Thermodynamic Fundamentals of Indirect-Evaporative Air Cooling and Specific Application Examples

Part-1. Thermodynamic Fundamentals of Indirect-Evaporative Air Cooling

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This article presents some results of the authors' several-decade-long extensive R&D in the field of air evaporative cooling technologies and techniques. The results of these efforts allowed the development of advanced, highly efficient, economical, and pollution-free Regenerative Indirect-Evaporative Air Coolers (RIEACs) of various designs. Some of these RIEACs are currently being manufactured and successfully used for different applications.

The continuously increasing imbalance between the energy demand and supply, together with escalating cost of the conventional energy resources - as well as growing environmental pollution - are forcing people to expand utilization of renewable energy resources, especially solar radiation, for the cooling and heating needs. Among "free" energy sources is a natural phenomenon: psychrometric non-equilibrium or "Psychrometric Temperatures Difference" - (PTD) of the unsaturated ambient air, containing variable amounts of water vapors. The PTD numerically represents a difference between values of the air's dry and wet bulb temperatures (t_c – t_M). In developed countries, the refrigeration-based air conditioning is one of the largest pieces of the total power pie consumption. For the globe's hot and dry regions, the PTD value of the summer ambient air could be as much as 25ºC, and that provides excellent opportunities for the wide economical and efficient usage of the evaporative air cooling technologies for air conditioning, process cooling, and other related applications. Usage of traditional refrigeration technology for these applications would significantly increase power consumption, hence an increase in environmental pollution produced by the thermal power generating plants.

An adiabatic or isenthalpic air cooling process occurs during direct contact of unsaturated air with water. It *is not a cold production process,* since the initial heat content (enthalpy) of the air within that process stays unchanged.

The air cooling process, utilizing its sensible PTD, could easily be realized in the indirect-evaporative air coolers where the ambient airstream flowing along the dry side of the heat transfer surface of the energy exchanger is usually cooled by another (auxiliary) interacting airstream flowing on the opposite (wet) heat-mass transfer side of the energy exchanger, due to evaporation of water from its wet heat-mass transfer surface.

Realization of the heat-moisture transport/transfer process taking place in the indirect-evaporative air cooling devices requires the presence of two energy exchanging airstreams interacting with each other as follows:

- The total main airstream (which, in this case, is the useful airstream, see Scheme 1a on Fig. 1) transfers its excessive heat to the evaporating water via forced convection through the dividing wall of the energy exchanger. That airstream, due to the sensible cooling process, decreases its dry bulb temperature and its heat content. Then, the cooled useful airstream can be directed into the warmer space, which requires cooling for assimilation of excessive heat, and, possibly, moisture.

- The auxiliary airstream (see Scheme 1a on Fig. 1) flows along the wet surfaces of the energy exchanger and due to the heat-mass transfer process taking place there, it absorbs certain amounts of evaporated water vapors (latent heat) coming from the wet surface of the energy exchanger's dividing wall due to energy (sensible heat) being transmitted from the warmer total main airstream. That energy exchange process between the total main airstream on the dry side, and the auxiliary airstream on the wet side of the energy exchanger, resulted in the dry (sensible) cooling of the total main airstream at its constant moisture content. At the same time, on the "wet" side of the energy exchanger's dividing wall, a heat-mass transfer process takes place, resulting in the auxiliary airstream's moisture content and temperature increase to such degree that this air is practically not suitable for cooling of the occupied space, and it has to be dumped outside. It should be specifically noted that the temperature of the evaporating water in the Regenerative Indirect-Evaporative Air Coolers (RIEAC) is always above the dew-point temperature (DPT) of the mentioned total main airstream.

For the conventional Indirect-Evaporative Air Coolers, which are equipped with either cross-or- counter-flow air-to-air energy exchangers, the theoretical limit of the lowest achievable temperature of the sensibly-cooled total main airstream is the wet bulb temperature of the auxiliary airstream. Usually, in the above-mentioned indirect-evaporative coolers, the "useful" airstream is ambient air, while the auxiliary airstream could be either the same ambient air or the building exhaust air. Instead of using the interacting airstreams circuitries of the conventional indirect evaporative air coolers, we propose a different and more efficient solution for interacting airstreams flow patterns as shown in Scheme 1c on Fig.1.

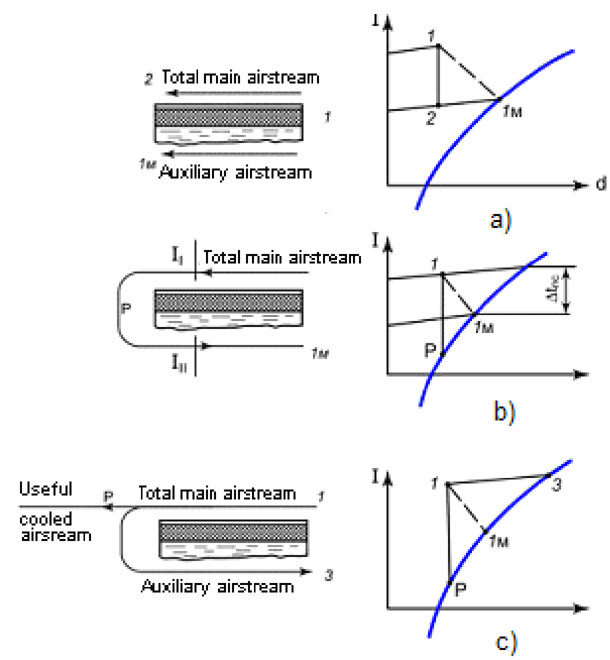

Fig. 1.

Fig. 1 schematically depicts individual energy transfer elements being used for several different patterns of the interacting airstreams.

To establish a proper method of getting an optimal air cooling effect based on utilization of the ambient air PTD, let's review the processes, which take place when an unsaturated airstream flows along the flat plate, one side of which is dry, while the opposite one is permanently wet (see Scheme 1b on Fig. 1).

Let's also assume that the plate's thermal resistance value is zero, the plate's surface area is infinite, and the heat exchange with circumambient is absent.

For the above assumed conditions, let's evaluate three specific configurations of the interacting airstreams (Schemes 1a, 1b and 1c on Fig.1) and define their applicability and theoretical lowest temperature limit of the cooled airstreams:

- **Fig. 1a.** The two interacting airstreams are at equal initial conditions and flowing parallel to each other in the same direction along the flat surfaces of the heat-mass-transfer element. The theoretical lowest limit temperature of the airstream being dry cooled, is its wet bulb temperature.
- **Fig. 1b.** The airstream to be cooled at first is moving along the dry heat transfer surface of the plate and then, at the end of the plate, it makes a 180^0 turn and continues to flow along the wet side of the plate. Thus, the auxiliary airstream exits the wet side of the plate at conditions where values of its dry and wet bulb temperatures become equal to each other, and, at the same time, both are equal to the wet bulb temperature of the intake air. In other words, the adiabatic air cooling process takes place. The temperature difference between the airstreams on both sides of the heat-mass transfer plate suggests that the heat flow moves from the airstream moving along the dry surface of the plate to the air stream moving along the wet side of the plate. The transmitted heat warms and evaporates water on the wet side of the plate, resulting in cooling of air, which is moving along the dry side of the plate. The water vapors are continuously swiped out and absorbed by the moving wet airstream.
- **Fig 1c**. The total main airstream is indirectly (sensibly) cooled (at its constant moisture content d=const) on the dry side of the energy exchanger. After that, its certain predetermined portion (an "auxiliary airstream") is extracted, adiabatically cooled on the wet side of the energy exchanger, and subsequently used for the sensible cooling of the total main airstream.

Let's denote enthalpies of the interacting airstreams as follows:

- I_1 - enthalpy of the ambient air at its dry & web bulb temperatures of t_{1db} & t_{1wb}* respectively enter into the energy exchanger
- I_{dp} - enthalpy of the ambient air at its dew point temperature
- I_2 - enthalpy of the sensibly cooled total main airstream at its splitting point
- I_3 - enthalpy of the auxiliary airstream exiting wet side of the energy exchanger.

*t_{1db} & t_{1wb} are respectively dry bulb and wet bulb temperatures of the ambient air entering into dry channels of the energy exchanger

From the energy balance equation $I_1 – I_{dp} = I_2 – I_{dp}$ it follows that $I_1 = I_2$. Thus, the value of the temperature difference between the dry and wet interacting airstreams, taken at any cross-section point of the energy transfer plate, equals to the PTD of the air being dry cooled.

Since the main airstream, moving along the dry heat transfer surface, is being cooled at its constant moisture content, the temperature difference value between the interacting airstreams across the energy transfer plate at the air's splitting/turning point would be equal to "zero". At the same time and at the same point, the dry bulb temperature of the cooled airstream would reach its dew-point value.

Thus, with accepted assumptions, the total main airstream flowing along the dry side of the energy transfer plate, while the opposite one is wet, is being cooled down to its DPT, while the auxiliary airstream moving on the wet side of the plate is increasing its moisture content and dry bulb temperature up to the parameters, corresponding to the wet bulb temperature of the ambient air entering the dry side of the energy transfer plate. The series of conducted experiments have proven the above statements. The test data of the experiments is presented in the Table 1A below.

Table 1A (SI Units)

№	№ of test regime	Parameters of the Interacting Airstreams of the Regenerative Indirect Evaporative Air Cooler									Air velocity in dry & wet channels
		Total main airstream entering dry side of the energy exchanger			Total main cooled airstream at the splitting point of the energy exchanger (air turning point)			Auxiliary airstream exiting the wet side of energy exchanger			
		t_{1db}, ºC	t_{1wb}, ºC	d_1, g/kg	t_{2db} ºC	t_{2wb}, ºC	d_2, g/kg	t_{3db}, ºC	t_{3wb}, ºC	d_3, g/kg	V, m/s
1	40	35	17.1	4.9	7.2	3.6	4.9	24	17.3	10.6	2.8
2	42	30.1	16.8	6.3	10.1	7.6	6.3	20	17.1	11	3.0
3	46	40	18.5	6.5	10.2	8.0	6.5	23.8	18.8	14.3	3.0

<u>Some data of the tested experimental Regenerative Indirect Evaporative Air Cooler, configured in accordance with the Scheme 1b on Fig. 1:</u>

1. Overall dimensions of the experimental Unit are:
   - Length L = 32" = 784мм
   - Width W = 8" = 196мм
   - Height H = 6" = 147мм

2. The air flow rate during all experiments was within the range of 170-190м^3 /hour (100-112 CFM).

3. The dew point temperature of the inlet air during all experiments was within the range of 3-8^{0}C (37.4 – 46.4^{0}F).

The main purpose of that particular test was to define the theoretical cooling limit temperature of the Regenerative Indirect Evaporative Air Cooler.

Tables 1A & 1B illustrate the character of changing parameters of the appropriate airstreams, which take place in the operational experimental RIEAC, configured per Scheme Fig. 1b. Tables' 1A & 1B data were obtained while testing the energy exchanger of the RIEAC, and the test data confirmed the assumption. The value of the dry bulb temperature of the cooled total airstream at the splitting point (auxiliary airstream offshoot point) is close to the DPT of the cooled air, while the value of the wet bulb temperature of the warm and wet auxiliary airstream exiting the energy exchanger approaches the value of wet bulb temperature of the intake air. It should be specifically mentioned, that within that process no refrigeration energy is produced, because enthalpies of the airstreams entering and exiting the energy exchanger stay invariable.

The above arguments and presented test data confirm an important fact that the lowest dry bulb temperature of the sensibly-cooled total main airstream does actually exist and it's located at the airstream's splitting point. Furthermore, its value approaches the airstream's DPT. However, the auxiliary airstream, after separating from the total main airstream at the separation point, moves along the wet side of the plate, where, due to the impact of its PTD, the auxiliary airstream simultaneously increases its dry bulb temperature and moisture content, as seen at point 3 (Fig. 1c). At presumed ideal conditions, the auxiliary airstream exists the wet side of the plate and its temperature and humidity values correspond to point 3 as follows: temperatures $t_1=t_3$ and relative humidity RH=100% (or $\varphi=1$). In that case, the following equation could be written: $I_3-I_{dp} > I_1-I_{dp}$.

From the energy balance it follows that the flow rate of the auxiliary airstream becomes less than the flow rate of the total main airstream. Hence, a certain portion of the sensibly-cooled total main airstream ("useful" airstream) could be utilized for some cooling needs. For instance, it could be used for space cooling. Then, the directional configuration of the interacting airstreams (Fig. 1b), which was reviewed earlier, could be transformed as follows: the sensibly-cooled total main airstream G_T exiting the dry surface of the energy exchange plate would be divided into two separate airstreams: the "useful" airstream G_o and the auxiliary airstream G_B. The "useful" airstream G_o is directed to the space requiring cooling, while the auxiliary airstream G_B makes a 180^0 turn and enters into the wet channel(s) of the energy exchanger and moves along the wet side of the energy transfer plates in counter flow to the total main airstream direction.

Let's name an "Ideal Model" a Model of such Indirect Evaporative Air Cooler, which cools the unsaturated total main airstream down to its dew-point temperature, when the PDT of the interacting airstreams is being fully utilized, and, simultaneously with that, the auxiliary airstream, while maintaining its saturation conditions, is gradually warming up, until its wet bulb temperature will approach the dry bulb temperature of the intake air.

The "Gross Useful Cooling Capacity" Q^{id}_o of the Ideal Model could be expressed by the equation (1):

$$Q^{id}_o = (G^{id}_o)\,(I_1 - I_{dp}) \qquad (1)$$

Where:
G^{id}_o – is a mass flow rate of the useful cooling airstream.

An Energy Balance equation for the Ideal Model could be written as follows:

$$(G_T)\,(I_1 - I_{dp}) = (G^{id}_B)\cdot(I_3 - I_{dp}) \qquad (2)$$

Where:
G_T - mass flow rate of the total main airstream
G^{id}_B - mass flow rate of the auxiliary airstream

Let's establish such a parameter as the "Ideal Specific Mass Flow Rates Ratio" (M^{id}), which represents a ratio between the mass flow rates of the useful and the total main airstreams:

$$M^{id} = G^{id}_{o} / G^{id}_{T}$$

Replacement members in the equations M^{id} with their meanings taken from the equations (1) and (2) would change the equation as follows:

$$M^{id} = G^{id}_{o} / G^{id}_{T} = (I_3 - I_1) / (I_3 - I_{dp}) \qquad (3)$$

The M^{id} defines a portion of the total main sensibly-cooled airstream (or useful cooling airstream), which could be used for the space (or other purpose) cooling.

From the equation (1) follows:

$$Q^{id}_{o} = (G^{id}_{o}) (I_1 - I_{dp}) = (G_{B}) (I_3 - I_1)$$

Then

$$M^{id} = (G^{id}_{B}) (I_3 - I_1) / (G^{id}_{B}) (I_3 - I_{dp}) = (Q^{id}_{o}) / (Q_T) \qquad (4)$$

Thus, the M^{id} characterizes potential cooling capabilities of the ambient air at given temperature and moisture content, and, as it follows from equation (3), its value depends only on the air's initial conditions.

Since the values of the dry bulb temperatures of the interacting total main and auxiliary airstreams are equal to each other at each and any cross-section of the Ideal Model's energy exchanger and the water evaporates into the saturated airstream, then, the heat and moisture transfer processes are proceeding quasi-statically, and it may be assumed that they are reversible. Therefore, according to the Second Law of Thermodynamics, it should maintain the following equality:

$$(\Sigma S_{out}) / (\Sigma S_{in}) = 1 \qquad (5)$$

Where:

ΣS_{out} and ΣS_{in} are respectively sums of the system's output and input entropies,

i.e. the entropy S of the mentioned cooling process is unchangeable.

With respect to the Ideal Model, the following equation could be written:

$$(\Sigma S_{out})/(\Sigma S_{in}) = [(M^{id})\cdot(S_{dp})\cdot(1 + M^{id}) (S_3)] / [(S_1) + (1 - M^{id}) (\Delta d) (S_w)] = 1 \qquad (6)$$

Where:

S_1 – is the entropy of air at point 1 on diagram on Fig. 1b.
S_{dp} – is the entropy of air at point P on diagram on Fig. 1b.
S_3 – is the entropy of air at point 3 on diagram on Fig. 1b.
S_w – is the entropy of the water vapors at point 3 on diagram of Fig. 1b.
Δd – is the difference between air moisture content at points 1 and 3 in diagram Fig. 1b.

The calculations performed, based on equation (6) for the various initial conditions of the ambient air, have shown that the obtained absolute values of the equation (6) are within the range of 1.007 ÷ 1.002. These results validate the assumption that the heat-mass exchange processes, occurring in the Ideal Model, are reversible. This allows achieving of the maximum air cooling effect via evaporating water into the unsaturated auxiliary airstream at the minimal energy and material consumption. Hence, the value of the thermodynamic perfection of the Ideal Model equals 1.0.

All of the above discussions, analysis, and conclusions combined together allowed establishing limiting capabilities of the Indirect-Evaporative Air Cooling Method, and they could be applied as a reference standard for any device of that kind.

The data obtained from tabulation of equation (3) with respect to function M^{id} for the range of ambient air parameters, applicable to the majority of Earth's climate zones, allowed us to develop the Diagram M^{id} (Fig. 2), establishing correlations between the M^{id} values and such ambient air parameters as its dry bulb temperature (t_1), relative humidity (φ), specific humidity (or total moisture content) (d), and enthalpy (I). The right-angled reference grid of the Diagram, Fig 2 below, is formed by the $M^{id\ lines}$ and they cover the entire area of the air's relative humidity lines between $\varphi = 100\%$ and $\varphi = 0\%$.

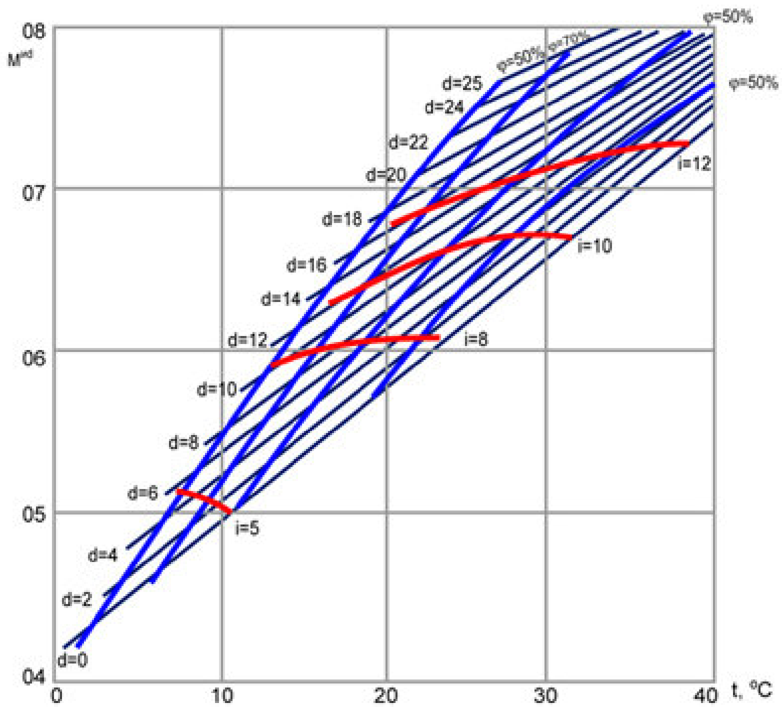

Fig. 2.

Analysis of the Diagram, shown in Fig. 2 allows us to make the following conclusions:

1. Decrease of the moisture content of the air to be cooled resulted in decrease of specific airflow rate of the total main airstream M^{id}. That could be translated into the following chain: getting cooled air at lower temperature would require respective increase of the auxiliary airstream G_B, and that would result in the respective decrease of the available "useful" cold airstream - G_o. In other words, lowering the temperature of the useful cooled airstream G_o, would result in the decreased available volume of the "useful" cooled airstream, as well as in the increased required energy input for "production" of the "useful" cooled air. All of the above is in direct correlation with the general thermodynamic principles.
2. The increased initial temperature of the total main airstream air to be cooled would result in the increased value of the specific flow rate of the total main airstream M^{id}. If temperature of air entering into the energy exchanger would reach $t_1 = 100°C$ (the

water boiling temperature at normal conditions), the water evaporation process would become a water boiling one. In this case, a necessity for the auxiliary air would be eliminated ($G_в = 0$), and $M^{id}_{100} = 1$.

3. At the constant moisture content of the total main airstream being cooled, the value of the ideal specific flow rate ratio (M^{id}) of that stream is approaching its minimal value on the saturated line: $t_1 = t_{dp}$ on the Diagram M^{id} (Fig. 2).
4. The values of the ideal specific air flow rates (M^{id}) for the total main airstream at its constant enthalpy are changing slightly. It's very remarkable seeing some interesting specifics in the Diagram M^{id} (Fig. 2). While moving along the line of the M^{id} = const: in the region of the air's high-moisture contents (d_1), a range of the M^{id} changes is insignificant, while, to the contrary of that, in the region of the air's low-moisture contents (d_1) the value of the M^{id} varies considerably.

Let's state the value of the thermodynamic perfection (or imperfection factor) Z of the Real Indirect-Evaporative Air Cooler's cooling process by comparing it with the Ideal Model. The direct comparison of the cooling capacities for the different temperature levels would be erroneous.

$$Z = E_Q^{act} / E_Q^{id} \qquad (7)$$

Where E_Q^{d} and E_Q^{id} – are the values of exergy relevant to the produced refrigeration energy Q^{act} and Q^{id} by the Real Regenerative Indirect Evaporative Air Cooler and by the Ideal Model respectively.

A general exergy equation could be expressed as follows:

$$E_Q = Q / \varepsilon_к \qquad (8)$$

Where:

- Q – is the amount of generated/produced refrigeration energy
- $\varepsilon_к$ is the refrigerating factor of the Carnot Cycle, taking into account the difference between an actual surrounding temperature and an average temperature of the obtained cold airstream

A substitution of some appropriate members in expression (8) with their matching meanings would result in the following:

$$E_Q^{act} = [(C_p)\cdot(G_o^{act})\,(T_1 - T_2)^2] / (T_1 + T_2) \qquad (9)$$

$$E_Q^{id} = (C_p)\cdot(G_o^{id})\,(T_1 - T_2)^2 / (T_1 + T_{dp}) \qquad (10)$$

where:

- **C_p** - Specific Heat of air at pressure P (in our case P = atmospheric pressure), kcal/(kg)(1^0C temperature change)
- **T_1** - is the initial temperature of the total main airstream entering into the energy exchanger, ^{0}K
- **T_2** – is the final temperature of the cooled total main airstream exiting the energy exchanger, ^{0}K
- **T_{dp}** – is the DPT of the total main airstream within the dry side of the energy exchanger, ^{0}K

Then:

$$Z = (G_o^{act}/G_o^{id})\,((T_1 - T_2) / ((T_1 - T_{dp})^2)\,((T_1 + T_{dp}) / (T_1 + T_2)) \qquad (11)$$

The first multiplier in equation (11) - (G_o^{act}/G_o^{id}) defines a ratio between the mass flow rates of the Actual and Ideal useful cooled airstreams at equal values of the Actual and Ideal total main airstreams, ($G_T^{act} = G_T^{id}$).

So, the ratio of the Ideal and Actual useful cooled airstreams could be expressed as follows:

$$\Lambda = (G_o^{act}/G_T) / (G_o^{id}/G_T) = M^{act} / M^{id} \qquad (12)$$

As it was mentioned earlier, the theoretical minimal temperature limit of the total cooled main airstream is its DPT. Hence, the second multiplier (($T_1 - T_2$) / ($T_1 - T_{dp}$)) in the equation (11) represents an effectiveness ratio (or temperature approaching factor) E_{dp} of the air cooling process relevant to the DPT of the total main airstream. So, the temperature approaching factor could be expressed as follows:

$$E_{dp} = (T_1 - T_2) / (T_1 - T_{dp}) \qquad (13)$$

For the range of the ambient air parameters near-Earth-surface typical and applicable for most of the Globe's climate zones, the numeric value of the third multiplier in the equation (11) is 1.0 and as follows:

$$(T_1 + T_{dp}) / (T_1 + T_2) \approx 1.0 \qquad (14)$$

Then, after proper substitution, the equation (11) of the thermodynamic perfection is transformed into equation below (15):

$$\mathbf{Z = \Lambda\, E^2_{dp}} \qquad (15)$$

So, the extent of thermodynamic perfection considers influence of the two following factors: the quantity of the "useful" cooled airstream, which could be utilized for the active cooling needs and the operating performance efficiency of the IEAC. The equation (15) is suitable for estimation of the thermodynamic perfection of any type of the indirect-evaporative air coolers.

At equal conditions of the ambient air, the extent of the thermodynamic perfection Z for the RIEAC would exceed the ones for the other comparable indirect evaporative air coolers.

In the case when the cooling space requires more comfortable (lower humidity) air conditions, or when the process cooling application prohibits elevated moisture content of the cooling airstream, the IEAC unit should be equipped only with the sensible cooling stage (the energy exchanger) which provides dry cooling of the total main and "useful" airstreams.

For boosting of the cooling output of the IEAC (in cases when the elevated humidity for the non-comfort space and process-cooling application is allowed), the lower dry bulb temperature of the sensibly-cooled discharged air could be achieved by means of installing downstream of the first-dry cooling stage (the energy exchanger) of the IEAC of an additional adiabatic air cooling section, i.e. converting the Unit into the two-stage Indirect-Direct Evaporative Air Cooler.

For the effective operation of the RIEACs in the regions with high–humidity ambient air, the units could be integrated with the inlet (ambient) air drying module. This combination allows it to significantly bring down the dry bulb temperature (down to +5ºC ÷ +7°C) as well as moisture content of the discharge cooled air. This approach greatly expands the use of this cooling technology for the controllable comfort space and process cooling.

The conducted experiments of that hybrid unit have demonstrated that the total main airstream at its inlet moisture content of **d** <2 g/kg could be cooled down to as low as 4-6°Cdb temperature.

Therefore, it could be stated, that the Performance Efficiency and Economics of the "pure" (without using any additional air drying means) Indirect-Evaporative Air Cooling Technology defines possibilities of its wide practical applications, which largely depend on the initial moisture content (or dew-point temperature) of the intake air.

The Indirect Evaporative Air Cooler during summertime is able to provide the indoor air temperature within 25-28°Cdb range at the corresponding relative humidity of ≤60 % if the value of the outdoor air moisture content (**d**) does not exceed 13g/kg.

At the considerable levels of the ambient air moisture content it is necessary to use either conventional refrigeration or RIEACs equipped with air pre-drying means.

Another important factor defining expediency of use of the RIEACs, is their overall performance efficiency being characterized by parameter ξ (Energy Efficiency Ratio), which is a ratio between the net heat assimilating (cooling) capacity (Q_{ncc}) of the unit and the total power input (TPI) into the operational Unit, including power draw by the fan(s), water circulating pump, controls, and misc.

$$\xi = Q_{ncc} / N_{tpi} \quad (16)$$

Where:
N_{tpi} –Total Power Input (TPI) into operating RIEAC, kW
Q_{ncc} – Net Cooling Capacity (NCC) of the useful cooled supply airstream, kW

$$Q_{ncc} = (G_o^{act}) (I^{act}_{indoor} - I_2), \text{kW}$$

Where:
I^{act}_{indoor} – an actual enthalpy of the indoor air of the space being served by the RIEAC.
I_2 - Enthalpy of the cooled total main air stream leaving the dry side of the energy exchanger. The fan's total pressure should be adequate for overcoming the resistance of all internal components (dry and wet sides of the energy exchanger, air pre-filter, etc.) of the RIEAC as well as providing a required external static pressure necessary for overcoming a resistance of the air distribution system, as well as a certain room back pressure, while delivering required volume of the cooled air down to the cooling space.

Total power draw (fan motor + controls + misc. power) by the operating RIEAC Unit:

$$N_{tpi} = N_{fan\ mot} + N_{contr.} + N_{misc}$$

Where:

$N_{fan\ mot}$ - Power draw by fan motor, kW
$N_{contr.}$ – Power draw by the onboard control system. kW
N_{misc} – Power draw by the misc. electrical components, kW

A value of the Energy Efficiency Ratio (ξ) primarily depends on a moisture content of the dry-cooled total main airstream. Table-2 presents the test values of the respective Energy Efficiency Ratios (ξ) being obtained for the constant air dry bulb temperature of 32ºC (entering the energy exchanger) at various moisture contents.

Table-2

Inlet air's moisture content, g/kg	2	4	6	8	10	12
Energy Efficiency Ratio, ξ	24	19	15.6	12.5	10	8.2

However, below are some limitations for the application of the indirect evaporative air cooling means:

- In some hot and humid regions, where the summer-time air moisture content

exceeds 13g/kg
- In premises with high moisture gain (Theatre Halls, Concert Halls, etc.)
- In premises, which require stable temperature and humidity control

A significant energy reduction could also be achieved by use of the hybrid cooling systems - combining the indirect evaporative air cooling means with the conventional mechanical refrigeration means.

Developed RIEAC is of simple general design and could successfully be used for residential, commercial and some industrial applications. The RIEAC could work productively and economically year-round:
- During the summertime it cools the makeup air
- During the wintertime it works as the energy recovery device; pre-heating the makeup cold air by utilizing thermal energy being extracted from the building exhaust air, and saving a significant amount of heating energy.

Conclusion:
1. Extensive analysis of the statistical climate data with respect to ambient air moisture content levels for the major world regions showed a possibility of applying the RIEACs without air drying means for providing optimal and/or admissible parameters of indoor air in premises within these territories (excluding the tropical and subtropical wet zones). The technology of the RIEACs is mature and available for the Global applications.
2. Dr. Tsymerman had patented (a list of patents is available upon request), designed, developed, and fabricated experimental RIEACs of different capacities and configurations for variety of applications. The results of extensive laboratory and field testing of the mentioned IREAC have confirmed that the actual performance characteristics and characteristics established theoretically are very close in value.
3. The RIEACs could with easy be integrated into the existing HVAC systems, which significantly improves their overall performance efficiency.

Information about the authors:

- *Dr. Tsymerman A.B. is a Director of the Thermo-physical Lab. He is the world-known authority in the area of the non-refrigeration cooling and relevant methods and technologies, holding over 100 national and international patents. Dr. Tsymerman had received his MSME degree in Refrigeration Technologies from the Odessa State Academy of Refrigeration Technologies in 1956. In 1985 he had received his PHD degree from the Odessa State Academy of Refrigeration Technologies. The thesis of his dissertation was "Theory and practical realization of the Regenerative Indirect Evaporative Cooling Method". Dr. Tsymerman A.B. is a founder and Director of the Thermo-physical Lab.*

- *Mr. Mike Reytblat, CPE had received his MS degree in Naval Engineering from the Odessa Naval Engineering Academy in 1956, and his MSME degree in HVAC and Gas Supply from the Odessa State University of Engineering and Construction in 1964. Currently Mr. Mike Reytblat is VP of Engineering with Intech-mark.*

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