Numerical simulation of thin water film evaporation

Ilona Machovská[1,*], Tomáš Hyhlík[1]

[1] CTU in Prague, Faculty of mechanical engineering, Department of fluid dynamics and thermodynamics, Technická 4, 166 07 Prague 6, Czech republic

Abstract

This work deals with the theory of thin water film evaporation and numerical simulation of this phenomena. The theory focuses on the mass transfer of water vapor from the film to the bulk flow of air. As possible mass transfer mechanism is considered convection – natural and forced. To solve specific problem there is used the equation to evaluate mass flux of water vapor from the film to the bulk flow of air. In this equation there appears mass transfer coefficient. Essential part of this paper is devoted to evaluation of mass transfer coefficient. At first the problem is solved using criterion equations to determine mass transfer coefficient. In the next step the same problem is solved using analogy between heat and mass transfer. There is numerically determined heat transfer coefficient solving natural convection in the closed volume or solving forced convection in the model of experimental rig. Target of this work is to evaluate the time of evaporation of the thin water film of known thickness from the surface of the specific temperature by the specific moist air conditions.

Keywords: numerical simulation; thin water film; mass transfer coefficient; evaporation

1. Introduction

Problem with thin water film creation and evaporation occurs in many technical applications. In some applications creation of thin water film is desirable from the point of view of heat and mass transfer between the film and bulk flow of air. As an example could be mentioned heat exchangers such as cooling towers. On the other hand there are applications in which the creation of the water film is undesirable and there is an effort to determine the time of evaporation of the water from the film to the air. This work examines the second group of applications.

Need of solution of thin water film evaporation proves number of papers devoted to this problem such as [1,2]. In most of this papers there is theoretical analysis of the evaporation process, numerical solution using different approaches and in some of them there is also experimental validation of the numerical solution [3,4]. This paper focuses mainly on the thin water film evaporation theory and the numerical solution on the specific geometry appropriate the solved application. Part of this work is also devoted to the experimental validation of the numerical models which will be introduced on following pages.

In [1] there is said that the evaporation process is led by the heat and mass transfer mechanism at the water-air interface but it is significantly influenced by surrounding conditions. Examining the real solved application and its working conditions, convective heat and mass transfer is considered. The theory focuses especially on the convective mass transfer respectively on the evaluation of mass flux of water vapor to the bulk flow of air. Convective mass transfer theory is well described in [5]. From the point of view of the solved application the natural convection is considered. Forced convection is considered in experimental setup.

2. Theoretical model

Theory deals with convective mass transfer respectively with evaluation of mass flux of water vapor from the film to the bulk flow of air. The scheme of the problem is shown in Fig. 1. There is a flat surface representing a plate of specified temperature on which is thin steady (meaning not falling) film of water. Plate with the water film is surrounded by moist air of specified temperature and relative humidity.

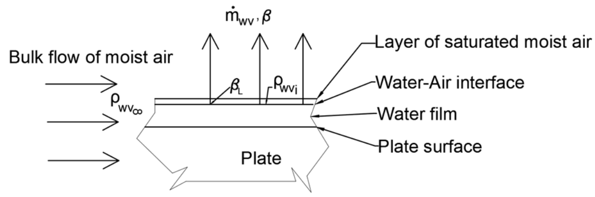

Fig. 1. *Thin water film evaporation scheme.*

The theory assumes that on the water-air interface there is very thin layer of saturated moist air. Mass transfer of water vapor takes place only between this layer and bulk flow of moist air. Another assumption is that the water vapor is dilute in the moist air. It means that the water vapor density is much smaller that the density of the moist air at the same temperature. Considering mentioned assumptions, mass flux of water vapor from the film to the air can be evaluated using following equation

$$\dot{m}''_{wv} = \beta(\rho''_{wvi}(T_p) - \rho_{wv\infty}(T_{ma})) \qquad (1)$$

where ρ''_{wvi} is saturation density of water vapor at the water-air interface and $\rho_{wv\infty}$ is density of water vapor in the bulk flow of moist air. Eq. (1) is valid under condition of

* Kontakt na autora: Ilona.Machovska@fs.cvut.cz

constant values of ρ''_{wvi} and $\rho_{wv\infty}$ over entire water-air interface. Symbol β denotes the average mass transfer coefficient defined as

$$\beta = \frac{1}{A}\int_A \beta_L dA \quad (2)$$

where β_L is local mass transfer coefficient at defined location on the water-air interface.

To determine values of water vapor densities ρ''_{wvi}, $\rho_{wv\infty}$, it is necessary to know the barometric pressure, relative humidity of the moist air flowing around the plate and temperatures of the moist air at the water-air interface and in the bulk flow of air [6]. Specific humidity of the moist air flowing around the plate is given by

$$x = 0.622\frac{p''_{wv}(T_{ma})\cdot\varphi}{p-p''_{wv}(T_{ma})\cdot\varphi} \quad (3)$$

where p''_{wv} is partial pressure of saturated water vapor defined as [7]

$$p''_{wv} = 10^z \quad (4)$$

where exponent z is a function of the temperature and it is defined as

$$z = 10.79586\left(1-\frac{273.16}{T}\right) + 5.02808\cdot\log\left(\frac{273.16}{T}\right) + 1.50474\cdot10^{-4}\left(1-10^{-8.29692\left(\frac{T}{273.16}-1\right)}\right) + 4.2873\cdot 10^{-4}\left(10^{4.76955\left(1-\frac{273.16}{T}\right)}-1\right) + 2.786118312 \quad (5)$$

Density of the moist air is given by

$$\rho_{ma} = \frac{p}{R\cdot T_{ma}}\cdot\frac{(1+x)\cdot M_a}{1+\frac{M_a}{M_{wv}}\cdot x} \quad (6)$$

where R is universal gas constant, M_a is molecular weight of dry air and M_{wv} is molecular weight of water vapor. Water vapor density in the bulk flow of moist air can be expressed as

$$\rho_{wv\infty} = \frac{x}{1+x}\rho_{ma}. \quad (7)$$

As was said the theory assumes that on the water-air interface there is very thin layer of saturated moist air. Another assumption is that the water film temperature as well as the temperature of the moist air at the interface is the same as the temperature of the plate T_p. Specific humidity of saturated moist air at the water-air interface is given by

$$x'' = 0.622\frac{p''_{wv}(T_p)}{p-p''_{wv}(T_p)} \quad (8)$$

Density of saturated moist air is given by

$$\rho''_{ma} = \frac{p}{R\cdot T_p}\cdot\frac{(1+x'')\cdot M_a}{1+\frac{M_a}{M_{wv}}\cdot x''} \quad (9)$$

Saturation density of water vapor at the water-air interface can be expressed as

$$\rho''_{wvi} = \frac{x''}{1+x''}\rho''_{ma}. \quad (10)$$

To determine mass flux of water vapor from the film to the air using Eq. (1), it remains to evaluate mass transfer coefficient β. Several methods how to determine mass transfer coefficient are introduced.

3. Mass transfer coefficient evaluation methods

Problem of evaluation of mass flux of water vapor from the film to the bulk flow of moist air is essentially the problem of determining mass transfer coefficient. As was said there are chosen several methods how to determine it. First method use criterion equations, second method in first step evaluate numerically heat transfer coefficient solving natural convection flow in appropriate fluid domain. Mass transfer coefficient is then determined using analogy between heat and mass transfer. Third method is experimental. Target of all three methods is to determine the time of evaporation of the water film of known thickness.

3.1. Mass transfer coefficient evaluation using criterion equations

Method of evaluation of mass transfer coefficient using criterion equations is chosen to quickly evaluate mass flux of water vapor from the film to the air and then to determine the time of evaporation. This method is also good to find the parameters that most influence the total evaporation time. Following calculations assumes constant value of mass transfer coefficient over the entire surface.

Mass transfer coefficient is evaluated using following equation [5]

$$\beta = \frac{Sh\cdot D_{wv}}{d}. \quad (11)$$

Mass transfer coefficient requires calculation of Sherwood number Sh and calculation of diffusion coefficient of water vapor to the air D_{wv}. Diameter d is the diameter of the plate. Sherwood number is obtained using following equation considering the plate, laminar flow and forced convection [5]

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

where Reynolds and Schmidt numbers are given by

$$Re = \frac{v\cdot d}{\nu} \quad (13)$$

$$Sc = \frac{\nu}{D_{wv}}. \quad (14)$$

Diffusion coefficient of water vapor to the air D_{wv} can be expressed as [5]

$$D_{wv} = \frac{0.926}{p\cdot 1000}\left(\frac{T^{2.5}}{T+245}\right). \quad (15)$$

Mass flux of water vapor from the film to the air is then obtained using Eq. (1) with mass transfer coefficient obtained from Eq. (11) and water vapor densities are given by Eq. (7) and Eq. (10). Total time of evaporation of the thin water film of known thickness can be expressed as

$$t = \frac{h_{wf} \cdot \rho_w}{\dot{m}''_{wv}} \tag{16}$$

where h_{wf} is the defined thickness of the water film and ρ_w is the density of water.

3.2. Mass transfer coefficient evaluation using analogy between heat and mass transfer

This method of mass transfer coefficient evaluation works with the analogy between heat and mass transfer. Most expressions for convective mass transfer coefficient β are determined from expressions for the convective heat transfer coefficient α [5]. Heat transfer coefficient is numerically evaluated solving the natural convection inside the closed volume. Closed volume is represented by the cube of dimensions to be 0,7x0,7x0,85m. Inside the cube there are three vertical flat discs of specified temperature. All of the cube walls are adiabatic except the right one which has specified uniform temperature. Fluid domain was designed to be appropriate the real solved application. Geometry of the fluid domain is shown in Fig. 2.

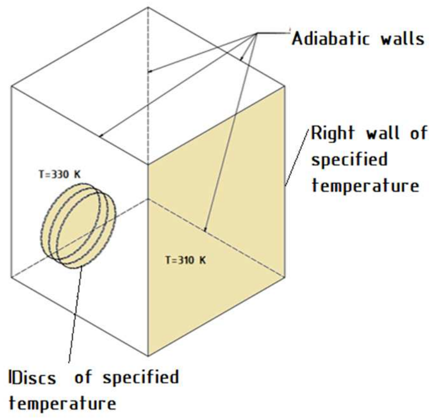

Fig. 2. *Geometry of the fluid domain.*

Flow inside the cube is caused only due to the different temperatures of the wall and plates. Moist air inside the cube is saturated, so in Eq. (7) to determine the water vapor density of bulk flow of air $\rho_{wv\infty}$, relative humidity φ is equal to 1.

Natural convection flow is numerically solved in commercial CFD software using Constant density equation of state, Boussinesq model of natural convection and viscous regime is set to be laminar according to the values of Grasshof and Prandtl number.

Heat transfer coefficient is determined from the value of surface heat flux $\dot{q}$ which is evaluated using Standard wall functions [8]. With known value of surface heat flux, heat transfer coefficient can be expressed as

$$\alpha = \frac{\dot{q}}{T_{wall} - T_{fluid}}. \tag{17}$$

In the denominator of right side of Eq. (17) there is the temperature difference between the wall and fluid which flows around the wall. Problem is where to specify the fluid temperature. In used commercial CFD software there are available values of Local heat transfer coefficient α_L and Local heat transfer reference temperature T_{ref}. These values are determined in the first cell next to the wall so they account with local variations in fluid temperature.

With known value of Local heat transfer coefficient, mass transfer coefficient is determined using the analogy between heat and mass transfer expressed as

$$Le_f = \frac{\alpha_L}{\beta \bar{\rho} c_p}. \tag{18}$$

where Le_f is Lewis factor, c_p is specific heat of moist air at constant pressure and $\bar{\rho}$ is average density of moist air given by

$$\bar{\rho} = \frac{\rho_{ma}(T_{ref}) + \rho_{ma}(T_p)}{2}. \tag{19}$$

Lewis factor is the dimensionless number which can be found in equations that describe heat and mass transfer. According to [9] values of Lewis factor are in the range of $\langle 0.5, 1.3 \rangle$. In following calculations Lewis factor is considered to be equal to 1 as the simplifying assumption. Then the mass transfer coefficient can be expressed as

$$\beta = \frac{\alpha_L}{\bar{\rho} c_p}. \tag{20}$$

With known value of mass transfer coefficient, mass flux of water vapor can be expressed as

$$\dot{m}''_{wv} = \beta \left(\rho''_{wvi}(T_p) - \rho''_{wv\infty}(T_{ref}) \right). \tag{21}$$

All the equations to evaluate water vapor densities $\rho''_{wvi}, \rho''_{wv\infty}$ and mass transfer coefficient β are implemented to the commercial CFD software. Time of evaporation of the film of known thickness can be determined using Eq. (16) or the evaporation rate of water vapor from the film to the air can be expressed as

$$\dot{h}_{wv} = \frac{\dot{m}''_{wv}}{\rho_w} \tag{22}$$

Evaporation rate of water vapor $\dot{h}_{wv}$ says how many meters of water film are evaporated per second.

3.3. Experimental evaluation of mass transfer coefficient

To validate previous calculations experimental test rig was designed. Test rig design is shown in Fig. 3. The test rig consist of the duct of rectangular shape. In the bottom of this duct there is placed the water tank. Rectangular duct passed into circular and at the end there is suction fan which provides moist air flow inside the duct.

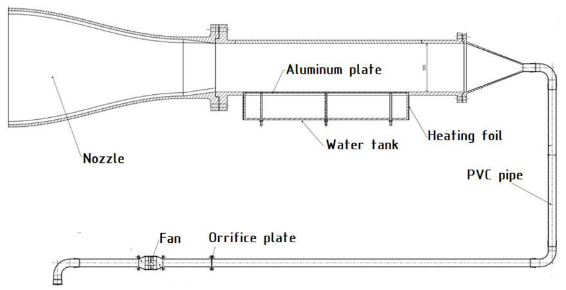

Fig. 3. *Test rig design.*

Water in the water tank is heated on constant temperature. Below the upper edge of the container there is the aluminum plate. Bottom side of this plate is in touch with water, on the top side of the plate there is thin water film of constant thickness. Keeping the constant water temperature in water tank provides constant temperature of the water film. There are installed temperature sensors on the bottom side of the aluminum plate to check the temperature distribution

From the water film evaporates water to the air. Measuring the humidity change of the air, required evaporation rate of water vapor is evaluated.

In Fig. 4 there is shown the control volume on which the conservation equations are solved.

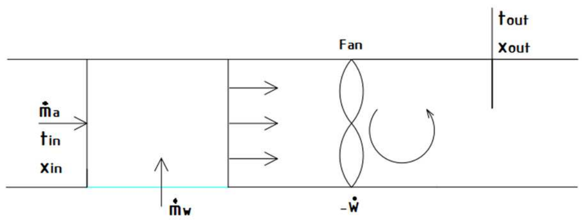

Fig. 4. *Control volume.*

There are two conservation equations for the control volume, continuity equation and energy conservation equation. Continuity conservation equation can be expressed as

$$\dot{m}_a(1 + x_{in}) + \dot{m}_{wv} = \dot{m}_a(1 + x_{out}) \qquad (23)$$

where $\dot{m}_a$is mass flow rate of dry air, $\dot{m}_{wv}$ is mass flow rate of water vapor from the film to the air and x_{in}, x_{out} is inlet, outlet specific humidity. Test rig dimensions were designed with respect to specific humidity change to aim measurable humidity change but avoid the saturation of the air. With known value of outlet specific humidity, mass flow rate of water vapor can be expressed as

$$\dot{m}_{wv} = \dot{m}_a(x_{out} - x_{in}) \qquad (24)$$

Energy conservation equation can be expressed as

$$\dot{Q} - \dot{W} + \dot{m}_a(h_{1+x})_{in} + \dot{m}_a(1 + x_{in})\frac{v_{in}^2}{2} - \dot{m}_a(h_{1+x})_{out} - \dot{m}_a(1 + x_{out})\frac{v_{out}^2}{2} + \dot{m}_{wv}h_{wv}(T_w) = 0 \qquad (25)$$

where $\dot{Q}$ is heat flux from water to the air, $(h_{1+x})_{in}$, $(h_{1+x})_{out}$ is moist air inlet, outlet enthalpy and $h_{wv}(T_w)$ is enthalpy of water vapor at water temperature. Outlet velocity v_{out} is measured using orifice plate and inlet velocity v_{in} is determined using continuity equation and know change in duct cross section area. Moist air enthalpies are given by

$$(h_{1+x})_{in} = (c_{pa} + x_{in}c_{pwv})T_{in} + x_{in}l_0 \qquad (26)$$

$$(h_{1+x})_{out} = (c_{pa} + x_{out}c_{pwv})T_{out} + x_{out}l_0 \qquad (27)$$

where l_0 is specific latent heat of water vaporization.

In the energy conservation equation also appears fan performance $\dot{W}$. It can be seen that the outlet temperature and outlet humidity are measured behind the fan because the fan provides mixing of the air so behind the fan there is uniform temperature and humidity field so the outlet values can be measured in one point of the duct cross section. Knowing the outlet temperature and using Eq. (25), heat flux $\dot{Q}$ can be determined. With known value of heat flux can be determined heat transfer coefficient using following equation

$$\alpha = \frac{\dot{Q}}{A_{WS}(T_W - T_{out})} \qquad (28)$$

where $A_{ws}$is the water surface area. Mass transfer coefficient is then evaluated using Eq. (20).

Test rig has been built and stands in the laboratories of Department of Fluid dynamics and thermodynamics. Currently the test rig is under testing. Picture of the test rig is shown in Fig. 5.

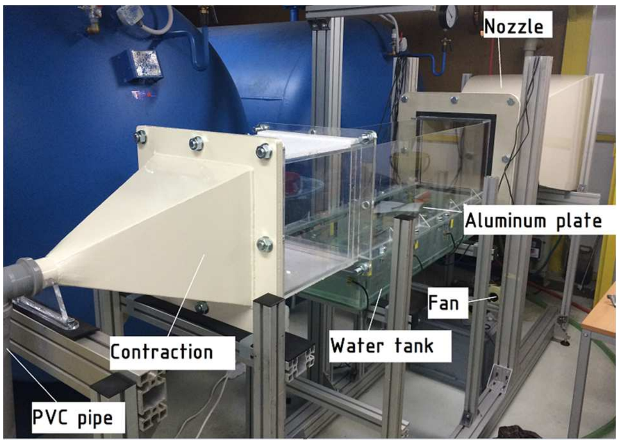

Fig. 5. *Picture of the test rig.*

4. Results

In this section are shown the results of evaluation of time of evaporation of water from the film to the air. At first there are shown the results using criterion equations then the result using analogy between heat and mass transfer and numerical simulation of natural convection. Results from the experiment are not yet available because the test rig is under testing.

4.1. Mass transfer coefficient evaluation using criterion equations results

As was said this calculation was done to find the parameters which influence the total evaporation time the most. In Fig. 6 there is shown how the time of evaporation depends on the specific humidity of the moist air flowing around the flat disc for different values of the moist air velocity. Values of moist are velocities were chosen to be appropriate the velocity of the flow inside the real solved application.

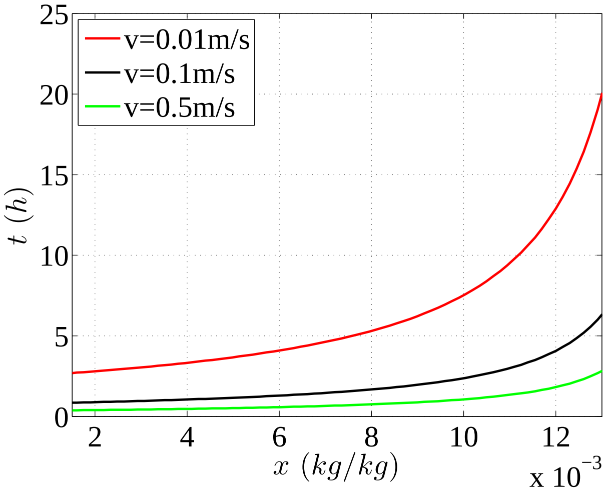

Fig. 6. *Time of evaporation dependency on specific humidity for different values of moist air velocity*

From the Fig. 6 can be seen that the time of evaporation rapidly increases with increasing specific humidity of the moist air. For this case temperature of the moist air remains the same so the relative humidity must also increase and moist air become saturated.

Other parameter which influence the total evaporation time is the temperature of the surface respectively the temperature of the flat disc. Theory assumes that the water film temperature and temperature of the layer of saturated moist air at the water-air interface have the same temperature as the surface. In Fig. 7 there is shown how the time of evaporation depends on the temperature of the flat disc.

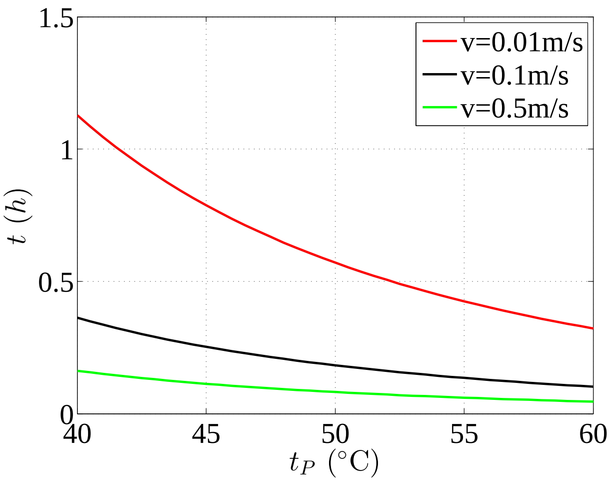

Fig. 7.*Time of evaporation dependency on the surface temperature for different values of moist air velocity.*

From the Fig. 7 can be seen that the time of evaporation decreases significantly with increasing the plate temperature. The plate temperature range was chosen to be equal to the surface temperature inside the real solved application. Total time of evaporation is adequate to the time of evaporation in the real application.

4.2. Mass transfer coefficient evaluation using analogy between heat and mass transfer results

In this section are shown the results solving numerically natural convection in the fluid domain shown in Fig. 2. Natural convection was solved using Boussinesq model in commercial CFD software In the post processing was evaluated mass transfer coefficient using analogy between heat and mass transfer, time of evaporation of the film of known thickness and evaporation rate of water vapor to the air.

Temperature of the flat discs was set to be 330 K (57°C), initial moist air temperature was set to be 293,15 K (20°C), moist air inside the volume is saturated so the relative humidity is equal to 1. At first were evaluated Local heat transfer coefficient and Local heat transfer reference temperature on the surface. These values are shown in Fig. 8 and Fig. 9.

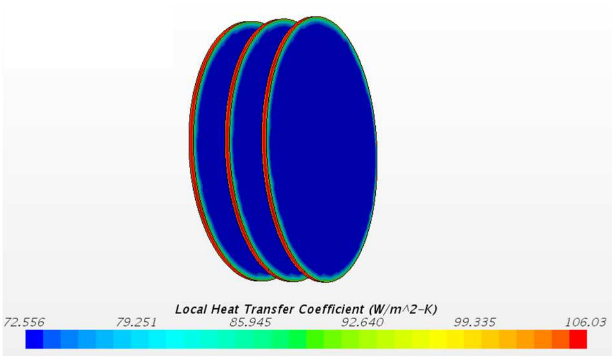

Fig. 8.*Values of Local heat transfer coefficient.*

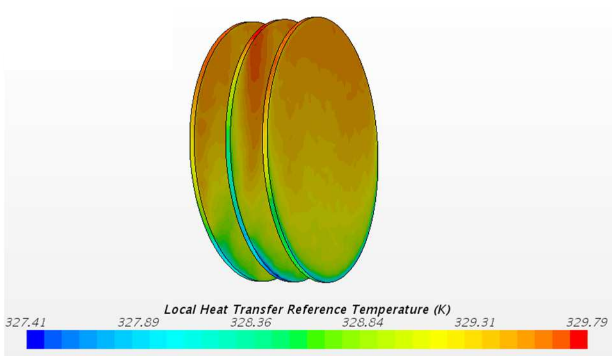

Fig. 9.*Values of Local heat transfer reference temperature.*

Values of mass transfer coefficient from water film to the air are shown in Fig. 10.

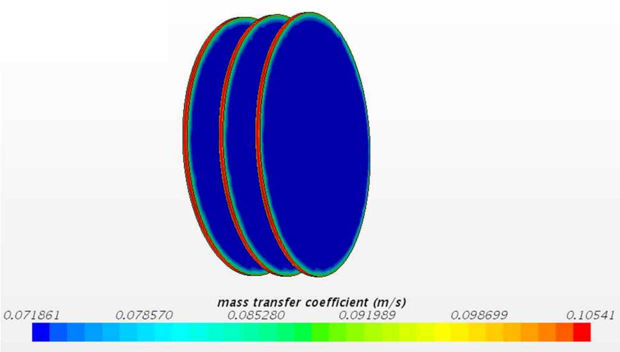

Fig. 10.*Values of mass transfer coefficient.*

Evaporation rate of water from water film to the air was evaluated using Eq. (22). Values of evaporation rate of water are evaluated in more relevant unit which say how many tenths of millimeters of water film are evaporated per hour.

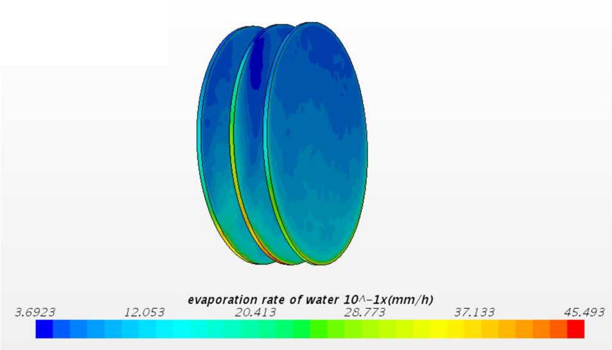

Fig. 11.*Evaporation rate of water from water film to the air.*

Using Eq. (16) was evaluated total time of evaporation of the water film of known thickness. Thickness of the water film was set to be 0,1 mm.

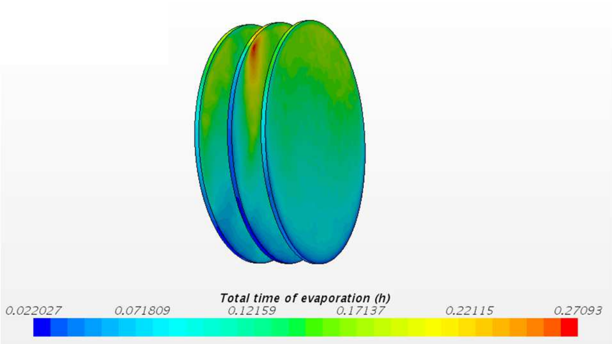

Fig. 12.*Total time of evaporation of water film of 0,1mm thickness.*

Values of mass transfer coefficient, local heat transfer reference temperature and total evaporation time were averaged per plate surface and compared with the values determined using criterion equations. These values were compared using the same conditions which are shown in Tab. 1.

Tab 1. *Conditions set for comparison.*

Plate temperature (K)	Initial moist air temperature (K)	Relative humidity (1)
330	293,15	1

Comparison of the values of mass transfer coefficient and total time of evaporation is shown in Tab. 2. In first row there are values determined using criterion equations, in second row there are values determined using analogy between heat and mass transfer. In the last column there are values of fluid temperature. These values are used in Eq. (1) as the moist air temperature respectively in Eq. (21) as the local heat transfer reference temperature.

Tab 2. *Comparison of mass transfer coefficient, total evaporation time and fluid temperature.*

mass transfer coefficient (m/s)	time of evaporation (h)	fluid temperature (K)
0,0023	0,117	293,15
0,074	0,121	329,27

Despite the fact that the values of mass transfer coefficient are different in order of magnitude, values of total time of evaporation are almost the same. Mass transfer coefficient evaluated using criterion equations is determined for fluid temperature to be equal to the moist air temperature. Mass transfer coefficient evaluated using analogy is determined using Local heat transfer coefficient and local heat transfer reference temperature (=fluid temperature). These values are evaluated in first cell next to the surface so value of local heat transfer reference temperature is effected by the flat discs temperature and it is different from moist air temperature. Difference in these two values of fluid temperatures blurs the difference in mass transfer coefficient and then the values of total time of evaporation for both solutions are almost the same.

5. Conclusions

Mass transfer coefficient evaluation using criterion equations and using analogy between heat and mass transfer was done in this work. Using the values of determined mass transfer coefficients, mass flux of water vapor from the water film to the air was evaluated. Specifying the value of water film thickness was determined the total time of evaporation of the water film. Values of the total time of evaporation are adequate to the real solved application.

The test rig to experimentally evaluate mass transfer coefficient was introduced. Governing equation to determine the mass flow rate of water and mass transfer coefficient were shown. Experimental results are not yet available because the test rig is still under testing.

Nomenclature

A area (m^2)
c_p specific heat at constant pressure ($J{\cdot}kg^{-1}\ K^{-1}$)
d diameter (m)
D diffusion coefficient ($m^2{\cdot}s^{-1}$)
h specific enthalpy ($J{\cdot}kg^{-1}$)
h_{wf} water film thickness (m)
$\dot{h}$ evaporation rate ($m{\cdot}s^{-1}$)
l_0 specific latent heat of vaporization ($J{\cdot}kg^{-1}$)
Le_f Lewis factor (1)
$\dot{m}$ mass flow rate ($kg{\cdot}s^{-1}$)
$\dot{m}''$ mass flux ($kg\ m^{-2}{\cdot}s^{-1}$)
p pressure (Pa)
$\dot{q}$ heat flux density ($J\ s^{-1}{\cdot}m^{-2}$)
$\dot{Q}$ heat flux ($J\ s^{-1}$)
Sc Schmidt number (1)
Sh Sherwood number (1)
R universal gas constant ($J\ kmol^{-1}{\cdot}K^{-1}$)
Re Reynolds number (1)
t time (s)
T temperature (K)
v velocity ($m\ s^{-1}$)
$\dot{W}$ performance ($J\ s^{-1}$)

α heat transfer coefficient ($J\ s^{-1}{\cdot}m^{-2}K^{-1}$)
β mass transfer coefficient ($m{\cdot}s^{-1}$)
ρ density ($kg{\cdot}m^{-3}$)
φ relative humidity (1)
ν kinematic viscosity ($m^2{\cdot}s^{-1}$)

a dry air
in inlet
ma moist air
out outlet
ref reference
w water
wv water vapor
WS water surface
" saturation state

References

[1] A.M. Raimundo, A.R. Gaspar, A.V.M. Oliveira, D.A. Ouintela, Wind tunnel measurements and numerical simulations of water evaporation in forced convection airflow, Int. J. Sci. 86 (2014) 28-40.

[2] J. Min, Y. Tang, Theoretical analysis of water film evaporation charactericties on an adiabatic solid wall, Int. J. Refrigration. 53 (2015) 55-61.

[3] S. Fischer, T. Gambaryan-Roisman, P. Stephan, On the development of a thin evaporating liquid film at a receding liquid/vapour interface, Int. J. Heat and mass transfer. 88 (2015) 346-356.

[4] A. Gonda, P. Lancereau, P. Bandelier, L. Luo, Y. Fan, S. Benezech, Water falling film evaporation on a corrugated plate, Int. J. Sci. 81 (2014) 29-37.

[5] ASHRAE, 2001. ASHRAE Handbook fundamentals. American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc, Atlanta (Chapter 6).

[6] ASHRAE, 2001. ASHRAE Handbook fundamentals. American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc, Atlanta (Chapter 1).

[7] D. G. Kröger, Air-cooled heat exchangers and cooling towers, Tulsa Penn Well corporation, Tulsa, 2004.

[8] *Cd-Adapco, Best practice workshop: Heat transfer.* [online]. http://www.cd-adapco.com/sites/default/files/Presentation/Heat-Transfer-Best-Practices-Nov2012_PE.pdf

[9] J.C. Kloppers, D.G. Kröger, The Lewis factor and its influence on the performance prediction of wet-cooling towers, Int. J. Sci. 44 (2005) 879-84.