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FILM CONDENSATION ON THE UNDERSIDE OF FLAT PLATE WITH AND WITHOUT THE PRESENCE OF NONCONDENSABLE GASES

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Abstract. *The effect of film condensation of pure steam and of mixtures of steam an air streams that reach cooled flat surfaces from beneath is investigated experimentally at atmospheric pressure. Different amounts of air concentration, surface sub cooling temperature levels and plate inclination are tested. The tested surfaces have different lengths, according to the plate inclination angle, in order to keep the same projected area in the direction of the stream. The heat transferred is measured by the stream mass flow rate and the temperature increase of the plate cooling water. This same information is obtained measuring the amount of condensate obtained times de latent heat of condensation. This heat transfer data is also compared with that obtained from the Nusselt model. It is observed that the two phase boundary layer, caused by the presence of non condensable gases close to the condensing surface, presents a lower effect in the decrease of the heat transfer rate when compared with free convection condensation*

Keywords*: Filmwise Condensation, Flat Plate, Noncondensable Gases, Inclined Surfaces*

1. INTRODUCTION

Condensation processes can be applied in a variety of engineering industrial and domestic applications such as air dryers, heat transfer devices, water recovery in cooling towers, etc. Therefore, the heat transfer mechanisms, as well as the condensate film formation induced by the condensation phenomena, have attracted a lot of research interest. Since the first work in the field of theoretical analysis of filmwise condensation developed by Nusselt in 1916, a number of workers such as Rohsenow [1], Sparrow and Gregg [2], Chen [3] and others have improved Nusselt's model, by removing some of the original restrictive assumptions.

The dropwise condensation has received attention from the researchers around the world due to its higher heat transfer coefficients when compared with the filmwise condensation. The main problem of this mode of condensation is related to the difficulty of keeping the drops formation for a long time. After a while, the surface becomes wet and the drops have problems in emerging again. Many authors have directed their researches to the development of new coating materials, different kind of substrates or other deposition methods, in order to provide to the surface a poor wetability characteristic and though obtain dropwise condensation for a longer period [4-10].

Although it is desirable to achieve dropwise condensation for industrial applications, it is often difficult to maintain these conditions. For this reason, porous medium in the filmwise condensation also has been investigated [11-20]. According to Wang et al. [17], who studied the condensation over a wavy surface covered by a porous medium, significant enhancement in the heat transfer can be reached with the increase of the waviness, due to the increase of the contact area. Although they did not consider the effects of capillary forces induced by the porous medium, Tong-Bou Chang [18] showed that the porous media effectively enhances the heat transfer performance of the surface. Renken and Raich [20] report 250% of condensation enhancement on a parallel flowing steam over a surface with a variety of thin porous coating (0 to 254 µm).

It is well know that the presence of non-condensable gases (NCG) in steam, even in small amounts, reduces drastically the condensation process due to the interface temperature decrease. Bum-Jin et al. [21] and Xue-Hu Ma [22] compared filmwise and dropwise condensation of water in a vertical flat plate with the presence of NCG over a range of 0 to 6% of NCG mass fraction. The gas boundary layer is destroyed due to the liquid interfacial dynamic in the dropwise condensation and, consequently, the effect of NCG in this mode of condensation is lower than for the filmwise condensation. One way to deal with this effect in the filmwise condensation mode is increasing the liquid film Reynolds number, so that the liquid film interface becomes wavy. Park et al. [23] conducted a series of experiments of filmwise condensation over a vertical flat plate in the presence of NCG on the wavy flow region. According to them, for a given air-mass fraction and a water vapor-gas mixture velocity, the overall heat transfer increases with the increment of the liquid film Reynolds number. Moreover, A. M. Zhu et al. [24] studied the effect of the condensation heat transfer rates under high amounts of NCG for a variety of vertical tube lengths. One of their results is that, as the Reynolds number of the mixture increases, the effect of the NCG in the condensation decreases. According to these authors, the gas boundary layer is destroyed by the mixture velocity, so that the vapor can reach the surface and condense easily.

While numerous experimental and theoretical investigations on condensation heat transfer enhancement in flat plates and pipes have been studied, only a few experimental researches were conducted on the counter-current mixture flowing over the condensate liquid film. Gerstmann and Griffith [25] worked on the condensation effect of FREON-113 on the underside of a cooled copper surface in different inclinations. Bum-Jin Chung et al. [26] conducted a series of experimental tests of condensation in this physical configuration under natural convection and with the presence of NCG varying between 0 to 6% for water vapor.

In the present paper, the effect of condensation heat transfer under forced convection in counter-current flow over the underside of a condensing plate under different inclinations, for several NCG mass fraction concentrations, is reported.

2. EXPERIMENTAL APPARATUS AND PROCEDURES

The experimental facility developed in this work consists of a test section and auxiliary equipments such as steam generator, cooling water system, NCG supplier and data acquisition system, which are grouped in three main sections: boiler, vapor supply line and test section. All the parts of this equipment are insulated with 50 mm rockwool to avoid energy loss to the environment.

The boiler consists of a container where water is heated up to its vaporization. It is made of galvanized carbon steel with 2 mm of thickness and with 600x500x270 mm of dimensions. The heating is provided by means of four electrical heater modules, which can be controlled (on or off), comprising three heaters each, that supply controlled heat power to the water to be vaporized. A level gauge is used to identify the level of water inside the boiler. Fig. 1 shows the schematic of this part of the device.

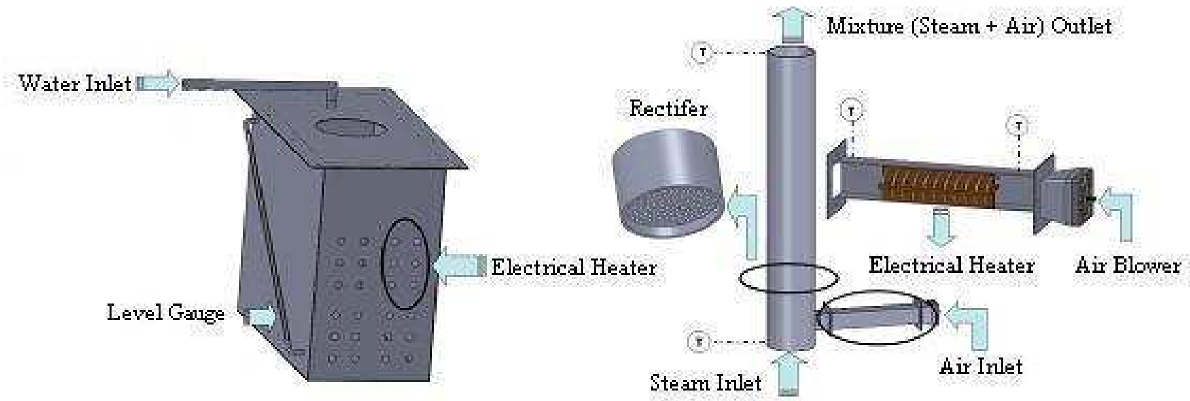

Figure 1. Schematic of the boiler (left) and the vapor and air supply system (right).

The vapor supply line, which also is shown in Fig. 1, consists of a main cylindrical vertical column and a small square section horizontal duct. Both ducts are made of galvanized carbon steel with 2 mm of thickness. The main column has 147 mm of inner-diameter and 1000 mm of height, whereas the square duct has 50 mm cross section internal side and 300 mm of length.

The square duct, which is connected to the column, is in charge to deliver the controlled amount of NCG gases to the stream. The NCG gas employed is the atmospheric air, which is introduced to the main cylinder by means of an air blower. Inside this duct, an electrical heater is responsible to heat this air to the same temperature of the steam generated by the boiler, so that the water vapor is kept in a dry saturated condition. Two k-type thermocouples were installed in the beginning and in the end of this square duct. So, knowing the power supplied to the air, which is made by means of a controlled electrical source, the thermal capacity of the air and the temperature difference measured by the thermocouples, the main column inlet NCG mass flow rate can be obtained through the following equation:

$$m_{NCG} = \frac{Q}{Cp_{air}\Delta T} \tag{1}$$

where Q is the heat provided to air through the electrical heater, Cp_{air} the heat capacity and ΔT the temperature difference measured by the thermocouples.

The main column receives the steam from the boiler located in its bottom and the controlled NCG just above the steam inlet. Just after the entrance of these two fluid lines, there is a flux rectifier that is in charge to organize the velocity profile and to aid in the homogenization of these two components. Two k-type thermocouples are located in the inlet of this main column, to make sure that steam produced by the boiler is in a saturated condition for the atmospheric pressure. Two more thermocouples are installed in the outlet of the column to guarantee that the mixture in this region has the same conditions as it is provided, in the bottom.

Therefore, the vapor and air mixture comes from the bottom of the vapor supply cylinder and leaves its top opening, reaching the condensing surface to be studied. The testing surface is installed over the lower side of a small heat exchanger, made by a hollow thermal insulated metallic parallelepiped box. Inside this box, cooling water circulates at controlled rates and temperatures, so that the amount of heat removed inside the box is known. As heat is removed from the surface, water vapor condensation happens in its face. As this cooling surface is not in horizontal position, the resulting condensate is drained and collected by means of gravity. The residual mixture flows through the lateral gaps (between the tested surface and the test section vertical walls) and reaches the environment at the top of the box.

One vertical glass wall is provided to the text section box to allow the visualization of the condensation phenomena over the tested surfaces. The other walls are made of galvanized carbon steel with 2 mm of thickness. The small heat exchanger, the test surface and the insulation are fixed by a pin located in the lateral sides on the metallic walls of the test section, allowing the variations of the slopes (see Fig. 2).

The heat exchanger cooling water is provided by a water bath, keeping the device at a controlled temperature level and allowing the study of the influence of sub-cooling temperature levels in the condensation. The cooling water mass flux is measured by a rotameter. Seven k-type thermocouples, connected to the data acquisition system, are designed to be embedded below the condensing surface, in order to measure the surface temperatures.

Fig. 2 shows the condensation surfaces to be tested. All the surfaces are rectangular and planar with 10 mm of thickness and 100 mm of horizontal length. As already mentioned and shown in Fig. 2, the surfaces are tested under different inclinations. So, different lengths of the surface were tested for each inclination angle in order to keep the same projected area in the direction of the steam and air mixture that hits the surface. The first surface of Fig. 2 has 116.2 mm of length and is tested in a condition of 30 degree of inclination in relation to the horizontal axes. The next surface has a length of 142.8 mm and is tested in 45 degrees, while the last one presents 200 mm of length and is tested at 60 degrees of inclination.

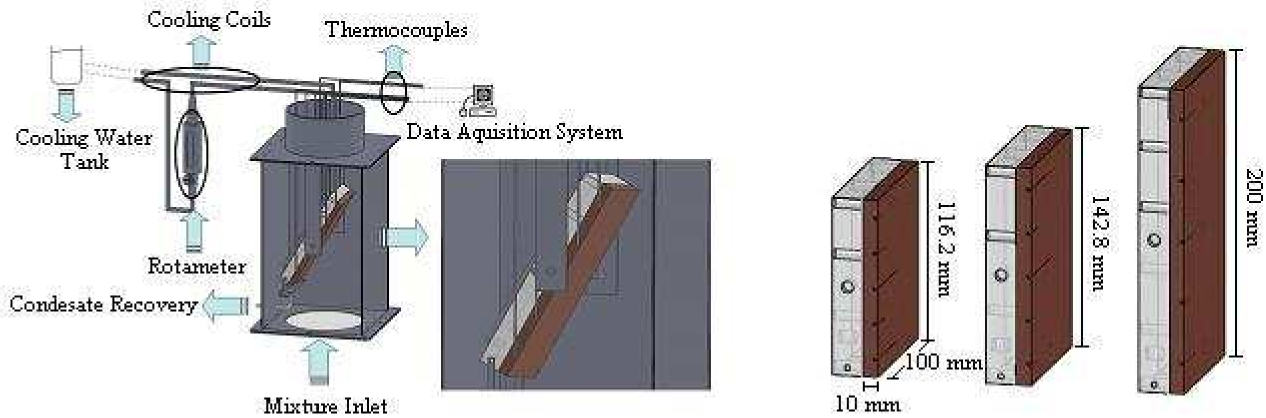

Figure 2. Testing box and tested surfaces.

According to some results from the literature [25 and 26], systematic reductions in the heat transfer rates are observed as inclination angles approaches to zero (horizontal position). This behavior was deducted theoretical for the case of a quiescent vapor condensing on a flat plate. Although Gerstmann and Griffith [25] and Bum-Jin Chung et al. [26] studied the condensation on the underside of flat plates in different inclinations using the same surface and, by modifying the mixture of water vapor and air flow conditions through deflectors, they changed the attack angle of the stream over the tested surface, so that the effect of vapor shear stress at the liquid interface could be neglect.

In the present study, it is observed, during the tests, that pure steam or the mixture of steam and NCG reach the surface with a considerable force, so that, normal and shear stress are noted .

Figure 3 illustrates the entire apparatus composed by the boiler, vapor supply line and test section ready for testing.

The tests were carried out at atmospheric pressure with a mass flow of 2.4 g/s and a mass fraction from 0 to 50% of NCG. The surface temperature varied between 35 and 85ºC. The heat transfer rates are evaluated by three ways for the pure steam case and by two ways for the NCG mixtures. The heat removed from the condensation plate by the cooling water was determined by:

$$q^{"} = \dot{m}_{water}\, Cp_{water}\left(T_{out} - T_{in}\right) \tag{2}$$

where m_{water} is the mass flow rate of the cooling water, Cp_{water} is the specific heat of the cooling water and T_{out} and T_{in} re the outlet and inlet temperature of the cooling water, respectively.

The second way to check the heat removed from the condensing surface is by measuring the liquid mass of condensate and the latent heat of condensation using the equation:

$$q^{"}_{condensate} = \dot{m}_{condensate}\, h_{vl} \tag{3}$$

where $m_{condensate}$ and h_{vl} are the condensate liquid mass rate collected from the plate and the latent heat of condensations of water, respectively.

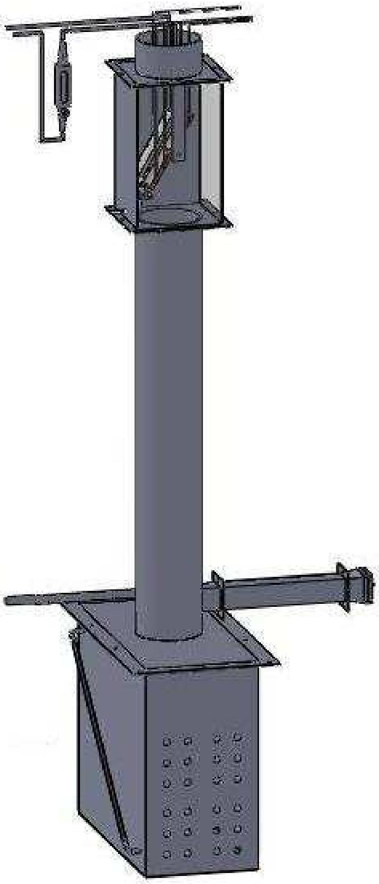

Figure 3. Entire experimental apparatus.

For the pure water vapor case, the Nusselt flat plate condensation analysis, reevaluated by Rohsenow [1], also was used to estimate the condensation coefficient of heat transfer *h*, which is given through the following equation:

$$h = 0.943\left[\frac{\rho_l\left(\rho_l - \rho_v\right)g\sin(\theta)\kappa_l^3\left(h_{lv} + 0{,}68Cp_l\Delta T\right)}{L\mu_l\Delta T}\right]^{\frac{1}{4}} \tag{4}$$

where ρ_l and ρ_v the density of the water and vapor, g the gravity acceleration, κ_l the thermal conductivity of water, h_{vl} the latent heat of condensation, ΔT the sub cooling temperature of the wall, μ_l the viscosity of water, L the length of the condensing plate and Cp_l the specific heat of the water. This model was developed for pure saturated quiescent vapor in contact with cooled condensing surface and will be used as a benchmark for the experimental data.

An uncertainty analysis was performed using the methodology suggested by Holmann [27]. The vertical bars on the data points in Figs. 4 to 9, represents these uncertainties. One can see that the uncertainties varied from 10 to 50% for the determination of the heat transfer rate using Eq. 2 and it is always less than 5% using Eq. 3. Larger uncertainties associated with Eq. 2 are observed for higher temperatures as the difference between the inlet and outlet cooling temperatures is very small. Also the mass flow rate is obtained using a rotameter and the uncertainty is associated with the measurement equipment. On the other hand, the uncertainties obtained using Eq.3 are smaller since, since in this case, the mass flow rate is determined by the collection of condensate for a long time period. Both the uncertainties of the timer and of the weighing scale are much smaller than the values of the measurement obtained form this devices. By this reason, the error bars for the heat measured by Eq. 3 is difficult to note in figures that follow.

3. EXPERIMENTAL RESULTS AND DISCUSSION

Fig. 4 shows plots of the heat rate, given by Eqs 2, 3 and 4, as a function of the surface temperature for the vertical case (90º), for pure vapor streams. The blue line represents the results obtained by the Nusselt analysis (Eq 4), while the red solid and open symbols represent the results obtained through Eqs. 2 and 3, respectively. The heat transferred obtained from Eqs. 2 and 3 shows a good agreement but both present a large disagreement with the Nusselt theory. This can be explained analyzing the test conditions which are quite different from the hypothesis adopted for the Nusselt model. In the present experimental study, the vapor was not quiescent, especially in the region surrounding the condensing surface. As it can be seen in Fig. 2 and 3, the test section is fully opened in the vapor inlet and outlet, which causes a large vapor flow. As the only obstacle of the moving vapor stream is the condensing surface, the part of the steam and air flux that does not reach this cooled surface, escapes by the openings between the sides of the plate testing box walls and also there is the stress caused by the vapor flow in the falling film, moving in counterflow directions. According to Fahgri [28], heat transfer rates, in filmwise condensation within a vertical thermosyphon, tends to decrease as the vapor velocity increases. The disturbance caused by the vapor shear stress at liquid-vapor interface tends to thicken the liquid film. Moreover, when the liquid film regime of flow becomes wavy, the interfacial shear tends to tear droplets of the wave crest, sending them upward to strike the end cap of the thermosyphon. Still in reflux condensation, Chen S. L. et al. [29] say that the phase change over the condensation liquid film also contribute for the augmentation of the shear stress. Although thermosyphon configurations are different from flat platesconsidered in this paper (in the thermosyphon case the space through which the vapor and the condensation falling film flows in counter current is narrow) the effect of the gas stream over the condensing film still can be observed in the present case, due to the position of the cooled condensing surface, which faces down and receives the vapor flow directly. Therefore, normal and shear stress are expected in the liquid-vapor interface, leading to decrease of the heat transfer coefficient, similary as observed for thermosyphons.

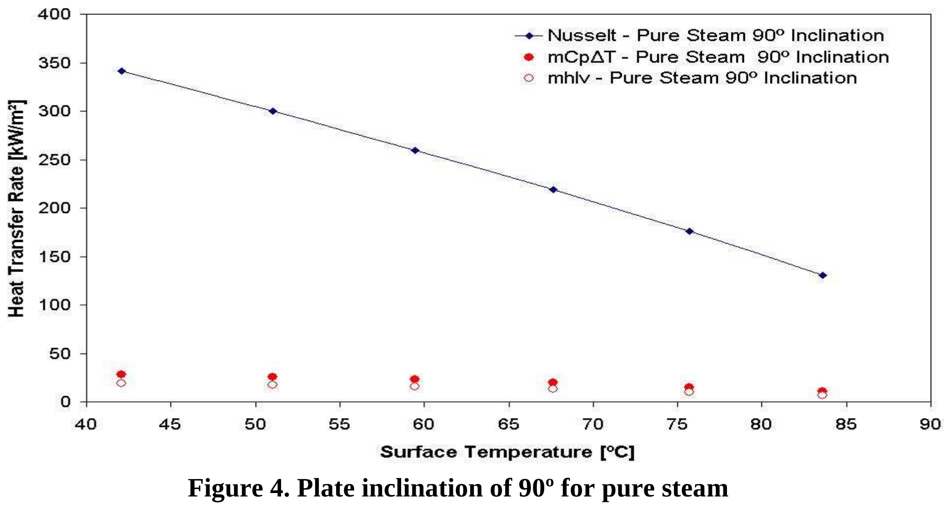

Figure 4. Plate inclination of 90º for pure steam

In Fig. 5, plots of the heat transfer rate as a function of the surface temperature are presented. The solid symbols represent the heat transfer measured by Eq. 2, while the hollow symbols represent with Eq. 3 data. One can observe that the heat transfer rate decreases, both for the vapor pure case and for the NCG and vapor case, as the surface temperature increase, as it is expected. The heat transfer rates also decrease systematically, when the mass fraction of air increase. In this case, the decay of the heat transfer rates by the presence of NCG is caused by the build up of NCG at the liquid interface, resulting in a reduction of the partial pressure of the steam, decreasing the saturation temperature, in which the condensation takes places. This behavior, for moving mass flow configurations, is not affected so hard as for the case of a quiescent mixture of vapor and NCG, as it was reported by the theoretical work conducted by Minkowycz and Sparrow [30] and Rose [31]. In their theoretical work, even with 0.5% of NCG mass fraction, the heat transfer could decrease to about 50% when compared with the pure steam condensation. Due to the test configuration, the reduction of the heat transfer rate for 20% of NCG mass fraction was about 20% of the pure vapor case, for the same range of wall temperatures, and about 40%, for the 50% of NCG mass fraction. This can be explained by A. M. Zhu [24], who states in his work that the velocity of the flowing mixture hitting the liquid film, helps spreading the NCG boundary layer over the liquid film interface, improving the heat transfer when compared to pure vapor condensation flows.

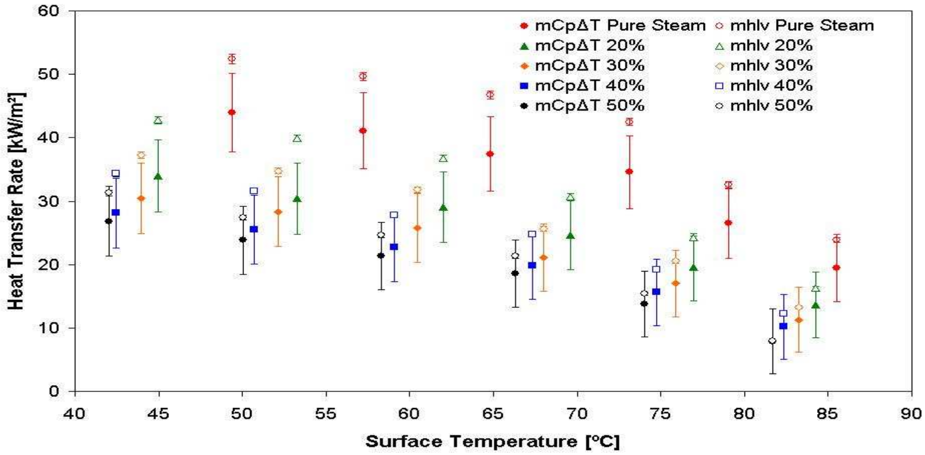

Figure 5. Plate inclination of 30º

Similar trends were observed for the cases where the plate presents the inclinations of 45º, 60º and 90º, shown in Figs. 6, 7 and 8, respectively. From these figures, one can see that, as the plate inclinations approach the vertical case (90º), the heat transfer rates decrease, for all condensing wall surface temperatures. In this case, the gravity, which is the force that pulls down the liquid film, does not play as an important role as in the case for a quiescent condensation vapor, adopted in the Nusselt theory. Due to the configuration of the condensation process of this work, the condensation heat transfer rates were higher for 30º than for the vertical case (90º). Actually, for small inclinations, the testing surface blocks and hold much more vapor than for the case with higher inclinations, increasing the heat transfer

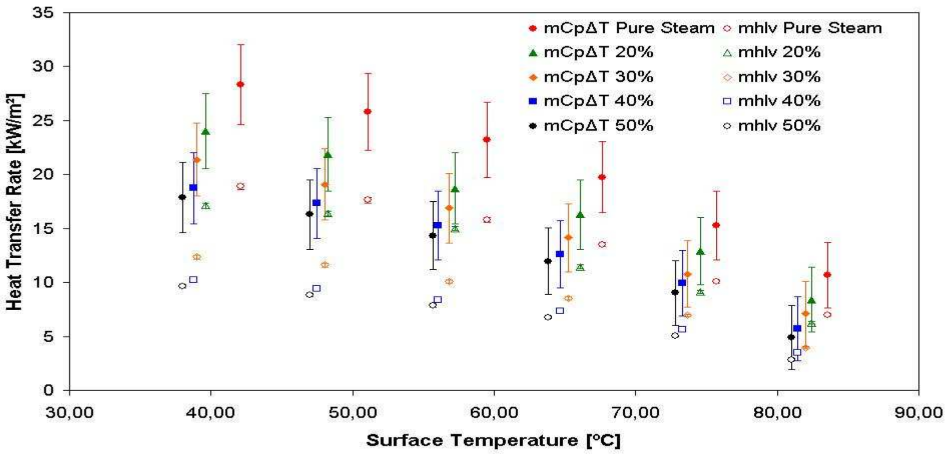

Figure 6. Plate inclination of 45º

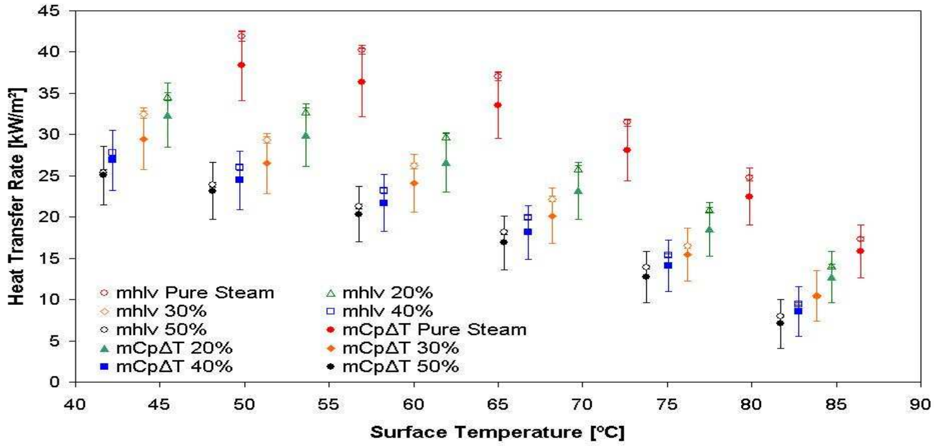

Figure 7. Plate inclination of 60º

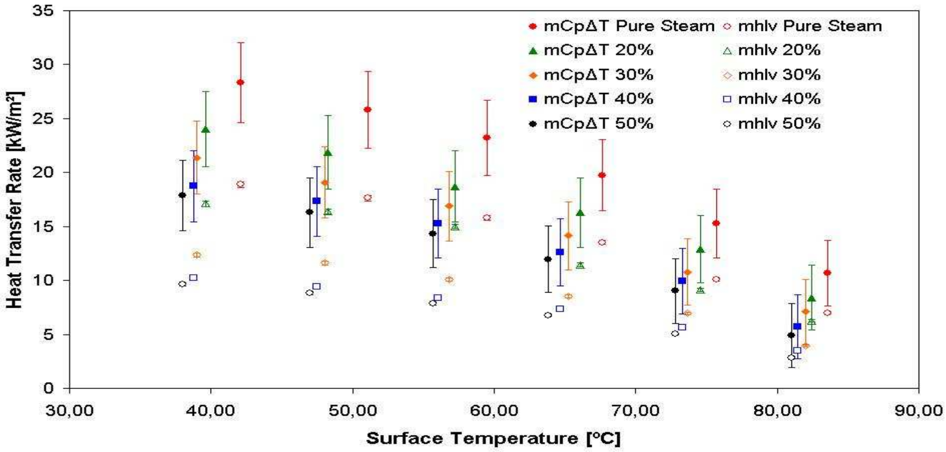

Figure 8. Plate inclination of 90º

Fig. 9 shows the results for the heat transfer rates, for pure vapor condensation obtained using Eqs. 2 and 3, as a function of the surface temperature and for different inclinations. In this graphic, is more evident that heat transfer rates are higher with some inclinations when compared with the vertical case, which does not match with the predictions of is the Nusselt theory.

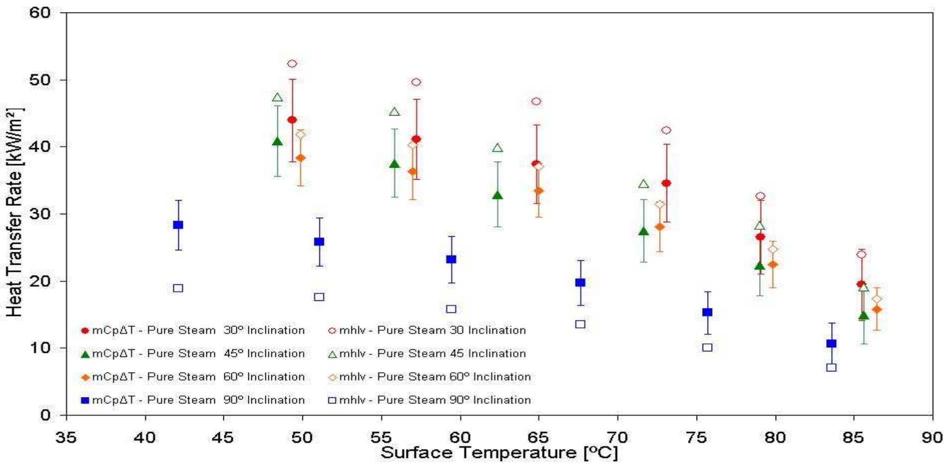

Figure 9 Pure Vapor

For all the data presented in Figs, 4 to 9, Eqs. 2 and 3 predict similar values of the heat transferred observing the cooling plate water flow mass and temperatures differences and by the amount of the condensate. This shows that the experimental apparatus is well designed and provide good data. Therefore, through these two methods one can double check of the test heat transfer results.

On the other hand, both the results provided by Eqs 2 and 3 show different trends for the vertical plate configuration when compared with the inclined cases. The heat transfer results obtained using Eq. 3 are always a little higher those obtained with Eq. 2 for the inclined plate cases. This trend is reversed for the vertical case. It is believed that this is because there is a certain amount of vapor that just heats the plate and are not condensated.

4. CONCLUSIONS

A set of film condensation experiments were carried out to study the effects of plate inclination, air mass fraction and sub cooling levels of the condensing surface. The condensation occurs in counter current forced convection film. The pure vapor or the mixture of vapor and air, always at atmospheric pressure, flow freely over the cooled surface. .

The test results for the pure vapor condensation showed a great disagreement with the Nusselt heat transfer rates, showing that the hypothesis of quiescent vapor in a region close to the condensing surface, cannot be used in the present experimental work. The calculation of the heat transferred, based on the heat removed by the water cooling system agrees very well with prediction based on the condensate collected, showing that the experimental apparatus is well designed and provides good data. The discrepancy of the data with the Nusselt analyses theory is caused by the vapor free flow over the cooled surfaces and the stress it provokes over the liquid film. The Nusselt theory also shows different effects of the inclination angles than observed in the data, where systematic increases of the heat transfer rates are observed when the angle from the horizontal reduces, unlike predicted by Nusselt, for the quiescent condensation case.

The presence of NCG showed to be an important effect in the decrease of the heat transfer rates. But the decrease in the heat transfer rate with the increase of the NCG in the vapor stream was lower than observed for free convection cases.

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