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Theoretical Foundation of Thermal Regulation of Working Liquid in the Hydraulic System

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Abstract

Theoretical researches about determination of the thermal power transferred to working liquid of a tractor hydraulic system from the developed thermal regulation system by thermal energy of lubricant system engine oil are stated. Use of thermal regulation system provides a warming up and maintenance of working liquid temperature in a hydraulic system within the range of 40…60oC and reduces achievement time of the recommended oil temperature level to 5…30 minutes of thermal regulation system continuous work at various reference temperatures of air.

Keywords: climatic conditions, the thermal regime of the hydraulic system, hydraulic state and the working fluid, the parameters and characteristics of hydraulic machines, technical and economic indicators

Introduction

At design of hydraulic systems in the cars operated outside it is necessary to consider influence of climatic conditions on a hydraulic system thermal condition. It has to remember that design features and an operating mode of a hydraulic system and the car also have impact on the thermal mode. Thus, there is a quality linkage between climatic conditions (environment), a design, a hydraulic system operating mode, on the one hand, and a thermal condition of a hydraulic system on another.

In this logical chain (climatic conditions – the thermal mode of a hydraulic system – a condition of the hydraulic equipment and working liquid – parameters and characteristics of a hydraulic system – technical and economic indicators of the car) the location of the temperature regulation device is clearly visible. Based on the analysis of the existing devices and systems of working liquid temperature regulation the system of thermal regulation in which to warm up working liquid internal reserves of engine, namely, thermal energy of lubricant system engine oil [1] are used was offered.

Because of a design feature of the developed thermal regulation system, during justification of hydraulic system operability increase , it is necessary to define communications between properties of engine lubricant system cooling devices and communications of external factors with criterion of a temperature dynamic characteristics, to define the settlement mode of the oil heater of lubricant system of engine of a tractor, to estimate influence of air temperature on change of the tractor hydraulic system thermal mode and to carry out calculation of the heat exchange equipment of working liquid thermal regulation system in a hydraulic system.

Theoretical foundation

In relation to lubricant system of the automobile and tractor engine, a basis of structural communications of system potential properties and external factors interrelation with criterion of a temperature dynamic characteristics is interaction of the engine which is emitting warmth and the radiator disseminating this warmth in air. Final criterion of this interaction it is necessary to accept the engine thermal condition determined by the established temperature of the t'W engine oil (at a certain temperature of t'L air) [2], when

$$Qen = Qr, \qquad (1)$$

where Qen – the thermal power given by the engine to engine oil, kJ/s; Qr – the thermal power disseminated by a radiator, kJ/s.

The thermal power of Qr which is taken away by a radiator in the surrounding atmosphere rather precisely can be determined by Newton's equation, however use of this equation always presents essential difficulties as values of average temperatures of liquid and air at the exit usually are not known in advance. Therefore, they most often pass from an average logarithmic temperature pressure to an arithmetic average which is the convenient parameter for an assessment of thermal efficiency of a radiator [2]

$$Q_r = \frac{\Delta t_{in}}{\dfrac{1}{k \cdot F_L} + \dfrac{1}{2 \cdot c_{pL} \cdot G_L} + \dfrac{1}{2 \cdot c_{pW} \cdot G_W}}. \qquad (2)$$

where Δt_{in} – initial difference of temperatures of liquid and air in a radiator, ºC; c_{pW} – specific heat of liquid, kJ/кг·ºC; G_W – consumption of working liquid of a radiator, kg/s; c_{pL} – specific heat of air, kJ/кг·ºC; G_L – consumption of the running air on a radiator, kg/s; k – coefficient of a heat transfer, $W/m^{2}\cdot{}^{o}C$; F_L – area of a radiator, m^2.

After some transformations we can receive the equation of this kind [2]

$$Q_r = \frac{1}{\dfrac{1}{k \cdot F_L} + \dfrac{1}{2 \cdot c_{pL} \cdot G_L} + \dfrac{1}{2 \cdot c_{pW} \cdot G_W}}, \qquad (3)$$

which considers all factors influencing the generalized indicator of a temperature dynamic characteristics of lubricant system cooling devices of the tractor or car engine.

At low temperatures the hydraulic system can be prepared for work on the rational thermal mode at the expense of an internal thermal emission, for example, of friction of liquid about pipelines and the hydraulic equipment walls, at the expense of external heat sources or at the expense of both sources. To approach the model of hydraulic system heat exchange to the processes taking place in the conditions of real operation it is necessary to consider both internal, and external sources of heat.

It is expedient to carry out calculation of parameters of a separate modular tractor hydraulic system of and technical and economic indicators of the car in the range of the established working liquid temperatures providing its working capacity. For the oil applied as working liquid of an experimental tractor hydraulic system it makes +5…+90°C [3]. And initial value of working liquid temperature was accepted equal to temperature of air or temperature up to which the car managed to be cooled.

The thermal power spent for heating of a hydraulic system is taken from the equation [4]

$$Q^{ha}_{heati} = Q_{rei} + Q_{cdi} - Q_{exi}, \qquad (4)$$

where Q_{ni} – the thermal power received by a hydraulic system in time $\Delta\tau_i$, kJ; Q_{exi} – the thermal power given by a hydraulic system to environment in time $\Delta\tau_i$, kJ,

$$Q_{exi} = k_{ha} \cdot F_{ha} \cdot (T_i - T_o) \cdot \Delta\tau_i, \qquad (5)$$

The thermal power received by a hydraulic system in time $\Delta\tau_i$, kW

$$Q_{rei} = N_{heati} \cdot \Delta\tau_i, \qquad (6)$$

where N_{heati} – the power lost in a hydraulic system (it is spent for heating), kW;

The power lost in a hydraulic system. It consists of the components [4]

$$Nheat = Np + Nd + Nc + Npl + Nbp + Nf, \qquad (7)$$

where Np, Nd, Nc, Npl, Nbp, Nf – losses of power in the pump, the distributor, a hydraulic cylinder, pipelines, the backpressure valve and the filter.

Having calculated consistently each component of a formula (7) we will be able to consider about the quantity of the thermal power received by hydraulic system working liquid as a result of its additional transfer, a sparging and the increasing friction about walls of pipelines and hydraulic units.

The calculations of the thermal power received from warming up devices are included into calculation of the hydraulic system thermal mode as separate blocks. The thermal power received by a hydraulic system from the warming up device can be determined by the formula [4]

$$Q_{hs} = k_T \cdot F_T \cdot (T_{1\text{-}2} - T_3), \tag{8}$$

where k – heat transfer coefficient, W/m^2·°C; $T_{1\text{-}2}$ – temperature of the hot heat carrier in front of the heat exchanger, °C; T_3 – temperature of the hot heat carrier after the heat exchanger, °C.

A practical solution to the problem

The heat exchanger offered by us constructively represents a hydraulic tank through which two heat exchange elements executed as system of tubes pass. For the most effective operation of the heat exchanger it is expedient to apply the countercurrent scheme of the heat carriers movement as at identical temperatures of the entering and leaving heat carriers (Δt) at a countercurrent it is always more, than at a direct flow. Thus, transfer of the same thermal stream of Q at the countercurrent scheme will require the heat exchanger of the smaller area and, besides, only in the countercurrent heat exchanger it is possible to heat the cold heat carrier to temperature of higher, than temperature of the heating heat carrier at the exit $t''_2 > t''_1$ [5].

On the strength of the listed conditions and features of a hydraulic system design, we accept the cross scheme the movement of heat carriers with number of the cross courses more than three. In this case the scheme of the movement can be considered purely countercurrent as the directions of the heat carrier and working liquid movement in a tractor hydraulic system are opposite. One of heat exchange elements is intended for working liquid heating, the second for it´s cooling in the conditions of the lowered or increased temperatures. The heating element is consistently included in lubricant system of the engine and can carry out functions of the regular oil heater of a tractor. The element for cooling of working liquid is connected to a pneumatic system receiver on the entrance to a hydraulic tank and freely reported with the atmosphere at the exit [1].

For the purpose of calculation simplification, conditionally we will divide the bringing and taking away branch pipes into sites, in which there are approximately identical diameters and mode of a current of heat carriers (fig. 1).

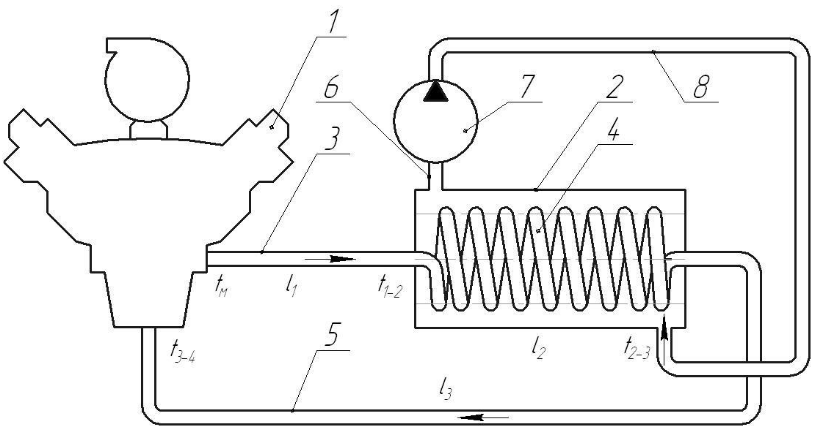

Fig. 1 – System of regulation of temperature of working liquid in a hydraulic system

1 – engine; 2 – hydraulic tank; 3 – pressure head hydraulic line; 4 – heat exchange element; 5 – drain hydraulic line; 6 – the soaking-up hydraulic line; 7 – gear pump; 8 – delivery hydraulic line.

It is obvious that on sites 1-2 and 3-4 where the movement of liquid happens in the flexible rubber reinforced hoses, heat exchange between the heating oil and the atmosphere is insignificant and during calculations it is possible to neglect losses of heat due to natural convection through their external surfaces, and oil temperature on the entrance equals to the heat exchanger the temperature of oil at the exit from the engine.

Calculation of the heat exchange device is reduced to the joint solution of the equations of thermal balance and a heat transfer in which values of temperatures of heat carriers after the heat exchanger are unknown. For determination of final temperatures of heat carriers we will use formulas [6]

$$T_{2-3} = T_м - \left(T_м - T_o\right) \cdot Z \,, \tag{9}$$

$$T_{2-3} = T_o + \left(T_м - T_o\right) \cdot Z \cdot \frac{G_м \cdot c_{pм}}{G_o \cdot c_{po}} \,, \tag{10}$$

where T_o – temperature of working liquid in a hydraulic tank, ^oC; $T_М$ – temperature of engine oil on an entrance to the heat exchanger, ^oC; Z – skilled coefficient; $G_М$ – mass consumption of engine oil, m^3/s; G_o – mass consumption of working liquid of a hydraulic system, m^3/s; $c_{pМ}$ – specific heat capacity with a constant pressure of engine oil, J/(kg·^{o}C); c_{po} – specific heat capacity with a constant pressure of working liquid, J/(kg·^{o}C).

We will consider change limits, a choice and calculation of the variables included into the equations (9) and (10).

Temperature of engine oil at the exit from the engine can be determined from a technical characteristics of the engine of the basic car. The maximum established temperature of engine oil in the diesel engine at negative temperatures of air shouldn't be lower 40os. At lower temperatures of engine oil operation of the engine becomes unstable, fuel consumption considerably increases (6…7%), extent of wear of engine details increases from 4 to 20 times in comparison with normal temperature conditions (85 … 90°C) [7, 8].

To warm up engine oil in the engine lubricant system, water or cooling liquid in the cooling system and to prepare the engine for perception of operational loadings before its start-up, they apply the special different starting heaters established on tractors. Thereby, they achieve an engine warming up till the temperature of oil, water or the low-freezing liquid equal 50 … 60°C, as much as possible reducing the engine operation period in the range of the underestimated working liquid temperatures. Besides, engine preparation time for perception of operational loadings, irrespective of air temperature, shouldn't exceed 30 minutes [9].

Tractors and hydraulic mobile machines are operated in winter conditions on different types of works. Generally all of them are interfaced to moving from an parking lot to a venue of works – a field, a farm, etc. Time spent on moving can be used for a warming up of hydraulic system working liquid before its intensive operation.

Thus, at calculation of working parameters of the heat exchanger it is expedient to proceed from an assumption that temperature of the heating heat carrier (engine oil) ($T_м$) up to the heat exchanger operation start reaches normal temperature condition – 85±5°C.

Diameter and length of sites of pipelines in each case are chosen structurally – depending on dimensions of the car, engine capacity and engine capacity, and also on location of a hydraulic tank.

The coefficient of a heat transfer of the coil pipe section can be calculated by the formula [4, 5, 6]

$$k_1 = \frac{1}{1/\alpha_1 + \delta/\lambda + 1/\alpha_2}, \tag{11}$$

where α_1 – coefficient of a heat irradiation of engine oil in the engine to a pipe wall, W/(m^2·oC); δ – average thickness of a wall of the considered section of the pipe, m; λ – coefficient of heat conductivity of a wall of a pipe, W/(m·°C); α_2 – heat irradiation coefficient from a pipe wall to air, W/(m^2·°C).

In the formula (11) value of δ and λ are known from a design of the heat exchange device, value of coefficient α_2 gets out depending on heat exchanger material. The heat irradiation coefficient α_1 for the turbulent mode of oil is defined on the basis of the criteria equation [10]

$$Nu_м = 0{,}021 \cdot \mathrm{Re}_м^{0,8} \cdot \mathrm{Pr}_м^{0,43} \cdot \left(\mathrm{Pr}_м / \mathrm{Pr}_c\right)^{0,25} \cdot \xi_1, \tag{12}$$

where Nu – Nusselt´s number, $Nu = \frac{\alpha_1 \cdot d_b}{\lambda_м}$; $\mathrm{Re}_М$ – Reynolds´ number, $\mathrm{Re}_м = \frac{\upsilon_м \cdot d_b}{\nu_м}$; $\mathrm{Pr}_М$, Pr_c – Prandtle´s number oils and walls of a coil respectively; ξ_1 – correction coefficient, for coil pipes $\xi_1 = 1 + d/R$; d_b – internal pipe diameter, m; $\lambda_м$ – thermal conductivity coefficient of the heat carrier, W/(m·°C); $\upsilon_м$ – speed of an engine oil current, m/s; $\nu_м$ – kinematic viscosity coefficient of the heat carrier, m^2/s.

Having substituted Nusselt´s, Reynolds´ and Prandtle´s numbers in the equation(12), having solved it concerning heat irradiation coefficient α_1 and having expressed dynamic viscosity (μ) through the kinematic – $\mu = \nu \cdot \rho$, we will receive

$$\alpha_1 = 0{,}021 \cdot \frac{\nu_м^{1,23} \cdot \rho_м^{0,43} \cdot c_{pм}^{0,43} \cdot \lambda_м^{0,57}}{\nu_м^{0,8} \cdot d_b^{0,2}} \cdot \left(\frac{\Pr_м}{\Pr_c}\right)^{0,25} . \tag{13}$$

As wall temperature and respectively and Prandtle´s number for a wall are unknown, it is expedient to keep the decision by method of consecutive approximations.

Conclusion

Thus, having determined temperature of engine oil before the heat exchanger (T_{1-2}) and after it (T_3), it is possible to determine the thermal power (Q_{cd}) transferred to a hydraulic system by the device of a working liquid warming up.

It is obvious that for each type of a tractor the system of thermal regulation will have various geometrical sizes depending on the area of heat exchange, diameter of the pipeline, speed of a liquid current in the channel, masses and density of the heat carrier.

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