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TECHNICAL NOTES

Effects of friction losses in water-flow pipe systems on the freeze-off conditions

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INTRODUCTION

THE FREEZING process of water flow involves interactions among the flow, the shape of ice and the heat transfer at the ice–water interface. Such interactions result in an instability of the interface. During melting or formation of ice, the dominant factor related to the instability is the heat flux from the interface into the ice. On the other hand, under steady-state conditions where the ice-side and water-side heat fluxes are equal, the influential factor is the thickness of the ice layer [1].

When the ice is thin, the thickness of the ice layer increases monotonously along the pipe. In that case the interface is stable and a number of experimental as well as theoretical studies have been reported. However, when the ice is thick, the layer becomes unstable and an ice-band structure which shows a typical feature of a flow passage with a cyclical variation in cross-section is observed [2, 3]. A separation of flow occurring at each ice-band gives rise to a large pressure drop and a reduction of flow rate. When the ice layer is thick, the reduction is accelerated and freeze-off of water flow sometimes takes place.

Hirata and Ishihara [4] examined the conditions for the onset of freeze-off of a pipe flow. They found that the freeze-off conditions were described in terms of a modified Reynolds number based on a total pressure at inlet of pipe, Re_p, and a cooling temperature ratio, θ; the former was a preventive factor and the latter a promotive one. In the present study the effect of friction factor in piping systems, which is usually caused by many bends, valves, etc., on the freeze-off conditions are examined experimentally. Analytical considerations based on the experimental results are also presented to find out nondimensional parameters governing the freeze-off conditions.

EXPERIMENTAL APPARATUS AND PROCEDURE

The experimental apparatus, as shown in Fig. 1, consisted of a test section, a friction-factor controlling valve and two circulation systems of water and coolant (30% $CaCl_2$ solution), of which temperatures were controlled by PID-controlled heaters and refrigeration units. The test section was constructed of two tubes in the vertical position; the inner one was a copper tube whose dimensions were 19.9 mm I.D., 697 mm long and 1.1 mm wall thickness; the outer one was a 36-mm-I.D. steel tube. The water in which freezing occurred was pumped through the copper tube and the coolant was circulated through the space between the two tubes. To examine the effect of friction factor on the freeze-off conditions, a sluice valve was installed in the water circulation system and the friction factor was changed by controlling the opening of the valve.

The wall temperature of the copper pipe, T_w, was evaluated from three thermocouples located along the pipe and the water temperature in the pipe, T_∞, was estimated from the mean value of the inlet and outlet of the test section. The total pressure at the inlet of the pipe, P_0, which was measured by using the outlet value as a datum, was controlled by the valve of the by-pass pipe. The experimental range covered was

$$D = 19.9\ \text{mm}: \quad 2.74\times10^4 \leqslant Re_p \leqslant 1.6\times10^5$$
$$3.08 \leqslant \theta \leqslant 19.8$$
$$0 \leqslant \xi \leqslant 3.6\times10^4$$

where Re_p is a modified Reynolds number defined by equation (5), θ is a cooling temperature ratio, $\theta = (T_f - T_w)/(T_\infty - T_f)$ and ξ is a friction-factor coefficient in piping systems.

NOMENCLATURE

- B — diameter ratio, d/D
- d — minimum diameter at contraction region of ice-band
- D — diameter of pipe
- g — acceleration of gravity
- h_w, $h_{w,p}$ — head losses in ice-band structure and in piping systems
- L — length of pipe
- n — number of ice-bands in a pipe
- P_0 — total pressure at the pipe inlet measured by using the outlet value as a datum
- Re_d, Re_D — Reynolds numbers, vd/ν, VD/ν
- Re_p — modified Reynolds number based on P_0, equation (5)
- T_f, T_w, T_∞ — freezing, pipe wall and water temperatures
- v — mean velocity of water at contraction region of ice-band
- V — mean velocity of water in a pipe without ice.

Greek symbols

- θ — cooling temperature ratio, $(T_f - T_w)/(T_\infty - T_f)$
- λ^* — thermal conductivity ratio of ice to water, λ_i/λ_w
- ν — kinematic viscosity of water
- ξ — friction-factor coefficient in pipe systems, equation (1)
- ρ — density of water.

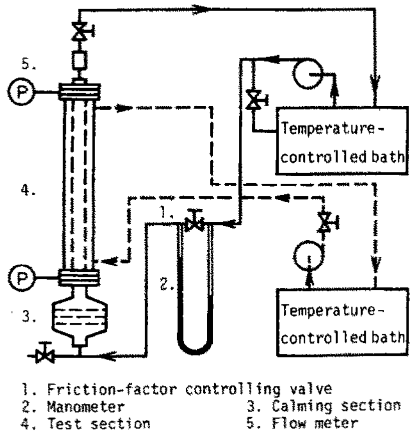

FIG. 1. Schematic representation of the experimental apparatus.

RESULTS OF ANALYSIS AND EXPERIMENT

Let $h_{w,p}$ be a head loss due to bends, valves, elbows, etc., in piping systems, then, the frction-factor coefficient, ξ, is defined as

$$h_{w,p} = \xi \frac{V^2}{2g} \tag{1}$$

where V is a mean velocity of water without ice formation and g is the acceleration of gravity. Consider that the ice-band is formed inside a pipe and that the thickness of the ice is large ($V^2 \ll v^2$), a relation between the mean velocity, v, at the contraction region of the ice-band and the total pressure, P_0, at the inlet of a pipe can be obtained from Bernoulli's equation as

$$\frac{P_0}{\gamma} - n\,h_w - h_{w,p} = \frac{v^2}{2g} \tag{2}$$

where n is the number of ice-bands in the pipe and $\gamma = \rho g$. The second term of the LHS describes a pressure drop due to the ice-band structure. The pressure drop in a pipe containing two-dimensional ice-bands is caused mainly by two factors. One is due to viscous drag at the ice–water interface and the other is pressure loss occurring in the sudden expansion downstream of each separation point. As the former is, in general, comparatively smaller than the latter, the head loss caused by each ice-band, h_w, will be

$$h_w = (1-B^2)^2 \frac{v^2}{2g} \tag{3}$$

where B is a diameter ratio, d/D. Substitution of equations (1) and (3) into equation (2) leads to

$$Re_d = \frac{B\,Re_p}{\sqrt{1+n(1-B^2)^2+\xi B^4}}. \tag{4}$$

In this expression Re_p is a modified Reynolds number defined by

$$Re_p = \frac{D}{\nu}\sqrt{2P_0/\rho}. \tag{5}$$

For a steady-state condition, a relation between the minimum diameter at contraction region and θ is given by ref. [4] as

$$\frac{Re_D}{\theta} = -\frac{2\lambda^*}{0.0045}\frac{B}{\ln B} \tag{6}$$

where λ^* is a thermal conductivity ratio of ice to water and Re_D is a pipe Reynolds number. Substitution of equations (4) and (6) into the relation of $Re_D = B\,Re_d$ yields

$$\frac{Re_p}{\theta} = -\frac{2\lambda^*}{0.0045}\frac{1}{B\ln B}\sqrt{1+n(1-B^2)^2+\xi B^4}. \tag{7}$$

It can be deduced that the freeze-off occurs when the thickness of ice exceeds the steady-state value expressed by equation (7). For given conditions of Re_p and θ, the criterion whether the freeze-off occurs or not can be given by $d\theta/dB = 0$ [4]. Differentiation of equation (7) with respect to B and introducing $d\theta/dB = 0$ leads to

$$n = -\frac{1+\ln B+\xi B^4(1-\ln B)}{(1-B^2)^2+(1-B^4)\ln B} \quad \text{for} \quad n > 0. \tag{8}$$

If the value of ξ is given, the limiting values for which a steady-state ice-band exists can be calculated from equations (7) and (8) by eliminating n and B. In Fig. 2, the predicted results of the onset of freeze-off are shown for three values of ξ, in which the abscissa and ordinate are expressed by the LHS terms of equations (7) and (8), respectively; n is represented as functions of θ and L/D [4]. It can be realized that the freeze-off easily occurs for the larger value of ξ, since the flow rate decreases with increasing ξ. In Fig. 2 the experimental data obtained for $\xi = 2600 \sim 3600$ are also shown; the solid and open points indicate a freeze-off and a steady-state ice-band, respectively. It is seen that the predicted result for $\xi = 3000$ coincides well with the experimental data.

A relation which describes the effect of ξ on the freeze-off conditions can be deduced as follows. On the assumption that the cross-sectional area of flow passage at the contraction region is negligibly small compared to that of the pipe, $B^2 \ll 1$, equation (7) can be reduced as

$$Re_p/(\theta\sqrt{\xi+1}) = -\frac{2\lambda^*}{0.0045}\frac{1}{B\ln B}\sqrt{(n+1)/(\xi+1)+B^4}. \tag{9}$$

Introducing $d\theta/dB = 0$ leads to

$$\frac{n+1}{\xi+1} = -\frac{(1-\ln B)B^4}{1+\ln B}. \tag{10}$$

From equations (9) and (10), it should be noted that the effect of ξ on the freeze-off conditions is represented approximately in terms of $\xi+1$. In Fig. 3 the predicted result is compared with the experimental data for a wide range of friction-

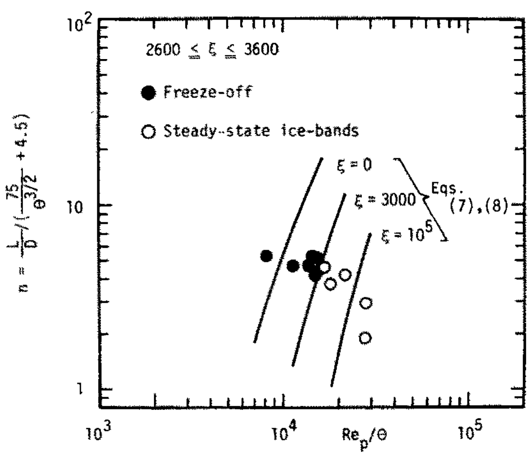

FIG. 2. Effect of friction-factor coefficient on the freeze-off conditions.

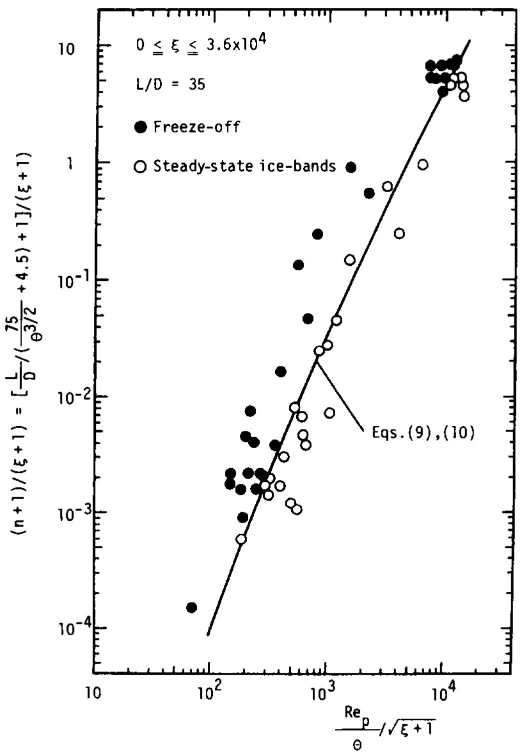

FIG. 3. A correlation of the freeze-off conditions for a wide range of ξ.

factor coefficient, $0 \leqslant \xi \leqslant 3.6 \times 10^4$. The upper region of the predicted line shows a freeze-off regime and the lower, a steady-state ice-band. Taking into account that the morphology of the ice layer under which the experimental conditions are close to the onset of freeze-off is often affected by small disturbances in temperature and flow conditions, it can be said that equations (9) and (10) give good predictions for the effects of the friction-factor on the freeze-off conditions.

CONCLUDING REMARKS

The onset of freeze-off was examined for the case that the friction-factor coefficient, ξ, in water-flow pipe systems was dominant. It has been ascertained both experimentally and analytically that the freeze-off easily occurs for a larger value of ξ and the effect of ξ on the freeze-off conditions can be given by equations (9) and (10).

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Boundary-layer treatment of film condensation in the presence of a solid matrix

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1. INTRODUCTION

LAMINAR, steady-state film condensation and boiling, along a plane surface submerged in a porous medium, have been studied analytically [1, 2] assuming that capillarity and non-Darcian effects are not significant.

The results, based on these assumptions, give the Nusselt number, condensate flow rate, and film thickness as a function of the subcooling (or superheating) parameter. The Prandtl number, which is introduced through the comparison between the thermal and momentum boundary-layer thickness, is not present because of the uniform film velocity distribution resulting from the application of Darcy's law. The capillary pressure, which is proportional to $\sigma(K/\varepsilon)^{-1/2}$ and depends on the saturation, can become significant at low permeabilities.

As the permeability increases, the inertia and boundary effects become important and for very high permeabilities the results based on no rigid matrix present [3, 4] must hold. However, since the film thickness decreases with an increase in permeability, then for the boundary-layer treatment to be valid, the small length scale associated with the microstructure of the rigid matrix must be much smaller than the film thickness. If this condition is satisfied, then the non-Darcy regime can be examined and, also, the parameters indicative of transition to the Darcian regime can be determined.

In this study the boundary layer and similarity treatment of film condensation in the absence of any solid matrix [3, 4] are extended to include the first-order resistance resulting from the presence of a solid matrix. This is done by applying an expansion method [5] (up to a third order) which has previously been used for treating natural convection in porous media [6]. The findings are then compared to those based on application of Darcy's law [1, 2].