THE THERMAL RESISTANCE OF FROST ON A FINNED AIR COOLER

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Results are given from measuring the thermal resistance of frost on finned air coolers.

One needs data on the current thermal resistance of continuously accumulating frost in calculations on heat transfer and performance for finned air coolers, particularly in order to choose optimum working conditions. There are very few papers on the resistance of frost [1-4], which do not deal with the effects of all the major factors on R_f and do not give practical recommendations on calculating it.

We have measured resistances for frost on finned air coolers having corridor-type tube bundles and various fin stacks. The apparatus was build up from sections of length 170 mm, which each contained two longitudinal rows of tubes and five transverse ones, diameter 25 mm and pitches 70 and 76 mm correspondingly. The flat solid fins were made of 0.4 mm steel. The set thermal conditions were provided by an independent refrigerator. The coolant was ammonia, and its temperature was measured along with the temperatures of the tube walls, ribs, and air by means of thermocouples. The heat load was determined from the heat balances for the air and coolant, and the flow rates were measured.

The air was supplied by a centrifugal blower having a dc drive, which provided smooth speed control; the flow rate was determined with calibrated nozzles, and the speed in the cooler slot channels was monitored by Pitot tubes. The system was fitted with an air heater and humidifier, as well as with an instrument for measuring the humidity. The apparatus and methods have been described in [5]. We measured not only the quantities required to calculate the heat-transfer coefficient but also the thickness of the frost on the fins. The air parameters varied as follows: mass flow rates from 2 to 13 kg/m²·sec, relative humidity from 0.74 to 0.96, temperature from 10 to −15°C, and mean temperature difference from 6 to 16°C.

The thermal resistance of the frost layer is

$$R_f = R - R_0 = k^{-1} - k_0^{-1}. \tag{1}$$

To eliminate errors in determining R_f, the values of the heat-transfer coefficients for the coolant appearing in k and k_0 were kept constant.

In most previous studies, a constant air flow was used, which can hardly be taken as a satisfactory method. The growth of frost on the fins reduces the active cross section in the slots and causes an ongoing increase in the air speed and aerodynamic resistance, this being the more so the smaller the fin pitch. In that case, the results for the frost resistance tend to be underestimated, because the frost density increases with the air speed, as does the effective conductivity [1, 2, 6]. Also, the condition w = var predetermines the additional errors in calculating R_f arising from differences in heat-transfer rate during the experiment. Under actual air-cooler working conditions, it is recommended that the blower characteristics should be such that the change in flow due to increase in aerodynamic resistance from the frost should be such that as far as possible the air speed in the slots remains constant [7], which provides for thermal and aerodynamic stability.

We therefore kept the air speed in the working cross section constant, which provides more generally valid information on R_f, since the frost formation rate and the physical parameters of the frost for w = const are independent of the design characteristics [3, 6].

Figure 1 shows the resistance variations for various air flow speeds, together with the analogous data of [4]; the air speed affects the frost resistance increment. The main increase in R_f occurs in the initial period (τ < 4 h), after which the rate of increase is much

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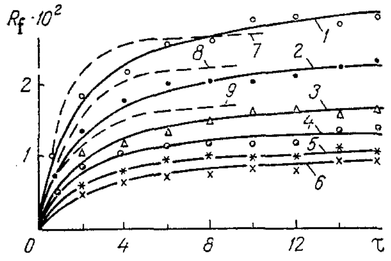

Fig. 1

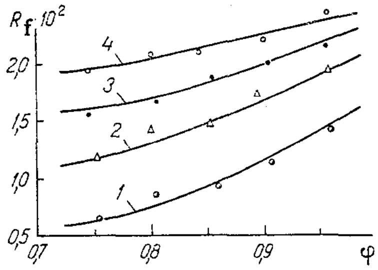

Fig. 2

Fig. 1. Time course of R_f, $\varphi = 0.85$; $c_t = 0.97$: 1-6) our experiments; 7-9) [4]; 1) $\rho w = 2.46\ kg/m^2 \cdot sec$; 2) 3.65; 3) 4.84; 4) 5.9; 5) 8.10; 6) 10.82; 7) 2.5; 8) 4.2; 9) 5.4. $R_f \cdot 10^2$ in $m^2 \cdot K/W$, and τ in h.

Fig. 2. Values of R_f for various relative humidities ($\rho w = 3.65\ kg/m^2 \cdot sec$ and $c_t = 0.97$): 1) $\tau = 1$ h; 2) 4; 3) 8; 4) 16.

less, not only because the frost thickness increase rate in this period is constant [6, 7] but also because the frost is more porous, and the thermal conductivity is then low [1, 2, 6]. At the start, the frost is formed at low density, and a small amount of water vapor gives a considerable thickness increase, with high porosity and low thermal conductivity. As the frost consolidates, the same thickness increase requires more vapor, so the rate of increase becomes less, while the effective thermal conductivity increases. Our results agree satisfactorily with those of [4]. There are unfortunately no other data for the purposes of this comparison.

It has been stated [1, 3] that the relative humidity affects the frost density and conductivity, but no explanation was given for the effects on R_f, as in the case of [4]. We found that an increase in relative humidity raises the resistance; the effects of φ were uneven in time, and were more pronounced in the initial period. Figure 2 shows typical effects of φ on R_f. The regularities [6, 7] show that at the start, the frost thickness increases more rapidly than the density and effective conductivity as φ increases, but subsequently, the rate of thickening and consolidation tend to equalize, and the effects of φ on R_f gradually become more comparable, which explains the curves in Fig. 2.

Also, the frost thickness increases with the air temperature and as the wall temperature is reduced [1, 6, 7], which is due to a feature of the crystal growth, where the main parts are played by the supersaturation, temperature difference, and heat of sublimation. As the above temperatures have opposite effects on the frost formation rate, one can incorporate their effects on R_f by means of a temperature coefficient c_t, which is the ratio of the absolute temperatures of the fin surface and the incoming air. The effects of this coefficient on the resistance are virually uniform throughout the operation.

The data gave the following formula for the thermal resistance:

$$R_f = 2(\rho w)^{-0,85}(0.025 + 0.027 \lg \tau)\tau^{-0,3}\varphi^{(4\tau^{-0,54})} c_t^{-1,7}. \tag{2}$$

By analogy with [8], it was very convenient to estimate the thermal economy of such a cooler by means of the dimensionless frost formation coefficient, which incorporates the initial surface state:

$$\psi_f = R_f(R_0 + R_f)^{-1} \quad (0 < \psi_f < 1). \tag{3}$$

From (3) one can readily calculate the current heat-transfer coefficient and thermal resistance:

$$k = k_0(1 - \psi_f) = k_0\eta, \tag{4}$$

$$R_f = \psi_f(k_0\eta)^{-1}. \tag{5}$$

The η of (4) and (5) is the current thermal efficiency of the cooler working with frosting. One can determine the heat-transfer coefficients for the clean surface from the formulas recommended in [9].

The nomogram of Fig. 3 has been constructed for R_f and ψ_f to simplify the calculations from (2)-(5).

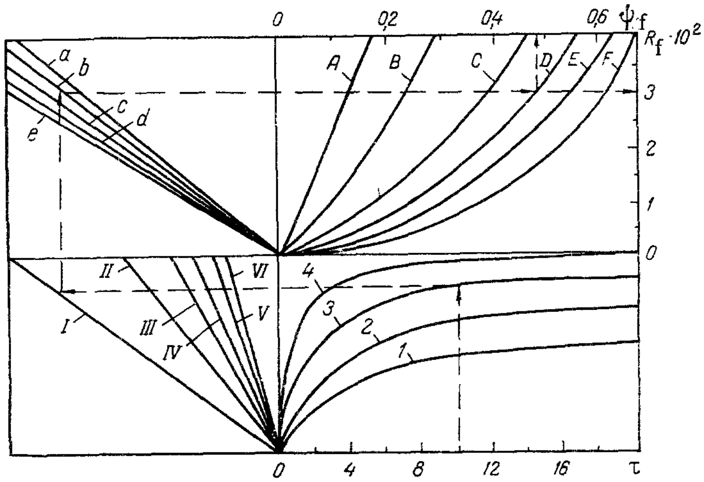

Fig. 3. Nomogram for determining R_f and ψ_f: 1) $\varphi = 0.7$; 2) 0.8; 3) 0.9; 4) 1; I) $\rho w = 2$ kg/m²·sec; II) 4; III) 6; IV) 8; V) 10; VI) 12; a) $c_t = 0.82$; b) 0.86; c) 0.9; d) 0.94; e) 0.98; A) $k_0 = 5$ W/m²·K; B) 10; C) 20; D) 30; E) 40; F) 50.

These results provide the necessary evidence not only in evaluating performance in finned air coolers during frosting but also in choosing optimum designs and working conditions.

NOTATION

R, k, thermal resistance and heat-transfer coefficient, respectively; η, w, φ, density, velocity, and relative humidity, respectively; τ, time, h. Subscripts: 0, initial value; f, under frost formation conditions.

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