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# HE II CO-CURRENT TWO PHASE FLOW AT HIGH VAPOR VELOCITIES

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**Abstract**

In the framework of LHC studies, we have performed several experiments on He II co-current two-phase flow. It was found that for high vapor velocities, the heat exchange capacity between the He II flow and the pipe wall is significantly better than what can be accounted for by the liquid to wall interface of a stratified two-phase flow pattern. This seems to indicate a transition from a pure stratified two-phase flow into either a partially annular two-phase flow or a stratified two-phase flow including liquid droplets in the vapor flow or a combination of the two. In the last two cases, it is assumed that liquid droplets which get dispersed on the tube wall increase the wetted surface. A new facility has been designed to analyze this flow behavior. High sensitivity capacitive liquid level sensors glued onto the inner wall of the pipe were used in order to detect a possible semi-annular flow pattern whereas light diffraction and scattering were used to detect liquid droplets. Finally, in addition to a circumferential heat exchange box, local heat exchange boxes located at different azimuth positions are added. Description of this new facility, calibration of the local heat exchange boxes and first results are presented.

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## ABSTRACT

In the framework of LHC studies, we have performed several experiments on He II co-current two-phase flow. It was found that for high vapor velocities, the heat exchange capacity between the He II flow and the pipe wall is significantly better than what can be accounted for by the liquid to wall interface of a stratified two-phase flow pattern. This seems to indicate a transition from a pure stratified two-phase flow into either a partially annular two-phase flow or a stratified two-phase flow including liquid droplets in the vapor flow or a combination of the two. In the last two cases, it is assumed that liquid droplets which get dispersed on the tube wall increase the wetted surface. A new facility has been designed to analyze this flow behavior. High sensitivity capacitive liquid level sensors[1] glued onto the inner wall of the pipe were used in order to detect a possible semi-annular flow pattern whereas light diffraction and scattering[2] were used to detect liquid droplets. Finally, in addition to a circumferential heat exchange box, local heat exchange boxes located at different azimuth positions are added. Description of this new facility, calibration of the local heat exchange boxes and first results are presented.

## INTRODUCTION

In the framework of LHC cooling scheme studies, we have investigated the thermohydraulic behavior of He II two-phase flow[3,4]. The huge amount of data acquired on previous Cryoloop experiments confirmed the pressure loss predictions given by the Andristos/Hanratty[5] model. However, apparent excess heat transfer (with respect to pure stratified two-phase flow) occurred at high Vgs (superficial vapor velocity), possibly due to an onset of change in flow pattern. The more commonly encountered transition is to annular flow with liquid beginning to creep on the wall. The other one is a transition to mist flow which appears for high Vgs and high qualities. In order to discriminate between these two possible transitions and build a physical model for wall heat transfer, our "Cryoloop" facility was upgraded. Specific devices designed to characterize mist and annular flow were added, whereas heat transfer measurements were maintained and improved.

## DESCRIPTION OF THE EXPERIMENT

### Test Loop

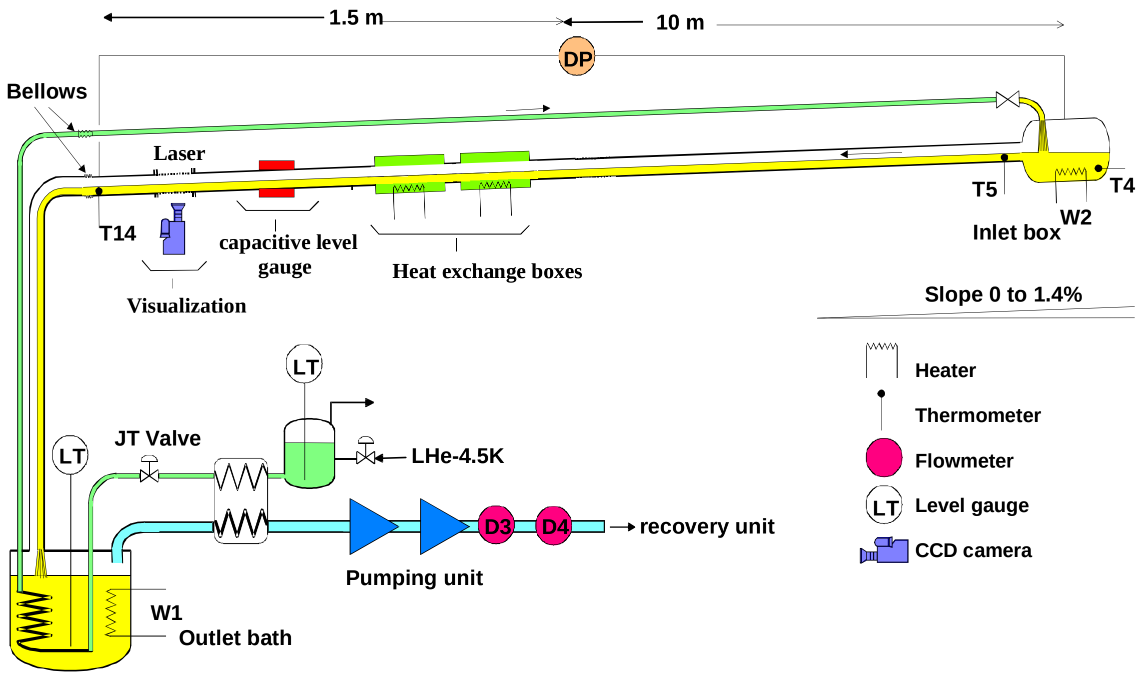


**FIGURE 1.** Test facility flow scheme

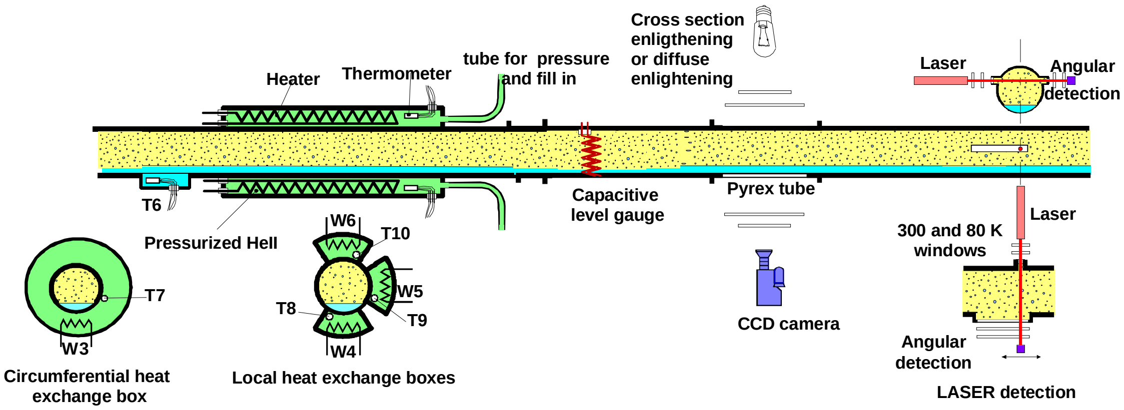


**FIGURE 2.** Details of the new instrumentation

The "Cryoloop II" (figure 1) has already been described previously[4] and mainly consists of an inlet box used to create the inlet quality, a 10 m inclinable straight line of dimensions 42x43 mm, and an outlet saturated bath where excess liquid is evaporated. During a run, total mass flow ($\dot{m}_1$) is kept constant using a fixed aperture for the J.T. valve. The inlet power ($W_2$) is used to increase step by step the quality. Finally, liquid dry out is obtained at a power $W_{2\_do}$ and used as a means to calculate heat losses ($W_{1osses}$ ) :

$$W_{1osses} = \dot{m}_1 L_{sat} - W_{2\_do} . \qquad (1)$$

The global measurements include the total mass flow rate (taken at room temperature), the saturated temperature, heating powers, temperature increase in the circumferential heat exchange box when power is on (which gives access to the thermal heat exchange). In addition, three local He II subcooled boxes located at different azimutal positions (figure 2) have been introduced in order to measure wall heat transfer at the bottom, the middle and the top of the pipe.

Four capacitance level gauges glued inside the inner pipe (see ref. 1 for details) were used to detect liquid film on the pipe wall. Furthermore, interfacial area of liquid droplets is determined by the analysis of light scattering (see ref. 2 for details).

## Calibration of the local heat exchange boxes

Calibrations were performed in a separate cryostat, where boxes are placed vertically. The boxes were filled with He II at 1 bar. The inner tube was entirely filled with saturated He II, so that the wetted perimeter was always equal to 100 %. The heat transfer coefficient of each box was measured for various temperatures (figure 3a). We measured separately the thermal conductivity of a copper sample taken from the central tube. Using a ANSYS thermal model, the Kapitza resistance between copper and He II was then adjusted to give an accurate prediction of the experimental data (see figure 3b).

Calculation of the wetted perimeter when the inner tube is only partly wetted by saturated He II is not straightforward and the numerical model (ANSYS code) was used to determine the relation between wetted inner perimeter and heat removed on each box (figure 4). This model takes into account the contribution of both the liquid and the vapor phase and of the radial conduction through the copper tube wall to determine the heat exchange capacity of a two-phase flow configuration. Due to conduction through the copper pipe wall, heat flux streamlines are not strictly radial (figure 4). Consequently heat flux on the bottom box varies over 20 % of inner wetted perimeter while 15 % is sufficient to cover surface in front of the box (figure 5). On the other hand heat deposited on middle box became to be extracted before liquid level on inner tube reached the low part of the box.

It was found that for vapor flow rates involved, heat exchange due to vapor flow was always negligible.

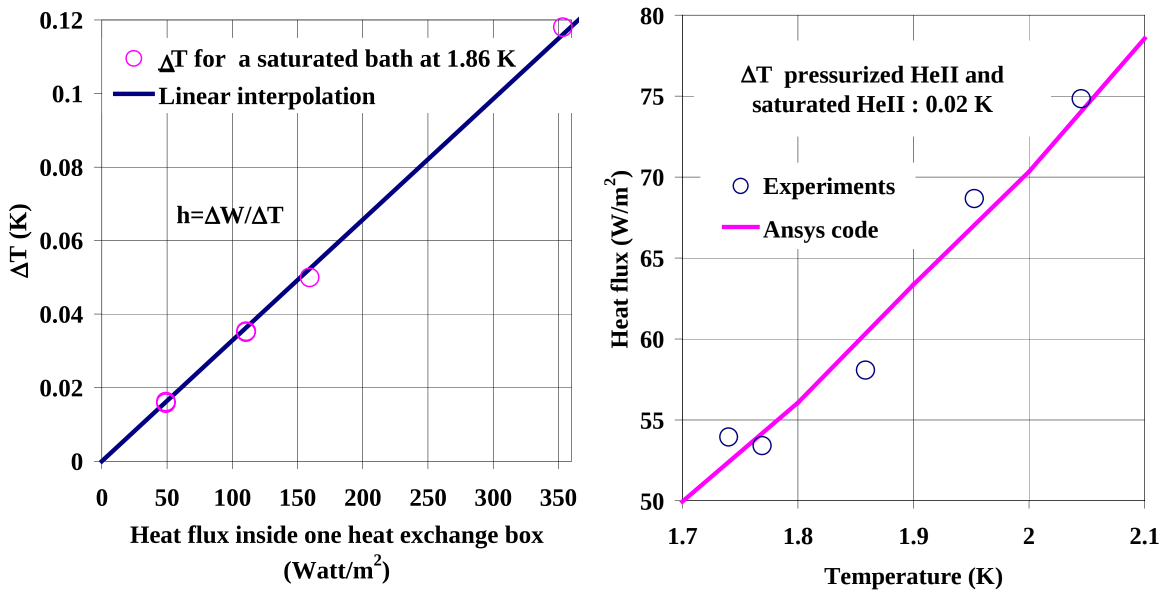


**FIGURE 3(a).** Calibration of thermal heat coefficient for the local boxes **(b).** Heat flux removed for a fully wetted perimeter

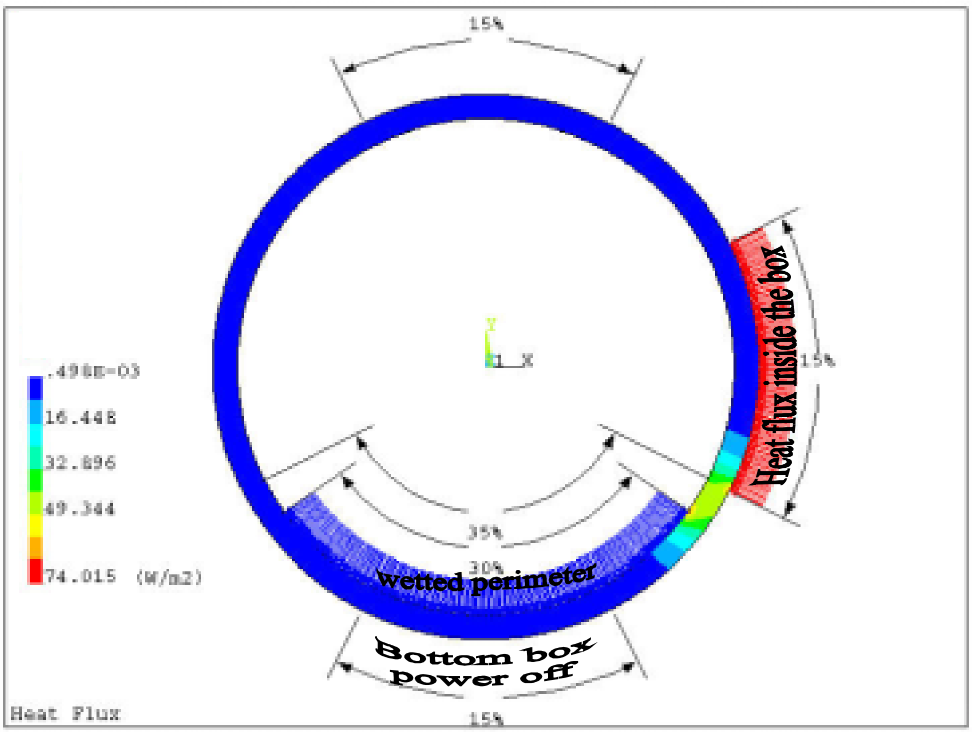


**FIGURE 4.** ANSYS heat flux calculation for $\Delta T$=0.02 K and a partly wetted inner perimeter

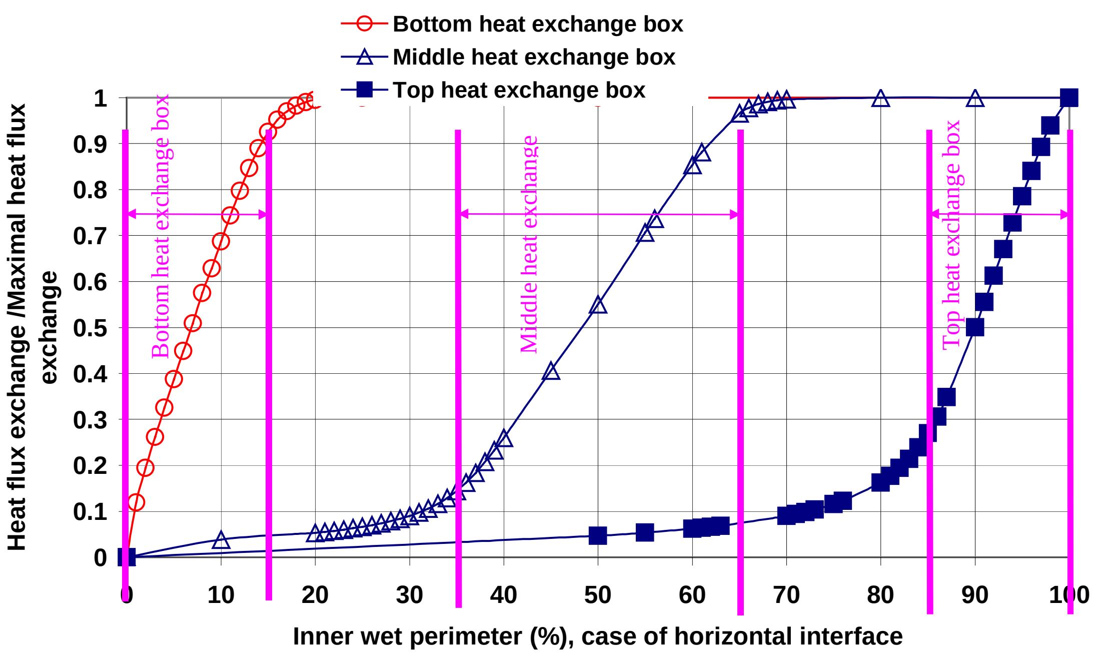


**FIGURE 5.** Relation between heat flux removed in local boxes and inner wetted perimeter

## Procedure for circumferential wall heat transfer capacity measurement

In case of "over wetting" due to liquid droplets impinging on the pipe wall, the thickness of the liquid layer is probably thin and can be dried out by sufficient heat flux. In order to check the dry out crisis, heat flux must be varied. Waiting to reach steady state is time consuming and a signal response procedure was adopted. The system is similar to a RC circuit, R being the thermal resistance between annular box and two phase flow and C the heat capacity of the box. The describing equation is

$$T' + \frac{M\, c_P}{h\, S_{wet}} \frac{\partial T'}{\partial t} = \frac{1}{h\, S_{wet}} W \text{ , where } T' = T_{box} - T_{sat} \tag{2}$$

$T_{box}$ represents the temperature inside the He II pressurized box, $T_{sat}$ the temperature of the He II saturated flow, M is the mass of superfluid inside the box, $c_P$ the corresponding specific heat and *h* the heat transfer coefficient calculated after calibration.

With the assumption of constant $c_P$ and *h*, the resolution of equation (1) in the case of a sinusoidal power injected ( $\overline{W}\cos(\omega t)$) leads to

$$T'(t) = \sqrt{\frac{1}{1+(a_0\,\omega)^2}}\;\frac{1}{h\, S_{wet}}\overline{W}\cos(\omega t+\varphi)\text{, where } a_0 = \frac{M\, c_P}{h\, S_{wet}} \text{ and } \tan\varphi = -a_0\,\omega \tag{3}$$

$\varphi$ corresponds to the phase shift and the term $\sqrt{\frac{1}{1+(a_0\,\omega)^2}}$ characterizes the damping of the system.

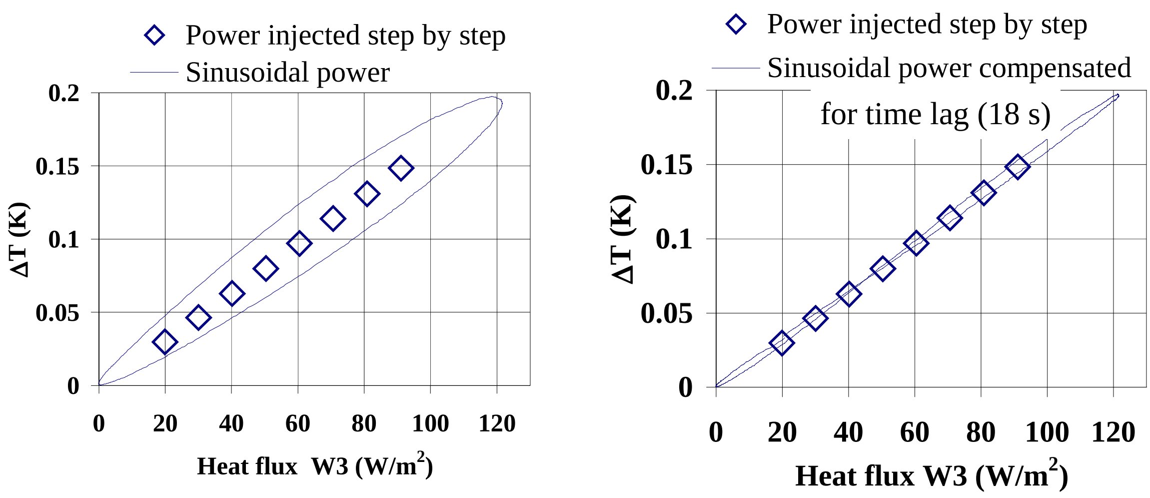


**FIGURE 6** Inlet power W2 100 Watt, Total mass flow 6.8 g/s, T° 1.8 K **(a).** Comparison between steady "step by step" heat flux increases and sinusoidal heat flux **(b).** Same comparison after phase shift adjustment

Typical values in our case are the following:

Volume of the box: 1.048 liter, pressurized He II temperature :1.85 K, and wetted perimeter of 22.5 %, one obtains for a period of 500 seconds a damping factor of 0.976 and a phase shift of 17.6 seconds. As the data acquisition period was equal to one second, the ΔT corrected from the shift phase was directly derived from temperature measurements taken 18 acquisition periods after power measurements.

Figures 6a and 6b show comparison between "step by step" heat flux increases and sinusoidal heat flux without and with rough phase adjustment. Agreement for the last one allows us to calculate the local slope, from which wetting is calculated.

## RESULTS

### Circumferential heat exchange box

This device was first used to check our previous measurements, i.e. the increase of wall heat transfer at high Vgs. Figure 7 shows the reproducibility of our results. The discrepancy for low Vgs is explained by slight differences in slope between the two experiments. This point having been established, the box was used to determine whether the "over-wetting" could be burnt and the corresponding heat flux needed for this.

In order to do this, the wetting was calculated using the local slope of the curve $\Delta T = f(W3)$. For a given heat flux we can assume that a part of it is used to burn the droplets, while the remaining heat flux acts as if there were no droplets.

At 0 Watt, as there was no "extra-wetting", the given wetting shown on figure 7b is almost constant and equal to the value indicated by the capacitive measurements. On the other hand, at high velocity (107 Watt and 1.8 K), the wetting decreases drastically as the heat flux goes from a few W/m$^2$ to 80 W/m$^2$ after which it reaches a plateau. However, the value of this plateau is not compatible with that obtained by capacitive measurements.

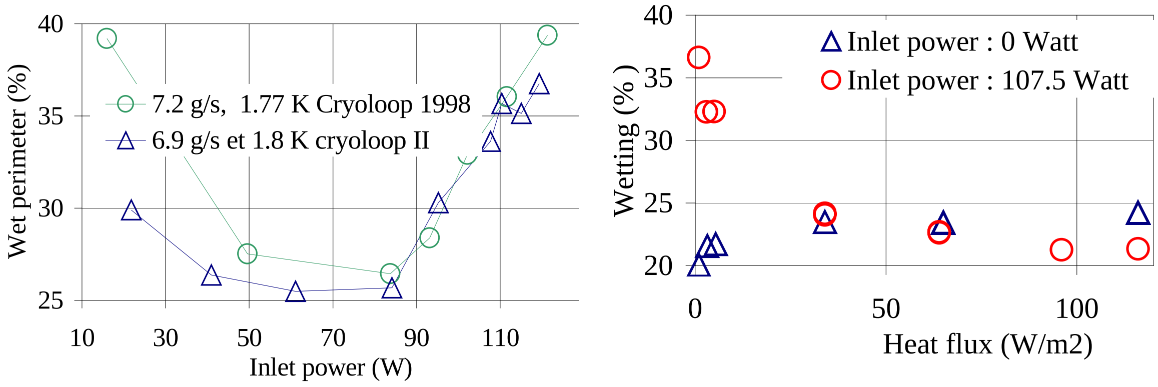


**FIGURE 7 (a).** Repeatability of heat transfer measurements after the installation of the new Cryoloop (10 m long instead of 20 m) Case of horizontal pipe, with heat flux W3 equal to 20 W/m2 **(b).** Influence of heat flux W3 on heat transfer measurements Case of 6.8 g/s, 1.8 K and 1.4 % slope.

### Local heat exchange boxes

For all our steady state measurements, the liquid level inside the pipe corresponds to a wetted perimeter higher than 15 % and the bottom heat exchange box remains always fully wetted. At low Vgs due to heat losses, the middle and top boxes always remain in normal helium state, which prevents any heat transfer measurements. For high vapor velocities, the fluid in the middle box comes into the superfluid state, whereas the top box remains in the normal state. This indicates the increase in wetted surface when vapor velocity is increased and the stratified nature of the process involved. The value found on the middle box for an inlet power of 100 Watt, a total mass flow of 6.8 g/s and a temperature of 1.8 K corresponds to an inner wetted perimeter equal to 30 %.

### Correlation of results coming from the different sensors

Figure 8 shows calculated wetting derived from different measurement methods. Capacitive measurements show a constant decrease of wetting as inlet power grows. This indicates that no significant thick film creeps along the inner pipe wall. The capacitive measurements are representative of liquid level inside the pipe. Shown also on the same figure is the difference between thermal measurements and capactive measurements, which we count as “over wetting”.

In ref 2 it has been shown that it is possible to convert the optical measurements performed on mist flow at the mid-height of the tube into the liquid-vapor interfacial density ($\Sigma$ ). $\Sigma = n 2 \pi r^2$, where n represents the number of droplets per unit of volume and r its radius. Laser light scattered ($\Sigma$ ) is also plotted on figure 8 after scaling all the data by an appropriate factor. It is noticeable that applying a simple coefficient on optical raw data gives such a good correlation with “over-wetting”.

Another run is plotted on figure 8b corresponding to a new slope of 1.4 %. Behavior is similar to that observed at zero slope. Residual discrepancies could be due to differences in total mass flow and saturated temperature.

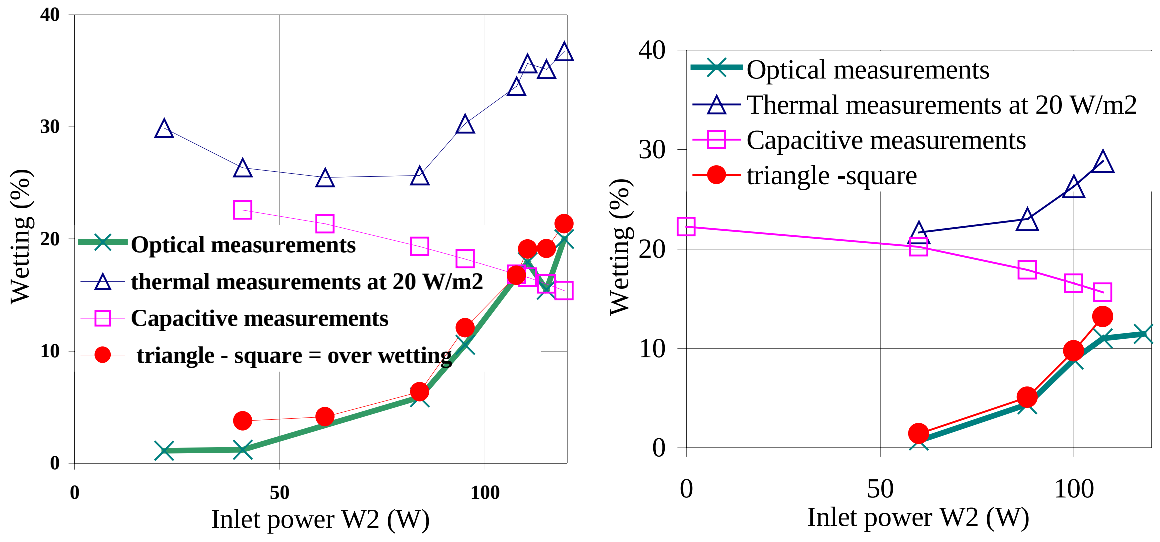


**FIGURE 8 (a).** Comparison between different ways to estimate the wetting in the case of horizontal line, 6.9 g/s and 1.8 K (**b).** Case of 1.4 % slope, 6.6 g/s and 1.8 K

## CONCLUSION

Liquid droplets were found present at high vapor velocities while no thick film was detected. Behavior of the droplet flow is consistent with the increase of wall heat transfer as both phenomena scale with the vapor velocity. However, there are still some apparent contradictions with an extra heat exchange due to liquid droplets, e. g. the difficulty to completely dry out the droplets. Analysis of all the data acquired is still in progress, especially concerning optical measurements.


## ACKNOWLEDGEMENTS

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