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THERMOHYDRAULIC CHARACTERISTICS OF REFRIGERATOR-RADIATOR DEVICES

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UDC 66.045.1:597.385.3

Generalized relations are derived for determining the thermal efficiency and the outlet temperature of tubes in radiative heat exchangers with a nonuniform distribution of heat carrier between these tubes.

Finned tubular refrigerator-radiator devices have found broad applications in various branches of engineering [1]. A segment of such a heat exchanger consists of a distributor and a collector connected through a row of tubes with crosspieces between them. Finning of tubes through which the heat carrier flows with heat-emitting crosspieces makes it possible to substantially reduce the metal content of a radiator and improve the energy characteristics of the overall heat-exchanger system.

In this study the thermohydraulic characteristics of radiators and the effect of a nonuniform distribution of heat carrier between their tubes on their thermal efficiency will be considered.

The thermal characteristics of radiative refrigerators with a uniform distribution of heat carrier between tubes have already been studied [2, 3]. In one study [3] the temperature fields over the width of a crosspiece as well as the dependence of the thermal efficiency of a radiator element with finning (Fig. 1) on thermophysical and geometrical parameters of tubes and crosspieces in the case of radiators with black surfaces were determined. Analogous studies were made [2] of diffusely emitting and absorbing gray surfaces.

Elsewhere [4] the results of studies made pertaining to the efficiency of finned tubular radiator elements with tubes at unequal temperatures were reported. The thermal efficiency of a radiator element η_e is defined as the ratio of the amount of heat emitted by it into the surrounding space to the amount of heat emitted by a perfectly black plate of width 2(R + L) at a temperature equal to that of the hotter tube.

The dimensionless temperature field over the width of a crosspiece is described by the equation

$$\frac{d^2\Theta_f}{d\bar{X}^2} = \varepsilon N (\Theta_f^4 - \beta_{f\,inc}) \quad (1)$$

with the boundary conditions

$$X = 0,\ \Theta_f = \Theta_1 = 1,\ X = 2,\ \Theta_f = \Theta_2. \quad (2)$$

M. I. Kalinin Dnepropetrovsk Institute of Railroad Transportation Engineers. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 41, No. 5, pp. 880-887, November, 1981. Original article submitted July 2, 1980.

The boundary-value problem (1)-(2) was simultaneously solved with the system of integral equations describing the radiative heat transfer from a given radiator element. The amount of heat reradiated from an element can be expressed in terms of the effective radiant fluxes from the tubes, these fluxes being functions of the angular coordinates α_1 and α_2:

$$\beta_{\mathrm{finc}} = \int_0^{\alpha_1(X)} \beta_{\mathrm{ef}_1} F_{X-\alpha_1} + \int_0^{\alpha_2(X)} \beta_{\mathrm{ef}_2} F_{X-\alpha_2}. \tag{3}$$

On the basis of energy balance, the effective radiant fluxes from the crosspiece and the tubes can be expressed through the relations

$$\beta_{\mathrm{ef\,f}} = \varepsilon\Theta_{\mathrm{f}}^4 + (1-\varepsilon)\beta_{\mathrm{finc}}, \tag{4}$$

$$\beta_{\mathrm{ef}_1} = \varepsilon\Theta_1^4 + (1-\varepsilon)\left\{\int_0^{\alpha_2(\alpha_1)} \beta_{\mathrm{ef}_2} F_{\alpha_1-\alpha_2} + \int_{X(\alpha_1)}^{2L} \beta_{\mathrm{ef\,f}} F_{\alpha_1-X},\right. \tag{5}$$

$$\beta_{\mathrm{ef}_2} = \varepsilon\Theta_2^4 + (1-\varepsilon)\left\{\int_0^{\alpha_1(\alpha_2)} \beta_{\mathrm{ef}_1} F_{\alpha_2-\alpha_1} + \int_0^{X(\alpha_2)} \beta_{\mathrm{ef\,f}} F_{\alpha_2-X}.\right. \tag{6}$$

The elementary angular coefficients of irradiance F_{i-j} for long cylindrical surfaces were determined from the well-known relation $F_{i-j} = {}^1/_2 d(\sin\varphi)$, suitable for calculating the angular coefficients of irradiance between infinitesimally small surface elements [3]. For instance, the elementary angular coefficient F_{α_1-X} is

$$F_{\alpha_1-X} = \frac{R\sin\alpha_1\,[x\cos\alpha_1 - R(1-\cos\alpha_1)]\,dx}{2\,[x^2 + 2R(x+R)(1-\cos\alpha_1)]^{3/2}}. \tag{7}$$

The nonlinear system of integrodifferential equations (1)-(7) was solved with the aid of a computer. System (3)-(7) was solved by the iteration method and system (1)-(2) was solved by the Runge–Kutta method.

The relations $\Theta(N, R/L, x/L, \varepsilon)$ and $\eta_e(N, R/L, \varepsilon, \Theta_2)$, based on calculations for several variants, are shown in Figs. 2 and 3. From these relations the temperature field in a radiator segment and the thermal efficiency of the latter when the tubes are at unequal temperatures because of a nonuniform distribution of heat carrier between them can be determined. First we will examine the reduction of thermal efficiency of a heat emitting element whose width is $2(R + L)$.

The amount of the heat emitted by a radiator element when $\Theta_1 = \Theta_2 = 1$ is

$$Q_0 = 2\sigma T_1^4 (R+L)\,\eta_{e0}. \tag{8}$$

When the tubes are at unequal temperatures, then the amount of heat emitted by a radiator element at the temperature of the first tube T_1 is

$$Q_1 = 2\sigma T_1^4 (R+L)\,\eta_{e1}. \tag{9}$$

The ratio Q_1/Q_0 characterizes the reduction of thermal efficiency of a radiator element due to a nonuniform distribution of heat carrier between the tubes, viz.

$$\psi_e = Q_1/Q_0 = \eta_{e1}/\eta_{e0}. \tag{10}$$

As the heat carrier flows through the radiator tubes, its temperature and thus also the temperature of the heat emitting surface decrease along the tubes from T_{in} to T_{out}. The amount of heat emitted by a tube is in this case determined by the mean-integral temperature $\bar{T}$, which can be calculated as follows. With the temperature difference between heat carrier and tube wall disregarded, the temperature variation in the heat carrier along a heat emitting tube is described by the differential equation of heat transfer

$$\frac{dT}{dx} = -\frac{\varepsilon\sigma}{Gc_p} T^4 P. \tag{11}$$

The solution to Eq. (11) yields the temperature distribution over the tube length

$$x = \frac{Gc_p}{3\varepsilon\sigma P T_{in}^3}\left(\frac{T_{in}^3}{T^3} - 1\right). \tag{12}$$

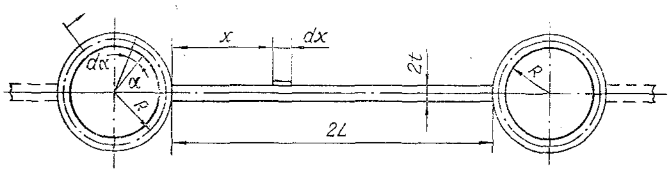

Fig. 1. Schematic diagram of a finned tubular radiator element.

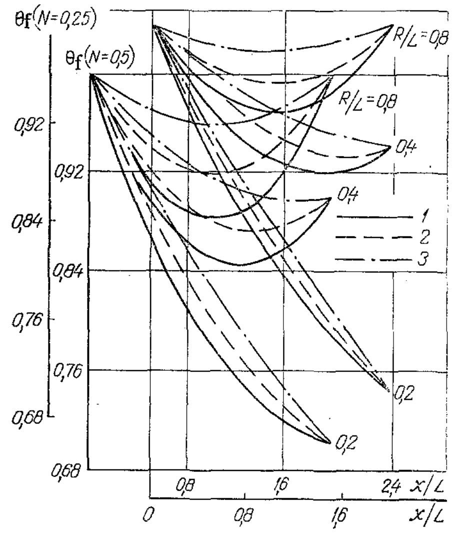

Fig. 2. Temperature field over the width of the crosspiece in a radiator element of infinite length with N = 0.25 or 0.5: 1) ε = 0.9; 2) ε = 0.5; 3) ε = 0.2.

The expression for the amount of heat emitted by a radiator tube can then be written as

$$Q_t = \varepsilon\sigma\bar{T}^4 PL, \tag{13}$$

where $\bar{T} = \bar{T}_{in}\left(\frac{3\beta^3}{1+\beta+\beta^2}\right)^{0.25}$ is the mean-integral temperature of the heat emitting tube and $\beta = T_{out}/T_{in}$.

The effect of a nonuniform distribution of heat carrier between the radiator tubes on the thermal efficiency will henceforth be analyzed on the basis of mean-integral temperatures of tubes.

A lower flow rate of heat carrier in any one tube (say the second tube) results in a lower temperature T_{out} of the heat carrier at the outlet from this tube and, consequently, in a lower mean-integral temperature $\bar{T}_2$. Therefore, to Eqs. (8)-(10) must be added closing equations which describe the temperature fields of the tubes, the dependence of the thermal efficiency η_e of a radiator element on the difference between temperatures Θ_1 and Θ_2 of the respective tubes and on the thermophysical characteristics of the crosspieces, and the dependence of the hydraulic nonuniformity of heat carrier distribution between tubes η_h on the layout of a radiator section (Z- or Π-connection into the system), on the geometrical dimensions of distributor and collector and of tubes, on the location of a given tube and radiator section, etc.

These relations will be expressed as

$$\bar{T}_1 = T_{in}\,[3\beta_1^3/(1+\beta_1+\beta_1^2)]^{0.25},$$
$$\bar{T}_2 = T_{in}\,[3\beta_2^3/(1+\beta_2+\beta_2^2)]^{0.25},$$

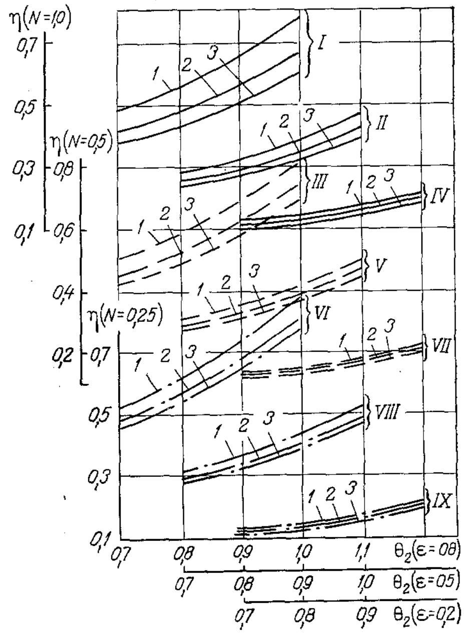

Fig. 3. Dependence of the thermal efficiency of a radiator element with tubes at unequal temperatures on Θ_2, ε, R/L, and N: I) N = 1.0; ε = 0.9; II) 1.0 and 0.5; III) 0.5 and 0.9; IV) 1.0 and 0.2; V) 0.5 and 0.5; VI) 0.25 and 0.9; VII) 0.5 and 0.2; VIII) 0.25 and 0.5; IX) 0.25 and 0.2; 1) R/L = 0.8; 2) 0.4; 3) 0.2.

$$
\begin{gathered}
Q_0 = 2G_1 c_p (T_{\text{in}} - T_{\text{out}_1}),\\
Q_1 = Gc_p (T_{\text{in}} - T_{\text{out}_1}) + G_2 c_p (T_{\text{in}} - T_{\text{out}_2}),\\
\eta_{e0} = f_1 (N,\ R(L,\ \varepsilon)),\\
\eta_{e1} = f_2 (N,\ R/L,\ \varepsilon,\ \Theta_2),\\
\eta_h = \Phi (C_x,\ f,\ \bar{z}_T),
\end{gathered}
\tag{14}
$$

where C_x characterizes the pattern in which the heat carrier is fed into and taken out from a radiator (emitter) section; f, ratio of the cross-sectional area of the radiator tubes to the cross-sectional area of the collector; and $\bar{z}_T$, referred hydraulic drag coefficient for a tube with the mean flow rate of heat carrier.

The solution to system (8)-(14) will, under the given assumptions, yield the mean-integral temperature of emitting tubes and thus also the effect of hydraulic unbalance on the thermal efficiency of a radiator element. In the case where the hydraulic unbalance (nonuniform distribution of heat carrier between tubes) cannot be eliminated, one can use the system of equations (8)-(14) for designing the geometry of tubes and crosspieces so that the thermal efficiency of the radiator will not drop by more than the permissible amount.

The dependence of the thermal efficiency of a radiator element on all the said factors can be expressed in the form $\psi_e = \Phi(N, R/L, \bar{T}/T_{in}, \eta_{he})$. On the basis of those relations one can determine the temperature field and the thermal efficiency of a radiator section assembled with any number of tubes.

The way in which the hydraulic connection scheme in a radiator section as well as the geometrical dimensions of the distributor and the collector determine the distribution of heat carrier between tubes can be established from known relations [5]. Let the total flow rate of the heat carrier in a radiator section, its temperatures T_{in} at the inlet and T_{out_0} at the outlet, assuming that there is no hydraulic unbalance ($\eta_h = 1$), also be given.

The thermal efficiency of a radiator section with hydraulic unbalance is defined as $\psi_s = Q_s/Q_0$, where Q_s and Q_0 denote the amounts of heat emitted by this radiator section with a, respectively, nonuniform and uniform distribution of heat carrier between tubes.

In the case of hydraulic unbalance there are k tubes in one part of a radiator section which operate with flow rates of the heat carrier below the given average ($\eta_h < 1$) and, excluding the tube with the average flow rate of the heat carrier, there are m = n − k − 1 tubes in the other part which operate with flow rates of the heat carrier above that average ($\eta_h > 1$). Summation over the corresponding heat-emitting elements of a radiator section, including the tubes which operate with a hydraulic unbalance factor larger and smaller than unity, will yield the expression

$$\psi_s = \frac{\sum_{i=1}^{k-1} a_i^4 \eta_{0i} \psi_{ei} + \sum_{j=2}^{m+1} a_j^4 \eta_{0j} \psi_{ej}}{a^4 (n-1) \eta_0}, \tag{15}$$

for the thermal efficiency of the entire radiator section. Here $a = \bar{T}/T_{in} = [3\beta_0^3/(1+\beta_0+\beta_0^2)]^{0.25}$; $\beta_0 = \frac{T_{out_0}}{T_{in}}$ is ratio of temperatures of the heat carrier at the outlet from a radiator section with a uniform distribution of heat carrier between tubes $T_{out,0}$ and at the inlet to the radiator section T_{in}; $a_i = \bar{T}_i/T_{in}$; $a_j = \bar{T}_j/T_{in}$; $a_1 = a$, a given quantity; $\eta_{0i} = f(N_i, R/L, \varepsilon)$, thermal efficiency of the i-th heat-emitting element with tubes at equal temperatures T_i; η_0, thermal efficiency of a radiator section with a uniform distribution of heat carrier between tubes;

$$\eta_{0j} = f_1(N_j, R/L, \varepsilon); \; N_i = \sigma L^2 \bar{T}_i^3 / \lambda t;$$

$\bar{T}_i$, mean-integral temperature of the i-th tube (determined by the temperatures of the heat carrier at the inlet to the radiator section T_{in} and at the outlet from the i-th tube $T_{out,i}$); $\psi_{ei} = \Phi(\bar{T}_i/T_{in}, N_i, R/L, G_{i+1}/G_i)$, thermal efficiency of the i-th radiator element taking into account the hydraulic unbalance $\eta_i = G_{i+1}/G_i$; G_i, flow rate of the heat carrier in the i-th tube;

$$\psi_j = \Phi_1(\bar{T}_j/T_{in}, N_j, R/L, G_{i-1}/G_i);$$

$$\bar{T}_i/T_{in} = [3\beta_i^3/(1+\beta_i+\beta_i^2)]^{0.25};$$

$$\bar{T}_j/T_{in} = [3\beta_j^3/(1+\beta_j+\beta_j^2)]^{0.25};$$

$$\beta_i = \begin{cases} \beta_0 & \text{for } i = 1, \\ 1 - \dfrac{1-\beta_{i-1}}{G_{i+1}/G_i}(2\psi_{ei} - 1) & \text{for } i \geqslant 2; \end{cases}$$

$$\beta_j = 1 - \frac{1-\beta_{j-1}}{(2\psi_{ej} - 1) G_i/G_{i-1}}.$$

The solution to Eq. (15) yields the decrease of thermal efficiency of a radiator section due to a nonuniform distribution of heat carrier between its tubes as well as the mean-integral temperature field in tubes and crosspieces of that radiator section, all these parameters needed for calculating the thermal stresses in components of the structure.

Calculating the thermal efficiency of a radiator consisting of many tubes is a rather laborious task, most expediently performed with the aid of a computer. The algorithm of this calculation reduces to the solution of a system of algebraic equations with values of η_{0i} and ψ_{ei} as input data.

For a quantitative evaluation of the effect of hydraulic unbalance on the thermal characteristics of a radiative heat exchanger, calculations were made to determine the thermal efficiency of tubes and the temperature of the heat carrier at the outlet from the tubes in a radiator section consisting of 23 tubes 8 mm in diameter, 1 mm in wall thickness, and 1400 mm long. The data on the distribution of heat carrier between tubes in such a radiator section, depending on the geometrical dimensions of distributor and collector as well as on the scheme of heat carrier feed and removal, are given in another study [5].

The results of calculations of the thermal efficiency are shown in Fig. 4 for a radiator section with the parameters $\varepsilon = 0.9$, $R/L = 0.4$, $N_0 = 0.5$, $a = 0.9$ (1) and $a = 0.8$ (2) and with various distributions of heat carrier between tubes. As the criterion for defining the distribution, nonuniformity index served its mean-integral value

$$S = \frac{1}{2} \int_0^1 /\eta_h - 1/ dx. \tag{16}$$

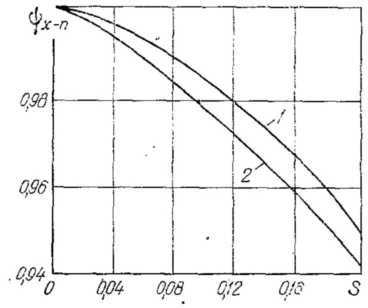

Fig. 4. Dependence of the thermal efficiency of the radiator section on the mean-integral nonuniformity index of heat carrier distribution between tubes.

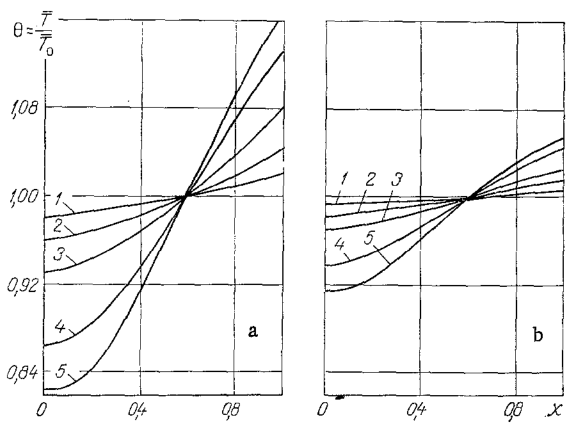

Fig. 5. Mean-integral temperature field in tubes of a radiator section (ε = 0.9, R/L = 0.4, N = 0.5) with various distributions of heat carrier between tubes, (a) a = 0.7 and (b) a = 0.9; 1) S = 0.02, 2) 0.044, 3) 0.082, 4) 0.156; 5) 0.2.

The calculations have revealed (Fig. 4) that in radiative heat exchangers a nonuniform distribution of heat carrier between tubes results in a relatively small reduction of their thermal efficiency. The maximum decrease of thermal efficiency for the given radiator with S = 0.2 and a = 0.8 was ~6%. According to the data in Fig. 5, however, the temperature of the heat carrier at the outlet from the tubes differ appreciably. The temperature difference between tubes at extreme ends of the radiator section with S = 0.2 and a = 0.7 was 34%. This gives rise to high thermal stresses in the structure, which certainly affect the service life of such a radiator section.

Increasing the flow of heat carrier through the radiator section will tend to equalize the temperatures at the outlets. However, this will also require more pumping power and thus reduce the overall efficiency of the system.

Therefore, providing for a uniform distribution of heat carrier between tubes in the design of radiative heat exchangers will ensure their higher thermal efficiency and operational reliability.

NOTATION

$N = \sigma L^2 T/\lambda t$, a parameter characterizing the thermal conductivity of a crosspiece; σ, Stefan–Boltzmann constant; L, width of a crosspiece; T, absolute temperature; λ, thermal conductivity; t, half-thickness of a crosspiece; R, radius of a tube; ε, emissivity of a surface; $\Theta = T/T_1$, referred temperature of a radiator element; η, efficiency of a radiator element; c_p, specific heat at constant pressure; G, mass flow rate of the heat carrier; P, perimeter; and η_h, hydraulic nonuniformity of heat carrier distribution between tubes.

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RADIATION STRUCTURE OF THE RELAXATION ZONE OF A SHOCK WAVE IN TWO-PHASE RAREFIED MEDIA

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UDC 533.6.011.72

Methods are presented for calculating the radiation structure of a shock front in a two-phase medium with low density and an example of such a calculation is given.

The problem of the role of radiation in gasdynamics of high-temperature two-phase media is always encountered when the radiation energy flux becomes comparable or exceeds the gasdynamic flux. Thus, if the equilibrium temperature behind the shock front propagating in such media is $\sim 10^3$ K, then radiation has a large effect on the structure of the front with densities in front of the wave less than 10^{-2} kg/m^3. The investigation of the radiation structure of the front of a shock wave in gas in the diffusion approximation involves an analysis of the properties of the transfer equation at $\pm\infty$ and joining the functions sought at the density jump [1, 2]. In this case, when radiation transfer occurs in a background of other relaxation processes, the analysis of the properties at $\pm\infty$ and the numerical solution of the problem as a whole, as noted in [3] and as will be evident in what follows, involve serious difficulties.

The radiation structure of a shock wave in a gas with particles was investigated previously in [4], wherein the interaction of retardation processes, heat exchange with the gas, and radiation of particles without taking into account their effect on the gas flow was determined. In this paper, the processes indicated are examined in a higher-order approximation and, in addition, their interaction with the gasdynamics is determined and two methods are proposed for calculating the radiation structure of the shock wave.

In what follows, we investigate the stationary structure of a shock front in a mixture of a gas and microscopic particles. It is assumed that all particles are spherical and have the same radius, and the gas and the particles are characterized by different temperatures and mass velocities. Phase transformations of the particles are not examined. The gas is assumed to be nonradiating and does not react with the particles, and it is also assumed to be nonviscous and nonthermally conducting. The effects of viscosity and thermal conductivity are taken into account only in the interaction between the gas and the particles.

The structure of the shock front is characterized by two regions of flow, separated by a density jump. In the region behind the density jump, the particles on interacting with the gas exchange heat and mechanical energy, and also loose or acquire energy through radiation. These processes, interacting with one another, determine the structure of the shock front in a given region. The radiation, leaving the surface of the discontinuity, is absorbed in front of the discontinuity by particles, which are thereby heated. In their turn, the particles heat the gas in front of the discontinuity, creating a pressure gradient in it, which puts the gas, and then the particles, into motion. At low density of the two-phase medium, the radiation can greatly change the

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Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 41, No. 5, pp. 888-896, November, 1981. Original article submitted September 5, 1980.

0022-0841/81/4105-1247$07.50