Dropwise Condensation Experiments with Humid Air at a Polymer Surface

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Abstract. A new test facility has been developed to investigate dropwise condensation heat transfer in a humid air environment. It is designed as a closed loop system in which air is circulated by a fan, enabling investigations in the following parameter ranges: velocity up to 20 m/s; Reynolds number up to 20,000; temperature 20 to 100 °C; relative humidity up to 100 %. Heat transfer measurements are done with a specifically designed micro sensor which is flush mounted at one of the vertical surfaces of a horizontal flow channel 12 mm x 32 mm (inner width and height, respectively) and covered at its air-side surface by a newly developed polymer layer containing 20 % of carbon nanotubes for improvement of the thermal conductivity. A total of 8 thermocouples is embedded inside the sensor. Their readings serve as input data to a numerical model which enables consideration of heat losses and evaluation of surface temperature and heat flux. The measuring system allows to analyse the effects of heat flux, air-to-wall temperature difference, absolute and relative humidity, and Reynolds number on the heat transfer coefficient. Single phase heat transfer results show excellent agreement with well established correlations for turbulent air flow. The onset of dropwise condensation was detected with very good repeatability. This paper covers details of the experimental device, measuring system and data evaluation including accuracy considerations. Single phase and preliminary dropwise condensation results with humid air are reported.

1. Introduction
Dropwise condensation of pure vapor is characterized by extremely large heat transfer coefficients which exceed those of the filmwise mode by one and more orders of magnitude. Hence it is not surprising that numerous attempts are reported which tried to establish stable conditions for this thermodynamic effect. Small contents of non-condensable gases, however, impede the vapor side transport process by an additional resistance for mass transfer from bulk to the phase interface resulting in a strong decrease of heat transfer coefficients. The difference between filmwise and dropwise condensation is diminished by this and the attractiveness of the latter mode is widely lost. Nevertheless humid air shows a big potential for energy recovery and it brings a completely different point of view. Increasing the humidity from zero, the poor single-phase heat transfer coefficients are found to be strongly improved at the onset of dropwise condensation. The literature offers only a few investigations of this promising effect. In 1956 Heyser [1] and Kirschbaum & Lipphard [2] studied humid-air condensation outside a vertical tube and they reported graphical representations of the various parameter effects. They found transition from dropwise to filmwise condensation when the heat flux, the air-to-wall temperature difference and also the liquefied mass flux exceed some limiting
values. They concluded that cooling of humid air at temperature differences below 15 K is of negligible interest. No further studies occurred until 1995 when van der Geld and his group [3-7] started to investigate dropwise condensation in polymer heat exchangers. Another recent publication comes from Rausch et al. [8] who studied drying of humid air in a heat exchanger made from ion implanted aluminium surfaces for getting dropwise condensation. Today, low temperature heating systems and plate heat exchangers with small temperature differences are state of the art pushing the interest in dropwise condensation of humid air.

2. Measuring system

Fig. 1 shows the experimental set-up with details of operation, measurements and visual observation tools. After assembling and testing of all measuring devices the entire system has finally been encased by thermal insulation. Humid air is circulated by a fan to an outer annular flow path which serves as a thermal protection system surrounding two concentric inner tubes. The air flow is then forced to return to the left hand side being separated into two parts, namely into an annular outer bypass which includes a valve for controlling the flow rate and the main flow through the center path containing the measuring system.

Figure 2. Sensor for heat transfer measurements with thermocouples and details of the arrangement
The core of the test facility consists of a horizontal flow channel 12 mm x 32 mm (inner width and height, respectively, see Fig. 2) with the cooled surface being flush mounted at one of the vertical side walls. The measuring system consists of a block sensor. It is made from steel with a central cylindrical rod surrounded by three tube-like jackets serving as protection shields. This arrangement is water cooled from the backside (right-hand side in Fig. 2) by using a water supply system from brass with a central inflow port and an annular system of small openings for the outflow of cooling water. An additional ring-shaped cooling channel surrounding the steel arrangement acts as an additional heat sink. This system is designed for getting a zero radial heat flow between the central rod and the first annular section providing adiabatic conditions for the heat flow measurements. Each of both front surfaces carries four embedded thermocouples. The radial temperature difference on the left hand side is used to control of adiabatic conditions, the difference from front to back side of the sensor serves for the heat-flux evaluation. The air-side surface is covered by a thin layer of polypropylene filled with 20 % Carbon Nanotubes with increased thermal conductivity [9].

The test arrangement enables condensation experiments with variations of surface temperature and heat flux, and furthermore variations of vapor temperature, humidity and velocity.

The mass flow rate of humid air is measured by a flow meter (type ST98, error within ± 2%) from Fluid Components International LLC, San Marcos. For determination of the thermodynamic state of the humid air entering the test section, the following properties are measured: pressure (PAA-35XHTT, Keller AG, Winterthur, ±1.5 hPa), temperatures (TMH GmbH, Hanau, ± 0.1 K), temperature of the dew point (MTR 2.1/SA 25 XU, CiS Forschungsinstitut für Mikrosensorik und Photovoltaik GmbH, Erfurt, ± 0.2 K). For the block sensor temperature, K type mineral insulated thermocouples are used with enhanced accuracy (TMH GmbH, Hanau, 0.5 mm, ≤± 0.01 K).

Dropwise-condensation experiments have been carried out within the following limits of the main parameters:

| Parameter                      | Range                      |
|--------------------------------|----------------------------|
| Humid air velocity             | 5 – 20 m/s                 |
| Mass flow                      | 8 – 29 kg/h                |
| Reynolds Number                | 5,000 – 20,000             |
| Temperature                    | 46 °C                      |
| Absolute humidity (x)          | 0.006 – 0.063              |
| Relative humidity (ϕ)          | 10 – 85 %                  |
| Cooling water temperature      | 22 – 45 °C                 |
| Air to cooling water temperature difference | 6 – 24 K                  |
| Air to wall temperature difference | 5 – 18 K                  |
| Heat flux                      | 500 – 3,500 W/m²           |

3.  Results and Discussion

3.1. Evaluation of data

The measurements have been evaluated for getting the following data:

a) Temperature and heat flux at the sensor surface:

Surface temperature and heat flux of the sensor can be evaluated from geometry, thermal conductivity and measured front and back side temperatures by application of Fourier’s law of heat conduction on both, the steel core of the sensor and the covering polymer layer. This is only possible for zero heat flow between the core and the surrounding protection shields. Preliminary experiments showed remaining heat losses, even for the case of zero radial temperature difference. This was the reason for the numerical solution of the inverse heat conduction problem using the measured temperatures as boundary conditions. Fig. 3 shows some results, obtained for various radial temperature differences between the inner two front side measuring locations (see Fig. 2) taken at 3 different Re numbers. The red and black lines represent numerical results for assumed cases with and without consideration of radial heat exchange, respectively. There is an obvious difference for the case of zero temperature
difference. The crossing point is found between about 0.005 and 0.010 K, valid for the temperature difference between the inner two front side temperature readings. Based on this experience, all measured data have been evaluated by application of the numerical model.

b) Heat transfer coefficient, Re and Nu numbers

\[ \alpha = \frac{\dot{q}}{\Delta T_{\text{air-wall}}} \quad \text{Re} = \frac{\dot{m}}{\mu_{\text{air}}} \frac{d_h}{A} \quad Nu = \frac{\alpha d_h}{\lambda_{\text{air}}} \]

where

- \( \dot{m} \) ... measured mass flow rate
- \( d_h, A \) ... hydraulic diameter (\( d_h = 17.45 \text{ mm} \)) and cross section of the inner channel
- \( \mu_{\text{air}}, \lambda_{\text{air}} \) ... dynamic viscosity and thermal conductivity of humid air, at \( T = (T_{\text{air}} + T_{\text{wall}})/2 \)

3.2. Consideration of uncertainties

The error analysis and calculation of uncertainties for the test rig was done according to the DIN 1319. The guideline contains rules for computation of different systematic and a coincidental contribution which occur in the measurement of individual measurands. In general the formulas are:

\[ u_{\text{rel}}(y) = \left[ \left( \frac{N}{\sum_{j=0}^{N} \left( \frac{\partial f}{\partial x_j} \right)^2 u_{\text{rel}}(x_j) \right) \right]^{1/2} \quad u(y)_{\text{rel}} = \frac{u(y)}{y} \]

Applied to the formulas known from subsection 3.1 it is possible to testify the uncertainty of measurands. Despite that an uncertainty calculation allows to analyze the contributions from single measurands within the measurement chain to the overall uncertainty. Fig. 4 show the expanded, overall relative uncertainty \( u_{\exp, ov, \text{rel}}(\alpha) \) (graph a.) of the heat transfer coefficient. A coverage factor of two was used which provides a level of confidence of round about 95% for normal contribution. Furthermore the figure contains individual components of \( u_{\exp, ov, \text{rel}}(\alpha) \). These individual components can be obtained by zeroizing uncertainties coming from individual measurands.
The figure 4 illustrates that at a temperature difference above 0.3 K the expanded overall uncertainty decreases almost logarithmically. Obviously this is due to the uncertainty in temperature measurement. For temperature differences above 0.3 K across the sensor it can be stated that the overall relative uncertainty of the heat transfer coefficient is better than ±6.5 %. The overall relative uncertainty of the heat transfer coefficient is better than ±4.5 % if the temperature difference exceeds 0.8 K.

3.3. Effect of heat flux and driving temperature difference from air to sensor surface

Fig. 5 shows heat flux effects on the heat transfer coefficient for some Re numbers (in the turbulent range) and with the absolute water content ranging up to 0.06 kg\textsubscript{vapor} / kg\textsubscript{air}. Starting the experiments from zero heat flux, only single phase convection is obtained with heat transfer coefficients which are almost independent of the heat flux, and they increase with the Re number, as expected. At some critical intensity of cooling, the heat transfer coefficient starts to increase which is a clear indication for beginning dropwise condensation. This has been confirmed by visual observations. The related critical heat flux decreases when the absolute humidity is raised from zero (at constant Re number).

Similar results are seen for the effect of the air-to-wall temperature difference as plotted in Fig. 6. Onset of condensation is obtained at driving temperature differences $\Delta T_{\text{air-wall}}$ of about 13 K at absolute humidity $x$=0.04, 9 K at $x$=0.05 and even smaller (maybe around 3 K) at $x$=0.06 which has, however, been outside the present experiments.
3.4. Single phase convection

As already seen there is a clear and expected Re number effect on single phase heat transfer. Respective results are plotted in Fig. 7 as Nu versus Re number together with the well recommended correlation for turbulent heat transfer taken from VDI-Wärmeatlas [10]. Excellent agreement is found for Re>10,000 with some deviations below this limit. This latter fact may be attributed to the flow situation which is characterized by a developing thermal boundary layer in a hydrodynamic fully developed air flow. In addition Fig. 7 includes a simplified equation (in red) correlating the measured data within ± 4 %.

3.5. Humidity effects on dropwise condensation

The humidity effect on heat transfer is seen in Figs 8 and 9. Both, the relative and the absolute humidity show insignificant effects in situations without condensation, i.e. at $\varphi<0.55$ in Fig. 8 and $x<0.38$ in Figure 9. Starting from the onset of condensation, a linear relationship is obtained for the humidity effects on heat transfer which is in good agreement with the literature [11]. In the experiments reported in Figs 8 and 9, air velocity (Re=14,000) and temperature ($T_{air} = 46 \, ^{\circ}C$) were kept constant with some variation of the driving temperature difference (most of the data were taken at $\Delta T_{air-water} = 16.7 \, ^{\circ}C$). For small vapor contents (i.e. $\varphi<0.55$ and $x<0.38$, respectively) the air is far below saturation, and condensation at the sensor surface requires a large air-to-wall temperature difference $\Delta T_{air-wall}$. Its critical value decreases with rising humidity. Beyond that point, the intensity of droplet formation (in terms of the condensed mass per time) is increased with the driving force, which may be expressed as the difference of the absolute humidities in the bulk and at the air-to-droplet interface ($\Delta x = x_{air} - x_{vap-liq-interface}$). The latter one does not deviate too much from the saturation state at wall surface temperature $T_{wall}$. It decreases for rising the air to cooling water temperature difference $\Delta T_{air-water}$ and simultaneously the heat flux. This effect is visible in Figure 8, when $\Delta T_{air-water}$ is increased from 15.5 K to 24 K for constant relative humidity $\varphi = 0.58$ showing a clearly improved heat transfer.

3.6. Re number effects on dropwise condensation

Fig. 10 shows condensation heat transfer coefficients plotted versus the Re number for two different absolute humidities and four different air-to-water temperature differences. Single phase results are not included.

Dropwise condensation is characterized by the growth of single small droplets which occasionally merge to large ones. The entire sensor surface may be subdivided into two different sub-areas which

![Figure 9. Absolute humidity effect (Re=14,000)](image)

![Figure 10. Coupled Effects on heat transfer](image)
are extremely fine dispersed: (a) The first one is wetted, representing the sum of the contact areas of a giant number of single growing droplets ranging from the nanometer to the millimeter scale. The extension of these contact areas depends on the wetting characteristics resulting from a balance of the surface energies of solid, liquid and vapor with an additional effect by gravity and vapor shear stress. (b) The second one may be considered to be dry representing the very small inter-droplet areas. The total heat transfer from air to the surface may be considered to be in parallel:

a) The “wetted path” is composed from three serial steps: (1) Convection transfer from bulk air to the air-liquid interface (surface of the droplets). This is governed by the Re number with parallel convection of heat (\( \dot{Q}_{\text{conv,1}} \), driven by \( \Delta T = T_{\text{air}} - T_{\text{vap-liq-interface}} \)) and mass (\( \dot{m}_{\text{conv,1}} \), driven by \( \Delta x = x_{\text{air}} - x_{\text{vap-liq-interface}} \)). (2) Liquefaction: the arriving vapor is liquefied at the droplet surface releasing the respective amount of latent heat of liquefaction (\( \dot{Q}_{\text{vap-liq}} \)). (3) Conduction of the sum of both heat flow rates (\( \dot{Q}_{\text{cond}} = \dot{Q}_{\text{conv,1}} + \dot{Q}_{\text{vap-liq}} \)) inside the droplet from its vapor side to its wall (sensor) side surface, driven by \( \Delta T = T_{\text{vap-liq-interphase}} - T_{\text{wall}} \).

b) The “dry path” is open for single phase convection (\( \dot{Q}_{\text{conv,2}} \)) directly from the bulk air to the sensor surface, driven by \( \Delta T = T_{\text{air}} - T_{\text{wall}} \). (Comment: going into more detail, one theory is that a thin adhesion layer in molecular scale is assumed to cover the dry patches between individual droplets, see e.g. [11-12] – this has not been considered in this paper).

The relative importance of the various contributions and the underlying effects depend on the various parameters to be studied. The share of the entire surface area available for the “dry path” is assumed to be close to 100 %. This allows a rough prediction of \( \dot{Q}_{\text{conv,2}} \) going this path based on the correlation given in Fig. 7. This has been done for all data points included in Figure 10 with the results plotted in black. The contribution of dropwise condensation to the measured heat transfer (red and green symbols) is obviously given by the difference between respective red/green and black points.

Having this in mind, we can now discuss the results presented in Fig. 10:

a) **Absolute humidity** – Results for two different humidities are included: the small one (\( x=0.04 \), red symbols) enables an only small driving force for the convective mass transfer (\( \Delta x = x_{\text{air}} - x_{\text{vap-liq-interface}} \)). The resulting heat transfer coefficients are low, and they are dominated by the single phase convection path (\( \dot{Q}_{\text{conv,2}} \)) including an only small contribution by condensation. The latter one disappears entirely at the meeting point of red and black data series. At even higher Re numbers only single phase convection is found (not shown in Figure 10). The situation is only slightly improved when the air to water temperature difference \( \Delta T_{\text{air-water}} \) is increased from 15.4 K to 17.1 K – yielding subsequently a decrease of the wall surface temperature, the droplet surface temperature, the absolute humidity and finally an increase of the driving force for the convective mass transfer from bulk to droplet (\( \Delta x = x_{\text{air}} - x_{\text{vap-liq-interface}} \)). Fig. 10 shows the strong effect of an increased absolute humidity of the bulk air up to \( x=0.06 \) (green symbols).

b) **Re number at small humidity** – Decreasing the Re number at the smaller humidity (red symbols) brings the expected decrease of the single phase heat transfer (black symbols) and an increase of the difference, i.e. the condensation contribution. The decrease of the Re number (i.e. convection heat transfer coefficient) at a constant air-to-water temperature difference (e.g. at \( \Delta T_{\text{air-water}} = 17.1 \text{ K} \)) results in an increased ratio of heat transfer resistances by convection and conduction inside the sensor – with a tendency for the sensor surface temperature to
decrease. The obtained result is governed by two contradicting effects: the driving forces for the convective heat and mass transfer processes to the droplet \( \Delta T = T_{\text{air}} - T_{\text{vap-liq-interface}} \) and \( \Delta x = x_{\text{air}} - x_{\text{vap-liq-interface}} \), respectively) will increase, but the corresponding coefficients for heat and mass transfer (\( \alpha \) and \( \beta \)) as weighting factors will decrease with decreasing Re number. Fig. 10 shows that the first ones are dominating in the considered situation.

c) **Re number at large humidity** – At the larger humidity (\( x=0.06 \)) an opposite effect is found – despite the same physical modifications in the transfer processes. This is due to the smaller effect on the driving force \( \Delta x = x_{\text{air}} - x_{\text{vap-liq-interface}} \) for mass transfer which is a priori at a much higher level, allowing the decreased mass transfer coefficient to dominate. The result is a slightly reduced contribution of condensation heat transfer when the Re number is decreased.

4. Conclusions

The experimental investigation of dropwise condensation on the surface of the considered polymers brought the following results:

It is shown by visual observation that dropwise condensation is actually present at the polymer surfaces. Single phase heat transfer has been investigated with some experiments for testing the facility. Heat transfer coefficients have successfully been determined, and they are found to be in excellent agreement with well established correlations for turbulent air flow. Onset of dropwise condensation has been detected with very good repeatability. The effects of heat flux, air-to-wall temperature difference, absolute and relative humidity, and Re number have been investigated in a limited range of the parameters. Further experiments are needed for provision of an enlarged data basis which is required for the formulation of a correlating equation.

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