ABSTRACT

An analytical study of condensation flows in the presence of non-condensable gases in a large containment vessel has been performed. Analysis of the experimental results is performed using extensive local measurements. Local total heat transfer coefficients and condensation mass-flow rates are determined using different approaches: coolant energy balance, degradation factors, heat and mass transfer analogy. For the two last approaches, different references or methods can be found in the literature and are analyzed in this paper. The first objective of this paper is to check if the condensation models generally used for analytical/laboratory experiments can be applied in a larger vessel. Experiments have been done in the TOSQAN facility and it can be seen that the condensation models used here are also valid for TOSQAN. The second objective of this paper is to evaluate the influence of different parameters used in the condensation models: the way of calculating the bulk mean temperature can have an influence for stratified tests; the influence of the compressibility effect is also non-negligible if tests are performed at higher pressure (4-5 bar); and the expression of the diffusion coefficient, which is a linear factor of the mass transfer rate, can have a significant role.

KEYWORDS

Condensation, TOSQAN, filmwise, containment, hydrogen risk.

1. INTRODUCTION

During the course of a severe accident in a Pressurized Water Reactor (PWR), large amounts of hydrogen could be generated and released into the containment building. The integrity of the containment could be challenged by hydrogen combustion (so-called hydrogen risk). As boundaries between the different combustion modes are characterized by narrow gas concentration bandwidths, gas distribution calculations are needed to generate the initial conditions of a subsequent combustion process. The TOSQAN project has been created to simulate these accidental thermal-hydraulic flow conditions and to study phenomena such as wall condensation in the presence of non-condensable gases. The latter phenomenon is a subject of many interests and extensive studies have been performed in the past (Dehbi, 1991, Huthiniemi and Corradini, 1993, Park et al., 1997, Peterson and Schrock, 1993; Anderson 1998 and 1998b). Some empirical correlations have demonstrated their limit for extrapolation under different thermal-hydraulic conditions and at different geometries/scales (Peterson, 1996). Furthermore, previous studies have been done either in simple configurations (horizontal/vertical plates, at smaller scale, under well-controlled flows or saturation) or at very large scale with low instrumentation density and poor flow control. The idea of the TOSQAN project is to perform an intermediate study between separate-effect tests and integral tests, considering three

1 Corresponding Author
phenomena (plume/jet flows, condensation on walls and natural convection), with a high density of instrumentation.

The first objective of this paper is to check if the models generally used for analytical/laboratory experiments can be applied in a larger vessel like the TOSQAN one. The second objective of this paper is to evaluate the influence of different parameters used in the condensation modelling. This study could help to evaluate the limit of condensation modelling used in safety codes. For these purposes, an analytical analysis of the experimental results is performed using extensive local measurements; local total heat transfer coefficients are determined using different approaches: coolant energy balance, degradation factors, heat and mass transfer analogy. For the two last approaches, different references or uses can be found in the literature and are analyzed in this paper.

2. DESCRIPTION OF THE TOSQAN FACILITY AND THE CONDENSATION TESTS

The TOSQAN experiment (Figure 1) is a closed cylindrical vessel (7 m$^3$, i.d. 1.5 m, total height of 4.8 m, condensing height of 2 m) into which steam or non-condensable gases are injected through a vertical pipe located on the vessel axis. This vessel has thermostatically controlled walls so that steam condensation may occur on one part of the wall (the condensing wall, CW), the other part being superheated (the non-condensing wall, NCW). Over 150 thermocouples are located in the vessel (in the main flow and near the walls). 54 sampling points for mass spectrometry are used for steam volume fraction measurements (Auban et al., 2003). Optical accesses are provided by 14 overpressure resistant viewing windows permitting non-intrusive optical measurements along an enclosure diameter at 4 different levels (LDV and PIV for the gas velocities, Raman spectrometry for steam volume fractions, Porcheron et al., 2002). Measurements are mainly located at the different positions presented in Figure 1. Temperature measurements used for the mean temperature calculations are located on the horizontal rows of 6 thermocouples presented on Figure 1 (thermocouples in the “injection zone” are kept from this mean value). More details on the calculation of this mean value will be given later in the paper.

Condensation tests presented here consist of a water steam injection into the enclosure that is initially filled with air at atmospheric pressure, the NCW and the CW having already reached their nominal temperature. After a transient stage corresponding to enclosure pressurization, a steady-state is reached when the water steam injection and the condensation flow rates are equal. This corresponds to constant enclosure total pressure and thermal equilibrium.

Nominal tests specifications are given in Table 1. These tests are performed by changing the injection mass-flow rate $Q_{\text{inj}}$, the NCW and CW temperatures $T_{\text{NCW}}$ and $T_{\text{CW}}$, and the air partial pressure. As a result, they have different gas mixture compositions, final pressures and gas superheating. This
thermodynamical conditions corresponds to the range of thermodynamical conditions that can occur in the reactor containment during a hypothetical accident. In this paper, tests ISP-P1, ISP-P2, ISP-P3 stand for different sequences of the International Standard Problem ISP-47 TOSQAN test (respectively for sequences S2, S4 and S6). Detailed air-steam mixture results of the ISP-47 test can be found in Malet et al., 2002. The TOSQAN facility is mainly used for the validation of ‘elementary’ phenomena (condensation, spray systems, sump-atmosphere interaction), so that it is not designed to represent a real reactor at a smaller scale. For example, the Grashof numbers \(10^9\) \(-10^{10}\) applied on the TOSQAN condensation walls are much lower than the one in the real reactor \(10^{15}\).

<table>
<thead>
<tr>
<th>Test n°</th>
<th>(Q_{inj}) (g/s)</th>
<th>(T_{NCW} (°C)) higher part of TOSQAN***</th>
<th>(T_{NCW} (°C)) lower part of TOSQAN***</th>
<th>(T_{CW} (°C))***</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>12.25 +/- 0.25</td>
<td>121.9</td>
<td>122.8</td>
<td>111.2</td>
</tr>
<tr>
<td>ISP-P2*</td>
<td>12.27 +/- 0.12</td>
<td>122.0</td>
<td>123.5</td>
<td>111.4** (107.8)</td>
</tr>
<tr>
<td>2</td>
<td>11.70 +/- 1.20</td>
<td>107.3</td>
<td>108.2</td>
<td>97.0</td>
</tr>
<tr>
<td>3</td>
<td>11.92 +/- 0.21</td>
<td>152.2</td>
<td>154.5</td>
<td>148.8</td>
</tr>
<tr>
<td>6</td>
<td>1.05 +/- 0.10</td>
<td>124.0</td>
<td>125.5</td>
<td>117.7</td>
</tr>
<tr>
<td>ISP-P1 and ISP-P3*</td>
<td>1.11 +/- 0.10</td>
<td>122.0</td>
<td>123.5</td>
<td>102.0** (101.8)</td>
</tr>
<tr>
<td>7</td>
<td>0.96 +/- 0.19</td>
<td>61.7</td>
<td>58.4</td>
<td>52.5</td>
</tr>
<tr>
<td>8</td>
<td>0.75 +/- 0.11</td>
<td>145.0</td>
<td>145.6</td>
<td>137.7</td>
</tr>
<tr>
<td>9b</td>
<td>0.95 +/- 0.16</td>
<td>87.1</td>
<td>89.5</td>
<td>53.1</td>
</tr>
</tbody>
</table>

*: test with 1.3 bar of air (instead of 1 bar of air initially in all other tests)
**: the CW temperatures given for the ISP-test were some corrected values (in parenthesis here) of this temperature, whereas the real measured ones are specified here
***: uncertainty on temperature measurements is +/-0.5°C

Table 1: Nominal condensation tests matrix in TOSQAN

Tests 6 to 9b are generally at the transition between jet and plume injection (low-momentum injection, Richardson number higher than 0.1) whereas tests 1 to 4 are turbulent jet tests (high-momentum injection).

### 3. PRESENTATION OF THE RESULTS

A general description of the main phenomena occurring during the steady-state of air-steam condensation tests is presented on Figure 2. The latter is obtained from numerical calculations using the TONUS 3D code (Malet et al., 2004). Four zones are identified: the injection zone, the recirculation zone (connected with the injection zone), the condensing region and the natural convective flow occurring below the injection. In this paper, characterization of the bulk flow and the condensing region will be performed from the experimental results.

Table 2 presents the range of typical experimental mean characteristics of the tests. A wide range of total pressure is investigated, allowing also to cover a large range of steam volume fraction, humidity and gas temperature. If mean values are considered, the tests are superheated \(T_{m} - T_{sat} > 0\), \(T_{m}\) being the mean gas temperature, \(T_{sat}\) the saturation temperature at the total pressure. It does not mean that locally, a test cannot be saturated. Furthermore, the condensing wall temperature \(T_{cw}\) is below the saturation temperature \(T_{sat}\), some tests being just at the limit, indicating very low condensation rates. The Reynolds number at the steam injection \(Re_{inj}\) shows a turbulent flow at the injection (during the steady-state), the Richardson number at the injection \(Re_{inj}\) shows that some tests are just at the limit between a jet and a plume, whereas others are pure jets. The Prandtl number \(Pr_{in}\) of the mixture at the
steady-state shows that it is rather a mainly air flow ($Pr_m = 0.72$) or that the dynamic and thermal boundary layer are similar ($Pr_m = 1$). The Schmidt number $Sc_m$ is quite constant from one test to another and is always lower than the Prandtl number, indicating that the Lewis number $Le_m$ is always lower than 1: the boundary layer might be superheated. The Nusselt $Nu$ and the Sherwood $Sh$ numbers are characteristics for condensation in the presence of non condensable gases.

![Schematic view of the zones corresponding to the main phenomena occurring during air-steam condensation tests.](image)

<table>
<thead>
<tr>
<th>Variable</th>
<th>Range</th>
<th>Variable</th>
<th>Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure $P$ (bar)</td>
<td>1.17 – 5.78</td>
<td>$Re_{inj}$</td>
<td>1 000 – 30 000</td>
</tr>
<tr>
<td>Mean gas temperature $T_m(°C)$</td>
<td>60 – 153</td>
<td>$Re_{inj}$</td>
<td>0.001 – 1</td>
</tr>
<tr>
<td>Mean condensing wall temperature $T_{cw}(°C)$</td>
<td>52 – 149</td>
<td>$Pr_m$</td>
<td>0.72 – 1.07</td>
</tr>
<tr>
<td>Mean steam volume fraction $X_s$ (%)</td>
<td>25 – 85</td>
<td>$Sc_m$</td>
<td>0.44 – 0.63</td>
</tr>
<tr>
<td>Mean relative humidity RH (%)</td>
<td>60 – 100</td>
<td>$Ra_{th}$</td>
<td>$5\times 10^7$</td>
</tr>
<tr>
<td>$T_m - T_{cw}(°C)$</td>
<td>3 – 27</td>
<td>$Ra_{inj}$</td>
<td>$10^7 - 2.2x10^{10}$</td>
</tr>
<tr>
<td>$T_{sat} - T_{cw}(°C)$</td>
<td>1 – 12</td>
<td>$Nu$</td>
<td>220 – 410</td>
</tr>
<tr>
<td>$T_{sat} - T_{cw}(°C)$</td>
<td>0.1 – 22</td>
<td>$Sh$</td>
<td>40 – 440**</td>
</tr>
</tbody>
</table>

*depending on the relations for the diffusion coefficients (see later in the paper)

**depending on the relations for the Nusselt and Sherwood numbers (see later in the paper)

Table 2: Range of the experimental values of different characteristics of the bulk flow.

In Table 3 are presented some characteristics of the condensing region. It can be seen that the condensate film Reynolds number (obtained from the measured condensation mass-flow-rate, the condensing surface and the wall temperature) is low. This indicates that the liquid film is laminar for all the “low mass-flow rate tests”, and in the transition zone to a wavy film for the high mass flow rate tests.

The Nusselt theory (Nusselt, 1916) for laminar film is theoretically not applicable here (theory mainly developed for pure steam and with a zero bulk velocity). However, in order to evaluate the range of order of some film characteristics, it is applied here. The average condensate film thickness is calculated using the measured condensation mass-flow-rate, and is found to be between 50 and 200 µm; the heat transfer coefficient (HTC) of the liquid film is high ($10^{-3} - 10^{-4}$ W/m²/K) comparing to the mean HTC in the gas obtained generally for condensation in the presence of non condensable gases (in this study, it will be seen in the next sections that the global HTC is one order of magnitude lower). As a result, one order of magnitude differences between the Nusselt number in the gas and in the liquid film is also found in these tests. Corrections of Kutateladze (1963) or Nozhat (1995) are applied on the calculations of the liquid film HTC in order to take into account the waviness of the film and the impact of these corrections is found to be negligible (lower than 10%) considering that the liquid film heat transfer is so high that the resistance offered by the liquid film does not need to be known very precisely.

An evaluation of the water-gas interface temperature is performed using the total heat flux from the Uchida correlation (see later in this paper) and the liquid film HTC. The difference between the
interface temperature $T_i$ and the condensing wall temperature $T_{cw}$ is of maximum 0.4°C. In safety codes, the interface temperature is generally not calculated because of high computational time required for this purpose (iterative method on each time step for the determination of $T_i$) and it is taken equal to the wall temperature. Since condensing wall temperature is of major influence on the level of pressure in the containment, a small variation of this value can have an impact. However, this problem is also connected with the whole problematic of the determination of the wall temperature if conduction in the walls is considered. Since conduction in the walls and heat losses can be another source of error for the wall temperature, the calculation of the interface temperature can be considered as a problem of second order. However, it is emphasized here that the interface temperature is rigorously not negligible.

Evaluation of the convective flux and the radiative flux is also performed. The convective flux is determined from thermocouples situated near the condensing wall (see next section for more information on the convective flux). The radiative flux is estimated from the mean bulk and wall temperatures, using the classical Stephan-Boltzman equation and assuming a low emission of the stainless steel surface (0.05). The convective HTC is on the order of magnitude expected for tests with condensation in the presence of non condensable gases, i.e. one order of magnitude lower than the global HTC (see section 5). The convective flux is low and the radiative flux, which can be considered here at its “maximum value” (the real emissivity should not exceed 0.05) is one order of magnitude lower than the convective flux.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>Condensate film Reynold number (-)</td>
<td>2 - 40</td>
</tr>
<tr>
<td>Condensate film thickness (mm)</td>
<td>0.05 - 0.2</td>
</tr>
<tr>
<td>Condensate film heat transfer coefficient (W/m²/K)</td>
<td>$6 \times 10^3$ - $2 \times 10^4$</td>
</tr>
<tr>
<td>Condensate film Nusselt number (-)</td>
<td>$10^4$ - $5 \times 10^4$</td>
</tr>
<tr>
<td>$T_i$-$T_{cw}$ (°C)</td>
<td>0 - 0.4</td>
</tr>
<tr>
<td>Experimental convective heat transfer coefficient (W/m²/K)</td>
<td>2 - 14</td>
</tr>
<tr>
<td>Experimental convective flux (W/m²)</td>
<td>9 - 120</td>
</tr>
<tr>
<td>Experimental radiative flux (W/m²)</td>
<td>1 - 8</td>
</tr>
</tbody>
</table>

Table 3: Range of the experimental values of different characteristics of the condensing zone.

In this section, it has been seen that several phenomena of secondary importance have been studied on the basis of experimental results and allow us to make some assumptions in the modelling of non condensable gases in the TOSQAN tests. However, it is emphasized that these usually negligible phenomena/parameters could all together lead to a more significant influence in some specific cases. Modelling should be performed having in mind these considerations. Analysis of the calculations from safety codes should be completed by an evaluation of these parameters/phenomena performed independently for each study. Guidelines for safety codes, indicating the limit of the code/modelling in given situations should be another way for a better analysis of the concerned phenomena.

### 4. CONVECTIVE HEAT TRANSFER COEFFICIENT

The objective of this section is to calculate a convective heat transfer coefficient HTC in order to verify if this value is, as expected in condensation studies, low compared to the global heat transfer coefficient. Furthermore, in multi-dimensional calculations which use heat and mass transfer analogy for the condensation modeling, applied on the first mesh cell close to the walls, this convective HTC can play a significant role on the bulk temperature (Malet 2004b).

The experimental convective heat flux $\Phi_{\text{conv-exp}}$ is approximated from the Fourier law used for conduction heat flux (it is thus assumed that the gas layer adjacent to the surface has no velocity) :

$$\Phi_{\text{conv-exp}} = \lambda \left( \frac{dT}{dr} \right)_{r=0}$$  \hspace{1cm} (1)
where $\lambda$ is the thermal conductivity of the mixture near the wall, and $r$ the normal distance to the wall ($r = 0$ at the wall). The temperature gradient is evaluated using a temperature profile near the condensing wall: this temperature profile is composed of 4 thermocouples where 3 of them are outer the boundary layer (the value measured on these thermocouples are constant values and are equal to the bulk value); as a result, only two thermocouples are used for this gradient: the one near the wall and the one on the wall. The convective HTC $h_{\text{conv-exp}}$ is then obtained using the following relation:

$$h_{\text{conv-exp}} = \frac{\Phi_{\text{conv-exp}}}{T_m - T_{cw}}$$

(2)

This method is clearly not precise, but the objective here is to evaluate a range of order of the convective flux. Furthermore, it is the best that can be done using the available instrumentation of TOSQAN.

The ‘theoretical’ convective HTC $h_{\text{conv-theor}}$ is defined as follows:

$$h_{\text{conv-theor}} = \frac{Ntu_{\text{th}}}{L}$$

(3)

where $L$ is the length of the condensing wall (2 m) and the Nusselt number $Nu$ is given by the following empirical expression for turbulent flow over a flat plate (Mc Adams, 1954):

$$Nu = a \left(Gr_{th}Pr_{m}\right)^{b} \quad a=0.13, \ b=1/3$$

(4)

where the thermal Grashof number $Gr$ and the Prandtl number $Pr$ are determined at the ‘film’ temperature (mean value between bulk and wall temperature). The thermal Grashof number is given by the following expression:

$$Gr_{th} = g \cdot \frac{\left(T_{m} - T_{f}\right) L}{v_{th} \cdot \nu_{th}}$$

(5)

where subscript $f$ stands for the “film” temperature, $m$ for the bulk mixture, $i$ for the interface, $\nu$ is the kinematic viscosity, $g$ is the gravity. The kinematic viscosity of the mixture is determined from classical expressions for mixture viscosity.

The use of the McAdams correlations can be justified since the TOSQAN Rayleigh numbers are between $10^9$ to $10^{10}$ (turbulent flow). The surface of TOSQAN is vertical as supposed for the McAdams correlation. However, it is the aim of this work to see if such a correlation for ‘basic’ cases can be used in a facility like TOSQAN.

The Prandtl number is given by:

$$Pr_{m} = \frac{\mu_{m} C_{p_{m}}}{\lambda_{m}}$$

(6)

where $\mu_{m}$ is the dynamic viscosity and $C_{p_{m}}$ is the specific heat of the mixture.

It should be noticed also that $h_{\text{conv-exp}}$ is a HTC at a position $z$, whereas $h_{\text{conv-theor}}$ is an average value over the plate. A factor $5/4$ should be applied to the local Nusselt number to get the average Nusselt number. Furthermore, it should be noticed that this ‘theoretical’ value is also based on experimental measurements.

Comparison between the experimental convective HTC and the theoretical one for the TOSQAN tests is presented on Table 4. It can be seen that the values are in the range of order of the ones expected for convective flow for condensation tests in the presence of non condensable gases. It has to be emphasized also here that the TOSQAN facility is not a small-scale facility and that it is not devoted for such fine determination on thermal boundary layers. However, considering the more industrial character of this facility, the theoretical results can be considered to be in good agreement with the experimental ones, except for test 7 and test 3. It can be noticed that these tests are the two extremes in wall temperature. Further investigations have to be done to understand the results of these two tests.
The 11th International Topical Meeting on Nuclear Reactor Thermal-Hydraulics (NURETH-11)
Popes’ Palace Conference Center, Avignon, France, October 2-6, 2005.

Convective heat transfer coefficient (W/m²/K)

<table>
<thead>
<tr>
<th>Test</th>
<th>TOSQAN experiments</th>
<th>Heat transfer modelling</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>4.6 (5.8)</td>
<td>5.1</td>
</tr>
<tr>
<td>2</td>
<td>4.8 (6.0)</td>
<td>4.5</td>
</tr>
<tr>
<td>3</td>
<td>2.3 (2.9)</td>
<td>6.3</td>
</tr>
<tr>
<td>6</td>
<td>4.8 (6.0)</td>
<td>4.1</td>
</tr>
<tr>
<td>7</td>
<td>14.3 (17.9)</td>
<td>3.2</td>
</tr>
<tr>
<td>8</td>
<td>7.3 (9.1)</td>
<td>4.9</td>
</tr>
<tr>
<td>9b</td>
<td>4.4 (5.5)</td>
<td>5.1</td>
</tr>
<tr>
<td>ISP-P1</td>
<td>6.5 (8.1)</td>
<td>5.2</td>
</tr>
<tr>
<td>ISP-P2</td>
<td>5.4 (6.8)</td>
<td>5.5</td>
</tr>
<tr>
<td>ISP-P3</td>
<td>6.8 (8.5)</td>
<td>5.0</td>
</tr>
</tbody>
</table>

* : value in parenthesis are the mean values (5/4 * the local value)

Table 4: Comparison between the experimental convective HTC and the theoretical one for the TOSQAN tests.

5. GLOBAL HEAT AND MASS TRANSFER COEFFICIENT

The experimental global HTC is determined experimentally by a coolant energy balance. The “theoretical” global HTC is obtained from empirical correlations.

A Coolant Energy Balance (CEB) can be performed to determine a global heat transfer coefficient (HTC) and to compare it to the one obtained by the modelling. The heat transfer coefficient $h_{CEB}$ is then determined as follow:

$$h_{CEB} = \rho_o \cdot C_{p_o} \cdot \frac{Q_o \cdot \Delta T_o}{S_c \cdot (T_{in} - T_{cw})}$$  \hspace{1cm} (7)

where indices o stands for the oil (coolant used between the TOSQAN walls to keep the wall temperature at a given constant value), $\rho_o$, $C_p$ and $Q$ are the density, specific heat, volumetric flow-rate of the oil, $\Delta T$ is the difference of temperature between the inlet and outlet of the coolant circuit and $S_c$ is the total condensing surface.

Empirical correlations are also used to determine the global HTC $h$. The relations used here are the well-known Tagami (1965) and Uchida (1965) correlations, the Kataoka (1992) correlation and the Dehbi (1991) correlation. These correlations are very simple to be used but are ‘facility dependant’; furthermore, Peterson (1996) shows that the well-known Uchida correlation was accurate only in specific cases.

The global HTC ($h$) is given by empirical correlation of the following type:

$$h = a_1 + a_2 \cdot Z^n$$ \hspace{1cm} (8)

with

$$Z = \frac{\rho_s}{\rho_s + \rho_g}$$ \hspace{1cm} (9)

where $\rho_s$ and $\rho_g$ are respectively the steam and non condensable gas densities, and $a_1$, $a_2$ and $n$ are empirical constants, given in Table 5 for three authors.

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>$a_1$ (W/m²/K)</td>
<td>11.351</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>$a_2$ (W/m²/K)</td>
<td>283.77</td>
<td>379</td>
<td>430</td>
</tr>
<tr>
<td>n</td>
<td>1</td>
<td>0.707</td>
<td>0.8</td>
</tr>
</tbody>
</table>

Table 5: Empirical coefficients of the Uchida (1965), the Tagami (1965) and the Kataoka (1992) correlations.

Dehbi (1991) found the following correlation for the global HTC for air-steam mixtures:
where P is the total pressure (in atmosphere), and L the condensing length (in meter).

Results obtained from the Uchida correlation are presented on Figure 3. The error bars are given for an Interval of Confidence (IC) of 95%. The low mass flow-rate tests are the ones for which the HTC by CEB is lower than 100 W/m²/K. In fact, for these tests, the value of $\Delta T_o$ is lower than 0.5°C, which is also equal to the experimental error on temperature measurement. It is thus difficult to validate these results with this method. However, for the high mass flow rate tests (HTC of CEB around 300-800 W/m²/K, tests 1, 2 and 3), it can be seen that the Uchida correlation is quite in good agreement with the experimental results in a range of +/-20%. Comparison with other empirical correlations is given on Table 6 for high mass flow-rate tests. Considering the experimental uncertainty, the Uchida, the Tagami and the Kataoka correlations lead to similar results. The Dehbi correlation leads to higher values of the global HTC.

The impact of the presence of non-condensable gases is of course directly seen in the global HTC, which is for all tests much lower than expected for pure steam. Peterson (1996) shows that the partial pressure of air could play a role in the validity of the Uchida correlation. This was not observed here, but the partial pressure of air varies here over a small range: between 1 and 1.3 bar.

![Figure 3: Results obtained on the global heat transfer coefficients using the Uchida correlation.](image)

Table 6: Relative differences (%) between the global HTC obtained from CEB and the one calculated from empirical correlations.

<table>
<thead>
<tr>
<th></th>
<th>Tests 1 (5 repeated tests)</th>
<th>Test 2</th>
<th>Test 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Uchida</td>
<td>-28</td>
<td>-19</td>
<td>-36</td>
</tr>
<tr>
<td>Tagami</td>
<td>-8</td>
<td>3</td>
<td>-11</td>
</tr>
<tr>
<td>Kataoka</td>
<td>5</td>
<td>17</td>
<td>-2</td>
</tr>
<tr>
<td>Dehbi</td>
<td>75</td>
<td>92</td>
<td>63</td>
</tr>
</tbody>
</table>

Several investigations are conducted in order to determine if the data processing of the experimental results could lead to some discrepancies in the determination of the global HTC by empirical correlations.

The repeatability of the results has been checked on several tests. It is shown that the results are quite similar for two tests of the same type. The analysis here is mainly function of the steam and air densities; parameters that influence the calculation of these variables have been investigating. Here, in the data analysis, steam concentration is obtained using pressure and temperature measurements, assuming perfect gas law. However, if the compressibility factor for steam is used, small differences in
the steam partial pressure can be obtained, leading to a difference in the mass transfer rate. This difference is presented on Figure 4 as a function of the total pressure. It can be seen that the relative difference on the condensation mass flow rate due to steam compressibility effect is lower than 10% for total pressures below 3 bar, and can be up to 40% for higher pressures (5.8 bar). This point can be extrapolated in other situation than TOSQAN and should be kept in mind if analysis of high-pressure sequences is performed considering perfect gas law.

![Influence of the compressibility factor](image)

Figure 4: Relative difference between condensation mass flow rates calculated using perfect gas law or taking into account the steam compressibility effect, depending of the TOSQAN vessel total pressure.

Another parameter that has been investigated concerns the modelling of the condensation mass flow rate. In the present study, the following relation has been used to calculate the condensation mass flow-rate $Q_{\text{cond}}$ from the global HTC using the total heat flux $\Phi$ given above:

$$Q_{\text{cond}} = \frac{\Phi \, S_c}{H_{\text{lat}}} \cdot F_{\text{cond}}$$  \hspace{1cm} (11)

where

$$\Phi = h \left( T_m - T_{cw} \right)$$  \hspace{1cm} (12)

and where $H_{\text{lat}}$ is the latent heat and $F_{\text{cond}}$ is a correction factor for superheating (Dabenne 1998). The latter is given by the following relations:

$$F_{\text{cond}} = c \left[ a + (1-a) \exp^{b \left( T_m - T_{sat} \right)} \right] \quad \text{if} \quad T_{sat} \leq T_m \quad \text{and} \quad T_{cw} + 10 \leq T_{sat}$$

$$F_{\text{cond}} = c \left[ a + (1-a) \exp^{-b \left( T_m - T_{sat} \right)} \right] \left[ 1 - \exp^{-\left( T_{sat} - T_{cw} \right)^2} \right] \quad \text{if} \quad T_{sat} \leq T_m \quad \text{and} \quad T_{cw} \leq T_{sat} \leq T_{cw} + 10$$

$$F_{\text{cond}} = 1 \quad \text{if} \quad T_m \leq T_{sat}$$  \hspace{1cm} (13)

where $a = 0.92$, $b = 0.5$, $c=1$ and $T_{sat}$ is the saturation temperature.

In some cases, simplification is made and the correction factor for superheating is taken equal to 1. Figure 5 shows the results obtained if such simplification is made. It can be seen that for low mass transfer tests, this simplification leads to notable discrepancies.
Another parameter that can be discussed in data processing is the way of calculating the mean values, here the mean gas temperature. Since over 100 thermocouples are present inside the TOSQAN vessel, different ways to calculate the mean temperature can be considered: the arithmetic way (Tm1), the volume ponderation way (Tm2, which was the one adopted in the whole study), the arithmetic mean temperature in the whole region facing the condensing wall (Tm3) and the mean temperature obtained from the thermocouples placed near the condensing wall (close to the wall but out of the boundary layer), facing the condensing wall (Tm4). Results are presented on Figure 6. It can be seen that the way of calculating a mean temperature plays a role but do not change the results drastically. The reason for that is that the TOSQAN tests are not stratified, except test 9b, for which the influence of the mean temperature is clearly visible. This confirms the limit of such analysis under stratified conditions. It should be kept in mind in experimental analysis of stratified tests that the results can depend on the way the mean temperature is defined. Such conclusions can be extrapolated to lumped parameter codes, for which adequate meshing should be used in order to have compartments without physical stratification in the concerned zone.

The influence of the thermodynamic conditions of the test on the agreement between the experiment and the correlations has been also investigated. The relative difference between the global HTC
obtained by CEB and the one obtained by the Uchida correlation has been plotted versus several thermodynamic parameters. The same exercise has been done for the relative difference between the measured condensation mass flow-rate and the condensation mass flow-rate obtained from the global HTC calculated using the Uchida correlation. The concerned thermodynamic parameters are the following: superheating, relative humidity, steam volume fraction, total pressure, mean gas temperature, gas-wall temperature difference \((T_{m} - T_{cw})\), thermal Grashof number, mass Grashof number, saturation-wall temperature difference \((T_{sat} - T_{cw})\), thermal stratification. No real relation has been found between these parameters and the relative difference for all tests except test 9b. This test is the only one that is highly stratified test, with local gas saturation (homogeneous condensation), having the highest gas-wall temperature difference. It is the only test that really does not match the empirical correlations for the global HTC. This study allows us to conclude that other tests should be performed under similar conditions in order to study how to adapt these correlations to such conditions.

6. MASS TRANSFER MODELLING

This modelling leads to the determination of the condensation mass flow-rate, which is compared to the experimental value measured in TOSQAN. Several steps in the modelling have to be distinguished:

- mass transfer flux expressions: these expressions are obtained by modelling resulting from the diffusion Fick law;
- mass transfer rate relations: they are obtained from semi-empirical relations generally obtained in heat transfer studies and applied here assuming heat and mass transfer analogy;
- expression of several parameters depending on pressure and temperature: diffusion coefficient, saturation pressure, etc.

In order to evaluate the impact of the different relations at each step, it is chosen to first present a so-called “reference modelling” (chosen after a literature survey on modelling of condensation in the presence of non condensable gases) and the results obtained using this modelling, and then to evaluate the influence of other relations used in each step.

6.1 Reference modelling

Several mass-flux expressions exist in the literature, depending mainly on the way the Fick diffusion law is derived. A good and global description is given in Bird et al. (1960). Applied to the case of steam condensation in the presence of air, and considering perfect gas law, the following expression for the mass flux \(j_{mass}\) is given:

\[
j_{mass} = \frac{Q_{cond}}{S_{c}} = \rho_{m}K_{m} \frac{(X_{s-o} - X_{s-i})}{1 - X_{s-i}}
\]

(14)

where \(X\) is the molar mass fraction, indices \(m\) stands for mea, value in the bulk mixture, \(s\) for steam, \(o\) for bulk, \(i\) for interface (here the internal TOSQAN wall), and \(K_{m}\) is the mass transfer rate. The latter can be defined as follows:

\[
K_{m} = \frac{ShD_{m}}{L}
\]

(15)

where \(Sh\) is the Sherwood number, defined, by analogy to the convective flux, as a function of the Grashof \(Gr_{m}\) and the Schmidt number \(Sc\) (Mc Adams, 1954):

\[
Sh = a (Gr_{m}Sc)^{b}
\]

(16)

and \(D_{m}\) is the molecular diffusion coefficient, calculated, in our reference modelling, using the Wilke and Lee (1965) relations (in Poling et al. 2000). The Schmidt number is defined on the basis of the mixture properties. The Grashof number is defined as in Bird et al. 1960, as a function of molar fraction differences. It is emphasized that several ways to write the Grashof number can be found in
the literature (on the basis of molar fractions or densities, or as the sum of mass and thermal Grashof numbers). This point is skept in this paper because the small differences on the calculated Grashof numbers would not lead to high variations in the mass transfer rate $K_m$: the latter is function of the Grashof number at a power value lower than 1. All the mixture properties are defined at the so-called film temperature, i.e. the mean value between wall and bulk temperature. Perfect gas law is applied for the data reduction from temperature and pressure measurements.

Results obtained on the TOSQAN tests using this reference modelling are given on Figure 7. The error bars are given for an interval of confidence IC of 67% (95% IC would not have been readable on this figure). Tests 1, reproduced several times to check the repeatability, are in a range of error of +/-20% regarding to the experimental data. Test 2 is also in this range if an interval of confidence of 95% is considered here. Test 3, 6, 7, 8 and the ISP-47 tests are also in a range of +/-20% of the experimental values, but it has to be noticed that their relative uncertainty is quite high. The reason for that is that the difference of molar fractions in the expression of the mass flux is low regarding to the uncertainty of the molar fractions themselves. However, it can be seen that this reference modelling leads to quite good results, except for test 9b, as it was found for the Uchida correlation, that remains the only test really out of the range of the modelling. Discussion on this test will be performed later in this paper.

![CHILTON/BIRD Modelling](image)

Figure 7: reference modelling for mass transfer modelling used for the analysis of the TOSQAN tests.

### 6.2 Expressions of the mass flux

Two other expressions of the mass flux are often used. The first one (called here COLLIER/STEPHAN) is valid for low mass transfer rate. It is described in two reference books on heat and mass transfer, specifically on condensation in the presence of non condensable gases (Collier, 1994, Stephan, 1992). It is given by the following relation:

$$j_{mass} = \frac{Q_{cond}}{S_c} = K_m \rho m \ln \left( \frac{1-X_{t+}}{1-X_{t-}} \right)$$

(17)

The second one (called here CHILTON/BIRD/HMT) can be applied for high mass transfer rates. It is derived from the CHILTON/BIRD relation (see section 6.1) and the complete mathematical derivation is given in Bird et al. (1960). The following correction is then applied to the mass transfer rate:

$$\dot{K}_m = \Theta K_m$$

(18)

where $\Theta$ is a correction factor given by:
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Popes’ Palace Conference Center, Avignon, France, October 2-6, 2005.

\[ \Theta = \frac{\ln(R+1)}{R} \] (19)

and \( R \) can be written as follow for air-steam mixture:

\[ R = \frac{X_{i-1} - X_{i-o}}{1 - X_{i-1}} \] (20)

What is called ‘high mass transfer rate’ is when a distortion of the velocity and the concentration profiles near the walls is observed, due to the mass transfer involved in the phase change. In this case, there is a kind of ‘suction’ effect of the dynamic and mass boundary layer. This is also referred as the DuFour effect or the analogy defect.

Results obtained on the TOSQAN tests using these expressions are given on Figure 8. It is not the goal of this study to say that one expression is better than another one. The objective here is more to evaluate the sensitivity to the modelling of the mass flux expressions. It can be seen that the High Mass Transfer rate modelling (CHILTON/BIRD/HMT) leads as expected to a higher condensation mass flow-rate: the variation is of maximum 15% on the considered tests. The COLLIER/STEPHAN model is mainly around 40% lower than the CHILTON/BIRD model. However, since it underpredicts the condensation mass flow-rate, it overestimates the pressure and is conservative for safety codes. It has also to be emphasized that a variation of +/-20% on the condensation mass flow-rate leads to a variation less than +/-0.05 bar of the total pressure, for the considered tests.

![Figure 8: Comparison of condensation mass flow-rate determined experimentally with the one determined by mass transfer modelling using different rigorous expressions of the mass flux.](image)

Furthermore, since these expressions of the mass flux can be re-written in term of partial pressure or mass fraction \( Y \), assuming perfect gas law, three other ways to develop them can be found in some publications (Hadida 1997, TONUS 2003):

\[ j_{\text{mass}}^i = \rho_i K_i \frac{(Y_{i-o} - Y_{i-1})}{1 - Y_{i-1}} \quad (\text{MASS}_i) \] (21)

\[ j_{\text{mass}}^m = \rho_m K_m \frac{(Y_{i-o} - Y_{i-1})}{1 - Y_{i-1}} \quad (\text{MASS}_m) \] (22)

\[ j_{\text{mass}}^{m_o} = \rho_{m_o} K_{m_o} (Y_{i-o} - Y_{i-1}) \quad (\text{MASS}_{m_o}) \] (23)

These three expressions are plotted on Figure 9 on the basis of the TOSQAN tests. Expressions Mass_i is very close to the CHILTON/BIRD modelling, the differences are around 7% to 35% lower than the reference modelling. MASS_m about 35% to 75% lower, and Mass_m_o 35% to 90%.
Figure 9: Comparison of condensation mass flow-rate determined experimentally with the one determined by mass transfer modelling using different expressions of the mass flux.

Other expressions of the mass flux, quite recent and based on an interesting modelling approach, can be found in Peterson and Schrock (1993) and Herranz et al. (1998). However, it was not considered in this paper but the authors think it would be necessary to add their model in a next future.

As it will be explained in the next sections, a mass flux expression that under-estimates the condensation mass flow-rate can be counter-balanced by an expression on the Sherwood number or by other expressions of different gas properties (like the diffusion coefficient). That is why, when condensation modelings of safety codes are described, it is important to describe not only the mass flux modelling but also the correlations and the relations used for the thermodynamic properties.

6.3 Relations for the mass transfer rate

The mass transfer rate is deduced from the Sherwood number, which can be expressed using several relations. Since the mass Rayleigh number of the TOSQAN tests is found to be between $10^7$-2.2 $10^{10}$, just at the transition between laminar and turbulent flow, correlations for both flows are considered here. Table 7 in appendix presents the relations used for turbulent flow and Table 8 for laminar flow. Results obtained on the TOSQAN tests using these relations are presented on Figure 10 and Figure 11.

It can be seen that for turbulent flows, the range of results obtained with the different expressions is larger than the one obtained for laminar flow. The differences between condensation mass flow-rate of the reference model and the one obtained using other turbulent flow relations for the Sherwood number is between 10 and 40%.

For the laminar flow relations, these differences are higher, between 30 and 40%. It can also be noticed that the results for laminar flow are quite homogeneous, indicating a convergence between the different relations. Furthermore, it can be conclude, even if the error bars are high for the TOSQAN tests, that the tests having a Rayleigh number close to $10^9$ (theoretical limit to the transition to turbulence), i.e. tests 6 and 7, do not validate the laminar relations for the Sherwood number. The flow might be turbulent for all the TOSQAN tests, which is a realistic conclusion regarding to the intrusive instrumentation in TOSQAN, especially the one near the walls, that can enhance the transition to turbulence.
6.4 Influence of other parameters

As it has been done for the global HTC modelling (section 5), several parameters can be investigated in order to check their influence on the condensation mass flow-rate. The repeatability has been checked and is found to be good. Taking into account the compressibility factor for the steam or using different ways of calculating the mean gas temperature leads to the same conclusions as in section 5.

The other parameters investigated here are the expression of the diffusion coefficient DC of the mixture. The Wilke and Lee (1965) (in Poling et al. 2000 method is used in the reference modelling. Results obtained for the condensation mass flow is compared to two other relations (the one of Fuller, 1969, in Poling et al. 2000) and the one of the TONUS-0D code, Dabbene 1998) and to a constant value ($10^{-5}$ m$^2$/s). The constant value is chosen in order to evaluate the necessity, in numerical codes, of calculating a more complex equation for the DC instead of using a constant value. The relative differences are presented on Figure 12. It can be seen that the two other expressions used for the DC
lead to 20\% underestimation, where the constant value of $10^{-5}$ m²/s lead to 60\% error at lower pressure up to 20\% difference at 4 bar.

![Graph](image)

**Figure 12:** Relative difference between the condensation mass flow-rate calculated using a DC from the Wilke & Lee relations and the ones calculated from other relations.

Furthermore, five expressions of the saturation pressure have been tested, for a range of temperature from 20 to 160°C. The difference between the calculated values is of maximum 0.02 bar.

At last, as it has been done is section 5, the influence of the thermodynamic conditions of the tests on the adequation between the experiment and the modelling has been investigated. The concerned thermodynamic parameters are the same as in section 5. No real relation has been found between these parameters and the relative difference for all tests except test 9b. This test is the only highly stratified test, with local gas saturation (homogeneous condensation), having the highest gas-wall temperature difference. This result confirms the need of other tests under stratified conditions.

7. CONCLUSIONS

This paper shows the complexity of a rigorous use of the heat and mass transfer modelling for condensation in containment applications. It presents an estimation of the differences introduced in the results for different modelling approaches. Concerning the mass transfer modelling, the approach does not only concern the way the diffusion mass flux is described, but also the heat and mass transfer correlations and the relations used to determine thermodynamic or gas characteristics such as the diffusion coefficient, which is a linear factor of the mass transfer rate. That is why, when condensation modellings of safety codes are described, it is important to describe not only the mass flux modelling but also the correlations and the relations used for the thermodynamic properties.

From the experimental point of view, it has been seen that the way of calculating the bulk mean temperature can have an influence for stratified tests. The influence of compressibility effect is also not negligible if tests are performed at higher pressure (4-5 bar).

This paper shows also that the condensation tests in TOSQAN are quite in good concordance with both modelling approaches (empirical correlations and mass transfer modelling). One test, which presents specific thermodynamic conditions, does not match any modelling. Some more tests of the same kind should be performed in order to understand this result.

Furthermore, this conclusion is valid for different variables (global HTC and condensation mass flow-rate). It is important to see that these approaches, developed on analytical or laboratory experiments, are also valid for a larger facility like TOSQAN, having a more industrial approach closer to the real reactor case.
NOMENCLATURE

Most of the mathematical terms are defined in the text.

Thermal Grashof number:
\[ G_m = \frac{g (T_w - T) L}{T_v \nu_h} \]

Mass Grashof number:
\[ G_m = \frac{g (X_n - X_c) L}{V_\nu} \left( \frac{M_s}{M_n - M_r} - X_c \right)^{-1} \]

Nusselt number:
\[ Nu = \frac{hL}{\lambda} \]

Mass Rayleigh number:
\[ Ra_m = G_m Sc_m \]

Injection Reynolds number:
\[ Re_{inj} = \frac{2 \rho_{inj} U_{inj} R}{\mu_{inj}} \]

Injection Richardson number:
\[ Ri_{inj} = \frac{(\rho_{inj} - \rho_m) g 2 R_h}{\rho_m U_{inj}^2} \]

Thermal Rayleigh number:
\[ Ra_{th} = Gr_{th} Pr_m \]

g: gravity
M: molar mass
Rc: steam injection radius
Dc: steam injection diameter
L: condensing length
U_{inj}: injection velocity
Q_{inj}: injection mass flow-rate
Vol: TOSQAN volume
\rho: density
\mu: dynamic viscosity
\nu: kinematic viscosity

Subscripts:

CW: condensing wall
f: film (mean value between bulk and interface)
g: non condensable gases
i: interface
inj: injection
m: mean value (relative to the bulk)
ma: mass
NCW: non condensing wall
o: bulk
s: steam
sat: saturation
th: thermal

REFERENCES AND CITATIONS


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Appendix

<table>
<thead>
<tr>
<th>Expression</th>
<th>Relation type</th>
<th>Coefficients</th>
<th>Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td></td>
<td>A = 0, B=0.13, n = 1/3</td>
<td>Mc Adams (1954) in Roshenow and Choi (1961)</td>
</tr>
</tbody>
</table>
| 2          | $Nu = A + B (Ra)^n$ | $A = 0$  
$B = \frac{0.0248 \cdot Pr^{1/15}}{[1 + 0.494 Pr^{2/3}]^{3/5}}$  
n = 2/5 | Burmeister (1983) |
| 3          |               | A=0, n=2/5  
$B = \frac{0.0295 \cdot Pr^{1/15}}{[1 + 0.494 Pr^{2/3}]^{3/5}}$ | Kays & Crawford (1993) |
| 4          |               | A=0, B=0.183, n=0.31 | Bayley (1955) in Burmeister (1983) |
| 5          | $Nu^{1/2} = A + B (Ra)^n$ | A=0.825, n=1/6  
$B = \frac{0.387}{1 + \left(\frac{0.492}{Pr}\right)^{1/27}}$ | Churchill & Chu (1975) in Burmeister (1983) |
| 6          | $Nu = A + B (Ra)^n$ | A=0.021, n=0.4 | Roshenow and Choi (1961) |

Table 7: Relations for the Nusselt number for turbulent flow over a flat plate.

<table>
<thead>
<tr>
<th>Expression</th>
<th>Relation type</th>
<th>A,B</th>
<th>Reference</th>
</tr>
</thead>
</table>
| 7          |               | $A = 0$  
$B = \frac{0.68 \cdot Pr^{1/4}}{(Pr+ 20/21)^{1/2}}$ | Burmeister (1983) |
| 8          |               | A=0, B=0.548, n=1/4 | Lorenz (1881) in Bird (1960) |
| 9          | $Nu = A + B (Ra)^n$ | A = 0, B=0.59, n=1/4 | Mc Adams (1954) in Roshenow and Choi (1961) |
| 10         |               | $A=0.68$  
$B=\frac{0.67}{1 + \left(\frac{0.492}{Pr}\right)^{1/27}}$,  
n=1/4 | Churchill & Chu (1975) in Burmeister (1983) |

Table 8: Relations for the Nusselt number for laminar flow over a flat plate.