FREE CONVECTIVE HEAT TRANSFER OF A PLANE NONISOTHERMAL SURFACE UNDER DIFFERENT ORIENTATIONS

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The turbulent free convective heat elimination from a nonisothermal slab surface is investigated experimentally.

A rise in the reliability of exploitation of shipboard power equipment requires the passage to closed cooling systems. The most promising in this area is utilization of shipboard jacketed heat-transfer apparatus (SJHTA) [1]. In the general case SJHTA is a labyrinth channel on the inner side of the ship hull jacket and enclosing elements of its assembly. The fresh water heated in the power plant is pumped through this channel whereupon its heat is transferred through the jacket to the water intake. The velocity of hot fresh water motion in the SJHTA channel is ordinarily about 1 m/sec, resulting in heat elimination by forced convection during development of the turbulent mode.

A ship can move or be in a berth. In the first case the heat elimination of the intake water is realized by means of forced, and in the second by free convection. It is well known that heat elimination is considerably more intense under forced convection than under free convection. Therefore, the most unfavorable SJHTA operating conditions are produced for a standing ship. Consequently, the mode mentioned is considered the main design mode. Computations executed in conformity with [1] show that comparatively high values of the heat elimination coefficient of 2.5-4.5 $kW/(m^2 \cdot K)$ are reached in the labyrinth channel. This indicates the decisive influence of the process of free convective heat elimination of the intake water on the heat elimination in the SJHTA.

The specific nature of the hot fresh water motion along the jacket results in the formation of complex temperature conditions on the outer surface. As is noted in [2], this substantially influences the development of the heat transfer process since it is associated with the deformations of the temperature distribution in the thermal boundary layer and its thickness. Consequently, it is impossible to utilize the well-known dependence for isothermal surfaces to analyze SJHTA. Preliminary computations permitted the finding that intake water heat elimination occurs by means of turbulent free convection. Computation methods for laminar free convection are studied and developed in the majority of papers on the heat elimination of plane nonisothermal surfaces.

An experimental investigation was undertaken of the free-convective heat transfer of a plane nonisothermal SJHTA jacket surface in order to set up the necessary dependences. To do this a plane steel 10 mm thick slab (corresponding to the ship jacket thickness) of 1 × 1 m dimensions was used. On one side of the slab was the labyrinth channel over which the hot fresh water flowed to create a temperature field on the other slab surface that is analogous to that existing in a real structure. The channel was carefully heat insulated to assure heat removal only through the slab. The model being investigated was in a tank of around 20-m^3 volume filled with sea water. The experimental installation permitted a change in the slab slope φ between the horizontal and the direction of the heat flux from $\varphi = -90°$ (heat elimination downward) to $\varphi = 90°$ (heat elimination upward). In addition, the possibility was provided for rotation of the model under investigation even in the plane of the heat-transfer surface.

The temperature was measured by using a Chromel—Copel thermocouple. Sixteen of them were caulked on the slab surface. Five more thermocouples were arranged along the motion path to check the change in the hot fresh water temperature directly in the labyrinth channel

Varied in the experimental investigation were the mean fresh water temperature $\bar{t}_{fw}$ = 29-97°C; the fresh water velocity w_{fw} = 0.3-2.9 m/sec, which corresponded to the range of numbers

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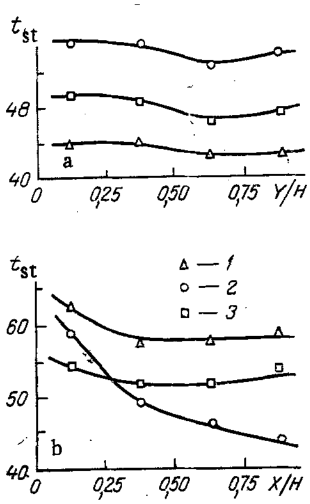

Fig. 1

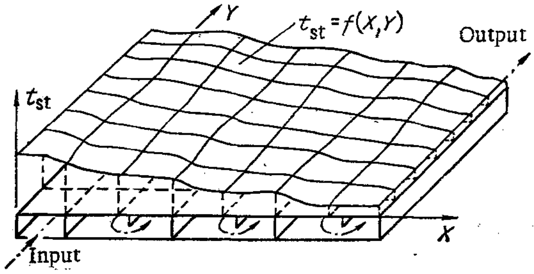

Fig. 2

Fig. 1. Dependence of the surface temperature t_{st} of the slab on the longitudinal X and Y coordinates: a) in the Y direction (where H = 1 m is the surface dimension) and b) in the X direction, H = 1 m [for $\varphi = 0°$; $t_{sw} \approx 20°C$; 1) $w_{fw} = 0.35$ m/sec; $t_{fw} \approx 54°C$; 2) $w_{fw} = 1.1$ m/sec; $t_{fw} \approx 71°C$; 3) $w_{fw} = 1.9$ m/sec; $t_{fw} \approx 60°C$].

Fig. 2. Simplified model of the surface temperature field of the model under investigation.

$Re = 8 \cdot 10^3 - 33 \cdot 10^4$, and the sea water temperature $t_{sw} = 7$–$27°C$. Eight different slopes were examined $\varphi = -90°$; -75; -60; -30; 0; 30; 60; 90°. For the vertical position ($\varphi = 0°$) the slab was installed in three locations differing by 90° because of rotation in the heat transfer plane, which assures a different relative motion of the hot fresh water in the channel and the free-convective sea water currents.

Measurements confirmed the presence of a variable temperature along the heat eliminating surface (Figs. 1 and 2). Its maximal change in the Y direction did not exceed 2-3°C. At the same time the change in t_{st} in the X direction (Fig. 2) turns out to be more substantial and reaches 8-18°C (Fig. 1b) as experiments showed. These data permitted representation of a somewhat simplified temperature distribution pattern along the slab surface (Fig. 2). The maximal change in t_{st} in the X direction takes place at the "root" of the labyrinth channel partition. This is explained by the fact that water having the highest temperature drop because of cooling flows on both sides of the partition. In addition the heat spillover in the body of the slab itself exerts a definite influence on the formation of the surface temperature boundary conditions because of its comparatively large thickness.

The slab heat elimination depends substantially on its slope (Fig. 3). The relative influence of the slope turns out to be distinct for different temperature heads between the surface and the sea water. Thus, for example, the heat elimination $\varphi = -90°$ for $\bar{t}_{st} - t_{sw} = 7°C$ is approximately 0.53 times and for $\bar{t}_{st} - t_{sw} = 45°C$ is 0.32 times less as compared with the heat elimination for $\varphi = 0°$. These relationships are quite close to the recommendations in [3]. At the same time the results obtained differ from the unequivocal recommendations [4] proposing that values 30% less be taken at $\varphi = -90°$ for $\bar{\alpha}$ than at $\varphi = 0°$.

Results of experiments on the free-convective heat elimination of a nonisothermal surface was extended on the basis of an extensively utilized criterial processing. The appropriate data processing for one of the angular positions is represented in Fig. 4, where they include different slab rotations in the heat transfer plane. The influence of this factor on the heat transfer is not established to the accuracy of the investigations performed. For the plate arranged vertically, the equation describing the heat elimination has the form

$$\overline{Nu} = 0.1\,(PrGr)^{0.33}\left(\frac{Pr_{sw}}{Pr_{st}}\right)^{-0.09}. \quad (1)$$

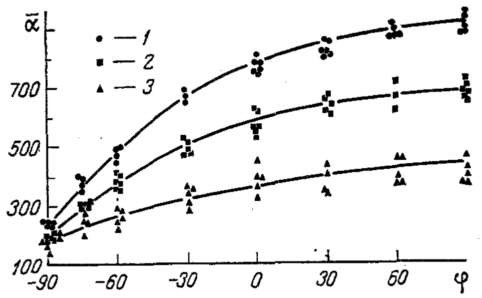

Fig. 3. Dependence of the heat-elimination coefficient $\bar{\alpha}$ of the slab surface on its slope φ for different temperature heads: 1) $\bar{t}_{st} - t_{sw} \approx 45°C$; 2) $\approx 23°C$; 3) $\approx 7°C$.

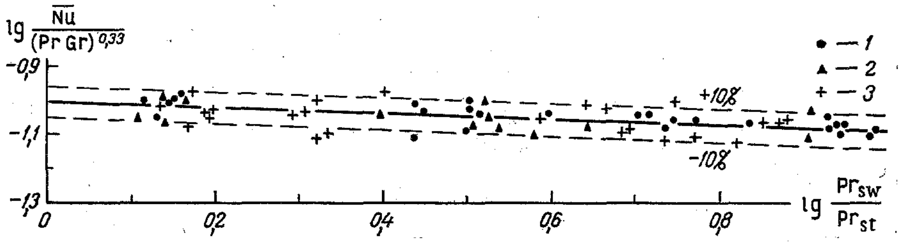

Fig. 4. Dependence of $\lg \frac{\overline{\mathrm{Nu}}}{(\mathrm{Pr\,Gr})^{0.33}}$ on $\lg \frac{\mathrm{Pr_{sw}}}{\mathrm{Pr_{st}}}$ (and $\varphi = 0°$; $w_{fw} = 0.35$–1.9 m/sec; $2.1 \cdot 10^{11} < \mathrm{PrGr} < 7.8 \cdot 10^{12}$): 1, 2, and 3 correspond to slab rotations differing by 90° in the heat-transfer plane.

The midpoint between $\bar{t}_{st}$ and t_{sw} was taken as the governing temperature when finding the numbers Pr and Gr while the characteristic dimension was the slab height (H = 1 m). Analogous criterial equations of the form

$$\overline{\mathrm{Nu}} = C\,(\mathrm{PrGr})^{k} \left(\frac{\mathrm{Pr_{sw}}}{\mathrm{Pr_{st}}}\right)^{-0.09} \tag{2}$$

are obtained for different slopes. Represented in the table are appropriate values of the coefficients C and k. These dependences are valid in the range $2.1 \cdot 10^{11} < \mathrm{PrGr} < 7.8 \cdot 10^{12}$ and $1 < (\mathrm{Pr_{sw}}/\mathrm{Pr_{st}}) < 16$.

Let us note that the exponent for the $\mathrm{Pr_{sw}}/\mathrm{Pr_{st}}$ is -0.09 in the criterial equations obtained. This quantity differs substantially from the well-known exponent 0.25 [4] that takes account of the influence of the so-called transverse isothermy of the boundary layer. As mentioned above, the surface nonisothermy (longitudinal nonisothermy) exerts substantial influence on the heat transfer. The nature of this isothermy for the model being investigated remains identical in all cases (see Fig. 2) since it is given by the hot water moving in the labyrinth channel. The influence of this factor on the heat transfer turns out to be proportional to the ratio $\mathrm{Pr_{sw}}/\mathrm{Pr_{st}}$. This explains the unusualness of the exponent for $\mathrm{Pr_{sw}}/\mathrm{Pr_{st}}$ and the fact that the heat elimination of a nonisothermal surface is described successfully without introducing additional simplexes including the longitudinal nonisothermy parameters in explicit form.

The exponent for PrGr for the angles $\varphi = -90°$ and $-75°$ differs from 0.33 which indicates that the heat transfer process is not self-similar, i.e., in these cases the magnitude of the characteristic dimension H exerts a definite influence on the heat elimination. Returning to the question raised above about the relationships between the magnitudes of $\bar{\alpha}$ for different φ, the essential dependence on the quantity H should additionally be mentioned. It is seen from the table that for $\varphi < 60°$ the quantity H results in degradation of the heat elimination. The necessity to take account of the dimensions when examining the heat elimination of surfaces oriented horizontally downward is also indicated in investigations in [5].

To determine the values of the coefficients C and k in (2) for slab slopes intermediate to those indicated in the table, appropriate continuous functions are found

TABLE 1. Values of the Coefficients C and k in (2)

φ, deg	−90	−75	−60	−30	0	30	60	90
C	1,4	0,22	0,065	0,087	0,1	0,11	0,12	0,12
k	0,2	0,28	0,33	0,33	0,33	0,33	0,33	0,33

for $-90° \leqslant \varphi < -60°$

$$\overline{\mathrm{Nu}} = 0{,}047\,[\mathrm{ctg}\,(100° + \varphi)]^{1.96}\,(\mathrm{PrGr})^{0.35[\mathrm{tg}(100°+\varphi)]^{0.31}} \left(\frac{\mathrm{Pr_{sw}}}{\mathrm{Pr_{st}}}\right)^{-0.09},$$

for $-60° \leqslant \varphi \leqslant 90°$

$$\overline{\mathrm{Nu}} = 0{,}117 \left[\cos\left(\frac{\varphi}{2} - 45°\right)\right]^{0.43} (\mathrm{PrGr})^{0.33} \left(\frac{\mathrm{Pr_{sw}}}{\mathrm{Pr_{st}}}\right)^{-0.09}.$$

Let us compare the obtained criterial equation for the vertical surface with known data. Thus, by comparison with a vertical isothermal surface [4] in the range of numbers $2.1 \cdot 10^{11} < \mathrm{PrGr} < 7.8 \cdot 10^{12}$ the dependence (1) yields a value of $\overline{\mathrm{Nu}}$ approximately 1.7 times less. Results are presented in [6] of a study of the free-convective heat elimination by a vertical nonisothermal surface in the turbulent mode. The nature of this nonisothermy differs from that under consideration in this investigation. The equation

$$\overline{\mathrm{Nu}} = 0{,}276\,(\mathrm{PrGr})^{0.285}$$

yields sufficiently close results in the mentioned domain of numbers PrGr. The difference does not exceed 30% here. In certain cases agreement of the results is observed ($\mathrm{PrGr} = 1.6 \times 10^{12}$). Other dependences established in [6] yield values of $\overline{\mathrm{Nu}}$ that are 1.8-2.8 times less as compared with (1).

The criterial equations obtained for the free-convective heat elimination of the nonisothermal surface under consideration, together with the results of the heat elimination investigation in the labyrinth channel [1], permitted development of a thermotechnical method of analyzing the SJHTA. We present certain results of comparing the design values of the heat transmission coefficients and the values obtained as a result of full-scale ship tests with such apparatus. Thus, in [7] there are test data on the ship "Ludwig Franzius" with a SJHTA inclined at the angle $\varphi = -30°$. The design and test data here differed by not more than 11%. The floating crane "Bogatyr'", the first domestic vessel fitted with two SJHTA in our land, was launched in 1983. These heat exchangers were arranged on the bottom ($\varphi = -90°$) and each has a 50-m² area. The full-scale thermotechnical test conducted with the author's participation showed that the heat transmission coefficient of the apparatus was about 90 W/(m²·K), which corresponded to the design data with up to 15% accuracy. This again confirms the correctness of the results obtained.

NOTATION

φ, slope of the heat-transfer surface, deg; t_{fw}, fresh water temperature, °C; w_{fw}, velocity of fresh water motion in the channel, m/sec; t_{st}, surface temperature, °C; t_{sw}, sea water temperature, °C; α, heat elimination coefficient, W/(m²·K); H, characteristic linear dimension, m; Nu, Nusselt criterion; Re, Reynolds criterion; Pr, Prandtl criterion; Gr, Grashof number; $\mathrm{Pr_{st}}$, $\mathrm{Pr_{sw}}$, Prandtl number at the surface and sea water temperatures; $\bar{\ }$, averaging sign.

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