



Improvement of Start Mode and Post-Start Heating of Diesel Engines

G.M. Krokhta, P.S. Vagaytsev, N.A. Usatykh, A.A. Galynskiy, E.N. Khomchenko
Novosibirsk State Agrarian University, Novosibirsk, Russia

ABSTRACT

Analysis of published data demonstrates that at present, the most common method of efficiency estimation of thermal machines is the method of thermal balance, which is characterized by certain drawbacks.

This work is aimed at development of partial procedure of estimation of quantity and quality of exergy fluxes in air and fuel supply systems and their influence on engine starting characteristics and incomplete fuel combustion during post-start heating.

During start, ignition in diesel cylinder is determined by air temperature at the end of compression (T_c) and heat consumptions for heating, evaporation, and overheating of fuel vapors. It is obvious that the exergy of fuel injected into cylinder in terms of quantity is significantly lower than that of compressed air, however, the fuel exergy mainly determines heat and mass transfer in the area of front of fuel sprays and, as a consequence, diesel starting properties and completeness of combustion during post-start period.

Analysis of exergy balance for fuel supply system demonstrates that atomization during start and post-start heating at negative ambient temperatures can be controlled with minimum consumptions of energy by fuel preheating in nozzle immediately before its injection into combustion chamber. Partial procedure has been developed based on exergy method allowing to estimate quantitatively the exergy fluxes in air and fuel supply systems and, thus, the rate of activity of fuel spray and energy supply of air charge during start and post-start heating of tractor engine.

Key words: Diesel engine, exergy, fuel supply, air supply, atomization, start, post-start heating.

1. INTRODUCTION

When machines operate at low ambient temperatures, peculiar attention should be paid to cold start of diesel engines, when the temperatures of cylinder head, coolant, oil, engine parts, battery are equal to ambient temperature.

Start of diesel engine at low ambient temperature without preheating is related with significant difficulties caused by decreased battery capacity, poor fuel pumping, temperature drop at the end of compression stroke, poor fuel spraying, increased oil viscosity.

Numerous studies are devoted to decrease in time consumption for engine preheating and improvement of its

operation efficiency during start and post-start heating at low ambient temperatures.

The researchers in [1-4] published results of theoretical and experimental studies of the influence of temperature of fuel and ambient air on start properties of diesel engines, various approaches were proposed to improve the start properties and engine efficiency during operation in post-start heating by fuel and air heating prior to injection into combustion chamber.

Thus, for instance Zhurba [1] developed the design and performed comparative tests of several heater types located on sprayer nozzle and in nozzle fuel channel allowing to improve atomization and, as a consequence, engine start properties. Herewith, processes in cylinders were estimated by efficiency determination.

However, in recent years the exergy analysis of thermodynamic systems becomes more and more popular. Thus, in [5, 6] the exergy method is applied for analysis of diesel engines operating on biofuel with various contents of components, and in [7-9] this approach is used for analysis of fuel and air supply.

On the basis of published data, the hypothesis was formulated about possible development of partial procedure of exergy analysis, which allowed to estimate the rate of activity of fuel spray and energy supply of air charge during start and post-start heating of tractor engine.

2. METHODS

Thermodynamic analysis of power systems allows to estimate the degree of perfection of working process in terms of heat transformations, to determine possible financial expenses, and to outline the ways to save energy resources consumed by a given system.

The simplest method to estimate the degree of perfection of thermodynamic systems is efficiency determination. For instance, concerning tractor engines, determination of effective efficiency (η_{eff}) is not difficult, though, estimation of its constituents (η_i and η_{mech}) is related with certain assumptions and, thus, is not highly accurate. In some cases, comparison of total efficiency of different designs of power assemblies is unacceptable. For instance, direct comparison of efficiency of thermal and electric engines is incorrect.

The most common estimation of efficiency of thermal machines is the method of thermal balance. Such approach can be exemplified by internal combustion engine (ICE) thermal balance, which accounts for amount of energy

supplied to the engine together with fuel as well as heat amount equivalent to mechanical energy taken from engine crankshaft. The difference between these two types of energy is the measure of perfection of thermal process.

Entropy and exergy methods are more perfect based both on the first and the second laws of thermodynamics. The entropy method is characterized by amount of effective work and entropy increment. Loss of productivity of working medium (ΔL) in thermodynamic system due to increase in entropy is expressed by the Guy–Stodola equation [10, 11]:

$$\Delta L = T_{amb} \cdot \Delta S, \quad (1)$$

where ΔS is the entropy increment of working medium, J/kg·K; T_{amb} is the ambient temperature, K.

A peculiar feature of the entropy method is that it operates with unified system of coefficients of productivity loss, which equal to the ratio of ΔL to total non-entropy portion of energy added to assembly [10]. The main difficulty of this method during computations is the absence of universal determination mechanism of ΔS , hence, ΔS should be determined separately for each actual process [10].

The exergy method is based on the first and the second law of thermodynamics, which allows to consider for quantitative and qualitative aspects of energy transformations and to estimate precisely perfection of any existing heat and power plants.

Exergy is the portion of energy which can perform maximum work during system transfer from preset state into equilibrium with all components of ambient environment. Another portion of energy is anergy which cannot be used for process needs according to the second law of thermodynamics since it is in equilibrium with the ambient parameters. Therefore, exergy depends directly on ambient parameters, which is the advantage of exergy method. This method allows to estimate actual degree of perfection of processes running in ICE [5-7]

This work is aimed at development of partial estimation procedure of quantity and quality of energy fluxes in air and fuel supply systems, as well as their influence on engine start properties and incomplete fuel combustion during post-start heating.

3. RESULTS AND DISCUSSION

Depending on the exergy fraction in total amount of energy, the energy resources can be subdivided into three types:

- Energy resources completely comprised of anergy. As is known, anergy cannot be used. This type can include, for instance, internal energy of ambient environment and work against ambient pressure.
- Energy resources completely comprised of exergy. They are characterized by zero entropy and, in the case of reversible thermodynamic process, can be unlimitedly converted into other types of energy. The second type includes electric and mechanical energies.
- Restricted conversable energy resources. Exergy for such types of energy is determined by special dependences on the

basis of parameters of ambient environment and energy. This group attracts the highest interest of researchers and includes exergy of heat flux, exergy of substance flow, and exergy of substance in closed space.

Anergy of heat flux A_x can be determined as follows:

$$A_x = Q_x \cdot \left(\frac{T_{amb}}{\bar{T}_{avg}} \right), \quad (2)$$

where Q_x is the energy of heat flux, J; T_{amb} is the ambient temperature, K; \bar{T}_{avg} is the average dynamic temperature of working medium, K.

Upon analysis of air and fuel supply systems, there are exergy fluxes of various quality, they can be estimated only after presentation in one scale.

Exergy of thermodynamic system can be determined by the three major equations [8, 12]:

1) Exergy of substance in closed volume (E_V):

$$E_V = (U_1 - U_{amb}) - T_{amb} \cdot (S_1 - S_{amb}) + P_{amb} \cdot (V_1 - V_{amb}) \quad (3)$$

where U_1 , S_1 , V_1 are the stagnation parameters of working medium (internal energy, entropy and volume of the system under given conditions); U_{amb} , S_{amb} , T_{amb} , V_{amb} , P_{amb} are the internal energy, entropy, temperature, volume and pressure of working medium under equilibrium conditions regarding ambient environment.

2) Exergy of substance flux including thermomechanical (physical) and chemical exergy. The exergy of flux differs from the exergy of substance in volume by amount of work consumed for displacement of substance, it is described as follows:

$$E_S = H_1 - H_{amb} - T_{amb} \cdot (S_1 - S_{amb}), \quad (4)$$

where H_1 , H_{amb} are the enthalpy of system under given conditions and equilibrium with ambient environment, respectively.

3) Flux of exergy transferred in the form of heat:

$$E_x = Q_x \cdot \left(1 - \frac{T_{amb}}{\bar{T}_{avg}} \right), \quad (5)$$

where Q_x is the amount of heat transferred via system boundaries at variable temperature.

Chemical exergy of fuel consumed by engine is determined as follows:

$$E_F = (1.02...1.064)H_U \cdot G_F, \quad (6)$$

where E_F is the chemical exergy of fuel, J/h; 1.02...1.064 is the coefficient depending on the ratio of chemical components in diesel fuel (1.04 is used in the computations); H_U is the lower heat of fuel combustion (42.7 MJ/kg is used in the computations); G_F is the fuel flow rate per hour, kg/h. Aiming at formalization of processes in combustion chamber at the end of compression stroke during engine start, it is required to consider for the following known facts.

Injection of cold fuel into hot air charge results in decrease in its temperature along the spray axis by 150...200°C [13]. The computations demonstrated that upon participation of total

charge in heat exchange, the decrease in its temperature due to heat consumption for heating, evaporation, and overheating of fuel vapors was not more than 20...30°C [14]. It is obvious that the heat and mass exchange processes are the most intensive in the front area of fuel sprays [15]. The injected cold fuel is heated, evaporated, and reaches the self-ignition point due to the energy of compressed air. Herewith, mainly large droplets are formed, their amount during cyclic supply is about 85...90%. The large droplets reach the walls of combustion chamber, where they are deposited forming cold film. Small droplets, comprised of 10...15% of cyclic supply, are decelerated in the combustion chamber, captured by air fluxes, mixed with air and distributed over total space of combustion chamber, forming combustion mixture. However, even these droplets are not always evaporated and combusted. During evaporation the fuel content in air increases, the ratio of reagents can reach stoichiometric value. This period is characterized by activity of pre-flame oxidation processes of hydrocarbon molecules with formation of peroxides. After accumulation of peroxides up to certain critical concentration, their decomposition starts and cold flame is initiated with subsequent transformation into hot flame. However, the initiated exothermal oxidation

can stop at any time due to instability of oxidation caused by violation of balance of the evolved heat and the heat consumed for heating and evaporation of fuel. Stability of oxidation at preset temperature at the end of compression stroke can be controlled by thermal preparation of fuel before its supply to combustion chamber [1, 2]. In this case, the fineness of fuel spraying increases with total heat content of fuel-air mixture at the end of compression stroke, the temperature drop between the air charge and the temperature along the axis of fuel spray decreases upon injection. Therefore, during start the ignition in diesel engine cylinder is determined by the air temperature at the end of compression (T_c) and heat consumption for heating, evaporation, and overheating of vapors of the fuel injected into cylinder. In addition, it is obvious from the aforementioned that the exergy of fuel injected into cylinder in terms of quantity is significantly lower than the exergy of compressed air, nevertheless, it is the exergy of fuel (thermomechanical exergy in this case) that determines heat and mass transfer in the front area of fuel sprays and, as a consequence, the engine start properties and completeness of combustion during post-start period.

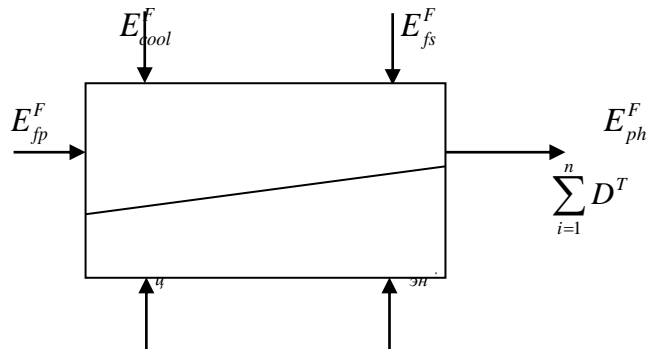


Figure 1: Schematic view of exergy fluxes in diesel fuel supply system.

For further computations, let us present the exergy fluxes in fuel supply system in the following form (Figure 1). With consideration for already mentioned data and the presented flowchart, it is possible to write the equation of exergy balance for fuel supply system:

$$E_{ph}^F = E_{fp}^F + E_{fs}^F + E_{cool}^F + E_{eh}^F + E_{cyc}^F - \sum_{i=1}^n D^F, \quad (7)$$

where E_{ph}^F is the total physical exergy of fuel supplied to the combustion chamber by sprayer, J/cycle; E_{fp}^F is the thermomechanical exergy of fuel at the outlet of high-pressure fuel pump, J/cycle; E_{fs}^F is the exergy of heat obtained by fuel from heated high-pressure fuel supply line, J/cycle; E_{cool}^F is the exergy of heat transferred from coolant to fuel in sprayer, J/cycle; E_{eh}^F is the exergy of heat transferred by electric heater, J/cycle; E_{cyc}^F is the exergy of heat supplied to fuel from working medium, J/cycle; $\sum_{i=1}^n D^F$ are the cumulative losses of exergy in fuel supply system of engine, J/cycle.

Let us consider the heat transfer in stationary regime (pre-start thermomechanical preparation), when ($U_T = 0$).

The thermomechanical exergy of fuel at outlet from high-pressure fuel pump with consideration for Eq. (3) can be expressed as follows:

$$E_{fp}^F = (U_1 - U_{amb}) - T_{amb} \cdot (S_1 - S_{amb}) + \int_0^\phi P_{inj} \cdot (V_1 - V_0) \cdot d\phi, \quad (8)$$

where U_1, S_1 are the internal energy and entropy of fuel temperature increase in the pump head, J/kg and J/kg·K, respectively; P_{inj} is the fuel pressure in sprayer during injection, Pa; ϕ is the angle characterizing duration of fuel injection.

It is obvious that upon start of cold engine, $T_{amb} \cdot (S_1 - S_{amb}) = 0$ and $U_1 - U_{amb} = 0$. Then the work of high-pressure pump can be written as:

$$E_{fp}^F = \int_0^\phi P_{inj} \cdot (V_1 - V_0) \cdot d\phi, \quad (9)$$

The expression under integral in Eq. (9) is the work of

sprayer per cycle. Hence, $E_{fp}^F = L_{fp}$. In its turn, the work of fuel pump per (L_{fp}) for mechanical high-pressure fuel pumps can be written as follows:

$$L_{fp} = L_{elas} + L_{cyc}, \quad (10)$$

where L_{elas} is the work against the spring elasticity, its tension determines the injection pressure per cycle, J/cycle. In the course of further consideration, it is assumed that the fuel pressure upon injection equals to sprayer spring tension (P_{tens}), that is, $P_{inj} = P_{tens} = \text{const}$. Then, after integration of Eq. (9) and a set of simple transformations, we obtain the equation for work L_{fp} :

$$L_{fp} = \frac{\kappa x^2}{2} + P_{inj} \cdot g_{cyc}, \quad (11)$$

where κ is the coefficient of quasi-elastic force of spring, N/m; x is the motion of sprayer needle, m; P_{inj} is the injection pressure, Pa; g_{cyc} is the fuel supply per cycle, m^3/cycle .

Since $G_F = \rho_f \cdot g_{cyc}$, let us write Eq. (8) for cyclic supply with consideration for Eqs. (10) and (11), when one section of fuel pump runs for one cylinder, as follows:

$$E_{fp}^F = [(U_1 - U_{amb}) - T_{amb} \cdot (S_2 - S_{amb})] \cdot \rho_f \cdot g_{cyc} + \left(\frac{\kappa x^2}{2} + P_{inj} \cdot g_{cyc}\right), \quad (12)$$

where ρ_f is the fuel density, kg/m^3 .

The exergy of heat obtained by fuel in fuel supply line per cycle can be computed as follows:

$$E_{fs}^F = [(U_2 - U_{amb}) - T_{amb} \cdot (S_2 - S_{amb})] \cdot \rho_f \cdot g_{cyc}, \quad (13)$$

where U_2, S_2 are the parameters of fuel at input to sprayer, J/kg, J/kg·K.

During start of cold engine and at initial time of its heating $E_{tn}^T = 0$ and only after some time interval the fuel temperature in fuel supply line starts to increase with regard to ambient temperature due to convective heat exchange.

The exergy of heat obtained by fuel in sprayer from engine cooling system during post-start heating can be determined as follows:

$$E_{cool}^F = Q_{cool} \cdot \left(1 - \frac{T_{in}}{T_{spr}^{cool}}\right) \quad (14)$$

where Q_{cool} is the heat supplied to fuel in the sprayer from the cooling system, J/cycle.

In its turn, the heat Q_{cool} is:

$$Q_{cool} = c_V^F \cdot m_F \cdot T_{spr}^{cool}, \quad (15)$$

where c_V^F is the heat capacity of fuel at constant volume, J/kg·K; m_F is the weight of fuel, kg; T_{spr}^{cool} is the temperature of fuel in sprayer box with consideration for heat transferred from engine cooling system, K.

Variation of fuel temperature in the sprayer resulted from heat transfer from coolant (ΔT_{cool}^F):

$$\Delta T_{cool}^F = T_{spr}^{cool} - T_{in}, \quad (16)$$

where T_{in} is the fuel temperature at nozzle inlet, K.

With consideration for Eqs. 15 and 16, Eq. (14) will be as follows:

$$E_{cool}^F = c_V^F \cdot \rho_F \cdot g_{cyc} \cdot T_{spr}^{cool} \cdot \left(1 - \frac{T_{in}}{T_{spr}^{cool}}\right), \quad (17)$$

or

$$E_{cool}^F = c_V^F \cdot \rho_F \cdot g_{cyc} \cdot \Delta T_{cool}^F. \quad (18)$$

Further thermal computations are performed according to the flowchart illustrated in Fig. 2. The electric heater is located in the nozzle fuel channel. During computations it is assumed that the channel is directed along the nozzle axis. During pre-start thermal preparation of fuel in nozzles by means of the electric heaters, a part of heat is consumed for fuel heating in sprayer channel (Q_{fc}^{eh}), another part is transferred by sprayer body to cylinder head and ambient environment.

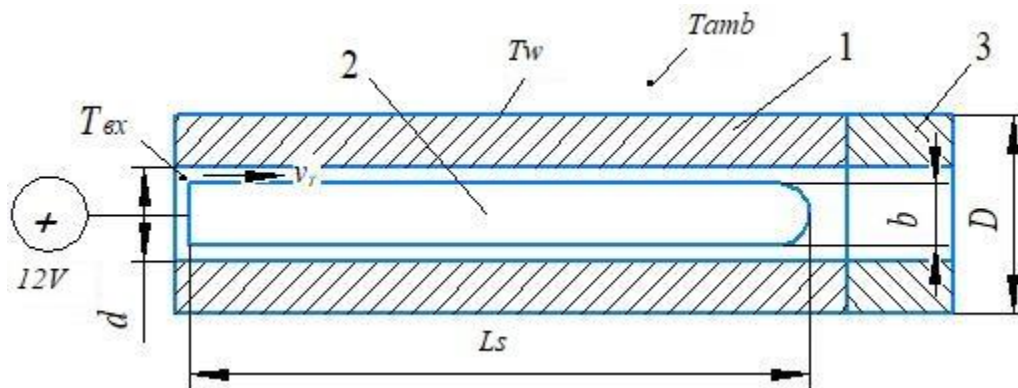


Figure 2: Schematic view of heat transfer from heater in fuel line of high-pressure pump to fuel during start: 1 - nozzle body; 2 - electric heater; 3 - cylinder head.

The equation of thermal balance of fuel pump element (fuel supply line and sprayer with electric heater) during pre-start

preparation will be as follows:

$$Q_{eh}^F = Q_{eh} - \sum_{i=1}^n Q_{rel}, \quad (19)$$

Taking into account that:

$$\sum_{i=1}^n Q_{rel} = Q_b + Q_h, \quad (20)$$

it can be written:

$$Q_{eh}^F = Q_{eh} - Q_b - Q_h, \quad (21)$$

where Q_{eh}^F is the heat supplied to fuel from the electric heater, J; Q_{eh} is the heat released by electric heater, J; $\sum_{i=1}^n Q_{rel}$ is the cumulative heat loss released by sprayer, J; Q_b is the heat dissipated by sprayer body into ambient environment, J; Q_{uh} is the heat released to cylinder head, J. The amount of heat transferred by electric heater in the time t :

$$Q_{eh} = I_{eh} \cdot U_{eh} \cdot t, \quad (22)$$

where I_{eh} is the current in heater circuit, A; U_{eh} is the voltage feeding the electric heater, V.

In addition, the heat consumed for additional fuel heating in fuel pump element in pre-start period by electric heater can be determined as follows:

$$Q_{eh}^F = c_V^F \cdot m_F \cdot \Delta T_{eh}. \quad (23)$$

The amount of fuel heated in the sprayer using electric heater (m_F):

$$m_F = \frac{\pi}{4} l_{eh} \cdot \rho_f \cdot (d^2 - b^2), \quad (24)$$

where b is the heater outer diameter, m; d is the internal diameter of sprayer channel, m; l_{eh} is the length of electric heater in high-pressure fuel pump, m.

$$\Delta T_{eh} = T_p^{eh} - T_{in}. \quad (25)$$

After substitution of Eqs. (24, 25) into Eq. (23) we have:

$$Q_b^{eh} = c_V^F \cdot \frac{\pi}{4} \cdot l_s \cdot \rho_f \cdot (d^2 - b^2) \cdot \Delta T_{eh} \quad (26)$$

Heat loss via sprayer wall (body) is determined as follows [16]:

$$Q_b = \frac{2\pi \cdot \lambda_K \cdot (T_s^{eh} - T_w) \cdot l_{eh}}{\ln \frac{d}{b}}, \quad (27)$$

where λ_K is the coefficient of heat conductance of sprayer body J/m·K; T_s^{eh} is the fuel temperature in sprayer after electric heater, K; T_w is the temperature of outer wall (sprayer body), K; l_{eh} is the length of heater, m.

In its turn, the temperature of outer wall is determined as follows [16]:

$$T_w = T_s^{eh} - \frac{q_i}{2\pi \cdot \lambda_K} \cdot \ln \frac{d}{b}, \quad (28)$$

where q_i is the heat flux per 1 m of heating element, J/m.

The heat flux is:

$$q_i = \frac{Q_b}{l_{eh}}, \quad (29)$$

During start and further operation of engine, when fuel flow speed is higher than zero, the heat transfer will be nonstationary. Then, the constituents of thermal balance (21) can be determined as follows.

The amount of heat obtained by fuel (J/cycle) is:

$$Q_{ph} = m_F \cdot c_s \cdot (T_s^{eh} - T_{in}) \quad (30)$$

Let us rewrite Eq. (30) with consideration for m_F as follows:

$$Q_{ph} = g_{cyc} \cdot \rho_f \cdot c_s \cdot (T_s^{eh} - T_{in}) \quad (31)$$

The values of c_s and other physical constants for diesel fuel are determined at average fuel temperature:

$$T_f = \frac{T_s^{eh} + T_{in}}{2}, \quad (32)$$

The fuel speed in nozzle channel is determined on the basis of absence of cycled fuel supply:

$$v_F = \frac{4 \cdot g_{cyc} \cdot n}{120\pi(d^2 - b^2)} \quad (33)$$

The Reynolds number for fuel is:

$$Re = \frac{v_F \cdot (d-b)}{\nu}, \quad (34)$$

where ν is the kinematic viscosity of fuel, mm²/s.

If $Re < 2 \cdot 10^3$, then the regime is laminar, and at $Re > 10^4$ it is turbulent.

The Nusselt number for fuel according to Mikheev [17] is:

$$Nu_f = 0.021 \cdot Re_{ef}^{0.8} \cdot P_{rf}^{0.4} \cdot \left(\frac{P_{rf}}{P_{rw}}\right)^{0.25} \cdot E_e, \quad (35)$$

where P_{rf}, P_{rw} are the Prandtl numbers for fuel; E_e is the adjustment coefficient.

The Prandtl number P_{rw} is determined with consideration for wall temperature. The ratio $\left(\frac{P_{rf}}{P_{rw}}\right)$ takes into account the influence of direction of heat flux.

The coefficient E_e at $\frac{l_{ph}}{d} > 50$ is assumed to be one. At $\frac{l_{ph}}{d} < 50$ its value is selected in Tables 10.1 and 10.2, PP. 189-190 [16].

The coefficient of fuel heat release to sprayer body wall α_T is determined as follows:

$$\alpha_T = \frac{Nu_f \cdot \lambda_F}{d-b}, \quad (36)$$

where λ_F is the coefficient of fuel heat conductance, J/m·K. Substituting the obtained data into Eq. (27), we obtain the following:

$$Q_b = \alpha_T \cdot F \cdot (T_s^{eh} - T_w), \quad (37)$$

where F is the surface area of heat release, m².

The temperature of outer wall of sprayer body is determined by Eq. (28).

Thus, using the proposed procedure and adopted to this case classical heat calculation of fuel pump element with electric heating of fuel, it is possible to determine not only the exergy of heat supplied from electric heater to fuel but also losses, using Eq. (21). The amount of heat released by electric heater in time equaling to cyclic supply (t_{cyc}) can be determined by Eq. (22).

The exergy of heat supplied to fuel in sprayer by heat transfer from working medium to sprayer nozzle is computed by means of energy balance of processes in cylinders. However, in this case the data of indicator diagrams of combustion and gas exchange are required, which is related with rather complicated experiments. In addition, it is difficult to provide validity of results due to determination of current and integral values of fuel combustion temperature.

At the same time, aiming at simplification of computations, it is possible to assume that the surface temperature of

cylinder head is already characterized by functional integral dependence on temperatures of working medium and ambient environment. Then the exergy of heat (E_{cyc}^F)

supplied from working medium to sprayer can be computed by Eq. (5) and flowchart in Fig. 3.

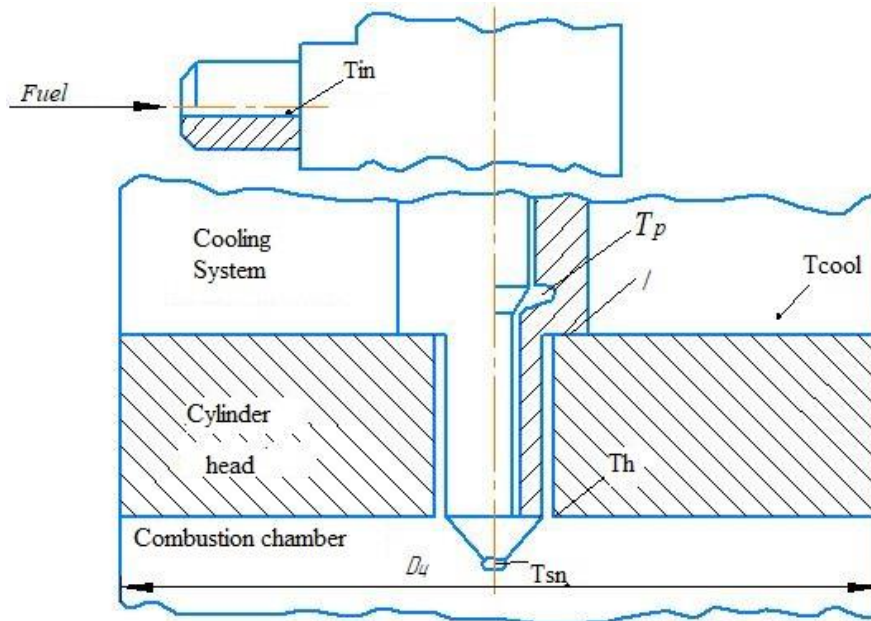


Figure 3: Schematic view of heat transfer from working medium to spray medium.

In subsequent considerations, it is assumed that the sprayer and the head are single entity, and the surface temperature of sprayer protruded part (sprayer nozzles) equals to the head temperature near sprayer (T_h), that is, $T_h \approx T_{sn}$. Then:

$$E_{cyc}^F = Q_{sn} \cdot \left(1 - \frac{T_p}{T_{sn}}\right), \quad (38)$$

where Q_{sn} is the fuel heat at output from sprayer nozzles, J/cycle; T_{sn} is the fuel temperature at output from sprayer nozzles, K.

$$Q_{sn} = c_p^F \cdot m_F \cdot T_h, \quad (39)$$

where T_h is the temperature of fire side of cylinder head, K.

After substitution of Q_{sn} and m_F into Eq. (37) we have:

$$E_{cyc}^F = c_p^F \cdot g_{cyc} \cdot \rho_f \cdot T_h \cdot \left(1 - \frac{T_p^{sh}}{T_h}\right). \quad (40)$$

Total physical exergy of fuel (E_{ph}^F) at outlet from sprayer nozzles can be determined by Eq. (4) for elemental fuel mass flow rate (dg_{cyc}) in the following form:

$$E_{ph}^F = \int_0^g [H_1 - H_0 - T_0 \cdot (S_1 - S_0)] \cdot \rho_f \cdot dg_{cyc}. \quad (41)$$

Exergy loss D^F can be obtained using the Guy–Stodola equation:

$$D^F = T_{amb} \cdot \Delta S, \quad (42)$$

where ΔS is the variation of entropy of all participating media.

In this case, it is suitable to determine exergy losses by equation of exergy balance. Let us rewrite Eq. (7) as follows:

$$E_{ph}^F = \sum_{i=1}^n E^F - \sum_{i=1}^n D^F, \quad (43)$$

hence:

$$\sum_{i=1}^n D^F = \sum_{i=1}^n E^F - E_{ph}^F, \quad (44)$$

where $\sum_{i=1}^n E^F$ is the cumulative exergy supplied to fuel in nozzle, J/cycle.

The above considerations are valid for liquid fuel. Thus, it is assumed that the fuel jet is decomposed after its discharge from sprayer nozzles. This period is characterized by fuel heating, its evaporation and overheating due to exergy of air compressed in cylinder and partially due to exergy obtained by fuel in high-pressure fuel pump. Heat consumption for fuel heating is determined as follows:

$$Q_h = c_s^F \cdot mf \cdot (T_{sn} - T_{boil}), \quad (45)$$

or

$$Q_h = c_s^F \cdot g_{cyc} \cdot \rho_f \cdot (T_{sn} - T_{boil}), \quad (46)$$

where T_{boil} is the average boiling point of diesel fuel, K.

In its turn, T_{boil} is determined as follows:

$$T_{boil} = \frac{1}{g} \sum_{i=1}^g T_{i}, \quad (47)$$

where T_1, T_2, \dots, T_g are the average final boiling points of 10, 20, ..., 90% of initial fuel amount, K.

The heat of vapor generation is determined as follows:

$$Q_r = \delta \cdot g_{cyc} \cdot \rho_f \cdot r, \quad (48)$$

where δ is the fraction of fuel evaporated during start in the range of 0.6...0.8; r is the specific heat of vapor generation, J/kg.

The heat consumed for overheating of fuel vapors is determined as follows:

$$Q_{oh} = c_{ohv} \cdot \delta \cdot g_{cyc} \cdot \rho_f \cdot (T_c - T_{boil}), \quad (49)$$

where c_{ohv} is the specific heat capacity of overheated fuel vapors, J/kg·K, T_c is the air temperature at the end of

compression stroke (during start it is assumed to be 623 K, and during post-start heating – 773 K). Further considerations are based on the flowchart of exergy

fluxes entering engine cylinder during start at the end of compression stroke (Figure 4). In this case, it is assumed the cyclic fuel supply starts and ends in piston UDC prior to initiation of combustion.

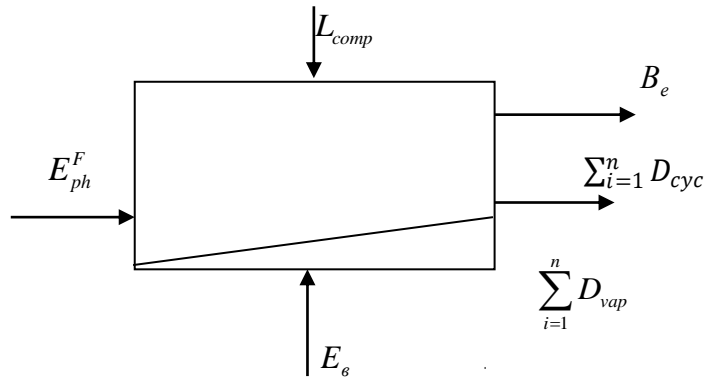


Figure 4: Schematic view of exergy fluxes in engine cylinder and the end of compression.

The exergy balance of fuel mixture for cylinder at the end of compression stroke after completion of fuel supply is written as follows:

$$E_{ph}^F + E_{air} + L_{comp} = B_e + \sum_{i=1}^n D_{cyc} + \sum_{i=1}^n D_{vap}, \quad (50)$$

where E_{air} is the exergy of air supplied to cylinder, J/cycle; L_{comp} is the work of air compression in cylinder, J/cycle; B_e is the exergy potential characterizing the degree of readiness of air fuel mixture at the beginning of oxidation, J/cycle; $\sum_{i=1}^n D_{cyc}$ is the exergy loss due to heat transfer via the walls of cylinder and combustion chamber, J/cycle; $\sum_{i=1}^n D_{vap}$ is the exergy consumption for heating, evaporation, and overheating of fuel vapors (internal loss), J/cycle.

Let us consider that the cumulative evaporation loss includes:

$$\sum_{i=1}^n D_{vap} = D_h + D_r + D_{ohv}, \quad (51)$$

where D_h is the exergy of heat consumed for fuel heating from T_{in} to T_{sn} , J/cycle; D_r is the exergy of heat of fuel vapor generation, J/cycle; D_{ohv} is the exergy of heat consumed for overheating of fuel vapors by $\Delta T = T_c - T_{boil}$, J/cycle.

Let us consider cold engine start without air preheating, thus, $E_{air} = 0$.

In the case when before start air supplied to combustion chamber is heated, the exergy of air is taken into account.

The work of air compression in cylinder (L_{comp}) is determined as follows:

$$L_{comp} = \frac{n}{n-1} \cdot \frac{\pi \cdot D^2}{4} \cdot S \cdot P_c \left[\left(\frac{P_c}{P_a} \right)^{\frac{n-1}{n}} - 1 \right], \quad (52)$$

where D is the piston diameter, m; S is the piston stroke, m; P_c is the pressure of compression end, Pa; n is the compression polytrope; P_a is the pressure of injection end, Pa.

Cumulative consumptions of exergy of heat for heating, evaporation, and fuel overheating are determined by Eq. (52) with consideration for Eqs. (5), (45), (46), (47), and

(48):

$$\sum_{i=1}^n D_{vap} = c_s^F \cdot g_{cyc} \cdot \rho_f \cdot T_{boil} \cdot \left(1 - \frac{T_{sn}}{T_{boil}} \right) + \delta \cdot g_{cyc} \cdot \rho_f \cdot r \cdot \left(1 - \frac{T_{sn}}{T_{boil}} \right) + c_{ohv} \cdot \delta \cdot g_{cyc} \cdot T_c \cdot \left(1 - \frac{T_{boil}}{T_r} \right). \quad (53)$$

Due to polytropic pattern of compression, there is significant heat loss via cold walls of cylinder, head and piston. Therefore, in subsequent computations $\sum_{i=1}^n D_{cyc}$ should be considered.

The exergy potential B_e is the sum of exergy of fuel injected into combustion chamber (B_e^F) and exergy of air charge (B_e^{air}). In terms of thermodynamic laws, the following equality is valid:

$$B_e = B_e^{air} + B_e^F. \quad (54)$$

At the same time, the exergy potential can be in fact the sum of exergies B_e^F and B_e^{air} only when the time of heat and mass exchange of the two systems is sufficiently high. During this time the existing concentration heterogeneity accompanied by temperature heterogeneity should be equalized over total volume of combustion chamber.

The processes in the combustion chamber are very short, which leads to heterogeneous composition of mixture and temperature. At the same time, Eqs. (7), (52), (53), and (54) make it possible to obtain the set of three equations:

$$\begin{cases} E_{ph}^F - \sum_{i=1}^n D_{vap} = B_e^F \\ E_{air} + L_{comp} - \sum_{i=1}^n D_{cyc} = B_e^{air} \\ B_e^F + B_e^{air} = B_e \end{cases}. \quad (55)$$

The obtained set reflects the process of atomization in terms of energy, which is extremely complicated in the theory of ICE.

It is known that atomization in engine cylinder (heating, evaporation, mutual diffusion, and mixing of fuel vapors with air) is carried out as a consequence of mutual heat and mass exchange of two simultaneous processes: thermomechanical activation of fuel, starting from high-

pressure fuel pump and to the combustion chamber, and thermomechanical preparation of air charge with its sufficiently developed turbulence at the end of compression stroke [7].

In order to estimate the level of fuel activity to oxidation and the degree of readiness of air charge to provide this process, we propose two exergy criteria:

$$K_e^F = \frac{B_e^F}{g_{cyc}^F} \quad (56)$$

and

$$K_e^{air} = \frac{B_e^{air}}{g_{cyc}^{air}}, \quad (57)$$

where K_e^F is the exergy criterion of activity of fuel spray in combustion chamber equaling to the fuel exergy per one mg of cyclic supply, J/mg; K_e^{air} is the exergy criterion of energy supply to air charge in combustion chamber equaling to the air exergy per one mg of cyclic fuel supply, J/mg.

Therefore, the proposed procedure of estimation of readiness of engine cylinder parameters to oxidation start makes it possible to analyze qualitative and quantitative variation of constituents of exergy balance of ICE compression cycle. The authors believe that the introduced exergy criterion makes it possible to estimate the engine readiness to start under various ambient temperatures. During pre-start preparation of engine the main criterion is its subsequent reliable start at minimum exergy consumptions. Thus, only evaporation of 60...80% of cyclic supply of the lowest boiling fuel fractions is proposed [3].

During post-start period, the main criterion is reduction of heating time and fuel consumption by engine. This should be achieved by increased completeness of combustion, decreased heat losses into ambient environment.

The proposed procedure can also be used for this operation mode, only instead of B_e the notion of exergy criterion of combustion perfection is introduced (B'_e). The value of this criterion can be easily determined by engine parameters corresponding to optimum heat regime using the proposed procedure. During post-start period, completeness of combustion can be achieved and maintained at the upper level by variation of exergy of air or fuel supplied to engine cylinder. The studies have demonstrated that variation of exergy of air or fuel by one and the same value influences differently engine start properties. Upon fuel heating in nozzles, lower heat is consumed in comparison with air heating, provided that the start properties are identical. Thermodynamic exergy of fuel is determined by the degree of pre-start preparation and pressure of fuel injected into combustion chamber.

4. CONCLUSION

1. Partial procedure has been developed based on exergy method allowing to estimate the rate of activity of fuel spray and energy supply of air charge during start and post-start heating of tractor engine.
2. Approaches are highlighted to improvement of operation efficiency of air and fuel supply system during start and post-

start heating of engine.

3. Analytical dependences are obtained allowing to estimate quantitatively and qualitatively the exergy fluxes in air and fuel supply systems.

REFERENCES

1. A.A. Zhurba. Improvement of preheating of tractor diesel engines in winter by fuel heating: Synopsis: cand. thesis. Novosibirsk, 1991.
2. G.M. Krokhta. Eksergeticheskiy metod i kriterii otsenki gotovnosti dizel'nogo dvigatelya k pusku [Exergy method and criteria of readiness of diesel engine for start]. Vestnik NGAU, vol. 18, no. 18, pp. 132-135, 2012.
3. G.M. Krokhta. Upravlenie kachestvom protsessa smeseobrazovaniya v period puska i poslepuskovogo progreva dizel'nogo dvigatelya [Quality control of atomization during start and post-start heating of diesel engine]. Traktory i sel'khozmashiny, no. 4, pp. 3-11, 2019.
4. G.S. Gambhir, D.L. Abata, D.J. Michalek. An experimental and computational investigation on the pre-heating of fuel to improve cold starting of diesel engines. ASME Internal combustion engine division spring technical conference, 2003. <https://doi.org/10.1115/ICES2003-0662>
5. B. Sayin Kul, A. Kahraman. Energy and exergy analyses of a Diesel engine fuelled with biodiesel-diesel blends containing 5% bioethanol. Entropy, vol. 18, no. 11, 2016. <https://doi.org/10.3390/e18110387>
6. F. Meisami, H. Ajam. Energy, exergy and economic analysis of a Diesel engine fueled with castor oil biodiesel. International Journal of Engine Research, vol. 15, no. 5, 2015.
7. M. Özkan. An exergy analysis of CI comparative study on energy and exergy analyses of a CI engine performed with different multiple injection strategies a part load: effect of injection pressure. Entropy, vol. 17, no. 1, 2015. <https://doi.org/10.3390/e17010244>
8. V.V. Sychev. Slozhnye termodinamicheskie protsessy [Complex thermodynamic processes]. Moscow, 1986.
9. A.N. Berdnik. Eksergeticheskiy metod otsenki effektivnosti sistem vozdukhosnabzheniya s odno- i dvukhstupenchatym nadduvom i s silovoi turbinoi [Exergy method of estimation of air supply systems with one- and two-stage supercharge and power turbine]. Fundamental'nye i prikladnye problemy tekhniki i tekhnologii, vol. 3, no. 305, pp. 45-53, 2014.
10. V.A. Kirillin, V.V. Sychev, A.E. Sheidlin. Tekhnicheskaya termodinamika [Technical thermodynamics]. Moscow: Nauka, 1979.
11. G.P. Gokhshtein. Sovremennye metody termodinamicheskogo analiza energeticheskikh ustanovok [Modern methods of thermodynamic analysis of power assemblies]. Moscow: Energiya, 1969.
12. R.W. Haywood. Equilibrium Thermodynamics for Engineers and Scientists. New York: John Wiley and Sons, 1980.

13. G.D. Chernyshev. Rabochii protsess i teplonapryazhennost' avtomobil'nykh dizelei [Working process and thermal loading of vehicle diesel engines]. Moscow: Mashinostroenie, 1986.
14. M.M. Vikhert, Yu.G. Grudskii. Konstruirovaniye vpusknykh sistem bystrokhodnykh dizelei [Designing intake systems of high speed diesel engines]. Moscow: Mashinostroenie, 1982.
15. Yu.B. Sviridov. Smeseobrazovaniye i sgoraniye v dizelyakh [Atomization and combustion in diesel engines]. Leningrad: Mashinostroenie, 1972.
16. B.N. Yudaev. Teploperedacha [Heat transfer]: Guidebook. Moscow: Vysshaya shkola, 1981.
17. M.A. Mikheev, I.M. Mikheeva. Osnovy teploperedachi [Fundamentals of heat transfer]. Moscow: Energiya, 1977.