temperature variations suggests the condition $sl^2/(4\pi^2 Kr^2 T_x) \ll 1$.

Finally, since many of the parameters determining ablation problems are poorly determined, the theory above may supply useful estimates for certain engineering requirements.

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ANALYSIS OF NUSSELT-TYPE CONDENSATION ON A TRIANGULAR FLUTED SURFACE

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NOMENCLATURE

- a, amplitude of flute [m];
- A, ratio of flute amplitude to pitch, a/p;
- g, gravitational constant [m s^{-2}];
- h_w, conductance of wall [W m^{-2} K^{-1}];
- H_v, latent heat of vaporization [J kg^{-1}];
- K, thermal conductivity [W m^{-1} K^{-1}];
- Nu, Nusselt number, Up/K_f;
- p, period (pitch) of flute [m];
- Re, Reynolds number, $4w/(X_L\mu)$;
- S, thickness of unfluted zone of tube wall [m];
- T, temperature [K];
- U, average overall heat-transfer coefficient [W m^{-2} K^{-1}];
- w, axial mass flow rate of liquid [kg s^{-1}];
- X_L, perimeter of half-flute [m].

Greek symbols

- δ, condensate-film thickness [m];
- δ_0, condensate-film thickness at midpoint of meniscus [m];
- λ, dimensionless physical property group, $(4\rho^2/\mu^2)g\delta_0^4/X_L$;
- μ, viscosity of condensate [Pa s];
- ρ, density of condensate [kg m^{-3}];
- σ, surface-tension coefficient [N m^{-1}];
- Ω, dimensionless physical property group, $K_f(T_v - T_c)/(\sigma H_v ap)$.

INTRODUCTION

CONDENSATION on a fluted surface and the resulting enhancement in the heat transfer coefficient was first recognized by Gregorig [1]; however, the concept was not fully applied until recently. Various theoretical studies [2–7] and experimental studies [8–12] show that the phenomena governed by surface tension can enhance the condensate-film coefficient. In a previous study in which condensation on a vertical 'cosine' fluted surface was analyzed, wall resistance was assumed to be negligible [2]. Edwards *et al.*, in their analysis of condensation on a horizontal tube with transverse flutes, mentioned the importance of the wall resistance [3]. Fuji and Honda solved, for a cosine flute, the difficult set of equations that numerically describes heat transfer in the condensate film and the tube wall, and they introduced the concept of a representative value for wall thickness [4]. However, the importance of wall resistance is not yet fully understood. The present study analyzes condensation on a vertical 'triangular' fluted surface and also investigates the importance of the wall resistance.

DESCRIPTION OF THE PROBLEM

Figure 1 shows condensation on a vertical fluted surface. The figure also defines three coordinate systems (a cylindrical system with a vertical z axis and two rectangular systems) as well as some important geometric parameters.

Unlike condensation on a smooth vertical surface, condensate film on a fluted surface in two directions: vertically (due to gravity) and horizontally from the crest to the low point of the trough (due to the surface-tension force). As a result, the condensate accumulates in the trough, leaving only a very thin film near the crest. The heat transfer coefficient of the condensate film is large near the crest, while it is small at the low point of the trough. This results in a circumferentially nonuniform heat transfer flux in the tube wall. It is expected that the temperature at the condensate–wall interface will vary from almost saturation temperature, T_v at the crest to some lower value at the bottom of the trough. It is also expected that heat fluxes at the coolant–wall interface will be, in general, circumferentially nonuniform, although the coolant heat transfer coefficient is assumed to be constant.

An exact analysis of this problem could be difficult, if not impossible. However, previous work [2] leads to some assumptions that simplify the problem somewhat. The principal assumption used in this analysis is that horizontal cross flow is negligible at the low point of the trough where the film is thick, while the vertical flow is negligible near the crest.

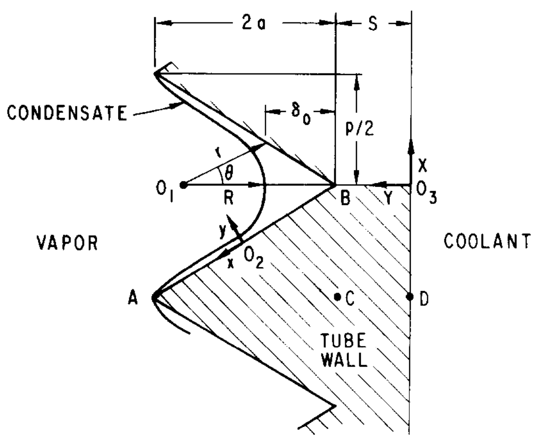

FIG. 1. Schematic diagram of triangular flute.

Therefore, the film thickness at the midpoint of the meniscus at a given axial flow rate of the liquid can be calculated by solving the gravity-controlled equation of motion. Local and average rates of condensation are calculated by solving the surface-tension governed equation of motion and the energy equations for the film and the metallic wall. (For the detailed analysis see ref. [2].)

RESULTS AND DISCUSSION

The equations developed in this analysis are solved using numerical methods. The required physical and geometrical parameters are generated by considering the condensation of ammonia and water vapors on aluminum surfaces with a thermal conductivity of 202 W m^{-1} K^{-1} and on titanium surfaces with a thermal conductivity of 16.6 W m^{-2} K^{-1}. This section discusses the complex phenomena of condensation, considering one factor at a time.

Axial flow rate in the trough

For a given value of the condensate-film thickness δ_0, the Reynolds number Re is calculated using the velocity profile obtained by solving the gravity-controlled motion equation. In this study, the Reynolds number is calculated for the fluted surface, not for its projection. For the triangular flute, a simplified correlation has been developed relating Re to a physical property group λ, with the ratio A (of flute amplitude a to pitch p) as a parameter. The correlation is given by

$$\lambda = (50\,A + 387\,A^{2.67})\,Re \tag{1}$$

and

$$\frac{Re}{Re_f} = \frac{\lambda}{\lambda_f} \tag{2}$$

where $A = a/p$ and the subscript f denotes the condition in which the condensate completely fills the triangular trough (flooding). Similar correlation has been developed for cosine flutes [2]. This correlation helps in calculating the maximum condensate-film thickness δ_0 for a given value of Re; δ_0 is then used in calculating the overall heat transfer coefficient.

Effects of the physical property group Ω

A detailed description of the physical property group has been given in the authors' previous paper on cosine flutes [2]. In the present analysis, Fig. 2 shows the effect of Ω on Nu for very large values of both the thermal conductivity K_S of the wall material and the coolant heat transfer coefficient h_c. For reduced Reynolds numbers Re/Re_f less than 0.05, the curves are nearly linear and parallel to one another. The effect of Ω, therefore, can be expressed in the following form:

$$Nu = f(Re/Re_f,\ A)\Omega^c. \tag{3}$$

The value of c is approximately -0.16.

For values of the reduced Reynolds number greater than 0.05, the Nusselt number drops significantly. The reason is that, for higher reduced Reynolds numbers, the condensate-film thickness changes rapidly when the flute is nearly flooded, from zero (at the crest) to a large value in the trough. (At a reduced Reynolds number of 0.3, the maximum condensate-film thickness is about 0.7 of the total depth $2a$ of the flute.) The effect of the physical group Ω is insignificant in the region of maximum film thickness. The type of deterioration in performance indicated by the drop in the Nusselt number is not observed in cosine flutes. It should be remembered that the drainage capacity (i.e. the axial flow rate for a given maximum condensate-film thickness) of a triangular flute is less than that of a cosine flute, other conditions being the same.

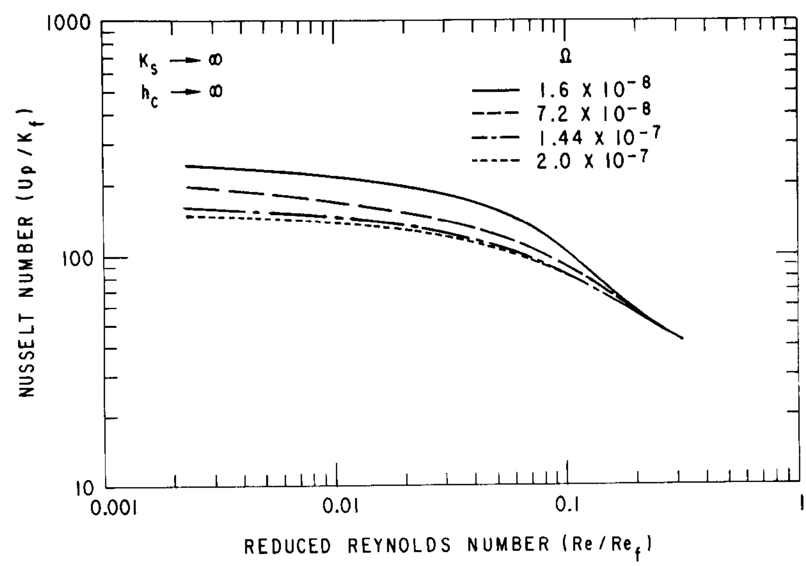

Fig. 2. Dependency of Nusselt number on physical property group Ω.

Table 1. Effective tube-wall conductance*

Re/Re_f	for $K_S \to \infty$	Up/K_f for Case 1	Up/K_f for Case 2	$h_w p/K_f$ for Case 1	$h_w p/K_f$ for Case 2
0	∞	97	1174	97	1174
0.026	130	33	81	44	214
0.063	105	29	69	40	197
0.085	93	27	62	38	184
0.130	74	23	47	34	129
0.240	50	18	31	29	80

*Physical parameters: a = 025 mm, p = 2.0 mm, S = 0.55 mm, $\Omega = 1.44 \times 10^{-7}$, K_S/K_f = 36 (Case 1) 439 (Case 2).

Heat transfer in the flute

In order to better understand the heat transfer process in the metallic wall, temperature profiles for a set of physical parameters are calculated. It is observed that the temperature profiles are not, in general, parallel to the coolant–wall interface, which indicates that heat transfer takes place in both the X and Y directions. Condensate-film thickness varies from zero at the crest to a maximum value at the low point of the trough. As a result, the temperature at the condensate–wall interface is nonuniform. Furthermore the area of the longitudinal section, which is the area available for heat transfer, increases from the crest to the coolant–wall interface. As the interface is approached, the temperature profile tends to become parallel to it. Thus, if the unfluted zone of the tube wall is thick enough, the temperature gradient is uniform at the coolant–wall interface, giving uniform heat flux.

The interaction of the heat transfer phenomena in the condensate-film and in the metallic flute is illustrated in Table 1, in which an effective wall conductance is calculated for various reduced Reynolds numbers. The condensate-film resistance (reciprocal of conductance) is calculated for very large thermal conductivity of the wall material, i.e. $K_S \to \infty$, and this resistance is subtracted from the overall resistance to find the wall resistance. An effective wall conductance for $Re/Re_f = 0$, is calculated by assuming wall thickness equal to $S + a$ (Fig. 1). In the absence of condensation of a vapor, the overall resistance will be minimum at the trough. However, even at low rate of condensation, the trough will be filled with condensate and a sharp drop in the wall conductance is observed. As the reduced Reynolds number increases, the effective wall conductance will drop further. Development of the condensate-film profile is, in general, affected by the heat transfer process in the metallic flute. As a result, by changing thermal conductivity of the flute, the condensate-film profile will change giving a different average film coefficient for the same Reynolds number.

CONCLUSIONS

Condensation of a vapor on a vertical fluted surface is theoretically analyzed. The Reynolds number and Nusselt number are calculated for a range of physical parameters and their effects are studied. The present analysis shows that the wall resistance and condensate-film resistance cannot be separated. The wall, in effect, becomes an integral part of the condensation process.

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