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# CHARACTERIZATION OF EVAPORATION RATE, TEMPERATURE, VELOCITY AND HUMIDITY FIELDS IN A COLD CHAMBER

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## ABSTRACT

In a food factory some pathogenic bacteria can grow even at low temperature, especially *Listeria Monocytogenes* which resists to temperature around 0°C. Currently, two ways are used to prevent bacterial growth; product treatment (pasteurization, pH decrease) and cleaning. However, this is not sufficient to eliminate all microorganisms in practice.

This work takes part of a four-year research project called EcoSec (2013-2016) in France. The practical objective is to propose an ecological and economical way to destroy *Listeria Monocytogenes* by humidity reduction. The principle idea is based on the dehumidification of the supply air of a plant. To achieve this objective, a method to predict the evaporation rate on wet surfaces in a refrigerated room was developed using experimental and numerical approaches (CFD, Computational Fluid Dynamics). The study was performed inside a cold room (volume 29m<sup>3</sup>) in which the temperature and velocity of air were controlled. The results show that the air humidity and its local velocity near the wet surface play a major role.

Keywords: food, cooling chamber, humidity, temperature, air flow velocity, food safety

## NOMENCLATURE

$C_{p_{air}}$	Specific heat of air at 5°C	(J.kg <sup>-1</sup> .K <sup>-1</sup> )
$C_{pl}$	Specific heat of the plate (stainless steel)	(J.kg <sup>-1</sup> .K <sup>-1</sup> )
$\frac{dm}{dt}$	Evaporation rate	(kg.s <sup>-1</sup> )
$e$	Water thickness	(m)
$h$	Average convective heat transfer coefficient	(W.m <sup>-2</sup> .K <sup>-1</sup> )
$hr$	Relative humidity	
$k$	Average mass transfer coefficient	(m.s <sup>-1</sup> )
$Le$	Lewis number	
$m(0)$	Initial water weight	(kg)
$m(t)$	Water weight at time $t$	(kg)
$m_{pl}$	Weight of the plate	(kg)
$M_w$	Water molar mass	(kg.mol <sup>-1</sup> )
$P_{sat}(T)$	Saturation vapor pressure at $Ta$	(Pa)
$R$	Ideal gas constant	(J.K <sup>-1</sup> .mol <sup>-1</sup> )
$S$	Surface of the plate	(m <sup>2</sup> )
$S_m$	Wet surface	(m <sup>2</sup> )
$t$	Time	(s)
$t_l$	Time at which wet surface starts to decrease	(s)
$Ta$	Air temperature	(°C)
$T_{pl}$	Plate temperature	(°C)
$T_{wb}$	Wet-bulb temperature	(°C)
$\rho_{water}$	Water density at 5°C	(kg.m <sup>-3</sup> )

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$\rho_{air}$	Air density at 5°C	(kg.m <sup>-3</sup> )
$\Delta H_v$	Enthalpy of vaporization	(J.kg <sup>-1</sup> )

## 1. INTRODUCTION

### 1.1. Background

The study of water evaporation inside a production plant is a topic of major interest to prevent *Listeria Monocytogenes* growth in the product. These bacteria survive especially thanks to water and high relative humidity inside the room.

The main objective of this work is to propose optimal operating conditions to reduce the humidity in a production plant and technical solutions to control food contamination. More precisely, the objective of this paper is to develop a numerical approach to predict the water evaporation rate in a cold chamber. This rate varies due to the position inside a cold room because of the heterogeneity of temperature, velocity and humidity inside the chamber. An evaporation model was developed and the predicted results were compared with the experimental ones. After validation, the model will be used to predict the evaporation rate in other conditions unexplored experimentally. This can help to propose operating conditions to completely dry the production plant as fast as possible after cleaning.

### 1.2. Literature review

Many studies have been performed concerning the prediction of the velocity, temperature and relative humidity fields in refrigerated equipment. Laguerre and al. (2009a [1], 2009b [2], 2010a [3], 2010b [4], 2012 [5]) and Hoang and al. (2012) [6] worked into the prediction of this parameters into refrigerated display cabinets, domestic refrigerators and refrigerated trucks.

For the numerical approach, the conservation equations of mass, momentum and energy are solved allowing the knowledge of the temperature, humidity, velocity fields. Sometimes evaporation/condensation at the product surface are also taken into account. Delele et al. (2009) [7] studied, for example, the influence of a humidification system to avoid fruits to be dried. A multiphase model was developed and the droplets evaporation was taken into account as well as the product and system characteristics (humidification rate, on and off cycle of the cold production system). The ambient conditions used in this study were: velocity (around 0.5m.s<sup>-1</sup>), temperature (20°C, 10°C, 1°C) and humidity (50%, 83.6% and 96%). This study shows a possibility of a CFD approach to analyze the influence of the humidification parameters and the possibility to define the optimal conditions. The coupling of heat and mass transfer equations has been studied in different conditions. Kaya and Dincer (2006) [8] studied, for example, the heat and mass transfer during a wet object drying. Several geometries were used and the object was placed inside a working area (0.6x0.2 m<sup>2</sup>) in which a low velocity (0.33m/s) was maintained. 2D simulations (commercial software Fluent) have been achieved to predict the heat and mass transfer coefficients. This study also shows the boundary layer and temperature field around this object.

Bower and Saylor (2009) [9] used an electronic balance to measure the evaporation rate of a bulk of water (different widths; 15.2cm, 30.5cm, 45.7cm and 60.9cm) under natural convection with different ambient conditions (22°C to 32°C and 51% to 58% relative humidity). These authors propose a correlation between the mass transfer and airflow ( $Sh = a * Ra^b * Sc^c$  with a, b and c constants determined experimentally).

An important point in the evaporation is also the radius of the droplets or the water film thickness. Kelly-Zion et al. (2013) [10] have shown that the smaller the radius of the droplet is, the smaller the evaporation rate will be. D'Agaro et al. (2006) [11] studied the evolution of the shape of a droplet during evaporation. Two periods were observed: one with constant wet surface and decrease of contact angle and one with decrease of wet surface and constant contact angle.

## 2. MATERIALS AND METHODS

### 2.1. Cold chamber

The experiments were carried out inside a cold chamber at Irstea where the temperature and velocity can be controlled. The cold chamber is 3.4m long, 3.4m wide and 2.5m high which represent a total volume of 29m<sup>3</sup> (Figure 1). The cooling unit is constituted of two axial fans of 30cm diameter which allow the flow rate to vary from 1000 m<sup>3</sup>.h<sup>-1</sup> to 2450 m<sup>3</sup>.h<sup>-1</sup>, and the average inlet air velocity from 2 to 4.8 m.s<sup>-1</sup> (same order of magnitude as in productions plants). The evolution of temperature and relative humidity inside the chamber was followed using thermocouples and hygrometers. The water was sprayed on a stainless steel plate (Width x Length x Thickness 0.9m x 0.9m x 0.001m) with a 10 cm thickness thermal insulation (extruded polystyrene) underneath and placed on an electronic balance located on the floor. The evolution of water evaporation rate could be followed by the recorded weight. The experimental evaporation rate at different inlet air velocities was compared with the one calculated from the developed model (presented in 2.4.).

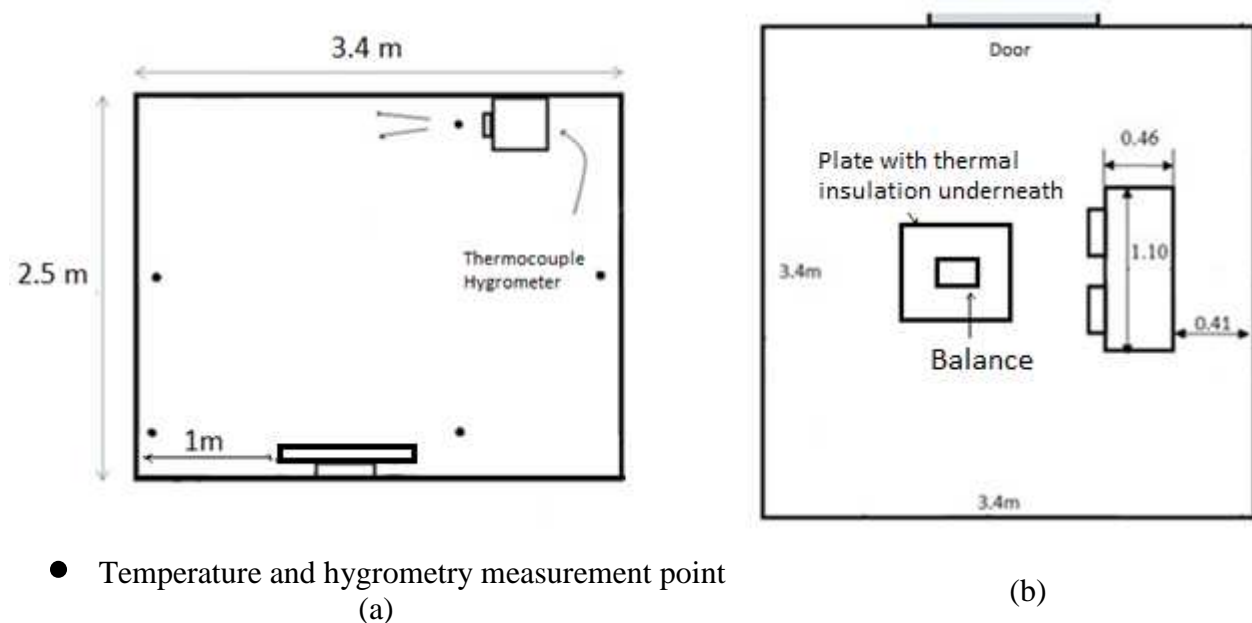


Figure 1 - Cold chamber (a) - Side view (b) - Top view

In order to prevent the influence of the air blowing inside the room from having an impact of the value measured by the balance, the ventilators were stopped every 20 minutes during 2 minutes and the weight was taken during this time.

### 2.2. Measurement devices

Five thermocouples (T type, 1 mm diameter, 0.2°C precision) and hygrometers (3% RH precision) were used to measure air temperature and relative humidity at different positions (Figure 1a). The thermocouples were calibrated at 5 different temperatures (-10°C, 0°C, 10°C, 20°C, 30°C) and the hygrometers at 3 relative humidity (11.3%, 75.5%, 97.7%).

The velocity measurement was performed with a hot wire anemometer (0.03 m.s<sup>-1</sup> precision) in front of the fans at different positions.

### 2.3. CFD simulation

In order to understand the phenomena, particularly the airflow pattern inside the cold chamber, simulation of the air flow was performed on Ansys-Fluent 14.5 with the standard k-ε turbulence model and the enhanced wall treatment. The mesh was based mainly on tetrahedral elements with refined prismatic cells nears the wall. It consisted in approximately 9.1\*10<sup>5</sup> elements. The inlet air temperature was fixed at 5°C and the upper part of the plate temperature at 10°C, with the thermal insulation underneath of 10 cm thickness. The convective heat transfer coefficient (h) was identified by the simulation and this value was used to develop the evaporation rate model (part 2.4.).

#### 2.4. Water evaporation rate model

The stainless steel plate was insulated at the bottom using extruded polystyrene. Thus, it can be considered that the exchanges by convection and evaporation with the air take place only on the top part of the wet plate (Figure 2).

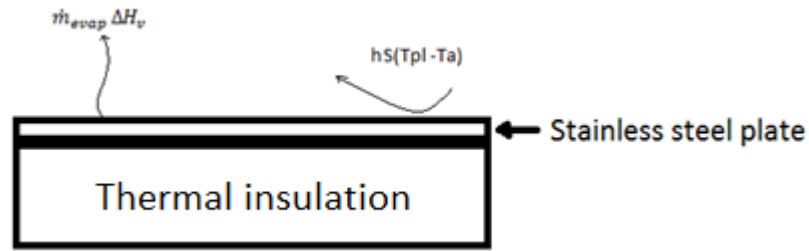


Figure 2 - Physical phenomenon on the plate

The equation 1 describes the variation of the water weight due to the evaporation. The equation 2 describes how heat accumulation in the plate is related to the difference between the heat exchange by convection and water evaporation.

$$\frac{dm}{dt} = -k \cdot S_m \cdot (C_{sat(T_{pl})} - C_{wa(T_a,hr)}) \quad (1)$$

$$\underbrace{m_{pl} \cdot C_{pl} \cdot \frac{dT_{pl}}{dt}}_{\text{Heat accumulation}} = \underbrace{h \cdot S \cdot (T_a - T_{pl})}_{\text{Heat exchange by convection}} - \underbrace{k \cdot S_m \cdot (C_{sat(T_{pl})} - C_{wa(T_a,hr)}) \cdot \Delta H_v}_{\text{Heat exchange by evaporation}} \quad (2)$$

With:

$$C_{sat(T_{pl})} = \frac{P_{sat}(T_{pl}) \cdot M_w}{R \cdot (T_{pl} + 273,15)} \quad C_{wa(T_a,hr)} = \frac{P_{sat}(T_a) \cdot hr \cdot M_w}{R \cdot (T_a + 273,15)}$$

As a first approach to predict the evaporation rate, the thermal inertia of the plate is considered negligible. Equation 2 becomes then:

$$h \cdot S \cdot (T_a - T_{pl}) = k \cdot S_m \cdot (C_{sat(t_{pl})} - C_{wa(T_a,hr)}) \cdot \Delta H_v \quad (3)$$

By replacing the equation 3 in the equation 1, the evaporation rate (dm/dt) can be calculated knowing the plate temperature.

At the beginning of evaporation, it can be assumed that the droplets or the puddles occupy almost the whole surface ( $S_m=S$ ). Therefore, the plate temperature is supposed to be constant and equal to the wet bulb temperature. The evolution of the water weight can then be predicted by the equations 4 or 5:

$$m(t) = m(0) - \left( \frac{h \cdot S \cdot (T_a - T_{pl})}{\Delta H_v} \right) \cdot t \quad (4)$$

$$m(t) = m(0) - \left( k \cdot S_m \cdot (C_{sat(T_{pl})} - C_{wa(T_a,hr)}) \right) \cdot t \quad (5)$$

It is to be emphasized that for  $0^\circ\text{C} < T_{pl} < 5^\circ\text{C}$ ,  $C_{sat(T_{pl})}$  can be calculated approximately by the following expression (obtained by linear regression):

$$C_{sat(T_{pl})} = 3,89 \cdot 10^{-4} T_{pl} + 4,78 \cdot 10^{-3} \quad (6)$$

It is to be reminded that the mass transfer coefficient ( $k$ ) can also be estimated from the convective heat transfer coefficient ( $h$ ) using the Lewis analogy (equation 7):

$$k = \frac{h}{\rho_{air} \cdot C_{p_{air}} \cdot Le^{2/3}} \quad (7)$$

Using the equations 3, 6 and 7, the wet-bulb temperature can be approximated by following expression:

$$T_{pl} = T_{wb} = \frac{\rho_{air} \cdot C_{p_{air}} \cdot Le^{2/3} \cdot T_a - (4,78 \cdot 10^{-3} - C_{wa}(T_a, hr)) \Delta H_v}{\rho_{air} \cdot C_{p_{air}} \cdot Le^{2/3} + 3,89 \cdot 10^{-4} \cdot \Delta H_v} \quad (8)$$

So, using together the equations 4 and 8, it is easy to predict the evolution of the mass of water at the beginning of evaporation.

It can be observed that, the higher the relative humidity, the lower the evaporation rate. Indeed, for a high humidity the wet-bulb temperature (or plate temperature) tends toward the air temperature. So that, according to equation 4, the mass of water decreases slower for higher air humidity.

### 3. RESULTS AND DISCUSSION

#### 3.1. Velocity field in the chamber

The velocity field inside the chamber obtained from CFD simulation (3D) is presented in figures 3 and 4 for an average inlet velocity of  $4.8 \text{ m.s}^{-1}$ . These figures show the directions of the airflow and the velocity magnitude. It is then possible to examine the position of high or low velocity zones. This is an important point because the velocity plays a major role for the determination of the convective heat transfer coefficient: the higher the local velocity, the higher the transfer coefficient. For the investigated position, the simulation showed that the convective heat transfer coefficient between the plate and the air is  $7.6 \text{ W.m}^{-2}.\text{K}^{-1}$ .

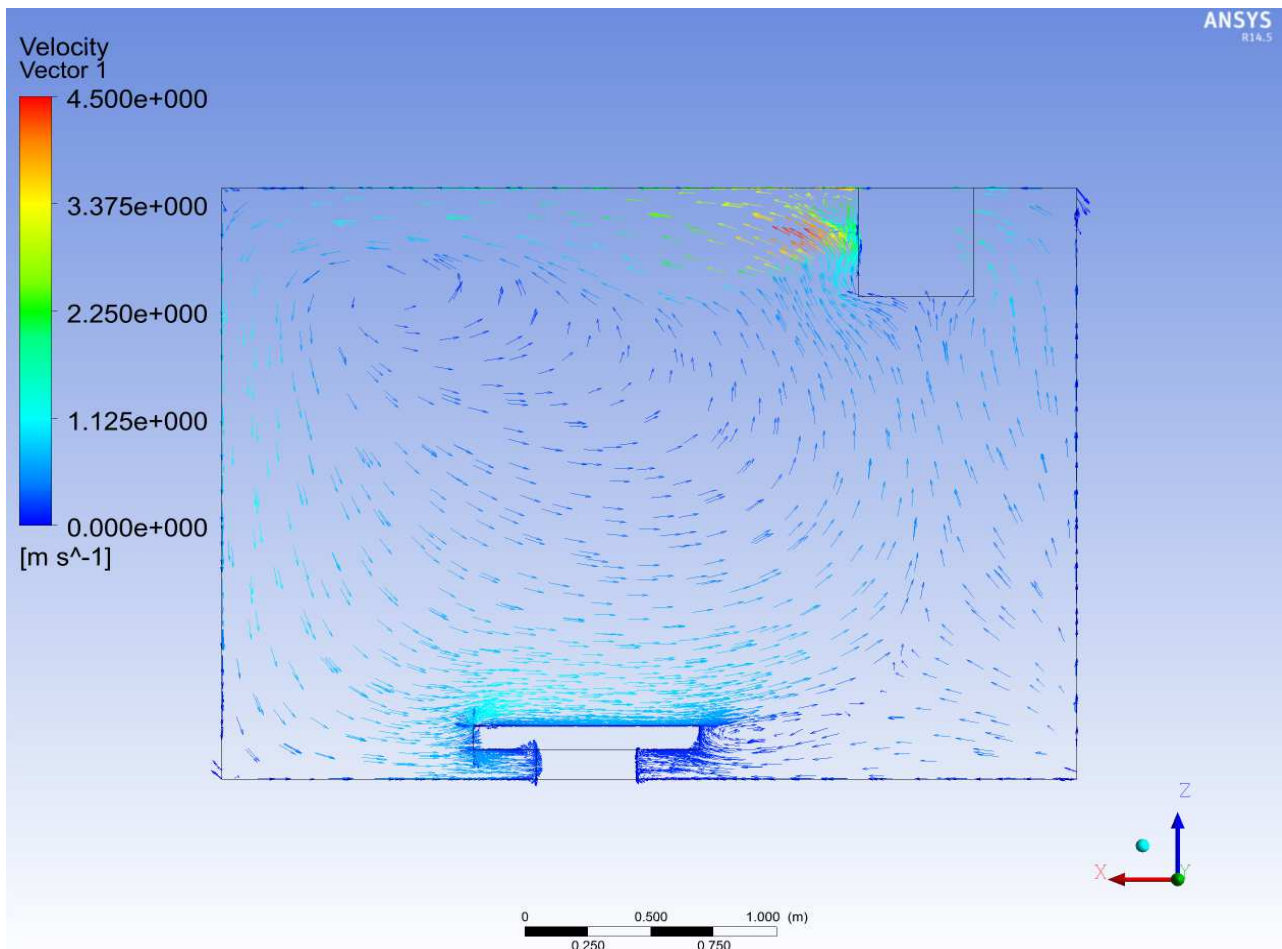


Figure 3 – Velocity vector Side view (symmetry plane)

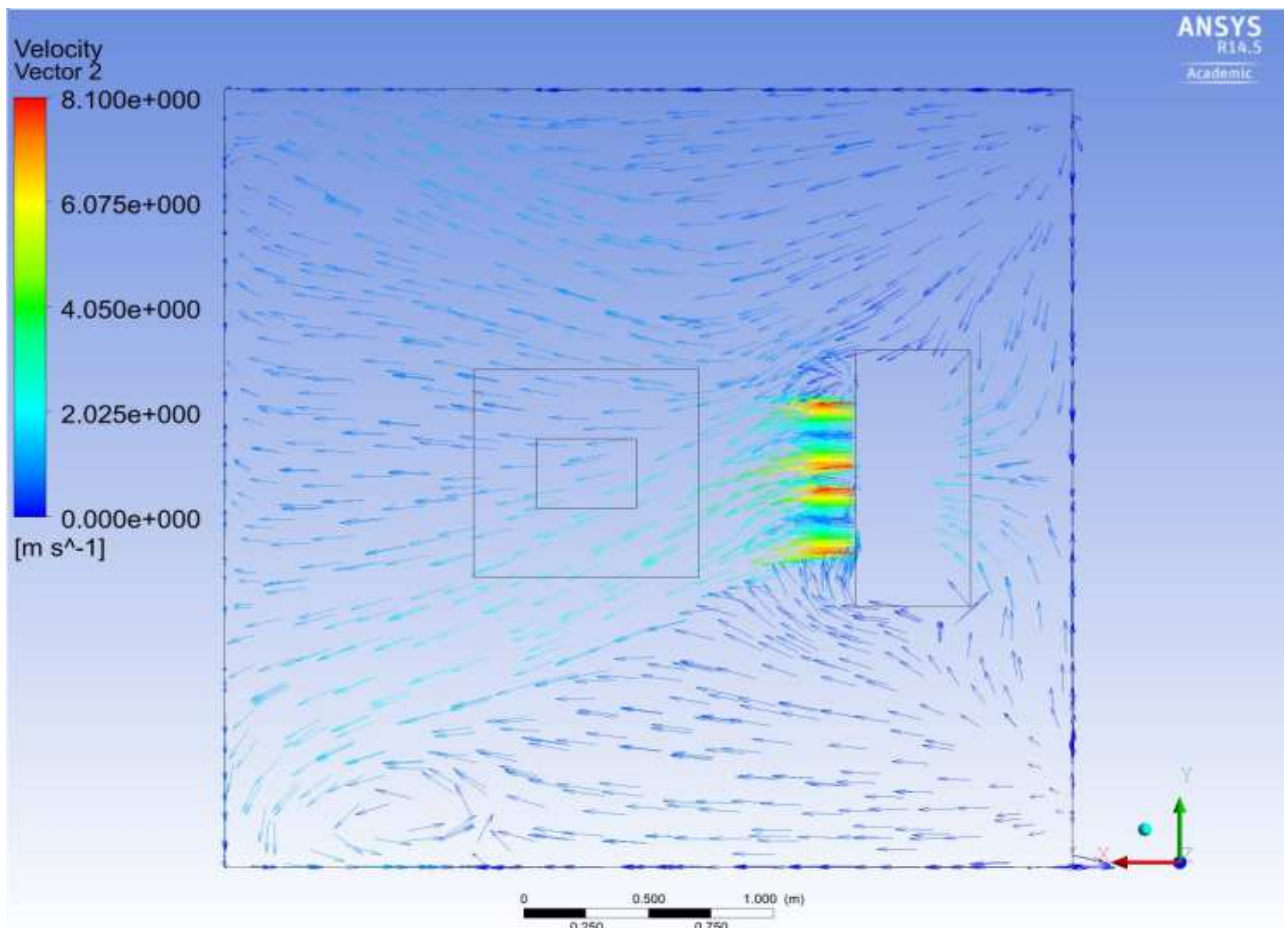


Figure 4- Velocity vector Top view (middle height of the fans)

### 3.2. Evaporation rate considering water droplets

The value of heat transfer coefficient obtained from the CFD simulation was applied to the model of water evaporation (eq. 4, constant wet surface) for 3 air inlet conditions:  $3.0 \text{ m}\cdot\text{s}^{-1}$ ,  $5.4^\circ\text{C}$ , 90% RH –  $4.0 \text{ m}\cdot\text{s}^{-1}$ ,  $5.4^\circ\text{C}$ , 90% RH and  $4.8 \text{ m}\cdot\text{s}^{-1}$ ,  $5.1^\circ\text{C}$ , 85% RH. The comparisons between the experimental and calculated data are shown in figure 5. The first part of the evaporation seems to be predicted correctly while the second part is under estimated by the model because the wet surface area is then decreasing. The influence of air velocity on the evaporation rate is well shown in this figure: the higher the inlet velocity, the faster the evaporation. This can be explained by the fact that a higher air velocity leads to a higher mass transfer coefficient.

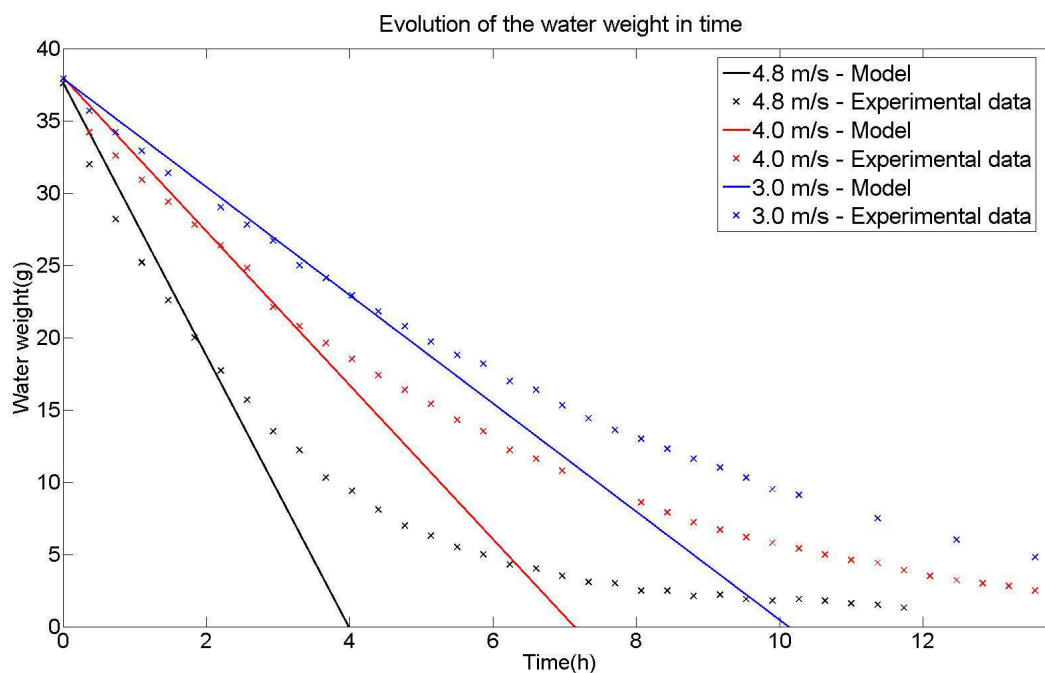


Figure 5 - Evolution of the water weight - Comparison with the developed model

#### 4. CONCLUSION AND PERSPECTIVES

A CFD simulation was carried out to better understanding some phenomena inside a cold chamber, especially for the airflow direction and velocity field. It allowed also determining the convective heat transfer coefficient which was then used for the prediction of water evaporation rate in the chamber. The model equations show the importance of the air relative humidity inside the room on the evaporation rate: the higher the air humidity, the lower the evaporation rate. The numerical and experimental results also showed the influence of airflow rate: the higher the air velocity of the fans, the higher the evaporation rate.

The first part of the evaporation was well predicted while the second part was under estimated by the model. To improve the model, it is necessary to understand the mechanism of droplet surface evolution which requires the development of an experimental setup. This will be carried out in the future.

Currently, the inlet air temperature and relative humidity were constants during experiments. These parameters will be varied allowing studying their impact on the evaporation rate at different positions in the chamber.

The result of this study leads to conclude that to avoid humidity stagnation in a production plant, one can increase the airflow rate or decrease the air relative humidity. The later solution is more comfortable for workers compared to the one of high air velocity.

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