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FLOW PATTERNS DURING CONDENSATION OF PURE REFRIGERANTS ON ENHANCED TUBES UNDER HIGH VAPOUR VELOCITY

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ABSTRACT

This work experimentally investigates the effect of vapour velocity during condensation of a refrigerant on the outside of a single horizontal enhanced tube with particular reference to the fluid-dynamic mechanisms induced by vapour shear stress, surface tension and gravity on the condensate flow.

More than 70 experimental runs were done in a transparent test section in order to directly observe the condensation of vapour, to take photographs and video-camera film, with two different extended tubes (an integral-fin tube with 1333 fins per meter and a commercial pin-fin type Hitachi Thermoexcel tube) in a wide operative range. These direct observations were implemented with the heat transfer coefficients previously measured by the authors with the same test tubes under the same operative conditions.

NOMENCLATURE

d	= Tube diameter
g	= Gravitational acceleration
h_{LG}	= Latent heat of vaporization
k	= Nondimensional group = $[(\Gamma/2)\,(g/\rho_L)^{1/4}]\,/\,\sigma^{3/4}$
$\dot{Q}$	= Heat flow rate
Re_f	= Film Reynolds number = $2\Gamma\,/\,\mu_L$
Re_V	= Vapour Reynolds number = $(\rho_V\, u_{max} d_O)\,/\,\mu_V$
s	= Fin spacing
u	= Vapour velocity
ϕ_f	= Flooding angle
Γ	= Condensate flow rate per unit length of the tube
σ	= Surface tension
ρ	= Density
λ	= Taylor wavelength
μ	= Dynamic viscosity
θ	= Fin half tip angle

Subscripts:

L	= condensate
max	= maximum value
O	= outside of finned tube at the fin tip
V	= vapour

INTRODUCTION

In film-wise condensation of a pure vapour, the heat transfer coefficients depend on the local thickness and conformation of the liquid film and on the type of condensate flow (laminar or turbulent). This work presents the results of an experimental investigation aimed at observing the condensate flow patterns obtained varying the vapour and condensate flow rates, during condensation of refrigerant 113 on the outside of a single enhanced tube.

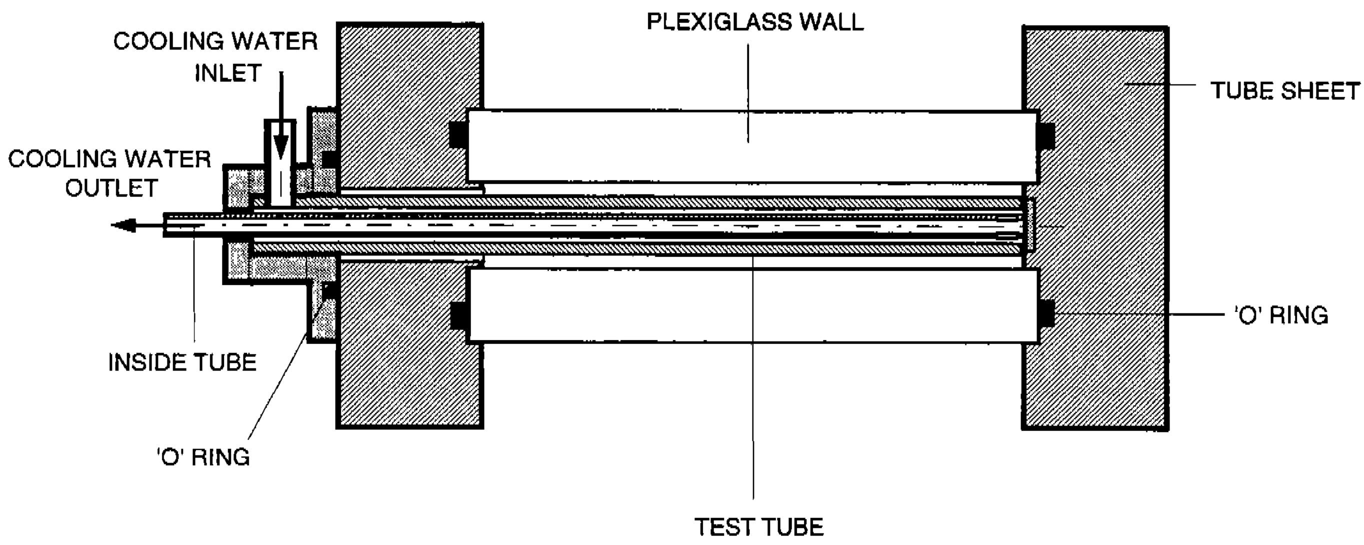


Fig.1 Experimental test section with transparent walls.

EXPERIMENTAL SETUP AND PROCEDURE

The experimental apparatus, fully described in previous papers [1, 2, 3], consists of a refrigerant loop in which dry saturated vapour, supplied by an electric boiler, goes through the test section partially condensing on a single horizontal extended tube cooled by water kept at constant inlet temperature.

The test section, sketched in Figure 1, has two transparent walls which allow to directly observe the condensation of the vapour and to take photographs and video-camera films.

The refrigerant vapour condenses on the outside enhanced surface of the test tube, while the cooling water flows inside the tube in a bayonet type configuration.

Two different extended tubes have been tested: the first one is an integral-fin tube with 1333 constant section fins per meter (fpm) and the second one is the commercial pin-fin type Hitachi Thermoexcel tube. Table 1 gives their main geometrical characteristics.

During each experimental run the condensate flow was filmed and photographed and, at the same time, the main experimental parameters such as refrigerant pressure and flow rate, cooling water flow rate, cooling water inlet temperature and temperature gain across the tube were measured, according to the procedures already described in [1, 2, 3]. These direct observations were implemented with measures of the heat transfer coefficients previously taken by the authors [1, 2, 3] with the same test tubes, instrumented with wall thermocouples, under the same operative conditions in a different test section with the same flow area as the present one, but without the possibility of visualization of the condensate flow pattern.

TABLE 1. Geometrical characteristics of the test tubes.

Test tube	Thermoexcel	1333 fpm
Fin type	Three-dimensional	Integral-fin
Tube material	Copper	Copper
Tube length [mm]	150.00	150.00
Outside diameter at the fin tip d_O [mm]	18.70	16.40
Outside diameter at the fin root [mm]	14.70	15.00
Fin pitch [mm]	1.25	0.75
Fin height [mm]	2.00	0.70
Effective/Nominal surface area	2.82	2.40

TABLE 2. Operative conditions during experimental runs.		
Test tube	Thermoexcel	1333 fpm
Number of experimental runs	30	42
Fluid	R113	R113
Vapour inlet pressure [kPa]	115-118	75-119
Maximum vapour velocity [m/s]	2.0-38.7	1.3-28.3
Re_f	120-224	37-254
$Re_V \cdot 10^{-5}$	0.29-5.50	0.16-3.56

The two-phase flow considered was characterized by two nondimensional groups: the vapour Reynolds number

$$Re_V = (\rho_V\, u_{max} d_O) / \mu_V \tag{1}$$

which refers to the maximum vapour velocity in the test section and to the vapour phase properties, and the film Reynolds number:

$$Re_f = 2\Gamma / \mu_L \tag{2}$$

where $\Gamma = \dot{Q}/(h_{LG}L)$ is the condensate flow rate per unit length of the tube and $\dot{Q}$ is the condensation heat flow rate exchanged in the tube derived from a thermal balance on the cooling water.
Table 2 reports the operative conditions of the experimental runs.

RESULTS AND DISCUSSION

As shown in [3], vapour velocity effects on condensation heat transfer appear to be different for each geometry tested, both in behaviour and magnitude. Vapour shear effects for the 1333 fpm integral-fin tube appear only at vapour Reynolds number Re_V higher than 100,000 with a 50% maximum heat transfer increase at the maximum Re_V tested (350,000), while they affect the commercial Hitachi tube in the whole operative range giving a maximum enhancement of 40% at the maximum Re_V (550,000).
The visual observations in the transparent test section show that the above heat transfer enhancement is due to changes in condensate flow patterns.
Figure 2 shows still photographs of the condensate flow patterns for condensation of refrigerant 113 on the 1333 fpm tube at near stationary vapour conditions for different condensate flow rates. The lower portion of the tube is completely flooded by the condensate, and the measured value of the flooding angle ϕ_f (i.e. the angle from the top of the tube below which the grooves between the fins are completely filled by the liquid) is 10-15% less than that calculated by the model of Honda et al. [4], that is

$$\phi_f = \cos^{-1} [(4\, \sigma \cos\theta) / (\rho_L g\, s\, d_O) - 1] \tag{3}$$

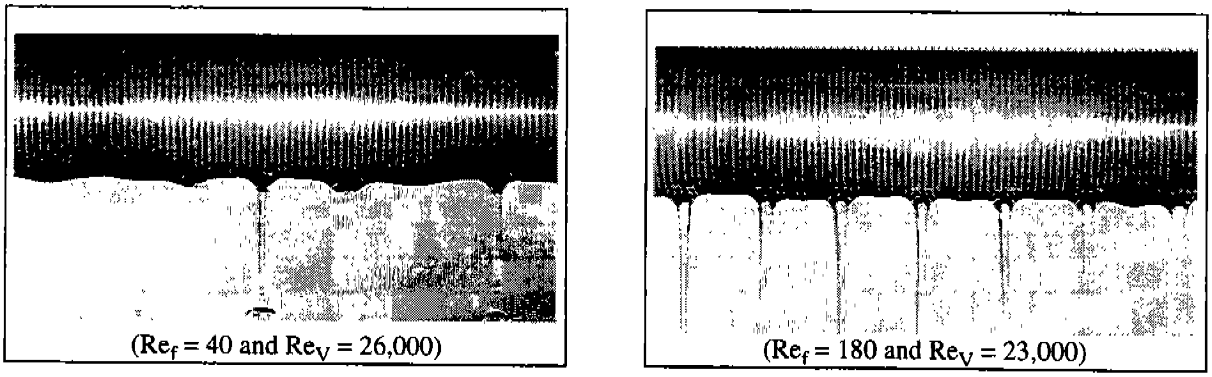
At very low condensation rates the liquid condensate leaves the bottom of the tube forming droplets, while when the condensate flow increases, liquid columns, almost equally spaced along the tube length, appear. The distance between two adjacent columns, around 8.5 mm, is very well predicted by the most unstable Taylor wavelength λ for a thin film [4], given by:

$$\lambda = 2\pi \sqrt{2\sigma / (\rho_L g)} \tag{4}$$

The transition between droplet and column flow patterns is controlled by the nondimensional group

$$k = [(\Gamma/2)\, (g/\rho_L)^{1/4}] / \sigma^{3/4} \tag{5}$$

and appears at around 0.1 in accordance with Honda et al. [5].

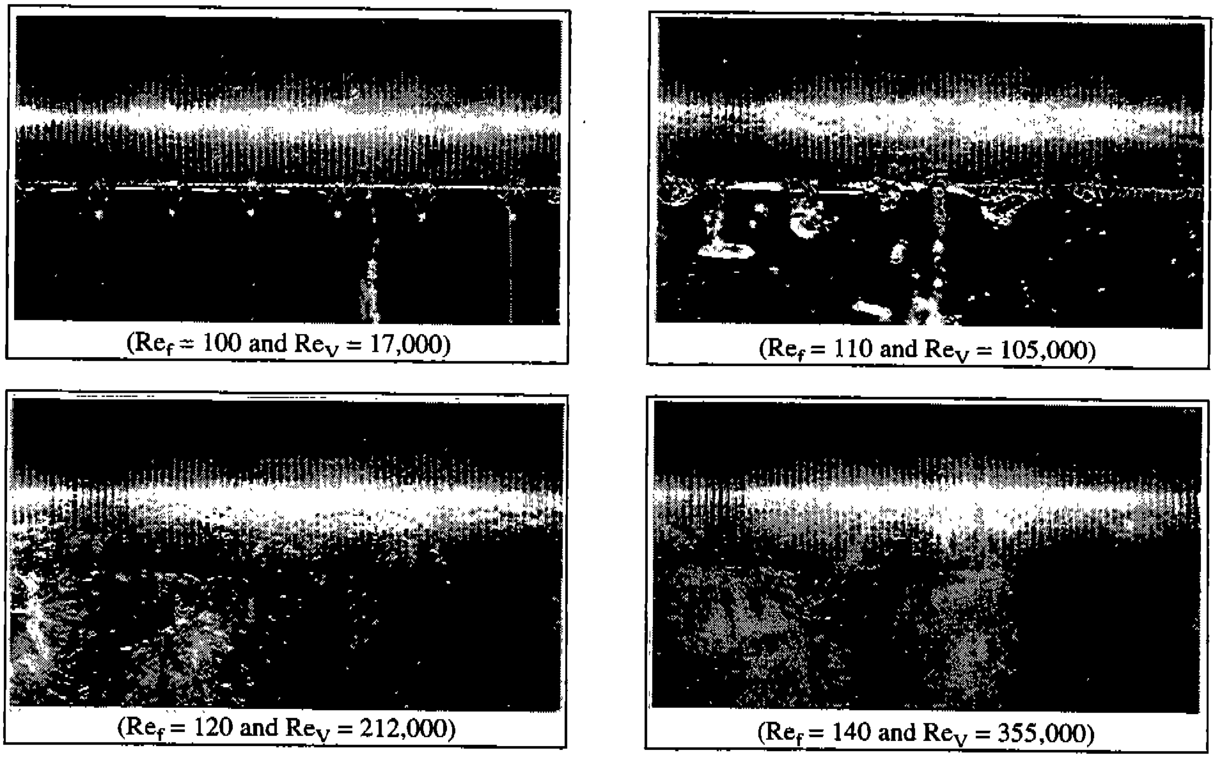


($Re_f = 40$ and $Re_V = 26{,}000$) ($Re_f = 180$ and $Re_V = 23{,}000$)

Fig. 2. Condensate flow patterns for condensation of R113 on a 1333 fpm integral-fin tube for different condensate rates.

Figure 3 shows still photographs of the condensate flow patterns for condensation of refrigerant 113 on the 1333 fpm tube at increasing vapour velocities. Starting from stationary conditions, when vapour velocity increases, the value of the flooding angle ϕ_f seems to slightly decrease due to vapour boundary layer separation, but the border between the flooded and unflooded portion of the tube is not a straight line, as with stagnant vapour, and becomes more and more wave-like.

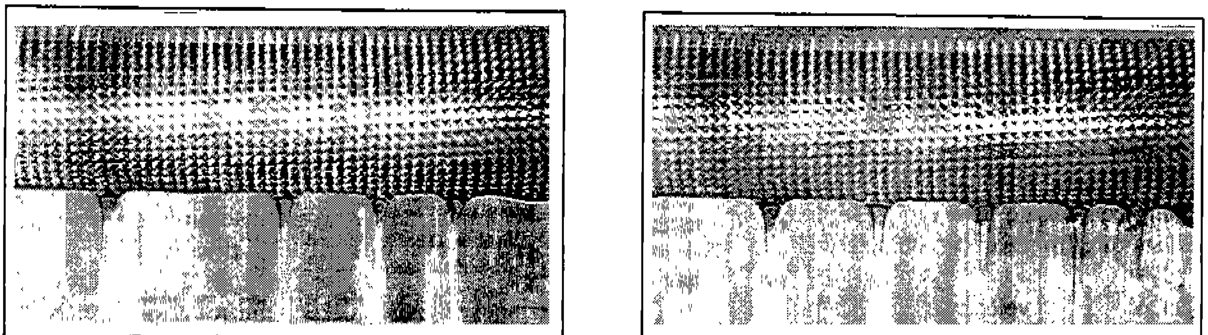
At vapour Reynolds number Re_V slightly greater than 100,000 (the value at which vapour velocity effects on the heat transfer rate appear) the liquid columns become less stable, their sites move along the tube and the distance between them is no longer constant: these phenomena produce local reduction of the liquid film thickness in the flooded region. When Re_V reaches approximately 200,000 the vapour shear induces waves at the gas-liquid interface in the flooded region, whereas the interaction between vapour and falling liquid columns produces a dispersed flow below the tube. At a higher vapour velocity the entrainment of condensate droplets in the vapour flow appears just below the flooding angle, reducing the thickness and generating turbulence in the liquid film in the bottom part of the tube. At the maximum vapour Reynolds number tested, $Re_V = 350{,}000$, a fine dispersion of droplets in the vapour flow is entrained mainly at the limit of the flooded region, where the separation of the vapour boundary layer occurs.



($Re_f = 100$ and $Re_V = 17{,}000$) ($Re_f = 110$ and $Re_V = 105{,}000$)

($Re_f = 120$ and $Re_V = 212{,}000$) ($Re_f = 140$ and $Re_V = 355{,}000$)

Fig. 3. Condensate flow patterns for condensation of R113 on a 1333 fpm integral-fin tube for different vapour velocities.



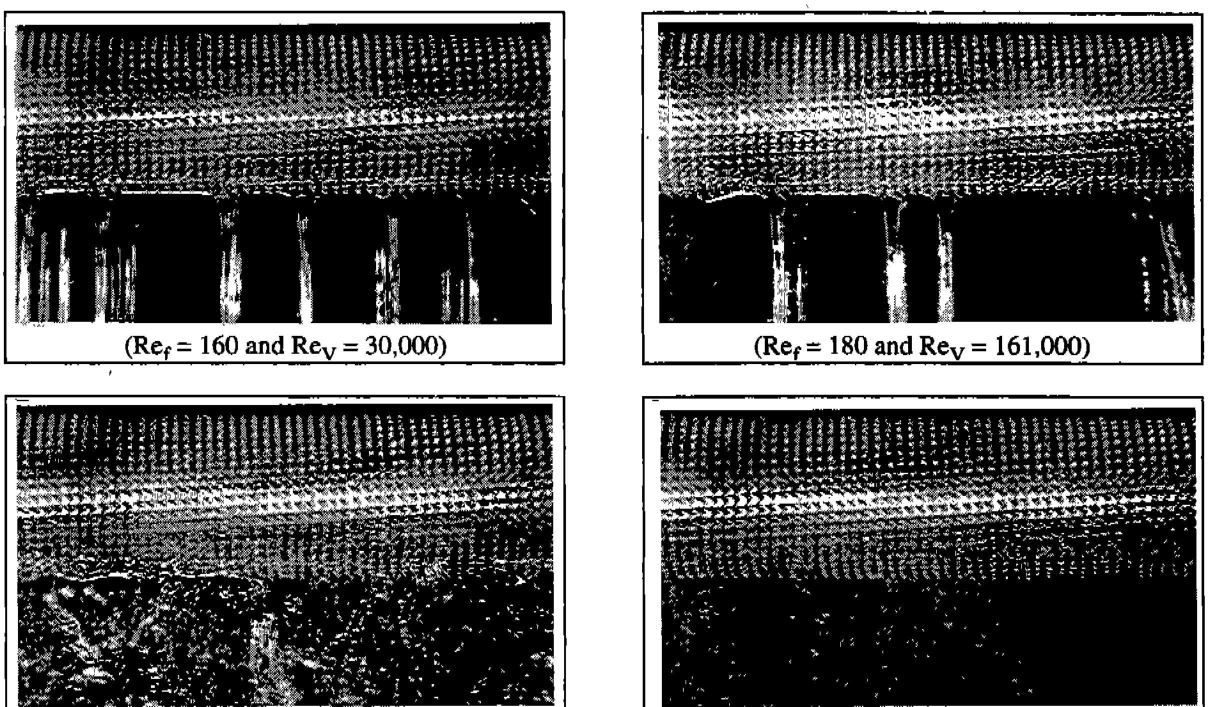
(Re$_f$ = 120 and Re$_V$ = 29,000) (Re$_f$ = 180 and Re$_V$ = 32,000)

Fig. 4. Condensate flow patterns for condensation of R113 on the Thermoexcel tube for different condensate rates.

Figure 4 and 5 show still photographs of the condensate flow patterns for condensation of refrigerant 113 on the Thermoexcel tube at different condensate rates and different vapour velocities, respectively.

At near stationary vapour conditions, the Thermoexcel tube gives a heat transfer rate which is 30% higher than that of the 1333 fpm integral-fin tube, so the condensate rate explored during experimental runs covers only the region of column flow pattern ($0.1 < k < 0.2$) and the liquid columns do not result equally spaced as happens with the integral-fin tube tested. This three-dimensional pin-finned surface is not affected by condensate retention in the bottom part of the tube and this contributes to explain its higher performance at stationary conditions with respect to integral-fin tubes.

The column flow results stable also for high vapour velocities (up to Re$_V$ = 160,000), while only at vapour Reynolds numbers higher than 350,000 the transition to dispersed flow is completed. These observations justify the lower effect of vapour velocity on the Thermoexcel heat transfer performance and the absence of singularities in the regular enhancement trend.



(Re$_f$ = 160 and Re$_V$ = 30,000) (Re$_f$ = 180 and Re$_V$ = 161,000)

(Re$_f$ = 190 and Re$_V$ = 328,000) (Re$_f$ = 210 and Re$_V$ = 543,000)

Fig. 5. Condensate flow patterns for condensation of R113 on the Thermoexcel tube for different vapour velocities.

CONCLUSIONS

The effects of vapour shear stress and operative conditions during condensation on two different enhanced tubes were experimentally investigated. The outside heat transfer enhancement with vapour velocity was correlated to characteristic changes in the condensate flow patterns.

During almost stagnant vapour condensation on the 1333 fpm tube, a transition between droplet and column flows occurred at increasing condensate flow rates. On the other hand, increasing the vapour velocity, a transition to dispersed flow was observed, starting from Re_V = 100,000 up to Re_V = 200,000. The flooding level was only slightly affected by vapour shear stress.

The effect of vapour velocity on the condensate flow for the Thermoexcel tube is less significant, producing a gradual transition from column mode to dispersed flow in the range of vapour Reynolds number from 160,000 to 350,000.

The above observations are in good agreement with Honda et al. [6, 7] investigations on condensate flow patterns in a bundle of finned tubes (integral-fin and three-dimensional). This work investigates a wider range of vapour velocities.

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