Mass transfer evaluation for condensation of humid air on a horizontal plate

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Résumé – Cette étude concerne la caractérisation du phénomène de condensation d'air humide sur un substrat solide et son contrôle par convection forcée en milieu confiné. Ainsi, un dispositif expérimental a été développé pour étudier l'hydrodynamique et le flux de matière (mesure du coefficient de transfert k_La) sur un substrat horizontal soumis à des conditions environnementales contrôlées. Les expériences ont été réalisées à température ambiante, pour une humidité relative de 35 à 65% et pour une vitesse moyenne d'écoulement d'air variant de 1.0 à 3.0 m.s^{-1}.

Nomenclatura

D	binary diffusion coefficient, m^2.s^{-1}
L	length of the plate, m
N	mass flux, kg.m^{-2}.s^{-1}
T	temperature, °C
U	mean air velocity, m. s^{-1}
ω	mass fraction
ρ	density, kg.m^{-3}
μ	dynamic viscosity, Pa.s
Dimensionless parameters	
Nu	Nusselt number
Sh	Sherwood number
Re	Renolds number
Sc	Schmidt number
Subscripts	
a	ambiant
c	set to the controller
d	dewpoint
s	plate surface
∞	mean value of free flow

1. Introduction

The condensation phenomenon of humid air on solid substrates can occur in many applications, and it is known as one of the most difficult problem to deal with for the improvement of the quality of air in a closed environment (space habitat, submarine, operation room, greenhouse, etc.), the habitability of crew compartments or the maintainability of electronic devices. It can cause corrosion, the development of mold and pathogen germs, etc. On glass it affects sun radiation.

The development of Earth like environment inside a closed-system for the progress of controlled ecological life support systems (CELSS) is a challenge today. This is a requirement for long-duration exploratory manned missions to fulfil the needs of a crew including nutritional demand, atmosphere regeneration, waste recycling and psychological support. One of the key elements for CELSS are plants, as they regenerate ambient air by photosynthesis, help water recovery by transpiration, supply fresh food or nutritional needs for crews and can be used for the recycling of wastes. A maximum of biological materials could be reused for plant cultivation thanks to various effective waste processing techniques [1-4].

Growing plants in space missions is a vital component and its performance in CELSS will be principally dependent on the progress of plant cultivation technology for space and the achievement of associated equipment. Higher plant growth in a greenhouse is optimized by the environmental conditions among which the influence of ventilation, gas/liquid transfer at their interface with air, condensation on windows (sun transmitter)... Moreover, condensation on walls or plant leaves has to be controlled as well as the ambient air for optimized living conditions within the spacecraft, even if the humidity level is not as high as in a greenhouse in order to prevent mould, rot or rust. Furthermore, forced convection is known to be a good solution to prevent condensation while maintaining optimized conditions for life.

Hence, the optimization of CELSS requires a global coupled hydrodynamic, heat and mass transfer modelling that could simulate precisely the atmosphere in spatial greenhouses, or in manned capsules. Further, the coupling with microbiological development models [1] will help for the protection of the crew from nosocomial infections, the optimization of the microclimate prevailing in a space greenhouse and a better control of higher plant growth.

The present study was motivated by the investigation of the coupling between ventilation (forced convection) and condensation inside CELSS. The experimental and theoretical modelling of CELSS requires a comprehensive understanding of the micro to the macro levels of gas/liquid phase transfer. Our purpose was to clarify the basic mechanisms concerning the coupling of heat and mass transfer during phases of condensation with a low Reynolds number turbulent flow, as well as the kinetics of the diverse phenomena interacting at different characteristic scales. The goal was to establish correlations between the fluxes of mass and heat, the relative humidity and the mean flow for the development of theoretical models based on local transfer coefficients. These models will later be inserted in numerical simulation software for the prediction of airflow and gas/liquid transfer on solid substrates.

The initial phase of this study was to design an innovative set-up and realize experiments of condensation on a small-size horizontal substrate of controlled temperature to describe and quantify accurately this heterogeneous transfer which develops. A device regulates the surface temperature below the dew point of the air and thus, leads to condensation phases.

We discuss herein, briefly, the experimental set-up developed at 1g to characterize the condensation mass flux in a controlled wind tunnel environment and the flow. The study of flow profiles on the surface of vertical and horizontal flat plates has already been performed in dry conditions [5-6]. This paper presents the results obtained for the condensation of humid air in various conditions when the mean flow velocity varies between 1 and 3 $m.s^{-1}$.

2. Definition of the experiment

2.1. Global experiment

To generate and control a flux of condensation of wet air on the surface of a flat plate, we have developed a system based on a controlled thermoelectric cooler. The temperature of the plate was kept constant in order to induce a stable flux of condensation on the active plate/air interface and the condensate mass was monitored by continuous precise weighing. The overall device was placed in a vein which hydrodynamics, temperature and humidity fields were controlled [7]. The wind tunnel facility is a closed loop, sealed and highly insulated that generates runoff from nearly laminar to highly turbulent regimes. The characterization of the flow, average speed and fluctuations was performed by hot wire anemometry. Figure 1 gives a photograph of the test chamber (80 cm x 80 cm cross section) and the active condensation unit with water condensate on it.

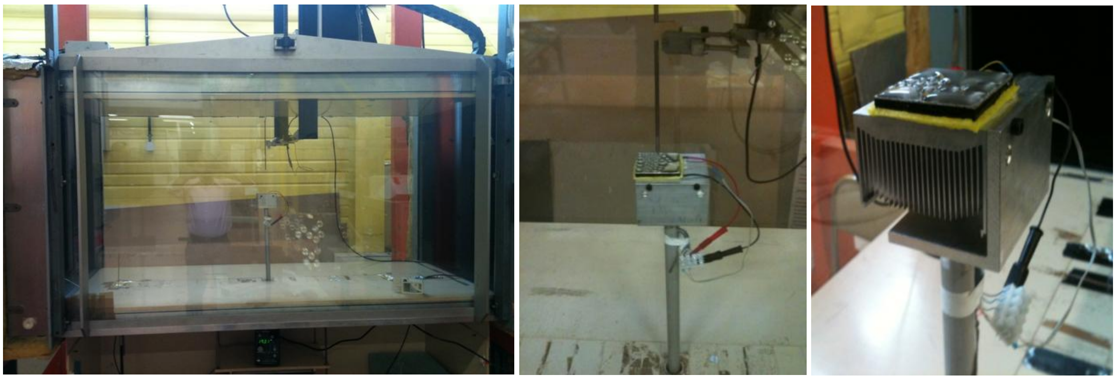

Figure 1. *Photograph of the test chamber and close look at the condensing unit in the presence of drops on top of the active surface.*

2.2. Description of condensation unit

The condensation unit was prepared after extensive experiments of the flat plate for its size and thickness, choice of metal substrate, choice of heat sink, temperature distribution at the air/solid interface in order to obtain a thermally homogeneous surface (which corresponds to the active side of the plate for condensation) [8].

A square plate of aluminium is bonded to a Peltier module of the same size (5x5 cm^2). The other face of the module is bonded to a heat sink (Figure 1) for the temperature of this side to be kept close to the ambient one (T_a, reference temperature). The temperature of the active surface (T_s) is controlled by the Peltier element which is regulated by means of a thermistor inserted into the small plate and a controller which adjusts the electric current supplied to the Peltier module [8]. The condensation unit is placed horizontally (active horizontal plate) at the centre of the test chamber (Figure 1.a-b) and maintained by a shaft fixed itself at the balance pan, located under the test chamber. The temperature regulation controller is also on the balance. The increase in mass is recorded by a precision (± 0.1 g) balance (Mettler 30).

Hence, on the cold side a temperature difference $\Delta T_s = T_d - T_s$ with the dew point (T_d) can be created and induce condensation, see Figure 2. During the experiments the controller temperature (T_c) is imposed and thus, the thermal contrast $\Delta T_c = T_d - T_c$ with the dew point (T_d), whereas ΔT_s is induced and not known. Aluminium was chosen for the plate for its high thermal conductivity in order to make the active surface to be as isothermal as possible and for its corrosion properties. It is noteworthy that the size of the aluminium plate should be small enough for the active surface to be considered thermally homogeneous and large enough for the amount of condensate to be weighed accurately on an electronic balance [8].

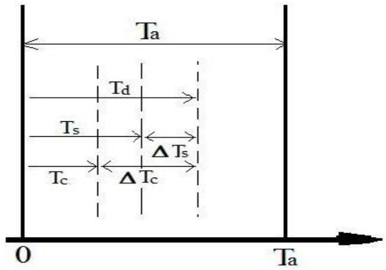

Figure 2. *Different temperature levels and thermal gradients considered herein.*

2.3. Mass condensate measurement method

The condensing unit including the temperature controller and all accessories were placed on the balance before condensation started. The weight of the condensing unit with all the supporting devices the temperature controller was approximately 11.8 kg before the fan was turned on. It was proven during calibration that the air flowing intensity had no influence on the weighing process once stability was reached. The least count of the balance was 0.1 g.

The controller temperature (T_c) was chosen relatively to the dew point temperature. The value of $\Delta T_c = T_d - T_c$ was selected such that, a sufficient amount of condensate could be produced at the end of the experiment. In fact condensation occurs when $\Delta T_s \geq 0$, but the thermal gradient within the condensation interface and the temperature measured just below the top of the upper surface of the plate need to be counted for, even though it is not known. Once condensation had started it was continued regularly till the end of the experiment without apparent interruption. The electronic balance showed an increase in weight [6].

3. Results and discussion

3.1. Amount of condensate versus time

More than 70 condensation experiments were performed on the horizontal plate at ambient temperature. Various condensation conditions (RH varied from 35% to 65%) were simulated at room temperature (around 20°C); the mean flow velocity ranged from 1 to 3 m/s. It corresponds to Reynolds numbers between 3.10^3 and 10^4 (L = 5 cm) and the Schmidt number was approximately 0.6, where

$$\mathrm{Re} = \left(\frac{\rho_\infty UL}{\mu_\infty}\right) \qquad Sc = \mu_\infty / (\rho_s D)$$

The experiments lasted from 3.5 to 8 h after condensation started. The maximum amount of condensate collected was 4.8 g on the plate and the minimum 0.8 g. The temperature and hygrometry of the wind tunnel were controlled well enough, except if there was a very large variation in the exterior ambient weather conditions (humidity or temperature), a noticeable variation in internal parameters was seen accordingly. The experimental setup of the wind tunnel was situated at INRA-Theix.

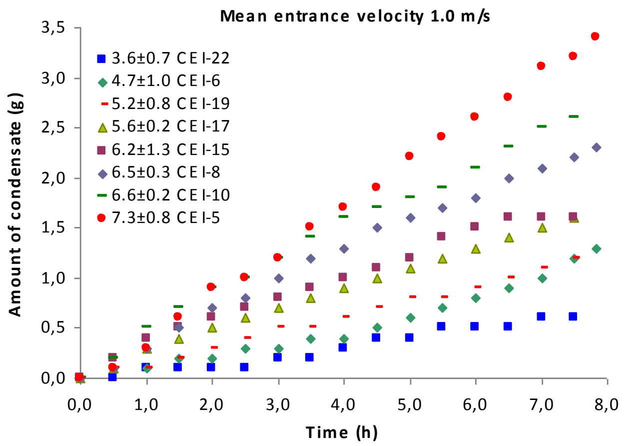

Figure 3. *Amount of condensate as a function of time for different average temperature differences (ΔT_c) under a 1.0 m/s mean entrance velocity.*

Figure 3 shows how the amount of collected condensate is significantly affected by a variation in different environmental parameters (for average temperature differences ΔT_c) versus time. The trends are slightly affected by the hygrometric conditions, which are not fully stable. However, Figure 3 indicates that on increasing the temperature difference, the average rate of collection of condensate also increases accordingly. Consequently, the main trend reflects the sensitivity of the slope and, thus, of the mass flux to the temperature difference as expected. All condensation rates reflect roughly linear global growths with time, whose slope increases with ΔT_c, as it generates the driving force through the partial pressure gradient. From all those plots we can deduce an average speed of condensation in various environmental conditions. These values are further used to deduce the mass flux on the plate.

3.2. Condensation patterns

Dropwise condensation occurred every time and it was visible (very small drops in size) after a few minutes of condensation conditions. Eye observations showed that inhomogeneous distributions of drops were observed spatially and also in size. The initial growth of drops and the coalescence process were found to be as described by Beysens [9], in which bigger drops attract the smaller ones that have grown in their vicinity and thus sweep off the metal surface around which allows the condensation process to continue with the nucleation of tiny drops, etc. It results in the appearance of very different sized drops on the surface.

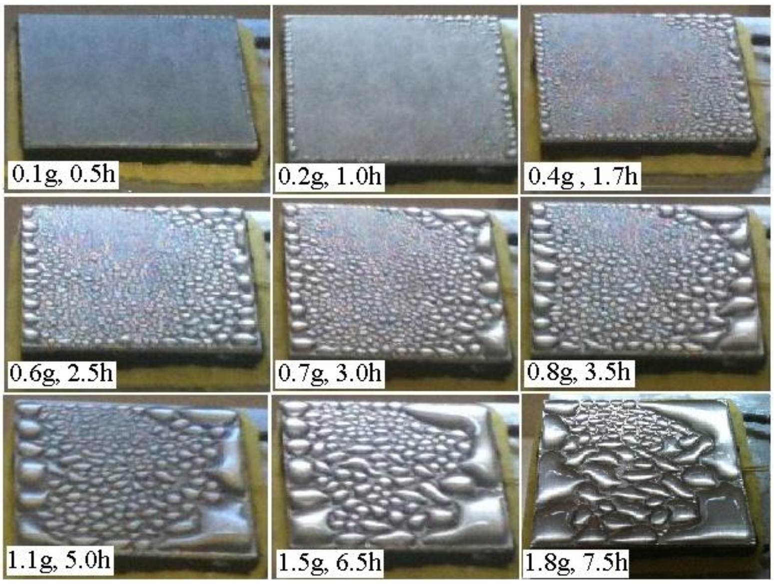

Figure 4. *Photographs of the flat plate covered with condensation at different instants of a trial.*

The surface of the aluminium metal flat plate was not well polished and only mechanical filing was done on the edges, and then just cleaned with ethyl alcohol by wipe out the substrate before starting each experiment. The hand filing of the substrate probably caused the drops to grow first on the edges all around the plate, as seen in Figure 4.

That experiment considered a mean velocity of 1 $m.s^{-1}$ and it was a good representation of the various observations, as very scattered drop configurations were obtained. The nature of the flow had probably a major effect on those patterns, and then on the heat-mass transfer at the surface of the plate, which shows that a perfect understanding of the global phenomena would require an accurate description of the 3D flow field above the plate, but it was not our initial purpose. It is also worth noting that the shape of the condensate is strongly influenced by the physico-chemical properties of the aluminium plate (contact angle, etc.).

3.3. Surface temperature estimation

The calculation of the condensation mass flux involves the partial pressure gradient at the liquid/air or metal/air interface where condensation proceeds, the area of the interface and the correlated mass transfer coefficient. As usual the area is not known and the goal is to determine the $k_L.A$ coefficient. The partial pressure of the air is deduced from the ambient temperature and the relative humidity, whereas neither the pressure nor the temperatures are known. Moreover, the only accurate data available is T_c which is the temperature measured by contact with the thermistor inserted 1 mm below the metal/air surface.

Consequently, the knowledge of the surface temperature T_s of the active condensing plate became a key issue for the estimation of the mass transfer coefficient, in addition to the mean values of the controller temperature T_c and of the ambient temperature T_a. The use of a surface sensor was not convincing because of the presence of forced convection [8]. This complexity encouraged us to try a few rules to estimate the surface temperature, keeping in mind that $T_c \leq T_s \leq T_a$. Finally, the best approximation for the surface temperature was given by the following rule [8]:

$$T_s = \frac{5 * T_c + T_a}{6} = T_c + 0.16\,(T_a - T_c) \qquad (1)$$

3.4. Mass flux evaluation for a mean flow velocity of 1 m.s^{-1}

The dependence of the mass flux as a function of the mean difference in partial pressure (between the saturated vapour partial pressure of the man flow and the partial pressure at T_s) is shown in Figure 5. The total area of the plate (5x5 cm^2) was considered to deduce the mass flux on the plate. Each experimental point corresponds to a 6-8 h experiment. The experimental value results from the algebraic average over the whole experiment duration of the data points evaluated at each time of measurement; the "slope" corresponds to the gradient of the curves plotted for the increase in weight with time for each experiment [8] (see 2.3).

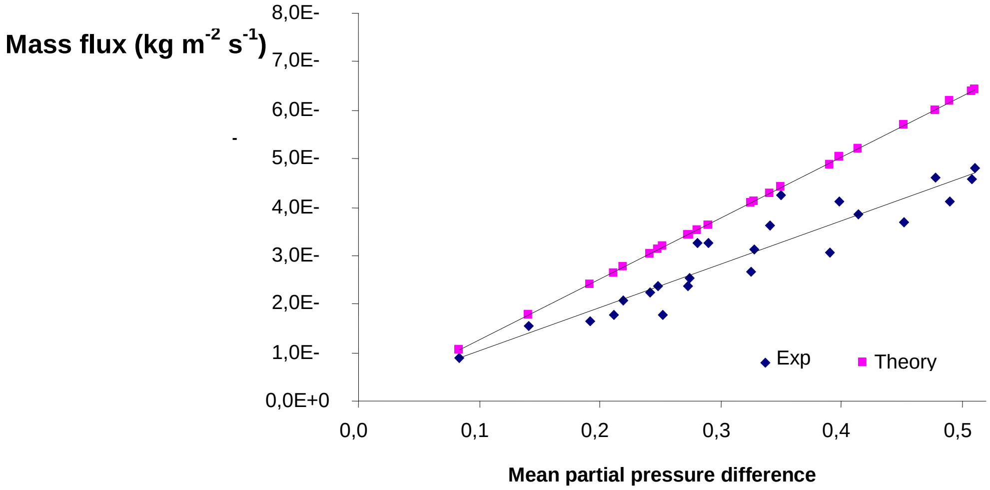

Figure 5. *Mass flux for a mean entrance velocity of 1.0 m/s as a function of the mean partial pressure difference.*

The improperly called "theory" data correspond to the arithmetic mean of points calculated using the environmental data recorded at the time of the experiment and by assuming a heat/mass analogy for the mass flux evaluation of a simple 1D stationary model. The heat flux is the one that would be calculated for a flat horizontal plate covered by a film of condensate when a laminar boundary layer flow develops on top [8].

The theoretical data show the best fits. The data are more scattered when the mass flux is plotted versus the average temperature difference ΔT_s, which indicates that the difference in partial pressure is a better indicator than the difference in temperature for the estimation of the mass flux. Indeed the driving force for condensation mass transfer (controlled by the diffusion of water vapour towards the condensation interface) is better modelled by partial pressure differences than temperature differences solely (no interaction with the relative humidity which is a key factor, as minor deviation in RH level (5% is enough) during an experiment influence significantly the condensation rate). As a matter of fact, a long variation in RH would act as a spurious mode, as the global problem is multi-parametric and the bi-dimensional plots given herein can only deal with 1 to 3 parameters only.

Variations in the experimental conditions can justify most of the scattering in data. The global set up did not allow us to be more precise and to conduct experiments at a specific ambient temperature coupled with a specific relative humidity over 7 hours as, even sealed and insulated, the air in the closed loop was slightly affected by outside environmental conditions after a few hours.

3.5. Influence of the mean entrance velocity

On increasing the mean entrance velocity inside the wind tunnel, the mass flux has a tendency to increase but not according to a linear variation [8]. The raise in mass transfer on increasing the mean velocity inside the wind tunnel is logical as the strengthening of the flow intensity over the active surface causes a thinning of the diffusion layer that develops on top.

The influence of the mean flow velocity, when it increases from 1 to 3 $m.s^{-1}$, is seen on Figure 6 as a variation of the Sherwood number versus the Reynolds number. The Sherwood number is defined as in Asano [10]:

$$Sh = \frac{N_A}{\rho D(\omega_\infty - \omega_s)/L}$$

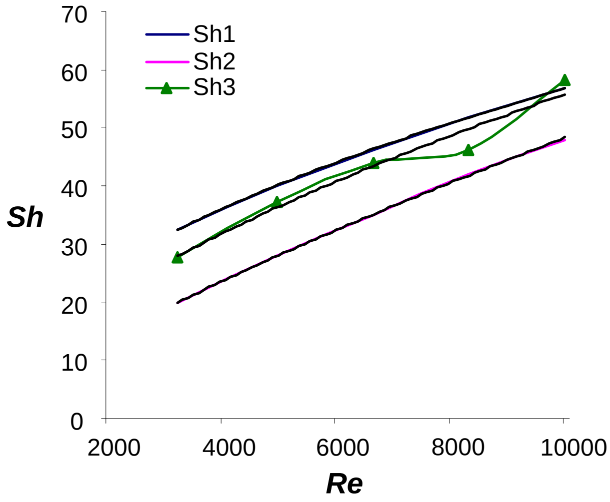

Figure 6. *Influence of the mean flow velocity on the mass.*

The curve Sh_1 corresponds to an improperly called "theoretical" one obtained with a laminar boundary layer type of approximation (film condensation over a flat plate and doing a heat/mass analogy for the estimation of the mass transfer); the curve Sh_2 is obtained with a mixed laminar/turbulent boundary layer approximation model. The experimental curve Sh_2 was obtained when including all experimental data. As a result of our experiments we can express the average Sherwood number as a dependence in $Re^{2/3}$:

$$Sh = 0.225 \mathrm{Re}^{2/3} Sc^{1/3} \tag{2}$$

4. Conclusion

The present study concerned the coupling between ventilation (forced convection) and condensation inside a closed ecological life support system. It focussed on the characterization of heat and mass transfer by condensation of humid air on solid substrates in a low *Re* number turbulent flow, configurations that can be encountered in many applications.

The innovative experimental set-up designed and data treatment have proved to be accurate for the evaluation of local mass transfer coefficients on a solid horizontal substrate.

A complex flow regime was observed over the plate with the development of a thick shear layer, similar to those observed with blunt-faced bodies. A rise in the strength of the flow intensity (measured with the Reynolds number, Re) caused an increase in mass flux with dependence in $Re^{2/3}$, which reflects a turbulent regime.

Acknowledgments

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