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AN ENERGY-BASED JUSTIFICATION FOR AND A COMPARISON OF THE METHODS OF TECHNICAL-ECONOMIC AND ENERGY OPTIMIZATION OF CONVECTIVE HEAT-EXCHANGE SURFACES

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Calculation relationships are offered and a comparison is carried out of the energy and technical-economic optimization of these methods, and the limitations imposed on the latter are also established.

We currently have on hand a tremendous volume of theoretical and experimental material dealing with studies into the convective exchange of heat in the forced motion of coolants through channels exhibiting a variety of surface shapes. Among the most urgent problems are those connected with selection of the most effective heat-exchange surfaces, as well as the determination of their optimum operating conditions in the nominal regime.

The most objective comparison of convective heat-exchange surfaces and their optimization, in our opinion, is offered by the energy method which essentially involves determining the relationship between the intensity of the convective heat exchange and the specific expenditures of power on the propulsion of the coolant through the channels of the heat-exchange surface, and namely:

$$\alpha = f\left(\frac{N}{F}\right). \tag{1}$$

These specific expenditures can be determined from the relationship

$$\frac{N}{F} = \xi \frac{\mathrm{Re}^3 \nu^3 \rho}{8 d_{\mathrm{eq}}^3}. \tag{2}$$

If for purposes of calculating the coefficient ξ we use the theoretical or empirical relationships of the form

$$\xi = \frac{A}{\mathrm{Re}^m}, \tag{3}$$

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then Eq. (2) may be represented as

$$\frac{N}{F}=\frac{A\,\mathrm{Re}^{3-m}\nu^{3}\rho}{8d_{\mathrm{eq}}^{3}}. \tag{4}$$

In determining the heat-transfer coefficient from the criterial equation of convective heat exchange

$$\alpha=\frac{\lambda}{d_{\mathrm{eq}}}c\,\mathrm{Re}^{n}\mathrm{Pr}^{k}\varepsilon_{t} \tag{5}$$

from the combined examination of relationships (4) and (5) we will obtain

$$\alpha=\frac{\lambda}{d_{\mathrm{eq}}}c\left[\frac{8d_{\mathrm{e}}^{3}\left(\frac{N}{F}\right)}{A\nu^{3}\rho}\right]^{\frac{n}{3-m}}\mathrm{Pr}^{k}\varepsilon_{t}. \tag{6}$$

Relationship (6) can be utilized for purposes of an energy comparison between the various heat-exchange surfaces forming the channels. The methodology involved in such a comparison has been detailed rather fully in [1].

As analysis will demonstrate, relationship (6) is exponential in nature. As the specific expenditures of power on the propulsion of the heat-carrying coolant increase, the intensity of the convective heat exchange initially rises sharply and the increase in the coefficient of heat transfer is subsequently retarded, with the intensity of heat exchange then tending to a limit value that is only slightly dependent on any increment to the expenditure of power on the propulsion of the heat-carrying coolant.

In order to establish a reasonable limit to the forced exchange of heat in the case of forced convection, it is our suggestion that we use a conditional optimum in the form of

$$\frac{d(\alpha)}{d\left(\frac{N}{F}\right)}=1, \tag{7}$$

i.e., a condition which with a 1 W/m² increment to the specific expenditures of power on the propulsion of the coolant through the channels of the surface yields an addition to the intensity of convective heat exchange by 1 W/(m²·K). The question of validating this "conditional optimum" has been dealt with in detail in [2].

Finally, the relationships for the determination of the optimum specific expenditures of power on the propulsion of the coolant and the corresponding Reynolds number condition in the case of the energy method of optimization will have the form:

$$\left(\frac{N}{F}\right)_{\mathrm{opt}}^{\mathrm{eq}}=\left\{\frac{n\lambda c}{(3-m)\,d_{\mathrm{eq}}}\left[\frac{8d_{\mathrm{eq}}^{3}}{A\nu^{3}\rho}\right]^{\frac{n}{3-m}}\mathrm{Pr}^{k}\varepsilon_{t}\right\}^{\frac{3-m}{3-m-n}} \tag{8}$$

and

$$\mathrm{Re}_{\mathrm{opt}}^{\mathrm{eq}}=\left[\frac{8d_{\mathrm{eq}}^{3}\left(\frac{N}{F}\right)_{\mathrm{opt}}^{\mathrm{eq}}}{A\nu^{3}\rho}\right]^{\frac{1}{3-m}}. \tag{9}$$

The velocity of coolant motion, corresponding to the optimum specific expenditures of power on propulsion of the coolant can be determined from the derived value of $\mathrm{Re}_{\mathrm{opt}}^{\mathrm{e}}$.

However, the results of the energy optimization must be verified on the basis of the technical-economic optimization indices.

As is well known, the most objective index of technical-economic effectiveness for any technical object is represented by the reduced annual expenditures which take into consideration the capital investment for the construction of the object and its technical operation. These questions have been dealt with in a number of references, for example [3-5], insofar as this pertains to heat-exchange apparatus.

Let us carry out the validation of the calculation relationships for the technical-economic optimization, basing our derivation of the specific expenditures of power on the propulsion of the coolant through the channels of the heat-exchange surface.

The reduced annual expenditures for the heat-exchange surface can be determined from the relationship

$$3_{red} = \frac{3_s}{Q_a} + \frac{3_s}{Q_a} + \frac{3_{eq}}{Q_s}. \tag{10}$$

Taking into consideration that the annual expenditures 3_s on the fabrication of the heat-exchange surface and 3_b on a drive-operated blower, the operational expenditures 3_e, as well as the annual quantity of heat transmitted by the heat-exchange surface can be represented by the equations:

$$3_{red} = (a+p)\, k_{red} F,$$

$$3_b = \frac{(a+p)\, k_b \left(\dfrac{N}{F}\right)^{te}_{opt} Fr}{10^3 \eta_e},$$

$$3_{eq} = \frac{k_e \left(\dfrac{N}{F}\right) F\tau}{10^3 \eta_e},$$

$$Q_a = \alpha \Delta t F \tau,$$

and relationship (10) for unilateral heat exchange can be represented in the form

$$3_{red} = \frac{(a+p)\, k_b 10^3}{\alpha \Delta t \tau} + \frac{(a+p)\, k_b \left(\dfrac{N}{F}\right)^{te}_{opt} r}{\alpha \Delta t \tau \eta_e} + \frac{k_{eq}\left(\dfrac{N}{F}\right)}{\alpha \Delta t \eta_e} \tag{11}$$

or with consideration of Eqs. (4) and (6):

$$3_{red} = B\left[\frac{(a+p)\, k_s 10^3}{\left(\dfrac{N}{F}\right)^{\frac{n}{3-m}}} + \frac{(a+p)\, k_b \left(\dfrac{N}{F}\right)^{te}_{opt} r}{\left(\dfrac{N}{F}\right)^{\frac{n}{3-m}} \eta_e} + \frac{k_{eq}\tau \left(\dfrac{N}{F}\right)^{\frac{3-m-n}{3-m}}}{\eta_e}\right], \tag{12}$$

where

$$B = \frac{d_{eq}}{\lambda c \Pr^k \varepsilon_t \Delta t \tau}.$$

It should be noted that the established power of the drive-operated blower is determined from the magnitude of the optimum specific expenditures of power. With a change in the expenditures on power for the propulsion of the coolant the magnitude of the established drive-operated blower power undergoes no change.

The optimum specific expenditures of power on the propulsion of the coolant through the channels of the heat-exchange surface in the case of technical-economic optimization are found from the condition

$$\frac{d(3_{red})}{d\left(\dfrac{N}{F}\right)} = 0. \tag{13}$$

Taking into consideration that $B \neq 0$, we obtain

$$\left(\frac{N}{F}\right)^{te}_{opt} = \frac{n k_s 10^3 \eta_e}{(3-m-n)\left[\dfrac{k_e \tau}{(a+p)} - \dfrac{k_b r}{(3-m-n)}\right]}. \tag{14}$$

The optimum values of the Reynolds numbers in the case of technical-economic optimization can be found from relationship (4):

$$\mathrm{Re}_{\mathrm{opt}}^{\mathrm{te}} = \left[\frac{8d_{\mathrm{eq}}^{3}\left(\frac{N}{F}\right)_{\mathrm{opt}}^{\mathrm{te}}}{A\nu^{3}\rho} \right]^{\frac{1}{3-m}}. \tag{15}$$

We can determine the corresponding velocity of coolant motion on the basis of the derived Reynolds number.

As we can see from relationship (14), the optimum value of the specific expenditures of power in the case of technical-economic optimization does not depend on the type of coolant nor on its thermophysical properties, the linear dimensions of the heat-exchange surface channels, but is wholly determined by the technical-economic characteristics of the surface and the coolant motion regime.

The optimum value of the Reynolds number and, consequently, the optimum velocity of coolant motion depend on the magnitude of the optimum specific expenditures of power on the propulsion of the coolant, i.e., on the above-noted quantities, on the determining linear dimensions of the heat-exchange surface, and on the thermophysical characteristics of the coolant.

Comparison of the theoretical relationships (8) and (14) demonstrates that they are different both in terms of structure and composition. One should assume that the magnitudes of the optimum specific expenditures of power in the various methods of optimization will also differ significantly relative to each other.

Of considerable interest is a comparison of the quantities $(N/F)_{\mathrm{opt}}^{\mathrm{eq}}$ and $(N/F)_{\mathrm{opt}}^{\mathrm{te}}$, as well as of the Reynolds numbers $\mathrm{Re}_{\mathrm{opt}}^{\mathrm{eq}}$ and $\mathrm{Re}_{\mathrm{opt}}^{\mathrm{te}}$ and the corresponding velocities of coolant motion for a given surface, in combination with the same coolant.

For purposes of comparing the indicated methods of optimization and in order to determine their suitability to practical calculations, let us examine the numerical example for heat exchange on one side of a heat-exchange surface in a water-water heat exchanger, whose basic technical-economic indices are taken from the source [6]: k_s = 24 rubles/m²; k_b = 100 rubles/kW; k_e = 1.5 kopeks/(kW·h); η_e = 0.7; a = 0.1 and p = 0.12 1/year; τ = 6000 h/year; d_{eq} = 11.6 mm. The surface type is a tube bundle and the flow takes place inside of the tube. The drive-operated blower power reserve coefficient is assumed to be r = 1.20. The thermophysical parameters of the coolant are taken from [7] in these calculations.

For the conditions cited above and for the turbulent regime of coolant motion (A = 0.3164, m = 0.25, n = 0.8) the optimum specific expenditures of power, the optimum Reynolds numbers and the corresponding velocities of distilled water motion, said water at an average temperatures of $\bar{t}_w$ = 20°C, in the case of energy and technical-economic optimization, have the following values:

$$\left(\frac{N}{F}\right)_{\mathrm{opt}}^{\mathrm{eq}} = 14695\ \mathrm{W/m^2};\quad \left(\frac{N}{F}\right)_{\mathrm{opt}}^{\mathrm{te}} = 19{,}8\ \mathrm{W/m^2};$$

$$\mathrm{Re}_{\mathrm{opt}}^{\mathrm{eq}} = 233820;\quad \mathrm{Re}_{\mathrm{opt}}^{\mathrm{eq}} = 21135;$$

$$w_{\mathrm{opt}}^{\mathrm{eq}} = 20{,}1\ \mathrm{m/sec};\quad w_{\mathrm{opt}}^{\mathrm{te}} = 1{,}82\ \mathrm{m/sec}.$$

As we can see from these quantities, the technical-economic optimization confirms the generally accepted practice of heat-exchange apparatus design in which the velocity of motion for the water through the tubes is assumed to be approximately equal to 2 m/sec. However, the energy optimization calls for specific expenditures of power on the order of 14.7 kW/m² and velocities of coolant motion on the order of 20 m/sec, which is unacceptable for actual purposes.

However, this conclusion pertains only to the example under consideration. When the specific cost of the heat-exchange surface is increased to k_s = 100 rubles/m², the optimum value of the specific expenditures on power in the case of technical-economic optimization for the surface under consideration, according to Eq. (14), increases to 82.5 W/m². The velocity of the coolant motion under these conditions must also rise to 3.06 m/sec. The parameters of energy optimization in this case remains unchanged.

With a specific surface cost of 300 rubles/m² the optimum specific expenditures on power for the case under consideration increase to $(N/F)_{opt}^{te} = 247.5$ W/m², while the velocity of coolant motion to attain these conditions must increase to 4.56 m/sec. An analysis of price lists for various types of heat-exchange equipment shows that these and even more expensive surfaces exist. Thus, the retail price of a PR 0.3-3-1 plate heat-exchange unit, according to the No. 23-03 Price List for January 1, 1982, lists a price of 1150 rubles for a heat-exchange surface 3 m² in size. The specific cost of the heat-exchange surface in this case amounts to more than 380 rubles/m².

As we can see from the above, the technical-economic optimization is an extremely important tool for purposes of analyzing the efficiency of the heat-exchange surface, but it cannot be accepted as the only decisive criterion.

With a change in heat-exchanger prices, or with a change in the cost of 1 kW·h of electrical energy, 1 kW of drive-operated blower power, of surface operating time, and other quantities in Eq. (14), the optimum value of the specific expenditures on power for the propulsion of the coolant will change markedly. Today's technical-economic optimum may prove to be nonoptimum tomorrow.

The parameters of energy optimization with a change in the technical-economic characteristics of the heat-exchange surface remain unchanged and represent a theoretical limit, although unattainable in the present case, for the intensification of convective heat exchange.

However, the parameters of energy optimization for this surface are not invariant. For another coolant or for the same coolant, but with a different average temperature, the same surface may exhibit different values for the optimum specific expenditures of power on the propulsion of the coolant.

Indeed, in the case, for example, of the motion of the oil of a spindle-shaped AU moving through the channels of the surface being examined here, and exhibiting a temperature of 60°C, the parameters of energy optimization will have the following values:

$$\left(\frac{N}{F}\right)_{opt}^{eq} = 709{,}6 \text{ W/m}^2; \quad Re_{opt}^{eq} = 7208; \ w_{opt}^{eq} = 5{,}78 \text{ m/sec.}$$

For the case under consideration the parameters of the technical-economic and energy optimization came very much closer together. However, the design of the surface for a nominal operating regime in the case cited here must be based on the permissible velocity of coolant motion through the channels of the heat-exchange surface.

When MS-20 oil with an average temperature of 60°C is in motion through tubes of a given diameter, the parameters of energy optimization will exhibit different values (the laminar regime):

$$\left(\frac{N}{F}\right)_{opt}^{eq} = 40{,}8 \text{ W/m}^2; \quad Re_{opt}^{eq} = 110{,}9; \ w_{opt}^{eq} = 0{,}87 \text{ m/sec.}$$

This last example demonstrates that the parameters of energy optimization may be lower than the parameters of the technical-economic optimization. Naturally, the best design variant most probably would involve agreement in the energy and technical-economic optima with velocities of coolant motion lower than the maximum technically justified velocity. The proposed calculation relationships enable us to determine these conditions.

The analysis presented above also demonstrates that the velocity of coolant motion through the channels of the heat-exchange surface may represent a limitation on the achievement of the technical-economic and energy optima. For these cases the nominal operating regime of the surface must be determined from the maximum technically validated velocity of coolant motion through the channels of the surface under consideration. This velocity must be at its maximum, and yet ensure prolonged and safe operation of the heat-exchange surface.

The determination of these velocities for specific coolants, taken at specific average temperatures, in combination with specific heat-exchange surfaces from specifically identified materials, is an unusually extensive problem and one that is extraordinarily important, particularly in connection with the development of automated systems for the design of heat-exchange apparatus.

NOTATION

α, coefficient of heat transfer; N/F, specific expenditure on power to overcome the hydraulic resistance of the heat-exchange surface channels; N, power expended on the propulsion of the heat-carrying coolant through the surface channels; F, magnitude of the heat-exchange surface; ξ, coefficient of hydraulic resistance for a unit of relative channel length in the heat-exchange surface; A, numerical coefficient expressing the relationship for the determination of the coefficient of hydraulic losses; ν, ρ, coefficients of kinematic viscosity and coolant density; d_{eq}, equivalent channel diameter in the heat-exchange surface; Re, Pr, Reynolds and Prandtl numbers; $\varepsilon_t = (Pr_\ell/Pr_w)^{0.25}$, parameter which takes into account the direction of the heat flow and the temperature increase; λ, coefficient of coolant thermal conductivity; c, criterial constant of the convective heat-exchange equation; d, differential symbol; $З_{red}$, reduced annual expenditures on the development and operation of the heat-exchange surface; $З_s$, $З_b$, $З_e$, annual capital expenditures on the development of heat-exchange surface and the drive-operated blower covering this surface, as well as the operational expenditures on the blower drive; Q_a, annual quantity of heat transferred through the heat-exchange surface; a, fraction of annual deductions for depreciation; p, coefficient of investment capital effectiveness; k_s, k_b, k_e, specific costs for the heat-exchange surface, for 1 kW of installed drive-operator blower power and for 1 kW·h of electrical energy; η_e, drive-operated blower efficiency; r, coefficient of drive-operated blower reserve; τ, duration of heat-exchange surface operation, h/year; Δt, average temperature difference; m, n, k, exponents of the similarity numbers. Subscripts: e, te, opt, ℓ, w, energy, technical-economic, optimum, liquid, wall; w, velocity of heat-carrying coolant motion.

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