HEAT OF A LAYER TRANSVERSELY FLOWING PAST A CYLINDER WITH CIRCULAR FINNING

V. A. Kalender'yan and V. V. Kornaraki

UDC 536.248:541.182.3

Results of investigations and generalized relations for calculating heat transfer of a dense layer with transversely finned cylinders are presented. Thermal efficiency is determined and recommendations are given for selecting the geometric parameters of developed surfaces of this type.

The present article is a continuation of the set of investigations of heat transfer between a dense unblown layer of granular material and transversely-circumfluent developed surfaces [1-3, et al.]. Their purpose is to study the effect of the size and shape of finning on the rate of heat transfer, to obtain design relationships, and to find ways of optimizing developed surfaces applicable to a dense layer.

Finning should satisfy the following requirements: a) provide separation-free circumfluence by the layer, which makes it possible to achieve a high rate of heat transfer with elements of small length; b) eliminate or reduce zones of stagnation and separation of the layer arising during transverse flow around circular cylinders and characterized, according to [4, 5, 2], by low heat transfer; c) provide sufficiently high efficiency of the fins; d) provide the possibility of obtaining large finning coefficients without noticeable loss of heat transfer. The fulfillment of the last requirement is especially desirable since for a dense layer moving under gravitational force an increase of the finning coefficient is not accompanied by an increase of energy expenditures for transport as is the case for other heat carriers.

The authors investigated single cylinders with transverse circular finning which to some extent satisfied the aforementioned conditions. In so doing we studied the effect of the rate of heat transfer on the height and spacing of the fins, diameter of the cylinder, and velocity of the layer, which was varied within $w = 0.32$-10.6 mm/sec. The geometric parameters of the finned cylinders are given in Table 1.

The method of the investigations, its substantiation, description of the experimental device, and measurement schemes are presented in [3]. In accordance with this method, we estimated not only the total thermal resistance but also its components due to convective heat transfer and thermal conductivity of the fins. The investigations were carried out during steady movement of the material by the steady-state thermal regime method with the heat flux directed from the surface to the layer. We determined: 1) the reduced heat-transfer coefficient $\alpha_{red} = Q/\bar{\vartheta}_b F_\Sigma$; 2) the average convective heat-transfer coefficient $\bar{\alpha} = Q/\bar{\vartheta} F_\Sigma$; 3) the efficiency of the fins $E = \vartheta_f/\vartheta_b$; 4) the correction for nonuniformity of the distribution of the rate of heat transfer over the surface of the fins $C = E/E_t$, where $E_t = \mathrm{th}[mh(1 + 0.8\log(D/d))]/mh(1 + 0.8\log(D/d))$ is the efficiency when $\alpha_f = \mathrm{const}$ [6].

The granular material (dry quartz sand) was a mixture with an average particle size $d_p = 0.24$ mm and the following fractional composition: 0-0.24 mm, 0.9%; 0.24-0.42 mm, 44.65%; 0.42-0.85 mm, 35.6%; 0.85-1.0 mm, 4.86%; more than 1 mm, 5.0% (particle size was averaged by volume). The effective heat conductivity of the layer, determined at a unit weight $\gamma_{vol} = 1600$ kg/m^3 which corresponds to steady movement, was $\lambda_{ef} = 0.32$ W/m·g and the effective thermal diffusivity was $a = 0.22 \cdot 10^{-6}$ m^2/sec.

An analysis of the primary experimental data shows that for all investigated cylinders the heat-transfer coefficient drops with an increase of the height of the fins and diameter of the cylinder and decrease of the spacing of the fins and velocity of the layer. The effect of the geometric and regime parameters on heat transfer is explained by their effect on the character of circumfluence. We can assume that

M.V. Lomonosov Technological Institute, Odessa. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 18, No. 1, pp. 52-58, January, 1970. Original article submitted March 25, 1969.

TABLE 1. Geometric Parameters of Finned Surfaces

Diameter of base d, mm	33,5	42,1	54,2
Height of fins h, mm	30 10 20 30 30 30	10 20 30	10 20 30
Spacing of fins t, mm	12 22 22 22 32 52	22 22 22	22 22 22
Finning coefficient φ	11 2,3 4,2 6,6 4,5 3,5	2,2 3,8 5,7	2,1 3,6 5,5
Relative height of fins h/d	0,9 0,3 0,6 0,9 0,9 0,9	0,24 0,48 0,72	0,18 0,37 0,56
Relative spacing of fins t/d	0,36 0,66 0,66 0,66 0,95 1,55	0,52 0,52 0,52	0,4 0,4 0,4

Note: Fin thickness δ = 2 mm; material steel St. 3.

circular fins, while not changing the qualitative pattern of flow past the cylinder (stagnant zone in the frontal part, flow past the side surface without separation, and separation of the layer in the stern zone), do introduce quantitative changes. These are reflected mainly in the size of the stagnant zone, its height and width increasing, the higher the fins, and the smaller the spacing between them, become. Similar phenomena are observed also on increasing the diameter of the cylinder. This leads to a decrease of heat released from the surface of the fins and base, to a decrease in $\bar{\alpha}$ and α_{red} (the latter quantity also reflects a change in the thermal resistance of conductivity of the fins as a function of their height and cooling conditions). Intensification of heat transfer upon an increase of velocity is due to an improvement of the release of heat from the portions past which flow occurs without separation (side surfaces of the cylinder and fins) and to a much lesser extent to some decrease of the size of the stagnant zone. We should point out that the considerations expressed follow from an analysis of the general regularities of the movement of the layer and are in need of refinement on the basis of data on local heat transfer. The rate of change of $\bar{\alpha}$ and α_{red} as a function of velocity is practically the same for all cylinders.

The experimental data were generalized on the basis of the results of an analysis by the method of the theory of similarity of a system of equations describing heat transfer of a layer with finned surfaces (equations of heat transfer, energy, motion, limit equilibrium, continuity, and heat conductivity of the fins). The components of this complex process (purely convective heat transfer and heat conductivity of the fins) were examined separately, which led to the following dimensionless equations:

$$\mathrm{Nu} = f_1\left(\mathrm{Pe}, \frac{h}{d}, \frac{t}{d}, \frac{d}{d_p}, \frac{t}{d_p}\right), \tag{1}$$

$$E = f_2(\mathrm{mh}, C), \tag{2}$$

$$C = f_3\left(\mathrm{mh}, \mathrm{Pe}, \frac{h}{d}, \frac{t}{d}, \frac{d}{d_p}, \frac{t}{d_p}\right). \tag{3}$$

Equation (1) reflects the effect on convective heat transfer of the velocity and properties of the granular material (Peclet number), geometric parameters of the surface (simplexes h/d, t/d), conditions of flow around it (simplex d/d_p), character of the movement of the layer in the spaces between the fins (simplex t/d_p). Equation (2) takes into account the efficiency of the fins as a function of their size, heat conductivity, and cooling conditions.

Since the character of flow of the layer around the fins permits expecting considerable nonuniformity of the distribution of heat transfer over their surface, its effect must be taken into account, which can be done by means of the correction C. As is known, nonuniformity causes redistribution of the temperatures in the fin and a change of efficiency in comparison with that calculated for conditions α_f = const. Correction C is a function of the character and degree of nonuniformity, which in turn depends on the velocity of the layer, size of the fins, and conditions of their circumfluence. Other conditions being equal, the value of the correction is also affected by the mh number. All these factors are taken into account by Eq. (3).

On the basis of the data obtained from Eqs. (1)-(3) we can calculate the reduced heat-transfer coefficient and the quantity of heat for finned surfaces made of any material:

$$\alpha_{red} = \bar{\alpha}\left(E_t C \frac{F_f}{F_\Sigma} + \frac{F_b}{F_\Sigma}\right), \tag{4}$$

$$Q = \alpha_{red} F_\Sigma \bar{\vartheta}_b \tag{5}$$

In accordance with dimensionless equation (1) we analyzed in sequence the effect of each of the characteristic factors on the rate of heat transfer. Figure 1a, b shows the particular dependences $\overline{\mathrm{Nu}}$ = f(h/d)

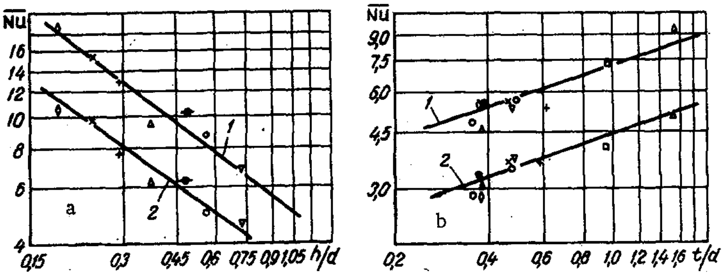

Fig. 1. Nusselt number vs simplex h/d (for t/d = 0.66 = idem) (a) and simplex t/d (h/d = 0.9 = idem) (b): 1) Pe = 700; 2) Pe = 100. See Fig. 2 for symbols.

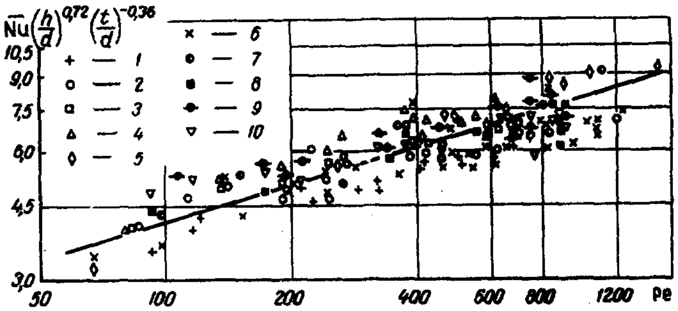

Fig. 2. Generalized relation of heat transfer of a dense layer with transversely finned cylinders: 1) t = 22 mm, d = 33.5 mm, h = 10 mm; 2) t = 12 mm; 3) t = 32 mm; 4) t = 52 mm; for 2-3) d = 33.5 mm, h = 30 mm; 5) h = 10 mm; 6) h = 20 mm; 7) h = 30 mm; for 5-6) d = 54.2 mm, t = 22 mm; 8) h = 10 mm; 9) h = 20 mm; 10) h = 30 mm; for 8-10) d = 42.1 mm, t = 22 mm.

and $\overline{Nu} = f(t/d)$ (the Peclet number is taken as a parameter). Their character follows from the effect of the spacing, height of the fins, and diameter of the cylinder described above. We see from the graphs that the exponents of the simplexes h/d and t/d are practically independent of the Peclet value. Within the investigated limits of variation of the parameter d/d_p (139–226) its effect on heat transfer with finned cylinders (unlike unfinned) is not detected. Nor are the conditions of movement in the spaces between fins affected in the investigated range of t/d_p (40–100). This can be explained by the fact that unconstrained movement occurs when $(t - \delta)/d_p > 30$.

Figure 2 presents the results of a generalization of all experimental data. The dependence in Fig. 2 can be described with a probable error of ±8% by the equation

$$\overline{Nu} = 1.38 Pe^{0.28}\left(\frac{h}{d}\right)^{-0.72}\left(\frac{t}{d}\right)^{0.36}, \tag{6}$$

which holds in the following limits: $0.18 \leq (h/d) \leq 0.9$; $0.36 \leq (t/d) \leq 1.55$; $70 \leq Pe \leq 1500$; $139 \leq (d/d_p) \leq 226$; $40 \leq (t/d_p) \leq 100$; $(t - \delta/d_p) > 30$. The diameter of the cylinder, average temperature of the material, and its velocity in the minimal section are taken as the characteristic parameters in Eq. (6).

As was indicated above, the correction C was calculated on the basis of experimental data on the temperature distribution in the fin. It is shown in [2] that in this method of determination the correction is somewhat conditional and takes into account not only the nonuniformity but also the difference between the weighted mean heat-transfer coefficient $\overline{\alpha}$ (with respect to which E_t is calculated) and the true coefficient of heat transfer to the surface of the fins themselves ($\overline{\alpha}_f$). However, for high finning coefficients, when the surface of the fins greatly exceeds the surface of the cylinder ($F_f \gg F_b$), the values of $\overline{\alpha}$ and $\overline{\alpha}_f$ are close and C reflects mainly the effect of nonuniformity.

The pattern of circumfluence described above indicates a noticeable change of the rate of heat transfer over the surface of the fins – over the radius and angle of rotation. The maximum nonuniformity radially

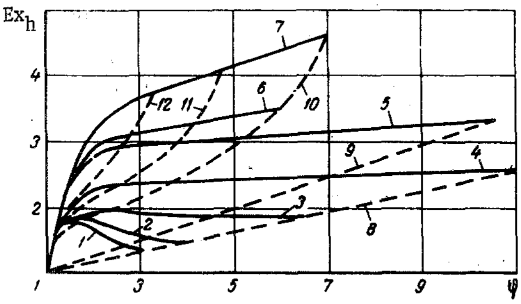

Fig. 3. Relative specific heat extraction as a function of the finning coefficient: 1) t = 52 mm; 2) 32; 3) 22; 4) 12 (for 1-4, d = 33.5 mm); 5) t = 8 mm, d = 22 mm; 6) respectively 6.6 and 15; 7) 6.6 and 10; 8) h = 30 mm; 9) 20; 10) 10; 11) 7; 12) 4.5 mm.

should occur in the frontal part, where the fin lies in the stagnant zone whose thickness at the base is considerably greater than near the apex. In the region of the equator and stern the fin is washed by moving material and therefore the change of character of movement and heat transfer radially cannot be very substantial. A maximum nonuniformity over the angle of rotation should also be expected between the frontal part and equator, where the size of the stagnant zone changes noticeably. We can assume that the heat-transfer coefficient reaches a maximum near the apex in the equatorial region and a minimum near the base in the frontal part. An increase of heat transfer from the base to the apex leads to a decrease of efficiency in comparison with its value at $\bar{\alpha}_f$ = const. This difference is more appreciable, the higher the fin and greater the velocity of the layer.

The method used did not permit revealing the effect of all factors figuring in Eq. (3). Treatment of the experimental data led to the following approximate formula for $0.5 \leq mh \leq 1.1$:

$$C = 1 - 0.1\,mh. \tag{7}$$

When $mh < 0.5$ we can take $C \cong 1.0$. Qualitatively, formula (7) agrees with the results obtained in [2] for straight longitudinal fins. However, it is in need of refinement on the bases of data on local heat transfer and solution of the problem of heat conductivity of circular fins for α_f = var. It should be noted that the analytic solution is extremely complicated owing to the two-dimensional field of variation of the heat-transfer coefficient.

It is of interest to compare the heat-transfer characteristics of finned and smooth cylinders. It can be performed on the basis of Eq. (6) and data of [7] for smooth cylinders and leads to the following generalized relation for the relative intensity of heat transfer:

$$\frac{\overline{\mathrm{Nu}}}{\overline{\mathrm{Nu}}_0} = 3.07\left(\frac{h}{d}\right)^{-0.72}\left(\frac{t}{d}\right)^{0.36}\mathrm{Pe}^{-0.03}\left(\frac{d}{d_p}\right)^{-0.39} \tag{8}$$

(here $\overline{\mathrm{Nu}}_0$ is the criterion of the average rate of heat transfer for a smooth cylinder when Pe = idem). Equation (8) is applicable within the same limits as (6). As we see from (8), the effect of circular finning on heat transfer is determined by its geometric parameters: relative height of the fins and relative spacing. The simplex d/d_p, which characterizes the conditions of flow around the smooth cylinder, also has an effect. The role of the Peclet number is small. Calculations show that when $h/d = 0.9$ and $t/d = 0.36$ the rate of heat transfer of a finned cylinder as compared to a smooth one drops 67% and when $h/d = 0.18$ and $t/d = 1.55$ it increases 62% ($d/d_p = 150$ and $Pe = 200$ were used in the calculations). Thus, low and widely-spaced fins promote an intensification of convective heat transfer owing to separation-free flow around their side surfaces and insignificant deterioration of the conditions of flow around the supporting cylinder.

This comparison still does not permit making a conclusion concerning the expediency of using finning and its optimal characteristics. Such a possibility is manifested on comparing finned and smooth surfaces with respect to specific heat extraction (per unit length of cylinder) at the same thermal heads and velocities of the layer. The relative specific heat extraction

$$\mathrm{Ex_h} = \frac{q_l}{q_{l_0}} = \frac{\alpha_{\mathrm{red}}F_\Sigma}{\bar{\alpha}_b F_b} = \frac{\alpha_{\mathrm{red}}}{\bar{\alpha}_b}\,\varphi \tag{9}$$

takes into account not only the conditions of convective heat transfer, just as Eq. (8), but also the effect of the thermal resistance of heat conductivity of the fins and the relationship of the surface of finned and smooth cylinders (finning coefficient φ). Equation (9) serves as an indicator of the thermal efficiency of developed surfaces. In accordance with (9), to increase the relative heat extraction it is necessary to increase the finning coefficient, which can be done by increasing the height or decreasing the spacing of the fins. But according to (8) this reduces heat transfer. The question arises, to what limits and how should the compactness of the surface be increased. The answer is given by Fig. 3, where the functions $\mathrm{Ex_h} = f(\varphi)$ are given for cylinders of various diameters when h = var, t = idem (solid lines 1-7) and when h = idem, t = var (dashed lines 8-12). The range of variation of the geometric parameters lies within the limits covered by Eq. (8). An analysis of curves 1-7 shows that when t = idem and φ increases owing to an increase of the height of the fins, the specific heat extraction at first increases very noticeably and then slows or stops. This is due to the fact that when certain values of φ are exceeded the decrease of $\bar{\alpha}$, E_t, and C become commensurable and even begins to predominate over the increase of surface. A decrease of the spacing when h = idem produces much better results; in this case the specific heat extraction increases in the entire range of φ (curves 8-12). This is explained by the fact that the simplex t/d affects only the values of $\bar{\alpha}$ (much more weakly than h/d), and E_t and C remain practically unchanged.

When φ = idem, maximum heat extraction is attained at minimum diameters of the cylinder. Thus, the smaller the diameter of the cylinder, the more expedient the use of fins. In this case it is necessary to use low fins, providing the necessary finning coefficient by reducing the spacing. We can recommend values $h \leq 15$ mm, $t \leq 10$ mm. When selecting the minimum values of spacing it is necessary to provide unconstrained movement in the spaces between the fins $(t - \delta)/d_p > 20$-30. The working conditions of finning when $10 \leq (t - \delta)/d_p \leq 20$ require study, since in this case heat transfer can be adversely affected by constraint of movement, leading to loosening of the layer and drop of its effective heat conductivity, which must be taken into account in the generalized relation. The use of $(t - \delta)/d_p < 10$ is not allowed owing to the possibility of the jamming of particles.

Thus, for a dense layer circular finning with rational geometric parameters permits increasing specific heat extraction by a factor of 3.5-4.5 and increasing considerably the compactness of the heat-transfer surface. This is all the more valuable since additional expenditures are not needed for transporting the granular material. In addition, still another advantage is attained in this case – more uniform heating of the material, which is important in a number of production processes. All conclusions are valid for various materials with good free-flowing properties. For each specific case this analysis of the thermal efficiency should be supplemented by a comparison on the basis of weight and cost indexes.

NOTATION

Q	is the total quantity of heat transferred from the finned cylinder to the layer, W;
q_l, q_{l0}	are the specific heat flux per unit length of finned and smooth cylinders, respectively, W/m;
F_Σ, F_f, F_b	are the surface of finned cylinder, fins, and base, respectively, m^2;
$\bar{\vartheta}$	is the weighted mean excess temperature of finned cylinder, °C;
$\bar{\vartheta}_b$	is the average excess temperature of base of cylinder, °C;
D	is the diameter of fins, m;
$mh = h\sqrt{2\bar{\alpha}/\delta\lambda}$	is the dimensionless complex;
$\overline{Nu} = \bar{\alpha}d/\lambda_{ef}$	is the Nusselt number;
$Pe = wd/a_{ef}$	is the Peclet number.

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