Flow boiling of R32 in a horizontal smooth tube of 6.0 mm internal diameter: heat transfer coefficient and pressure drop

R MASTRULLO, A W MAURO and L VISCITO

Department of Industrial Engineering, Federico II University of Naples, P.le Tecchio 80, 80125, Naples (Italy)

E-mail: alfonswilliam.mauro@unina.it

Abstract. This paper presents a series of experiments on flow boiling heat transfer coefficient and pressure drop using refrigerant R32. Tests were performed in a smooth horizontal stainless steel tube with an internal diameter of 6.00 mm, with the heat directly applied on the test section tube through DC current. Trends of the heat transfer coefficient and pressure drop data are displayed for different operating conditions. Specifically, the saturation temperature was fixed to 25 and 35 °C, the heat flux varied from 5 to 50 kW/m² and the mass velocity was fixed to the low values of 150 and 250 kg/m² s. The effects of the operating parameters on the heat transfer results are also shown.

1. Introduction

The research on flow boiling of refrigerants is of primary importance in several fields, such as air conditioning, refrigeration, nuclear and cooling electronic systems. Therefore, the capability to determine the two-phase heat transfer and pressure drop behavior is important for the correct design of evaporators and heat spreaders systems. The European Union has already planned a gradual removal of high-GWP substances in the next years [1] and R32 is one of the candidates for the substitution of R410A in the air-conditioning sector, due to its relatively low GWP index of 675 when used as pure fluid. Moreover, the peculiar thermodynamic characteristics of R32 make its employment in domestic refrigerators and roof-top systems particularly appealing [2]-[3], due to a reduction of the fluid mass charge, a higher compactness of heat exchangers and an overall increased efficiency.

Different studies are accessible in open literature coping with flow boiling of refrigerants in macro and mini tubes of different geometries and configurations, but relatively few data are currently available for refrigerant R32 and most of them present flow boiling data for R32 included in HFC mixtures.

Shin et al. [4] studied convective boiling heat transfer of different pure refrigerants and refrigerants mixture, including R32, in a horizontal seamless stainless steel tube with an inner diameter of 7.7 mm. The saturation temperature was fixed to 12 °C, whereas the heat flux and mass flux varied from 10 to 30 kW/m² and from 424 to 742 kg/m² s, respectively. The authors found the heat transfer coefficient strongly dependent on heat flux in the low-quality region and almost constant at higher vapor qualities.

Li et al. [5] presented flow boiling data for R1234yf/R32 mixtures in a smooth horizontal tube with an inner diameter of 2 mm. The experiments were conducted for two mass concentrations (80/20 and 50/50), by varying the heat flux from 6 to 24 kW/m² and the mass flux from 100 to 400 kg/m² s. The
evaporation temperature was fixed to 15 °C. The authors found that the heat transfer coefficients of the mixtures were about 20-50% lower than those of pure R32.

Kondou et al. [6] studied flow boiling heat transfer coefficient and pressure drop of R1234ze/R32 mixtures in a microfin tube of 5.21 mm internal diameter at a saturation temperature of 10 °C. They varied the mass flux from 150 to 400 kg/m² s, for two different heat fluxes of 10 and 15 kW/m². In case of pure R32, the authors found a small influence of both vapor quality and mass velocity.

Azzolin et al. [7] presented flow boiling data of a zetropic mixture of R1234ze/R32 (50/50) in a single microchannel of 0.96 mm internal diameter. Tests were conducted at a saturation temperature of 26.2 °C, mass velocities from 300 to 600 kg/m² s and heat fluxes from 27 to 165 kW/m². The authors found a strong influence of the imposed heat flux above all other parameters, especially in the low-quality region.

Jige et al. [8] studied flow boiling of pure R32 in smooth tubes of different diameters (1.0, 2.2 and 3.5 mm), examining the influence of mass velocity (from 50 to 600 kg/m² s) and of heat flux (from 5 to 40 kW/m²) on the heat transfer coefficient and the pressure drop. They found that the heat transfer coefficient increased with reduced tube diameter and that the pressure drop was enhanced for higher mass velocities and vapor qualities and lower tube diameters.

This work presents two-phase flow boiling heat transfer coefficient and pressure drop data for refrigerant R32 in a smooth horizontal stainless steel (SS316) tube with an internal diameter of 6.00 mm. Heat transfer coefficient data are taken at both top and bottom surface of the heated tube. Different trends were shown at several operating conditions, by changing the saturation temperature (25 and 35 °C), heat flux (from 5 to 50 kW/m²) and mass velocity (150 and 250 kg/m² s), to focus the attention on the effect of the heat flux and of the asymmetric annular flow structure on the local heat transfer coefficients.

2. Experimental apparatus
The test facility has been assembled in the Refrigeration Laboratory, at the Industrial Engineering Department, Università degli Studi di Napoli Federico II. The experimental test rig consists of a main refrigerant loop, in which the test section inlet conditions in terms of saturation temperature, mass flux and vapor quality are set and monitored, and a secondary water close loop, whose temperature is controlled by a thermostatic bath. Both loops are schematized in Fig. 1.

2.1. Main and secondary loops description
The black line in Fig. 1 portrays the main refrigerant loop, that is made up of a magnetic gear pump, a preheater, a diabatic test section, a brazed plate condenser, a tube-in-tube sub-cooler, a liquid receiver and a micrometric throttling valve.

The sub-cooled refrigerant passes through the magnetic gear pump, whose velocity can be set from 1650 to 3400 rpm by means of an inverter coupled with the electrical motor, able to provide volumetric flow rates within a range of 1.3-2.5 dm³/min. A throttling valve on the liquid line is employed during the experiments to adjust the system pressure and the mass flow rate to the desired values. The liquid then enters the preheater section, in which the heat is supplied by two fiberglass heating tapes (each of them with a nominal heat power of approximately 900 W at 240 V and 25 °C, as indicated by the manufacturer). The fluid then passes through an adiabatic, smooth, horizontal part, whose length is able to obtain a fully developed flow at the inlet of the tube section. A diabatic test section of 192.5 ±0.47 mm allows the heat transfer coefficient and pressure drop measurements. Another micrometric throttling valve at the test section outlet is used for the single-phase tests to adjust the system pressure and mass flow rate. The liquid/vapor refrigerant mixture is condensed with a plate heat exchanger and then flows into a liquid receiver. The working fluid is then sub-cooled by means of a double pipe heat exchanger before the pump suction head which closes the loop. A by-pass circuit that recirculates part of the refrigerant flow into the liquid receiver allows the investigation of very low mass fluxes.
The secondary fluid in the heat exchangers is demineralized water, whose temperature is controlled by a thermostatic bath. Its circuit is displayed with a blue line in Fig. 1. Both the tube-in-tube and the brazed plate heat exchangers may be excluded from the water flow by means of two ball-cock valves. The secondary fluid specific volume variations are restricted with the use of an expansion vessel.

**Figure 1.** Schematic view of the experimental plan

2.2. Test section

The test section employed in this work is a smooth, horizontal, circular stainless steel (type 316) tube with an internal diameter of 6.00 ±0.05 mm and an outer diameter of 8.00 ±0.05 mm. Fig. 2 exposes a picture and a schematic view of the test section with its geometric characteristics. The heat is applied to the fluid by Joule effect, by means of a DC power supply unit and two copper electrodes welded on the external tube surface (see points A and F in Fig. 2), placed at a distance of 192.5 ±0.47 mm one another. At approximately the same distance, two ports are employed for the pressure drop measurements. Two different measurement positions are provided along the tube, which are respectively 77.2 ±0.41 mm and 154.5 ±0.61 mm far from the diabatic inlet section (see C and D in Fig. 2). At each point, two T-type thermocouples are placed on the top and bottom surface of the tube for the measurement of the outer wall temperature. A high temperature epoxy resin and a Kapton adhesive layer guarantee the sensors fastening and their electrical insulation from the heated tube. The measurement points for the DC voltage is not located on the copper electrodes, since they might suffer of locally concentrated tension drop. Two measurement wires were instead clamped at a certain distance (see points B and E in Fig. 2) to guarantee the heat flux uniformity. The SS316 test was supplied with DC current (in the range 0-8 V and 0-300 A), and its electrical resistance is estimated to be 5.1 ±0.084 mΩ. A suitable amount of synthetic rubber (λ = 0.040 W/m K at 40 °C) covers the test section and the whole experimental facility in order to minimize the heat losses. The preheater section, due to the higher thermal loads handled, was firstly covered by a high-temperature insulation wool (λ = 0.050 W/m K at 200 °C).
2.3. Measurement instrumentation

With the exception of the test section, all the temperature measurements in the test rig were obtained with high-accuracy flat ceramic resistance thermometers, carrying an overall uncertainty of ±0.180 °C.

The absolute pressure measured at the inlet of the test section, preheater and liquid receiver were obtained with absolute pressure transducers within the range 0-50 bar. Their overall uncertainty, taking into account the non-linearity and repeatability effects is ±0.5% of the read value.

Four T-type thermocouples were employed for the outer wall test tube temperature measurements. All sensors were calibrated in the range 5-55 °C by using a high-accuracy resistance thermometer. Their B-type uncertainty is assumed to be ±0.1 °C.

The mass flow rate was measured by means of a Coriolis flow meter, calibrated up to 2% of the full scale, with a maximum uncertainty of 1% of the measurement.

Two differential pressure transducers were employed for the measurement of the test section pressure drop. The transducers were calibrated with two different full scale of 30 kPa and 50 kPa, respectively, and with an overall uncertainty of ±0.05 kPa.

A digital wattmeter provided the heat load applied to the preheater section, by separately measuring the voltage (100 mV-500 V) and current (1 mA-16 mA). Its uncertainty is 1% of the reading, as indicated by the manufacturer.

The DC voltage applied to the test section was evaluated with an electrical voltage transducer of 0-5 V carrying an uncertainty of ±0.03% of the reading, while the DC current was directly measured by the DC power supply unit (0-300 A). The manufacturer guarantees an overall uncertainty of 1% of the measurement.

2.4. Experimental procedure and data acquisition

All two-phase tests were recorded in steady state conditions. The saturation temperature, mass flux, heat flux and vapor quality were remotely set by Labview software and Arduino One controller.

Specifically, the saturation temperature was imposed by setting the water temperature in the thermostatic bath; the mass velocity was instead controlled by changing the inverter frequency of the motor coupled with the magnetic gear pump. Small adjustments of the mass flow rate and system pressure (i.e. saturation temperature) were possible by manipulating the main circuit by-pass valve and...
the micrometric throttling valve. The heat flux applied to the test section was imposed by remotely changing the voltage from the DC power supply unit. Precisely, the low electrical resistance of the steel tube required very low voltages (up to 1.0 V) and high currents (more than 100 A). The fiberglass heating tape of the preheater section was instead supplied with AC power. By varying the applied voltage by means of a TRIAC electronic unit (up to 240 V), different thermal loads were set to obtain a desired vapor quality at the inlet of the test section.

Data from sensors were recorded with an acquisition frequency of 1.0 Hz and the arithmetic average over 5 minutes was taken as nominal value of the sample. The system was considered stabilized when the calculated deviation of the main parameters was sufficiently low (±0.2 °C for the wall outer temperature obtained by the T-type thermocouples, ±0.1 bar for the inlet saturation pressure, ±10 kg/m² s for the mass velocity, ±1.0 kW/m² for the imposed heat flux).

3. Method

3.1. Data reduction and uncertainty analysis

The local heat transfer coefficient is evaluated with the Newton equation:

\[ h_i = \frac{q}{T_{\text{wall}},i - T_{\text{fluid}},i} \]  

(1)

where the subscript i refers to the specific measurement location (point C or point D in Fig. 2). The heat flux is evaluated as:

\[ \dot{q} = \frac{V_{BE}}{\pi d_{BE}} \frac{l}{I} \]  

(2)

in which \( V_{BE} \) is the voltage applied in the section \( BE \) and \( I \) is the DC current occurring in the test tube.

The wall temperature \( T_{\text{wall}},i \) is evaluated from the measured wall outer temperature \( T_{th},i \) by considering 1-D heat transfer and uniform generation in the metal tube:

\[ T_{\text{wall}},i = T_{th},i + \frac{V_{BE} l}{\pi \lambda_{\text{tube}} d_{BE}} \frac{(D/2)^2}{D} \left( 1 - \log \left( \frac{D}{2} \right) \right)^{-1} \]  

(3)

In the above equation, \( d \) and \( D \) represent the inner and outer diameter of the tube, respectively. \( \lambda_{\text{tube}} \) is the tube thermal conductivity, considered equal to 16.26 W/(m K) for all the experiments performed. The fluid saturation temperature \( T_{\text{fluid}},i \) at the measurement points is evaluated by considering a linear pressure drop from the tube inlet.

The local vapour quality at the measurement points is:

\[ x_i = \frac{i_{li},i - i_{lv},i}{i_{lv},i} \]  

(4)

in which \( i_{li},i \) and \( i_{lv},i \) are the local saturated liquid and vapor enthalpies and \( i_i \) is the local enthalpy at the measurement point, which is computable as:

\[ i_i = i_{in} + \frac{\dot{Q}_i}{\dot{m}} \]  

(5)

where \( \dot{Q}_i \) is the heat load applied up to the measurement section, obtained as a linear interpolation of the total heat applied, \( \dot{m} \) is the refrigerant mass flow rate and \( i_{in} \) is the test section inlet enthalpy, whose value is obtained from an energy balance applied to the preheater section.
In the above equation, \( \dot{m} \) is the preheater inlet enthalpy (obtained with measured temperature and pressure) and \( \dot{Q}_{\text{preh}} \) is the preheater load directly measured by the digital wattmeter.

All refrigerant thermodynamic properties are evaluated with the software REFPROP 9.0, developed by NIST [9]. The whole data reduction is carried out with MATLAB software [10].

The uncertainty analysis of the results was also carried out in this paper. The instrumental B-type uncertainties were composed to the standard deviation evaluated in the recording time for all the measured parameters. For the derived quantities, the law of propagation of uncertainties was implemented. The final results provided a maximum heat transfer coefficient uncertainty equal to ±35% and an average uncertainty of ±15% for all the operating conditions far from the occurrence of dry-out. In this condition, substantial fluctuations of the wall temperatures were detected, increasing the A-type contribution and therefore the overall sample uncertainty. The maximum uncertainty recorded for pressure drop is ±45% in case of low vapor qualities and low pressure gradients, while the average uncertainty in the high vapor quality region was ±8%.

3.2. Test section validation
Preliminary tests were performed in liquid single-phase with refrigerant R134a in order to verify the energy balance in the preheater and in the test section. Moreover, liquid single-phase heat transfer coefficients were obtained and compared with the Dittus-Boelter correlation [11].

Specifically, 16 tests were conducted to verify the correct insulation of the preheater, using mass flow rates from 30 to 40 g/s, heat loads from 100 to 600 W and temperatures from 32 to 38 °C. The difference between the tube and the ambient temperature is approximately the same encountered in the two-phase tests performed for this paper. Fig. 3a compares the electrical heat imposed versus the effective heat absorbed by the fluid. The heat dispersed, calculated by the energy balance, is approximately 7% of the total heat imposed for all the single-phase tests and therefore the real heat considered in the data reduction has been corrected accordingly.

As regards the test section, 29 liquid single-phase tests with refrigerant R134a were carried out, using mass flow rates from 4 to 31 g/s, heat fluxes from 3 to 42 kW/m\(^2\) and inlet temperatures from 18 to 36 °C. The energy balance led to an agreement always better than 5% and therefore the heat losses were neglected. On the other hand, the liquid heat transfer coefficients compared with the Dittus-Boelter, as portrayed in Fig. 3b, showed deviations lower than 15%.

![Figure 3](image-url)

**Figure 3.** R134a liquid single-phase tests. (a) Energy balance on the preheater section. (b) Experimental and predicted liquid single-phase heat transfer coefficient on test section, for the two measurement points, top and bottom data.
4. Results

4.1. Local heat transfer coefficients

Six different operating conditions were considered for the two-phase heat transfer experiments, by using two different inlet saturation temperatures $T_{\text{sat}}$ (25 and 35 °C) and two mass velocities $G$ (150 and 250 kg/m$^2$s), by varying the heat flux $q$ from 5 to 50 kW/m$^2$. Fig. 4 shows the local heat transfer coefficients with their uncertainty for an inlet saturation temperature of 35.1 °C, a mass velocity of 152 kg/m$^2$s and an imposed heat flux of 5.1 kW/m$^2$. The results are shown for measurement points C and D (see Fig.2): as it can be seen, except for the small differences due to vapor quality variations, the trends are in line as a double check of the results from different thermocouples and locations. This behavior was also recorded for other operating conditions and, for this reason, all the remaining diagrams of this section will refer only to the first measurement point C. The top heat transfer coefficients $h_{\text{top}}$ tend to initially decrease up to a vapor quality of 0.4, then a mild increase is detected before the final drop, that in this case occurs for $x=0.80$. The bottom heat transfer coefficients $h_{\text{bot}}$ are instead always increasing with vapor quality up to the dry-out condition, which occurs for higher qualities ($x=0.92$ in this case). Both $h_{\text{bot}}$ and $h_{\text{top}}$ curves almost blend one another, with the small differences included in the uncertainty range.

![Figure 4. R32 local heat transfer coefficient evaluated at $T_{\text{sat}}=35.1\,^\circ\text{C}$, $G=152\,\text{kg/m}^2\,\text{s}$ and $q=5.1\,\text{kW/m}^2$ for measurement points C and D. a) top surface. b) bottom surface](image)

The effect of the heat flux on the top and bottom heat transfer coefficient is shown in Fig. 5a, in which the saturation temperature and the mass velocity are fixed to 25.2 and 150 kg/m$^2$s, respectively. When the heat flux changes from 5 to 20 kW/m$^2$, $h_{\text{bot}}$ increases (+50\% at low vapor qualities and +25\% at $x=0.87$), whereas the heat flux has a negligible effect on $h_{\text{top}}$ and the recorded differences between the values are included in the error band, suggesting a poor contribution of the nucleate boiling mechanism on the upper surface of the tube, for this range of values. Remarkably, the top and bottom contributions have different trends: in particular, while that at the bottom the heat transfer coefficient increases monotonically with the vapor quality, the one at the top has a strong dependence on the flow pattern (film thickness and slug frequencies in the slug region; asymmetry and fluid velocity into the asymmetric annular flow: these results are consistent with previous similar studies, Grauso et al. [12]). To highlight the contribution of the heat flux on the heat transfer at the bottom, Fig. 5b displays instead $h_{\text{bot}}$ at the same conditions for heat fluxes up to 50 kW/m$^2$. The enhancement of the heat transfer coefficient is significant, while the vapor quality at the dry-out incipience remains substantially the same ($\approx 0.92$) up to 20 kW/m$^2$, whereas it drops to 0.78 at $q=50\,\text{kW/m}^2$. 

7
Figure 5. R32 local heat transfer coefficient evaluated at $T_{\text{sat}} = 25.2^\circ\text{C}$, $G = 150 \text{ kg/m}^2\text{s}$ for different heat fluxes. a) Top and bottom heat transfer coefficients for heat fluxes of 5 and 20 kW/m$^2$. b) Bottom heat transfer coefficients for heat fluxes from 5 to 50 kW/m$^2$.

The effect of the saturation temperature (equal to 25 and 35 $^\circ\text{C}$) on $h_{\text{bot}}$ and $h_{\text{top}}$ is displayed in Fig. 6, for a heat flux of 20.3 kW/m$^2$ and an average mass flux of 152 kg/m$^2\text{s}$. While the bottom heat transfer coefficient (see Fig. 6a) is always enhanced (about +20%) with increasing saturation temperature, the effect on the top heat transfer coefficient (see Fig. 6b) changes according to the vapor quality. For $x < 0.3$, the saturation temperature has a positive effect on $h_{\text{top}}$, while the trend is reversed for higher vapor qualities. As suggested by Grauso et al. [12], a higher reduced pressure causes a higher liquid density and therefore a lower fluid velocity, resulting in a lower heat transfer coefficient during annular flow. Both curves reach the same heat transfer coefficient when dry-out occurs (at approximately $x = 0.90$).

Figure 6. R32 local heat transfer coefficient evaluated at $q = 20.3 \text{ kW/m}^2$, $G = 152 \text{ kg/m}^2\text{s}$ for two different saturation temperatures. a) Bottom heat transfer coefficients for $T_{\text{sat}}$ of 25 and 35 $^\circ\text{C}$. b) Top heat transfer coefficients for $T_{\text{sat}}$ of 25 and 35 $^\circ\text{C}$.

The effect of mass velocity on bottom and top heat transfer coefficient is portrayed in Fig. 7a and Fig. 7b, respectively. In these tests, the saturation temperature is fixed at 35 $^\circ\text{C}$ and the heat flux at 5.1 kW/m$^2$. As regards the bottom section (see Fig. 7a), the mass flux increases the heat transfer efficiency and its effect is more evident for higher vapor qualities. The top heat transfer coefficient (see Fig. 7b) is instead significantly improved (more than +100%) with increasing mass flux in case of vapor qualities included between $x = 0.25$ and $x = 0.90$, whereas $h_{\text{top}}$ is penalized for lower vapor qualities.
Figure 7. R32 local heat transfer coefficient evaluated at $T_{\text{sat}} = 35.2^\circ\text{C}$, $q = 5.1 \text{ kW/m}^2$ for different mass fluxes. a) Bottom heat transfer coefficients for $G$ equal to 150 and 250 kg/m$^2$/s. b) Top heat transfer coefficients for $G$ equal to 150 and 250 kg/m$^2$/s.

4.2. Adiabatic pressure gradients

Under the same operating conditions for the heat transfer, data were also used for the evaluation of the adiabatic pressure gradient along the test section. The following diagram refers to the total pressure gradient measured by the pressure transducer with the lower full scale of 30 kPa and divided over the lube length. Due to the very small variation of the vapor quality between the inlet and outlet section of the tube under adiabatic conditions, the momentum contribution was neglected.

Fig. 8 exposes the total pressure gradient as a function of the local vapor quality, for a saturation temperature of 35 °C. The blue and red markers refer to mass velocities of 150 and 250 kg/m$^2$/s, respectively. As expected from previous studies [13], the pressure drop increases with ongoing evaporation for both mass fluxes, even if the influence of vapor quality is greater for the higher mass flux of 250 kg/m$^2$/s. Pressure drop is also significantly increased with mass velocity: at $x = 0.6$, the pressure gradient for $G = 250 \text{ kg/m}^2$/s is approximately 180% higher than that obtained for $G = 150 \text{ kg/m}^2$/s. Finally, the maximum was found for lower a lower vapor quality (0.78 instead of 0.82) for $G = 250 \text{ kg/m}^2$/s.

Figure 8. R32 total pressure gradient vs local vapor quality, for $T_{\text{sat}} = 35 ^\circ\text{C}$. The mass velocity is fixed to 150 and 250 kg/m$^2$/s.
5. Conclusions

New experimental data on local two-phase heat transfer coefficients and adiabatic pressure gradients were collected in this work, for a horizontal tube of 6.00 mm internal diameter. The main conclusions can be summarized as follows:

- The heat flux enhances the bottom heat transfer coefficient and has instead a poor effect on $h_{top}$, which does not seem to be significantly affected by the nucleate boiling contribution. Consistently, also a variation from low to higher reduced pressures lead to higher bottom heat transfer coefficients, whereas $h_{top}$ is conversely reduced (at least in the annular flow region) due to the reduction of the liquid velocity caused by the increased liquid density. Also, the trend of $h_{top}$, being this heat transfer contribution related only to the convection, is strongly dependent on the flow regime.

- The bottom heat transfer coefficient is increased with higher mass flux, with greater differences for higher vapor qualities, whereas $h_{top}$ is enhanced only in the annular flow region.

- The total pressure gradient is significantly dependent on vapor quality, showing a maximum for vapor qualities approximately equal to 0.80. The mass velocity (from 150 to 250 kg/m² s) also increases the pressure drop, which passes from 0.5 to 1.6 kPa when the vapor quality is fixed to 0.60.

References

[1] Regulation (EU) No 517/2014 of the European Parliament and the Council of 16 April 2014 Off. J. Union.

[2] Shuxue X, Guoyuan M, Qi L and Zhongliang L 2013 Int. J. Th. Sc. 68 103

[3] Han X H, Wang Q, Zhu Z W and Chen G M 2007 App. Th. Eng. 27 2559

[4] Shin J Y, Kim M S and Ro S T 1997 Int. J. Ref. 20 267-275

[5] Li M, Dang C and Hihara E 2012 Int. J. Heat Mass Tr. 55 3437

[6] Kondou C, Baba D, Mishima F and Koyama S 2013 Int. J. Ref. 36 2366

[7] Azzolin M, Bortolin S and Del Col D 2016 Int. J. Th. Sc. 110 83

[8] Jige D, Sagawa K and Inoue N 2017 Int. J. Ref. 76 206

[9] Lemmon E W, Mc Linden M O and Huber M L 2009 Standard Reference Database 23

[10] MATLAB release. Natick, Massachusetts, United States, The MathWorks, Inc.

[11] Dittus F W and Boelter L M K 1930 Univ. Calif. Publ. Engin. 2443

[12] Grauso S, Mastrullo R, Mauro A W and Vanoli G P 2013 Int. J. Heat Mass Tr. 56 107

[13] Grauso S, Mastrullo R, Mauro A W and Vanoli G P 2014 Exp. Th. Fl. Sc. 52 79