta W_s}{A_r} = \frac{k_1}{2\pi}\left(1 - k_{em}\right) = \frac{0.04 + \dfrac{7.22}{i_c}}{\pi}\left(1 - k_{em}\right), \tag{4}$$

where A_r - the total work force of external resistance to the movement of the vehicle; k_{em} - the proportion of torque on the wheels created by the electric motor,

$$k_{em} = \frac{M_{em} \cdot n_1}{r_d \cdot \Sigma P_r}. \tag{5}$$

It can be seen from equation (4) that with an increase in the coefficient ked the relative additional energy loss decreases due to the ICE torque variation. When k_{em} = 1 value $\eta_{\Delta W}$ = 0.

However, in the well-known works [1 – 3], the effect of the elastic coupling between the tractive force and the movement of the car on the additional energy losses is not investigated.

The aim of the study is to increase the energy efficiency of cars by reducing the additional energy losses associated with the longitudinal flexibility of the chassis.

To achieve this goal it is necessary to solve the following task: determine the additional energy loss for the combined electromechanical drive of the driving wheels of the car.

If there is an elastic connection between the traction force and the movement of the car, the movement of the latter can be represented as complex [15]. At the same time, the portable movement is uniform, and the relative movement is oscillatory (Fig. 1).

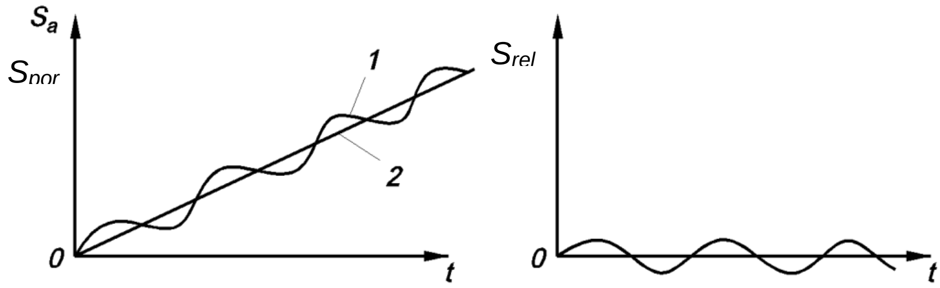

Figure1. The steady-state movement of the car with the elastic connection between the traction force and its movement: a - a graph of absolute (1) and portable (2) movements; b - relative motion graph

Portable movement of the car is described by the following equation

$$S_{por} = V_{por} \cdot t = \overline{V}_a \cdot t = S\ , \tag{6}$$

where V_{por} - the speed of the portable movement

$$V_{por} = \overline{V}_a\ ; \tag{7}$$

$\overline{V}_a$ - the average speed of the steady motion of the car; t – time.

The equation of the dynamics of the relative movement of the car can be represented as the equation of the dynamics of the relative movement of the car can be represented as

$$m_a \cdot \ddot{S}_{rel} + \alpha_{lon} \cdot \dot{S}_{rel} + c_{lon} \cdot S_{rel} = P_w - \Sigma P_r\ , \tag{8}$$

where m_a - the mass of the car; c_{lon}; α_{lon} - coefficients of stiffness and viscous friction, due to the tangential flexibility of the tires and the longitudinal flexibility of the suspension; P_w - traction force of the car.

3. Determination of additional energy loss for the mechanical transmission of the car

Equation (8) can be represented as

$$\ddot{S}_{rel} + 2n \cdot \dot{S}_{rel} + k^2 \cdot S_{rel} = \frac{P_w - \Sigma P_r}{m_a}\ , \tag{9}$$

here $2n$ - the damping coefficient,

$$2n = \frac{\alpha_{lon}}{m_a}\ ; \tag{10}$$

k - the frequency of free (natural) vibrations of the elastic system,

$$k = \sqrt{\frac{c_{lon}}{m_a}}\ . \tag{11}$$

The solution to the differential equation (9) is the sum of the total and particular solutions. Since with forced oscillations over time ($t \to \infty$), the general solution Sot tends to zero, then the general solution S_{rel} will be equal to the particular solution $(S_{rel})_0$. With the unevenness of the torque of the internal combustion engine, the total tractive force on the driving wheels of the vehicle can be represented by the following relationship:

$$P_w = \overline{P}_w + A_p \cdot \sin(\Omega_M \cdot t), \tag{12}$$

where $\overline{P}_w$ - the average value of the traction force; A_P - the oscillation amplitude of the traction force; Ω_M - the circular frequency of the internal combustion engine torque.

At the steady state of the car movement

$$\overline{P}_w = \Sigma P_r \,. \tag{13}$$

Equation (9) taking into account relations (12) and (13) takes the form

$$\ddot{S}_{rel} + 2n \cdot \dot{S}_{rel} + k^2 \cdot S_{rel} = \frac{A_p \cdot \sin(\Omega_M \cdot t)}{m_a} \,. \tag{14}$$

The particular solution of a second-order inhomogeneous differential equation (14) is obtained in the form

$$S_{rel} = (S_{rel})_0 = \frac{A_p}{m_a \cdot \left(k^2 - \Omega_M^2\right)^2 + 4n^2 \cdot \Omega_M^2} \cdot \left[\left(k^2 - \Omega_M^2\right) \cdot \sin(\Omega_M \cdot t) + 2n \cdot \Omega_M \cdot \cos(\Omega_M \cdot t)\right]. \tag{15}$$

Or, having made transformations, we will receive

$$S_{rel} = (S_{rel})_0 = \frac{A_p}{m_a \cdot \sqrt{\left(k^2 - \Omega_M^2\right)^2 + 4n^2 \cdot \Omega_M^2}} \cdot \sin(\Omega_M \cdot t + \varphi), \tag{16}$$

where φ - the angle of phase shift between the oscillations of the traction force and the relative movement of the car S_{rel},

$$\varphi = arctg\left(-\frac{2n \cdot \Omega_M}{k^2 - \Omega_M^2}\right). \tag{17}$$

From the expression (16) we determine the amplitude of oscillations of the relative movement of the vehicle

$$A_S = \frac{A_p}{m_a \cdot \sqrt{\left(k^2 - \Omega_M^2\right)^2 + 4n^2 \cdot \Omega_M^2}} \,. \tag{18}$$

The speed of the relative movement of the car

$$V_{rel} = \frac{dS_{rel}}{dt} = \frac{A_p \cdot \Omega_M}{m_a \cdot \sqrt{\left(k^2 - \Omega_M^2\right)^2 + 4n^2 \cdot \Omega_M^2}} \cdot \cos(\Omega_M \cdot t + \varphi). \tag{19}$$

Absolute vehicle speed

$$V_a = V_{por} + V_{rel} = \overline{V}_a + \frac{A_p \cdot \Omega_M}{m_a \cdot \sqrt{\left(k^2 - \Omega_M^2\right)^2 + 4n^2 \cdot \Omega_M^2}} \cdot \cos(\Omega_M \cdot t + \varphi). \tag{20}$$

In fig. 2 shows a graph of the change in the absolute speed of the car. In fig. 2 shows a graph of the change in the absolute speed of the car.

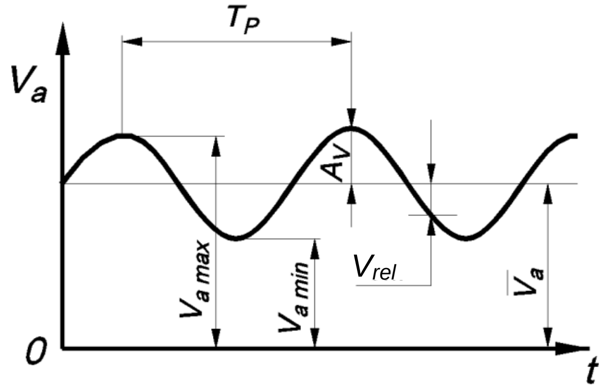

Figure 2. Tachogram of the complex movement of the vehicle when the traction force fluctuates

When the vehicle speed fluctuates, the level of its kinetic energy fluctuates. Compared with the uniform movement (portable), the additional energy consumption for the relative movement of the vehicle in one cycle of oscillation of the traction force will be equal (see fig. 2)

$$\Delta W_c = \frac{m_a \cdot \left(V_{a\max}^2 - V_{a\min}^2\right)}{2} = \frac{m_a \cdot \left(V_{a\max} - V_{a\min}\right) \cdot \left(V_{a\max} + V_{a\min}\right)}{2} = m_a \cdot \Delta V_a \cdot \overline{V}_a = 2 \cdot m_a \cdot \overline{V}_a \cdot A_V , \quad (21)$$

where ΔV_a - the maximum change in vehicle speed in one cycle of change in tractive force, $\Delta V_a = V_{a\,max} - V_{a\,\min}$; $\overline{V}_a$ - the average speed of the steady movement, equal to the speed of the portable movement,

$$\overline{V}_a = \frac{V_{a\max} - V_{a\min}}{2} ; \quad (22)$$

A_V – oscillation amplitude of the relative vehicle speed.

From the equation (19) we define

$$A_V = \frac{A_p \cdot \Omega_M}{m_a \cdot \sqrt{\left(k^2 - \Omega_M^2\right)^2 + 4n^2 \cdot \Omega_M^2}} . \quad (23)$$

After substituting the equation (23) into the expression (21), we obtain

$$\Delta W_c = 2A_p \cdot \overline{V}_a \cdot \frac{\Omega_M}{\sqrt{\left(k^2 - \Omega_M^2\right)^2 + 4n^2 \cdot \Omega_M^2}} \quad (24)$$

Additional energy consumption during t

$$\Delta W_t = \Delta W_ц \cdot \frac{t}{T_p} , \quad (25)$$

where T_p - the period of oscillation of the traction force,

$$T_p = \frac{2\pi}{\Omega_M} . \quad (26)$$

The expression (25) with (24) and (26) takes the form

$$\Delta W_t = \frac{A_p \cdot \overline{V}_a \cdot t \cdot \Omega_M^2}{\pi \cdot \sqrt{\left(k^2 - \Omega_M^2\right)^2 + 4n^2 \cdot \Omega_M^2}} . \quad (27)$$

Given that the path traveled by the car (its mileage) is determined by the dependence (6), we obtain the additional energy consumption depending on its mileage

$$\Delta W_S = \frac{A_p \cdot \Omega_M^2 \cdot S}{\pi \cdot \sqrt{\left(k^2 - \Omega_M^2\right)^2 + 4n^2 \cdot \Omega_M^2}} . \tag{28}$$

The amplitude of oscillations of the traction force on the driving wheels is defined in [2, 3]

$$A_p = \frac{\overline{M}_i \cdot u_{tr} \cdot \eta_{me} \cdot \eta_{tr}}{2r_d} \cdot \left(0.08 + \frac{14.44}{i_c}\right) \cdot \left[1 - \frac{1}{\eta_{me} \cdot \eta_{tr} \cdot \left(1 + \frac{m_a \cdot r_k^2}{I_{red}^w}\right)}\right] , \tag{29}$$

where $\overline{M}_i$ - the average indicated engine torque; u_{tr} - gear transmission ratio; η_{me}; η_{tr} - mechanical efficiency of the engine and transmission; r_k - the kinematic radius of the wheel; I_{red}^w - reduced to the wheels the moment of inertia of the transmission and the rotating mass of the engine.

The expression (28), after substituting equation (29) into it, transforms to the following form

$$\Delta W_S = \frac{\Omega_M^2 \cdot S}{\pi \cdot \sqrt{\left(k^2 - \Omega_M^2\right)^2 + 4n^2 \cdot \Omega_M^2}} \cdot \frac{\overline{M}_i \cdot u_{tr} \cdot \eta_{me} \cdot \eta_{tr}}{2r_d} \cdot \left(0.08 + \frac{14.44}{i_c}\right) \cdot \left[1 - \frac{1}{\eta_{me} \cdot \eta_{tr} \cdot \left(1 + \frac{m_a \cdot r_k^2}{I_{red}^w}\right)}\right] . \tag{30}$$

Circular frequency of oscillation of torque [2, 3] Circular frequency of oscillation of torque [2, 3]

$$\Omega_M = \frac{\overline{\omega}_e \cdot i_c}{2} , \tag{31}$$

where $\overline{\omega}_e$ - the average angular velocity of the shaft of the ICE,

$$\overline{\omega}_e = \frac{\overline{V}_a \cdot u_{tr}}{r_k} . \tag{32}$$

In the equation (30)

$$\frac{\overline{M}_i \cdot u_{tr} \cdot \eta_{me} \cdot \eta_{tr}}{r_d} = \overline{P}_w . \tag{33}$$

Transforming equation (30) into account the expressions (31) - (33), we get

$$\Delta W_S = \frac{S \cdot \overline{P}_w \cdot \left(0.04 + \frac{7.22}{i_c}\right) \cdot \left[1 - \frac{1}{\eta_{me} \cdot \eta_{tr} \cdot \left(1 + \frac{m_a \cdot r_k^2}{I_{red}^w}\right)}\right]}{\pi \cdot \sqrt{\left(\frac{k^2}{\Omega_M^2} - 1\right)^2 + \frac{16 \cdot n^2 \cdot r_k^2}{\overline{V}_a^2 \cdot u_{tr}^2 \cdot i_c^2}}} . \tag{34}$$

Given the ratio

$$W_{por} = S \cdot \Sigma P_r = S \cdot \overline{P}_w , \tag{35}$$

convert (34) to the form

$$\frac{\Delta W_S}{W_{por}} = \frac{\left(0.04 + \frac{7.22}{i_c}\right) \cdot \left[1 - \frac{1}{\eta_{me} \cdot \eta_{tr} \cdot \left(1 + \frac{m_a \cdot r_k^2}{I_{red}^w}\right)}\right]}{\pi \cdot \sqrt{\left(\frac{k^2}{\Omega_M^2} - 1\right)^2 + \frac{16 \cdot n^2 \cdot r_\kappa^2}{\overline{V}_a^2 \cdot u_{tr}^2 \cdot i_c^2}}} , \tag{36}$$

where W_{por} - energy losses for the portable movement of the car.

In the equation (36), the value $\frac{m_a \cdot r_k^2}{I_{red}^w}$ can be expressed in terms of the coefficient δ_{rot} accounting for the rotating masses of the transmission and engine

$$\frac{m_a \cdot r_k^2}{I_{red}^w} = \frac{1}{\delta_{rot} - 1} \quad . \tag{37}$$

The coefficient of accounting for rotating masses can be determined by the well-known [16] formula

$$\delta_{rot} = 1 + A_1 + A_2 \cdot u_g^2 \quad , \tag{38}$$

where A_1 - the coefficient of accounting for the rotating mass of the transmission associated with the wheels with a constant gear ratio; A_2 - factor accounting for the rotating mass of the transmission and the engine associated with the wheels of a variable gear ratio; u_g - gear ratio.

The expression (36) with (37) and (38) takes the form

$$\frac{\Delta W_S}{W_{por}} = \left(1 - \frac{1 - \dfrac{1}{1 + A_1 + A_2 \cdot u_g^2}}{\eta_{me} \cdot \eta_{tr}} \right) \frac{0.04 + \dfrac{7.22}{i_c}}{\pi \cdot \sqrt{\left(\dfrac{k^2}{\Omega_M^2} - 1 \right)^2 + \dfrac{16 \cdot n^2 \cdot r_k^2}{\overline{V}_a^2 \cdot u_{tr}^2 \cdot i_c^2}}} \quad . \tag{39}$$

To estimate the additional energy losses caused by the oscillations of the traction force, when conducting a preliminary analysis, we can assume that the elastic coupling is $n = 0$ and there is no effect of rotating transmission masses, i.e. $\delta_{rot} \approx 1$. Subject to accepted assumptions, equation (39) will be simplified

$$\frac{\Delta W_S}{W_{por}} = \frac{0.04 + \dfrac{7.22}{i_c}}{\pi \cdot \left| \dfrac{k^2}{\Omega_M^2} - 1 \right|} \quad . \tag{40}$$

In fig. 3 and fig. 4 shows dependency graphs $\frac{\Delta W_S}{W_{por}} = F\left(\frac{k}{\Omega_M}\right)$ for different numbers of cylinders i_c of an internal combustion engine. The relative increase in additional losses $\frac{\Delta W_S}{W_{por}}$ as can be seen from fig. 3 and fig. 4, higher at $\frac{k}{\Omega_M} < 1$ than at $\frac{k}{\Omega_M} > 1$. Obviously, to reduce the ratio in accordance with the equation (11), it is necessary to increase the coefficient of tangential rigidity of tires and reduce the longitudinal compliance (increase the longitudinal rigidity) of the suspension.

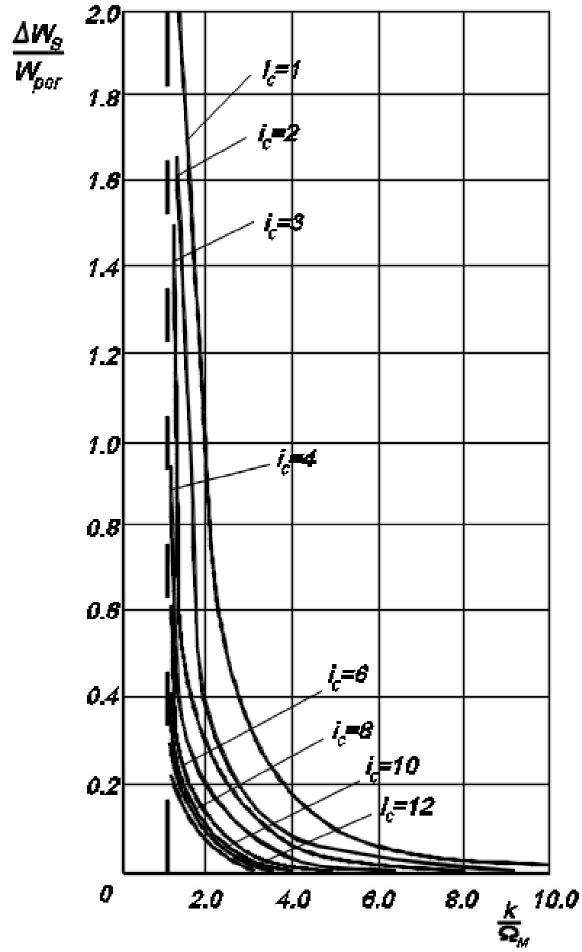

Figure 3. Dependence of relationship $\frac{\Delta W_S}{W_{por}}$ on relationship $\frac{k}{\Omega_M}$ with different values i_c and $\frac{k}{\Omega_M} > 1$

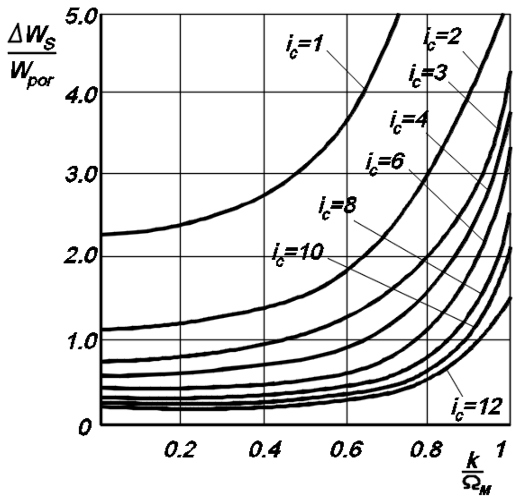

Figure 4. Dependence of relationship $\frac{\Delta W_S}{W_{por}}$ on relationship $\frac{k}{\Omega_M}$ with different values ic and $\frac{k}{\Omega_M} < 1$

From the equation (23) it can be seen that with absolutely rigid tires and suspension (κ = 0; n = 0) it will become (1), which corresponds to the result obtained earlier in the work [2].

4. Determination of additional energy loss for the combined electromechanical drive of the vehicle drive wheels

When using the combined electromechanical drive of the drive wheels, the amplitude of the oscillations of the traction force decreases [1]. In the work [1], the dependence was obtained to determine the amplitude of oscillations of the traction force in the case of using a combined electromechanical drive

$$A_p = \frac{k_1}{2r_d}\cdot\left(\bar{M}_i\cdot u_{tr}\cdot\eta_{me}\cdot\eta_{tr} - M_{em}\cdot n_1\right) = \frac{k_1}{2}\cdot\Sigma P_r\cdot\left(1-k_{em}\right) = \left(0.04+\frac{7.22}{i_c}\right)\cdot\left(1-k_{em}\right)\cdot\Sigma P_r\,. \tag{41}$$

After substituting the expression (41) into the equation (28), we get

$$\Delta W_S' = \frac{\left(0.04+\frac{7.22}{i_c}\right)\cdot S\cdot\Sigma P_r\cdot\left(1-k_{em}\right)\cdot\Omega_M^2}{\pi\cdot\sqrt{\left(k^2-\Omega_M^2\right)^2+4n^2\cdot\Omega_M^2}}\,. \tag{42}$$

Transforming the equation (42), we obtain the relative additional energy loss transforming the equation (42), we obtain the relative additional energy loss

$$\frac{\Delta W_S'}{S\cdot\Sigma P_r} = \frac{\Delta W_S'}{A_r} = \frac{\Delta W_S'}{W_{por}} = \frac{\left(0.04+\frac{7.22}{i_c}\right)\cdot\left(1-k_{em}\right)}{\pi\cdot\sqrt{\left(\frac{k^2}{\Omega_M^2}-1\right)^2+\frac{4\cdot n^2}{\Omega_M^2}}}\,. \tag{43}$$

Assuming the assumption that there is no damping (n = 0), we transform (43) to

$$\frac{\Delta W_S'}{W_{por}} = \frac{\left(0.04+\frac{7.22}{i_c}\right)\cdot\left(1-k_{em}\right)}{\pi\cdot\left|\frac{k^2}{\Omega_M^2}-1\right|}\,. \tag{44}$$

Comparing the expressions (40) and (44) with each other, we can conclude that the use of a combined electromechanical drive allows one to reduce (compared with a mechanical drive) additional energy losses. When k_{em} = 1 (with all-electric drive), the indicated losses are equal to zero.

Authors should discuss the results and how they can be interpreted in perspective of previous studies and of the working hypotheses. The findings and their implications should be discussed in the broadest context possible. Future research directions may also be highlighted.

5. Conclusions

Longitudinal flexibility of the running gear of the car when the traction force on the wheels fluctuates leads to an increase in additional energy losses. These losses increase as the frequency of oscillations of the torque of the internal combustion engine approaches the frequency of free (natural oscillations) of the running gear of the car in the longitudinal direction.

When designing cars, it is necessary to strive to ensure that the ratio $\frac{k}{\Omega_M}$ is significantly greater than one. With $\frac{k}{\Omega_M}>4$ relative additional energy losses $\frac{\Delta W_S}{W_{por}}$ does not exceed the value of 0.2, and when $\frac{k}{\Omega_M}>6$ - does not exceed the value of 0.1.

When using a combined electromechanical drive of the drive wheels as well as in the case of a mechanical transmission of a car, resonance is dangerous, i.e. equality $k = \Omega M$. But with the increase in the share of torque k_{em} on the wheel generated by the electric motor, the relative additional energy losses for the movement of the car are reduced. When $k_{em} = 1$, these losses are equal to zero.

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