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3D MODEL IN SIMULATION OF HEAT AND MASS TRANSFER PROCESSES IN WET COOLING TOWERS

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Velimir Stefanović, Gradimir Ilić, Mića Vukić,
Nenad Radojković, Goran Vučković, Predrag Živković

Faculty of Mechanical Engineering, University of Niš, FR Yugoslavia

Abstract. *In this paper results of a 3-D numerical simulation of heat and mass transfer processes in wet cooling towers are presented. Physical and mathematical models are presented in suficient measure. All relevant factors and data needed in solving of partial differential equation systems presented in this paper, are given. For solving of this problem special GROUND file in PHOENICS 3.3 software program is formed.*

Key words: *numerical model, simulation of heat and mass exchange, cooling tower.*

1. INTRODUCTION

In direct contact of humid air and water surface simultaneous heat and mass exchange is taking place between these two media. Total heat flux is an algebraic sum of conductive, convective, radiative and evaporative (condensative) heat fluxes. When water is cooled in wet cooling towers the total heat flux can be assumed as a sum of convective and evaporative one, since conductive and radiative heat fluxes take part only in large open poools.

Convective heat transfer is characterised by temperature difference of water and air, while evaporative heat transfer takes place because of evaporated water mass transfer into humid air.

This paper deals with numerical simulation of heat and mass transfer in wet cooling towers. For that purpose, 1-D and 3-D numerical methods are developed.

Considering carefully performed experiments, which included wide range of working parameters, authors had possibility to unite all relevant influencing parameters to heat and mass transfer process in the fill of wet cooling tower.

Results achieved in the improvement of wet cooling towers calculation are obvious, but there is still enough place for improvement of physical, mathematical and also numerical models used in calculations.

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That's why authors looked for a modest contribution in improvement of calculation, especially in 3-D numerical model, trying to catch change of heat and mass transfer coefficients in whole fill volume, that is shown with numerical experiment results given in this paper.

2. PHYSICAL MODEL OF HEAT AND MASS TRANSFER

Physical model of heat and mass transfer on boundary surface between water and air is shown in Fig. 1. It's supposed that water drop, or water film is surrounded with uniform saturated air film on water temperature. It's also supposed that water temperature on drop surface is equal to temperature in section. Same presumption is valid for water film of differentially small length which is shown in the same figure. According to Berman [3], this presumption is valid for heat flux less than 4000kJ/m^{2}h.

If nonsaturated air is in contact with water surface which temperature is higher of air temperature ($t_w > t_v$), both heat fluxes are directed from water surface to air.

If temperatures of air stream and water surface are equal ($t_w = t_v$), convective heat flux equals zero ($\dot{q}_k = 0$), and overall heat flux from water surface is reduced to evaporative heat flux ($\dot{q}_k = \dot{q}_i$). In case of saturated air the equilibrium is reached, and in a case of non - saturated air, water is cooled by evaporation.

When non - saturated air is in contact with water surface, water can be cooled even if air temperature is higher than water surface temperature, until $|\dot{q}_i| > |\dot{q}_k|$. Namely, in that case convective heat flux is directed from humid air to water surface, so overall heat flux from water surface is actually difference between evaporative heat flux and convective one. Water surface temperature at which $|\dot{q}_i| = |\dot{q}_k|$, $\dot{q} = 0$ respectively, is limit for cooling of water under these conditions.

It can be said that the limit temperature for cooling of water is always some lower than humid air temperature, and in limit case is equal to humid air temperature, if it is saturated.

Finally, heat transfer physical model can be mathematicaly written down in the following form

$$\dot{q}^{II} = \dot{q}_k^{II} + \dot{q}_i^{II}, \tag{1}$$

where

$$\dot{q}_k^{II} = \alpha(t_w - t_v) \tag{2}$$

is convective heat flux, following Newton's law, and

$$\dot{q}_i^{II} = \dot{m}_{vl}^{II}\, r \tag{3}$$

is surface evaporation heat flux for water because of converting part of liquid into vapour and diffusive and convective mass (vapour) exchange, taking that water evaporation is done on temperature t_w. Expression $\dot{m}_{vl}^{II}$ [kg/m^{2}s] represent mass transfer per unit area.

Phisical model of mass transfer shown in the same figure is considered over the control volume in which mass $(\dot{m}_{vl}^{II} + \beta_x x_v)$ is entering, and mass $(\beta_x + \dot{m}_{vl}^{II})x_{vs}$ is leaving. For steady state

$$\dot{q}^{II} = \alpha(t_w - t_v) + \beta_x(x_{vs} - x_v)\, r\, . \tag{4}$$

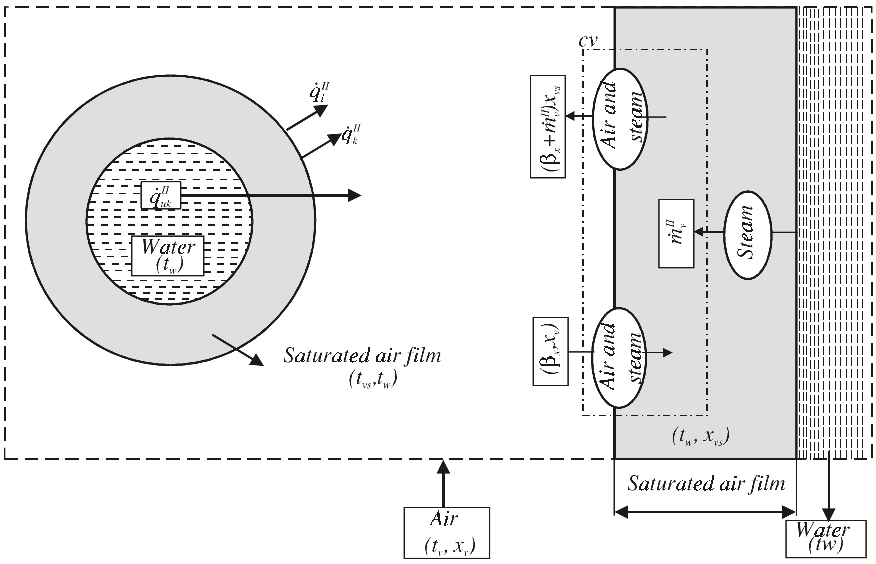


Fig..1. Physical model of the evaporation of water into non - saturated air

Using Lewis number, $Le = \alpha/\beta_x c_{pv} = 1$, and relations for specific enthalpy of air, we get known Merkel heat flux relation

$$\dot{q}^{II} = \beta_x (i_{vs} - i_v) \tag{5}$$

Equation (5) stands for nonsaturated air. However, air in the tower often becomes saturated. In that case, further evaporation of water is performing in the form of fog.

Mass of evaporated water is no longer proportional to term $[x_{vs}(t_w) - x_v(t_v)]$, but is proportional to term $[x_{vs}(t_w) - x_{vs}(t_v)]$. Fog content in air is given as $[x_{vs}(t_v) - x_v(t_v)]$.

Physical model of water evaporation into saturated air is shown in Figure 2.

Heat transfer coefficient α, and mass transfer coefficient β_x, under global consideration of phenomena, are get by an experimental way and their values are the mean ones. Local level considerations are making need to know values of these coefficients in every point of considered domain.

In the fill of a wet cooling tower of finite dimensions and under usual water and air mass flows of the same order of magnitude, water never reach temperature of cooling limit (which is presented by a wet bulb temperature of athmospheric air t_v^*), but only tends to it. Temperature difference between cooled water t_{w2} and wet bulb temperature of air at the fill inlet is called "cooling zone highness", while realized water temperature drop from inlet to outlet, $(t_{w2} - t_{w1})$, is called "cooling zone wideness" or cooling range.

Greater neighbouring to the cooling limits, which means smaller temperature difference $(t_{w2} - t_v^*)$, for defined relation between water and air mass flows ($\dot{m}_w / \dot{m}_a$) can be achieved by enlargement of the fill highness, i.e. by enlargement of contact water – air area with unchanged fill cross section. Cooling of water in the fill of the cooling tower up to cooling limit temperature assumes infinite dimension of fill.

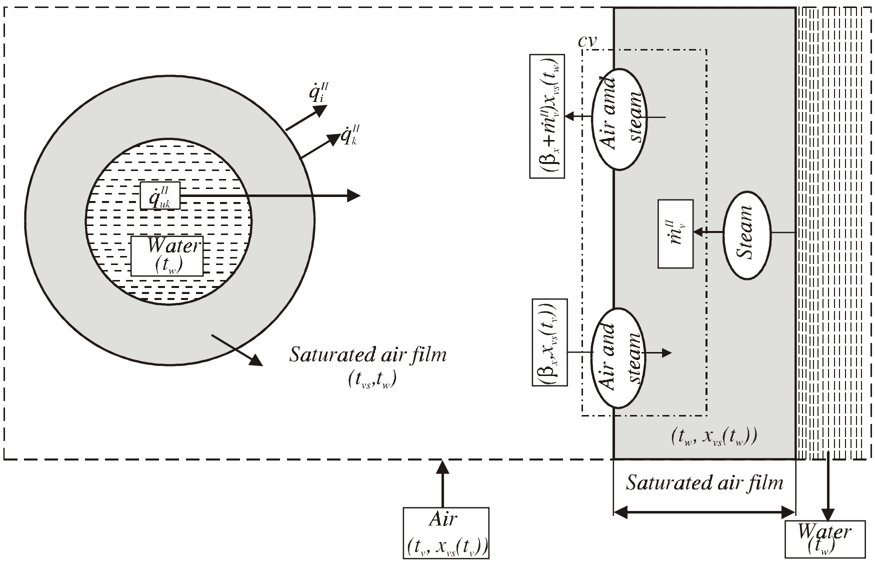


Fig. 2. Physical model of heat transfer from water into saturated air

3. MERKEL'S MAIN EQUATION

In the theoretical formulation of water cooling in counter current cooling tower two heat and mass transfer „driving forces" are defined. For convective heat transfer temperature diffreence between water and air flows have been taken. Mass transfer is expressed in dependence of vapour density differences at phase contact and in the air flow.

This formulation is considered quiet correct. For calculation of cooling towers two empirical values are needed: heat transfer coefficient and mass transfer coefficient.

In the simplified formulation, which was first established by Merkel, convective and evaporative heat transfer is taken by one coefficient, defined as „heat and mass transfer coefficient". "Driving force" is, in this case, the enthalpy difference between saturated air at phase contact and unsataraated air in bulk flow. Simplified equation was named as Merkel's main equation.

In world known standards (DIN, CTI) for calculation of counter current cooling towers, the method based on formulation of Merkel's evaporative water cooling was used, and, therefore, was considered conventional because of wide use.

Starting from relation

$$\dot{m}_a di_v = \beta_{xV}(i_{vs} - i_v)\, dV = \dot{m}_w di_w \,, \tag{6}$$

we get Merkel's main equation in recognisable form

$$\frac{di_w}{i_{vs} - i_v} = \frac{\beta_{xV} dV}{\dot{m}_w} = \frac{\beta_x dF}{\dot{m}_w} \,. \tag{7}$$

By integration of this equation over the variation of temperature t_{w2} to t_{w1}, we get:

$$\int_{t_{w2}}^{t_{w1}} \frac{c_{pw} dt_w}{i_{vs} - i_v} = \frac{\beta_{xV} V}{\dot{m}_w} = \frac{\beta_x F}{\dot{m}_w} \,. \tag{8}$$

This integral was denoted, once, by Berman [3] with greek symbol Ω. However, in world literature in the honour of Merkel it was named Merkel's number and was denoted as *Me*, i.e.:

$$Me = \int_{t_{w2}}^{t_{w1}} \frac{c_{pw} dt_w}{i_{vs} - i_v} = \frac{\beta_{xV} V}{\dot{m}_w} = \frac{\beta_x F}{\dot{m}_w} \,. \tag{9}$$

Analytical solution of this integral is not possible because of complex dependance of entalphy i_{vs} on temperature t_w.

Graphical interpretation of integral (9) is represented on Figure 3. Merkel's number represents area under the curve.

Intensification of heat and mass transfer in cooling towers can be achieved by installing fills which enable greater contact surface of water and air per unit volume of fill. In practice, two basic fills, film and drop, are used.

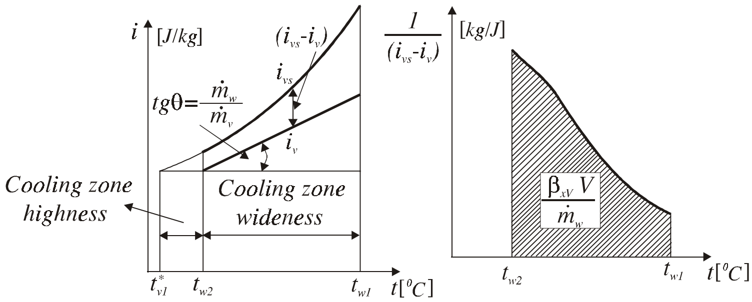


Fig. 3. Temperature – entalphy diagram after Sherwood and graphical expression of the Merkel's integral

Lowe and Christie [6], in the paper which is already concerned classical in the literature, have performed experimental investigations in order to determine heat and mass transfer coefficient β_{xV} for various types of film and drop fills. They have noticed the dependence of β_{xV} coefficient on ratio of water to air flow which can be expressed in the following form

$$\beta_{xV} = A(\dot{m}_w^F)^m (\dot{m}_v^F)^n \,, \tag{10}$$

or

$$\beta_{xV} / \dot{m}_w^F = A(\dot{m}_w^F)^{m+n-1} (\dot{m}_w^F / \dot{m}_v^F)^{-n} \,. \tag{11}$$

For most fills is $m + n = 1$, so we obtain simplified form

$$\beta_{xV} / \dot{m}_w^F = A(\dot{m}_w^F / \dot{m}_v^F)^{-n} \,. \tag{12}$$

If we define air number $\lambda = \dot{m}_v^F / \dot{m}_w^F$, we finaly obtain

$$\beta_{xV} / \dot{m}_w^F = A\lambda^n . \tag{13}$$

Dimensionless form of equation (13) is used in literature. If we take that

$$\dot{m}_w^F = \dot{m}_w / F_T = \dot{m}_w h_I / V , \tag{14}$$

where F_T is cross section area of tower in the level of fill, and h_I is entire fill highness. Based on equation (13) the equation (14) becomes:

$$\beta_{xV} V / \dot{m}_w = \beta_x F / \dot{m}_w = A\lambda^n h_I . \tag{15}$$

Right side of equation (15) is called ***fill characteristic*** in the literature. Coefficients A and n are determined only experimentally with investigation of different kinds of fills. Coefficient λ, which is essential relation between water and air mass flows is called ***air number*** in the literature.

Based on equations (9) and (15) we finally get the equation:

$$Me = \int_{t_{w2}}^{t_{w1}} \frac{c_{pw} dt_w}{i_{vs} - i_v} = \frac{\beta_{xV} V}{\dot{m}_w} = A\lambda^n h_I . \tag{16}$$

With this equation we are establishing relation between ***Merkel's number*** (integral) and ***fill characteristic***.

4. 3-D MATHEMATICAL MODEL

General mathematical model of transport processes in cooling tower is three - dimensional, because mass, heat and momentum conservation equation definition is done in rectangular coordinate system (*x, y, z*). Mass and heat conservation equations for water are given only in the water stream direction (*z*). Tower is symetrical relative to the vertical axis (Figure 4).

Conservation equations in this case are:

Mass of air

$$\frac{\partial}{\partial x}(\rho_v u) + \frac{\partial}{\partial y}(\rho_v v) + \frac{\partial}{\partial z}(\rho_v w) = \dot{m}_{vl}^{III} \tag{17}$$

Mass of water

$$\frac{\partial}{\partial z}(\rho_w w_w) = -\dot{m}_{vl}^{III} \tag{18}$$

Air momentum in x direction

$$u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} + w\frac{\partial u}{\partial z} - \frac{\partial}{\partial x}\left(\mu_{ef}\frac{\partial u}{\partial x}\right) - \frac{\partial}{\partial y}\left(\mu_{ef}\frac{\partial u}{\partial y}\right) - \frac{\partial}{\partial z}\left(\mu_{ef}\frac{\partial u}{\partial z}\right) = -\frac{\partial p_v}{\partial x} - f_x \tag{19}$$

Air momentum in y direction

$$u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y} + w\frac{\partial v}{\partial z} - \frac{\partial}{\partial x}\left(\mu_{ef}\frac{\partial v}{\partial x}\right) - \frac{\partial}{\partial y}\left(\mu_{ef}\frac{\partial v}{\partial y}\right) - \frac{\partial}{\partial z}\left(\mu_{ef}\frac{\partial v}{\partial z}\right) = -\frac{\partial p_v}{\partial y} - f_y \tag{20}$$

Air momentum in z direction

$$u\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial z}-\frac{\partial}{\partial x}\left(\mu_{ef}\frac{\partial w}{\partial x}\right)-\frac{\partial}{\partial y}\left(\mu_{ef}\frac{\partial w}{\partial y}\right)-\frac{\partial}{\partial z}\left(\mu_{ef}\frac{\partial w}{\partial z}\right)=-\frac{\partial \rho_v}{\partial z}-f_z \tag{21}$$

Enthalpy of air

$$\frac{\partial}{\partial x}(\rho_v u i_v)+\frac{\partial}{\partial y}(\rho_v v i_v)+\frac{\partial}{\partial z}(\rho_v w i_v)-\frac{\partial}{\partial x}\left(\Gamma_{ef}\frac{\partial i_v}{\partial x}\right)-\frac{\partial}{\partial y}\left(\Gamma_{ef}\frac{\partial i_v}{\partial y}\right)-\frac{\partial}{\partial z}\left(\Gamma_{ef}\frac{\partial i_v}{\partial z}\right)=\dot{q}^{III} \tag{22}$$

Enthalpy of water

$$\frac{\partial}{\partial z}(\rho_w w_w i_w)=-\dot{q}^{III} \tag{23}$$

Moisture content in air

$$\begin{gathered}\frac{\partial}{\partial x}(\rho_v u x_v)+\frac{\partial}{\partial y}(\rho_v v x_v)+\frac{\partial}{\partial z}(\rho_v w x_v)-\\-\frac{\partial}{\partial x}\left(\Gamma_{ef}\frac{\partial x_v}{\partial x}\right)-\frac{\partial}{\partial y}\left(\Gamma_{ef}\frac{\partial x_v}{\partial y}\right)-\frac{\partial}{\partial z}\left(\Gamma_{ef}\frac{\partial x_v}{\partial z}\right)=\dot{m}_{vl}^{III}\end{gathered} \tag{24}$$

Beside the conservation equations, humid air equation of state was used. In previous equations, the following coefficients are defined:

$$\mu_t=c_\mu\rho_v\frac{k^2}{\varepsilon} \qquad \Gamma_t=\mu_t/\mathrm{Pr}_t \tag{25}$$

where: k - turbulent kinetic energy, and ε - disipation.

By introducing effective turbulent viscosity the equations for turbulent flow are reduced to the same form of laminar flow equations by coefficients:

$$\mu_{ef}=\mu+\mu_t \qquad \Gamma_{ef}=\mu_{ef}/\mathrm{Pr}_t\,. \tag{26}$$

Main properties of defined mathematical model are:

- steadyness,
- heat and mass conservation equations for air are linked by convective fluxes, ($\rho_v u$, $\rho_v v$ and $\rho_v w$),
- momentum conservation equations are pressure linked,
- nonuniform air flow,
- nonuniform water flow,
- three - dimensional change of depending variables for air flow,
- three - dimensional change of depending variables for water flow,
- mass and heat transfer (terms $\dot{m}_{vl}^{III}$ and $\dot{q}^{III}$) are defined separately for rain zone and fill zone,
- flow resistances (f_x, f_y, and f_z) are defined separately for every zone,
- nonuniformity of entering air parameters,
- air pressure is relative to ambient air pressure,

– heat and mass transfer have local character,
– change of water flow parameters is practically three - dimensional although used mathematical model is one - dimensional,
– in this paper k-ε turbulent model is assumed in which turbulent viscosity is expressed through kinetic energy and its dissipation rate, for which special transport equations are solved, while diffusion coefficient are expressed through turbulent viscosity and turbulent Prandtl number.

Boundary and initial conditions

Parameters of water and air in the tower inlet are:
– water mass flow, $\dot{m}_{w1}$ (or rain density $\dot{m}_{w1}^{F}$)
– hot water temperature (entrance), t_{w1}
– dry bulb atmospheric air temperature (t_{v1}) and
– wet bulb atmospheric air temperature (t_{v1}^{*}) or relative humidity, (φ_1)
– atmospheric air pressure, p_{v1}
– air number, λ
– main dimensions, x_T, y_T, z_u, z_k, z_I, z_{iz}
– entering air stream direction.-
Other boundary conditions for every dependent variable are given on Figure 4.

Thermodynamic relations of saturated air for moisture, enthalpy and density of surrounding atmospheric air are used in calculations (equations 31 - 36)

Moisture of saturated air

In expression for heat flux we must know humidity of saturated air, which is function of temperature and pressure.

Partial pressure of steam in saturated humid air is determined by integration of Clausius – Clapeyron equation:

$$\frac{dp_{ps}}{dT_v} = \frac{r\rho_v}{T_v}\,. \tag{27}$$

where by using the equation of state

$$\rho_v = \frac{p_{ps}}{R_v T_v}\,, \tag{28}$$

and expression for latent heat of evaporation given as function of air temperature:

$$r = A_o - B_o T_v\,, \tag{29}$$

and where constants are: $A_o = 3.148856 \cdot 10^6$, $B_o = 2.372 \cdot 10^3$.

For r [J/kg] and T_V [K], values for A_{o} and B_{o} are given for temperature range 0 - 80°C.

By integration of equation (27) we get:

$$p_{ps} = p_{ref} \cdot \exp\left\{ \frac{M_w}{R_u} \left[A_o \left(\frac{1}{T_{ref}} - \frac{1}{T_{vs}} \right) - B_o \left(\ln T_{vs} - \ln T_{ref} \right) \right] \right\}, \tag{30}$$

where T_{ref} and p_{ref} are referent values of absolute air temperature and pressure (T_{ref} = 273K, p_{ref} = 610Pa).

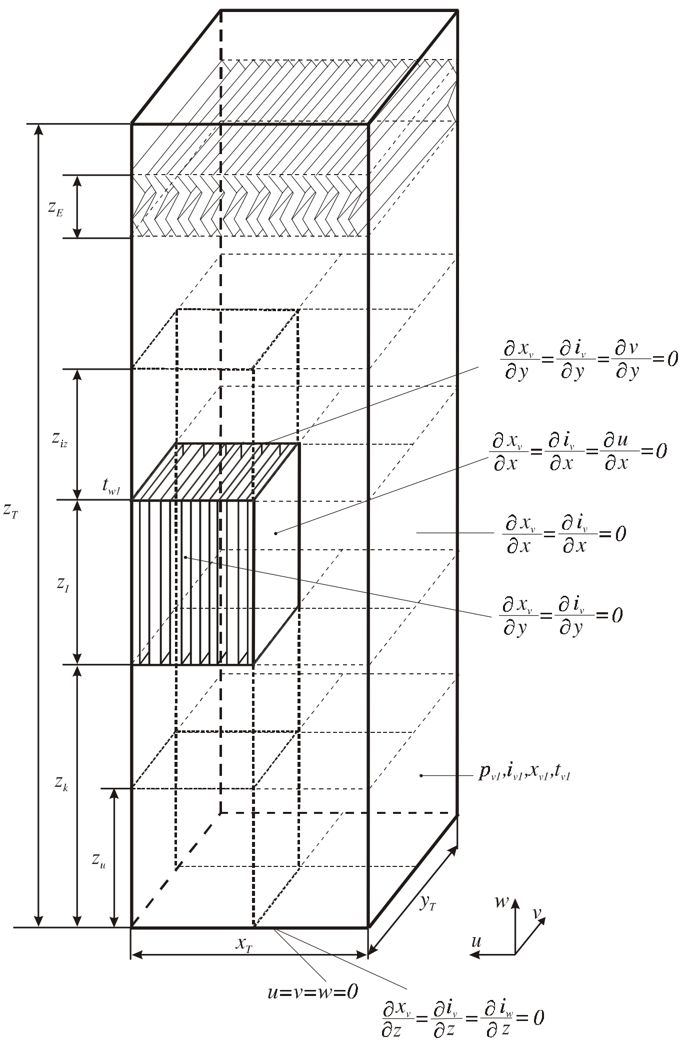


Fig. 4. Characteristical dimensions and boundary conditions for cooling towers

Absolute humidity of saturated air is determined by equation:

$$x_{vs} = 0{,}622 \frac{p_{vs}}{p_v - p_{vs}} \tag{31}$$

and enthalphy of saturated air:

$$i_{vs} = (c_{pa} + x_{vs}\, c_{pp})\; t_v + x_{vs}\, r \;. \tag{32}$$

Parameters of air on the tower inlet

Absolute humidity of entering air is:

$$x_{v1} = 0.6222 \frac{p_{p1}}{p_{v1} - p_{p1}} \tag{33}$$

where:

$$p_{p1} = \varphi_1 p_{ps1} \ . \tag{34}$$

Entering air enthalphy is:

$$i_{v1} = (c_{pa} + x_{v1}\, c_{pp})\ t_{v1} + x_{v1}\, r \ . \tag{35}$$

Density of entering air is:

$$\rho_{v1} = \left(\frac{p_{a1}}{R_a} + \frac{p_{p1}}{R_p} \right) / (273{,}15 + t_{v1}) \ . \tag{36}$$

5. Determination of Heat and Mass Transfer Coefficients and Resistance Coefficients

Heat and mass transfer coefficients in fill

In 1961. Lowe and Christie [6] performed laboratory measurements on large number of fill types for counter current cooling towers, by using Merkels number as method in data processing.

Graphical presentation of equation (16) is given in Figures (5) and (6) for some types of fills. Some types of fills were tested by other authors, showing the deviation of coefficients *A* and *n* up to 10%.

Fills with water film flow enable further progress in improvement of thermal characteristics. In the beginning flat or wavy asbestous - concrete plates, put vertically in the air stream, have been used. Development of plastic materials enables manufacturing of fills with different configurations, with small weight and small price.

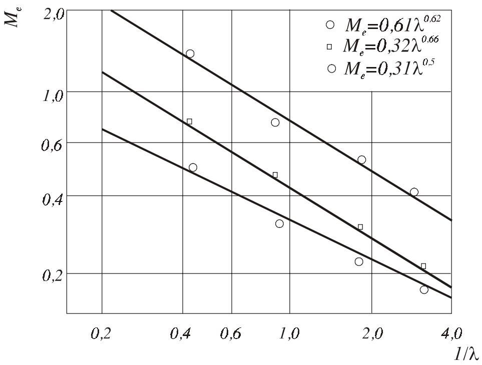


Fig. 5. Merkel's number for film and drop type fills [6]

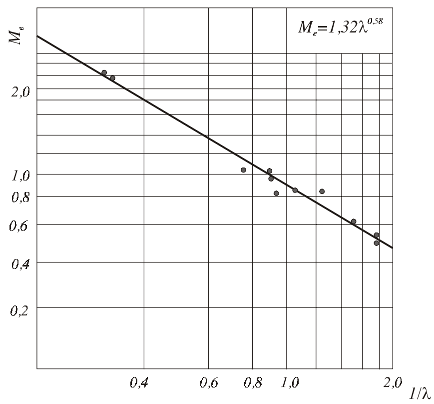


Fig. 6. Merkel's number for wavy plastical fill [9]

Heat and mass transfer coefficients in the rain zone

In the cooling tower, heat and mass transfer, except in the fill zone (FZ) is happening also in spray zone (SZ) and in entering air zone (EAZ), which are defined as rain zone.

Conventional method for cooling tower calculation do not consider heat and mass transfer in the rain zone. Although, in the higher capacity towers rain volume is considerably exceeding fill volume, and must be taken into consideration. In that case, using of conventional calculation method leads to incorrect results.

Equations which describe two – phase flow (sprayed water drops in air) are very complex and their analytical solution is practically imposibile to get.

Newer papers are considering the higher consistency and logic of conservation equation derivation.

Air resistance coefficient

Owerall air resistance coefficient in cooling tower depends on towers construction and density of rain. Exact solution of resistance coefficient value is practically imposibile to get, so the most reliable way of determination of this coefficient is by measurement on already built towers or models. Owerall coefficient is the sum of local coefficients.

Majumdar and others [7, 8] are giving the resistance coefficient two - dimensionally in following form

$$\int f_x dV = \frac{1}{2}(N\rho u^2 \Delta V + N_1 \rho u^2 \Delta A + N_{el} \rho u^2 \Delta A) \tag{37}$$

$$\int f_y dV = \frac{1}{2}(N\rho v^2 \Delta V + N_1 \rho v^2 \Delta A + N_{el} \rho v^2 \Delta A) \tag{38}$$

Analogically, for the third dimension we get:

$$\int f_z dV = \frac{1}{2}(N\rho w^2 \Delta V + N_1 \rho w^2 \Delta A + N_{el} \rho w^2 \Delta A) \tag{39}$$

where ΔV is control cell volume, and ΔA control cell area normal to velocity component. Coefficient N is representing fill resistance, and it is defined over unit length of air stream. N_{el} is eliminator resistance, and N_l resistance of other parts of towers volume and they are given undimensionally.

Coefficient of fill resistance N is determined on the basis of experimental data and it is determined in this paper.

6. ANALYSIS AND COMPARING OF THE RESULTS

Esence of this paper is the determination of local heat and mass transfer coefficients and confirmation of assumption that they are in direct correlation with magnitude of contact area between phases, i.e. with reestablished concentration and temperature gradients.

On the basis of both the real experiment and the numerical experiment including 1-D and 3-D model (which structures are not given here), the authors can withdraw some interesting conclusions.

In the absence of the valid experimental results giving us the basis for valid conclusion about established pressure and velocity field, the indirect possibility for proving only remain i.e. the analysis of the agreement between experimentally obtained and predicted temperature profiles.

Results of a numerical experiment based on 3-D model leads us to conclusion that heat and mass transfer coefficients are changeable over the whole fill volume. Values of these coefficients are changing in the cross section in the range of 1.5–10%, and 1–6% along the height.

Considering that in all known models of calculation, the influence of change of heat and mass transfer coefficients is not taken into account, i.e. mean values of these coefficients are considered, we can conclude that with this aproximation the mistake corresponding to change of this coefficients in the cross section and along the height.

The more detailed results about the temperature profiles and other characteristics are given in [12]. Over 40 experimental runs were done and only a part is given here. The choice is made due to its characteristics. In the following Figures is given the comparision between real and numerical experiments (with 1-D and 3-D numerical model).

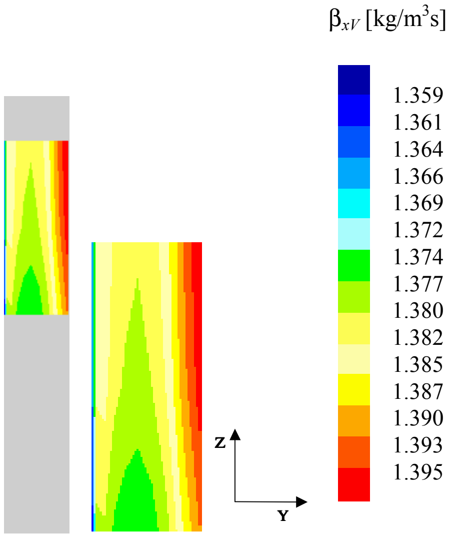


Fig. 7. Local mass transfer coefficient - vertical cross section

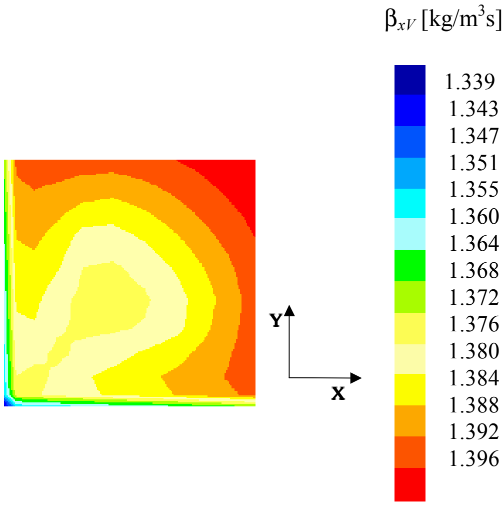


Fig. 8. Local mass transfer coefficient - horizontal cross section at packing inlet

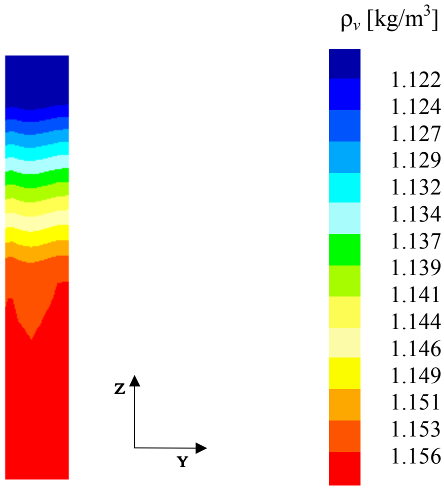


Fig. 9. Air density variation - vertical cross section

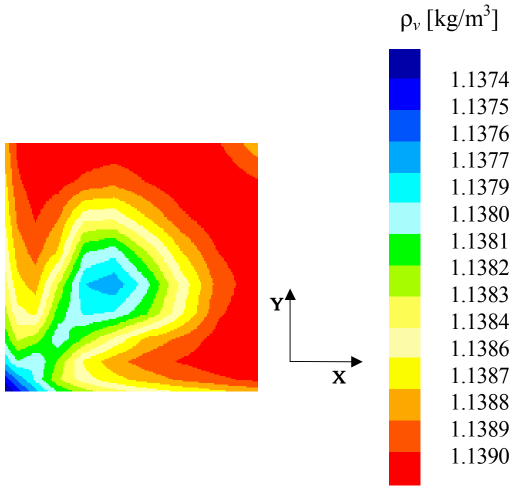


Fig. 10. Air density variation - horizontal cross section

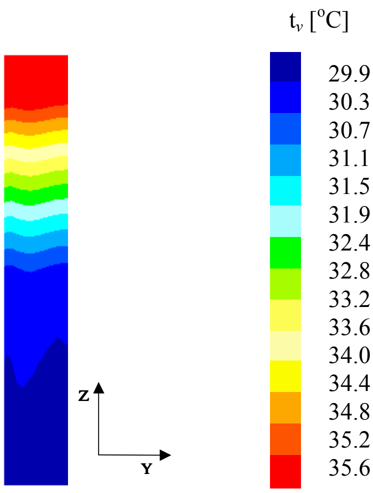


Fig. 11. Air temperature variation - vertical cross section

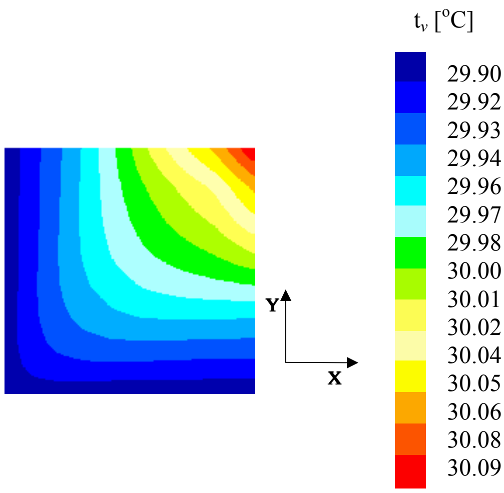


Fig. 12. Air temperature variation - horizontal cross section at packing inlet

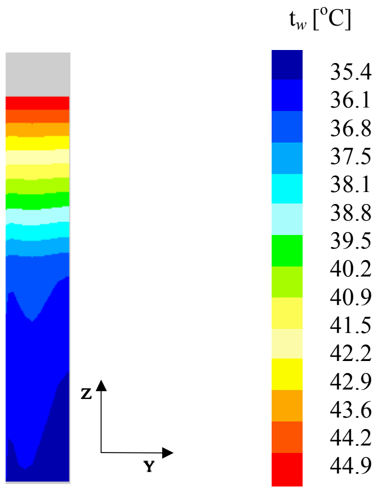


Fig. 13. Water temperature variation - vertical cross section

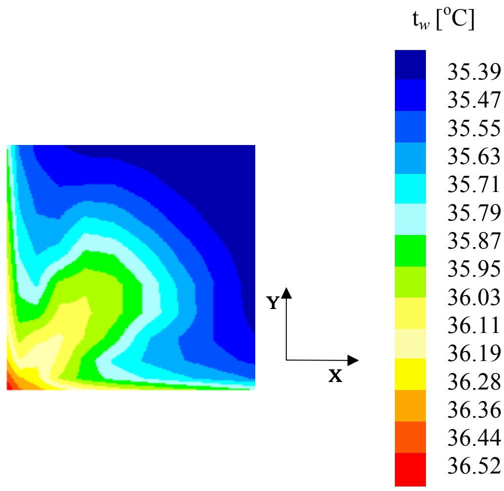


Fig. 14. Water temperature variation - horizontal cross section at packing outlet

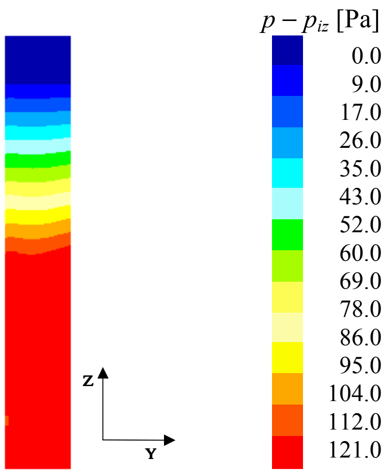


Fig. 15. Relative pressure - center cross section in x axis direction (x–5)

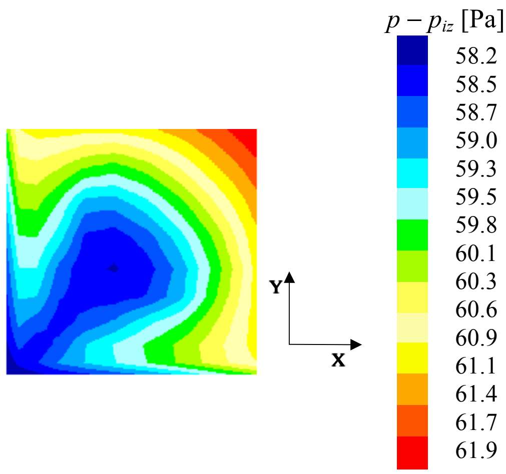


Fig. 16. Relative pressure - center cross section in z axis direction (z–50)

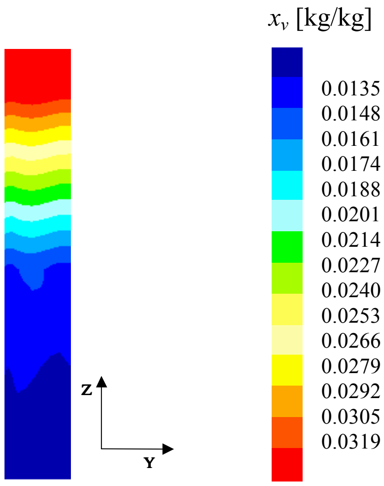


Fig. 17. Absolute moist air humidity - center cross section in x axis direction (x–5)

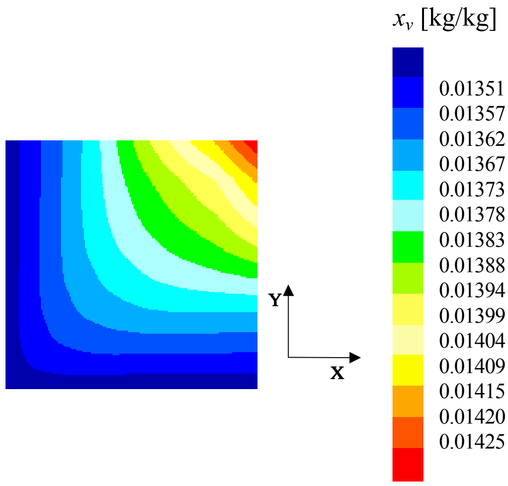
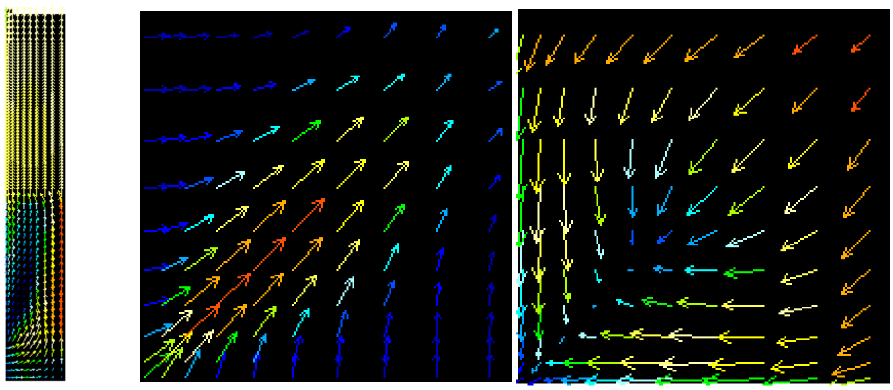


Fig. 18. Absolute moist air humidity - tower inlet (z–1)



min 0.23458m/s
max 3.2833m/s
- fill cross section
(x–5)

min 0.84846m/s
max 1.705m/s
- tower inlet (z–1)

min 1.6508m/s
max 2.7566m/s
- fill inlet (z–25)

Fig. 19. Vector presentation of velocity field

7. CONCLUSION

In this paper special attention is paid to modern methods of solving of diferential equations, with help of computer.

At the end, this paper only confirms, with its essence, all necessity of gathering, wherever it is necesary and unavoidable as in this case, the experimental and quality numerical results in the aim of as better knowing of phenomenon of this or similar type.

The rain droplets have been lifted from the lower edge of packing up to 1/2 of fill height. This phenomenon justifies the assumption of changeable phase contact surface across the whole volume especially along the height of fill. The determination of the phase contact surface area is almost impossible so the majority of the authors in this field are defining the volume averaged heat and mass transfer coefficients in tower fill.

On the basis of analysis of many papers, the authors have concluded that the influence of the phase contact surface variation in the fill is no important, and that contribution of phase contact changing is negligible in comparison to the totally transfered heat and mass, so the averaged values of heat and mass transfer coefficients have been accepted across the fill volume.

The heat and mass transfer coefficients are changeable not only along the fill height but also across the cross section, so the assumption of its invariance leads to error which has the order of magnitude of about 6%, according to our analysis. The final consequence is the increasing of the fill volume.

In 1-D numerical model the influence of the magnitude of the phase contact surface variation, is assumed to be the function of the z-coordinate, and so the good agreement between experiment and prediction has been achieved.

The application of a very sophisticated 3-D numerical model has contributed to the better understanding of such a complex phenomenon as the heat and mass transfer in two phase flow is.

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3-D MODEL SIMULACIJE PROCESA PRENOSA MASE I TOPLOTE U VLAŽNIM RASHLADNIM TORNJEVIMA

Velimir Stefanović, Gradimir Ilić, Nenad Radojković, Mića Vukić, Goran Vučković, Predrag Živković

U ovom radu prikazani su rezultati numeričke simulacije na 3D numeričkom modelu procesa razmene toplote i mase u vlažnim rashladnim tornjevima. U neophodnoj meri prikazan je fizički i matematički model procesa. Dati su svi relevantni činioci i podaci. Koji su potrebni u postupku rešavanja sistema parcijalnih diferencijalnih jednačina, prikazanog u ovom radu. Za rešavanje ovog problema skodiran je poseban GROUND fajl u okviru softverskog paketa PHOENICS 3.3.

Ključne reči: *numerički model, simulacija razmene toplote i mase, rashladni toranj*