Experimental investigation of evaporation from horizontal water films

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Abstract

Experimental study deals with two cases of low speed flow in horizontal rectangular duct with mass transfer from thin water film in regime of mixed convection: $Gr_m/Re^2 > 10$ and $0.1 > Gr_m/Re^2 < 10$. Measured water evaporation rate is for both cases compared with correlations found in literature. Discussed is also when the contribution of natural or forced convection can be neglected. For case $Gr_m/Re^2 > 10$, vertical temperature field was measured. Forced convection flow combined with buoyancy induced secondary flow leaded to 3D complex flow with stationary reversed flow below the top desk. Highest temperature fluctuations was measured above the fluid film.

Key-words: mixed convection, mass transfer, evaporation, experiment, criterion equation

1. Introduction

Evaporative processes plays important role in many industrial applications (chemical vapour deposition reactors, evaporative cooling, nuclear spent-fuel disassembly basin loses etc.). Due to complexity of the phenomena and geometries, solutions based on criterion equations are limited to simplified cases only and methods using CFD (Computational fluid dynamics) modelling are being introduced. Problem of such a models is that the usage is not universal and they have to be validated with experimental measurement. For validation purposes, an experimental rig was designed and built. The details of design of the test rig was described in [1]. As validation case, convective mass transfer from water film placed in rectangular duct was selected, the water film is evaporating to low speed airflow.

The aim of this study is to compare obtained data from measurement with correlations of researchers, who studied this topic to make sure that measurements were correct.

The conditions in the channel: airflow with hot water film at the bottom creates mixed convection flow, which can be characterized by ratio Gr_m/Re^2. Gr_m is Grashof mass transfer number and Re stands for Reynolds number. In section 2, mass transfer with ratio $Gr_m/Re^2 > 10$ is examined, section 3 is about mass transfer with ratio $Gr_m/Re^2 \in (0.1; 10)$.

2. Mixed convection with $Gr_m/Re^2 > 10$

Opposite to the experiments found in the literature [2, 3], where balances are used for measurement of depletion of water due to evaporation during 24 hour period, a psychrometric method for measurement of evaporation rate is used for this purpose. The evaporation rate is evaluated from the difference of specific humidity of passing air stream over hot water film. Conservation of mass must be valid in the test section:

$$\dot{m}_{da}(1 + x_{in}) + \dot{m}_{wv} = \dot{m}_{da}(1 + x_{out}) \qquad (1)$$

from which, the evaporation rate $\dot{m}_{wv}$ can be expressed:

$$\dot{m}_{wv} = \dot{m}_{da}(x_{out} - x_{in}) \qquad (2)$$

To correctly evaluate the evaporation rate, the test section of the experimental set-up has to meet requirement on change of specific humidity. If the change would be very small, the error of measurement could be very high. On the other hand, the airflow could become supersaturated, which is the state of moist air that can not be measured by chosen humidity sensors - psychrometers. The design stage of the test rig was covered in [1].

Schematic of apparatus is shown in fig. 1. The water tank's area used for measurements has dimensions of 1000mm x 300mm. The tank is heated by resistance heating foils located at the bottom and on sides of the water tank with possible regulation of input power for maintaining the water at steady temperature. To create a water film, thin aluminium plate is dipped into the tank with 18 digital thermometers Dallas Ds18b20 mounted on the bottom of the plate to monitor the temperature. The sensors are located in a grid of 3x6 for evaluation of uniformity of temperature's horizontal stratification. The air flow is straighten by a nozzle.

Specific humidity is measured by psychrometers, each one consists of two sheathed RTD probes (PT1000) - wet and dry thermometer. The wet thermometer is wrapped by a moisten sock (by distilled water). Two psychrometers are located in outlet pipe for measuring the outlet specific humidity, velocities in the pipe are sufficient to ensure right function of wet bulb thermometer. Psychrometer measuring inlet specific humidity is located above the entrance of the nozzle and the air current for wet-bulb thermometer is propelled by external fan.

The mass flow rate is measured by orifice plate. Orifice plate is a thin plate with a hole in it, centred rounded in this application, designed by CSN EN ISO 5167-2 [4]. It is used for reducing static pressure, which is measured by differential pressure transducer Setra 265. The orifice plate is located in the outlet pipe (fig. 1). The pressure drop was designed to 85Pa for velocity $0.1 m \cdot s^{-1}$ in the test section (due to contraction after the test section, the velocity in front of the orifice plate is approx. $4.6 m \cdot s^{-1}$).

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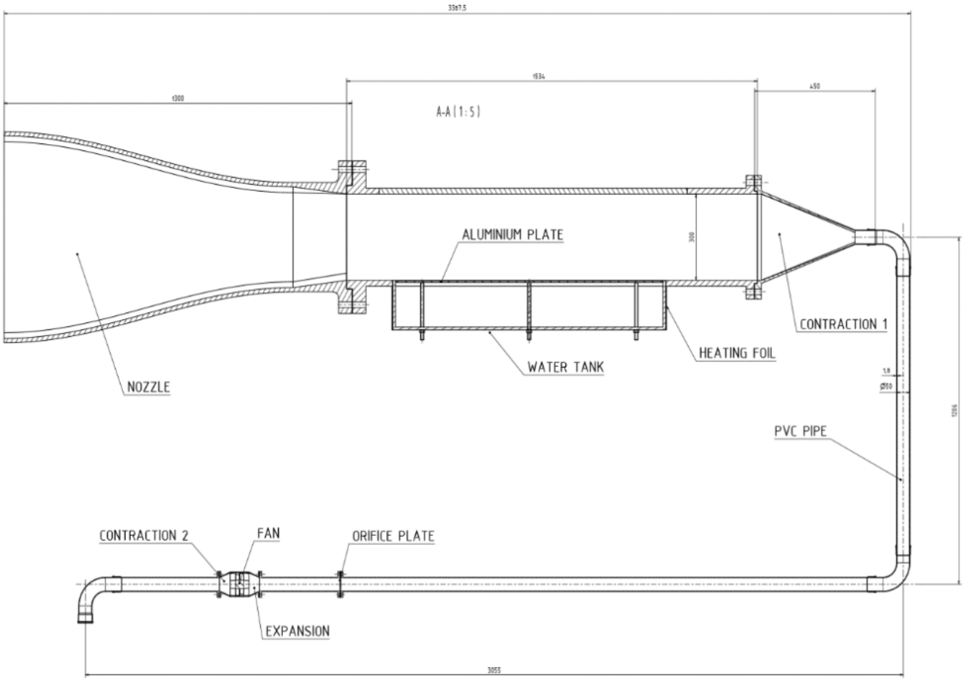

Fig. 1. Schematic of test rig with main dimensions

As data acquisition set is used a system developed by workers of department 12112, data are gathered through Matlab environment, using Instrument control toolbox. Each of the sensor (psychrometers, pressure transducer, chip set reading temperatures from digital thermometers) has unique fixed address for identification and data are transferred by ethernet cable.

2.1. Measurement procedure and evaluation

Before each measurement, water containers for moistening the socks of wet-bulb thermometers was checked whether still contains water to secure right function of psychrometers. After that, the water tank was heated to required temperature. Then, demanded flow rate in the test rig was set by reducing voltage of axial fans propelling the test rig. If the conditions in the test rig are stabilised: mass flow rate, temperature of the water film, outlet dry-bulb and wet-bulb temperature must be steady, the measurement starts. Usually the measurement lasts from 5 - 10 minutes.

At first, from data set were filtered badly measured data (see fig. 2), then time-averaged of measured value is made. From processed data, the specific humidity is evaluated as [5]:

$$x = \frac{2501,6 - 2,3263(T_{wb} - 273,15)}{2501,6 + 1,8577(T_{db} - 273,15) - 4,184(T_{wb} - 273,15)} \cdot \frac{0,62509 p''_{wv-wb}}{p - 1,005 p''_{wv-wb}} + \frac{1,00416(T_{db} - T_{wb})}{2501,6 + 1,8577(T_{db} - 273,15) - 4,184(T_{wb} - 273,15)} \quad (3)$$

Mass flow rate of humid air is evaluated according to [4]. From that, mass flow rate of dry air is calculated:

$$\dot{m}_{da} = \frac{\dot{m}_{ma}}{1 + x_{out}} \quad (4)$$

After evaluation of previously mentioned properties, the evaporation rate is calculated by above written equation 2.

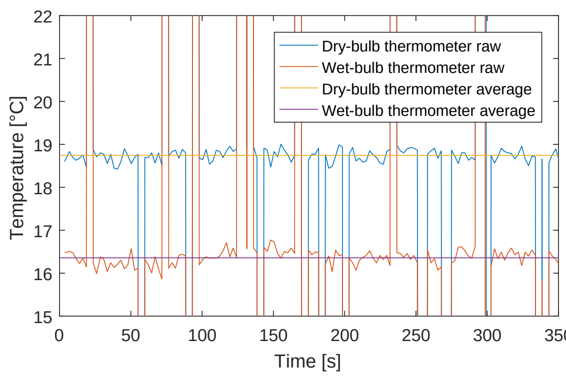

Fig. 2. Example of signal processing

2.2. Results of evaporation rate measurement and discussion

In this subsection, results from measurement of evaporation rate are shown and discussed. Measurements were carried out during a year to get variety of inlet conditions for future comparison with CFD results (ambient temperature was changing in range from 14°C to 28°C and relative humidity was varied from 30% to 55%). All evaluated points are shown in fig. 3, dependency of evaporation rate on specific humidity and ambient temperature was neglected, which is the cause of multiple results for one temperature.

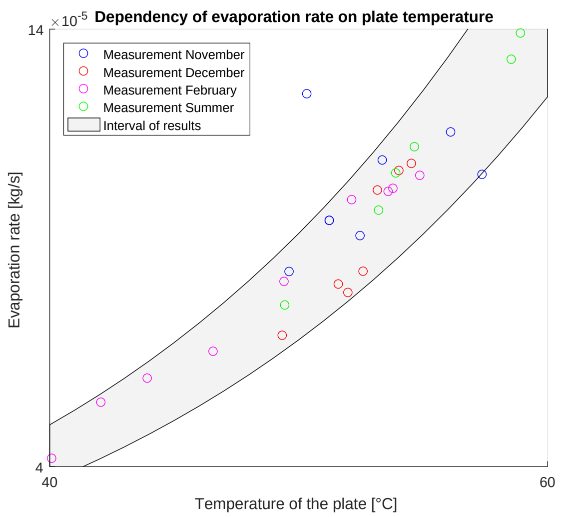

Fig. 3. *Experimental data measured during half-year period*

To be able of identifying the dependency od evaporation rate on water temperature, it is necessary to separate the effect of different inlet conditions. This was carried out by conducting set of measurements during single day. Such a measurement was realized twice for two mass flow rates (10g/s and 15g/s). Results are shown in fig. 4 and 5 and they are compared with mathematical model based on criterion equations, which was also covered in [1].

Even though, the water is evaporating to the air stream, the contribution by forced convective evaporation is negligible and experimental data are compared to free convection only. Ratio Gr_m/Re^2 is used as criteria to distinguish whether the effect of natural or forced convection in mass transfer is dominant (or mixed convection is present) [2, 3, 6]. In heat transfer, the ratio Gr/Re^2 is in some papers related to mixed convection called Richardson number [7] (Gr is heat transfer Grashof number).

$$Ri = \frac{Gr}{Re^2} \tag{5}$$

If the ratio Gr_m/Re^2 is smaller than 0.1, forced convection is the main mass transfer process. If it lays in interval of <0.1; 10>, none of the convective modes can be neglected. Sherwood number is then calculated as a non-linear combination of Sherwood number for free convection and Sherwood number for forced convection. If the ratio is above 10, the contribution of forced convection can be neglected.

The ratio Gr_m/Re^2 for presented data is ranging from 25 to 125, thus only natural convection is considered.

Criterion equation for laminar free convection used for comparison:

$$Sh = 0.54 \cdot (Gr_m \cdot Sc)^{1/4} \tag{6}$$

Criterion equation for turbulent free convection used for comparison:

$$Sh = 0.14 \cdot (Gr_m \cdot Sc)^{1/3} \tag{7}$$

Laminar free convection occurs, if the product of $Gr_m \cdot Sc < 2 \cdot 10^7$, size of transient region is not mentioned. Measured data has the product of $Gr_m \cdot Sc \approx 10^9$, which indicates turbulent character of natural convection.

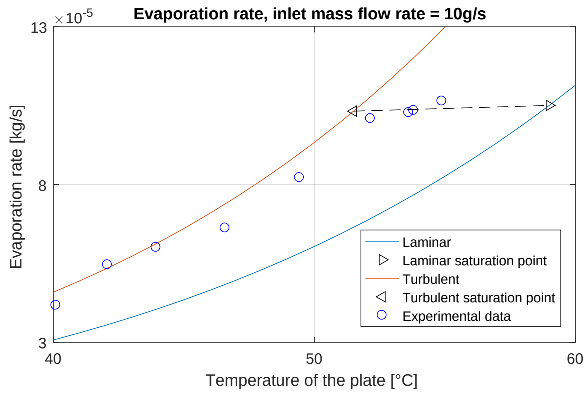

Fig. 4. *Results of evaporation rate measurement with comparison with criterion eq. for flow rate 10g/s*

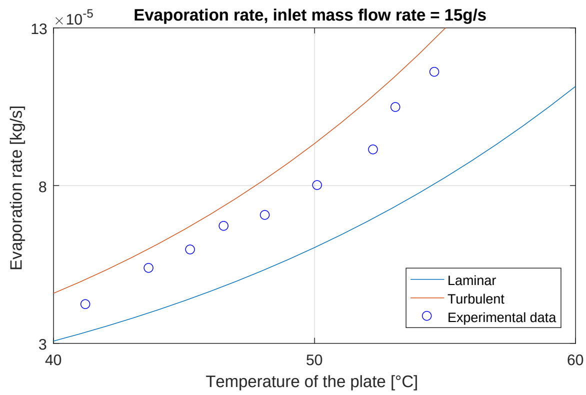

Fig. 5. *Results of evaporation rate measurement with comparison with criterion eq. for flow rate 15g/s*

Measured data (fig. 4 and 5) also confirm turbulent regime, even though in case of higher flow rate, there is deviation from predicted value by mathematical model. Comparison also proves, that considering only natural convective mass transfer is adequate. The most probable explanation of the deviation of experimental data from the mathematical model is small inaccuracies in evaluating properties of moist air, the most problematic is diffusion coefficient of water vapor in air. In the current version of the model, equation (8) is used, found in [8]:

$$D_{v,a} = \frac{2.07^{-5} \cdot 101325}{p} \left(\frac{T_{ma}}{293.15}\right)^{2.072} \tag{8}$$

Opposite to the previous version of mathematical model mentioned in [1] using diffusion coefficient from [9]:

$$D_{v,a} = \frac{0.926}{1000 \cdot p} \left(\frac{T_{ma}^{2.5}}{T_{ma} + 245}\right) \tag{9}$$

Last equation, which was tested, was found in [5]:

$$D_{v,a} = \frac{0.04357 \cdot T_{ma}^{1.5}}{p \cdot \left(29.9^{1/3} + 18.8^{1/3}\right)^2} \cdot \left(\frac{1}{28.97} + \frac{1}{18.016}\right)^{0.5} \tag{10}$$

Comparison of the solution is in fig. 6. Worst results are obtained with usage of equation (9). The smallest difference between measured data and calculation is with use of equation (10) combined with equation (7) (criterion equation for Sherwood number for free convection in turbulent regime) and with use of equation (9) combined with equation (6) (criterion equation for Sherwood number for free convection in laminar regime). Since $Gr_m \cdot Sc > 2 \cdot 10^7$, which means turbulent natural convection, equation (8) or (10) should be used.

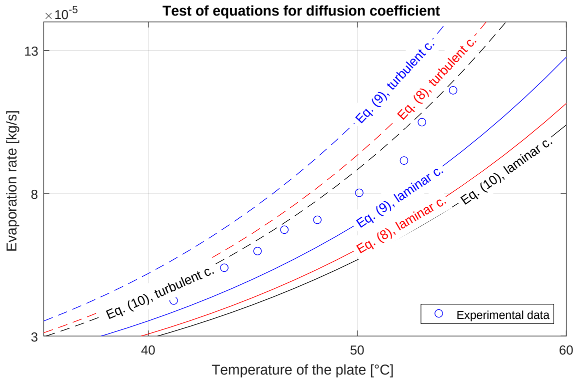

Fig. 6. *Comparison of calculation of evaporation rate using different equations for diffusion coefficient,* $\dot{m} = 15g/s$

Further research in diffusion coefficient would be needed for precise evaluation of evaporation rate, because the solution is very dependant on it and in literature exists several relationships and correlations. But for that, more data would be needed to be sure that the problem is not in other area. But solution using equation (8) is quite precise in case of flow rate of 10g/s. In the future, it would be interesting the measure also higher flow rates and see, if higher velocity, still in regime of natural convection, would have any effect on evaporation rate. But for verification whether the data was measured correctly, such comparison was sufficient.

2.3. Temperature field measurement

Second part of validation procedure was investigation of convective flow field. Air flow starts to interact with the buoyancy driven secondary flow (term Poiseuille-Benard flow is used in [10]), which leads to a complex 3D flow behaviour. Due to low velocities in the test rig, around $0.1\frac{m}{s}$, theirs measurement is quite problematic, only sophisticated method can be used: LDA or PIV, but several other problems exist such as fogging of the plexiglas in the bottom half of test section due to evaporation and large area which has to be traversed.

Instead of velocity field, temperature field was chosen for the comparison. 10 thermocouples, type K, were used for the measurement. Data were acquired via National Instruments's DAQ system combined with Matlab. Probes are mounted to a stand, located one above each other as a vertical line probe in a distance of 31mm. Lowest probe was at height of 10mm, highest was at height of 290mm. Measurements were conducted in streamwise vertical plane, examined area was plane starting 300mm in front of the fluid film and ending 200mm behind the fluid film (1500mm x 280 mm). To cover whole area, traversing with a stand is needed. In each position, it was waited for reaching thermal stability before actual measurement of temperature. Then, in Matlab, data were averaged, merged and interpolated on the measurement grid.

Two regime were examined, with heated water film (variant denoted as "moist") and variant with only a heated plate without the film (denoted as "dry"). The motivation was using single-component fluids in CFD, thus it was necessary to determine, how big temperature difference the water vapour cause. Only mass flow rate of 10g/s was measured.

2.4. Results and discussion

Results from the measurement are in fig. 7 and 8, for the two measurements was $\Delta T = T_{boundary} - T_{ambient} \approx 35°C$, $T_{ambient}$ was ranging from 16 to 17°C. The temperature field of both variants ("moist" and "dry") has very similar characteristic in both cases and is strongly influenced by the natural convection. The peak of the stream's core is shifted toward the heated plate, which caused very steep temperature gradient above the heated plate / hot fluid film. The heated region below the top desk steps in to area <-300mm; 0mm> (in upwind direction). That is caused by reversed flow along the top surface. The flow reversal makes the fluid in the upper region hotter than that below due to the convective heat transfer from the downstream fluid. This corresponds to conclusions of researchers studied this topic (mixed convection flow with $Gr/Re^2 > 10$), mostly interested only in velocity field. In majority of experimental studies, as fluid, water is used. Numerically was carried out studies both for Pr=0.7 and Pr=5, approximately equally. Two explanation of the shifted core and backflow are mentioned:

1. In [11] is concluded that fluid in lower part is hotter due to the heated plate, which leads to decrease of viscosity of the fluid. As consequence reduction of flow resistance arrived in lower region and hence, higher velocity. To satisfy conservation of mass, velocity has to be decreased in the upper part.
2. In [12] is the asymmetry of the velocity profile attributed to rising plumes and falling fluid parcels creating vertical motions which creates positive streamwise pressure gradient. The crossflow motion accelerates the axial flow in the lower half and retards it in the upper region.

Obtained temperature fields are opposite to the first statement, the fluid is hotter in the upper part. The asymmetry and reversed flow is more likely to be a product of complicated 3D flow behaviour because of buoyancy forces.

Main differences between temperature field with and without evaporation is in upper region. Differences are also at heigh of 10mm, but there is problem with steep temperature gradient, where even a small displacement of the probe can cause an error in order of degrees. The fluid is enhanced by the hot water vapour, which leads to the increase of the temperature.

The signal from the probes was almost stable in both cases ("moist" and "dry") see fig. 9 and 10 except the lowest thermocouple, 10mm. The temperature at that height is fluctuating, the changes are almost 5°C for the "moist" case, 2°C for the "dry". At height of 41mm above the surface, the signal was much less transient, fluctuations was around 0.5°C for both cases.

The "moist" temperature signal from 10mm height was subjected the to spectral analysis using Fast-Fourier transformation, but any dominant frequency wasn't discovered.

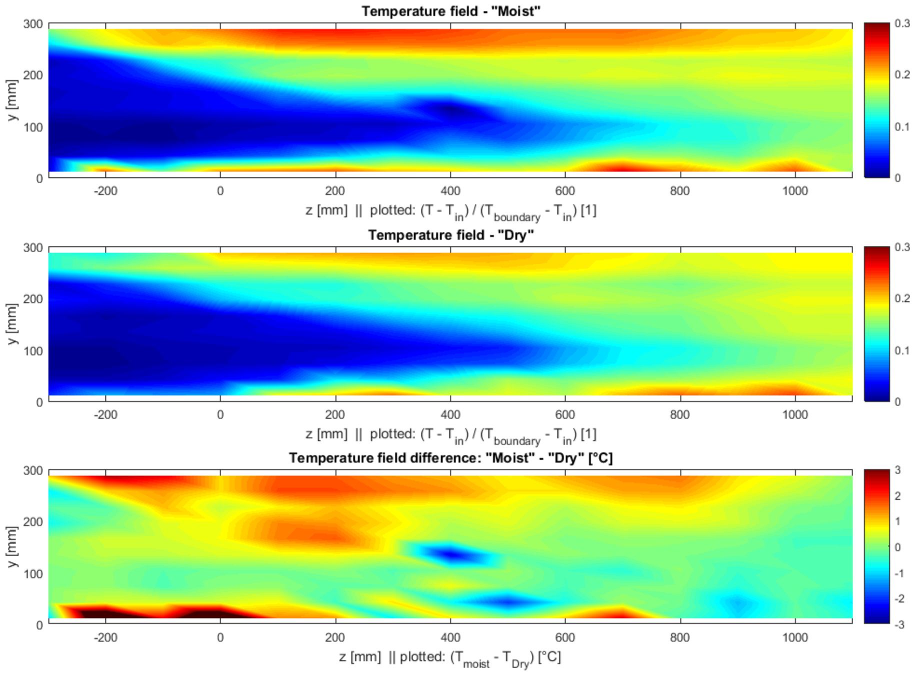

Fig. 7. *Vertical temperature field measurement 1: up - moist variant (water film as boundary), middle - dry variant (heated plate as boundary), down - temperature difference between the two variants*

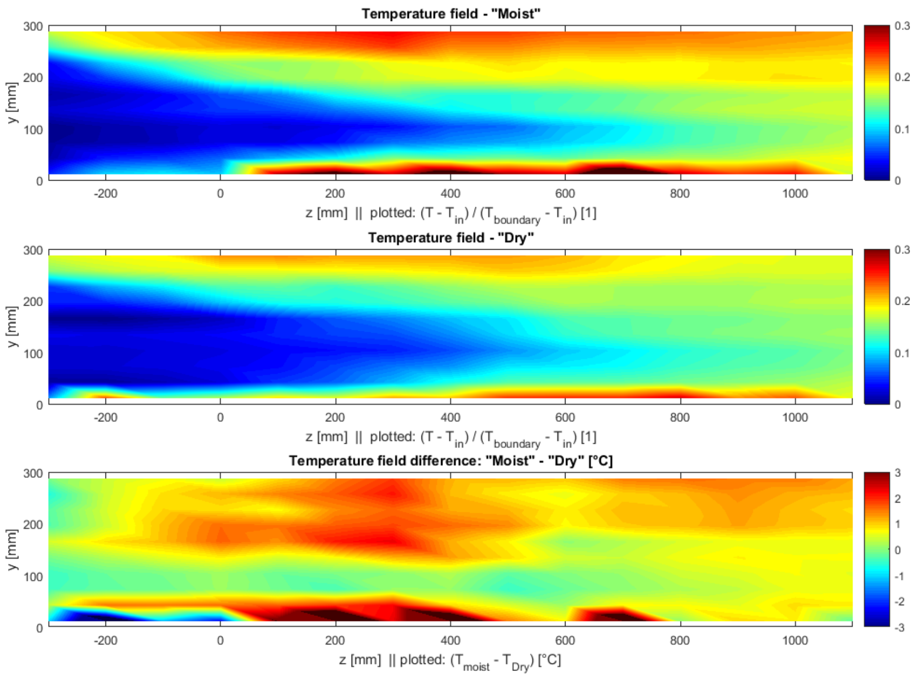

Fig. 8. *Vertical temperature field measurement 2: up - moist variant (water film as boundary), middle - dry variant (heated plate as boundary), down - temperature difference between the two variants*

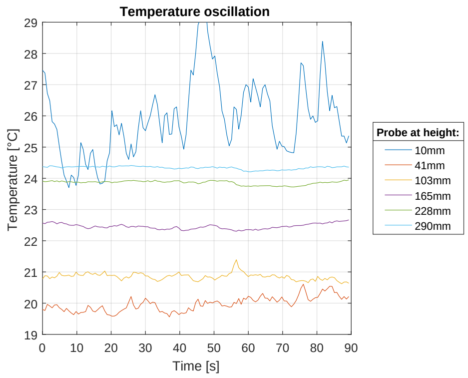

Fig. 9. *Temperature fluctuations at x=500mm for moist variant (water film as boundary)*

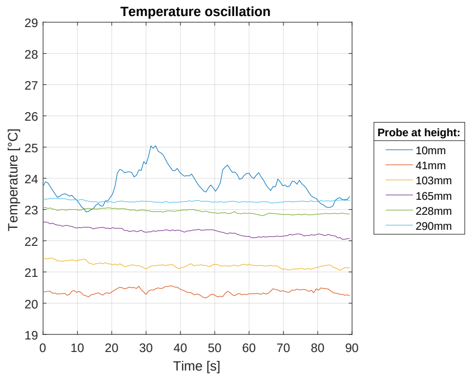

Fig. 10. *Temperature fluctuations at x=500mm for dry variant (heated plate as boundary)*

Results from the two measurements show that neglecting evaporation and solving the problem only as single-component fluid could cause an error of 2-3 °C in calculation of temperature field. Also show a complex flow behaviour with creation of reversed flow below the top desk, transient above the water film surface. In case of only heated plate, the transient behaviour was smaller.

3. Mixed convection with $0.1 > Gr_m/Re^2 < 10$

For second test case, water tank was removed and replaced by a stand with cone hole at the top. Water, heated to a defined temperature, is flowing around the inner wall of the stand. The stand is made from duralumin and with wall thickness of 1.5mm. The heating provides uniform temperature stratification on the surface of the stand (necessary for future comparison with CFD results). The water is heated by electric flow-heater (3kW input power) controlled by PID regulator using solid state relay. Input to the regulator is from thermocouple, type K inserted to pipeline. The circuit is propelled by circulation pump. The modified apparatus is shown in fig 11.

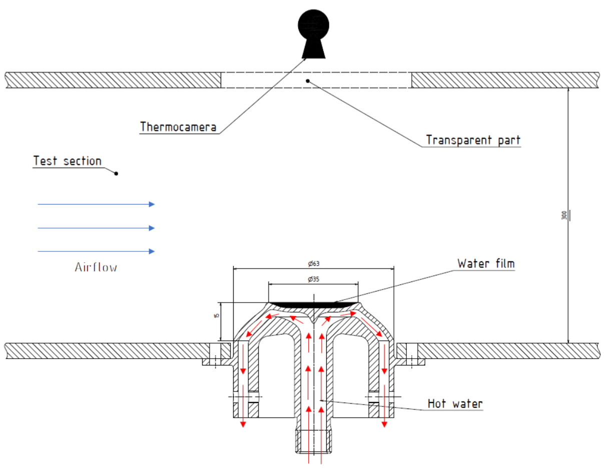

Fig. 11. *Rig modification*

Evaporation rate from the water film is very small (maximum diameter is 35mm) so above presented psychrometric method can't be used. It was decided to measure evaporation rate optically. It was necessary to overcome several difficulties: First is to ensure that spherical cup is not created instead of flat water surface. It is necessary to have clean solid surface and use distilled water. Second issue is how to capture the water film with camera. At first, coloured water combined with photo camera was tested, but results weren't good. Then, thermocamera FLIR was used. The water film was captured very good, but it was necessary to find a material transparent for the camera, because the rig has to be closed. Food foil was used. The optical method, before the measurement, was also compared with external balances with readability 0.1g. The deviation of values obtained by optical method and by balances was $\Delta m_{max} \approx 0.1g$, which is in order of the reading error.

The airflow above the stand with film has the same flow rates is in previous case ($\dot{m} = 10g/s$ and $\dot{m} = 15g/s$)

These low flow rates combined with small diameter of the water film of high temperature leads to mixed convection regime with ratio $Gr_m/Re^2 \in (0.1; 10)$ thus neither contribution of natural or forced convection can't be neglected. The Sherwood number is then calculated as [6]:

$$Sh_{mixed} = \left(Sh_{free}^n + Sh_{forced}^n\right)^{1/n} \tag{11}$$

According to [2, 3], $n \in (1; 2)$ for mass transfer and it is function of Gr_m/Re^2 and in each paper, experimental correlation is introduced. Presented case is problematic, because diameter of the film is changing during the measurement. Since the diameter is in Re and Gr_m on different order, the ratio Gr_m/Re^2 is also changing during the measurement. The evaporation rate is changing as well. The mathematical model was modified to cover the changes of water surface thanks to known dimensions of the cone hole. The schematic diagram is in fig. 12.

At first, the initial diameter is set according experimental data. From the diameter, height and mass of the film is calculated. Then dimensionless numbers (Re, Sc, Gr_m), from criterion equations Sherwood number for free, laminar and forced, laminar convection form which Sh number for mixed convection is interpolated. From Sh, mass transfer coefficient β is evaluated and evaporation rate is establish. Per given time-step (in range of seconds), evaporated mass is calculated, from which depletion of height is determined. With new height, new radius is set and the loop is running till $m_{res} \longrightarrow 0$.

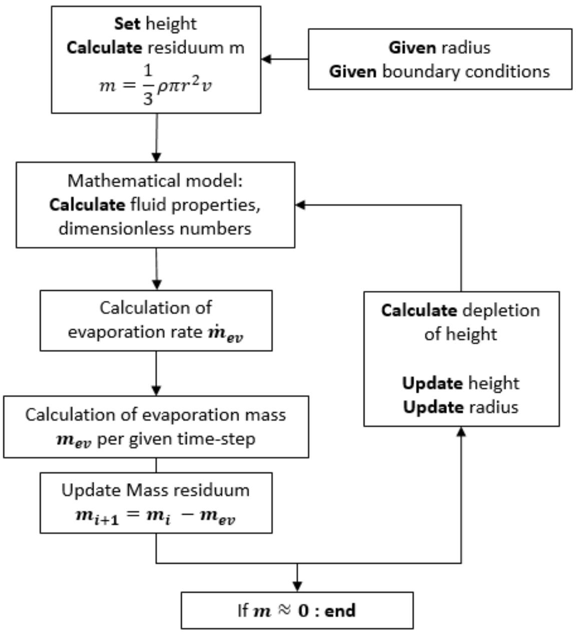

Fig. 12. *Calculation schematic of based criterion equation mathematical model*

3.1. Measurement and processing

At the start, required temperature of the stand on the PID regulator and pressure drop on the orifice plate is set. For measuring the inlet and rig conditions, same DAQ set is used is for previous case. The film is capturing by thermocamera. Periodic capturing with 1 or 2 minute interval is set. On temperature scale, maximum temperature is set lower than is the temperature of the water film. That makes the film more brighter in the image which is used later for the processing. After the conditions in the test rig are stabilized, film from distilled water is created and the measurement starts and continues till the film dries out.

The images obtained by camera are processed by Matlab through Image Processing Toolbox. The images are loaded to Matlab environment and function for finding circles and determining its radius is used:

By user defined ratio mm / px, diameter in mm are calculated and mass residuum on the stand are established.

3.2. Results and discussion

Obtained time dependency of evaporation rate for four cases compared with results from mathematical model are shown on fig. 14 - 17. The mass of examined film was $(1.15 \pm 0.05)g$. Measured data are compared with solution from criterion equation using equation (11), four variants of exponent n was considered:

1. $n = 1$
2. $n = 2$
3. n is function of Gr_m/Re^2, Sh_{mixed} is calculated according to [2]: $Sh = Sh_{Free} \cdot \left[1 + f\left(Gr_m/Re^2\right)\right]$ where $f\left(Gr_m/Re^2\right) = 0.543 - 0.408 \cdot ln\left(Gr_m/Re^2\right) + 0.0826 \cdot \left[ln\left(Gr_m/Re^2\right)\right]^2$
4. n is function of Gr_m/Re^2, Sh_{mixed} is calculated according to [3]: $n = -0.042 \cdot \left(Gr_m/Re^2\right)^2 + 0.583 \cdot \left(Gr_m/Re^2\right) + 1.182$

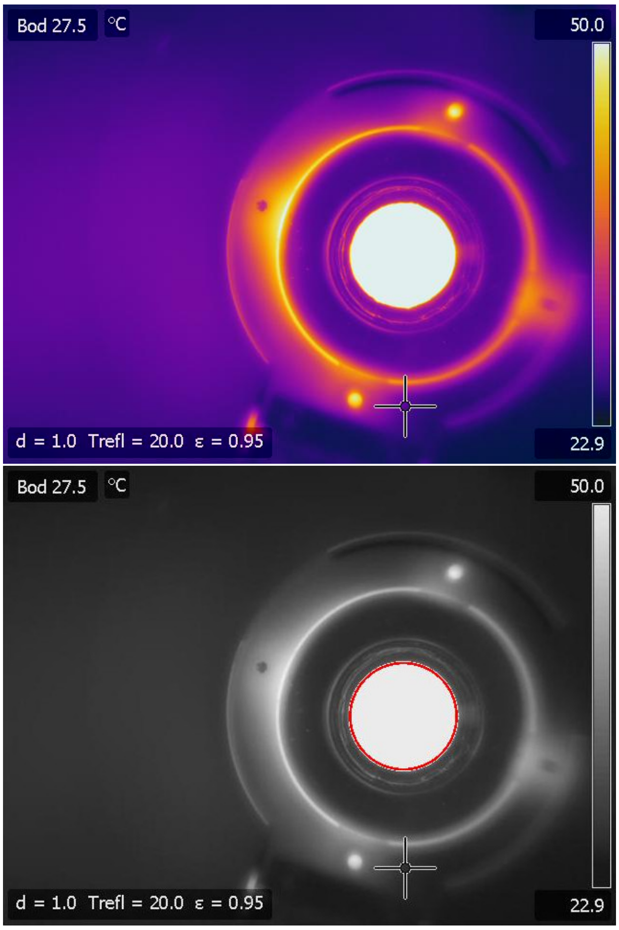

Fig. 13. *Up - film capturing using thermocamera, down - processing via Matlab*

Natural convection is considered as laminar and calculated according equation (6), forced convection is considered also laminar:

$$Sh = 0.664 \cdot Re^{1/2} \cdot Sc^{1/3} \qquad (12)$$

The trend of experimental data and solution from mathematical model is the same. Best match is with $n = 1$, except case $t_{stand} = 60,°C$, $\dot{m} = 15g/s$ (fig. 17), where shorter evaporation time is predicted.

Correlations found in [2, 3] and fixed $n = 2$ predicts similar values, but predicted evaporation time differs from experiment about 30%.

Volume velocity established from the mass flow rate on orifice plate was used for calculations, but the stand will affect the behaviour in the boundary layer and the velocity. A PIV measurement or CFD simulation would be needed to see the magnitudes of the velocity above the fluid film.

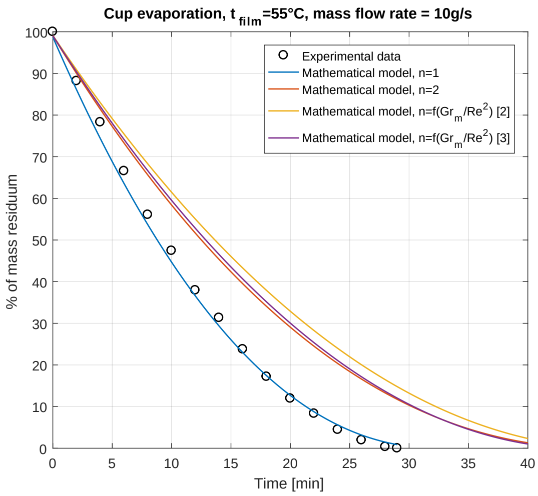

Fig. 14. *Depletion of mass by time,* $T_{film} = 55\,°C$, $\dot{m} = 10g/s$

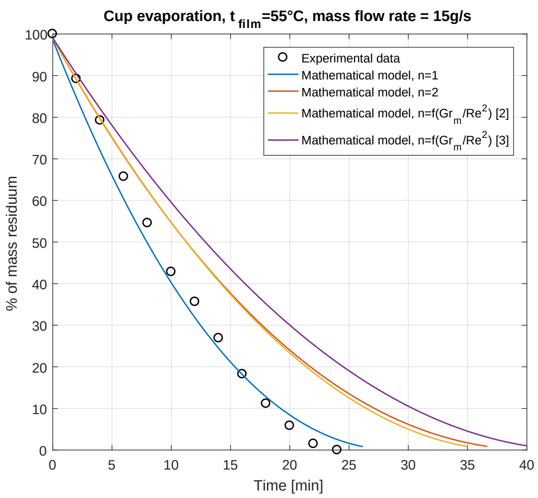

Fig. 15. *Depletion of mass by time,* $T_{film} = 55\,°C$, $\dot{m} = 15g/s$

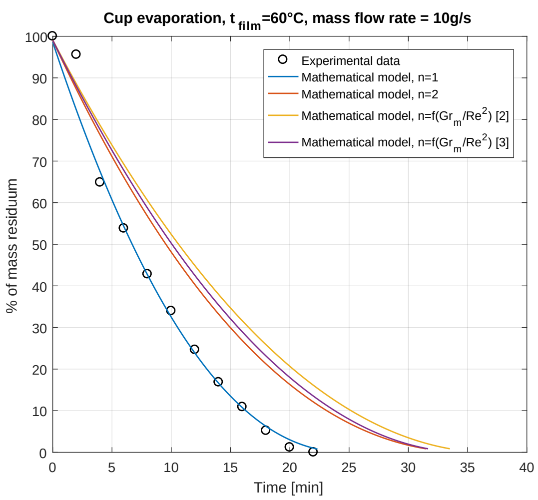

Fig. 16. *Depletion of mass by time,* $T_{film} = 60\,°C$, $\dot{m} = 10g/s$

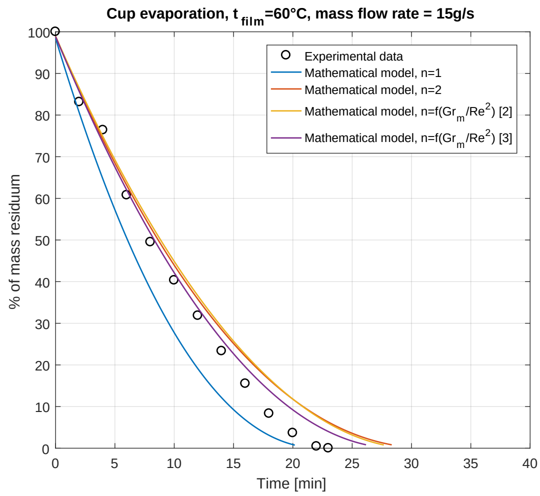

Fig. 17. *Depletion of mass by time,* $T_{film} = 60\,°C$, $\dot{m} = 15g/s$

Results implies that for this type of application, plain summation of Sh_{Free} and Sh_{forced} should be used. The prediction is sensitive to used diffusion coefficient in mathematical model as in previous case.

4. Conclusion and future steps

The main task of the measurement was obtained data which will be used for validation of CFD model. All collected data were discussed and compare to findings of researchers who were examined this topic.

For both regimes of mixed convection ($Gr_m/Re^2 > 10$ and $0.1 > Gr_m/Re^2 < 10$), convective mass transfer was measured and compared to solutions using criterion equation. For the first case can be found in literature that contribution of forced convection is very small and can be neglected. This finding corresponds with measured data, which are matching with calculations using criterion equation for Sherwood number for turbulent natural convection. 3 equations for diffusion coefficient were tested. Results show a dispersion in prediction of evaporation rate. An investigation of conditions for which the relationship for diffusion coefficient was determined would be needed for selecting the most suitable for this application.

Two variants of temperature field was examined, with water film as the bottom surface and only heated plate. For both cases, the vertical temperature field is strongly affected by the secondary flow induced by buoyancy forces. The airflow becomes hotter in the upper region. The leads to creation of stationary reversed flow below the top desk. High fluctuations of the temperature was observed only 10mm above the fluid film. The temperature difference between both variants is maximally 2-3 degrees.

For second mixed convection case, $0.1 > Gr_m/Re^2 < 10$ depletion of mass of water film by time is examined and compared to criterion equation based mathematical model, but modified to solve the evolution of mass in time. The trend of model with experiment is the same and from comparison, very good match can be seen for coefficient $n = 1$. For correlations found in literature, the mismatch in evaporation time is up to 30% to experimental data.

In the future, PIV measurement will be conducted to measure velocity field above the stand. Also measurement of temperature field is planned.

Nomenclature

D diffusion coefficient (m^2/s)
$\dot{m}$ mass flow rate (kg/s)
n coefficient (1)
p pressure (Pa)
p'' saturated vapour pressure (Pa)
T temperature (°C)
t time (s)
x specific humidity (kg/kg)

β mass transfer coefficient (m/s)

Gr Grashof number (1)
Gr_m Grashof mass transfer number (1)
Pr Prandtl number (1)
Re Reynolds number (1)
Ri Richardson number (1)
Sc Schmidt number (1)
Sh Sherwood number (1)

da dry air (1)
db dry-bulb thermometer (1)
in inlet (1)
ma moist air (1)
out outlet (1)
wb wet-bulb thermometer (1)
wv water vapour (1)

References

[1] Jakub Devera. "Experimental verification of numerical thin film evaporation model". In: *Studentská tvůrčí činnost: sborník* (2016).

[2] Michael T. Pauken. "An experimental investigation of combined turbulent free and forced evaporation". In: *Experimental Thermal and Fluid Science* (1998).

[3] Amid Jodat. "An experimental study of ability of similarity theory to predict water evaporation rate for different convection regimes". In: *International journal of multidisciplinary sciences and engineering* (2012).

[4] *ČSN EN ISO 5167-2. Měření průtoku tekutin pomocí snímačů diferenčního tlaku vložených do zcela zaplněného potrubí kruhového průřezu - Část 2: Clony.* Praha: Úřad pro technickou normalizaci, metrologii a státní zkušebnictví, 2003.

[5] Detlev G. Kröger. *Air-cooled heat exchangers and cooling towers, Volume II.* Tulsa: PennWell, 2004. ISBN: 1-59370-019-9.

[6] Frank P. Incropera and T. L. Bergman. *Fundamentals of heat and mass transfer. 7th ed.* Hoboken, NJ: Wiley, 2011. ISBN: 978-0470-50197-9.

[7] S. M. Elsherbiny, M. A. Teamah, and A. R. Moussa. *Experimental mixed convection heat transfer from an isothermal horizontal square cylinder.* Experimental Thermal and Fluid Science, 2016.

[8] Pavel Šafařík and Magda Vestfálová. *Termodynamika vlhkého vzduchu.* ČVUT, 2016. ISBN: 978-80-01-06020-9.

[9] T.K. Sherwood and R.L. Pigford. *Absorption and extraction.* McGraw-Hill, 1952.

[10] Christophe BONNEFOI et al. *Poiseuille–Benard instability in a horizontal rectangular duct water flow.* International Journal of Thermal Sciences, 2004.

[11] Ahmed ELATAR and Kamran SIDDIQUI. *The effect of mixed convection on the structure of channel flow at low Reynolds numbers.* International Journal of Heat and Fluid Flow, 2014.

[12] D.K. Choi. and D.H. Choi. *Developing mixed convection flow in a horizontal tube under circumferentially non-uniform heating.* International Journal of Heat and Mass Transfer, 1994.