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Surface nonisothermness effect on profiled surface condensation enhancement

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Abstract—The numerical solution results of condensation on fin, taking fin and tube wall nonisothermness into consideration, are presented. The problem is solved in a two-dimensional statement. Solution results show considerable two-dimensionality for finned tube temperature distribution. Fin wall nonisothermness reduces fin vapor condensation efficiency

INTRODUCTION

As it is known, surface nonisothermness considerably affects the efficiency of finning used to enhance the condensation. Temperature distributions along the height of various configuration fins for the case when variations of local heat transfer coefficient are determined by the Nusselt equation and wall temperature (constant along the fin thickness) are obtained in the papers [1–3].

In papers [4, 5], where surface tension force effect on finned tube condensation enhancement was studied, the wall nonisothermness influence was taken into consideration by dependences, obtained, for instance, in ref. [6] for the case of finned surface convective heat transfer, when the fin heat transfer coefficient is constant along the fin height.

At the same time it is known that one-dimensional solutions bring considerable error, even for the case when the heat transfer coefficient is constant along the fin height. In the present work on fin temperature, two-dimensional influence is considered for the case of condensation, when the heat transfer coefficient changes along the fin height, and also depends upon the local temperature difference.

PROBLEM STATEMENT

The present paper deals with the problem of condensation of pure static vapor on finned surfaces of two types:

(1) with the fins produced by the deformation of tube wall (Fig. 1a);

(2) with the wire fins [Fig. 1(b)].

Total thickness of tube condensers tube wall (δ_w+h) [Fig. 1(a)] is much less than its diameter in most of the practice cases. It allows us to assume this cylindrical wall is plane. Figure 1 shows part of the longitudinal section of the tube wall, whose fin height is negligible in comparison with its fin length. The thermal conductivity equation for such a wall has following form:

$$\frac{\partial^2 T_w}{\partial x^2}+\frac{\partial^2 T_w}{\partial y^2}=0. \qquad (1)$$

In accordance with Fig. 1, the following boundary conditions are used:

$$x_w=0 \quad x_w=\frac{t}{2} \quad \frac{\partial T_w}{\partial x}=0 \qquad (2)$$

$$y_w=0 \quad \lambda_w\frac{\partial T_w}{\partial y}=-\alpha_o(T_w-T_o) \qquad (3)$$

$$y_w=\Delta_w=f(x_w) \quad \lambda_w\frac{\partial T_w}{\partial y}=-\frac{\lambda_1}{\partial_{fl}}(T_s-T_w). \qquad (4)$$

During the condensation on the wire finned surface [Fig. 1(b)] we neglected the condensation on the wire itself. Thus, boundary conditions (2) and (3) are the same, and equation (4) will be valid with $y_w=\Delta_w$. The thickness of the condensate film δ_{fl}, which is in equation (4), is determined by the Nusselt formula for filmwise laminar condensation on a vertical plate

$$\delta_{fl}=\left[\frac{4\lambda_1\nu_1(T_s-T_w)}{r_1F}\right]^{0.25}, \qquad (5)$$

where

$$F=\rho_1 g+\mathrm{d}P/\mathrm{d}\xi. \qquad (6)$$

$\mathrm{d}P/\mathrm{d}\xi$ is a capillary pressure gradient caused by variable curvature of condensate film. There are various ways to calculate the value of $\mathrm{d}P/\mathrm{d}\xi$. In the examined case, only the numerical values of $F\geqslant\rho_1 g$ are important.

The thermal conductivity equation was solved numerically by the relaxation method. Condensate

NOMENCLATURE

$Bi = \bar{\alpha}_{is}\delta_w/\lambda_w$ $Bi_h = \bar{\alpha}_{is}h/\lambda_w$ $Bi_t = \bar{\alpha}_{is}t/2\lambda_w$ biot numbers
- d tube diameter [m]
- d' wire diameter [m]
- g gravitational acceleration [m s^{-2}]
- F resultant force [N]
- h fin height [m]
- K overall heat transfer coefficient [W m^{-2} K^{-1}]
- $2l$ fin pitch [m]
- $N_f = (2\bar{\alpha}_{is}h^2/\lambda_w t)^{0.5}$
- P pressure [N m^{-2}]
- Q heat flux [W]
- q heat flux based on nominal surface area [W m^{-2}]
- r vaporization heat [kJ kg^{-1}]
- $Re = vd_i/\nu$ Reynolds number
- T temperature [K]
- $\Delta T = T_s - T_w$ wall-steam temperature difference [K]
- $\Delta T_o = T_s - T_o$ wall-cooling water temperature difference [K]
- v velocity [m s^{-2}]
- X dimensionaless co-ordinate
- x co-ordinate [m]
- Y dimensionless co-ordinate
- y co-ordinate [m].

Greek symbols
- $\bar{\alpha}_{is}$ average isothermal heat transfer coefficient [W m^{-2} K]
- $\bar{\alpha}_o$ average experimental heat transfer coefficient [W m^{-2} K]
- $\bar{\alpha}_v$ vapor-to-condensate heat transfer coefficient [W m^{-2} K]
- α_x, α local heat transfer coefficients [W (m^2 K)$^{-1}$]
- γ_h own meanings
- δ thickness [m]
- Δ total tube wall thickness [m]
- ε_f finning efficiency
- Θ relative temperature difference
- λ thermal conductivity [W^2 m^{-1} K^{-1}]
- ν kinematic viscosity [m s^{-1}]
- ξ running coordinate [m]
- ρ density [Kg m^{-3}].

Subscripts
- f fin
- fl film
- i inner surface
- is isothermal
- l liquid
- o cooling medium, outside surface
- s saturation
- v vapor
- w tube wall
- x co-ordinate
- y co-ordinate
- 1 one-dimensional model
- 2 two-dimensional model.

physical properties, upon which condensation enhancement depends considerably, were changed in a wide range: coolants, ammonia and water were used. The wall thermal conductivity coefficient λ_w changed from 12 to 400 W m^{-1} K; fin height h—from 1 to 3 mm and fin root thickness t—from 0.67 to 4.8 mm ($t \geqslant 2$ mm was used for wire finned surface, Fig. 1b), $F = 10^4 \ldots 10^6$ N m^{-3}.

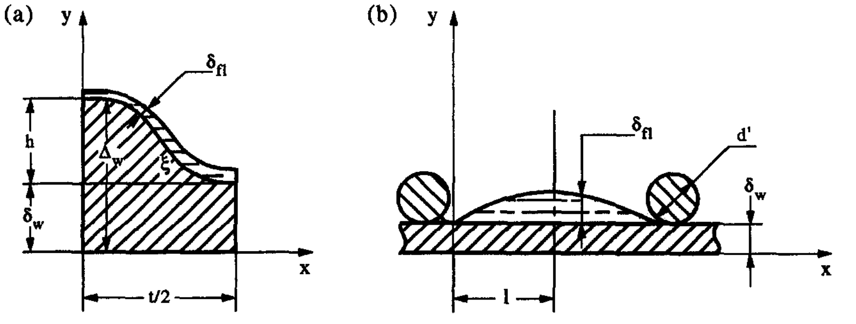

Fig. 1. Finned surface cross sections: (a) condensate film, (b) condensation surface.

SOLUTION RESULTS

It was shown in ref. [7] that for the case when the heat transfer coefficient is constant along the fin height, the account of the two-dimensional temperature pattern effect on it results in a considerable decrease (up to 70%, depending on Bi_t numbers) of the finning efficiency. It is shown in ref. [6], that circular fin and tenon characteristics from paper [9] differ slightly from straight fin characteristics presented in ref. [7].

In reality, with fin vapor condensation, the local heat transfer coefficient $\alpha = \lambda_l/\delta_{fl}$ varies with the fin height. The heat transfer coefficient dissimilarity effect on a one-dimensional fin temperature pattern is considered in many papers. In refs. [6, 10], the problem was solved for the case when α doesn't depend upon ΔT, and varies with the fin height by the power law.

As $\alpha \sim \xi^{-0.25}$ during the condensation, i.e. α variations with ξ are comparatively small, thus, fin height temperature distribution does not differ considerably from corresponding distributions, when α = constant (Fig. 2). Rectangular fin efficiency values do not differ from each other if we compare the cases of α = variable and α = constant [6]. It is shown in paper [6], that the efficiency correction factor, concerned with fin form doesn't exceed 8% with $N_f \leqslant 3$.

The temperature pattern two-dimensional influence in both α = variable and α = constant [7] cases is fairly considerable (Fig. 3). There are various t and λ_w rectangular fin characteristics in Table 1. All the table data are obtained with the use of equation (5) with ΔT_o = 1–2 K and water physical properties at T_s = 373 K. In the table, Θ_1 and Θ_2 are relative temperature differences, calculated, respectively, by one-dimensional and two-dimensional models; indices of $0.1h$ and $0.5h$ correspond, respectively, to the point's fin tip and fin middle parts. The heat transfer coefficients $\alpha = \lambda_l/\delta_{fl}$, being included in N_f, Bi_h and Bi_t, are determined by equation (5) with $\xi = t/2$ with

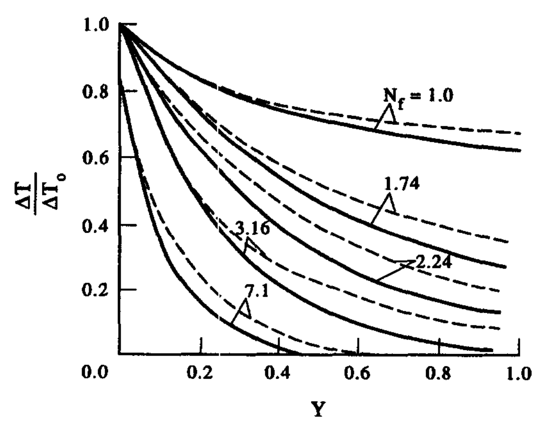

Fig. 2. Temperature differences in one-dimensional rectangular fin: continuous lines for vapor condensation calculations [2, 4]: dashed lines for calculations when α = constant along the fin height h [6]. (1) $N_f = 1$, (2) 1.74, (3) 2.24, (4) 3.16, (5) 7.1.

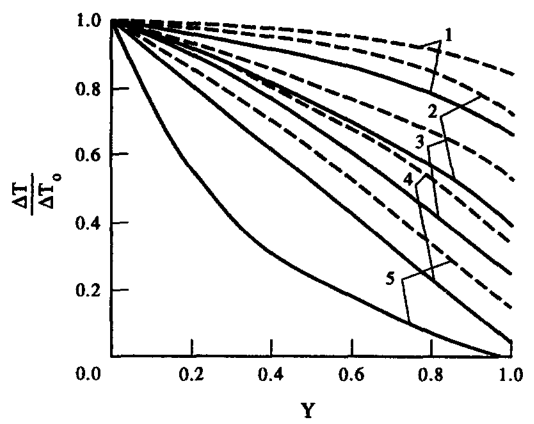

Fig. 3. Fin height temperature distribution in triangular fin: two-dimensional fin is shown by continuous lines, one-dimensional fin by dashed lines. (1) $N_f = 0.415$, (2) 0.58, (3) 0.91, (4) 1.3, (5) 1.84.

assumptions that $\Delta T = T_s - T_w$ = constant and equals ΔT averaged by the fin height.

It follows from the table, that Bi_t, plotted vs fin root t [7] cannot be used as a fin unambiguous characteristic. Thus, efficiencies of fins 2 and 8, having practically the same values of λ_w and Bi_t, are considerably different from each other. It is impossible to compare the fins by their Bi_h, whose h is a characteristic dimension. It is seen, for instance, from comparison of fins 5 and 8, where the bigger the Bi_h, the bigger the $\Theta_{0.1h}$ and $\Theta_{0.5h}$. The same statement is valid for fins 6 and 7. N_f, which decreases (Θ_2/Θ_1) ratio with its growth in all the cases, is most sensible in fin thermal characteristic.

The ratio of average heat fluxes with the same ΔT_o when $0.8 \leqslant N \leqslant 5$ can be determined by the following formula:

$$\frac{q_2}{q_1} = 0.931 N^{-0.4}. \tag{7}$$

This formula is obtained as a result of triangular fin calculation processing (Fig. 3). It is shown in refs. [6, 8], that for one-dimensional fins of various configuration and with equal heights and cross-sectional areas, these fins may differ from each other with their fin efficiencies by more than 25% when $N_f > 2.5$. This restriction is valid when $\lambda_w < 20 \ldots 40$ W m^{-1} K concerning steam condensation with $\Delta T = 1K$ on a fin of fin root $t > 0{,}5 \ldots 1$ mm and h = 1–2 mm.

For a wire finned tube (δ_w = constant and $t = 2 \ldots 5$ mm), Fig. 1(b), temperature distribution along the x-coordinate (Fig. 4), with $x/l > 0.05$ is approximated by the following dependence:

$$\frac{\Delta T}{\Delta T_o} = \Theta = (x/l)^{0.067 Bi}. \tag{8}$$

Here $Bi = \bar{\alpha}_{is}\delta_w/\lambda_w$; where $\bar{\alpha}_{is}$ is the average heat transfer coefficient obtained under conditions of $x = l$ and $\Delta T = \Delta T_o$. It follows from equation (8), that the

Table 1. Fin characteristics

Fin number	h mm	t mm	λ_w W m^{-1} K^{-1}	N_f —	Bi_h —	Bi_t —	$(\theta_2/\theta_1)_{0.1h}$ —	$(\theta_2/\theta_1)_{0.5h}$ —
1	1	2.4	400	0.43	0.22	0.27	$\frac{0.78}{0.87}$	$\frac{0.86}{0.97}$
2	1	0.67	110	0.82	0.96	0.8	$\frac{0.41}{0.54}$	$\frac{0.65}{0.85}$
3	1	2.4	50	1.33	2.14	2.7	$\frac{0.12}{0.31}$	$\frac{0.45}{0.54}$
4	1	2.4	30	1.57	3.0	3.6	$\frac{0.09}{0.23}$	$\frac{0.36}{0.48}$
5	2	4.8	30	1.96	4.6	5.52	$\frac{0.03}{0.12}$	$\frac{0.23}{0.3}$
6	1	2.4	12	2.53	7.53	9.0	$\frac{0.01}{0.07}$	$\frac{0.2}{0.26}$
7	2	2.4	30	2.9	5.05	3.0	$\frac{0.0}{0.036}$	$\frac{0.115}{0.21}$
8	5	2.4	110	3.37	3.2	0.76	$\frac{0.0}{0.019}$	$\frac{0.094}{0.16}$

ratio of average temperature differences of the equal ΔT when T_w = constant and T_w = variable, will be determined by the following dependence:

$$\tilde{\Theta} = \frac{\overline{\Delta T}}{\Delta T_o} = \frac{1}{1+0.067Bi}. \quad (9)$$

Nonisothermness parameter $\tilde{\Theta}$ determined by dependence (9), allows one to take into approximate account the nonisothermness effect on the condensation intensity surface of various thermal conductivity walls. It is known that efficiency of heat exchange surface finning ε_f is determined by the following dependence:

$$\varepsilon_f = \frac{Q}{Q_{is}}, \quad (10)$$

where Q and Q_{is} are the heat fluxes transferred, respectively, by isotherm and nonisotherm heat exchange surfaces with equal meanings of ΔT_o.

Thus, dependence (10), having taken equation (9) into account (when the surfaces are equal to each other) may be transformed in the following way:

$$\frac{Q}{Q_{is}} = \frac{\bar{\alpha}\overline{\Delta T}}{\bar{\alpha}_{is}\overline{\Delta T}_{is}} = \frac{\bar{\alpha}}{\bar{\alpha}_{is}}\tilde{\Theta}. \quad (11)$$

The heat flux correlation for heat exchange surfaces of the walls of various thermal conductivity is obtained from equation (11), when $\bar{\alpha} = \bar{\alpha}_{is}$:

$$\tilde{q} = \tilde{\Theta} = \frac{q}{q_{is}} = \frac{1}{1+0.067Bi}. \quad (12)$$

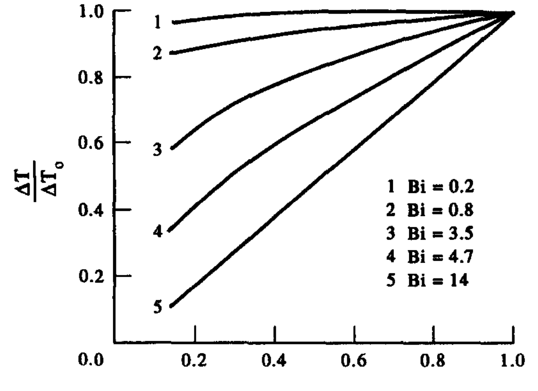

Fig. 4. Surface temperature distribution along the x-coordinate [Fig. 1(b)] on a section of 0–l between the wire fins.

EXPERIMENTAL INVESTIGATION OF NONISOTHERM SURFACE CONDENSATION

Determination of heat transfer coefficients in a case when T_w = variable is complicated by the problem of correct measurement of wall temperature. For the low fin tubes, that are normally used in condensers, an exact wall temperature determination is impossible nowadays. That is why, for such tubes of low thermal conductivity, the indirect method (Wilson's) is used to determine vapor-to-wall average heat transfer coefficients $\bar{\alpha}_v$ by the measured average overall heat transfer coefficient K, and enough exactly determined by known methods the heat transfer coefficient to the cooling medium, $\bar{\alpha}_o$.

However, even in this case, if the tube is finned by wall deformation (Fig. 1a), a difficulty in determination of the tube wall thermal resistance appears. That is why condensation on wire finned tubes of the same dimensions was investigated in our experiments. We investigated the condensation of pure steam at $P = 1.01 \ldots 1.02$ bars and steam velocity $v_v = 0.2$ m s^{-1} on single tubes. Cooling water flowed inside the tubes of inner diameter $d_i = 13$ mm and outer $d_o = 16$ mm with the velocity $v_l = 17.6$ m s^{-1} at average temperatures from 30 to 50°C. Finning was made by wire of $d' = 1.5$ mm, coiled on the tube with the distance between the fins $l = 10$ mm. Experimental tubes were made from brass, copper–nickel–iron alloy (CNI-5-1) and German silver with values of λ_w, respectively, of 100, 50 and 25 W m^{-1} K. For average wall temperature measuring, the resistance thermometer was put spirally into a wall of a smooth brass tube, that allowed us to determine the average wall temperature and heat transfer coefficients $\bar{\alpha}_v$ and $\bar{\alpha}_o$. Comparison of $\bar{\alpha}_o$ values obtained experimentally with $Re \geqslant 10^4$ (for water) with ones obtained from known calculation dependences for liquid turbulent flow in tubes, showed that their difference doesn't exceed 5%. This fact allowed us to determine the finned tube $\bar{\alpha}_o$ in experiments by calculation dependence.

Figure 5 shows values of experimental $\bar{\alpha}_v$ for various single tubes with spiral wire finning. It is clearly seen from Fig. 5 that a decrease of tube material thermal conductivity reduces heat transfer.

The experimental lines of $\tilde{\Theta} = f(Bi)$ for the equal $\bar{\alpha}_v$ may be obtained by dependence equation (12) with the use of experimental data from Fig. 5. Figure 6 shows the values of $\tilde{\Theta} = \tilde{q}$ (for the case when value of $\bar{\alpha}_v$ are equal and with $\tilde{q}_{is}$ from equation (12) levelled to q, obtained on brass tube), depending upon the Bi, obtained by use of Fig. 6's experimental data; and also the calculated line, built by equation (12).

The stronger effect of the tube wall thermal conductivity on condensation intensity can be explained in the following way. During the numerical solution of the problem, the film thickness changing law was given to us by the equation (1), i.e. film thickness $\delta_{fl} \sim x^{-0.25}$. In reality, as it is shown in refs. [11, 12],

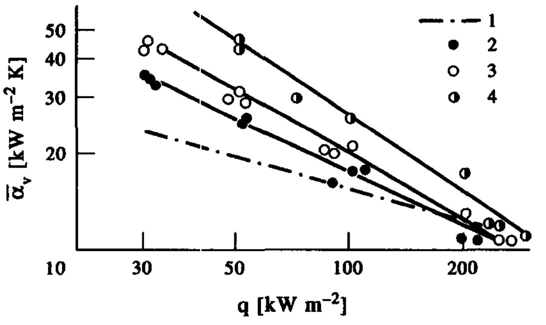

Fig. 5. Effect of heat flux q on average heat transfer coefficients $\bar{\alpha}_v$ during the condensation on a single tube: (1) smooth tube (data obtained by the Nusselt formula), (2)–(4) spiral finning tubes made from: (2) German silver, (3) copper–nickel–iron alloy, (4) brass.

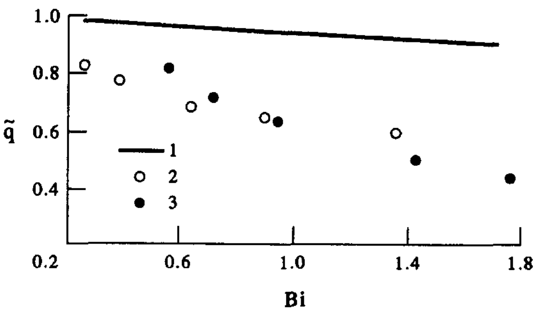

Fig. 6. $\tilde{q}$ vs Bi for experimental data presented in Fig. 5: (1) calculated line, (2), (3) experimental lines [(2) copper–nickel–iron alloy tube, $\lambda_w = 50$ W m^{-2} K, (3) German silver tube, $\lambda_w = 25$ W m^{-2} K^{-1})].

owing to the surface tension effect, δ_x depends upon x to a greater degree. It brings a greater degree of wall nonisothermness.

CONCLUSION

A two-dimensional thermal conductivity equation for tube finned wall is solved during the vapor condensation on it. Considerable fin wall nonisothermness, which increases with the non-dimensional number N_f, is noted.

Solution of the finned wall thermal conductivity one-dimensional equation gives a lower degree of nonisothermness than the two-dimensional equation does.

Experiments on pure steam condensation on wire finned tubes having various thermal conductivity were done and it was noted that the lower the tube thermal conductivity, the lower the heat transfer.

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