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Theoretical Substantiation of the Advisability of Using Adhesives When Sealing the Core of Car Radiators and Diagnosing Radiators with a Thermal Load

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Abstract *- In this work, a method for diagnosing radiators has been developed, which consists in exposing the radiator to a heat load and measuring heat transfer when the balance of the heat exchange process is achieved. A complex means of technological and information support for diagnosing radiators have been created, including means for modeling heat load, registering, processing, and storing the obtained values of parameters of the technical condition of radiators. A mathematical model of heat transfer from a radiator is obtained, reflecting the features of its functioning in conditions of operational pollution.*

Keywords — *heat transfer, heat transfer coefficient, total thermal resistance, aerodynamic resistance, hydraulic resistance.*

Abbreviation: *MV - motor vehicles; ICEs - internal combustion engines; TM - technical maintenance; VP - Vehicle passport;*

I. INTRODUCTION

The efficiency of motor vehicles (MV) equipped with internal combustion engines (ICEs) is largely determined by the technical condition of the cooling system units, which maintains a given thermal mode of engine operation. One of the most heat-loaded units of the system is the radiator, the contamination of the internal and external heat-transfer surfaces of which leads to engine overheating, power loss, an increase in fuel consumption (by an average of 5-6%), detonation, and increased oil burnout. Overheating of the engine leads to increased wear of the elements of the cylinder-piston group and premature engine failure. The only diagnostic parameter that indirectly reflects the influence of a large number of operational factors on the thermal regime of the internal combustion engine and can be measured while the vehicle is moving is the temperature of the coolant at the radiator inlet [1,2]. A visual assessment of the condition of the working surfaces of the radiator does not allow for a quantitative assessment of contamination and the degree of their influence on the output parameters. Monitoring the condition of internal heat transfer surfaces is particularly difficult due to the complexity of their shape and inaccessibility for visual observation. Well-known control methods and recommendations for maintaining the operability of the cooling system date back to the 70s - 80s of the last century and do not correspond to the changed operating conditions due to the transfer of ICE cooling systems to work with antifreeze and new radiator designs [3,4]. Because of the above, this study is relevant to the development of diagnostic methods, tools, and algorithms necessary to ensure an objective assessment of the technical condition of radiators.

II. MATERIALS AND METHODS

A. Investigation of car failures during its service.

According to the theory of reliability, a car as a technically complex object can fail. The failure of the car is a random event, and the time (T) before its occurrence is a random value. It is impossible to predict a refusal; one can only characterize the occurrence or non-occurrence of a given event with any probability [5,6,7].

In this regard, the probability of failure-free operation P (t) for some fixed time t is called the probability that the time T> t, i.e. (1)

$$P(t)=P(T>t) \tag{1}$$

Where T is a random variable equal to the operating time to failure, the probability of uptime is a decreasing function of time with the following properties (2):

$$0<P(t)<l, P(0)=1, P(+\infty)=0 \tag{2}$$

If we assume that the car can be in only one of two states related to operability (operable or inoperative), then the probability of no-failure operation P (t) can be related to the probability of failure by the following relations (3):

$$P(t)=1-F(t)=1-\int_0^t f(t)dt=\int_t^\infty f(t)dt \quad (3)$$

Where $F(t)$ is the probability of failure, equal to the probability that the meantime to failure will be less than the current value of *t*? A graphical representation of the probabilities of uptime and failure is shown in Fig. 1. [8,9].

The distribution density of the operating time to failure is related to the probability of failure-free operation by the ratio (4)

$$f(t)=\frac{dF(t)}{dt}=-\frac{dP(t)}{dt} \quad (4)$$

The failure rate $\lambda(t)$ is the conditional density of the probability of a vehicle failure, determined under the condition that the failure did not occur before the considered moment. The failure rate is used to characterize the vehicle's reliability and is defined as the ratio of the distribution density to the probability of the vehicle's no-failure operation (5):

$$\lambda(t)=\frac{f(t)}{P(t)} \quad (5)$$

The failure rate characterizes the proportion of products that have failed per unit of time since time t, referred to the number of products that are operational from time *t*.

The failure rate is estimated by the formula (6)

$$\lambda(t)=\frac{N(t)-N(t+\Delta t)}{\Delta t N(t)}=\frac{\Delta N}{\Delta t N(t)} \quad (6)$$

Where $N(t)$ is the number of workable objects at time t; Δt - rather small time interval; ΔN is the number of failures over the period Δt.

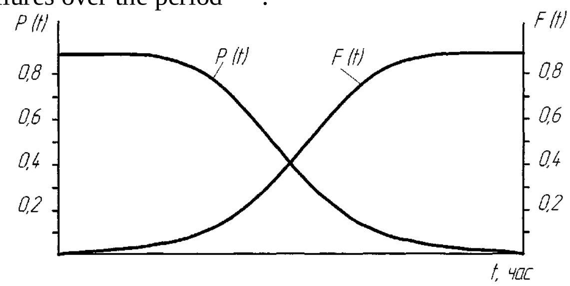

Fig. 1. The graph of the relative position of the probability of failure-free operation P(t) and failure F(t)

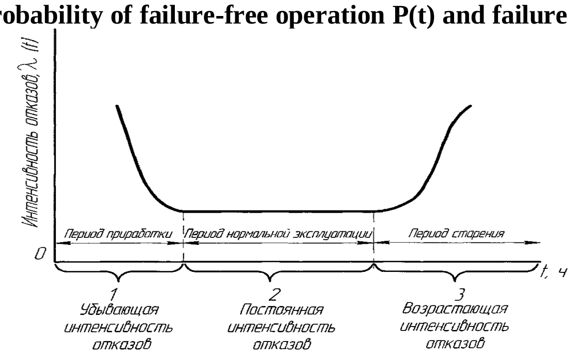

Fig. 2. Graph of changes in the rate of car failures

$$\boldsymbol{P(t)=exp\left[-\int_0^t f(t)d(t)\right]} \quad (7)$$

or

$$\boldsymbol{P(t)=e^{-\int_0^t (t)dt}} \quad (8)$$

The failure rate $\lambda(t)$ is the main indicator of the reliability of elements of complex systems, which is a car. This is due to the following circumstances:

– by the known intensity $\lambda(t)$, it is easy to evaluate the remaining reliability indicators, both of its elements and in general;
– the failure rate $\lambda(t)$ has good clarity;
– the failure rate is not difficult to obtain experimentally.

The failure rate changes its value as the vehicle is operated and is described by the U-shaped curve shown in Fig. 2. which has three characteristic sections: the running-in period, the period of normal operation, the period of wear [10,11].

The vehicle's running-in period is characterized by a decreasing failure rate. This intensity is due to defects in manufacturing, assembly, and adjustment. Often the duration of this period is associated with the terms of the warranty service of the object when the elimination of failures is made by the manufacturer. During normal operation, the failure rate remains practically constant $\lambda(t) \approx \text{const}$. This period begins immediately after the running-in period and ends immediately before the aging period, while failures are random and appear suddenly due to random load changes, non-compliance with operating conditions, unfavorable external factors, etc. It is this period that forms the basis of the vehicle's service life. An increase in the failure rate is observed during the aging period of the car, which is accompanied by catastrophic wear, which $\Lambda(t)$ grows rapidly.

B. Analysis of the causes of vehicle failures and their classification.

Failures are due to the occurrence, the nature of the manifestation, the relationship, groups of complexity, the method of detection, the method of elimination, and there are also resource and degassing. Due to the occurrence of failures, they are divided into structural, production, and operational. Constructive failures arise due to imperfection or violation of established rules and (or) norms for the design of an object. Production - arise as a result of the improper designation of technological processes for the manufacture or restoration of parts and assembly of a car or are the result of a violation of the adopted technology, unsatisfactory quality of the material of parts or coatings applied to them, the use of insufficiently accurate measuring instruments, non-compliance with technical requirements for the manufacture and assembly of elements, as well as and the manufacture and assembly of elements and the object as a whole. Operational failures occur due to the use of facilities in conditions for which they were not intended, violations of operating rules (unacceptable overloads, failure to comply with maintenance rules, untimely adjustments, use of non-compliant fuel and lubricants, failure to comply with transportation and storage rules). By the nature of the manifestation, failures are divided into gradual, sudden, and intermittent. Gradual failure occurs as a result of a gradual change in the values of one or more given object parameters. The reasons for this failure are natural aging and wear (increase in clearances, weakening of landings). Typical examples of gradual engine failures include extreme wear of parts and connections, increased oil consumption, low pressure in the lubrication system, reduced power, etc.

A sudden failure is characterized by an abrupt change in the values of one or more specified parameters of the object. Its main feature is the probability of occurrence $F(t)$ within a given period. Examples of such failures include thermal cracks in part due to an interruption in the lubricant supply; part breakdowns due to improper operating methods of the machine or the occurrence of overloads; deformation or breakage of parts that have fallen into unforeseen working conditions. In this case, the failure occurs, as a rule, suddenly, without previous symptoms of destruction and does not depend on the degree of deterioration of the product. An intermittent failure is a recurring self-eliminating failure of an object of the same nature; it occurs many times and is eliminated by itself. According to the relationship, failures are divided into independent and dependent. Independent failure is a failure of an object that is not caused by a failure of another object, and dependent is a failure not caused by other failures. An independent failure of an element is caused by the loss of the operability of this particular element and is not a consequence of the loss of operability of another element of the technical system.

According to the groups of complexity, failures are divided into three groups. Failures of the first group of complexity are eliminated by replacing or repairing parts located outside the units or assembly units. The second group - by replacing or repairing easily accessible assembly units and units with the opening of the internal cavities of the main units. Failures of the third group of complexity are eliminated by disassembling the main units in the conditions of repair and technical enterprises. The method of detection distinguishes between explicit and latent failures. Explicit failure is detected visually or by standard methods and means of control and diagnostics during the preparation of the object for use or in the process of its intended use. Latent failure cannot be detected visually or by standard methods and means of monitoring and diagnostics. It is detected during maintenance or special diagnostic methods. In addition to the generally accepted listed criteria by which failures can be classified, an additional criterion should be introduced at the place of failure elimination. The place of elimination of the failure can be a vehicle passport (VP), a private garage, parking lot, or carriageway where the failure occurred. The urgency of eliminating the failure on the roadway is associated with the growing popularity of cars with an automatic transmission, towing of which is, as a rule, prohibited, and evacuation requires additional time and money.

C. Investigation of the factors affecting the heat transfer of the radiator core.

Analysis of the methods for repairing radiators showed that any of them provides for a change in the conditions of heat transfer. The use of polymer materials for the repair of radiators is no exception. To reduce the effect of the proposed repair method on reducing the efficiency of heat transfer, knowledge and understanding of the patterns of heat transfer that occur in the radiator are required. For automobile engines, the parameters of the radiator of the cooling system are chosen to provide the required heat dissipation when the car is moving in an increased gear at a low speed of 12-15 km/h and an ambient temperature of +40 °C [12].

When calculating the design of radiators, three groups of indicators are used. General data on the core of the radiator.

H - radiator height;

B - the width of the radiator;

l - radiator depth;

F_{fr} - a frontal area of the radiator;

F_{coo} - total cooling surface area.

The geometry of the cooling surface elements.

b - the size of the cooling tube in the depth of the radiator;

c - the size of the cooling tube along the front of the radiator;

t_{fr} - step along the front;

t_{dep} - depth step;

δ_{tube} - cooling tube wall thickness;

δ_{pl} - cooling plate thickness.

Coefficients that characterize the layout feature of the radiator design:

- the compactness of the radiator is estimated by the volumetric compactness factor (φ).

$$\varphi = \frac{F_{cool}}{V_{cool}}, m^2 / m^3 \tag{9}$$

Where V_{coo} - the geometric volume of the radiator.

- the structure and shape of the cooling surfaces are determined by the finning coefficient (ξ_{finn}), which is determined by the ratio of the cooling surface washed by the cooling air (F_cool) to the cooling surface washed by the coolant (F_{liq}):

$$\xi_{rib} = \frac{F_{cool}}{F_{liq}} \tag{10}$$

In tubular-plate radiators, the cooling tubes can be arranged in relation to the flow of cooling air in a row, staggered and staggered at an angle (Fig. 1). Their designs use flat plates or plates with bent notches (turbulators), which form narrow and short air channels located at an angle to the flow of cooling air. The compactness factor of tubular-plate radiators, depending on the core design, is 440-850 m^2/m^3, and the finning factor is 2.5-5.4.

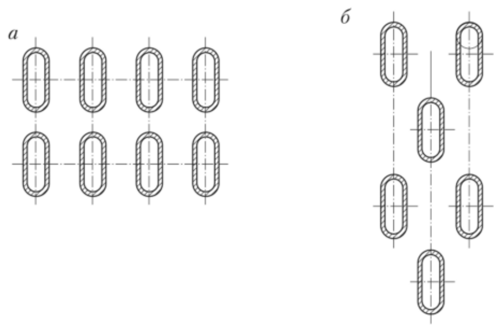

Fig. 3. Arrangement of tubes in radiators.
a) in-line arrangement of tubes b) staggered arrangement of tubes

In the tubular-strip radiator cores, the design of the cooling tubes practically does not differ from the tubes used in the tubular-plate radiators, but they are located only in a row. Tubular-strip radiators have a higher value of both the compactness factor, equal to 1100-1400 $m^2/m^{3,}$ and the finning factor, equal to 5-11.5.

Depending on the design of the radiator, they are set by the dimensions of the cooling elements, for tubular-plate radiators (fig. 2).

The size of the cooling plate t_{fr} along the front of the radiator is chosen in such a way that a finite number of cooling elements are located per size (B) in one row of the radiator. The height of the element (h) is selected in the same way, but with such a calculation that a finite number of cooling elements fit into the dimension (H) on one cooling tube. For tubular-tape radiators (Fig. 3), they are additionally set with a corrugation pitch t_1, but with such a calculation that a finite number of elements fit into the size of the radiator H on one cooling tube. To calculate the area of the cooling surface, take the surface of the calculated element of the radiator. For tubular-plate radiators

$$F'_{fre} = \left[2(b+c)h - 2(b+c)\delta_{pl} + 2\left(t_{fr}\cdot t_t - bc\right)\right] \quad (11)$$

For tape-and-tube radiators

$$F'_{fre} = \left[4l_t \cdot t_t + 2(b+c)\cdot t_t\right] \quad (12)$$

Knowing the area of one cooling element, the area of the cooling surface of the radiator is determined, which can be determined based on the expression

$$F = \frac{\gamma \cdot Q_l}{K\left(T_{mi.l} - T_{mi.air}\right)} \quad (13)$$

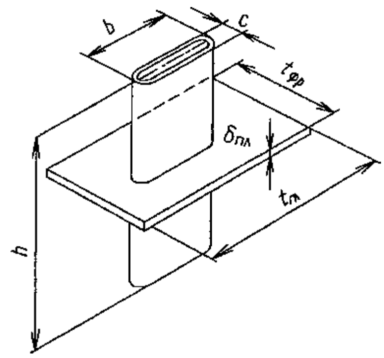

Fig. 4. Cooling element of the grille of tubular-plate radiators.

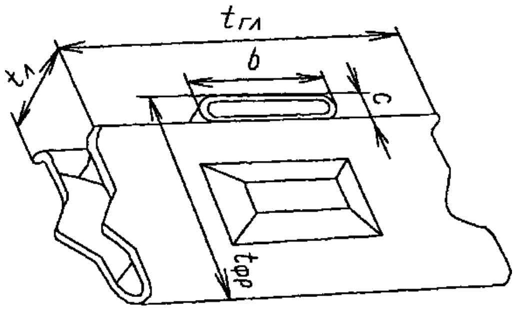

Fig. 5. Cooling element of the radiator grille.

where $\gamma = 1{,}1$ operational safety factor;

Q_{liq} - the amount of heat removed by the liquid, J/s;

K - the heat transfer coefficient of the radiator, $W/(m^2K)$;

$T_{av.liq}$ - the average temperature of the liquid in the radiator, K;

$T_{av.air}$ - the average temperature of the air passing through the radiator,

The use of a polymer material significantly reduces the cooling area of the design element, which will be equal to $ht_{fr.}$ However, if the design of the radiator and the location of the damage allows, then the cooling area can be increased due to the ribbing of the restored section. Analysis of the initial parameters for calculating the parameters shows that the radiator is designed for extreme operating conditions at an ambient temperature of +45 °C and an atmospheric pressure of 715 mm Hg. Art. In this case, the radiator area is calculated with an operational safety factor. Even though the use of the proposed method provides for a reduction in the area of the cooling surface, the initial parameters for calculating the radiator make it possible to repair a certain percentage of the radiator area using the proposed method. The permissible percentage of the repair area must be determined experimentally [13,14].

D. Substantiation of the boundary values of the criterion of the radiator operability in operation.

The amount of heat removed by the engine into the coolant is determined by [15,16]:

$$Q_{eng} = 632 a N_e \quad (14)$$

where:

$$a = \frac{Q_{eng}}{632N_e} = f\left(N_e, n_{eng}, t_L^{'}, t_W^{"}, G_W\right) \quad (15)$$

Where: a - experimental coefficient; N_e - engine power, W; n_{en} - crankshaft rotation frequency, rpm; t_L', t_W'' - air temperature at the inlet and into the radiator and liquid at the exit from the radiator, °C; G_W - mass airflow, kg/s. Coefficient a, which reflects the ratio of the heat transferred to the coolant to the heat converted into useful work, cannot serve as a convenient criterion for the performance of a radiator due to the complex dependence in real operating conditions on many factors. The values of this coefficient fluctuate over a wide range. For the maximum load mode, a = 0.8 ... 1.4 (for carburettor engines), and a = 0.45 ... 0.9 (for diesel engines). For guaranteed heat dissipation by the radiator, the maximum values of the coefficient are taken: a = 1.4 (for carburetor ICEs) and a = 0.9 (for diesel ICEs).

Thus, the limiting values of the heat transfer of the radiator:

$$Q_r^{carb} \geq 885N_{e\,max}\, u\, Q_r^{dis} \geq 569N_{e\,max} \quad (16)$$

However, the criterion values of the heat transfer of the radiator performance according to the formula (8) are suitable for use only for the case of the radiator functioning as part of the cooling system. Internal combustion engines. When the radiator is dismantled from the car to measure its heat transfer at the stand, the regime of the aerodynamic and hydraulic flow of coolants will change significantly. It is known that the potential properties of a radiator in an automobile will depend on many factors. A quantitative measure for assessing the degree of reduction in heat transfer when installing a radiator on a car is:

$$Q_{en} = Q_r = \chi_\Sigma Q_r^{'} \quad (17)$$

Where: Q_r', is the potential heat transfer of the radiator, kW; χ_Σ is the integral coefficient for the realization of the potential properties of the radiator as part of the car. The parameters affecting the realization of the potential properties of the radiator, taken into account by the coefficient χ_Σ, include:

a) the coefficient of realization of thermal properties during operational pollution, numerically equal to the coefficient of purity η_3 [17,18]:

$$x_1 = \eta_3 = \frac{k_r}{k_0} \quad (18)$$

b) the nosing coefficient, which is numerically equal to the value of the decrease in heat transfer due to additional aerodynamic resistance Δ_{pL} caused by a change in the aerodynamic drag mode of the radiator when it is installed on the car:

$$\chi_1 = \eta_k \quad (19)$$

In this case, criterion equations (18) for a radiator diagnosed at the stand will have the form:

$$Q_r^{'karb} \geq \frac{885N_{e\,max}}{\chi_\Sigma}\, u\, Q_r^{'dis} \geq \frac{569N_{e\,max}}{\chi_\Sigma}$$

or

$$Q_r^{'karb} \geq \frac{885N_{e\,max}}{\eta_3 + \eta_\kappa}\, u\, Q_r^{'dis} \geq \frac{569N_{e\,max}}{\eta_3 + \eta_\kappa} \quad (20)$$

E. Substantiation of a method for quantitatively assessing the performance of radiators in operation

a) Theoretical foundations for the quantitative determination of heat transfer under conditions of forced movement of coolants.

For different heat carriers (in this case, the coolant), heat transfer occurs differently and in a peculiar way. For each of them, physical properties are a function of temperature, and some also of pressure. The mathematical description of the heat transfer process consists of:

- heat conduction equations;
- equations of motion;
- complexity equations;
- heat transfer equations;
- uniqueness equations.

To date, analytical solutions of the system of differential equations of convective heat transfer have been obtained only for a limited number of simplest problems with the introduction of certain simplifying assumptions. This situation is explained by the great complexity of the equations, as well as the complexity and versatility of the described processes themselves. Due to the limited possibilities of the analytical solution of the above differential equations, the experiment is of great importance in the study of heat transfer processes. Experimental study of complex heat engineering processes, depending on a large number of factors, has a high cost, duration, and labor intensity.

For a certain class of experimental problems in which heat transfer occurs under conditions of forced movement of coolants, the similarity theory is used. Such processes of heat transfer can be considered as those under which the condition is fulfilled: $\mathrm{Re} = idem$; $\mathrm{Pr} = idem$. In this case, the Reynolds number (Re) determines the hydromechanical similarity of coolant flows:

$$\mathrm{Re} = \frac{\omega_0 l}{v} \quad (21)$$

Where ω_0 is the characteristic, usually the average speed of movement of a liquid or gas at the initial section of the system; l is the characteristic geometric size of the system; v is the coefficient of kinematic viscosity of the coolant. The Prandtl number (Pr) is a thermophysical characteristic of the coolant. It is composed only of physical parameters:

$$\mathrm{Pr} = \frac{\mu c_p}{\lambda} = \frac{\mathrm{v}}{a} \quad (22)$$

Here: $\upsilon = \mu/\rho$ and $a = \lambda/c_p\rho$ - thermal diffusivity, the numerical value of which is given in the tables. If the

Re numbers are equal, the condition for the *Pr* numbers being equal ensures thermal similarity, i.e., the similarity of the fields of temperature heads and heat fluxes in the entire volume of the systems under consideration. According to the theory of similarity, similar processes should have the same determined similarity numbers. In the processes of convective heat transfer, the Nusselt number Nu, which characterizes the intensity of the process of convective heat transfer, is used as the determinable one:

$$Nu = \frac{\alpha l}{\lambda} \qquad (23)$$

Thus, the condition for the identity of the similarity numbers (Pr, Re = idem) is the condition for the invariance of the defining similarity numbers. This ensures the similarity of processes. The similarity equation for convective heat transfer processes with the forced movement of the coolant, which is inherent in the working process of the internal combustion engine cooling radiator, has the form:

$$Nu = f\left(\mathrm{Re},\ \mathrm{Pr}\right) \qquad (24)$$

However, as noted by the majority of experts, the use of similarity criteria is possible only if the physical parameters of the medium and thermal engineering constants are strictly maintained. With a significant change in properties, a strict similarity of various processes, as analysis shows, in the general case becomes impossible. These circumstances do not allow the use of analytical dependences for the working process of the radiator when constant and stochastic changes in the flow regimes of coolants occur.

b) Analytical description of the heat transfer process of the radiator.

Based on the well-known formula that quantitatively describes heat transfer:

$$Q_{p\tau} = k_T F \overline{\Delta} t_{\log} \qquad (25)$$

it follows that with known values of the area *F* and the mean logarithmic head t_{log}, it is possible to quantify the intensity of heat transfer by the value of the heat transfer coefficient k_T:

$$k_r = 1/\left[\left(\frac{1}{a_w} + \frac{\delta}{\lambda}\right)\cdot\psi + \frac{1}{\alpha_L}\right] \qquad (26)$$

where: a_w, α_L - coefficients of heat transfer from the side of liquid and air, respectively, W/m ·°C;

ψ - coefficient of ribbing;

δ — wall thickness of the cooling element (tube), m;

λ - coefficient of thermal conductivity, W/(m*°C).

For a clean wall of the heat-transfer surface, it was found that the influence of the component $\delta\psi/\lambda$, which is the thermal resistance of the structural material of the cooling tube, is only about 0.5% of the total thermal resistance, and it is neglected. Expression (13) is simplified:

$$k = \frac{1}{\dfrac{1}{\alpha_w} + \dfrac{1}{\alpha_L}\psi} \qquad (27)$$

Under operating conditions, the value of the heat transfer coefficient decreases due to the accumulation of various contaminants and defects in the structure of structural materials on the working surfaces. Figure 6 shows the diagrams of the heat transfer process through the clean (a) and dirty (b) walls.

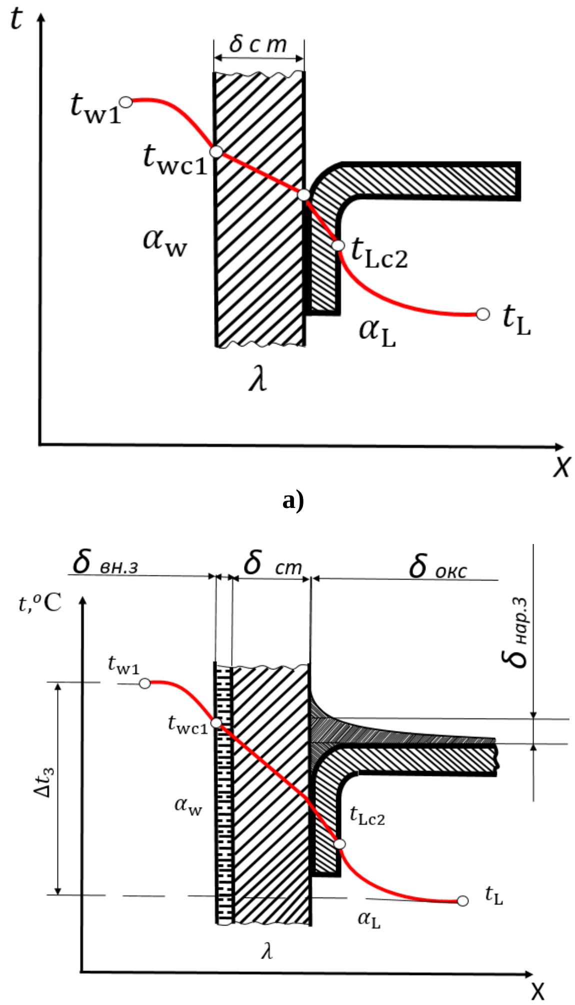

Fig. 6. Heat transfer through a clean (a) and dirty (b) radiator wall:

t - temperature °C; α - heat transfer coefficient W/m² °C; λ - coefficient of thermal conductivity, W/(m°C); ψ - coefficient of ribbing; δ — wall thickness, m; κ - heat transfer coefficient, W/m² °C; R - thermal resistance, m² °C/W; ξ3 is the coefficient of contamination of the finning surface. O - initial value; τ is the current value; вн.з - internal pollution; нз - external pollution; Art. - wall; W - cooling medium - water; L - cooling medium - air; окс. - oxide and other films; з - pollution.

Contamination in the form of deposits formed on the surfaces of radiators, as a rule, has low thermal conductivity and high thermal resistance. In this regard, the influence of the coefficient of thermal conductivity of the contaminated wall λ_{pol} cannot be ignored due to the low values of the coefficients of thermal conductivity of the main pollutants. It is enough to give this value for scale: $\lambda_{pol} = 1{,}32 \dots 3{,}4$, which is two orders of magnitude lower than brass, as the main structural material of cooling tubes. Therefore, for a dirty wall, the heat transfer coefficient will take the form:

$$k = \frac{1}{\frac{1}{\alpha_w} + \frac{\delta_{ex\,p}}{\lambda_{in\,p}} + \frac{\delta_{wa}}{\lambda_{\text{ст}}} + \left(\frac{\varepsilon_з \delta_{ex\,p}}{\lambda_{\text{ex p}}} + \frac{1}{\alpha_L}\right)\Psi} \qquad (28)$$

Here: $\lambda_{in\,p}, \lambda_{ex\,p}, \lambda_{wa}$ - coefficients of thermal conductivity of internal, external contaminants and walls of the cooling tube, W/m²•°C; $\delta_{in\,p}, \delta_{in\,p}, \delta_{wa}$ - thickness of internal, external contaminants and walls of the cooling tube; $\varepsilon_з$ - coefficient of surface contamination.

When considering the working process in a contaminated radiator, it is assumed that the following reasons lead to a decrease in the heat transfer coefficients α_w and α_L on the cooling surfaces:

1) deposition of a layer of pollution with high thermal resistance;

2) a decrease in the speed of the coolant (air or liquid), and sometimes even the termination of its flow into certain channels due to their complete blockage (most often in the initial section);

3) a change like the flow through the channels of the cooling surface due to local (local) deposits of the pollutant.

The listed reasons most often manifest themselves in aggregate, causing an integral negative effect, manifested in a decrease and loss of working capacity. To compensate for the influence of operational pollution at the design stage, a reserve $\Delta Q_{ppo} = \Delta Q_{po} - \Delta Q_{eng}$ of heat transfer is laid in an amount of at least 10% of the maximum calculated value of heat released from the internal combustion engine. The heat transfer reserve for a new (reference) radiator is a guarantee of its operable state throughout the service life with operating time t in the interval 0≤τ≤T:

Transformation

$$Q_{дв} \le Q_{дв} + Q_{ppr} \le Q_{p0} \qquad (29)$$

(16) when subtracting a constant gives:

$$0 \le \Delta Q_{pp\tau} \le Q_{pp0} \qquad (30)$$

After dividing (17) by ΔQ_{pp0}, we obtain the expression for the condition of the radiator operability in relative, dimensionless units:

$$0 \le q_{p\tau} \le l \qquad (31)$$

where $q_{p\tau}$ is a parameter reflecting the change in the heat transfer reserve of the radiator during operation.

The rate of the decrease in the heat transfer reserve from the operating time can be described by the following differential equation:

$$\frac{dq_{p\tau}}{d_\tau} = \frac{d(k_\tau F \Delta t)}{d_\tau}, \qquad (32)$$

moreover,

$$\mathrm{k_r} = \frac{l}{R_\tau} \qquad (33)$$

where: R_τ - total thermal resistance created by operational pollution, (m. °C)/W;

T - operating time to the limit state. It is natural to assume that $q_{pt} > 0$ is a decreasing function of the operating time. The solution of the differential equation (24) can give a very approximate result when using various methods of iteration or mathematical modeling of real operating conditions. Integration of the same equation gives a set of solutions depending on the initial conditions, the determination of which is possible only when the surfaces are cleaned to the initial level, which is very difficult. If the radiator is permanently malfunctioning under operating conditions, it is dismantled to restore its performance by cleaning. In this case, it becomes possible to clean the outer surface of the radiator and restore the shape of the finning plates of the air channels.

As a result, with a constant front area of the radiator (*F=const*), the influence of aerodynamic resistance on the mean log temperature head Δt is excluded, the differential equation (19) can be simplified:

$$\frac{dq_{p\tau}}{d_\tau} = \frac{dk_\tau}{d_\tau} = \frac{d\left(\frac{l}{R_\tau}\right)}{d_\tau} \qquad (34)$$

Statistical studies have shown that the total thermal resistance after the operating time of the radiator m under specific operating conditions will be:

$$R_r = R_{r\,max} \cdot \left(1 - e^{-B\tau}\right) \qquad (35)$$

Where: $R_{r\,max}$ - maximum total thermal resistance, to which contamination curves tend to asymptotically approach over time (at the maximum possible deposit thickness); *B* is a constant determined experimentally through the values of thermal resistances in the time interval of operation. In operation, along with the process of stochastic pollution of the cooling surfaces, they are periodically cleaned to a level determined by the cleaning methods and the nature of the accumulated pollution.

This process also has a stochastic nature, which introduces additional complexity in determining the heat transfer reserve of the radiator. Figure 7 shows a graphical interpretation of the process of contamination and cleaning of the radiator in operation.

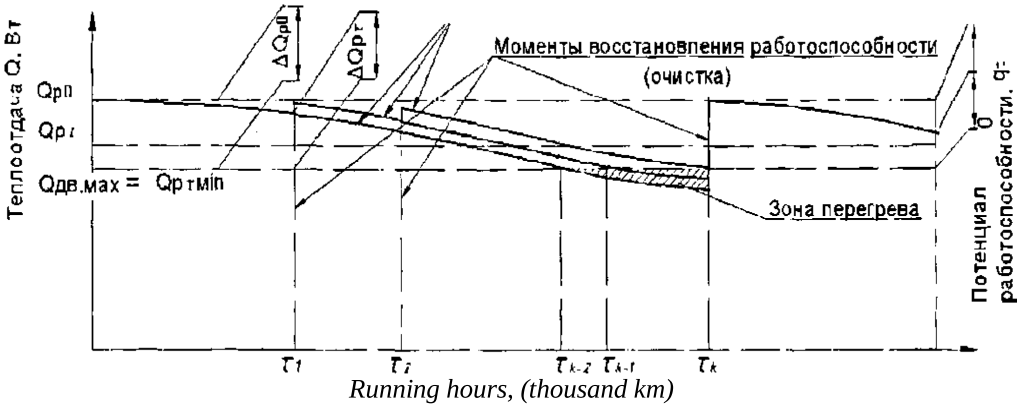

Fig. 7. Graphic interpretation of the process of contamination and cleaning of the radiator in operation:
Q_{p0}, $Q_{p\tau}$ - initial and current value of heat transfer from the radiator, W;
$Q_{en.\ max}$ maximum value of engine heat transfer, W;
$\Delta Q_{p0,}$ $\Delta Q_{p\tau}$ is the heat transfer reserve of the new and operating radiator, W;
τi - operating time of the radiator, thousand km

Thus, for a dirty wall, the heat transfer coefficient after a time interval τ is determined as:

$$k_r = \frac{l}{R_{cm\tau + \psi R_{w\tau} + R_{w\tau} + R_{L\tau}}}. \qquad (36)$$

Determination of partial thermal resistances will make it possible to determine the intensity constant of the pollution process:

$$\tau = \frac{\ln\left[\dfrac{l}{l - \dfrac{R_\tau}{R_{\tau max}}}\right]}{\mathrm{B}} \qquad (37)$$

Determination of the *B* coefficient makes it possible to predict the operating time of the radiator to the limiting state, at which it is necessary to carry out the appropriate preventive actions.

$$\tau = \frac{\ln\left[\dfrac{l}{l - \dfrac{R_\tau}{R_{\tau max}}}\right]}{B} \qquad (38)$$

The above formulas illustrate the limitations of their use for the analytical method for determining the operating time m to the limiting state of the radiator without experimental determination:

- heat transfer coefficients a_w and a_L for the liquid and air sides of the wall of a clean and, a_w , a_L of a dirty radiator;

- partial and total thermal resistances, R_{W_r}, R_{L_r}, R_τ;

- pollution intensity factor B for specific operating conditions.

F. Development of a mathematical model of the working process of a radiator in conditions of operational pollution

The development of a mathematical model of the radiator working process with a description of the structure of functional links between the varied composition of significant factors and the output parameter is presented in the form of a classic "black box" (Fig. 8).

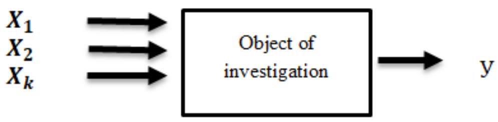

Fig. 8. Scheme of the research object used in the development of a mathematical model.

The use of the stated position allows using a mathematical model to express the functional relationship between the implementations of the input factors and the output parameter in the form of a functional

$$y = f\left(x_1, x_2, \ldots, x_n\right) \qquad (39)$$

Functional (40) characterizes the response surface in vector space. A polynomial equation of the following form was used as a response:

$$y = в_0 + \sum_{i=l}^{k} в_i x_i \qquad (40)$$

where *y* is the optimization parameter;

x_i - factors varied during the experiment, *i*=1,2,.. .*k*;

βo, βi - theoretical regression coefficients for linear interactions.

After establishing, as a result of an active experiment, the numerical values of the regression coefficients, equation (41) can be written as:

$$y = b_o + \sum_{i=1}^{k} b_i x_i \qquad (41)$$

Where y is the calculated value of the optimization parameter; b_o, b_i - estimates of the regression coefficients; k is the number of factors.

Based on possible cases of malfunction of the radiator in operation, namely:

- change in aerodynamic resistance due to contamination of the front of the radiator with coarse structural pollutants (organic fibers, leaves, needles, insects, etc.);
- change in hydraulic resistance due to contamination of the radiator tube sheet with various deposits;
- a change in the thermal conductivity of the wall due to the growth of fine-structured pollutants from the outside (road dust, oil-mud deposits, specific pollution, etc.);
- a change in the thermal conductivity of the wall due to the growth of various deposits on the inside (scale, corrosion products, etc.);
- change in aerodynamic resistance due to deformation of the inlet channels of the radiator due to mechanical damage to the finning of the tubes.

The accepted model for five factors will be:

$$y = b_o x_o + b_1 x_1 + b_2 x_2 + b_3 x_3 + b_4 x_4 + b_5 x_5 \qquad (42)$$

Where b_1, b_2, b_3, b_4, b_5 - coefficients for linear effects.

Thus, the diagram of the object of study in an active experiment is represented by a picture (Fig. 9).

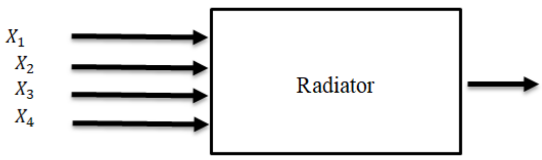

Fig. 9. Scheme of the research object

a) Ranking factors.

Based on the results of the analysis of scientific sources, the following factors were selected:

X_1 - additional aerodynamic resistance associated with clogging of the front of the radiator with a large-structured contaminant;

X_2 - additional hydraulic resistance associated with clogging of the inlet channels of the hydraulic path;

X_3 - fine-structured contamination of the outer surfaces of the radiator (road and soil dust);

X_4 - deposits on the internal surfaces of the radiators;

The developed questionnaire included the indicated five factors (k = 5), their dimensions, and levels of variation. The group included 8 specialists (m = 8), each of whom was assigned a conditional number.

Index and positions of the formed expert group:

1 - chief designer for automotive heat exchangers;
2- chief engineer of a motor transport company;
3 - scientist;
4 - metalworker-coppersmith;
5 - auto mechanic-repairman;
6 - driver A;
7 - driver B;
8 - driver V.

The results of the questionnaire are shown in Table 1.

TABLE I
RANK MATRIX

Эксперты	Факторы (к=4)			
(ш=8)	X,	X_2	X_3	x_4
1	1	5	3	4
2	1	5	3	4
3	1	5	3	4
4	1	5	3	4
5	2	4	1	5
6	2	5	3	4
7	1	4	3	5
8	1	4	3	5
$\sum^{m} a_y$	10	37	22	35
Δ_i	-14	13	-2	11
Δ_i^2	196	169	4	121

The sum of the ranks for each factor is shown in the matrix. Mathematical processing of the ranking results gave the following results:

$$T = \frac{10 + 22 + 35 + 37}{4} = \frac{120}{4} = 24$$

S= 196+4+121+169 = 554.

Concordance factor:

$$W = \frac{12 * 554}{64 * (125 - 4)} = 0{,}865$$

$$X^2_{расч} = \frac{12 * 554}{8 * 4 * 6} = 27{,}7$$

Accepted: a=0,05; f=4-1=3.

$X^2_{0,05;4} = 9{,}488$

$X^2_{calc} = 27{,}7 > X^2_{0,05;4} = 9{,}488$

The ratio of the critical and calculated values X^2 shows that, with a confidence level of 95%, the opinions of experts on the influence of factors on the optimization parameter are consistent with the concordance coefficient W = 0.865.

The constructed rank diagram is shown in Fig. 10.

The diagram shows that the distribution of factors corresponds to an uneven exponential law. Factors X_1, X_3, and X_4 can be attributed to the noise field and excluded from the experiment since the vast majority of modern cooling systems run on antifreeze.

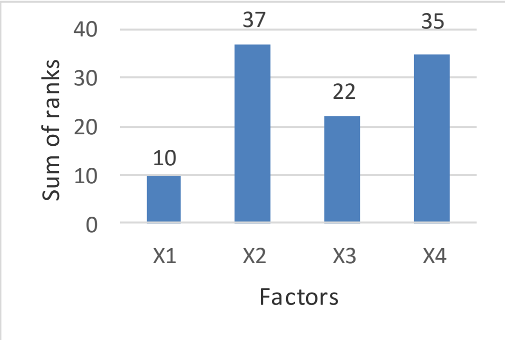

Fig. 10. Rank chart

In this case, the problem of intensive scale formation on the inner surfaces of the radiator does not arise. The deformation of the fins for most of the radiator designs is a legally correctable defect and can be eliminated by straightening even without dismantling the radiator.

b) Significant factors, their dimension, and levels of their variation.

The factors, their dimensions, and the levels of their variation affecting the heat transfer of the radiator are reflected in Table 2.

TABLE III
FACTORS, THEIR DIMENSIONS, AND LEVELS OF VARIATION AFFECTING THE HEAT TRANSFER OF THE RADIATOR

Factors	Lower level -1	Top-level + 1	Main level 0	Variation interval
X_1 - additional resistance associated with additional material	0	12	6	5
X_2 - additional hydraulic resistance associated with clogging of the inlet channels of the hydraulic path,%	0	22	11	11
X_3 - fine-structured contaminatio n of the lower and upper radiator tanks	0	33	21	24

.

c) Regression equation and matrix for planning a full-factor experiment.

For the linear model used in what follows.

$$y = b_0 + b_2 x_2 \qquad (43)$$

An estimate of the regression coefficients was established, and its adequacy to the experimental data was checked. Variations of factors are accepted at two levels. The number of experiments $N = 2^4 = 16$, the number of parallel observations in each experiment n = 3. The total number of experiments is 24.

The optimization parameter in equation (38), which is influenced by the above factors, is the heat transfer of the radiator, Q

The matrix for planning a full-factor experiment in coded values and a matrix with named values of factors and the results of the experiment are presented in the table. 3 and 4.

TABLE IIIII
EXPERIMENT PLANNING MATRIX

Experience number	X_0	X_1	X_2	X_3	$\bar{Y}_u$
1	+	-	-	-	$\bar{Y}_1$
2	+	+	-	-	$\bar{Y}_2$
3	+	-	+	-	$\bar{Y}_3$
4	+	+	+	-	$\bar{Y}_4$
5	+	-	-	+	$\bar{Y}_5$
6	+	+	-	+	$\bar{Y}_6$
7	+	-	+	+	$\bar{Y}_7$
8	+	+	+	+	$\bar{Y}_8$

TABLE VIV
MATRIX FOR PLANNING THE EXPERIMENT IN THE NAMED VALUES OF THE FACTORS

Experience number	X_0	X_1	X_2	X_3	$\bar{Y}_u$
1	+	0	0	0	$\bar{Y}_1$
2	+	+50	0	0	$\bar{Y}_2$
3	+	0	+10	0	$\bar{Y}_3$
4	+	+50	+10	0	$\bar{Y}_4$
5	+	0	0	+0,005	$\bar{Y}_5$
6	+	+50	0	+0,005	$\bar{Y}_6$
7	+	0	+10	+0,005	$\bar{Y}_7$
8	+	+50	+10	+0,005	$\bar{Y}_8$

Thus, this chapter presents an analysis of the factors influencing the change in the technical condition of the radiator during operation; general view of the mathematical model describing the influence of operating factors on the performance of the radiator; plan for conducting a full-factor experiment, the purpose of which is to clarify the coefficients included in the mathematical model.

III. RESULTS AND DISCUSSION

The assessment of the degree of clogging of the tube sheet was carried out by direct measurement of the area of the free cross-section of each of the tubes and calculation after summing up the relative area of clogging $\Delta S_{W\tau}$ according to the method, the measurement results are summarized in the table.

Fig. 11. Fragment of the radiator tube sheet in the area of the inlet pipe

TABLE V
RESULTS OF MEASUREMENTS OF THE DEGREE OF CLOGGING OF THE TUBE SHEET OF RADIATORS

Radiator no.	Measurement results			Radiator no.	Measurement results		
	S_{w0}, мм^2	$S_{w\tau}$, мм^2	$\Delta S_{w\tau}$, %		S_{w0}, мм^2	$S_{w\tau}$, мм^2	$\Delta S_{w\tau}$, %
1	8090	7184	11,2	16	7562	7145	5,5
2	7562	7297	3,5	17	7562	6790	10,2
3	7562	6458	14,6	18	8090	7532	6,9
4	8968	8060	10,1	19	8968	8452	5,8
5	7562	5898	22,0	20	7562	6252	17,3
6	4924	4456	9,5	21	7562	5792	23,4
7	8968	8555	4,6	22	8090	6181	23,6
8	4924	4238	13,9	23	4924	4650	5,6
9	7562	7123	5,8	24	4924	4235	14,0
10	7562	7012	7,2	25	7562	6400	15,4
11	7562	6508	13,9	26	8968	8604	4,1
12	7562	6418	15,1	27	7562	7040	6,9
13	8090	7063	12,7	28	7562	6728	11,0
14	8968	7984	11,0	29	4924	4215	14,4
15	8968	7744	13,6	30	4924	4530	8,0

Chevrolet Spark radiator; Chevrolet Aveo; Chevrolet Cobalt; Chevrolet Gentra;

A graphical representation of the experimental data is shown in Figure 12.

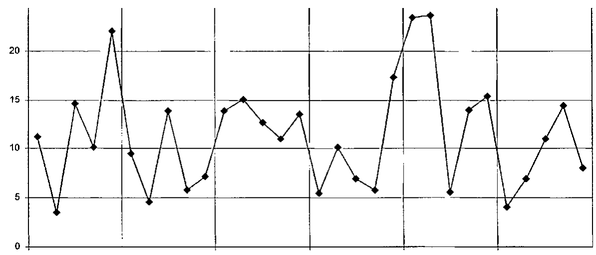

Fig. 12. Graphical representation of the measurement data of the channel area of the hydraulic path of radiators with operating time blocked by operational contamination (%)

The average value of the additional hydraulic resistance associated with clogging of the tube sheet channels is taken as a zero level when simulating operational pollution and conducting a full-factor experiment.

IV. CONCLUSIONS

Analysis of the obtained experimental results shows:

1. Implementation of the developed method for quantitative assessment of the integral characteristics of radiators - heat transfer, confirms the proposed theoretical provisions.

In particular, the method makes it possible to determine the coefficient of realization of thermal properties in terms of contamination, which is numerically equal to the coefficient of purity $X_{2}= \eta = \frac{\kappa 2}{\kappa}$, which is the initial parameter for the criterion equation for the performance of radiators in operation.

2. The developed diagnostic stand allows registering the change in heat transfer under the conditions of a simulated heat load with the implementation in the numerical and graphical form of the regularity: $Q = {}_{z}P_{\Sigma} = f(\tau,\Delta t)$, according to which, at Δt = *const* in the state of balance of the heat and mass transfer process, it is possible to establish the numerical value of heat transfer $Q_{\cdot p\tau}$ of the heat transfer coefficient kτ and thermal resistance R_τ.

3. The developed method and diagnostic stand for determining the operability of radiators allows you to

create a mathematical model of the working process of heat transfer from a radiator.

4. The resulting coefficients of the linear regression equation are obtained, reflecting the degree of influence of each of the factors on the integral parameter of heat transfer from radiators.

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