Heat exchange processes in automotive internal combustion engines

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Abstract. Combustion chamber diameters, an advance angle of fuel injection, injection rate, minimum ratio of air excess, methods of forcing, and oil cooling affect the heat exchange processes of engines. The method of thermal calculation of the working cycle makes it possible to consider the change in the physical properties of the working fluid, the effect of heat transfer between the working fluid, and the environment during the implementation of the working cycle. The main parameters of the gas at characteristic points of the indicator diagram are determined as a result of performing a thermal calculation, which makes it possible to assess the cycle perfection degree. Some of the main parameters of the working fluid (pressure, temperature) and the nature of their change can serve as input data when calculating engine parts for strength. The amount of mechanical work obtained in the cycle and the value of gas volume at the expansion process end demonstrates not only efficiency, but also the dimensions and weight of the engine, namely, indicators that affect the overall layout of machines.

1. Introduction

Hydrocarbon fuel, charged into the car’s tank through the fuel system, enters the cylinder of the internal combustion engine and is burned in the cylinder’s combustion chamber. As a result of fuel combustion, the temperature rises sharply up to 2000 - 3000 °C. High pressure that acts on the engine piston is formed from the ratio of changing parameters of the working fluid. According to the first law of thermodynamics, all supplied heat is directed to heating the engine, which is carried away into the environment through the power system and to effective mechanical work that the engine piston receives. There are three cycles in the internal combustion engine. Every cycle corresponds to a specific type of engine, and thermal calculations are carried out due to their laws. The topic is related to the engine cycle, with the supply of heat at constant volume and constant pressure, since all contemporary diesel engines operate this way.

The scientific hypothesis of the article is to determine the factors influencing the efficiency of ICE and the heat exchange process of the working fluid of diesel ICE.

2. Materials and methods

We consider theoretical cycles, that is, thermodynamic - circular processes of converting thermal energy into mechanical work at the first stage in order to study the processes occurring in the cylinders of internal combustion engines and to clarify the cycle features. The following assumptions are made when considering theoretical cycles: the economic and power indicators of internal combustion engines operating on different cycles are difficult to compare in real life. In these conditions, the
peculiarity of the separate process course of the working cycle or a part of the engine design can affect final comparison outcomes. Therefore, the primary indicators of various cycles at the first stage are considered in theoretical conditions, when every cycle is carried out in the most favorable conditions, in an imaginary heat engine.

At the second stage (under the conditions of an imaginary heat engine), the coefficients considering the actual conditions are introduced into theoretical dependencies:

1. The cycles are closed and run with a constant amount of the same working fluid, which is taken as an ideal gas.
2. The fuel combustion process in the cylinder is replaced by an instant heat supply from an external hot source, and the exhaust gas release process is replaced by an instant heat transfer to the refrigerator.
3. The processes of compression and expansion proceed without heat exchange with the environment; they are assumed to be adiabatic.
4. The heat capacity of the working fluid (gas) throughout the entire cycle is taken to be constant, independent of temperature.
5. The cycles are considered to be closed, occurring at a constant amount of ideal gas.

In accordance with these assumptions, the theoretical cycle is a closed cycle carried out in an imaginary heat engine with a constant non-replaceable portion of the working fluid. Due to the closed nature, the processes of combustion and release of the working fluid during the actual cycle are replaced by the supply and removal of heat. The processes of compression and expansion are assumed to be adiabatic providing maximum heat use; the process proceeds without heat exchange with the environment.

Theoretical cycles have a minimum amount of losses that are in strict accordance with the second law of thermodynamics. Existing ICEs operate in one of three cycles that have their own specific features.

The theoretical cycles of ICE differ in the nature of the processes of supplying heat and releasing it to a cold source.

Three theoretical cycles are used in reciprocating ICEs: modern diesel engines include a constant volume and constant pressure cycle. This cycle is called mixed.

All currently produced diesel engines operate with the given cycle.

3. Theoretical justification
The heat from fuel combustion goes to heat the engine, which is carried away into the environment, and part of the heat produces mechanical work.

Pulsating heat exchange in the order of engine operation is brought into action between a hot working fluid with a temperature of T2, T3, T4 and a coolant with a temperature of T31 with a view to cool the engine cylinder. Heat transfer passes through the cylinder wall surfaces of the engine, in which the heat transfer process is carried out from the working fluid to the coolant. The engine cylinders are located vertically parallel to each other on a common liquid jacket, and thence the coolant goes away to the radiator.

Heat transfer advantage:
- high coefficient of heat transfer due to the high speed of both heat transfer fluids.

Heat transfer disadvantage:
- difficulty in cleaning the cooling system space.

Research conditions:
The stationary mode of the heat transfer process is subject to consideration. The heat transfer coefficient is constant and the heat capacity of the heat fluid flows is constant. The heat transfer surface does not change. The movement of heat fluid flows is described by a hydrodynamic model of ideal displacement. The working fluid is equal to one kilogram of ideal gas. The property of this body is attributed to one kilogram of air. The processes of fresh air intake and gas exhaust are not taken into account, since these processes are not thermodynamic. The combustion process is replaced by the
supply of heat from the hot source, and the exhaust process by the removal of heat to the refrigerator. The process of compression and expansion is carried out without heat exchange with the environment. The chemical composition of the working fluid does not change over the entire cycle.

The cycle consists of two adiabats, two isochores and one isobar.

Cycle parameters:
1. Compression ratio \( \varepsilon = \frac{V_1}{V_2} \);
2. Pressure increase rate \( \lambda = \frac{p_2}{p_c} \);
3. Pre-expansion ratio \( \rho = \frac{V_{x_1}}{V_p} \);
4. Amount of supplied heat \( q_1 = C_v(T_3 - T_2) \); (1)
5. Amount of dissipated heat \( q_2 = C_v(T_5 - T_1) \); (2)
6. Mechanical work cycle \( A = q_1 + q_1^1 - q_2^1 \);
7. Thermal coefficient of efficiency is defined as the ratio of the useful heat to the supplied heat at a constant volume and pressure of the working fluid.

\[
\eta_t = \frac{q_1 + q_1^1 - q_2^1}{q_1 + q_1^1};
\]

The indicator diagram of the theoretical cycle is shown with a mixed heat supply. Gas is injected into the cylinder when the piston moves from the V.M.T. in N.M.T. at constant pressure close to atmospheric. Adiabatic compression of the air entering the cylinder (curve 1-2) occurs when the piston moves backward from N.M.T. to V.M.T. The supply of part of the heat from the hot source to the gas at constant volume (line 2-3) and part of the heat \( Q_{pp} \) at constant pressure (line 3-4) occurs at the end of compression upon arrival of the piston in the V.M.T. The gas pressure at point 3 will be higher if the temperature is brought higher at constant volume. Then there is an adiabatic expansion of the working fluid with the performance of mechanical work (curve 3-4) and the removal of heat \( Q \) to a cold source at a constant volume (line 5-1).

The efficient cycle work is determined by the area \( z - z^1 - b - a - c - z \) (Fig.1).

The mixed cycle is a theoretical cycle for non-compressor diesel engines. All modern tractor and automotive diesel engines operate on a mixed cycle. The compression ratio must be at least 14 for self-ignition of the injected fuel. The values of temperature and pressure at the end of the compression process are similar to the previous formulas:

We replace the temperature of the working fluid at the characteristic points T2, T3, T4, T5 through the temperature at the initial point T1, compression ratio \( \varepsilon \), pressure increase ratio \( \lambda \), preliminary expansion ratio \( \rho \).

Compression ratio means the amount of compression of the working fluid in the ICE combustion chamber. The given value is one of the most significant characteristics of the engine. Its choice depends on the method of mixing and the type of fuel.

In the thermal TS diagram, the entropy \( S = \text{Const} \) is constant along the lines -2 and 3-4, along the lines 2-3, 3-4, 5-1, the entropy changes as the product of heat capacity of the corresponding process by the natural logarithm due to the temperatures ratio of characteristic points of the engine cylinder \( \Delta S = C_v \ln \frac{T_3}{T_2}, \Delta S = C_p \ln \frac{T_4}{T_3} \) and \( \Delta S = C_v \ln \frac{T_5}{T_1} \).
4. Results and discussion
The charging process of a cylinder is different in a supercharged engine than in a naturally aspirated engine. Compressed air in the compressor with a pressure of 0.13 \( \pm \) 0.40 MPa enters the engine cylinder, then a high pressure is formed in the compression stroke of up to 11.0 MPa, which makes it possible to supply a large volume of fuel. The area of the indicator diagram is increased as a result of the boost; therefore, the engine power significantly grows. The heat exchange of the internal combustion engine takes place as a result of fuel combustion with the impact of a pulsating jet against the cylinder vertical wall. Uniform movement of the coolant occurs on the other side of the cylinder. Comparatively high rates of heat and mass transfer in the immediate vicinity of the impact point are among the advantages of jet collision. Pulsating jets of liquid falling on horizontal substrates with a pulsation frequency of about 100 Hz increase the efficiency of heat transfer.

Working fluid inside the cylinder
The cylinders of the engines are located vertically, parallel to each other and are washed around with coolant. The cylinder is heated in the order of the engine operation at two strokes, from temperature $T_1$ to temperature $T_2$ on the compression stroke, and in the combustion stroke at constant volume within the temperature range from $T_2$ to $T_3$ and at constant pressure from $T_3$ to $T_4$. There is no heat transfer at the stroke of the working stroke (Fig. 1. line z1b), it goes through an adiabatic process. The liquid transfers the produced heat under the pump pressure to the radiator and further to the environment. These heat exchange processes are not revealed in this paper. The following assumptions are made to construct a system of equations for the mathematical description of the heat transfer process in a heat exchanger:

- stationary mode of the heat transfer process is considered;
- heat transfer coefficient is constant and known;
- heat capacity of the heat fluid flows is constant;
- heat exchange surface is evenly distributed along the engine cylinders;
- movement of heat fluid flows is described by a hydrodynamic model of ideal displacement.

Working fluid

The Figure shows the heat transfer circuit of the engine used for the mathematical description.
Designations:
hot stream
\( T_1 \) – initial temperature, °C (at the beginning of air compression)
\( T_2 \) – final temperature, °C (at the end of air compression)
\( T_2' \) – initial temperature, °C (at the beginning of fuel combustion)
\( T_4 \) – final temperature, °C (at the end of fuel combustion)
\( q_1 + q_1' \) – heat consumption for the engine internal energy
cold stream
\( T_5 \) – initial coolant temperature, °C (at the end of p.x.)
\( T_1 \) – final coolant temperature, °C (at the end of the exhaust)
\( q_2 \) – coolant heat consumption
\( F \) - engine cylinder area
\( K_t \) - heat transfer coefficient
\( \delta \) – wall thickness of the engine cylinder

The heat balance equation for the hydrodynamic model of ideal displacement and the expression for the local intensity of heat transfer (\( \Delta q \)) for the flow of the working fluid of an internal combustion engine are presented in the following form: [1,5]

\[
\Delta q = \frac{dt F}{d\delta \xi}; \\
\Delta q = K_t (T_2 - T_1); \\
\Delta q_1 = K_t (T_3 - T_2); \\
\Delta q_1' = K_t (T_4 - T_3);
\]

The highest rate of heat transfer is observed in the stagnant zone and in the development zone of the thermal boundary layer. One of the essential factors limiting the rate of heat transfer upon the jet impact is the splashing of liquid from the wall. Spattering is observed for turbulent jets of liquid from the wall and large distances from the nozzle to the wall. It was assumed that the spray occurred downstream of the radial position based on the measurement of the velocity of sprayed droplets \( r_s = 5.76 \text{ dN} \) [1,2,4].

where: \( \text{dN} \) - nozzle orifice diameter.

Spattering is quantified by the sprayed mass fraction, which is the ratio between the mass flow rates of the sprayed and falling liquid. Correlation describes the mass fraction of the sprayed fuel, which is determined for the injector nozzle by the following formula \( \frac{L}{dN} \). Hence, the mass fraction of \( \xi \) enlarges with an increase in the Weber number for the jet. The use of pulsations with frequencies
up to 220 Hz leads to an increase in heat transfer in the stagnant zone up to 100%. Strengthening in this case is associated with renewal of the thermal boundary layer, and becomes dominant at the Strouhal number \( Sr \geq 35 \). Strouhal number is determined by the frequency of pulsations, characterized by the length and speed of the jet:

\[
Sr = \frac{L}{V}dN;
\]

The heat balance equation and expressions for the local heat transfer rate for the heat fluid flow of the engine cooling system are written as follows:

\[
\Delta q = \frac{\partial t}{\partial s};
\]

\[
\Delta q_2 = K_t (T_1 - T_5);
\]

Taking into account that local intensities of both flows are the same in magnitude, we write the following expression: [1,3]

\[
\Delta q + \Delta q_1 + \Delta q_2 = \Delta q_2;
\]

Besides, the compression ratio is selected owing to the presence or absence of boost, engine speed, cooling system, and other factors.

\[
\frac{V_a}{V_c} = k;
\]

We get the following value from the ratio of changing parameters

\[
\frac{V_a}{V_c} = \frac{P_c}{P_a};
\]

Hence, the pressure at the end of the compression cycle has

\[
P_c = Pa \Xi_k;
\]

Maximum pressure of the mixed cycle is at point 3

\[
P_Z = \lambda P_a;
\]

Temperature at V.M.T. end of the engine compression and isochoric process

\[
T_2 = \Xi k; T_3 = \lambda \Xi k - 1T_1; (5)
\]

Temperature at the end of the heat supply process is:

\[
T_4 = \rho T_3 = \rho \lambda \Xi k - 1T_1; (6)
\]

Pressure at the end of adiabatic expansion is:

\[
P_b = \frac{P_3 \rho}{\Xi_k};
\]

Temperature at the end of adiabatic expansion is calculated by the formula:

\[
T_5 = \frac{\rho k}{\Xi_k} \frac{\rho \lambda \Xi k - 1T_4 V_1}{V} = \frac{\rho \lambda \Xi k - 1T_4 V_1}{V} = \lambda \rho k T_1; (7)
\]

We decipher the thermal efficiency of the mixed heat supply cycle and define it using the formula as follows (4 we put 1, 2, 3)

\[
\eta = \frac{C_p(T_3 - T_2) + C_p(T_4 - T_3) - C_p(T_5 - T_4)}{C_p(T_3 - T_2) + C_p(T_4 - T_3)} = 1 - \frac{C_p(T_5 - T_3)}{C_p(T_3 - T_2) + C_p(T_4 - T_3)}; (8)
\]

We put the obtained values of temperatures of the working fluid into the formula of thermal efficiency of the engine cycle (8) for the characteristic points (5, 6, 7) and obtain the final expression by the mathematical solution.

\[
\eta = 1 - \frac{T_5 - T_1}{T_3 - T_2 + \frac{C_p(T_4 - T_3)}{C_p}} = 1 - \frac{\lambda \rho k T_1 - T_4}{\lambda \Xi k - 1T_1 - T_4 \Xi k - 1T_1 + \frac{C_p}{C_p} (\rho \lambda \Xi k - 1T_1 - \lambda \Xi k - 1T_1)};
\]
We simplify the temperature of the characteristic point $T_1$, and put the expression outside the parenthesis by the engine degree.

We obtain the efficiency of the cycle of an internal combustion engine with a mixed heat supply in the final form:

$$\eta_t = 1 - \frac{T_1(\lambda \rho^k - 1)}{C_v k T_1(\lambda - 1) + C_p \lambda \varepsilon^{k-1} T_1(\rho - 1)};$$

$$\eta_{t_{cs}} = 1 - \frac{1}{\varepsilon^{k-1} \lambda - 1 + k \lambda (\rho - 1)};$$

Compare with the previously derived formula for the efficiency of diesel engines:

$$\eta_{t_p} = 1 - \frac{1}{\varepsilon^{k-1} k (\rho - 1)};$$

In the mixed cycle, increase in the compression ratio improves economic and power indicators of the engine. However, as the compression ratio grows, the increase in heat utilization gradually slows down, and becomes almost imperceptible after values of the compression ratio of $10^{\frac{1}{2}} - 12$. The compression ratio value greater than 15 in diesel engines is explained by the desire to facilitate the start of cold engines. The temperature of the compression end goes up with increasing compression ratio, which ensures self-ignition of the fuel even at low temperatures of the cylinder walls and intake air.

The thermal efficiency of engines operating with a cycle at a constant pressure is less at the same compression ratios in the cycles of internal combustion engines than an engine with a mixed heat supply. We can obtain the following inequation:

$$\eta_{t_{cs}} > \eta_{t_p}$$

5. Conclusion

From the obtained equation of the cycle thermal efficiency, we claim that the use of heat in the mixed cycle depends on the compression ratio, preliminary expansion and pressure increase, as well as the adiabatic index. Thus, the more the compression ratio is formed in the combustion chamber of the engine $\varepsilon$, the higher the thermal efficiency is. The pressure ratio $\lambda$ has a similar pattern to the compression ratio of the engine. The larger the expansion ratio ($\rho$) is, the lower the thermal efficiency we get.

The thermal efficiency for combustion engines ranges from 25% to 42% (maximum). The efficiency decreases depending on the operating conditions, therefore the engineering service must ensure the correct technical operation of the vehicles. The rest of the heat from fuel combustion is carried away to the environment through the engine cooling system.

The temporary temperature difference between the working fluid at the nozzle inlet and the cylinder wall depends on the distance for various pulsation frequencies and is called the wall temperature. The average temperature difference between the heated wall and the working fluid at the inlet to the nozzle hole for diverse pulsation frequencies at a working cycle is 50%, and for a coolant is 100%. The wall temperature is measured above and below the target point, and the distance from the hole to the target is $L/dN$.

The initial data for the calculation and study of the heat exchanger of a car engine, designed for cooling with a liquid heated by a working fluid due to combustion, are equal to each other. As a result of non-observance of the given equality, the car engine overheats or overcools.

References

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