

EXPERIMENTAL INVESTIGATION ON PURE STEAM AND STEAM-AIR MIXTURE CONDENSATION INSIDE TUBES

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ABSTRACT

Experiments on steam condensation inside inclined tubes were carried out with the following aims: a) to investigate the physical phenomena involved in condensation of steam within tubes; b) to study the influence of the geometry (namely, tube inclination) on the heat transfer rate, also in presence of high concentration of non-condensables; c) to develop models and heat transfer correlations for these conditions; d) to produce a database for modeling in-tube condensation with high percentage of non-condensable gases.

Steam and steam-air condensation experiments were carried out in gravity controlled stratified flow regime inside a horizontal and inclined tube (22 mm inside diameter) and the average heat transfer coefficient has been evaluated. For pure steam condensation, the experimental data were compared with literature correlations and their agreement has been verified, suggesting some minor modifications. A limited influence of tube inclination on heat transfer has been observed: condensation in the presence of non-condensable gases is not sensibly affected by inclination, especially at high gas concentrations. Two empirical correlation are proposed to be used in the preliminary design of a condenser in a passive containment cooling system, as in thermal-hydraulic simulation, especially in transient conditions, when a high gas concentration is present.

1. INTRODUCTION

Following an eventual loss of a water cooled nuclear reactor primary loop integrity, primary coolant is collected in the reactor containment and decay heat must be removed from the reactor containment itself. Several passive containment cooling systems (PCCS) have been proposed for the next generation of nuclear reactor plants. For example, in the Simplified Boiling Water Reactor (SBWR), the Isolation Condenser (IC) includes heat exchangers permitting the heat transfer via in-tube steam condensation from the dry-well to the ultimate heat sinks, which may be the suppression pool or even a water pool outside the containment.

Another type of innovative reactor containment emergency cooling foresees the removal of heat from the reactor containment through an intermediate cooling loop which draws heat from the containment by means of stagnant water / flowing water heat exchangers or condensing steam / flowing water heat exchangers. The heat removed is utilized to heat up an external pool, where boiling takes place at atmospheric pressure and the steam produced is sent to a condenser externally cooled by air. Some designs of this condensing unit are conceived to operate with high concentration of non-condensable gases, in order to avoid an operation of the whole system under vacuum - or pressurized - conditions, with a negative effect for the overall reliability.

In order to allow a suitable design of all these components, a fundamental aspect to be analyzed is the behavior of steam condensation when non-condensable gases are present.

The main objectives of the theoretical and experimental research project were:

- To investigate the physical phenomena involved in steam condensation within tubes.

- To study the influence of the geometry (namely, the tube inclination) on the heat transfer rate, also in presence of high concentration of non-condensable gases.
- To develop models and heat transfer correlations for the given conditions.
- To produce a database for modeling in-tube condensation with high percentage of non-condensables.

The results have been useful to evaluate the effect of high-concentration non-condensable gases in a condensing steam stream, with respect to the overall heat transfer coefficient. These have allowed to identify the preliminary design requirements for inclined-tube, high non-condensables concentration steam condensers, operating at atmospheric pressure, to be used for totally static reactor containment emergency cooling and decay heat removal systems.

2. THE TEST FACILITY

Preliminary experiments were conducted in the past [1] with a simple test section, to refine measurement procedure and the test matrix. A new experimental facility was then realized and described in detail in [2]

The in-tube heat transfer performance was investigated in the following conditions:

- Horizontal or inclined tubes with angles of 5°, 15°, 30° and 45° degrees with respect to the horizontal;
- In-tube water flow;
- In-tube vapor flow;
- In-tube air-vapor mixture flow in steady state conditions with gas concentration ranging from 0% to 100%;
- External water cooling in forced convection.

The main components of the experimental facility are (Figure 1):

- an atmospheric steam generator, electrically heated, with power ranging from 375 W to a maximum of 6 kW;
- a condensation test section, consisting of a stainless steel tube contained inside a Plexiglas structure in order to generate an annulus. Tube dimensions are: ID/OD= 22/25 mm, length=1587 mm. The coolant water flows through the annulus between the Plexiglas structure, with a circular cavity of 32 mm in diameter, and the 25 mm stainless steel tube, so that the gap size between the two tubes is 3.5 mm;
- a supporting structure;
- an auxiliary air supply system dedicated to provide a fixed air flow rate to the test section. Tests in steady state conditions can be performed. The main components of this system are: a pressurized air cylinder, an air filter, a pressure-reducer, three air flow meters, pipes and nozzles;
- a forced convection cooling system, with a water flow meter;
- an auxiliary hot water system to supply hot water inside the stainless steel tube in order to perform the external heat transfer coefficient characterization tests;
- Instrumentation and a data acquisition system.

The operating conditions of the test section were:

pressure:	about 0.1 MPa
air flow rate:	from 0 to $6 \cdot 10^{-4}$ kg/s
steam flow rate:	from $2.3 \cdot 10^{-4}$ to $2.3 \cdot 10^{-3}$ kg/s
air content:	from 0% to 100%
mixture velocity:	from 0.85 to 5.55 kg/m ² s
average heat flux:	from 7500 to 136,000 W/m ² .

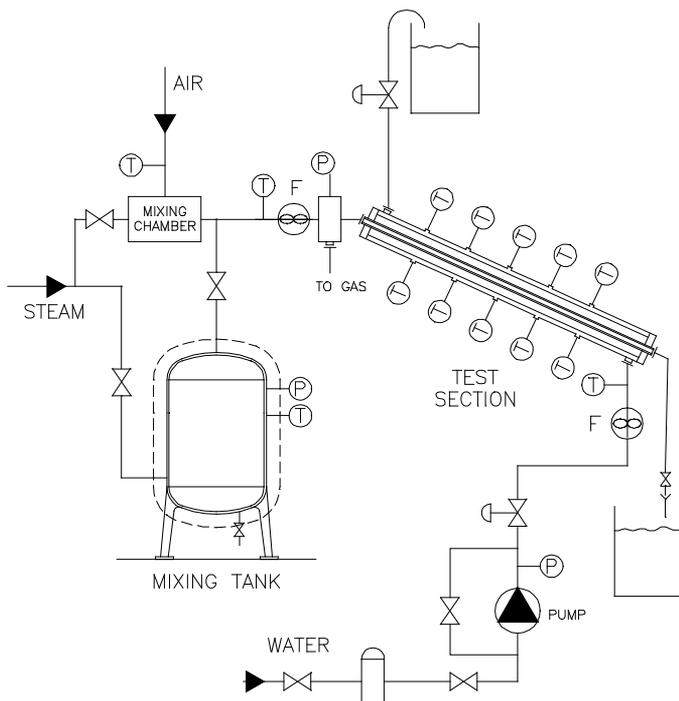


Figure 1 - Test facility

The experimental facility instrumentation includes pressure, temperatures and flow rates measurement with suitable equipment and the measure of air concentration in the mixture through a dedicated gas-chromatography system.

Air flow rate was measured through a volumetric flow rate transmitter in the auxiliary air supply line. The instruments have an hysteresis and a repeatability error of 0,5%. A vapor volumetric flow rate transmitter at the outlet of the steam generator was installed, with an uncertainty lower than 2% of the measured value. A Coriolis law-based flow meter was used in the water cooling loop, certified with an instrument accuracy of 0.15% of the measured value. Finally, two different volumetric flow meter (one for large flow rates and temperature lower than 80°C and one for low flow rates and temperature up to 140°C) were installed in the auxiliary loop, used to perform external coefficient characterization tests; their accuracy was certified within 0.5% of the measured value.

Thermocouples were installed in the test section, in 33 different locations:

- 9 T-type thermocouples located inside the stainless steel tube, with the aim of measuring the centerline tube temperature (fig. 2.9).
- 11 T-type thermocouples are soldered on the stainless steel tube surfaces, seven on the upper side and four on the lower side of the tube surface.
- 11 T-type thermocouples inside the annulus, seven on the upper side and four on the lower side.
- 2 T-type thermocouples at the inlet and the outlet of the test section cooling water loop.

All these thermocouples were certified with a three points calibration (0° C, 50° C, and 100° C), with an accuracy of 0.1 °C. The measure of the mixture air content concentration was also performed through a gas-chromatography system. The measurement point was located in the upper side of the mixing tank, in order to measure the mixture composition before the test section inlet. Water has been chosen as cooling medium due to its quite high heat transfer coefficient, even if the coolant flows in laminar or transient conditions.

3. CHARACTERIZATION TESTS

The main objectives of the characterization tests were:

- the verification of the temperature values accuracy provided through the thermocouples located inside the test section;
- the experimental evaluation of the heat transfer coefficient inside the annular gap.

The experimental measurement of the external heat transfer coefficient was needed because of the cooling water flow pattern in the annulus. In fact the cooling water flow rate was very low, and the external heat transfer coefficient evaluation in the annulus might introduce many uncertainties, using correlations available in literature, because the hydrodynamic conditions (Reynolds number) range around the laminar and the transition zone. To minimize uncertainties in the condensation heat transfer coefficients measurement, several tests have been carried out (characterization tests) to experimentally evaluate the heat transfer coefficient in the test section annulus. These tests were performed with different cooling water flow rates and using hot water in forced convection as heating medium, inside the test tube. Thanks to the use of Petukhov-Gnielinski correlation, for the internal heat transfer coefficient evaluation, and the employment of the energy balances, the external heat transfer coefficient could be reliably estimated.

Tests with several cooling water flow rates, test section inclinations of 0°, 15°, 30°, and 45° and hot fluid temperatures of 50°C, 75°C, 85°C and 95°C were performed. In Figure 2 the non-dimensional group $Nu/(Pr^{1/3} * (\mu/\mu_w)^{-1.31})$ is plotted versus cooling water Reynolds number and a fitting curve is evaluated. This figure shows that the non-dimensional group $Nu/(Pr^{1/3} * (\mu/\mu_w)^{-1.31})$ can be considered a function of the cooling water Reynolds number, and the test section inclination has no appreciable effects.

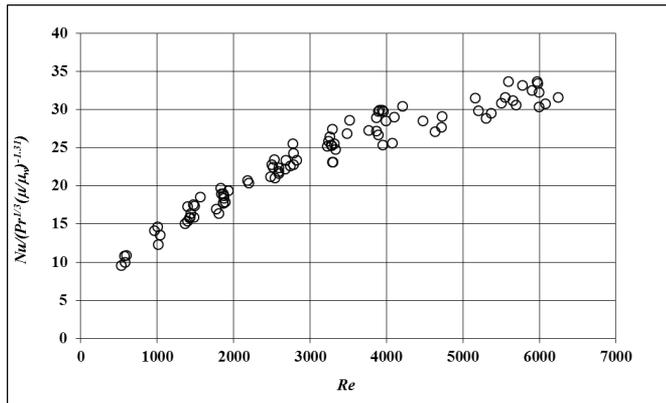


Figure 2 – Heat transfer coefficient in the annulus

Considering all the experimental data, the following correlation was obtained:

$$Nu_f = 0.452 \cdot Re_f^{0.495} \cdot Pr_f^{\frac{1}{3}} \cdot \left(\frac{\mu_b}{\mu_w} \right)^{-1.31} \quad (1)$$

where cooling water properties are evaluated at the film temperature and the bulk temperature as indicated in the correlation. This correlation fit all the experimental data with an $R^2 = 0,963$ and R.M.S. = 5,6%.

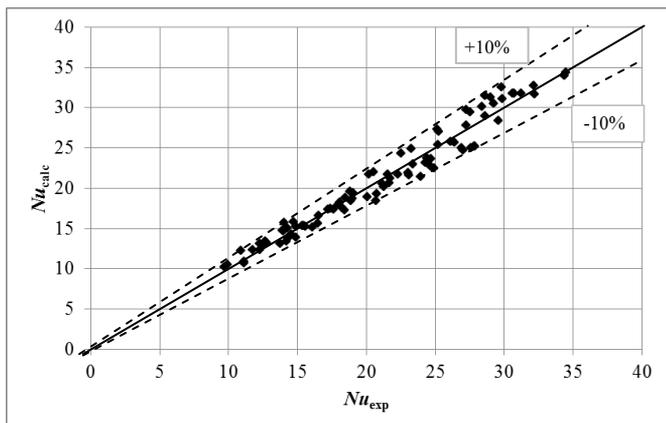


Figure 3 – Calculated vs experimental Nu in the cooling annulus

4. TESTS WITH PURE STEAM CONDENSATION

All the tests have been carried out in gravity controlled stratified flow regime, according to the Breber et al. [3] flow pattern map.

In pure steam condensation tests, once the steady state conditions were reached, the cooling water flow rate was adjusted to obtain a better estimation of the condensation length.

The condensation length was maintained as close as possible to some reference values. Thermocouples inside the stainless steel tube have a distance of 25 cm. When the steady state conditions were reached (before cooling water flow rate adjustment), the condensation length value had to be between two values corresponding to the location of the first internal thermocouple measuring a temperature lower than 100° C and the location of the preceding thermocouple. In this way the condensation length value was known, as a first approximation, with an uncertainty of 25 cm without performing any calculation. To reduce uncertainty, the cooling flow rate was adjusted until the thermocouple indicating 100° C changed its value or the other thermocouples reached the value of 100° C, to identify the full condensation limit as close as possible to a thermocouple. Obviously the cooling water flow rate was increased or decreased according to several parameters: temperature value of the thermocouple, power level, cooling water flow rate, and test section inclination. Condensation lengths were controlled to be as large as possible, to minimize the error correlated with their measurement, at the same time the cooling water flow rates were adjusted as low as possible in order to have a wide temperature difference between the test section cooling water inlet and outlet.

A too low cooling water flow rate (and as a consequence a large condensation length and a high temperature difference) yielded to a non-uniform cooling water flow rate, air formation and stagnation, and hence a unrealistic measurement of the refrigerant temperatures along the test section. Also the power level and the test section inclination affect the condensation length value.

Tests with pure vapor were carried out at the power level (i.e. vapor flow rate) of 100% (with a nominal power of 6 kW), 80%, 70%, 60% and 50%.

Tests were also performed with several cooling water flow rates with the test section inclination of 0°, 15°, 30° and 45°.

Mean values and standard deviations of the collected data were evaluated. The mean temperature values were then corrected with the calibration factors (evaluated for each thermocouple with preliminary tests) and the cooling water temperature profiles in different circumferential and axial locations) were obtained fitting the average experimental values. The cooling water flow rate was not uniformly distributed around the annulus, because of gravity force, annulus non-uniformity and eventual air stagnation, and the flow rates in the upper and in the lower side of the annulus were evaluated using temperature functions, temperature measured values and thermal balance equation on the test section. Once the cooling water flow rates in the two parts of the test section were evaluated, the coolant average temperature around the annulus could be calculated in each axial location.

Table 1 summarizes the main test conditions with pure steam.

Table 1 - Tests with pure steam

N° of tests	Inclination between tube and horizontal	Steam mass flux [kg/m²s]
26	0°	1.2-5.45
9	5°	2.34-3.36
30	15°	0.85-5.48
14	30°	2.34-5.37
9	45°	2.34-3.35

4.1 Data reduction

The overall test section was divided into two parts: the condensation region and the subcooling region. Condensation and subcooling heat transfer rates were evaluated through steam flow rate and compared with values obtained from condensate measurements.

The total exchanged power in the tube was evaluated and its values were compared with the coolant heat transfer rate. Only tests where the differences between these methods were lower than 8% were accepted.

The coolant temperature average value corresponding to the condensation limit and subcooling region was evaluated using the subcooling power and coolant inlet temperature to the test section.

$$T_{cw,j} = \frac{P_{sub,c}}{\Gamma_{cw} \cdot C_{p,cw}} + T_{cw}^{in}$$

The evaluation of the condensation length was obtained using condensate mass measurements and integrating coolant temperature profile until the length L_{cond} provided power balance between the condensation power and the coolant power in the condensation region.

$$P_{cond,c} = \Gamma_v \cdot \lambda = \int_0^{L_{cond}} \Gamma_{cw} \cdot C_{p,cw} \cdot \frac{dT}{dx} dx$$

where $T(x)$ is the average annular temperature profile evaluated fitting the temperature data.

Figure 4 shows the condensation length as a function of steam flow rate, for a fixed coolant flow rate.

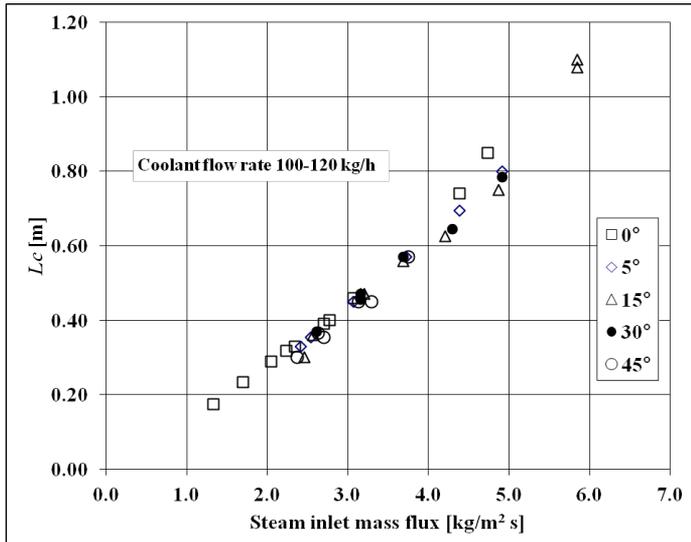


Figure 4 – Condensation length in the tube

Once the condensation length was evaluated, the average heat flux in the condensation and subcooling regions could be calculated:

$$q_{cond}'' = \frac{P_{cond,c}}{S_{cond}} \quad q_{sub}'' = \frac{P_{sub,c}}{S_{sub}}$$

with reference to the external surface.

The value of the overall heat transfer coefficients in the condensation and subcooling regions were calculated using the evaluated local heat flux and the logarithmic mean temperature difference (LMTD) in counter-current-flow:

$$LMTD = \frac{(T_v^{in} - T_{cw}^{out}) - (T_v^{out} - T_{cw}^{in})}{\ln\left(\frac{T_v^{in} - T_{cw}^{out}}{T_v^{out} - T_{cw}^{in}}\right)}$$

$$U_{e,c} = \frac{q_{cond}''}{LMTD_c} \quad U_{e,s} = \frac{q_{sub}''}{LMTD_s}$$

In the geometrical condition presented by the test section the overall heat transfer coefficient can be expressed through the relationship:

$$U_e = \frac{1}{\left(\frac{1}{h_i} \cdot \left(\frac{r_e}{r_i}\right) + \frac{r_e \cdot \ln(r_e/r_i)}{k_{acc}} + \frac{1}{h_e}\right)}$$

To evaluate the internal heat transfer coefficient h_i in the two regions, the cooling water heat transfer coefficient h_e in the has to be preliminarily calculated, using eq (1), where the wall and film temperatures (to evaluate the condensate film properties) are evaluated through an iterative procedure.

Once the external heat transfer coefficient in the two regions was evaluated, the internal heat transfer could be evaluated using the relation:

$$h_i = \frac{1}{\left(\frac{1}{U_e} - \frac{r_e \cdot \ln(r_e/r_i)}{k_{acc}} - \frac{1}{h_e}\right)} \cdot \frac{r_i}{r_e}$$

4.1 Results for pure steam condensation

An effect of tube inclination has been observed in pure steam condensation tests. In Figure 5 the mean values for each inclinations are reported and the higher values are observed for the 45° inclination, the lower values for the horizontal position. Other inclinations (5°, 15° and 30°) show similar average values. As expected, the condensate film thickness affects strongly the heat transfer coefficient.

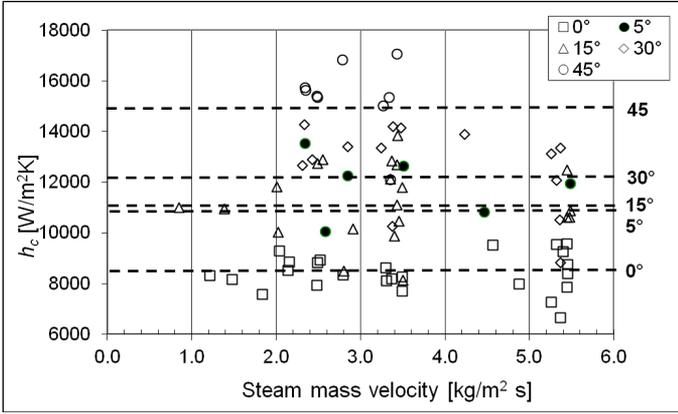


Figure 5 – Pure steam condensation heat transfer coefficient at different inclinations.

During condensation within horizontal tubes, when the vapor velocity is very low, the flow will be dominated by gravitational forces, and stratification of the condensate will occur, as schematically shown in Figure 6. In this case, the condensate forms a thin film on the top portion of the tube walls and is drained around the periphery due to gravity, toward the bottom of the tube, where a layer of condensate collects and flows axially due to shear forces.

The results obtained experimentally have been compared with the condensation heat transfer coefficient calculated using some correlations available in literature. These correlations are essentially modified Nusselt correlations:

- Chato's correlation [4]:

$$Nu_c = \frac{d}{k_f} F \left[\frac{\rho_f (\rho_f - \rho_v) g \cdot \cos \theta \cdot \lambda' \cdot k_f^3}{d \mu_f (T_{vi} - T_w)} \right]^{\frac{1}{4}} \quad (2)$$

where F is reported in Table 2 as a function of ϕ (Figure 6). To evaluate the average heat transfer coefficient along the tube, a mean value $F = 0.557$, corresponding to an average angle $\phi = 60^\circ$ as suggested by Chato, was used.

Table 2 - F factor as a function of the angle ϕ , for a laminar, stratified flow inside a horizontal tube

ϕ	$F(\phi)$	ϕ	$F(\phi)$	ϕ	$F(\phi)$
0°	0.725	70°	0.517	130°	0.248
10°	0.712	80°	0.476	140°	0.199
20°	0.689	90°	0.433	150°	0.150
30°	0.661	100°	0.389	160°	0.100
40°	0.629	110°	0.343	170°	0.050
50°	0.594	120°	0.296	180°	0.000
60°	0.557				

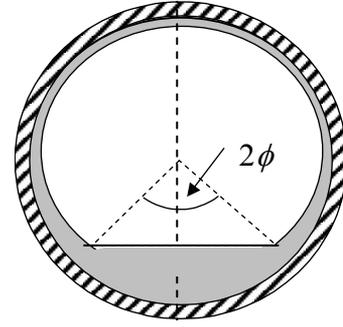


Figure 6 - Chato's model

- model inspired to the Butterworth and Owens' equations [2], simplified for very low flow rates:

$$Nu_c = \left(1 - \frac{\phi}{\pi}\right) \cdot Nu_c^{up} + \frac{\phi}{\pi} \cdot Nu_c^{down} \quad (3)$$

where:

Nu_c^{up} is the same as in the Chato's model, and

$$Nu_c^{down} = 0.024 \cdot Re_f^{0.8} Pr_f^{0.45} \cdot \psi(\rho) \quad (4)$$

where the liquid Reynolds number Re_f is evaluated referring to a liquid flowing with the same total flow rate in the tube and the function $\psi(\rho)$ is locally evaluated, following the analogy between liquid film flow and single phase flow in a pipe [6], as:

$$\psi(\rho) = \left(\frac{\rho_f}{\bar{\rho}}\right)^{0.5} \quad (5)$$

in which $\bar{\rho}$ is the homogeneous mean density of the vapor-liquid mixture.

The average value of the function $\psi(\rho)$ along the tube is:

$$\bar{\psi}(\rho) = 0.5 \cdot \left(1 + \left(\frac{\rho_f}{\rho_v}\right)^{0.5}\right)$$

The angle ϕ can be obtained with the following relations:

$$1 - \frac{\phi}{\pi} = 0.27 \cdot Re_v^{0.1} \quad \text{if } Re_v^{0.6} \cdot Re_f^{0.5} \leq 6.4 \cdot 10^{-5} \cdot Ga$$

$$1 - \frac{\phi}{\pi} = \frac{1.74 \cdot 10^{-5} \cdot Ga}{(Re_v \cdot Re_f)^{0.5}} \quad \text{if } Re_v^{0.6} \cdot Re_f^{0.5} \geq 6.4 \cdot 10^{-5} \cdot Ga$$

where the Galileo number is defined as:

$$Ga = \frac{d^3 \cdot \rho_l \cdot (\rho_f - \rho_v) \cdot g}{\mu_f^2}$$

For inclined tubes, with an angle θ with respect to the horizontal, a modified Nusselt correlation has been also tested for the mean value of the heat transfer coefficient:

$$\bar{Nu}_c = 1.2 \cdot d \cdot \left[\frac{\rho_f (\rho_f - \rho_v) g}{\mu_f^2 (Re_f)} \right]^{\frac{1}{3}} \quad (6)$$

where $Re_f = 2 \frac{W_f}{\mu_f \cdot L} \cos(\theta)$, applicable if the

following relation is satisfied: $\frac{L}{d} \geq 1.8 \cdot \tan(\theta)$.

The comparison between the experimentally obtained condensation heat transfer coefficient and the results obtained through correlation use, shows that calculated values over predict the values obtained for inclinations $\theta=0^\circ, 5^\circ, 15^\circ$ and 30° , while there is a good agreement between the experimental and calculated values with an inclination of 45° . The reason is related to the correlation structure used to calculate the condensation coefficient with inclined tubes. In fact this correlation has been proposed to calculate the average heat transfer coefficient on the external surface of a round horizontal or inclined tube, where condensate produced does not accumulate, but it is drained out due to gravity forces. Inside the tubes, condensate is collected in the tube bottom region itself, and it reduces the condensation heat transfer coefficient averaged on the whole surface. The correlation cannot account for the condensate layer, so it overpredicts the condensation heat transfer coefficient inside the tube. On the other hand, with increasing tube inclination, the condensate collected in the bottom region is drained out, and the calculated condensation heat transfer coefficient using the previous correlation and the experimentally obtained condensation heat transfer coefficient become similar, hence the greater the inclination is, the greater the agreement between condensation heat transfer coefficients are. For an inclination $\theta = 45^\circ$, experimental and theoretical condensation heat transfer coefficients assume the same values.

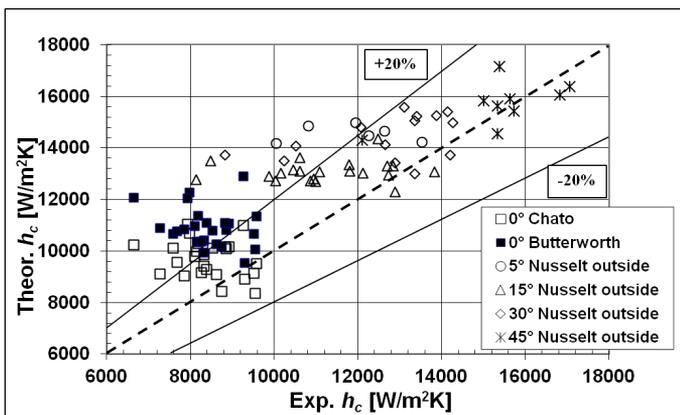


Figure 7 - Original correlations (Chato, Butterworth, Nusselt) Vs Experimental h_c (inclination as parameter)

For $\theta = 5^\circ, 15^\circ$ and 30° inclination, the Nusselt correlation has been modified as follows:

$$Nu_c = 1.2 \cdot d \cdot \left[\frac{\rho_f (\rho_f - \rho_v) g}{\mu_f^2 (Re_f)^{1.143}} \right]^{\frac{1}{3}} \quad (7)$$

to obtain a good agreement with the experimental results (Figure 8)

Chato's correlation has also been modified introducing a factor 0.837, and a better agreement with the experimental data was obtained (see Figure 8).

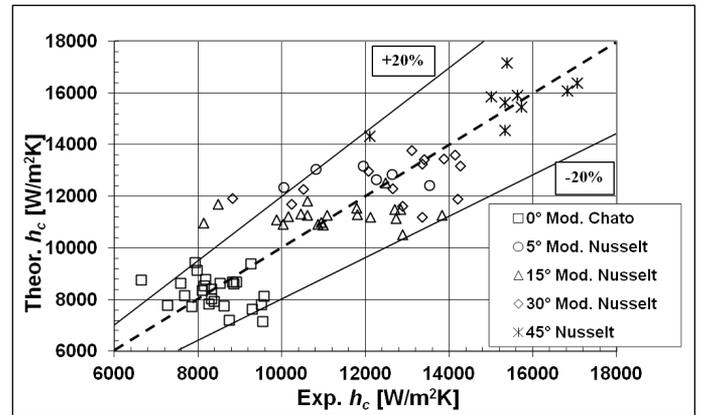


Figure 8 - Modified correlations (Chato, and Nusselt) Vs Experimental h_c (inclination as parameter)

Tests with a different test section inclination indicate that the condensation heat transfer coefficient increases with the inclination. The condensation heat transfer coefficient increase due to the tube inclination was well predicted through the correlations available. In fact, with increasing tube inclination, the condensate collected at the tube bottom increases its velocity, enhancing the average condensation heat transfer coefficient.

5. TESTS WITH NON-CONDENSABLE GASES

The aim of vapor-air mixture tests was to evaluate the condensation mean heat transfer coefficient with non-condensable gases in the mixture. Cooling water flow rates were selected to minimize refrigerant stratification inside the annulus and at the same time to obtain a consistent condensation surface.

Inlet air-vapor composition was obtained with several combinations of air and steam flow rates. Table 3 summarizes the tests performed.

Data reduction has been carried out as in pure steam tests.

Table 3 - Tests with non-condensable gas

N° of tests	Inclin. [°]	Gas concentration [%]	Steam mass flow [kg/m²s]
14	0°	1.8-46.9	2.69-3.85
10	5°	3.3-25.1	3.25-4.8
33	15°	1.5-71.5	1.61-4.47
8	30°	2.0-50.7	2.24-3.44
7	45°	1.7-25.1	2.46-3.4

5.1 Results for the air-steam mixture condensation

Tests with vapor-air mixtures show that non-condensable gases have a strong influence on the condensation heat transfer coefficient. With the same conditions (test section inclination, power level and cooling flow rate), the higher air percentage is, the lower the condensation heat transfer coefficient becomes. The influence of non-condensable gases in the mixture on condensation heat transfer coefficient is always very effective at low non-condensable gas concentration in the mixture, while as non-condensable gas percentage increases, the effect on condensation heat transfer

coefficient becomes less important in all experimental conditions.

The condensation heat transfer coefficient decreases to 50% of its original values when the air percentage in the mixture changes from 0% to 2%, while with an air percentage of 10% the condensation heat transfer coefficient is reduced to 10% of the condensation heat coefficient with pure steam.

Increasing the air percentage, the condensation heat transfer coefficient decreases again, but to reduce it with another magnitude order (1% of the pure vapor condensation coefficient), the mixture air percentage has to reach the value of 50%. By further increasing the air mixture concentration the heat transfer coefficient decreases slowly.

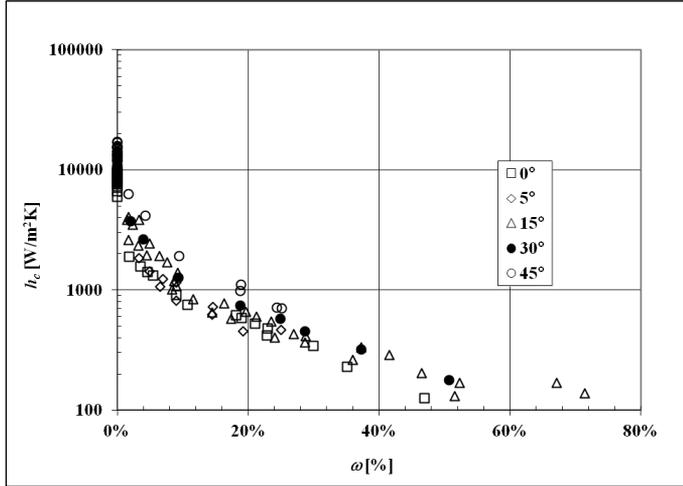


Figure 9 Heat transfer coefficient Vs ω [%] (inclination as parameter)

Tests with different inclinations show that condensation heat transfer coefficients increase with inclination of the test section with low air mixture concentration (Figure 9). This effect has not been observed at higher gas concentrations.

As expected, increasing non-condensable percentage inside the steam-air mixture, the thermal resistance of condensate film becomes negligible in comparison with the thermal resistance of the steam-gas mixture and hence the test section inclination has no appreciable effect on the condensation heat transfer.

In Figure 10, the complete experimental data set is reported, as the ratio of the air-steam heat transfer coefficient and the value for pure steam condensation. A simple fitting of data was attempted and in the figure, the following regression curve is also shown, valid for $\omega_{in} > 0.01$:

$$\frac{\bar{Nu}_c}{\bar{Nu}_{c0}} = 0.0118 \cdot \omega_{in}^{-0.86} \quad (8)$$

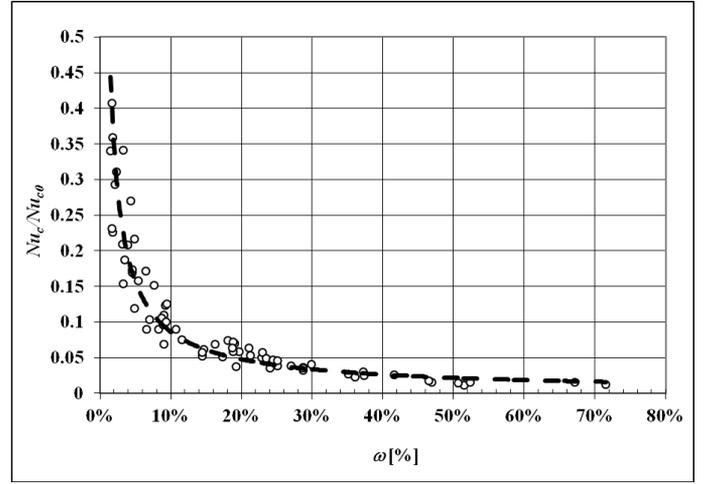


Figure 10 - Average Nu_c/Nu_{c0} Ratio Vs inlet non-condensable gases content

In the past, the most often used semi-empirical correlations to evaluate the condensation heat transfer coefficient in presence of high gas concentration (in particular, in containment thermal hydraulic analyses) and derived from the experimental database of Uchida [7], Tagami [8] and Kataoka [9], were the following:

$$\bar{h}_{Uchida} = 380 \cdot \left(\frac{\omega}{1-\omega} \right)^{-0.7} \quad (9)$$

$$\bar{h}_{Tagami} = 11.4 + 284 \cdot \frac{\omega}{1-\omega} \quad (10)$$

$$\bar{h}_{Kataoka} = 430 \cdot \left(\frac{\omega}{1-\omega} \right)^{-0.8} \quad (11)$$

Experimental data for condensation inside horizontal and inclined tubes were used to obtain a similar correlation, as the mixture flow rates in the experiments are quite low and gravity controlled conditions occurred. For all the experimental points, at all the inclinations used for the tube, the following empirical correlation to evaluate the average heat transfer coefficient along the tube was obtained:

$$\bar{h}_C = 209.3 \cdot \left(\frac{\omega_{in}}{1-\omega_{in}} \right)^{-0.725} \quad (12)$$

Figure 11 shows a comparison between the above mentioned correlations eqs. (9),(10),(11) and (12).

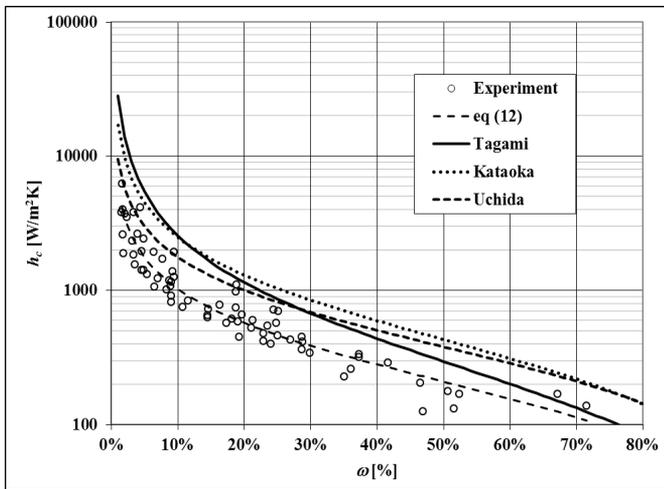


Figure 11 - Comparison of average HTC correlations in the presence of non-condensable gases

CONCLUSIONS

Pure steam and steam-air mixture condensation experiments were carried out in gravity controlled stratified flow regime inside an horizontal and inclined tube (22 mm inside diameter) and the average heat transfer coefficient was evaluated. For pure steam condensation, data was compared with literature correlation predictions (Chato and modified Nusselt correlations) and their agreement was verified, suggesting some minor modifications. An influence of tube inclination on heat transfer was observed. Inclination of the tube seems to have no influence on condensation in the presence of non-condensable gases, especially at high gas concentrations. Two simple empirical correlations have been obtained and could be used in preliminary design or system simulation in transient conditions of a condenser in a passive containment cooling system.

NOMENCLATURE

C_p specific heat at constant pressure	[J/(kg K)]
D, D_h diameter, hydraulic	[m]
h heat transfer coefficient	[W/(m ² K)]
k thermal conductivity	[W/(m K)]
L length	[m]
q'' heat flux	[W/m ²]
R universal gas constant	[J/(mol K)]
T temperature	[K]

Greek letters

Θ tube inclination with respect to horizontal	
λ latent heat	[J/kg]
λ' mod. latent heat $\lambda' = \lambda \left[1 + 0.68 \left(\frac{c_{p,f}(T_{vi} - T_{wi})}{\lambda} \right) \right]$	[J/kg]
μ dynamic viscosity	[kg/(m s)]
ρ density	[kg/m ³]
ϕ angle of the liquid region at the bottom of the tube	
ψ function of density	

ω mass fraction

Subscripts

0	reference state
c	condensation
e	external
exp	experimental
g	gas phase; gas mixture
i	internal
in	inlet
ms, sub	subcooled; sensible heat transfer
w	tube wall; water
v	vapour, steam

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