Modeling the vehicle operational efficiency in the "Vehicle - Infrastructure" system

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Abstract. The article is devoted to the study of vehicle operational efficiency in certain infrastructural environment conditions. The vehicle operational efficiency is evaluated according to dynamic, economic and environmental criteria using a mathematical model of the "Vehicle-Infrastructure" system. The mathematical model describes the main processes in the studied system: the conversion of input energy into mechanical energy of propulsion system, mechanical energy transmission to the vehicle wheels, the conversion of wheels rotation into vehicle translational motion, change of vehicle motion modes and directions according to a given motion law in certain infrastructural environment conditions. The presented structure of the mathematical model contains generalized dependences between the main parameters of the system processes and feedbacks implementing the target functions of thermal control of individual propulsion subsystems and vehicle modes to achieve the desired values of energy consumption and emissions per transport work unit. The results of vehicle operational efficiency evaluation using verified model in the implementation of individual algorithms for thermal development control of propulsion subsystems (engine and catalytic converter) in the heating mode are given.

1. Introduction

The efficiency of a vehicle use in operating conditions significantly depends on the degree of control processes optimization [1-3].

Management of vehicle operational efficiency involves a controlled change of the input parameters of the main processes occurring during the vehicle operation in accordance with the operating conditions. It is based on embedded target functions into control devices of input parameters that take into account certain efficiency criteria.

A system approach is used to comprehensively take into account all the factors influencing the management of the vehicle operational efficiency [4]. In accordance with this, the processes of vehicle management are defined depending on the level of implementation of the main morphological features of the functional elements of the "Vehicle - Infrastructure" system.

The "Vehicle - Infrastructure" system as the object of study describes the processes of vehicles motion with modern (heat, electrical, hybrid) propulsion systems in a given infrastructure environment. The initial results of the processes are vehicle performance, energy consumption and...
emissions per unit of mileage or transport work. The system determines the value of targets (restrictions) of operational efficiency and process control parameters (feedbacks) based on current information of the internal monitoring system of the technical condition and modes of the vehicle, as well as information of the external infrastructure environment that determines the actual operating conditions (road, traffic, meteorological etc.) [5, 6].

In previous studies, the authors have developed means and systems for monitoring the efficiency of vehicles and their propulsion systems in operating conditions [7, 8], presented the overall functional structure and distribution of the main morphological features of the "Vehicle - Infrastructure" system [9], which combines four main processes: conversion of output energy into mechanical (propulsion system); transfer of mechanical energy to the vehicle wheels (transmission); conversion of rotational movement of wheels into translational movement of the vehicle (chassis); change of modes and directions of motion according to the set motion law as a result of interaction of vehicle with a transport infrastructure (infrastructural environment).

The purpose of this study is to develop a mathematical model of the "Vehicle - Infrastructure" system, which will allow its practical use in on-board and server systems that provide high efficiency of vehicle operation.

2. Enlarged structure of the mathematical model of the "Vehicle - Infrastructure" system

The model describes the main processes of the system (Fig. 1). At level A, generalized mathematical dependences describing the processes of propulsion system are presented. The input data are the parameters of the environment \( p_0, T_0 \), the functional dependence of the energy supplied to the propulsion system on the speed of the output shaft and the degree of load \( E_{in} = f(n_e, E_{load}) \), the block of vehicle constant parameters \( A = \{u_0, C_s, F, r_w, \eta_{fr} \} \), total mass \( M_t \) and mass of the equipped vehicle \( M_0 \), initial temperatures of engine coolant \( t_c \) and catalytic converter \( t_k \). The working process of the propulsion system is described as a functional dependence of torque on the speed, load and thermal mode, the amount of energy supplied, energy consumption to ensure the operation of the system to maintain the thermal state of the engine and catalytic converter \( M_t(M) = f(n_e, E_{load}, E_{in}, t_c, t_k) \). The rotation speed of the output shaft is represented by the dependence \( n_e = f(V, u, t_0) \). At the output of level A, the specific energy consumption and emissions by the propulsion system are determined using the dependences \( g_e = f(E_{in}, M_t(M), n_e) \), \( g_i = f(G, M_t(M), n_e, E(t_k)) \), which describe these indicators as a function of the absolute energy consumption and emissions, torque and speed of the output shaft, the catalytic converter efficiency. The final values of the engine coolant temperatures \( t_c \) and the catalytic converter \( t_k \) are described depending on the time functions of change of these temperatures, the intensity of the thermal state system \( \{t_c, t_k\} = f(t_c(t), t_k(t), \Delta t_c(t), \Delta t_k(t), t_{TA}) \). Based on the obtained values of specific energy consumption and emissions, intensities of engine coolant and catalytic converter temperatures, the target function of intensity control of the thermal state system is formed, which is presented at level B as \( n_{TA} = f(g_e, g_i, \frac{dt_c}{dt}, \frac{dt_k}{dt}) \).

At level C, the process of vehicle transmission is described. The input parameters are the values of torque and speed of the output shaft of the engine \( n_e, M_e \). The working process is described by the dependences of the torque and speed of the driving wheels of the vehicle on the input parameters of the level C, the gear ratios and efficiency of transmission \( n_w = f(n_e, u_i, u_0) \), \( M_w = f(M_e, n_w, u_i, u_0) \). The gear ratio is determined at level D using the target function, which describes the minimum required value of the gear ratio depending on environmental parameters, constant parameters of the vehicle, speed, road resistance coefficient, torque and speed of the engine output shaft in the form \( u_i^{min} = f(M_t, p_0, T_0, A, V, \eta, M_e^{max}(n_e^{min})) \).
Level E describes the process of the chassis of the vehicle during acceleration to a given speed. The input parameters are the values of torque and speed of the drive wheels \( n_w, M_w \). Depending on these parameters, as well as the dynamic radius of the wheel, vehicle weight, road resistance coefficient, gear ratio, constant vehicle parameters, environmental parameters, speed, the current speed and acceleration of the vehicle are determined \( V_a = f(n_w, r_w), \frac{dV_a}{dt} = f(M_w, M_A, \psi, u_i, A_i, p_0, T_0, V_a) \). The control of the vehicle acceleration process is described at level F using the target function \( \{E_m, u_i\} = f(dV_a/dt_{\text{max}}, g_{\text{min}}^e, g_{\text{min}}^u) \), which provides control of the supplied energy in the propulsion system and the gear ratio with maximum acceleration and minimum emissions and energy consumption.

Level G describes the process of interaction of the vehicle with the infrastructure environment. The input data are the speed and acceleration of the vehicle, the input force of the brake system, the position of the steering actuator. The interaction process is described by the target function of the value of the maximum possible speed of the vehicle in these infrastructural conditions depending on air temperature, road resistance coefficient, GPS coordinates, minimum energy consumption, emissions, travel time as \( V_{a_{\text{max}}} = f(T_0, \psi, GPS, g_{\text{min}}^e, g_{\text{min}}^u, t_{\text{min}}) \). The initial data are the time dependences of the speed \( V(t) \), the distance traveled \( S(t) \), energy consumption \( g_e = f(E_m, S(t)) \), emissions \( g_i = f(G_i, S(t)) \). Control of the propulsion system, transmission, brake system and steering is described at level H. Control of the propulsion system and transmission is performed under the conditions \( E_{\text{in}} = f(M_e = M_{\text{res}}), u_{\text{in}}, i \geq u_{\text{in}} \). The target control function of the brake system is presented depending on the degree of energy recovery of the brake system, the deceleration of the vehicle in the form \( P_{\text{in}} = f(E_{\text{in}} = 0, E_{\text{rec}}^\text{max}, \frac{dV_a}{dt_{\text{min}}}) \). The steering function is represented as \( \psi_S(t) = f(GPS) \).

3. Modeling of processes of the subsystem "Propulsion System" (level A)

The input parameters during the simulation of the propulsion system processes are the rotation speed of the output shaft \( n_e \) and the position of the control lever of the propulsion system \( \theta_{\text{contr}} \). As a function of these two parameters, the engine load parameter is determined \( E_{\text{load}} = f(n_e, \theta_{\text{contr}}) \). Depending on the type of propulsion system, the position of the control lever \( \theta_{\text{contr}} \) for example, the degree of opening of the throttle valve is determined (gasoline and gas internal combustion engines with spark ignition). The absolute pressure in the intake manifold \( p_k \) can be determined as a load parameter in a spark ignition internal combustion engine.
The amount of energy consumed by the engine depending on the speed and load mode $E_{in} = f(n_e, E_{load})$ is based on the dependences of air flow $G_{AIR}$ in this mode, the excess air coefficient $\lambda$, which is determined by the engine control system and affects the amount of fuel consumed $G_{fuel}$, and fuel combustion heat $h_l$.

Propulsion system energy consumption, MJ/h:

$$E_{in} = \frac{G_{AIR} \cdot h_l}{\lambda \cdot l_0},$$  \hspace{1cm} (1)

where $l_0$ is the theoretically required amount of air for combustion of 1 kg of fuel, kg/kg.

However, the energy consumption will increase proportionally with increasing speed, which will lead to a large error when approximating the data. Therefore, to determine the dependence of the indicated torque $M_i = f(n_e, E_{in})$ on the speed of the engine and the energy consumed with the fuel, it is advisable to use the fuel consumption per cycle $g_c$.

Fuel consumption per cycle by the engine, mg/cycle:

$$g_c = \frac{G_{AIR} \cdot 10^6}{30 \cdot \lambda \cdot l_0 \cdot i_c \cdot n_e},$$  \hspace{1cm} (2)

where $i_c$ is the number of engine cylinders.

The moment of mechanical losses depends on the speed, load and temperature mode of the engine. When using the means of thermal preparation of the engine, the losses for the drive of the additional pump that circulates the working fluid of the thermal accumulator (or working fluid from the engine lubrication or cooling system) to transfer stored heat to engine systems are also added to total mechanical losses. In the general case, the dependence of the mechanical losses of the engine has the form $M_M = f(n_e, E_{load}, l_c, n_{TA})$.

In case of use of the electric drive of the additional pump of the thermal accumulator of cooling liquid, the power taken away from the engine, kW:

$$N_{TA} = \frac{V_f \cdot P_{cool} \cdot 10^3}{\eta_p \cdot \eta_g},$$  \hspace{1cm} (3)

where $V_f$ is the circulating fluid flow rate, m$^3$/s;

$p_{cool}$ is coolant pressure during its pumping through the thermal accumulator, $p_{cool} = 0.05$ MPa;

$\eta_p$ is the pump mechanical efficiency, $\eta_p = 0.9$;

$\eta_g$ is the automotive generator efficiency, $\eta_g = 0.7$.

The most common vane pumps. In this case, the circulation flow of coolant through the thermal accumulator will depend on the working volume of the pump, which determines the volume of liquid supplied by the pump per revolution and the rotation speed of the drive shaft $n_{TA}$, m$^3$/s:

$$V_f = V_{rf} \cdot \frac{\pi \cdot n_{TA}}{30},$$  \hspace{1cm} (4)

where $V_{rf}$ is the volumetric flow of the pump, m$^3$/rev. For engines in M1 cars, a pump with a blade diameter of 100 mm will create a sufficient volumetric flow. In this case, the volumetric flow will be $2.4 \cdot 10^{-5}$ m$^3$/rev.

The moment of losses on a crankshaft created by the additional pump, Nm:
The effective torque is determined from the known values of the indicated torque and the moment of mechanical losses of the engine

\[ M_{TA} = \frac{N_{TA} \cdot 9550}{n_e} = V_{rf} \cdot \frac{\pi \cdot n_{TA} \cdot p_{cool} \cdot 10^3 \cdot 9550}{\eta_p \cdot \eta_g \cdot n_e} = 1.9 \cdot \frac{n_{TA}}{n_e}. \]  \tag{5}

The circulation flow rate and the moment of mechanical losses generated by the pump will be determined by the rotation speed of the drive shaft of the pump of this design. On the other hand, the circulating fluid flow rate will depend on the required intensity of the coolant temperature rise due to the use of the thermal accumulator in terms of ensuring the minimum specific energy consumption and emissions.

In turn, the value of the coolant temperature during the heating period will depend on the intensity of heat loss during the operating cycle in the engine cylinders, which depend on the operation mode of the engine, and the intensity of heat supply from the thermal accumulator. In the general case, the value of the coolant temperature during the heating period is described by the dependence, K [10]:

\[ t_e = T_0 + \int_0^{\tau_H} \frac{(q_T + q_{TA})}{m_{cool} \cdot C_{cool}} d\tau, \]  \tag{6}

where \( T_0 \) is the ambient temperature (initial coolant temperature), K; \( \tau_H \) is engine heating time to operating temperature 90 °C (363 K); \( q_T \) is heat flow due to heat loss of the working fluid in the cylinder into the coolant, W; \( q_{TA} \) is heat flow due to heat transfer from the thermal accumulator to the coolant, W; \( m_{cool} \) is mass of coolant in the system, kg, \( m_{cool} = 8 \) kg; \( C_{cool} \) is heat capacity of the coolant, J/(kg K), \( C_{cool} = 4187 \) J/(kg K).

Heat flow due to heat loss of the working fluid in the cylinder into the coolant, W:

\[ q_T = \frac{q_c \cdot n_e \cdot i_c}{120}, \]  \tag{7}

where \( q_c \) is the amount of heat lost from the working fluid in the cylinder for one working cycle, J.

Heat flow \( q_{TA} \) due to heat transfer from the thermal accumulator to the coolant is determined by the dependence, W:

\[ q_{TA} = \alpha_{TA} \cdot F_{TA} \cdot (T_{TA} - t_e), \]  \tag{8}

where \( \alpha_{TA} \) is the heat transfer coefficient from the walls of the heat exchanger of the thermal accumulator to the coolant, W/(m²K);

\( F_{TA} \) is surface area of the walls of the heat exchanger of the thermal accumulator in contact with the coolant, m²;

\( T_{TA} \) is the temperature of the walls of the thermal accumulator, which is determined by the temperature of the phase transition of the heat-accumulating material, K.

Hydroquinone \( C_6H_4(OH)_2 \) with a phase transition temperature of 445 K is used as a heat-accumulating material [10].

To determine the heat transfer coefficient, the theory of similarity is used [11], which makes it possible to determine it depending on the physical properties and mode of movement of the coolant.

Concentrations of harmful substances in exhaust gases are determined, first of all, by their dependence on the mode of operation of the engine, its type, the type of fuel used and the composition of the fuel-air mixture [12]. Such concentrations can be determined on the basis of measurements in
untreated exhaust gases by the catalytic converter. Concentrations of the main harmful components are described for each type of the engine and a kind of the used fuel by dependences:

\[ C_i = f(n_e, p_k) \cdot k_{i}^f \cdot \left(1 - \frac{E_i}{100}\right), \]  

(9)

where \( k_{i}^f \) is a coefficient that takes into account the effect of the composition of the fuel-air mixture on the concentration; \( E_i \) is efficiency of conversion of the i-th substance, \( \% \) [10, 13].

Emissions of harmful substances from exhaust gases are determined on the basis of certain concentrations of harmful substances at the outlet of the exhaust system, fuel and air consumption and molar mass of harmful substances \( G_i = f(C_i, G_{\text{fuel}}, G_{\text{AIR}}, \mu_i) \) [12].

4. Modeling of processes of the subsystem "Vehicle" (level C, E)

The operation of this subsystem and its model is based on the principle of ensuring maximum efficiency of achieving a given speed in these road conditions, which takes into account the maximum possible acceleration with the minimum possible fuel consumption and emissions.

The model describes the main stages of achieving a given speed value:
- acceleration of the engine in the idling mode;
- starting from the place and acceleration before the first gear change;
- gear shift;
- acceleration to the set speed.

The last two stages are periodically repeated in the process of reaching a given speed according to its value.

The input data are the parameters of the propulsion system: rotation speed \( n_e \), the position of the control lever \( \varphi_{\text{contr}} \), the moments of masses inertia of the engine \( J_e \), the driven part of the clutch \( J_{cl} \) and wheels \( J_w \), the effective torque of the engine \( M_e \); time parameters: current time \( t \) and estimated time interval \( dt \); vehicle speed parameters: initial speed \( V_a \), set maximum speed \( V_{a\text{max}} \) (as the value of the maximum possible speed in these infrastructure conditions), minimum speed on the i-th gear \( V_{a\text{min}} \); transmission parameters: gear ratios \( u_i \), final drive ratio \( u_0 \), transmission efficiency \( \eta_{tr} \), clutch friction moment \( M_{cl} \), maximum clutch friction moment \( M_{cl\text{max}} \); parameters of the chassis of the vehicle: dynamic and rolling radius \( r_w \) of the wheel; infrastructure parameters: the current coefficient of road resistance \( \psi \) and the moment of resistance of the vehicle is reduced to the crankshaft \( M_{\text{res}} \); vehicle mass \( M_a \).

The output parameters are the speed \( V_a \) and acceleration \( \frac{dv_a}{dt} \) of the vehicle.

The vehicle motion begins with the process of accelerating the engine at idle. The rotation speed of the crankshaft in this mode [14] is described by equation, rpm:

\[ n_e = n_e^0 + M_e \cdot 30 \cdot \frac{dt}{\pi \cdot J_e}, \]  

(10)

where \( n_e^0 \) is the value of the engine shaft speed at the beginning of the estimated time interval \( dt \), rpm;
\( dt \) – estimated time interval, s.

Determined as a result of control actions the position of the control lever of propulsion system, deg:

\[ \varphi_{\text{contr}} = \varphi_{\text{contr}}^0 + V_{\varphi\text{contr}} \cdot dt, \]  

(11)
where $\varphi_{\text{contr}}^0$ is position of the control lever of propulsion system at the beginning of the estimated time interval $dt$, deg;

$V_{\varphi_{\text{contr}}}$ is speed of movement of the control lever of propulsion system, deg/s.

Based on this, the current value of the effective torque of the engine is determined.

Next is the transition to the next stage of starting and acceleration to the first gear change.

At this stage, the minimum required value of the gear ratio $u_i^{\text{min}}$ to start in these conditions depending on the maximum possible engine torque is determined.

The actual gear ratio under the term $u_i > u_i^{\text{min}}$ is selected.

Next, the clutch towing condition is determined as the absolute difference between the current engine shaft speed and the speed of the driven part of the clutch at a given gear and vehicle speed

$$n_e = \frac{30 \cdot u_i \cdot u_0 \cdot V_a}{3.6 \pi r_w} > 100.$$  

Determination of the engine shaft speed and acceleration of the vehicle in the case of a towing clutch is carried out depending on the current value of the friction moment of the clutch $M_{cl}$.

In the case where $M_{cl} < M_e$ the rotation speed of the engine shaft with towing clutch is defined as, rpm:

$$n_e = n_e^0 + \left( M_e - M_{cl} \right) \cdot \frac{30 \cdot dt}{\pi \cdot J_e}. \quad (12)$$

Acceleration of a vehicle with a towing clutch, m/s$^2$, is defined as [15]:

$$\frac{dV_a}{dt} = \frac{\left( M_{cl} - M_{res} \right) \cdot r_w \cdot u_i \cdot u_0 \cdot \eta_{tr}}{M_a \cdot r_w^2 + \Sigma J_w}. \quad (13)$$

In the case where $M_{cl} > M_e$ the vehicle acceleration will be determined as for the case of locked clutch [15], m/s$^2$:

$$\frac{dV_a}{dt} = \frac{\left( M_e - M_{res} \right) \cdot r_w}{J_e \cdot u_i \cdot u_0 + \frac{M_a \cdot r_w^2 + \Sigma J_w}{u_i \cdot u_0 \cdot \eta_{tr}}}. \quad (14)$$

When the process of towing the clutch stops, the engine speed corresponds to the speed with the locked clutch [15], rpm:

$$n_e = \frac{30 \cdot u_i \cdot u_0 \cdot V_a}{\pi \cdot r_w}, \quad (15)$$

where $V_a$ is the speed of the vehicle, m/s.

In this case, the value of torque on the engine shaft is determined depending on the position of the control lever of propulsion system, the optimal value of which describes the target function $\varphi_{\text{contr}}^{\text{opt}} = f \left( \frac{dV_a}{dt}^{\text{max}}, g_e^{\text{min}}, g_i^{\text{min}} \right)$.

In the case of the vehicle motion with the disengaged clutch, the acceleration is defined as, m/s$^2$:

$$\frac{dV_a}{dt} = -\frac{M_{res} \cdot u_i \cdot u_0 \cdot \eta_{tr} \cdot r_w}{M_a \cdot r_w^2 + \Sigma J_w}. \quad (16)$$
Vehicle speed, km/h, depending on the value of acceleration is defined as:

\[ V_a = V_{a0} + 3.6 \cdot \frac{dV_a}{dt} \cdot dt, \quad (17) \]

where \( V_{a0} \) is vehicle speed at the beginning of the estimated time interval \( dt \), km/h.

To determine the possibility of engaging a higher gear, two conditions are tested: achieving the minimum speed at \((i+1)\)-th gear \( V_a > V_{a_{\text{min}}u_{i+1}} \) and the sufficiency of engine torque when engaging \((i+1)\)th gear to overcome the moment of resistance on the \((i+1)\)th gear \( M_e(V_a, u_{i+1}, \varphi_{\text{contr max}}) > M_{\text{res}}(V_a, u_{i+1}) \). If these conditions are met, the gear is switched.

5. Modeling of processes of the subsystem "Infrastructure" (level G)

The process of interaction of the vehicle with the infrastructure is characterized by the moment of resistance of external forces reduced to the engine shaft in these infrastructural conditions, which are determined by the dependence \[15\], N\cdot m:

\[ M_{\text{res}} = \frac{M_a \cdot g \cdot \psi_{\text{GPS}} + C_x \cdot \rho_0 \cdot F_a \cdot \frac{V_a^2}{2}}{u_i \cdot u_0 \cdot \eta_r}, \quad (18) \]

where \( C_x \) is the coefficient of aerodynamic drag of the vehicle;
\( \rho_0 \) is ambient air density, kg/m\(^3\);
\( F_a \) is cross-sectional area of the vehicle, m\(^2\);
\( V_a \) is vehicle speed, m/s.

The coefficient of road resistance \( \psi_{\text{GPS}} \) takes into account the rolling resistance and the lifting resistance of the road, which depends on the angle of longitudinal slope of the road \( \alpha \). The angle \( \alpha \) is determined based on the analysis of GPS coordinates of the current position of the vehicle using specialized software \([16]\).

Thus, depending on the specific infrastructural conditions, the appropriate speed of the vehicle can be determined, which maximizes the value of the corresponding target function.

To verify the adequacy of modeling indicators, a comparison of KIA CEED fuel consumption dependences during modeling of its motion in the UNECE № 83-05 driving cycle was performed.

As the results of the adequacy test show, the total error of mathematical modeling, determined by the value of the total fuel consumption per cycle is 3.4% (686.3 g of fuel measured experimentally; 709.8 g of fuel is modeled), which indicates a sufficiently high adequacy of the mathematical model.

6. The research results

The study, performed using a mathematical model allowed to establish appropriate limits for changing the speed of the circulating pump with certain design parameters for the system of accelerated heating of the engine of the vehicle category M1.

The results of the study are presented in Fig. 2. The operating time of the engine at different speeds of the circulation pump is taken to be the same, which corresponds to the heating time \( \tau_H \) of the engine to operating temperature without a thermal accumulator.

As can be seen from the presented results, increasing the speed has a significant impact on reducing CO emissions. However, the impact on other engine performance is ambiguous. In general, the effect of significant improvement of all environmental and fuel-economic indicators can be achieved only at minimum idle speeds (Fig. 2 a, c, e). In this mode, the optimal speed of the pump shaft is \( n_{\text{TA opt}} = 3700 \text{ rpm} \). For other engine modes, the optimum speed of the circulation pump shaft is determined by the minimum fuel consumption during the heating period.
The study of the influence of the vehicle control algorithm on energy consumption and emissions of harmful substances was carried out by modeling these indicators during the vehicle motion on the experimental route. Two variants of the control system are investigated:

- basic control (the mode of operation of the propulsion system and transmission of the vehicle is controlled by the driver independently);
- optimized control of the vehicle and the thermal state of its propulsion system (thermal and load mode of the propulsion system is determined by the control system as a result of the analysis of the current traffic conditions of the vehicle).

The results of the study are shown in Fig. 3.

**Figure 2.** Influence of circulating pump speed on emissions of CO and CmHn (a, b), NOx and CO2 (c, d) and energy consumption (e, f) during engine heating at idle: a, c – ne = 1000 rpm; b, d, f – ne = 1500 rpm
As can be seen from the presented data, the optimization of vehicle management allows to improve the time frame of the output of the vehicle performance to the desired and/or limited indicators. Thus, the optimal control of the thermal state of the vehicle allows to reduce the time to reach the value of limited mileage CO emissions by 25% (from 1000 to 750 s); CnHm emissions by 21% (from 475 to 375 s); NOx emissions of 30% (up to 1150 s). In this case, the desired value of CO2 emissions is achieved in the range from 650 to 1100 s.

In general, optimizing the management of the thermal state of the vehicle propulsion system can improve efficiency by up to 8.7% (reduce energy consumption from 37 to 33.75 MJ).

7. Conclusions
To study the impact of different control variants on the operational efficiency of vehicles in accordance with the dynamic, economic and environmental criteria, an enlarged structure of the mathematical model was developed. The model describes the main processes of the system, their input values and outputs, feedbacks using functional dependencies and target functions that take into account the design parameters of vehicles, environmental and road conditions of their operation, technical condition of vehicle elements and target values of efficiency criteria depending from which parameters of management of the basic processes are defined and the speed motion law of vehicles on a route is realized.

Within the framework of the enlarged structure of the mathematical model of the “Vehicle-Infrastructure” system, algorithms for modeling its individual subsystems, in particular, propulsion system, vehicle and infrastructure, have been developed. The total error of mathematical modeling, determined by the total vehicle fuel consumption for the driving cycle is 3.4%.
The optimal speed modes of operation of the circulation pump of the thermal treatment system in accordance with the speed and load modes of operation of the engine are set. In general, the effect of significant improvement of all environmental and fuel-economic indicators can be achieved only at minimum idle speeds. As the crankshaft speed and engine load increase, the expediency of using the heat treatment system decreases significantly.

The evaluation of optimal vehicle management algorithms to reduce fuel consumption and emissions of harmful substances on the experimental route was performed. Optimization of vehicle thermal management can significantly improve the basic economic and environmental performance, in particular, reduce the time to achieve limited values of harmful emissions by 21-30%, achieve the desired value of CO\(_2\) emissions and improve energy efficiency by 8.7%.

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