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EVAPORATIVE COOLERS OF WATER AND AIR FOR COOLING SYSTEMS. ANALYSIS AND PERSPECTIVES

The concept of evaporative coolers of gases and fluids on the basis of monoblock multichannel polymeric structures is presented. Different schemes of indirect evaporative coolers, in which the natural cooling limit is the dew point of the ambient air, are discussed. In such systems the cooling temperature is lower than the wet bulb temperature of the ambient air. Special attention is paid to the recondensation of water vapor for deep evaporative cooling. It is shown that for the solution of the recondensation problem it is necessary to vary the ratio of the contacting air and water flows, particularly in each stage of the multistage system. Recommendations for the deep cooling process implementation in the evaporative coolers of gases and liquids are given.
Keywords *evaporative cooler, multichannel packing, polymeric materials, coupled heat and mass transfer, recondensation*

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РОЗРОБКА НИЗЬКОТЕМПЕРАТУРНИХ ВИПАРНИХ ОХОЛОДЖУВАЧІВ ГАЗІВ ТА РІДИН. АНАЛІЗ МОЖЛИВОСТЕЙ ТА ПЕРСПЕКТИВИ РОЗВИТКУ.

Розроблено концепцію створення випарних охолоджувачів газів та рідин на основі багатоканальних насадочних структур з полімерних матеріалів. Розглядаються варіанти різних схемних рішень випарних охолоджувачів непрямого типу, в яких межею охолодження є температура точки роси зовнішнього повітря в автономному варіанті використання, тобто виявляється нижче температури по мокрому термометру. Особливу увагу приділено проблемі реконденсації водяної пари при глибокому випарному охолодженні. Показано, що одним із рішень цієї проблеми є варіювання співвідношення витрат контактуючих потоків води та повітря, зокрема, для кожного ступеня охолодження багатоступінчастого охолоджувача.
Ключові слова: *випарний охолоджувач, багатоканальна насадка, полімерні матеріали, спільний тепло- та масоперенос, реконденсація.*

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I. INTRODUCTION

The interest to the opportunities the evaporative cooling steadily increases, that is caused by their low energy consumption and environmental cleanness [1-2, 7-10]. Evaporative water and air coolers (EC) can be used in autonomous mode in refrigeration and air conditioning systems, as well as in dessicant-evaporative coolers, based on heat-driven absorption cycle, when preliminary drying of the air provides high efficiency of subsequent EC in refrigeration systems and heat and humidity treatment of the air in air conditioning systems.

Direct evaporative coolers of air (DEC), water cooling towers (CTW), as well as indirect evaporative coolers (IEC) found wide practical application in different areas. The opportunities of such coolers for possible cooling temperature level are limited by the wet bulb temperature of ambient air t_{wb}, which is the natural limitation of the cooling. This considerably depends on the climatic conditions. Besides the value of t_{wb}, the limitation of the cooling in EC is also determined by the ratio of gas and liquid in heat and mass transfer device. The real value of the temperature limitation of cooling is a little higher than t_{wb}; this should be considered in calculations and design of EC [1].

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The area of practical application of the EC methods is determined by systems with cooling towers and air coolers, refrigeration systems with the cooling of condenser, air conditioning systems for temperature and humidity air handling.

The decreasing of the temperature level of cooling also provides the decreasing of the water quantity used in EC for the compensation of the evaporated water (up to 20-25%) [1-3]

II. SCHEMATIC DIAGRAMS OF SOLAR COOLING SYSTEMS ON THE BASIS OF HEAT-DRIVEN ABSORPTION CYCLE AND EVAPORATIVE COOLERS OF LIQUIDS AND GASES

The concept of solar liquid-desiccant cooling and air conditioning systems designing is shown in Figs. 1 and 2. The principle of indirect regeneration of desiccant is used in dehumidifying part of such systems. The dehumidifying part (left side in Figs. 1 and 2) consists of desorber-regenerator (DBR), absorber-dehumidifier (ABR), solar heating system with solar collectors and tank-accumulator with additional heating source of traditional type (gas or electric heater), heat exchanger of "weak cooled desiccant and strong hot desiccant" flows (HEx), and technological cooling tower for absorber cooling (CTWt).

Dehumidifying cycle: Fresh outside air (state 1) while drying in the absorber (process 1-2) decreases its moisture content and the value of dew point temperature, that provides significant potential for the further cooling. Strong and hot desiccant M and weak cold desiccant N interchange the heat in heat exchanger. When moisture is absorbed from the air by desiccant in the absorber the heat is released. The cooling of the absorber provides an approaching to the isothermal process of absorption and increases the efficiency of the process and the efficiency of the whole system [1]. The cooling of absorber could be realized by outside heat exchanger or this can be absorber conjugated with CTWt. In order to remove the dissolved water vapor from the weak desiccant solution N and to make it strong the energy from the solar heating system is supplied to the desorber (DBR).

In cooling part of the solar absorption system the following solutions are considered: Fig. 1 – chiller air cooler (Ch-Rg) and Fig. 2 – chiller water cooler (Ch-Rw). According to Fig. 1A, the dry air from the absorber enters air-water heat exchanger, where it temperature becomes lower. That the total air flow is divided into two flows: the primary cooled air flow P, which enters the cooling space providing air conditioning, and the secondary air flow S, which enters a direct evaporative cooler (DEC), where it contacts with the film of the water. During the coupled heat and mass transfer, the evaporative cooling of contacting water and air takes place. Cooled but humid secondary air flow cannot be used for the air conditioning purposes, thus it is released into the environment, and the chilled water from the DEC is supplied to the Hex to cool the air from the absorber. In order to use the temperature potential of the exhaust cooled secondary air, an additional air-air heat exchanger can be used (Fig. 1B). In both cases (Fig. 1A and Fig.1B) the cooled aid is supplied to the cooling space for the air conditioning purposes (chiller air cooler (Ch-Rg)).

In chiller water cooler (Ch-Rw) the cooled water and air are supplied to the cooling space (Fig. 2). A part of the air dry air from absorber is supplied to the cooling tower to cool the water needed for the fan coil of the cooling space (Fig. 2A). And the other part of the dry air is supplied to the DEC and than enters the cooling space. From Fig. 2B it can be seen that a part of the dry air enters CTW and the other part directly enters the cooling space. A number of heat exchangers is used to increase the efficiency of the system.

Before in [1, 2, 5 and 6] the comparative analysis of the possibilities of solar liquid-desiccant cooling and air conditioning systems with direct and indirect regeneration of the desiccant was made. Flat plate solar collector (SC) can be used in systems with indirect regeneration of the desiccant, and gas-liquid solar collector regenerator is used in systems with direct regeneration [1]. Each solution has advantages and disadvantages. The results of the present research cover the study of the solar systems with direct regeneration of the desiccant.

III. DESIGN OF THE HEAT AND MASS EXCHANGERS FOR DEHUMIDIFYING AND COOLING CYCLES OF SOLAR COOLING SYSTEM

Chiller air cooler Ch-Rg is composed from evaporative water cooler and water-air heat exchanger, in which the air from EC is cooled at constant moisture content. This decreases the wet bulb temperature, the natural limit of cooling level is decreased, and it can reach the dew point temperature. Chiller water cooler Ch-Rw is composed from CTW and water-air heat exchanger. This provides cooling of water temperature lower than the wet bulb temperature of the ambient air.

Constructive execution of packing for all heat and mass transfer devices (HMTD) of the dehumidifying and cooling cycles is unified. A multichannel monoblock structures made from polymeric materials are used for the packing. This creates the series of channels in which the liquid film (water in EC and desiccant in absorber and desorber) flows down the walls. The mode of contact gas and liquid flows can be counterflow as well as cross-flow. HMTD with high density of packing layers are used [1, 2, 5, 6, 9 and 10]. HMTD can be of direct type (CTWs and air coolers), as well as indirect type, when several processes are realized in one device. For example, the primary air flow (P) is cooled due to evaporative cooling of the liquid film in neighboring alternate channels, when the liquid film interacts with the secondary air flow (S). The separating thin wall can be made from polymeric material because its thermal resistance is comparable to the thermal resistance of liquid film.

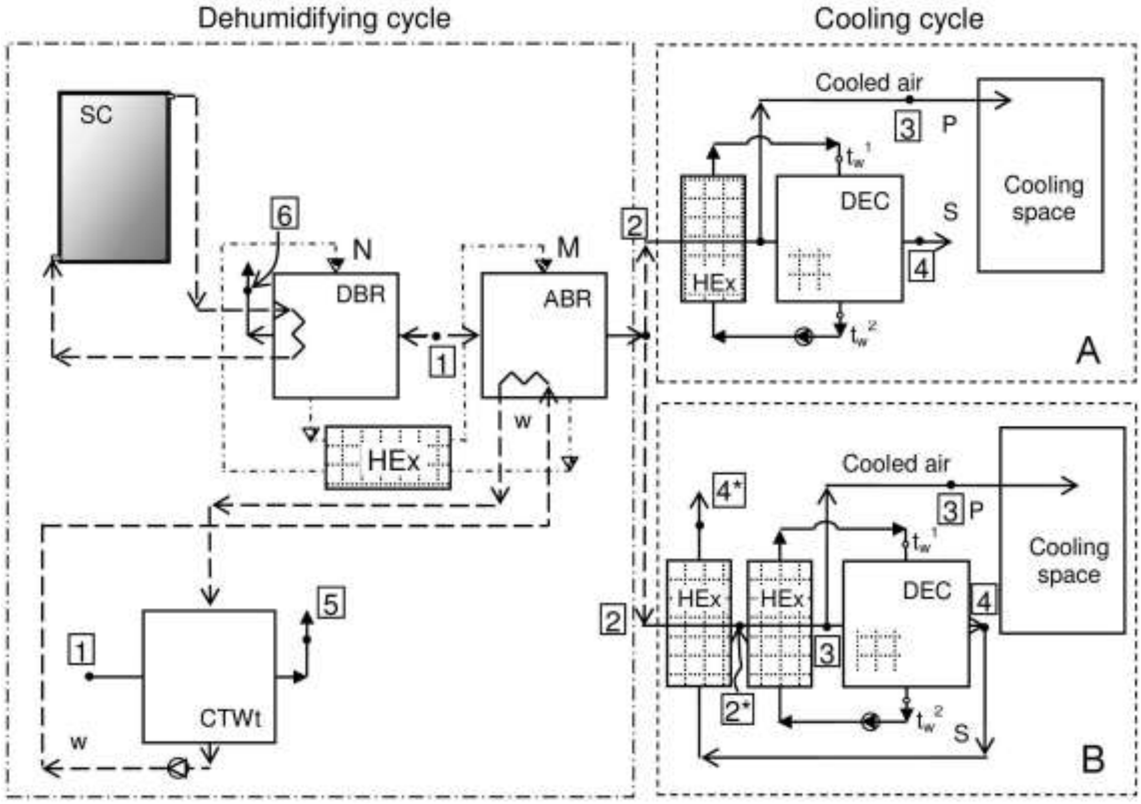

Figure 1− Schematic diagram of the solar heat-driven absorption system with indirect desiccant regeneration on the basis of chiller air cooler Ch-Rg.

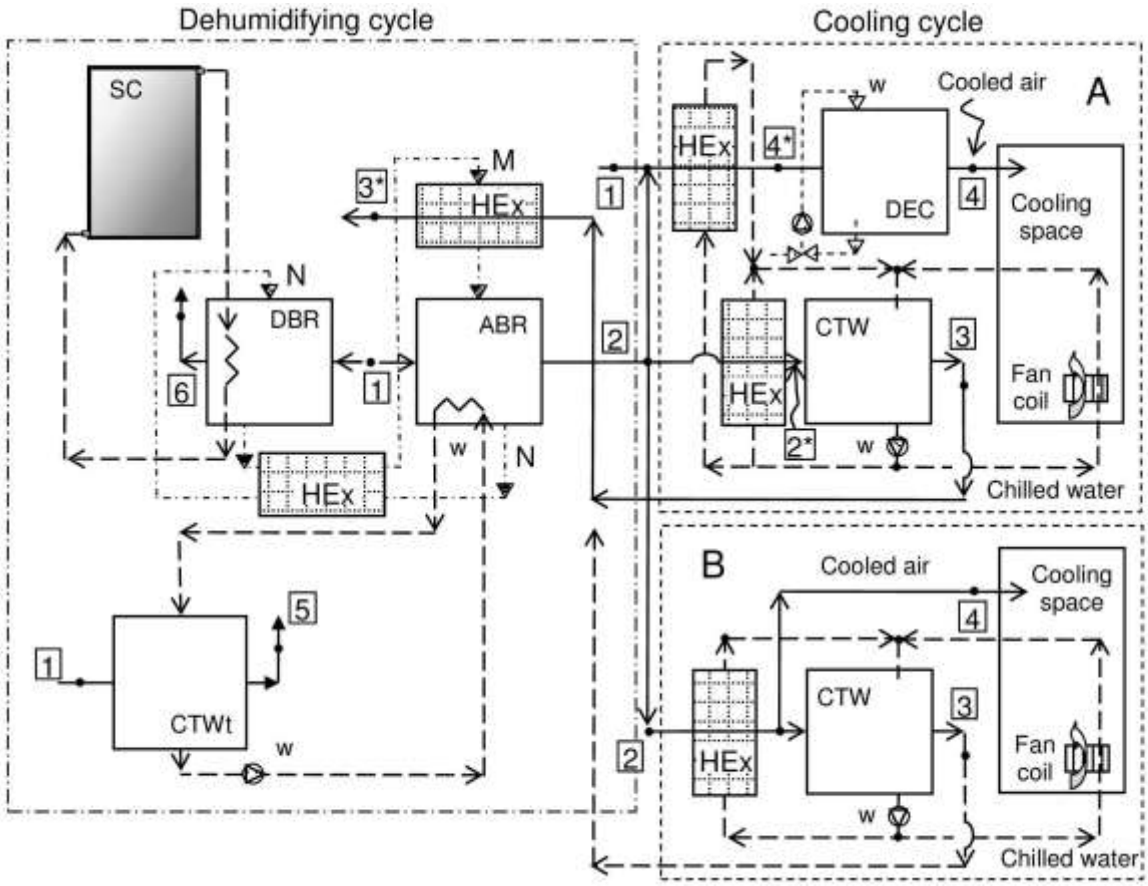

Figure 2 − Schematic diagram of the solar heat-driven absorption system with indirect desiccant regeneration on the basis of chiller air cooler Ch-Rw.

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Such a way an absorber with inner heat exchanger (Figs. 1 and 2), in which the heat, released during absorption of the water vapor by the film of desiccant, is removed by cooled water from technological CTWt. Desorber is made the similar way, where heat from the solar heating system is supplied through the channels. All the HMTDs of dehumidifying and cooling cycles can be incorporated in one cooling unit, which can be placed on the roof of the building as well as inside it.

Solar heating system is based on the application of flat plate solar collectors SC, in which all the elements are made from polymeric multichannel monoblock structures [1, 2, 5 and 6].

IV. THEORY THE HEAT AND MASS TRANSFER IN EVAPORATIVE COOLERS

The process of coupled heat and mass transfer during evaporative cooling is discussed in this section on the example of the direct evaporative cooling of water in CTW. The decreasing of the water temperature is reached by the combined influence of the following processes: 1) heat transfer during contacting (heat transfer due to thermal conductivity and convection); 2) heat transfer due to radiation; 3) surface evaporation of water into the air flow (diffusion of water vapor in the air). The main role here plays the evaporation from the surface (70-80% of heat released from water).

A total quantity of heat, released from water, can be found from:

$$dQ_{\Sigma} = dQ_{\alpha} + dQ_{\beta} \quad (1)$$

$$dQ_{\alpha} = \alpha_g \left(t_w - t_g\right) dF_{\alpha} \quad (2)$$

The assumption is made, that the temperature gradient along the depth of the liquid film is absent and its thermal resistance equals zero: R_w = 0. In studies of [1, 11] it was shown that in general for polytropic process $R_w \neq 0$ и $R_{\Sigma} = R_g + R_w$. The velocity of the vapor molecules from adjoining steam and gas layer being transferred to the air is proportional to the difference ($p^*_g - p_g$), where p_g is a partial pressure of steam in the air located at substantial distance from the water surface (in the core of the air flow). The quantity of the evaporated liquid can determined from :

$$dg_{\beta} = \beta_p \left(p_g^* - p_g\right) dF_{\beta} \ , \quad (3)$$

where β_p – is a mass transfer coefficient divided by full partial pressure difference of water steam (kg/(m^2·s)). The heat consumed during evaporation can be found from:

$$dQ_{\beta} = r \cdot dg_p = r \cdot \beta_p \left(p_g^* - p_g\right) dF_{\beta} \quad (4)$$

Total transferred heat is determined from the following equation:

$$dQ_{\Sigma} = \alpha_g \left(t_w - t_g\right) dF_{\alpha} + r \cdot \beta_p \left(p_g^* - p_g\right) dF_{\beta} \quad (5)$$

The partial pressure difference as a motive power of the mass transfer process can be substituted by the difference of moisture content $\Delta x = \left(x_g^* - x_g\right)$. In this case Eq. (5) will take the following form:

$$dQ_{\Sigma} = \alpha_g \left(t_w - t_g\right) dF + r \cdot \beta_x \left(x_g^* - x_g\right) dF \quad (6)$$

Here the assumption is made, that $F_{\alpha} = F_{\beta} = F$. This factor usually is ignored; in [1, 2] it was shown that for packing with tight structure the influence is great.

$$dQ_{\Sigma} = \beta_x \left[\frac{\alpha_g}{\beta_x} \left(t_w - t_g\right) + r \left(x_g^* - x_g\right) \right] dF \quad (7)$$

where:

$$le = \frac{\alpha_g}{\beta_x} \cong c_p^* \quad (8)$$

For the system of water-air the ratio of heat and mass transfer coefficients is constant. This is the expression of the similarity of the heat and mass transfer process, which take place in the dynamic field of temperatures and moisture contents. The existence of such similarity, which can be expressed by Lewis relation *le*, is dependant from the actuality of the undergoing in the system processes, from the ratio of heat and mass transfer surfaces. It cannot be applied if saturated wet air is used, when «recondensation process» takes place in the region near saturation curve. Neglecting the dependence of *r* from the temperature, the following equation can be obtained:

$$dQ_{\Sigma} = \beta_x \left[c_p^* \left(t_w - t_g\right) + r \left(x_g^* - x_g\right) \right] dF \quad (9)$$

$$dQ_{\Sigma} = K_h \left(h_g^* - h_g\right) dF \quad (10)$$

where K_h – is the total coefficient of heat and mass transfer (according to [11]), divided by enthalpy difference. It shows the coupled heat and mass transfer process intensity, which is defined by joint mechanism of convection and diffusion. Eq. (9) is the main equation of the «the method of enthalpy potential». This equation helps to make it easier the calculation of the heat and mass transfer process, because only one driving force is used instead of two driving forces, which is enthalpy head. Only one coefficient of K_h is used instead of two coefficients of transfer α_g and β_x. When it is necessary to consider the thermal resistance of the liquid film ($R_w \neq 0$), Eq. (10) will take on the following form:

$$dQ_{\Sigma} = \beta_h \left(h_g^+ - h_g\right) dF \quad (11)$$

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where h_g^+ is the enthalpy for $t_g = t^*$ and $\varphi_g = 100\%$.

The analysis of the coupled heat and mass transfer. when direct contacting of gas and liquid takes place, was carried out for the following assumptions :

- the liquid flow rate is constant ($\Delta G_w = 0$); during evaporation or condensation this flow rate will be changed;

- error from the assumption of substitution of dp by dx_g and from the influence of the Stephen mass flow is not big (convective mass flow appearing from the impenetrability of the liquid surface for the air flow; the law of the one-sided diffusion of Stephen);
- assumption, that empirical relation of Lewes equals one ($le=1$). This point is related to the question of assumption about equality of the exchange surfaces ($F_\alpha = F_\beta = F$).
- thermal resistance of the liquid film is negligible $R_w=0$; for polytropic process in system of water-air the thermal resistance of the system is dispersed uniformly between two phases [1, 11];
- additional error can be take place when the driving force is averaged $\overline{\Delta h_g}$.

Approximate methods and the inaccuracy of the averaging are studied in [1]; the approximate methods are based on substitution of the equilibrium curve by straight-line, parabolic or exponential dependence; the values of r and c_p are considered as constants for the design range of main parameters.

The main contribution to the summary error is brought by the assumption $R_w = 0$ and $\Delta G_w = 0$. The value of the error can be from 10% to 15 % [1]. It is necessary to mention that without simplifying background it is impossible to get the Eq. 10.

The equation of enthalpy balance has the following view:

$$\begin{aligned} &G_{w1}c_w\left(t_w^1 - t_0\right) + G_g\left[r_0 x_g^1 + c_p^{*1}\left(t_g^1 - t_0\right)\right] = \\ &= G_{w2}c_w\left(t_w^1 - t_0\right) + G_g\left[r_0 x_g^2 + c_p^{*2}\left(t_g^2 - t_0\right)\right] \end{aligned} \qquad (12)$$

$$G_w c_w dt_w = G_g dh_g \qquad (13)$$

This is the equation of the «working line» of the evaporating cooling process. The main Merkel's equation with consideration of Eq. (13) can be written as follows:

$$\frac{K_h F}{G_w} = \int_{t_w^2}^{t_w^1} \frac{c_w dt_w}{h_g^* - h_g} \qquad (14)$$

The right part of Eq. 14 includes only thermodynamic parameters of the flows; the left part includes constructive and operational characteristics of EC. This makes the Eq. (14) convenient for practical calculations. The value of $K_h{\cdot}F/G_w=K_v$ has a name of «evaporation criteria».

For two relevant cases ($\alpha_l=\infty$ $(R_l = 0)$ and $\alpha_l\neq\infty$ $(R_l \neq 0)$) the main equation of the «method of the enthalpy potential» will be written as Eq. (15):

$$\frac{1}{FK_h} = \frac{1}{F\beta_h} + \frac{\overline{m}}{F\alpha_w}, \quad R_\Sigma = R_g + \overline{R_w} \qquad (15)$$

where $\overline{m}$ is the value accounting the saturation line curvature (tangent slope of the saturation line).

Eq. (15) is the equation of the «additivity of phase resistance». It connects the total thermal resistance in the system ($R\Sigma=1/FK_h$) with thermal resistance of air and water phase $R_g=1/F\beta_h$ and $\overline{R}_w = \overline{m}/F\alpha_w$, respectively. The influence of the resistance of gas or liquid film is determined by the solubility of the gas in liquid. In [1, 2] it is stated, that R_w can be up to 50% from R_Σ. In monograph [11] it is stated that R_w can be 27-46% from the total resistance of the enthalpy transfer between phases.

V. RESULTS OF THE PRELIMINARY ANALYSIS OF THE PERFORMANCE OF SOLAR COOLING SYSTEMS ON THE BASIS OF HEAT DRIVEN ABSORPTION CYCLE AND EVAPORATIVE COOLERS

The analysis of the solar aircooler Ch-Rg for different initial moisture contents of the air x_g^1 is shown in Fig. 3 ($l = G_P/G_S = 1.0$). Here the analysis is based on the previously obtained in ONAFT experimental data about efficiency on the coupled heat and mass transfer process for EC [2, 3]. Initial conditions for point 2 ($x_g^1 = 3$ g/kg) were obtained in solar absorption desiccant system Solar / Ch-Rg (Fig. 3). LiBr (H_2O+LiBr+$LiNO_3$) [2] was used as a desiccant. During the dehumidifying process in absorber (process 1-2), the temperature of the dried air is increased. The temperature rise could be decreased by the application of the technological cooling tower (CTWt) – by the increasing of the contacting flows ratio ($l = G_g/G_w$); but this will result in inevitably increase of power consumption of the system. The following results are obtained for approaching the cooling limit of $\Delta t^{**} = (t_P - t_{dp})$:

$x_g^1 = 16$ g/kg: $t_P = 24.3^0$C, $\Delta t^{**} = 4.0^0$C;
$x_g^1 = 3$ g/kg: $t_P = 10.0^0$C, $\Delta t^{**} = 14.0^0$C.

From Fig. 3 it can be found that the level of primary flow cooling decreases rapidly when the initial moisture content of air decreases, but the degree of approaching the cooling limit is also decreased. The curve for the change of state of secondary air flow G_S (process 2«P» - 2«S») is sequentially cambered in direction of «acute angle» of the h-x diagram of the humid air. Then this curve meets the cooling limit and further follows along the line of φ = 100%. The authors of [2, 3, 11] stated, that during this conditions the «recondensation» of the water vapor can take place, resulting in decreasing of the

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efficiency of EC. For low values of initial moisture content of air it is possible to decrease the value of the primary and secondary air flow ratio $l = G_P/G_S$.

The analysis of the operation of water chiller Solar / Ch-Rw for different initial moisture contents x_g^1 is shown in Fig. 6 (for the same air flow ratios $l = G_g/G_{l\Sigma} = 1.0$ and similar distribution of cooled in EC water between circulating cycles in water-air and «product» heat exchangers $l^* = G^1{}_l/G^2{}_l = 1.0$).

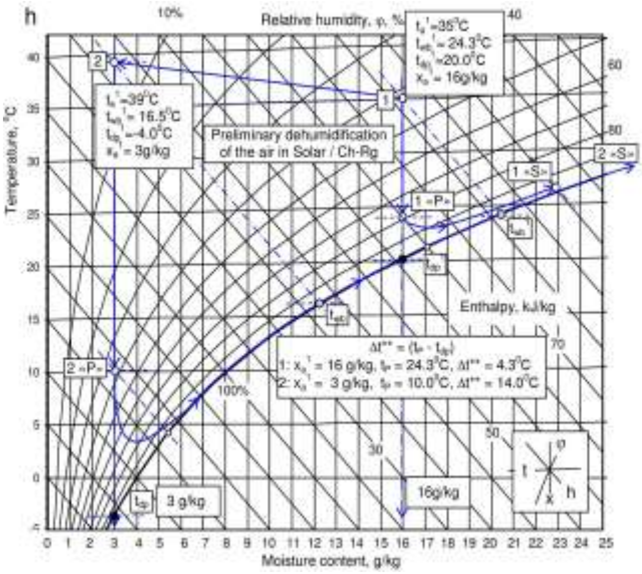

Figure 3 − Performance analysis of the Ch-Rg ($l = G_P/G_S = 1.0$).

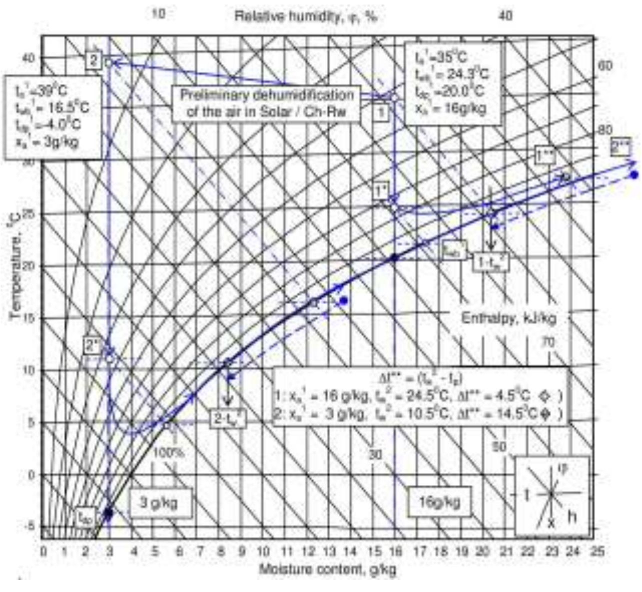

Figure 4 − Performance analysis of the Ch-Rw ($l = G_a/G_{w\Sigma} = 1.0$ and $l^* = G^1{}_w/G^2{}_w = 1.0$)

Initial conditions for point 2 ($x_g^1 = 3$ g/kg) were obtained in solar absorption desiccant system Solar / Ch-Rw (Fig. 4). The following results are obtained for approaching the cooling limit of $\Delta t^{**} = (t_w^2 - t_{dp})$:

$$x_g^1 = 16 \text{ g/kg},\ t_l^2 = 24.5^0\text{C},\ \Delta t^{**} = 4.5^0\text{C};$$

$$x_g^1 = 3 \text{ g/kg},\ t_l^2 = 10.5^0\text{C},\ \Delta t^{**} = 14.5^0\text{C}$$

From Fig. 4 it can be found that the level of water cooling decreases rapidly when the initial moisture content of air decreases, but the degree of approaching the cooling limit t_p is also decreased. The approaching degree of cooling in Ch-Rw is a little lower than in Ch-Rg. The curve for the change of state of secondary air flow in CTW is sequentially cambered in direction of the «angle» of the h-x diagram of the humid air. Then this curve meets the cooling limit of t_p. For low values of initial moisture content of air it is possible to increase the value of the ratio $l^* = G^1{}_w/G^2{}_w$ in the cycles of the cooling water for the efficiency increasing. The comparison of Ch-Rg and Ch-Rw shows that they allow production of «product» flow with almost the same cooling temperature and the ratio of «product» and «general» flow is similar (with similar power consumption). Solar absorption desiccant system (with Ch-Rg or Ch-Rw) can provide the temperature of the air $t_P = 10.0\ ^0$C and water $t_w^2 = 10.5^0$C, that significantly increases the capabilities of the practical application of the EC methods in cooling and air conditioning systems.

CONCLUSIONS

The transition from traditional indirect evaporative cooling of water or air to the evaporative cooling of ambient air with preliminary cooling (chiller air-coller Ch-Rg or water cooler Ch-Rw) allows decreasing of the achieved temperature of the cooled air; the limitation for such cooling is the temperature of dew point of the ambient air; this sufficiently increases the capabilities of practical application of such new evaporative coolers,

Solar liquid-desiccant cooling and air conditioning systems are designed and the main creation principles of such systems are discussed. Indirect regeneration (reduction) of the desiccant is used in dehumidifying part composed of desorber-regenerator, absorber-dehumidifyer, solar heating system. The cooling system can be composed from chiller air-cooler Ch-Rg, when the primary cooled air flow is delivered to the cooling space, or from chiller water-cooler Ch-Rw, when a cooling space is supplied by chilled water.

The main requirement for implementation of evaporative cooling in such coolers is the necessity of contacting flows variation (primary and secondary air flows) and water flow rate. Without such variation of flows the efficiency of cooling could be decreased.

The recondensation problem can influence the efficiency of Solar / Ch-Rg or Solar / Ch-Rw greatly. In order to solve this problem the ratio of the contacting flows for evaporative cooling should be found properly.

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NOMENCLATURE

F area (m^2)
c_p constant pressure specific heat (kJ kg^{-1} K^{-1})
G mass flow rate (kg s^{-1})
p pressure (Pa)
Q heat flow (W)
h enthalpy (kg kJ^{-1})
r heat of vaporization (kJ kg^{-1})
t temperature (^oC or K)

Greek letters
α heat-transfer coefficient (W m^{-2} K^{-1})
β *ma*ss transfer coefficient (kg m^{-2} s^{-1})
φ relative humidity (%)

Subscripts
a air
g das
P primary
S secondary
wb wet bulb
dp dew point
w water
1 entrance
2 exit

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РАЗРАБОТКА НИЗКОТЕМПЕРАТУРНЫХ ИСПАРИТЕЛЬНЫХ ОХЛАДИТЕЛЕЙ ГАЗОВ И ЖИДКОСТЕЙ. АНАЛИЗ ВОЗМОЖНОСТЕЙ И ПЕРСПЕКТИВЫ РАЗВИТИЯ.

Разработана концепция создания испарительных охладителей газов и жидкостей на основе многоканальных насадочных структур из полимерных материалов. Рассматриваются варианты различных схемных решений испарительных охладителей непрямого типа, в которых пределом охлаждения является температура точки росы наружного воздуха в автономном варианте использования, то есть оказывается ниже температуры по мокрому термометру. Особое внимание уделено проблеме реконденсации водяных паров при глубоком испарительном охлаждении. Показано, что одним из решений этой проблемы является варьирование соотношения расходов контактирующих потоков воды и воздуха, в частности, для каждой ступени охлаждения многоступенчатого охладителя.

Ключевые слова: *испарительный охладитель, многоканальная насадка, полимерные материалы, совместный тепло-и массоперенос, реконденсация.*

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