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Investigations on pool boiling of refrigerant R141b outside a horizontal tube

Boiling produces vapor with a phase change by absorbing a consistent amount of heat. Experimentation and modeling can help us better understand this phenomenon. The present study is focused on the heat transfer during the nucleate pool boiling of refrigerant R141b on the surface of a horizontal copper tube. The results of the experiment were compared with four correlations drawn from the literature, and the critical heat flux was examined for different pressures and also compared with the predicted values. Simulating boiling with two-phase models allowed us to infer the plot of the temperature distribution around the tube and compared it to results from other work.

Nomenclature

Symbol	Designation	Unit SI
A	coefficient	–
C	coefficient	–
C_{sf}	Rohsenow coefficient	–
c_p	heat capacity	J/kg K
D	tube diameter	m
E	energie	J
e	enthalpie	J
F	force	N
g	cceleration due to gravity	m/s^2

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h	heat transfer coefficient	W/m^{2}K
K	coefficient	–
L	latent heat of vaporization	J/kg
l_c	capillary length	m
M	molecular weight	kg/mol
m	mass	kg
p	pressure	Pa
Q	heat	J
q	surface heat flux	W/m^2
S	heat source	J
T	temperature	K
t	time	s
u	velocity	m/s
V	volume	m^3
v	velocity	m/s
x	coordinate	m
y	coordinate	m

Greek symbol

α	volume fraction	–
Δ	difference	–
ε	quadratic roughness	µm
ρ	density	kg/m^3
λ	thermal conductivity	W/m K
μ	dynamic viscosity	Pa·s
ν	cinematic viscosity	m^2/s
σ	surface tension	N/m
τ	stress strain tensor	–

Indices

c	critic
cal	calculated
dr	drift
eff	effective
exp	experimental
k	phase k
l	liquid
lift	lift
m	mixture
max	maximum
p	phase p
q	phase q
r	reduced
ref	reference
v	vapor
sat	saturated

1. Introduction

Vapor is produced by boiling in evaporators, which find their applications in the refrigeration industry. During the phase change, the liquid absorbs a large quantity of heat, however, this transfer mode is difficult to define because it depends on several parameters such as the saturation pressure, the thermo-physical properties of the fluid, the characteristics of the surface, the properties of the material, dimensions, orientation, thickness, surface quality and microstructure as underlined by Dhir [1], and Pioro [2].

Characterization of the heat transfer requires knowledge of thermal behavior, hence the search for correlations for the prediction adapted to the fluid. Baki and Aris [3] experimentally examined the boiling of the refrigerant R141b outside a horizontal tube, Baki et al. [4, 5] analyzed the influence of the diameter of the heating element on the boiling of pure fluids and have proposed a correlation that can determine the coefficient of heat transfer. Baki [6–8] compared the experimental boiling data with those of values predicted by the correlations taken from the literature, and Baki [9] compared the experimental data of the pool boiling liquid hydrogen with known correlations and studied the critical flux and the heat transfer coefficient.

Deb et al. [10] carried out experiments to study the impact of surface wettability on nucleate boiling of refrigerant R141b at atmospheric pressure. Khliyeva et al. [11] examined the vessel boiling of refrigerant R141b for various pressures and heat flows, and its mixing with a surfactant. Abdullah and Sudarmanta [12] used refrigerant R141b as the working fluid to produce electrical energy with the organic Rankine cycle. Li et al. [13] experimentally studied the pool boiling of refrigerant R141b at atmospheric pressure on a surface covered with copper foam.

Many correlations have been developed to predict the heat transfer coefficient; the most used are those of Rohsenow [14], Labuntsov [15], and Cooper [16]. It is well known that the correlation developed by Rohsenow [14] for the estimation of heat flux during nucleated boiling depends on the fluid and surface combination, and gives good performance with low error. All these correlations are used in a general framework whatever the type of heating, plates, wire, or tube. Baki et al. [5] developed a correlation to predict the heat transfer coefficient applicable to horizontal tubes, which depends on the diameter of the heating element.

When a tube is electrically heated with a uniform flow, a temperature distribution is noted around the diameter of the tube and depends on the orientation angle. Cornwell [17] was the first one who examined this variation. Dominiczak and Cieśliński [18] experimentally studied the temperature distribution around the circumference of a horizontal tube during the boiling of refrigerant R141b and water.

The study of the critical heat flux, the CHF, determines the limit during boiling before the onset of the crisis. Fukuda and Sakurai [19] studied the effect of the diameter of the heating element and effect of the pressure on the critical heat flux.

Kandlikar [20], in his article, explained the phenomenon of the boiling crisis and presented the effect of the pressure on the CHF.

In this study, we carried out experiments of boiling the refrigerant R141b, outside a horizontal tube, with several pressure levels. The obtained results, which allowed us to determine the heat transfer coefficient as a function of the heat flux, were compared with four correlations from the literature. By increasing the heat flux, we have drawn a curve relating the pressure to the critical heat flux, CHF. Likewise, we simulated boiling with a CFD code using two-phase models.

2. Review of empirical correlations

Boiling outside the horizontal tubes finds its application in various equipment of industry, such as the evaporators and the steam generators. To examine this problem, we are interested in quantifying heat transfer by using the important parameters that affect pool boiling.

A set of empirical correlations has been proposed to assess the heat transfer rate during nucleate pool boiling on a heated surface submerged in a liquid [5, 14–16]. The heat transfer was studied with the use of various modes and it was found that the principal parameters, which influence pool boiling, are the heat flux, the saturated pressure, the thermo-physical properties, and the characteristics of the surface of boiling.

We then selected four correlations.

2.1. Rohsenow correlation (1952) [14]

$$\left[\frac{c_p \Delta T}{L}\right] = C_{sf}\left[\frac{q}{\mu L}\left(\frac{\sigma}{g\left(\rho_l - \rho_v\right)}\right)^{0.5}\right]^n \left(\frac{c_p \mu}{\lambda}\right)^{m+1}. \tag{1}$$

The Rohsenow equation (1) remains the most frequently used and popular among the correlations for the evaluation of the heat transfer during pool boiling, and it will be presented in its most known form. This correlation covers a broad range of fluids, operating parameters and pressure values. It is a function of the thermophysical characteristics of the fluid and contains three coefficients, m, n and C_{sf}. The coefficient m is equal to 0.7 for any fluid except for water, where is equal to zero; n is taken equal to 0.33 and C_{sf} is the combination coefficient of liquid – heating surface; when it is not known, it is estimated as 0.013 for a first approximation.

2.2. Labuntsov correlation (1972) [15]

$$h = 0.075\left[1 + 10\left(\frac{\rho_v}{\rho_l - \rho_v}\right)^{0.67}\right]\left(\frac{\lambda^2}{\nu\sigma T_{\text{sat}}}\right)^{0.33} q^{0.67}. \tag{2}$$

The equation of Labuntsov (2) is presented in the form of the heat transfer coefficient in the function of only the thermophysical characteristics of the fluid and the heat flux raised to the power of 0.67, without any coefficient. It generally gives the minimum value of the coefficient compared to other correlations, it applies to all the fluids, within a wide range of operating characteristics and pressures.

2.3. Cooper correlation (1984) [16]

$$h = Ap_r^{(0.12-0.21\log_{10}(\varepsilon))}\left(-\log_{10}(p_r)\right)^{-0.55}M^{-0.5}q^{0.67}. \tag{3}$$

The equation of Cooper (3) enables us to obtain directly the heat transfer coefficient in relation to the reduced pressure, the roughness of the heating surface, the molecular mass, and the surface heat flux. Roughness, when it is not known, we assume ε equal to the value of 1 μm. In fact, according to the correlation, roughness has little influence on the result of the heat transfer coefficient. The coefficient A is equal to 55 for the copper tubes.

2.4. Baki et al. correlation (2014) [5]

$$\frac{hD}{\lambda} = 0.5\left(\frac{qD}{\mu L}\right)^{0.67}\left(\frac{\mu c_p}{\lambda}\right)^{0.40}\left(\frac{p}{p_c}\right)^{-0.10}\left(\frac{\varepsilon}{D}\right)^{0.07}\left(\frac{lc}{D}\right)^{-0.20}. \tag{4}$$

The correlations quoted above (1)–(3) are used for the determination of the heat transfer coefficient for any immersed heating surface, whatever the form: plates wire or tube. Equation (4) of Baki et al. [5] includes the diameter effect of a horizontal tube, as well as roughness, the pressure, and the thermophysical characteristics of the fluid; the roughness ε is assumed 1 μm when its value is not known.

3. Experimental setup

We proceeded to experiment with the nucleate boiling on a tube immersed in a liquid refrigerant R141b. The boiling apparatus comprises an electrical heating element in copper and is horizontally mounted in a cylindrical vertical chamber made of glass. The measured temperature is the average of readings from four Type-K thermocouples. It is displayed on a temperature indicator; the thermocouples are placed at the surface, at 90°from the lowest position of the diameter, two at each side. It was estimated that the temperature was determined with an accuracy of ±0.5 K.

The electrical supply to the heater element is controlled by a potentiometer, the consumption is displayed by a power meter, and it was determined with an accuracy of 3%. At the upper end, there is connected a condenser and a nickeled copper tube coil in which cooling water passes. This coil condenses the steam produced during boiling and returns it at the bottom of the room to be evaporated again (see the schematic diagram in Fig. 1). The heating surface is cylindrical in shape with a diameter of 12.7 mm and a working length of 42 mm, which results in a surface of 0.0018 m^2. The total volume of the chamber is 1.5 liters, and it must be completed at the third one, so that the heating tube is submerged. The air above the liquid should be evacuated; the liquid temperature is shown using a diving thermometer in the liquid and another one connected to the computer.

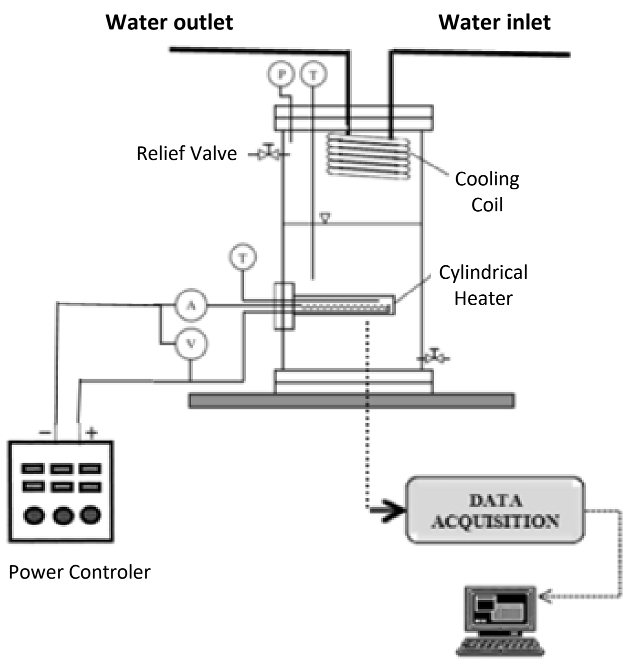
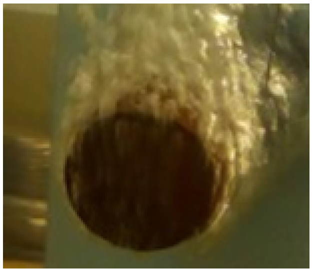


Fig. 1. Schematic diagram of the boiling apparatus

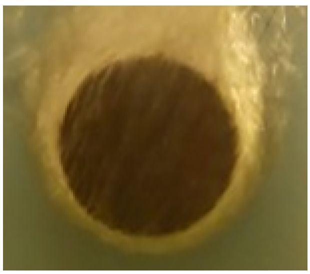
The experiment was carried out for four pressure levels, 150, 200, 250 and 300 kN/m62 and the data were collected by incrementing the heat flux per stage. The temperature of saturation of R141b for each pressure was respectively 44, 53, 61, and 67°C; a dial gauge indicated the pressure in the room. The series of measurements obtained relates to only the section of nucleate boiling. The average heat flux was ranging from 10 to 150 kW/m^2 and the difference between the wall temperature and the liquid saturation temperature was between 8 and 18°C.

Photos were taken when handling the experiment for different heat flux rates and a pressure of 130 kN/m^2. Figs 2a–2d show the intensity of the boiling when the heat flux increases.



(a) q = 34.44 kW/m^2

(b) q = 63.33 kW/m^2



(c) q = 101.67 kW/m^2

(d) q = 173.33 kW/m^2

Fig. 2. Photos taken during the handling of experiments for P = 130 kN/m^2

4. Result and discussion

4.1. Results of the experiment

The study relates to heat transfer during pool nucleate boiling outside a submerged horizontal tube for the pressure range from 150 kN/m^2 300 kN/m^2. The results of the experimental measurements are indicated above in Fig. 3 that shows the relationship between the temperature difference between the wall and the fluid saturation temperature and the heat flux for different pressures.

When the temperature increases, the heat flux increases; for a given temperature, when the pressure increases, the evacuated heat flux is much more important. When the temperature increases by 15 K, an amount of heat flux of 70 kW/m^2 is absorbed, and when the pressure goes from 150 to 200 kN/m^2 this quantity is increased by 50%, and after the pressure reaches 250 kN/m^2, the heat flux is improved by 92%. At 300 kN/m^2, the heat flux doubles reaching up to 104% compared to the first value.

The whole of the data was presented in the three-dimensional graph of Fig. 4 showing the relationship between temperature, pressure, and heat flux. The graph clearly shows the effect of pressure; for low-temperature gradient, when pressure increases, the heat flux increases more rapidly.

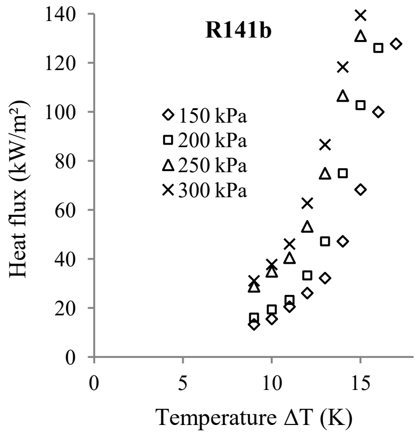


Fig. 3. Evolution of the heat flux according to the temperature

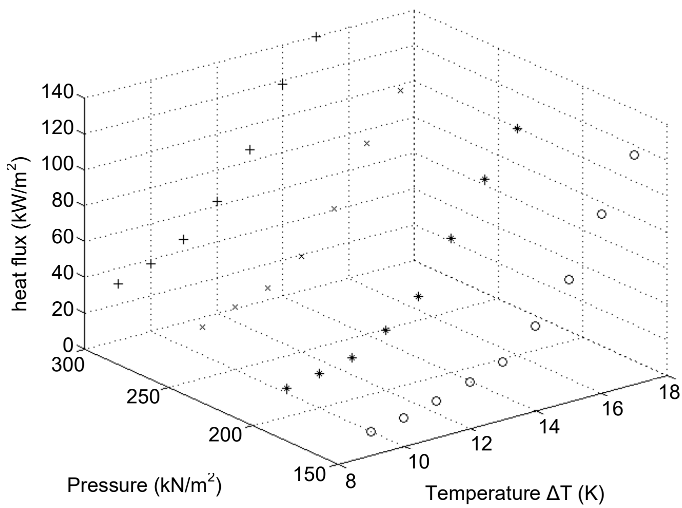


Fig. 4. Evolution of the heat flux according to the pressure and temperature

4.2. Determination of the coefficient C_{sf}

Calculating the heat transfer coefficient with the correlation of Rohsenow [14] requires knowledge of the C_{sf}coefficient, which depends on the couple of the heating surface and the fluid. In our experiment, we used the fluid R141b on a copper surface. This surface/fluid couple has not been tested in previous experiments, we then used the data collected from own experience and we plotted $\left[\frac{c_p \Delta T}{L}\right]$ in the function of $\left[\frac{q}{\mu L}\left(\frac{\sigma}{g(\rho_l-\rho_v)}\right)^{0.5}\right]^{0.33}\left(\frac{c_p\mu}{\lambda}\right)^{1.7}$, as shown in Fig. 5.

The coefficient C_{sf} was obtained by means of the least squares method, and was equal to 0.048.

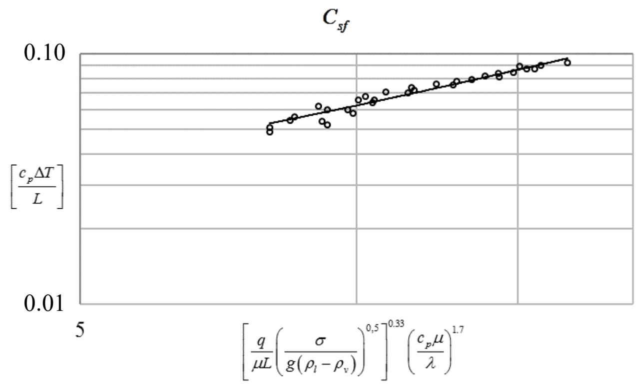


Fig. 5. Determination of the coefficient C_{sf}

4.3. Comparison of correlation with data

The experimental results of heat transfer during nucleate boiling were compared with the data of correlations provided by literature. The results of nucleate boiling away points of the onset nucleate boiling and the critical heat flux is given by the relationship of the heat flux coefficient versus surface heat flux, in the form such as of $h \propto q^{0.67}$.

The deviation between experiment and calculation will be determined as the relative difference between the heat transfer coefficient obtained by the experiment and calculation from the value of the experiment, as shown in the following equation (5):

$$\text{deviation} = \frac{h_{\text{exp}} - h_{\text{cal}}}{h_{\text{exp}}} \times 100. \tag{5}$$

Comparisons of the experiment results and the data calculated from the correlations of Rohsenow [14], Labuntsov [15], Cooper [16], and Baki et al. [5] are shown in the following Figs 6a–6d.

4.4. Critical heat flux

With the same experimental device, we proceeded to the study of critical heat flux. By varying the pressure, we obtained the curve shown in Fig. 7, which illustrates the relationship between the critical heat flux depending and the pressure. The increase is similar to that presented by Kandlikar [20] in his article for the case of the pool boiling for another liquid. Increasing the pressure influences positively the increase of critical heat flux with a slope of 0.39. We compared the results of

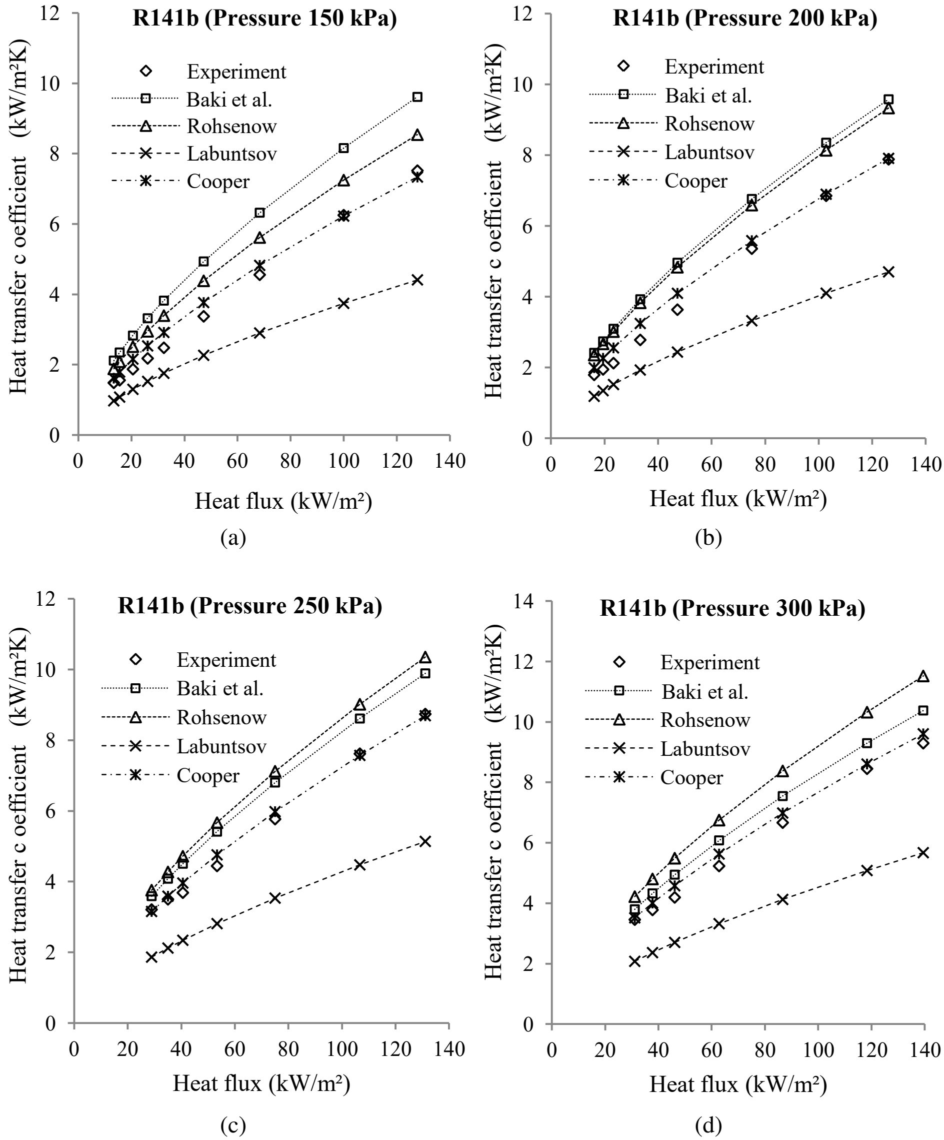


Fig. 6. Comparison of experimental data with correlations

the experiment with the Kutateladze correlation [21], which gives the CHF based on thermophysical characteristics of the fluid. The coefficient C was taken equal to 0.118, as it was stated by Rohsenow et al. [22]; the results are significant since the discrepancy is between 12% for low pressure and 9% for higher pressures.

$$q_c = C\rho_v^{1/2} L\left(g\left(\rho_l - \rho_v\right)\sigma\right)^{1/4}. \tag{6}$$

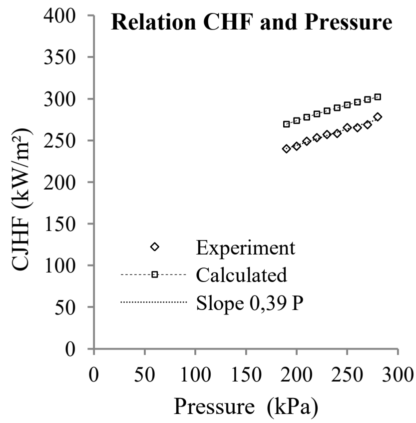


Fig. 7. Variation of CHF with the pressure

5. Simulation

The simulation was done using the CFD code based on numerical methods of finite volume, and the drawing and mesh were generated by Gambit. After the introduction of the initial values and the specifications of the boundary conditions for laminar flow, the control of the solution, as well as initialization, must be specified before starting of computing. The diphasic Mixture model is used with default settings, a UDF function is introduced to deal with the mass transfer of the liquid phase to the vapor phase and the second diphasic model simulated is Eulerian, Non-equilibrium boiling.

5.1. Simulation domain

Taking advantage of the symmetry, the control domain selected is of dimensions 13 mm $\times$ 26 mm (1D $\times$ 2D), with a half-cylinder placed in the middle of the height. The low part of it is the entry of the fluid with null velocity (velocity inlet), the two sides are symmetries (symmetry), and the upper part is the exit (outlet pressure) and the tube is the part heated with a heat flux (wall). The calculations are made in Cartesian coordinates in two dimensions. A triangular grid was selected with a step of 0.2 on the height and the same step on the basis. Adding up 14562 elements one obtains the diameter of the tube equal to 13.52 mm. The studied process is pool boiling with the R141b refrigerant at the atmospheric pressure and at the saturation temperature; the heat flux will then be injected through the surface of the tube.

5.2. Initial conditions

A heat flux leaving the tube (wall), $q = 14200$ W/m^2, is initialized with $u = 0$, $v = 0$ at the entry of the domain, i.e., at $y = 0$. Above the domain, one has $u = 0$,

$v = 0$ at $y = y_{max}$; at the levels of symmetries one has $u = 0$, $v = 0$, at $x = 0$ and at $x = x_{max}$. The temperature of the liquid in all the domain is $T = T_{sat}$. All the results presented have been computed after 300 iterations with a step of time of 0.01 s.

5.3. Grid influence

Two other meshes were tested, with a step of 0.22 according to x and 0.18 to the y-axis, then another one 0.18×0.22, always with triangular meshes, giving a number of cells of 16144 and 13865, respectively. Simulation was made with the Mixture model, and the result obtained gave a distribution of the temperature around the tube, as indicated in Fig. 8. One observes that, for the grids of 13865 and 14562 elements, the distribution of the temperature is practically similar, contrariwise for the grid of cells number of 16144 in which the temperature is lower by 2 to 3 K. We may then conclude that the influence of grid is small.

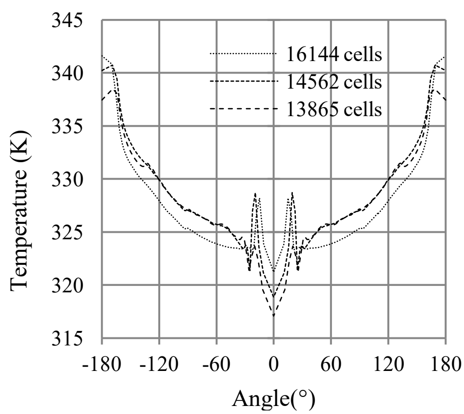
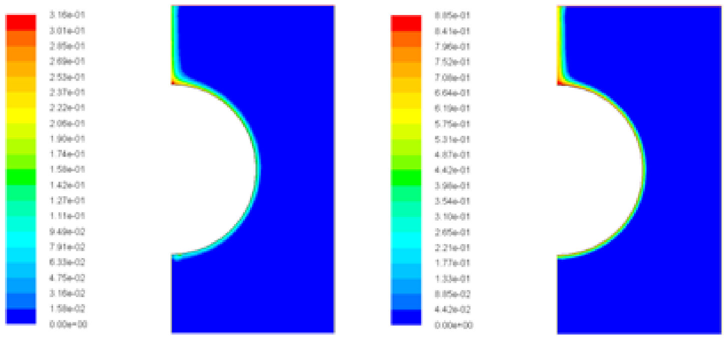


Fig. 8. Mesh influence

5.4. Simulation result

After a 3 seconds simulation, we obtained volume fractions around the tube, due to the production of vapor starting from a hot surface, as shows in Figs 9a and 9b. The vapor follows the circumference of the tube and gathers above, taking the shape of a plume. The Eulerian model makes it possible to produce more vapor so the volume fraction is definitely higher than that of Mixture model. The latter one indicates a fraction which reaches 0.32 and stagnates at the point higher of the tube, while the fraction of the Eulerian model follows the plume at a certain distance and its fraction exceeds 0.88.

The temperature of the wall around the tube for the two diphasic models, indicated in Fig. 10, was compared with the experimental data of Dominiczak and



(a) Mixture (b) Eulerian

Fig. 9. Contour of the volume fraction

Cieśliński [18]. It was noted that the Eulerian model presents peaks of temperature on the top and in the bottom of the tube, that is at 0 and 180°, but the other values are consistent with the experimental data. The Mixture model gives a shift of 10 K compared to the experimental data but it does not show peaks at the extreme points of the tube.

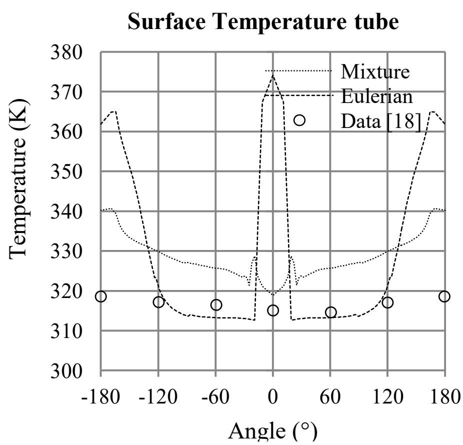


Fig. 10. Temperature of the wall tube

Computing at 13 mm from the center of the tube towards the exit of the domain, we traced the velocity and volume fraction profile of the two models, as shown in Figs 11a and 11b. We noticed that the plume had a width of 2 mm in the two cases, and the velocity of the moving flow and the volume fraction of vapor was more important for the Eulerian model. The last one produces more vapor, so the volume fraction reaches 0.88. The velocity is greater than 0.4 m/s for the Eulerian model, four time higher than that in the Mixture one.

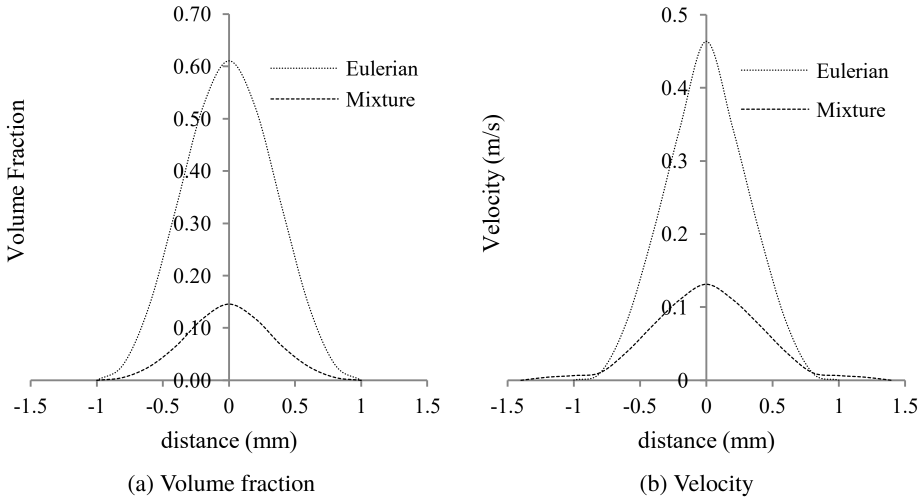


Fig. 11. Velocity and volume fraction above the tube

6. Conclusion

- Experiments on nucleate boiling of a refrigerant R141b outside a horizontal tube were conducted with four pressure levels and different rates of heat flux.
- The coefficient of Rohsenow correlation [14] was determined based on the results of the experiment and was found equal to 0.0048 for the couple R141b/copper.
- The results of the heat transfer coefficient obtained from the experiment were compared with four correlations known from the literature, and have shown a good agreement.
- Experimental results showed that the pressure acts positively on the critical heat flux. These data were compared with the Kutateladze correlation [21], giving good results with margins of error of 9 to 12%.
- A simulation of boiling outside of a horizontal tube with two diphasic models was performed; the temperature distribution obtained was compared with the experiment results known from literature and gave convincing results with a small discrepancy.
- The volume fraction and velocity of the flow at the exit of the domain were compared between the two models.

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