THEORETICAL DESCRIPTION OF THERMAL PROCESSES IN THE ENGINE COOLING SYSTEM OF THE TZST CE-220 COTTON PICKER

Tojiboyev Sherali Imomali o'g'li
Axunov Javlon Abdujalilovich
Fergana Polytechnic Institute, Fergana, Uzbekistan
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Abstract: In this article, how thermal processes occur in the cooling system of tractor engines and how to make a separate system for each of the processes, a separate solution has been found for thermal processes in all parts of the engine in the overall system. In addition, the temperature change in the cooling system at different ambient temperatures was calculated.

Key words: thermodynamics, specific enthalpy, heat balance, heat transfer surfaces.

Introduction

When theoretically studying a hydraulic drive operating at different ambient temperatures, it is important to study the thermodynamic processes occurring in it [3].

An analysis of the condition of cooling systems and their elements during operation in the conditions of the Central Asian region shows that machines moving in conditions of high temperature and dusty air have a very low service life, reliability and reduced performance [1]. It has been established that in conditions of high temperature and dusty air, the service life of mobile machines is reduced by 2 times, energy efficiency deteriorates by 22-25%.

It has been established that the law of change in liquid temperature does not significantly affect the nature of unsteady processes in liquid channels [4] and they can be characterized by a small error in the average specified liquid temperature along the channel.

Since temperature is one of the most profound concepts of thermodynamics, the thermal process occurring in a cooling system can be described by the equation.

$$m * di = dq_B + dq_{atrof-muhit} - dq_{rad} \qquad (1)$$

Here is di - the change in enthalpy of the system, $i -$ specific enthalpy, $m -$ mass of the system, $dq_B -$ internal heat transfer in the system, $dq_{muhit} -$ heat transferred during heat exchange between the system and the environment, $dq_{rad} -$ heat absorption in the water radiator.

Let us write down each term of equation (1), when calculating

$C_p = const$ va $di = C_p dT$

$$\begin{cases} m * di = m_s * c_s * dT_s + m_a * c_a * dT_a \\ dq_B = Q_B * dt \\ dq_{atrof-muhit} = k * F * \left(T_{atrof-muhit} - T_s\right) * dt \\ dq_{rad} = Q_{rad} * dt \end{cases} \qquad (2)$$

Here are $m_s, m_a -$ the liquid and aggregate masses; $c_s, c_a -$ specific heat capacity of liquids and aggregates; $dT_s, dT_a -$ current values of temperature rise of liquids and units;

$T_s, T_{atrof-muhit}$ – current values of liquid and ambient temperature; F - surface area of the external heat transfer system; k - average heat transfer coefficient to the environment; Q_B – current value of thermal power in the system; Q_{rad} – current value of the heat capacity of the liquid radiator.

Taking into account (2), equation (1) can be written as follows:

$$m * c\frac{dT_i}{dt} = Q_B + kF\left(T_{atrof-muhit} - T_i\right) - Q_{rad} \qquad (3)$$

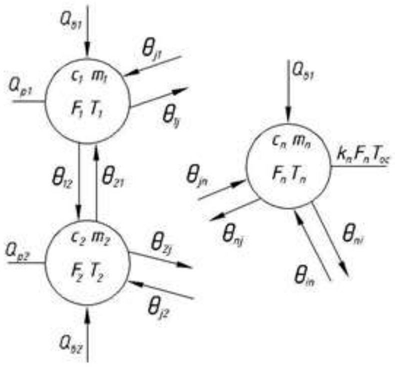
Here is m, c – the mass and specific heat capacity of the cooling system.

Let us assume that the cooling system under consideration consists of "n" sections having different current average temperatures T_1, T_2, ...T_n (Scheme 1). Let's imagine that the average heating level at any section of the flow in a section does not differ from the average mass temperature of the sections, and heat exchange between sections is carried out only due to forced convection of the liquid.

Taking into account these assumptions, equation (3) takes the following form:

$$m_i * c_i\frac{dT_i}{dt} = Q_{Bi} + k_iF_i\left(T_{atrof-muhit} - T_i\right) - \sum_j\left(\theta_{ji}T_j - \theta_{ij}T_i\right) - Q_{rad\ i}, i,j = 1,n \ (4)$$

θ_{ji}, θ_{ij} –average specific (temperature-dependent) heat fluxes per cycle transferred by a liquid from the i - part to the j - part and vice versa during forced convection.



Scheme 1. Diagram of internal heat transfer in the cooling system.

θ_{ij} the cost can be determined as follows

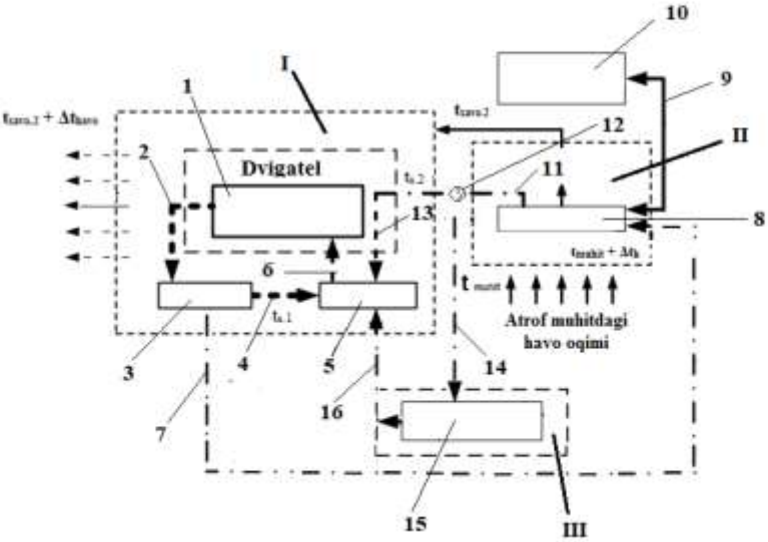
$$\theta_{ij} = Q_{suy\ i}\rho_{suy}c_{suy} \qquad (5)$$

Here is ρ_{suy} – the density of the working fluid; $Q_{suy\ i}$ – i is the average fluid flow across the section.

Thus, equation (4) allows us to obtain a reliable picture of thermal processes in the cooling system of automobiles.

Temperature change in the cooling system at different ambient temperatures.

Let's consider the cooling system shown in Diagram 2.



Scheme 2. Thermal balance diagram of the cooling system

Conventionally, the cooling system can be divided into three sections, section 1 consists of a cooling jacket 1, internal pipes 2,4,6, thermostat 3 and water pump 5. Section 2 consists of hoses 7,9,11,13, radiator 8, expansion tank 10 and heat exchanger 12. The third section consists of hoses 14, 16 and cabin radiator 15.

$$\begin{cases} i_1 \frac{dT_1}{dt} + (kF_1 + \theta_2)T_1 + \theta_2 T_2 + \theta_3 T_3 = Q_{B_1} \\ i_2 \frac{dT_2}{dt} + \theta_2 T_1 + (kF_2 + \theta_2)T_2 + \theta_3 T_2 = 0 \\ i_3 \frac{dT_3}{dt} + (kF_3 + \theta_3)T_3 + \theta_3 T_2 = 0 \end{cases} \qquad (6)$$

The areas of the outer heat transfer surfaces of sections F_I, F_{II}, F_{III}-1, 2 and 3 are determined by the following expressions

$$F_I = F_1 + F_2 + F_3 + F_4 + F_5 + F_6$$
$$F_{II} = F_7 + F_8 + F_9 + F_{10} + F_{11} + F_{12} + F_{13}$$
$$F_{III} = F_{14} + F_{15} + F_{16}$$

where F_1...F_{16} is the area of the external heat transfer surfaces of individual elements with positions 1-16; T_1, T_2 and T_3 - temperature rises in areas defined by these expressions

$T_1 = \tilde{T}_1 - T_{atrof-muhit}$, $T_2 = \tilde{T}_2 - T_{atrof-muhit}$ va $T_3 = \tilde{T}_3 - T_{atrof-muhit}$

i_1, i_2 va i_3 - a is the specific heat capacity of the sections; it is defined by the following expressions:

$i_1 = \sum_{j=1}^{P} c_j m_j$, $i_2 = \sum_{g=1}^{n} c_g m_g$ va $i_3 = \sum_{y=1}^{k} c_y m_y$

Where is c_j, c_g, c_y - the mass heat capacity of materials; m_j, m_g, m_y- - mass of materials; , Q_{B1}, Q_{B2} and Q_{B3} are the average indicators for the period of heat release in time units, which are determined by the following expressions.

$Q_{B1} = p_1 \frac{Q_1}{\eta_{n.umum\ f.i.k}} - Q_{B2}$, $Q_{B2} = p_2 * Q_2$, $Q_{B3} = p_3 * Q_3$

Here is p_1 —the pressure at the pump outlet; Q_1, $\eta_{n.umum\ f.i.k}$ — actual flow and total efficiency factor of the pump; p_2, Q_2 —pressure and flow in the second section;

p_3, Q_2 —pressure and flow in the third section; $\theta_1, \theta_2, \theta_3$ —the average specific heat flows between sections for each cycle (depending on the temperature level) are determined by the expression.

$$\theta_1 = Q_1 \rho_{suy} c_{suy}, \theta_2 = Q_2 \rho_{suy} c_{suy}, \theta_3 = Q_3 \rho_{suy} c_{suy}$$

If we insert the character

$$a_{11} = \frac{kF_1 + \theta_2}{i_1};\ a_{22} = \frac{kF_2 + \theta_2}{i_2};\ a_{13} = \frac{kF_3 + \theta_3}{i_3}. \qquad (7)$$

$$a_{12} = -\frac{\theta_2}{i_2};\ a_{21} = -\frac{\theta_2}{i_1};\ a_{31} = -\frac{\theta_3}{i_1};\ a_{23} = -\frac{\theta_3}{i_2};\ a_{33} = -\frac{\theta_3}{i_3}$$

then the system of equations (6) can be written in the following form

$$\begin{cases} \frac{dT_1}{dt} + a_{11}T_1 - a_{21}T_2 - a_{31}T_3 = \frac{Q_{B_1}}{i_1} \\ \frac{dT_2}{dt} - a_{12}T_1 + a_{22}T_2 - a_{23}T_2 = 0 \\ \frac{dT_3}{dt} + a_{13}T_3 - a_{33}T_2 = 0 \end{cases} \qquad (8)$$

An approximate numerical solution to the system of equations (7) is based on the principle of dividing the entire duration of the transition process into small time intervals. During the selected time interval, the heat transfer parameters are assumed to be constant and equal to the values they take at the reached temperature. Thus, for each time interval, constant integration, stationary temperature, etc. are calculated. The calculation error can always be reduced to the error of the original data by reducing the size of the segments.

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