1992

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PURE REFRIGERANT CONDENSATION ON A SINGLE INTEGRAL FINNED TUBE: VAPOUR VELOCITY EFFECTS

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ABSTRACT

The experimental set-up used to measure heat transfer coefficients during pure vapour condensation of halogenated refrigerants on the outside of a single horizontal finned tube is described.

Two different series of experimental runs are fully reported. The first series refers to refrigerant 11 with vapour pressure ranging from 109 to 198 kPa, condensation temperature difference varying from 4.4 to 11.7 °C and maximum vapour velocity ranging from 1.9 to 26.2 m/s. The second series is relative to refrigerant 113 condensing around 110 kPa with average vapour to wall temperature difference from 5.2 to 16.8 °C and maximum vapour velocity from 2.6 to 29.5 m/s.

The experimental data obtained is compared with the most recent theoretical models for condensation on a single integral finned tube. The effects of vapour shear stress on condensation heat transfer are particularly investigated and the results obtained are presented.

NOMENCLATURE

c_p	=	Specific heat capacity (J/(kg K))
d	=	Tube diameter (m)
E1	=	Mean percentage deviation
E2	=	Absolute mean percentage deviation
h	=	Fin height (m)
L	=	Tube length (m)
$\dot{m}$	=	Mass flow rate (kg/s)
Np	=	Number of experimental runs
Nu	=	$\alpha d_O/\lambda_L$ = Nusselt number
p	=	Fin pitch (m)
Q	=	Heat flow rate (W)
Re_V	=	$\rho_V u_{MAX} d_O/\mu_V$ = Reynolds number
S	=	Heat transfer surface (m^2)
t	=	Fin thickness (m)
T	=	Temperature (K)
u	=	Vapour velocity (m/s)
α	=	Film heat transfer coefficient ($W/(m^2\ K)$)
Δ	=	Difference
ρ	=	Density (kg/m^3)
λ	=	Thermal conductivity (W/(m K))
μ	=	Dynamic viscosity (kg/(m s))

Subscripts

C	=	coolant side
cal	=	calculated
exp	=	experimental
ln	=	logarithmic
L	=	condensate
MAX	=	maximum value
O	=	outside of finned tube at the fin tip
S	=	saturated vapour
st	=	stationary vapour
W	=	tube wall

INTRODUCTION

Integral finned tubes are very commonly used in refrigeration, air conditioning and chemical industries to improve the efficiency of shell and tube condensers.

Many experimental and theoretical investigations have analysed the effects of fin geometry, tube material and condensing fluid properties on the condensation heat transfer on integral finned tubes, but only few studies, /1,2,3/, have considered the effect of vapour shear stress at the vapour-liquid interface.

This paper describes an experimental apparatus for measuring heat transfer coefficients during condensation of pure refrigerants, R11 or R113, on a single horizontal, integral finned tube under high vapour velocity. The experimental results are analysed to correlate heat transfer enhancement with vapour velocities.

EXPERIMENTAL SET-UP

The experimental apparatus illustrated in figure 1 consists of a pump circulated loop of pure refrigerant and a forced circulation loop of coolant.

In the primary loop saturated vapour of a pure refrigerant, 11 or 113, flows downwards through a test section consisting of ten dummy tubes and one measuring tube (see figure 2). The test tube, which is made of copper and integral finned, presents the following geometry: length 150 mm, diameter at the fin tip 16.4 mm, internal diameter 10 mm, trapezoidal cross section fins with pitch 0.75 mm, height 0.70 mm, average thickness 0.22 mm, semi-vertex angle 10 deg, ratio between effective surface and nominal envelope surface area (nominal surface area of a smooth tube having the diameter of the finned tube at the fin tip) equal to 2.43. This tube, which is cooled by water, is equipped to measure the surface temperature with eight copper-constantan thermocouples, 0.128 mm in diameter and teflon-coated, inserted and soldered into four equidistant axial holes, each 1 mm in diameter and 27 mm deep, drilled on both sides (inlet and outlet) of the tube wall. A twisted-tape insert is placed inside the tube to mix the flowing water and generate turbulence. The uncooled dummy tubes are smooth tubes partially housed inside the test section wall with the same length and outside diameter of the instrumented tube. The walls of the test section are made of PVC of 40 mm thickness and insulated to prevent heat loss. The pure refrigerant, supplied by an electric boiler with 40 kW maximum power, passes through a wire-mesh demister, a centrifugal liquid separator, a perforated plate and a calming section, before reaching the instrumented tube, to ensure that dry-saturated vapour will distribute uniformely in the test section. Two manometric connections to a mercury U tube manometer and to a strain gage pressure transducer are placed on the test section, above and below the instrumented tube, to determine the pressure profile of the condensing vapour. Both manometers are heated to avoid vapour condensation. The two pressure measurements differ within ± 200 Pa. Atmospheric pressure is measured by means of a Fortin barometer. The refrigerant at the outlet of the test section is completely condensed in an auxilary condenser and then sent by a pump to the boiler, passing through one of the three microturbine flow meters which measures its mass flow rate within ± 1%. The total refrigerant mass flow rate through the test section is also calculated by evaluating heat balance at the boiler. These two different measures always differ by less than ± 3%.

In the forced convection loop the cooling water is kept at constant temperature (ranging from 5°C to 30°C) in an insulated tank by means of a chiller and by electric heating elements controlled by a proportional regulator. The coolant is pumped from the tank to the test tube and then back to the tank, passing through a volumetric flowmeter and a receiver. Its flow rate is directly measured by the flowmeter and also derived from the weight of the water collected in a fixed time. The two measurements differ within ± 0.5%. The volumetric meter is expected to give a maximum error of ± 0.2% of the measured flow rate. The water temperature gain across the instrumented tube is measured by means of a differential four-junction copper-constantan thermopile, while the perfect mixing of the water is obtained in two mixing chambers placed in series at the outlet of the pipe. The accuracy of the heat flow measurement chain (thermopiles-mixing chambers-flowmeter) has been tested by a specific device. A smooth tube, similar to the test one, heated by an electric resistance and thermally insulated has been inserted in the coolant loop to substitute the test section. The heat flow rate, derived from the balance over the cooling water, agreed with the input electric power within 0.5%.

HEAT TRANSFER COEFFICIENTS

Before starting the experimental runs some preliminary tests were performed to evaluate the heat loss through the test section and to check the thermocouple readings. The test section was fed by refrigerant vapour at constant pressure while the instrumented tube was not cooled. The difference between the vapour saturation temperature derived from the pressure reading and the tube wall temperature measured by the termocouples ranged from 0.03 to 0.09 K for refrigerant 11 and around 0.2 K for refrigerant 113. The greater discrepancy found for the R113 tests was due to a reduction of the thermocouples' reference-point accuracy which declined from 0.02 K to 0.1 K.

The experimental runs include heat transfer measurement during refrigerant 11 and refrigerant 113 pure vapour condensation on the single instrumented, cooled, integral finned tube. The heat flow rate Q exchanged in the tube is calculated from the heat balance over the cooling water:

$$Q = \dot{m}_C c_{pC} \Delta T_C \tag{1}$$

where ΔT_C is the water temperature gain measured by the thermopile, $\dot{m}_C$ is the cooling water flow rate and c_{pC} is the specific heat capacity of the cooling water. The average condensation heat transfer coefficient refers to the fin envelope surface area S and to the logarithmic average temperature difference ΔT_{ln} between the vapour and the tube wall:

$$\alpha = Q / (S \, \Delta T_{ln}) \tag{2}$$

with $S = \pi \, d_O \, L$ and

$$\Delta T_{ln} = (T_{WO}-T_{WI})/\ln[(T_S-T_{WI})/(T_S-T_{WO})] \tag{3}$$

where L and d_O are respectively the tube length and the tube's outside diameter at the fin tip, T_S is the average vapour saturation temperature derived from the pressure profile, T_{WI} and T_{WO} the average surface temperatures evaluated by the thermocouples placed in two distinct sections, at 27 mm from the inlet and from the outlet of the tube, respectively. To improve measurement accuracy the pressure profile in the test section is measured by two distinct devices while the coolant flow rate and the refrigerant flow rate are determined by two different procedures as explained above. The operating conditions relative to the data obtained with refrigerant 11 and refrigerant 113 are reported in table I. The vapour velocity is calculated using the refrigerant mass flow rate measurements, and refers to the minimum cross-section area.

Table I. Operative conditions of experimental runs

Fluid	R-11	R-113
Number of runs	34	41
Vapour inlet pressure (kPa)	109-198	104-125
Mean temperature difference (°C)	4.4-11.7	5.2-16.8
Maximum vapour velocity (m/s)	1.9-26.2	2.6-29.5
Heat flux density (kW/m^2)	53.8-117.2	64.3-147.2
Maximum mass flux ($kg/(m^2s)$)	20-203	23.4-236.1
$Re_V \cdot 10^{-5}$	0.2-2.9	0.3-3.5

CORRELATIONS AND EXPERIMENTAL RESULTS

Several theoretical models which refer to pure vapour condensation on a single horizontal, integral finned tube have been developed as from 1948, when Beatty and Katz /4/ proposed their correlation based on a Nusselt-type analysis. This approach, which considers only the effect of gravity in the drainage of the condensate, results inadequate for condensation of high surface tension fluids on high fin density tubes /5/. Surface tension affects condensation heat transfer by reducing the condensate film thickness on the fin flanks and causing a retention of condensate in the lower part of the tube. At present two models, the Honda et Al. 1987 model /6,7/ and the Adamek and Webb 1990 model /8/ give a better prediction for the condensation heat transfer coefficient of halogenated refrigerants on a single horizontal finned tube. Both of them consider the combined effect of surface tension and gravity as well as the effects of fin efficiency and of conduction in the tube wall but not the effect of vapour shear stress at the interface. Figures 3, 4 and 5 report the present experimental data plotted on the coordinates of Nu_{exp}/Nu_{cal} vs. Re_V, where Nu_{exp} is the mentioned experimental Nusselt number, Nu_{cal} is the Nusselt number evaluated by the above theoretical models and Re_V is Reynolds number relative to maximum vapour velocity, to the properties of the vapour phase and to the diameter of the finned tube at the fin tip. Table II gives the mean percentage deviation E1 and the absolute mean percentage deviation E2, defined as follows:

$$E1=(100/Np)\sum_{i=1}^{Np}(1-\alpha_{cal}/\alpha_{exp}) \qquad E2=(100/Np)\sum_{i=1}^{Np}|1-\alpha_{cal}/\alpha_{exp}| \tag{4}$$

This comparison shows that the Honda et Al. 1987 model /6,7/ reproduces the present data in the low vapour velocity range very well, while it is not satisfactory at higher vapour velocities. Up to Re_V = 100000 it overpredicts present work experimental data by about 7%, whereas at the highest vapour velocities (i.e. Re_V = 350000) it underestimates experimental data by approximately 50%. The Beatty and Katz 1948 model /4/ always underestimates experimental data and particularly those at higher vapour velocities, while the Adamek and Webb 1990 model /8/ generally overpredicts experimental results.

Table II Comparison between theoretical models and present experimental data

Model	R11		R113	
	E1	E2	E1	E2
Beatty and Katz 1948 /4/	31.94	31.94	37.97	37.97
Honda et Al. 1987 /6,7/	4.08	11.74	12.78	14.97
Adamek and Webb 1990 /8/	-13.38	16.38	-1.69	14.97

Vapour velocity effects on condensation heat transfer are shown in figure 4 which reports the present experimental data plotted on the coordinates Nu_{exp}/Nu_{st} vs. Re_V, where Nu_{st} is the experimental Nusselt number at low vapour velocity, with $u_{MAX} = 2$ m/s and Re_V around 30000. For comparison the trend for condensation on a smooth tube, where the Nusselt number under vapour velocity is computed by Honda et Al. 1986 /9/ model is also reported, whereas the Nusselt number for stationary vapour is derived from the classical Nusselt equation /10/. This plot shows that the condensation heat transfer enhancement for finned tubes due to vapour velocity is less significant than for a smooth tube but it is still remarkable. This effect appears at Reynolds number Re_V higher than 100000 and produces an increase of around 50%, with respect to stationary vapour, for Rev around 350000. A statistical analysis of present work experimental data for $Re_V > 100000$ shows a dependence of Nu/Nu_{st} on $Re_V^{0.325}$.

Figure 5 reports the present film heat transfer coefficients α vs. condensation temperature difference ΔT for condensation of refrigerant R113 at two different maximum vapour velocities, 24.5 m/s and 10.5 m/s. A statistical analysis of all the present data, both at high and low vapour velocity, shows the dependence of α on $\Delta T^{-0.25}$, as suggested by Nusselt for condensation on a smooth tube with stationary vapour. In particular for R-11 data the exponent is -0.28, while for R113 data it is -0.23.

CONCLUSIONS

Heat transfer enhancement due to vapour shear stress at the interface during condensation on a single finned tube results less significant than for condensation on a single smooth tube, however it is quite considerable. For present experimental conditions this effect appears at Reynolds number Re_V around 100000 and the enhancement ratio Nu/Nu_{st} shows a dependence on $Re_V^{0.325}$.

At lower vapour velocity, $Re_V < 100000$, the Honda et Al. 1987 model /6,7/ reproduces the present data with a mean percentage deviation around - 8% and it results, at present, as the best prediction model for condensation heat transfer on a single horizontal integral-finned tube of stationary pure vapour.

Further experimental work is being planned in order to evaluate the combined effect of vapour velocity and condensate inundation during condensation of pure vapour on a bundle of finned tubes.

ACKNOWLEDGMENTS

Research carried out with partial financial aid of C.N.R., contract 86.02563.07.

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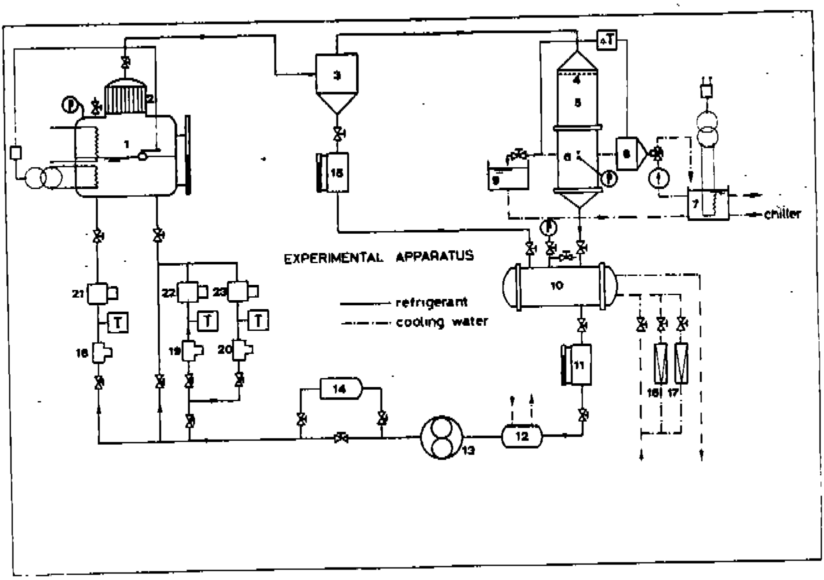

1 Boiler, 2 Wire-mesh, 3 Centrifugal separator, 4 Perforated plate, 5 Calming section, 6 Test section, 7 Cooling water, 8,9 Cooling water distributor and receiver, 10 Auxiliary condenser, 11,15 Liquid receiver, 12 Cooler, 13 Rotary pump, 14 Drier, 16,17 Water rotameter, 18,19,20 Filter, 21,22,23 Microturbine flow meter.

Figure 1 Diagram of the experimental apparatus.

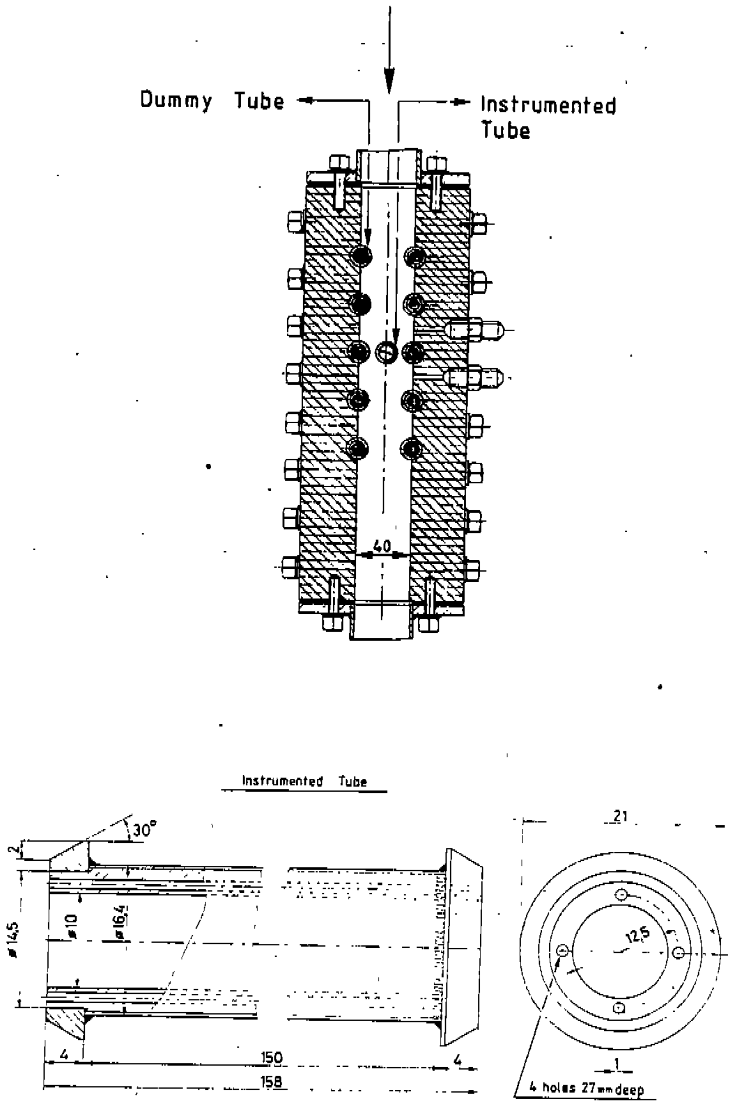

Figure 2 Diagram of the test section (measures in millimeters).

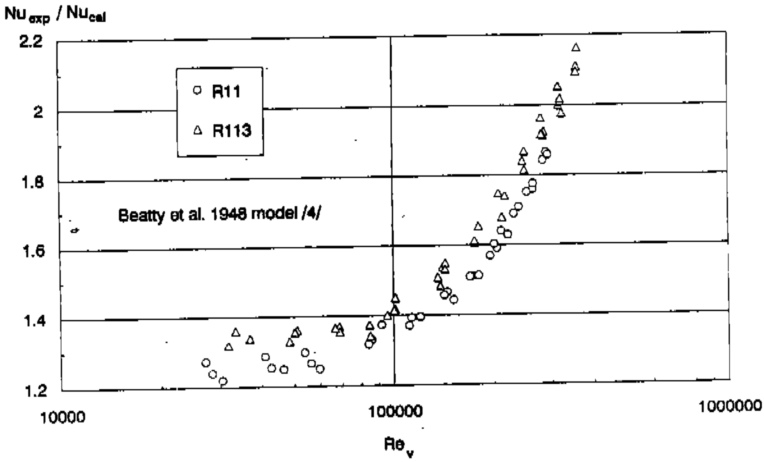

Figure 3 Present work experimental data plotted on the coordinates Nu_{exp}/Nu_{cal} vs. Re_V: comparison with Beatty and Katz 1948 model /4/.

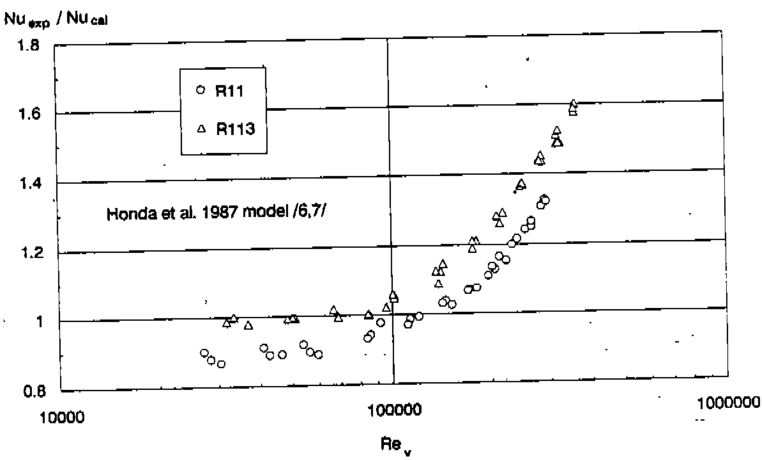

Figure 4 Present work experimental data plotted on the coordinates Nu_{exp}/Nu_{cal} vs. Re_V: comparison with Honda et Al. 1987 model /6,7/.

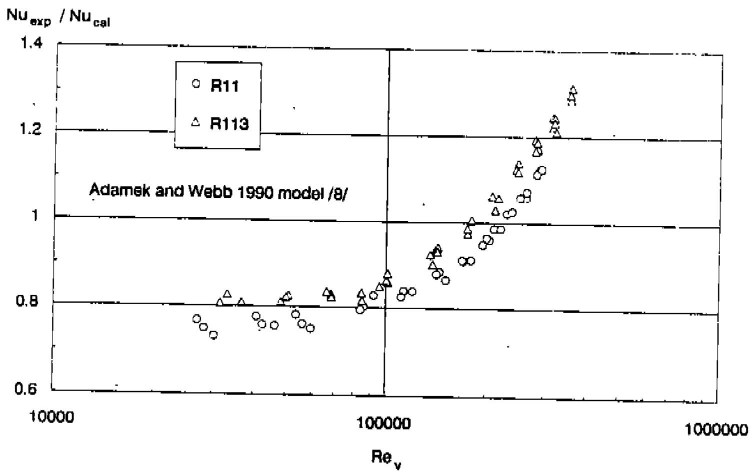

Figure 5 Present work experimental data plotted on the coordinates Nu_{exp}/Nu_{cal} vs. Re_v: comparison with Adamek and Webb 1990 model /8/.

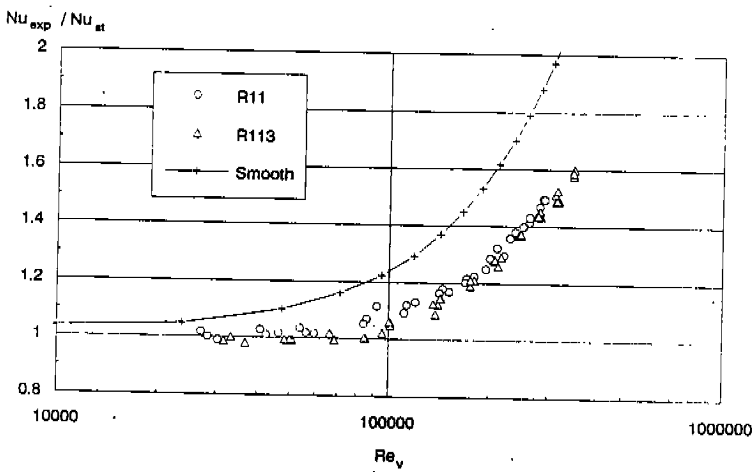

Figure 6 Present work experimental data plotted on the coordinates Nu_{exp}/Nu_{st} vs. Re_v. and compared with the smooth tube trend.

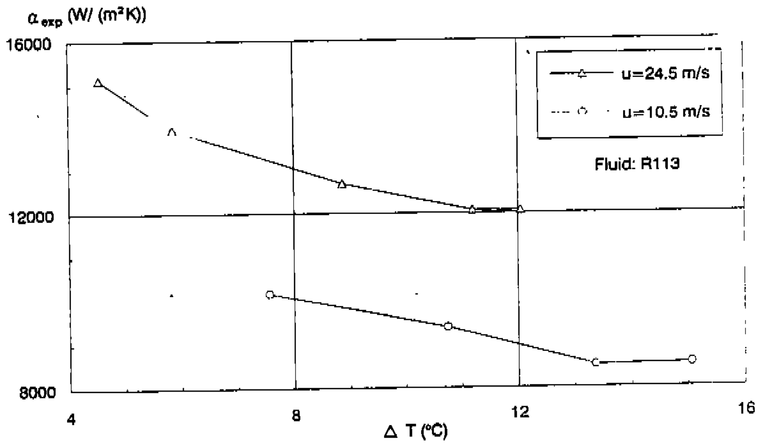

Figure 7 Present work experimental data plotted on the coordinates α vs. ΔT for condensation of refrigerant 113 under two different vapour velocities, 24.5 and 10.5 m/s.