UDC 629.017 DOI: 10.30977/AT.2219-8342.2024.55.0.04

Cycle mechanical coefficient of useful effect of motor-transmission installations of transportation and tractor vehicles

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Annotation. Problem. *When determining the efficiency coefficient of the motor-transmission systems of cars and tractors, it is necessary to consider not only the losses due to viscous and dry friction but also the losses caused by the circulation of potential and kinetic energy within the transmission. These losses result from the use of typical internal combustion engines in motor-transmission systems. Currently, existing studies do not take into account the dynamic cyclic energy losses in the transmission, which prevents an accurate assessment of the cyclic efficiency coefficient.* ***Purpose.*** *The purpose is to determine the cyclic efficiency coefficient of motor-transmission systems with internal combustion engines, taking into account the cyclic dynamic efficiency coefficient of the transmission in vehicles equipped with internal combustion engines.* ***Methodology.*** *The approaches adopted in this work to achieve the stated goal are based on the theoretical foundations for determining the dynamic efficiency coefficient, the work balance in the motor-transmission system of a car (or tractor) over one oscillation period of the indicator torque and the angular velocities of the engine's crankshaft during one oscillation period of its torque.* ***Results.*** *The cyclic mechanical efficiency coefficient of motor-transmission systems in wheeled machines has been determined, accounting for the losses associated with the acceleration of moving masses and the torque irregularities inherent to internal combustion engines. A relationship has been established in which the cyclic coefficients of dynamic and elastic losses are equal to zero when the torque irregularity coefficient is zero, a characteristic typical of electric motors.* ***Originality****. The results of the study provide a general understanding of the proportionality of the cyclic dynamic coefficient of mechanical losses in the transmission to the torque irregularity coefficient and the difference between the squares of the circular frequencies of torque oscillations and the natural (free) oscillations of the transmission input shaft.* ***Practical meaning.*** *The obtained results can be recommended for identifying frequency coincidences that lead to resonance, where the cyclic coefficients of elastic and dynamic losses increase sharply.*

Key words: *motor-transmission installation, internal combustion engine, mass, dynamic efficiency, energy, friction, crankshaft, oscillation, torque.*

Introduction

The efficiency coefficient (EC) is an indicator of the energy efficiency of machines and mechanisms. When evaluating the efficiency coefficient of the motor-transmission systems of cars and tractors, it is necessary to consider not only losses due to viscous and dry friction but also losses caused by the circulation of potential and kinetic energy within the transmission. These losses result from the use of internal combustion engines (ICE) in motor-transmission systems, which serve as a source of oscillations in the indicator torque.

An analysis of known research results revealed that dynamic cyclic energy losses in the transmission have not been accounted for in previous studies, preventing an accurate assessment of the cyclic EC.

This study refines the cyclic EC of motor-transmission systems with ICEs by determining the cyclic dynamic EC. The dynamic EC is a component of the overall EC of the motor-transmission system, taking into account losses caused by the circulation of the kinetic energy of rotating transmission masses and the translational motion of the vehicle's or tractor's mass.

Analysis of publications

The efficiency coefficient (EC) is one of the indicators of the energy efficiency of motor-transmission systems in transport and traction machines [1], [2]. In [3], the concept of dynamic EC was introduced, which accounts for energy losses caused by oscillations in the rotational speed of transmission masses.

The study [1] provides a formula for calculating the instantaneous overall EC of the transmission.

$$\eta_{tr}^{mgn} = \frac{N_k}{N_e} = 1 - \frac{N_{st}}{N_e} - \frac{N_{vt}}{N_e} - \frac{N_{vr}}{N_e} = = 1 - \left(\psi_{tr}^{st} + \psi_{tr}^{kin} + \psi_{tr}^{din}\right), \quad (1)$$

where N_k – power at the driving wheels; N_e – effective engine power; N_{st} – power consumed to overcome dry friction forces; N_{vt} – power consumed to overcome viscous friction forces; N_{vr} – power consumed to accelerate the rotating masses of the transmission; $\psi_{tr}^{st}, \psi_{tr}^{kin}, \psi_{tr}^{din}$ – power loss coefficients of the engine for overcoming dry and viscous friction forces, as well as for accelerating the rotating masses of the transmission.

Since losses due to dry friction do not depend on the speed of the rotating transmission elements, the coefficient for overcoming dry friction forces in work [1] is referred to as the static loss coefficient. The coefficient for overcoming viscous friction forces, which depends on speed, is called the kinematic loss coefficient. The coefficient for power loss due to the acceleration of rotating masses, which depends on acceleration, is referred to as the dynamic loss coefficient in work [1].

In work [1], it is shown that for stable transmission operation, the sum of the static, kinematic, and dynamic loss coefficients must be less than one.

$$\psi_{tr}^{st} + \psi_{tr}^{kin} + \psi_{tr}^{din} \le 1. \quad (2)$$

Considering that

$$\psi_{tr}^{st} = 1 - \left(\eta_{tr}^{st}\right)_{mgn}, \quad (3)$$

$$\psi_{tr}^{kin} = 1 - \left(\eta_{tr}^{kin}\right)_{mgn}, \quad (4)$$

$$\psi_{tr}^{din} = 1 - \left(\eta_{tr}^{din}\right)_{mgn}, \quad (5)$$

we transform equation (1) from [1] into the following form

$$\eta_{tr}^{mgn} = \eta_{tr}^{st} + \eta_{tr}^{kin} + \eta_{tr}^{din} - 2 \ge 0. \quad (6)$$

Inequality (6) characterizes the condition for stable transmission operation in terms of power indicators.

In works [1], [2], the impact of elastic links on energy losses in the transmission due to the variation of engine torque according to a periodic law, typical for internal combustion engines (ICE), has been determined. By approximating the periodic function of torque variation with a harmonic law, [1], [2] have defined the cyclic elastic efficiency coefficient of the transmission

$$\eta_{tr}^{upr} = 1 - \frac{A_{me} \cdot \left(1 - \dfrac{A_{me}}{2 \cdot M_{sopr}}\right)}{J_p \cdot \bar{\omega}_e \cdot \omega_m \cdot \pi \cdot \left(\dfrac{k^2}{\omega_m^2} - 1\right)}, \quad (7)$$

where A_{me} – amplitude of the oscillations of the effective torque of the ICE; ω_m – angular frequency of torque oscillations; J_p – the moment of inertia of the transmission reduced to the crankshaft of the internal combustion engine (when installed on the vehicle, the translationally moving mass of the machine is taken into account); M_{sopr} – the resistance torque, reduced to the crankshaft of the internal combustion engine, is assumed to be equal to the average engine torque during steady-state operation – in this case, it is considered constant.; $\bar{\omega}_e$ – the average angular velocity of the crankshaft of the ICE; k – the angular frequency of the natural (free) oscillations of the transmission input shaft.

For an internal combustion engine, the amplitude of the oscillations of the effective torque can be determined as

$$A_{me} = A_{m_i} \cdot \eta_{mdv} = 0.5 \cdot \bar{M}_i \cdot \eta_{mdv} \cdot K_i, \quad (8)$$

where η_{mdv} – mechanical efficiency of the internal combustion engine; $\overline{M_i}$ – average indicator torque of the ICE; A_{m_i} – amplitude of the oscillations of the indicator torque of the ICE; K_i – torque irregularity coefficient [1], [2]

$$K_i = \frac{M_{i\max} - M_{i\min}}{\overline{M}_i}; \qquad (9)$$

where $M_{i\max}$, $M_{i\min}$ – maximum and minimum values of the torque over one cycle of oscillation.

However, in works [1], [2], the cyclic dynamic efficiency coefficient of the transmission has not been determined, which prevented deriving an expression for the overall cyclic efficiency coefficient of the motor-transmission system. A significant number of studies have been dedicated to evaluating the energy efficiency and vibration stability of the system, both in our country [4-6,15] and abroad [7-14, 16].

Purpose and Tasks

The purpose of the study is to determine the cyclic efficiency coefficient of motor-transmission systems with internal combustion engines (ICE), taking into account the cyclic dynamic efficiency coefficient of the transmission.

To achieve this goal, it is necessary to determine the cyclic dynamic efficiency coefficient of the transmission in a vehicle with an internal combustion engine.

Determination of the cyclic efficiency coefficient of the motor-transmission system of a vehicle

The work balance in the motor-transmission system of a vehicle over one period of oscillation of the indicator torque can be determined as the work of the engine A_e distributed among the following tasks

$$A_e = A_i \cdot \eta_{mdv} = A_p + A_{st} + A_v + A_{din} + A_u. \qquad (10)$$

In equation (10), the following are denoted: A_p - the useful work performed by the motor-transmission system over one period of oscillation of the indicator torque; A_{st} and A_v - the work spent to overcome dry and viscous friction forces over one period of oscillation of the indicator torque; A_u - the work spent to overcome elastic deformation forces over one period of oscillation of the indicator torque; A_{din} - the work spent to accelerate the rotating and gradually moving masses over one period of oscillation of the indicator torque.

It should be noted that the period of oscillation of the effective torque is equal to the period of oscillation of the indicator torque.

By dividing both the left and right sides of equation (10) by A_e, we obtain:

$$1 = \left(\eta_{tr}\right)_c + \psi_{tr}^{st} + \psi_{tr}^{kin} + \psi_{tr}^{din} + \psi_{tr}^{u}. \qquad (11)$$

Alternatively, considering the relationship between the efficiency coefficient and the loss coefficients, we can transform equation (11) into the following form

$$\left(\eta_{tr}\right)_c = \eta_{tr}^{st} + \eta_{tr}^{v} + \eta_{tr}^{din} + \psi_{tr}^{u} - 3 \ge 1. \qquad (12)$$

Equation (12) represents the condition for stable operation of the motor-transmission system of the vehicle in terms of energy indicators.

Determination of the cyclic dynamic efficiency coefficient of the motor-transmission system

Over one period of oscillation of the torque, the angular velocity of the crankshaft will decrease within the range [$\omega_{e\min}$; $\omega_{e\max}$].

The maximum and minimum angular velocities of the crankshaft, as defined in works [1], [2], can be determined using the following equations

$$\omega_{e\max} = \overline{\omega}_e + \frac{\overline{M}_i \cdot R_i}{\overline{\omega}_e \cdot i_c \cdot J_p \cdot \left(1 - \left(\frac{2 \cdot k}{\overline{\omega}_e \cdot i_c}\right)^2\right)}, \qquad (13)$$

$$\omega_{e\min} = \overline{\omega}_e - \frac{\overline{M}_i \cdot R_i}{\overline{\omega}_e \cdot i_c \cdot J_p \cdot \left(1 - \left(\frac{2 \cdot k}{\overline{\omega}_e \cdot i_c}\right)^2\right)}, \qquad (14)$$

$$J_p = J_p^{ICE} + J_{pI}^{tr} + J_{pII}^{tr} + \frac{m \cdot r_k^2}{u_0^2 \cdot u_k^2} \qquad (15)$$

where i_c – the number of cylinders in an ICE; J_p^{ICE} – the moment of inertia of the moving masses of the ICE reduced to the crankshaft; $J_{pI}^{tr}, J_{pII}^{tr}$ – the moments of inertia of masses, related to variable and constant gear ratios, reduced to the crankshaft of the engine; m – the mass of the vehicle; r_k – the kinematic radius of the driving wheels of the vehicle; u_k, u_o – the gear ratios of the gearbox and the final drive.

The change in the angular velocity of the crankshaft of the internal combustion engine over one cycle of torque oscillation leads to a change in the kinetic energy of the moving masses of the transmission and the vehicle within the range [$W_{k\min}$; $W_{k\max}$]

$$W_{k\max} = \frac{J_p \cdot \omega_{e\max}^2}{2}, \qquad (16)$$

$$W_{k\min} = \frac{J_p \cdot \omega_{e\min}^2}{2}, \qquad (17)$$

The change in the kinetic energy of the moving masses is not used beneficially and, therefore, is lost. This change in the kinetic energy of the moving masses is the loss.

$$\Delta W_k = W_{k\max} - W_{k\min} = J_p \cdot \bar{\omega}_e \cdot \Delta\omega_e, \qquad (18)$$

where $\Delta\omega_e$ – the difference between the maximum and minimum angular velocities of the crankshaft over one period of torque oscillation,

$$\Delta\omega_e = \omega_{e\max} - \omega_{e\min}, \qquad (19)$$

Equation (19), taking into account (13) and (14), will take the following form

$$\Delta\omega_e = \frac{2 \cdot \bar{M}_i \cdot R_i}{\bar{\omega}_e \cdot i_c \cdot J_p \cdot \left(1 - \left(\frac{2 \cdot k}{\bar{\omega}_e \cdot i_c}\right)^2\right)}. \qquad (20)$$

After substituting equation (20) into equation (18), we will obtain

$$\Delta W_k = \frac{2 \cdot \bar{M}_i \cdot R_i}{i_c \cdot \left(1 - \left(\frac{2 \cdot k}{\bar{\omega}_e \cdot i_c}\right)^2\right)} = A_{din}. \qquad (21)$$

Thus, the dynamic cyclic loss coefficient can be expressed as

$$\psi_{tr}^{din} = \frac{A_{din}}{A_e}. \qquad (22)$$

The effective work of the internal combustion engine (ICE) over one period of torque oscillation, previously defined in works [1], [2], is determined by the following relation

$$A_e = \bar{M}_i \cdot \frac{2 \cdot \pi \cdot \bar{\omega}_e}{\omega_m} = 2 \cdot \pi \cdot \bar{M}_i \cdot \frac{2 \cdot \bar{\omega}_e}{\bar{\omega}_e \cdot i_c} = 4 \cdot \pi \cdot \frac{\bar{M}_i}{i_c}. \qquad (23)$$

After substituting equations (21) and (23) into equation (22) and performing the transformation, we will obtain:

$$\psi_{tr}^{din} = \frac{0.5 \cdot K_i}{1 - \left(\frac{2 \cdot k}{\bar{\omega}_e \cdot i_c}\right)^2} = \frac{0.5 \cdot K_i}{\left(\frac{\bar{\omega}_e \cdot i_c}{2}\right)^2 - k^2}. \qquad (24)$$

The cyclic dynamic efficiency coefficient of the motor-transmission system of the vehicle.

$$\eta_{tr}^{din} = 1 - \psi_{tr}^{din} = 1 - \frac{0.5 \cdot K_i}{1 - \left(\frac{2 \cdot k}{\bar{\omega}_e \cdot i_c}\right)^2}. \qquad (25)$$

From equation (25), it can be seen that under the condition

$$K_i = 2 \cdot \left[1 - \left(\frac{2 \cdot k}{\bar{\omega}_e \cdot i_c}\right)^2\right]. \qquad (26)$$

the value $\eta_{tr}^{din} = 0$.

Equation (7), taking into account equation (8), will take the following form

$$\eta_{tr}^{u} = 1 - \frac{\bar{M}_i \cdot \eta_{mdv} \cdot K_i \cdot \left(1 - \frac{K_i}{4}\right)}{\pi \cdot J_p \cdot \bar{\omega}_e^2 \cdot i_c \cdot \left(\left(\frac{2 \cdot k}{\bar{\omega}_e \cdot i_c}\right)^2 - 1\right)}. \qquad (27)$$

By substituting equations (25) and (27) into equation (12), we will obtain

$$\left(\eta_{tr}\right)_c = \eta_{tr}^{st} + \eta_{tr}^{v} - \frac{K_i}{\left|\left(\frac{2 \cdot k}{\bar{\omega}_e \cdot i_c}\right)^2 - 1\right|} \times \times \left(\frac{\bar{M}_i \cdot \eta_{mdv} \cdot \left(1 - \frac{K_i}{4}\right)}{\pi \cdot J_p \cdot \bar{\omega}_e^2 \cdot i_c} + 0.5 \right). \quad (28)$$

In equation (28), $\eta_{mdv} = 1$ can be assumed. In this case, the dissipative losses in the engine can be attributed to the dissipative losses in the transmission.

From expressions (27) and (28), it is evident that when $K_i = 1$, the elastic cyclic efficiency coefficient of the motor-transmission system equals one, and the loss coefficient is zero. The specified irregularity coefficient $K_i = 4$ is close to the irregularity coefficient of a 4-cylinder engine, where it equals 4.7. Thus, for 4-cylinder engines, energy losses due to overcoming elastic deformations of transmission elements are nearly zero. This is likely one of the reasons why 4-cylinder engines have become widely adopted in automotive and tractor manufacturing.

Analysis of equation (24) indicates that the cyclic dynamic coefficient of mechanical losses in the transmission is proportional to the irregularity coefficient of torque and inversely proportional to the difference of the squares of the circular frequencies of torque oscillations and the natural (free) oscillations of the transmission input shaft. When these frequencies coincide, resonance occurs, leading to cyclic coefficients of elastic and dynamic mechanical losses in the transmission. For $K_i = 0$ (characteristic when using electric motors), the cyclic coefficients of dynamic and elastic losses are zero (see equation (28)).

Conclusion

The determined cyclic mechanical efficiency of the motor-transmission systems in wheeled vehicles accounts for the energy losses during the acceleration of moving masses, considering the torque irregularity characteristic of internal combustion engines (ICEs).

The cyclic dynamic coefficient of mechanical losses in the transmission is proportional to the torque irregularity coefficient and inversely proportional to the difference between the squares of the circular frequencies of torque oscillations and the natural (free) oscillations of the transmission input shaft. When these frequencies coincide, resonance occurs, leading to a sharp increase in the cyclic coefficients of elastic and dynamic losses.

For an ICE torque irregularity coefficient of $K_i = 4$, the cyclic dynamic coefficient of mechanical losses is close to zero. The value $K_i = 4$ closely aligns with $K_i = 4.7$, characteristic of a 4-cylinder engine, which is considered one of the reasons for the popularity of such ICE configurations.

When the torque irregularity coefficient is zero, as is typical for electric motors, the cyclic coefficients of dynamic and elastic losses are also zero (see equation (28)).

Conflict of interests

The authors declare that there is no conflict of interests regarding the publication of this paper.

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Цикловий механічний коефіцієнт корисної дії моторно-трансмісійних установок транспортно-тягових машин

Анотація. Проблема. *При визначенні коефіцієнта корисної дії моторно-трансмісійних установок автомобіля та тракторів необхідно враховувати не тільки втрати на в'язке та сухе тертя, але також і втрати, зумовлені циркуляцією у трансмісії потенційної та кінетичної енергії, які є результатом використання в моторно-трансмісійних установках двигунів внутрішнього згоряння. На даний час у відомих дослідженнях не враховані динамічні циклові втрати енергії у трансмісії, що не дозволяє точно оцінити цикловий коефіцієнт корисної дії.* ***Мета.*** *Мета полягає в тому, щоб визначити цикловий коефіцієнт корисної дії моторно-трансмісійних установок із двигунами внутрішнього згоряння з урахуванням циклового динамічного коефіцієнта корисної дії трансмісії автомобіля з двигуном внутрішнього згоряння.* ***Методологія.*** *Прийняті в роботі підходи до вирішення поставленої мети базуються на теоретичних основах визначення динамічного коефіцієнта корисної дії, балансу робіт у моторно-трансмісійній установці автомобіля за один період коливань індикаторного крутного моменту, кутових швидкостей колінчастого валу за один період коливань крутного моменту.* ***Результати.*** *Визначено цикловий механічний коефіцієнт корисної дії моторно-трансмісійних установок колісних машин, що враховує втрати на розгін мас, що рухаються, з урахуванням нерівно-мірності крутного моменту, властивої двигунам внутрішнього згоряння. Визначено залежність в якій циклові коефіцієнти динамічних і пружних втрат рівні нулю, при коефіцієнті нерівномірності моменту, що* *крутить, рівному нулю, що характерно для електродвигунів.* ***Оригінальність.*** *Результати проведеної роботи дають загальне уявлення про пропорційність циклового динамічного коефіцієнта механічних втрат у трансмісії до коефіцієнту нерівномірності крутного моменту і різниці квадратів кругових частот коливань крутного моменту та власних (вільних) коливань вхідного валу трансмісії.* ***Практичне значення.*** *Отримані результати можуть бути рекомендовані при визначені збігу частот що приводить до резонансу, у якому різко зростають циклові коефіцієнти пружних і динамічних втрат.*

Ключові слова: *моторно-трансмісійна установка, двигун внутрішнього згорання, маса, динамічний ККД, енергія, тертя, колінчатий вал, коливання, крутний момент.*

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