Renewable energy utilization in indirect evaporative air coolers under combined airflow conditions

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SUMMARY

The present report concerns mathematical simulation of coupled heat and mass transfer in the heat exchangers used in air conditioning units for the purpose of evaporative air cooling. Heat and mass transfer model based on the one-dimensional transfer model has been developed for thermal calculations of indirect evaporative cooler performance. On the basis of numerical investigations of “classical” heat exchangers designs the new design of air cooler is proposed. The original arrangement of matrix makes possible to realise two-stage indirect evaporative air cooling under combined parallel-counter air flow conditions and raise the degree of air cooling and thermodynamic perfection of this unit at the expense of efficient phase change heat utilisation. The received results show high efficiency of indirect evaporative air cooler under combined parallel-counter airflow conditions and offer scope for estimation of optimum operating conditions range variations and suitable climatic zones for the proposed unit.

INTRODUCTION

Keeping a safe and clean environment is currently becoming a more and more serious problem. One of the possibilities to decrease the pollution is the reduction of conventional energy consumption on behalf of more environmental friendly solution such as the increasing of renewable energy share in the total primary energy structure. Evaporative air cooling is recognized as a key demand-side “opportunity technology”, which offers compelling energy, economic development and environmental benefits at the expense of utilization of thermodynamic unbalance of the atmospheric air.

In the case of direct evaporative cooler, the air stream to be cooled is in direct contact with a liquid water film and cooling is accomplished by the adiabatic heat exchange between the air stream and the liquid water film. The evaporation of water in the air stream leads to reduction of dry-bulb temperature and increasing absolute humidity of the supplied air. In general, the maximum possible reduction in air dry-bulb temperature depends on the difference between the dry-bulb and the wet-bulb temperature of the air stream. The direct type is simple but introduces moisture into the air-stream while cooling it adiabatically. In many applications, however, the increase of humidity in a supply air stream is not desirable. A promising way to increase the thermal performance of evaporative cooling systems is to decrease the dry-bulb temperature of the air at constant absolute humidity. Such kinds of processes can be realized with the help of indirect evaporative air coolers (IEC) [1–7].

Conventional integrated IEC has two distinct air passages, one named as the primary and the other as the secondary air passage. The primary air is usually outdoor air that is supplied to the room after it has been cooled by the air in the secondary air passages through the heat
The secondary air can be either outdoor air or room exhaust air. The surface of the secondary air passage is moisturized, so that heat and mass transfer takes place between the wet surface and the secondary air. The dry primary air stream passes through the heat exchanger and uses the cooled wet secondary stream as its heat sink. Thus, IEC involves sensible cooling of the 'primary' air by an evaporatively cooled secondary air stream. As a result, the temperature of both streams decreases. The two air streams never mix or come in direct contact.

Traditionally, the heat exchanger type employed for IEC is a plate heat exchanger. The evaporative cooling of the wet stream takes place within the IEC itself along the wet passages. Then, the humidified air is dumped into the atmosphere.

There are many construction designs of combined IEC, improving of which goes in the following directions [6–9]:

- application of multistage cooling process within the confines of one unit;
- using capillary porous plates as a structural material for the heat transfer matrix, allowing moisture to fully cover the surface of the secondary channels at the minimum value of water consumption;
- implementation of rational combined management of airflows conditions, allowing to lower the temperature level of the received cold (air coolant);
- searching optimum operating conditions and suitable climatic zones for IEC on the basis of using the modern methods of polyoptimization.

Taking into account that there are merits and demerits for each type of heat exchanger, researches should be carried out not only for the developed air coolers, but also for heat exchangers, realizing "traditional" management of airflows conditions (Fig. 1). Such kind of research allows one to estimate an overall performance of new IEC and their conformity to the predicted purposes and to determine suitable climatic zones of their rational application.

![Figure 1. Schematic diagrams of typical IEC designs. a) Direct-flow, b) Counter-flow, c) Regenerative heat exchanger [6].](image)

However, the substantiation of potential efficiency of heat and mass transfer process and rational performance of the considered units calls for a detailed theoretical study. With this purpose mathematical model of heat and mass transfer was developed for the combined IEC on the base of differential equations of the α-model [4, 5, 10]. It is necessary to note, that mathematical modelling of heat and mass transfer in modern IEC appears rather complicated [1–5, 11]. First of all, it is connected with an increase of the number of airflow passages and, consequently, the problem order.

The present paper is concerned with mathematical simulation of coupled heat and mass transfer in the units, used in air conditioning for the purpose of indirect evaporative air cooling.
METHODS

In many problems, including the said one, an engineer is not interested in receiving thermodynamic variables distributions normal to the heat exchanger plate surface, but bulk average values. Therefore, we gave preference to a one-dimensional transfer model (α-model) [4, 5, 10]. In this case, air stream in matrix passages is considered as a gaseous fluid flow with constant temperature and mass transfer potential in direction normal to plate surfaces, which are equalled to bulk average values.

Now, let us formulate a number of assumptions, concerning the peculiarities of examined heat and mass transfer processes and taken as the base of developed model in the most universal case (Fig. 2):

- driving force of mass transfer is a gradient of absolute humidity of air;
- kinetic properties of air flow and water film are constant, equalled to bulk average values;
- longitudinal molecular diffusion of water vapor in air and longitudinal heat transfer at the expense of thermal conductivity are negligible;
- no wall, air and fluid thermal and moisture diffusivity in the flow directions;
- unrestricted number of passages – \( n \);
- consumed water rate corresponds to sufficient evaporation and keeping up the material of plates in hygroscopic saturated condition, that causes the following water film to air flow heat capacity rate ratio \( \frac{W_w}{W} \ll 1 \);
- the passage walls are impervious to mass transfer;

According to the assumptions of α-model combined heat and water vapour transfer in in the \( j \)th passage of the plate IEC is described by partial differential equations of energy and mass balances developed for the air stream

\[
\left( \frac{W_x}{W} \right)_j \frac{\partial t_j}{\partial X} + \left( \frac{W_y}{W} \right)_j \frac{\partial t_j}{\partial Y} =
\]

\[
= NTU_{j,j+1}(t_{nj,j+1} - t_j) \left( 1 + \left( \frac{c_g}{c_p} \right)_{j} \left( \sigma/Le \right)_{j+1,j} \left( d_{nj-1,j+1} - d_j \right) \right) +
\]

\[
+ NTU_{j,j-1}(t_{nj,j-1} - t_j) \left( 1 + \left( \frac{c_g}{c_p} \right)_{j} \left( \sigma/Le \right)_{j+1,j} \left( d_{nj+1,j} - d_j \right) \right) \text{, (1)}
\]

\[
\left( \frac{W_x}{W} \right)_j \frac{\partial d_j}{\partial X} + \left( \frac{W_y}{W} \right)_j \frac{\partial d_j}{\partial Y} =
\]

![Figure 2. Schematic diagram of multichannel plate IEC.](image)
and equations, describing conditions of energy flow rate continuity on the phase boundary

\[
\left( \frac{\bar{W}_{j+1}}{\bar{W}_o} \right) \text{NTU}_{j+1,j} \left[ \left( t_{uw,j+1,j} - t_{j+1} \right) + \left( 1/c_{pj+1} \right) \left( H_{fj} \sigma / \text{Le} \right)_{j+1,j} \left( d_{uw,j+1,j} - d_{j+1} \right) \right] +
\left( \frac{\bar{W}_{j}}{\bar{W}_o} \right) \text{NTU}_{j,j+1} \left[ \left( t_{uw,j,j+1} - t_{j} \right) + \left( 1/c_{pj} \right) \left( H_{fj} \sigma / \text{Le} \right)_{j,j+1} \left( d_{uw,j,j+1} - d_{j} \right) \right] = 0 , \quad (3)
\]

\[
\left( \frac{\bar{W}_{j-1}}{\bar{W}_o} \right) \text{NTU}_{j-1,j} \left[ \left( t_{uw,j-1,j} - t_{j-1} \right) + \left( 1/c_{pj-1} \right) \left( H_{fj} \sigma / \text{Le} \right)_{j-1,j} \left( d_{uw,j-1,j} - d_{j-1} \right) \right] +
\left( \frac{\bar{W}_{j}}{\bar{W}_o} \right) \text{NTU}_{j,j-1} \left[ \left( t_{uw,j,j-1} - t_{j} \right) + \left( 1/c_{pj} \right) \left( H_{fj} \sigma / \text{Le} \right)_{j,j-1} \left( d_{uw,j,j-1} - d_{j} \right) \right] = 0 , \quad (4)
\]

where  \( j = 1..., n \) – the number of the channel (air stream), \( (j,j-1), (j,j+1) \) - the indexes describing interrelation between the \( j^{th} \) air stream and water film on the plate surfaces of the \( j^{th} \) passage, accordingly adjacent to (with) the \((j-1)^{th}\) and \((j+1)^{th}\) channels, \( \bar{W}_o \) - heat capacity rate of the base air stream, \( W/ K \).

Derivation of the energy balance equations for water films (3), (4) is carried out on the base of the following additional assumptions: temperature difference across the plate and water film on the plate has been assumed to be negligible because of the small thickness of the plate (\( \text{Bi} \ll 1 \)) and liquid film (\( \bar{W}_{uw,j+1} \ll \bar{W}_j \bar{W}_{uw,j-1} \ll \bar{W}_j \)), and, hence, the local temperature of water film can be taken to be the same as the local temperature of the plate

\[
t_{uw,j+1} = t_{uw,j+1,j} , \quad t_{uw,j} = t_{uw,j-1,j} . \quad (5)
\]

Thus, the set of simultaneous equations, describing heat and mass transfer in IEC, includes \( n \) differential equations of energy and mass balances for the air streams (1), (2), \( n \) the algebraic equations of energy balance for water films (3), (4) and \( n \) conditions for local temperature of water films (5). To derive the solution the set of equations (1)–(5) is supplemented by the non-linear empirical relation between absolute humidity of saturated air over liquid film and its temperature \( d_{uw,sat} = f(t_u) \) [12], the boundary conditions, establishing initial thermodynamic parameters values of exchanged air streams on the entries to appropriate channels of IEC. Taking into account the big variety of IEC construction designs and characters of plate moisturizability , the boundary conditions should be taken up for each type of heat exchanger separately. The above formulated nonlinear boundary problem is nonlinear and can not be solve on analytic way. Therefore simultaneous integration of the equations set (1)–(5) with proper boundary conditions was made by numerical methods.

Numerical simulation was executed with the help of the one-step fourth-order Runge-Kutta method, distinguished by sufficient accuracy and stability at the decision of similar type problems. The values of local plate temperature \( t_{uw} \) satisfied by equation (3) and (4) were determined on each step of integration by means of Wegstein’s iteration method used for solving general nonlinear equations of the form \( t_{uw} = f(t_{uw}) \). The described algorithm was realised in digital computer program including 14 modules. The computer program evaluation tests were performed with the help of calculations of different variants, describing problems with familiar analytical solutions [4, 5, 9]. The accuracy of numerical method was checked by computing the identical variants on various space-time domain grids. Convergence test
showed that the optimum number of grid nodes and, consequently, the minimum time of computation within the given tolerance of accuracy (0.5% thermal and mass balance error) was reached at the step size of 0.02 in the direction of axis $X$.

**RESULTS and DISCUSSION**

On the basis of numerous calculations it was established that combined heat and mass transfer in the investigated IEC is characterised by complex and diverse temperatures and humidity profiles, which depend upon the combinations of thermodynamic, hydrodynamic and other factors.

The established distribution of air streams and water film thermodynamic parameters in the direction of primary airflow (Figs. 3, 4) necessitates the need to take into account the changes of plate temperature. Largely it concerns to regenerative IEC, in which changes of water film temperature reach $(6−11)°C$ (Fig. 3. $b$) and to a lesser degree – to direct-flow IEC, where these changes are in an interval $(1÷5) °C$ (Fig. 3. $a$).

![Figure 3. Effect of heat capacity rate ratio $\frac{W_2}{W_1}$ on the liquid film temperature distribution in the direction of primary airflow for two IEC construction designs.](image)

![Figure 4. Paths for the ‘primary’ and secondary air streams on the $i$-$d$ chart for two IEC construction designs.](image)
In the case of the regenerative type of IEC the decrease of water film temperature in the direction of primary airflow allows the lowering of the temperature of the received cold (see Figs. 3, b, 5).

As shown in the Fig. 4, combined heat and mass transfer in the investigated IEC differs substantially in the case of direct-flow and regenerative type. The character of heat and mass transfer in counter-flow IEC needs special discussion.

The analysis of temperature distribution in secondary counter-flow passages brings out the existence of two particular heat and mass transfer areas (2c-O and O-2c in Fig 4.b). The air temperature decreases when the secondary air stream enters the channel (2c-O), reaching a minimum value at a certain distance from the entrance, and then going up in the remainder of the channel (O-2c). This can be explained as follows: the high values of temperature and absolute humidity differences at the entrance of secondary counter-flow channels (2c-O) cause intensive water evaporation and secondary airflow cooling as a result. At the same time liquid film temperature \( t_{w} \) gradually increases because virtually constant sensible heat flow from adjacent primary air stream begins to exceed the evaporative cooling effect. Therefore, the temperatures of the secondary air stream \( t_{2} \) and water film \( t_{w} \) at a certain distance from the flow inlet become equal (point O in the Fig. 4.b). As a result of water film heating on the terminal parts of the plate the temperature of liquid film \( t_{w} \) becomes higher then airflow temperature \( t_{2} \). Thus combined heat and mass transfer process at the exit part of secondary counter-flow channels is characterised by the change of sensible heat exchange direction.

Figure 5 compares the thermal performances of three investigated IEC models for the ranges of heat capacity rate ratio \( 0 \leq \frac{W_{2}}{W_{1}} \leq 1 \).

The received dew-point temperature effectiveness functions of dimensionless operating parameters (Fig. 5.a) are the most advantageous for the regenerative model of IEC. However, from the point of view of cooling capacity \( Q \) such construction design of IEC are worse then direct-flow and counter-flow heat exchangers (Fig. 5.b).

According to the above specified characteristics of the heat and mass exchange processes the new construction of heat exchanger (Fig. 6) was proposed [7]. This construction – owing to the rational implementation of the combined management of airflow conditions (direct-flow
and counter-flow) – results in the following advantages of two-stage cooling process: effective heat and mass exchange at the entrance areas of plates at the parallel airflow conditions bundle and regenerative counter-flow on the terminal parts of the matrix. The heat exchanger contains shell 1, pallet 2, suction branch for total airflow 3, and downstream fairings for primary 4 and secondary 5 airflows. Heat transfer bundle of a regular structure located in package 1 is composed of capillary porous plates 6, one side of which is covered by moisture-resistant coating. Plates are arranged so that they form alternate "dry" 7 and "wet" 8, 9 channels for passing to total I and secondary II, III airflows consequently. There are dividing partitions 10 for organising necessary directions of airflows.

![Diagram](image)

Figure 6. IEC under combined airflow conditions [7]. a) Front view, b) Top view.

The air cooler works as follows. Outside air enters the heat exchanger through suction branch 3 and then is divided on parallel secondary II and total I airflows, the first of which, passing through the "wet" channels 8, causes liquid film to be cooled by evaporation and then is discharged through downstream fairing 5 into the atmosphere. The total airflow I is cooled in "dry" air passages 7, formed by moisture-resistant surfaces of plates 6, due to its heat exchange with secondary airflows II and III passing through "wet" passages, and on the exit from downstream fairing 4 is divided on the primary IV and secondary regenerative III airflows. The primary airflow IV is delivered to air conditioned space, and secondary countercurrent airflow III (after the cooling of water film via evaporation in channels 9) is discharged through downstream fairing 5 in atmosphere. The increasing of the air cooler capacity in comparison with the regenerative indirect evaporative air cooler [6] becomes possible due to the reduction of the secondary regenerative airflow consumption without decreasing total airflow temperature difference. Optimising these distributions with the help of mathematical response surface models offered scope to determine rational performance characteristics of this IEC and the variation range of optimal operating conditions (1,5 <NTU₁< 2,5; 0,8<W₂/W₁< 1; 0,14<W₃/W₁< 0,27; 0,4<l< 0,55).

**NOMENCLATURE**

\[ c_p = \text{specific heat of the air, } J/(\text{kg dry air} \cdot \text{K}); \]
\[ c_v = \text{specific heat of the water vapor, } J/(\text{kg} \cdot \text{K}); \]
\[ d = \text{absolute humidity of air (air humidity ratio), kg/kg dry air (g/kg dry air)}; \]
\[ E_{dpt} = \frac{(t_{e,x} - t_{x})}{(t_{t,x} - t_{dpt})} = \text{dew-point temperature effectiveness, dimensionless}; \]
\[ F = \text{heat and transfer area of matrix, exposed to an air stream, } m^2; \]
\[ G = \text{air mass flow rate, kg/s}; \]
\[ H = \text{latent heat of phase change, } J/\text{kg}; \]
\[ i = \text{specific enthalpy of the air-water vapor mixture, } J/(\text{kg dry air}); \]
\[ l = \text{length of parallel airflow zone, } m; \]
\[ l = \text{relative length of parallel airflow zone, dimensionless}; \]
\[ L_X = \text{plate length in } X \text{ direction, } m; \]
\[ L_Y = \text{plate length in } Y \text{ direction, } m; \]
\[ Q = \text{...} \]
heat flow rate, W/m²; \( t \) = temperature, °C; \( w \) = speed of air flow m/s; \( \overline{W} \) = heat capacity rate = \((G \cdot c_p)\), W/K; \( X \) = axial coordinate along the plate in the direction of primary airflow, m; \( \overline{X} = X/L_x \), \( \overline{Y} = Y/L_y \) = relative coordinates, dimensionless; \( \alpha \) = sensible heat transfer coefficient, W/(m²·K); \( \beta \) = mass transfer coefficient, kg/(m²·s); \( \varphi \) = relative humidity, percent; \( \sigma \) = moisturizability of plate (\( \sigma = 0,0 \) : dry; \( \sigma = 1,0 \) : completely wet), dimensionless.

**NONDIMENSIONAL COMPLEXES**

\[ \text{NTU} = \frac{\alpha \cdot F}{(G \cdot c_p)} ; \text{Le} = \frac{\alpha}{(\beta \cdot c_p)} . \]

**SUBSCRIPTS**

\( \text{dpt} \) = dew-point temperature; \( e \) = state of air stream at the entrance of passage; \( f_g \) = vapor-liquid; \( g \) = water vapor; \( p \) = at constant pressure; \( \text{sat} \) = saturation state; \( x \) = state of air stream on the exit of passage; \( w \) = surface of plate; \( 1 \) = primary airflow; \( 2 \) = secondary parallel airflow; \( 3 \) = secondary regenerative counter-airflow.

**REFERENCES**