Thermodynamic Fundamentals of Indirect-Evaporative Air Cooling

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This article presents some results of 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 which 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, increase 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 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 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 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.

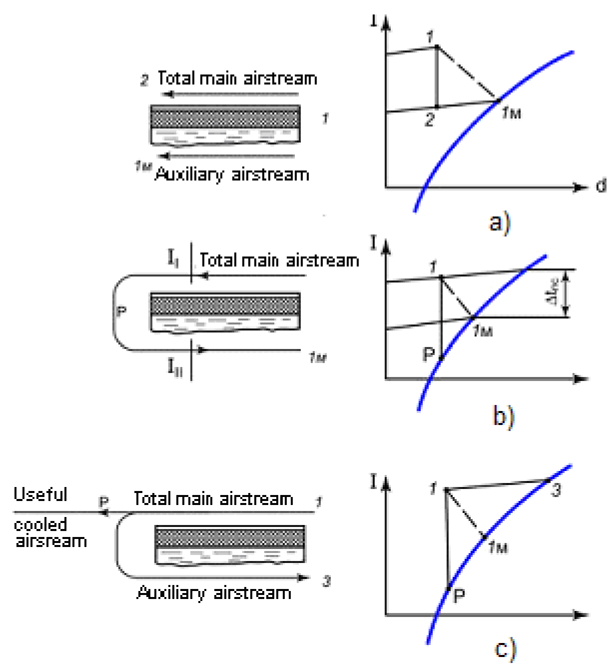

Fig. 1.

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, on 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 dew-point temperature, 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 Tables 1A (SI Units) and 1B (British Units) 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

Table 1B (British 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 the dry side of the energy exchanger			Total main cooled airstream at the splitting point of the energy exchanger (air turning point)			Auxiliary airstream exiting wet side of the energy exchanger			
		t_{1db}, ºF	t_{1wb}, ºF	d_1, gr	t_{2db} ºF	t_{2wb}, ºF	d_2, gr	t_{3db}, ºF	t_{3wb}, ºF	d_3, gr	V, fpm
1	40	95	63	34.3	45	38.5	34.8	75.2	63	74	551
2	42	86	62	44	50	45.8	44	68	62.7	77	590
3	46	104	65	45.5	50.4	46.4	45.5	75	66	100	590

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

1. Over all 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 Regenerative Indirect Evaporative Air Cooler, configured per Scheme Fig. 1b. Tables' A & 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 dew-point temperature 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 dew-point temperature. 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 psychrometric temperature difference (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 φ =1). In that case, the following equation could be written: I_3-I_{dp} > I_1-I_{dp}.

From the energy balance 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 psychrometric temperature difference 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:

$$(G_т)\ (I_1 - I_{dp}) = (G^{id}_в)\cdot(I_3 - I_{dp}) \qquad (2)$$

Where:
$G_т$ - mass flow rate of the total main airstream
$G^{id}_в$ - 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:

$$M^{id} = G^{id}_o / G^{id}_т$$

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

$$M^{id} = G^{id}_o / G^{id}_т = (I_3 - I_1) / (I_3 - I_{dp}) \qquad (3)$$

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:

$$Q^{id}_o = (G^{id}_o)\ (I_1 - I_{dp}) = (G_в)\ (I_3 - I_1)$$

Then

$$M^{id} = (G^{id}_в)\ (I_3 - I_1) / (G^{id}_в)\ (I_3 - I_{dp}) = (Q^{id}_o) / (Q_т) \qquad (4)$$

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 φ = 100% and φ = 0%.

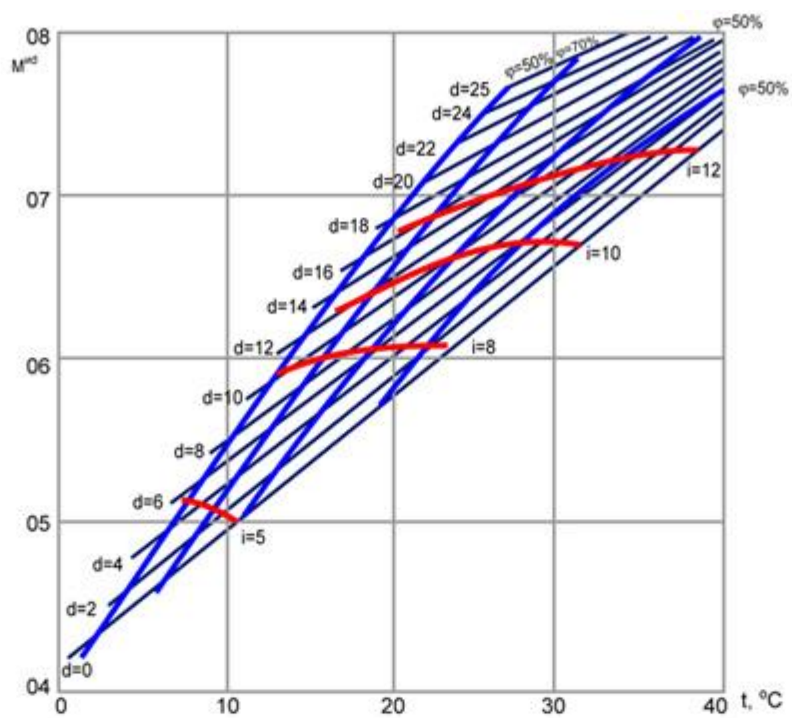

Fig. 2.

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 $G_в$, and that would result in the respective decrease of the available "useful" cold airstream - G_o. In other words, lowering 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

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 Dew Point temperature 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 dew-point temperature. 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 dew point temperature 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 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 regenerative indirect-evaporative air coolers in the regions with high–humidity ambient air, the units could be integrated with the inlet (ambient) air drying module. This combination allows 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 is greatly expands the area of applications of that method, allowing use of this cooling technology for the controllable comfort space and process cooling. For instance, it could be used for cooling of juicy vegetative raw materials and other products, which do not require freezing temperatures for storage.

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 summer time 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 Regenerative Indirect Evaporative Air Coolers equipped with the air pre-drying means.

Another important factor defining expediency of use of the Regenerative Indirect Evaporative Air Coolers (RIEAC), 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

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

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 and 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 + pump motor + controls + misc. power) by the operating RIEAC Unit:

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

Where:

$N_{fan\ mot}$ - Power draw by fan motor, kW
$N_{pump\ mot}$ – Power draw by pump 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

Fig.3 Chart below illustrates a correlation between the temperature of the cooled total main airstream exiting the energy exchanger and the inlet air's moisture content. During that test the airstream's entering dry bulb temperature was maintained constant at 40ºC at varying moisture content.

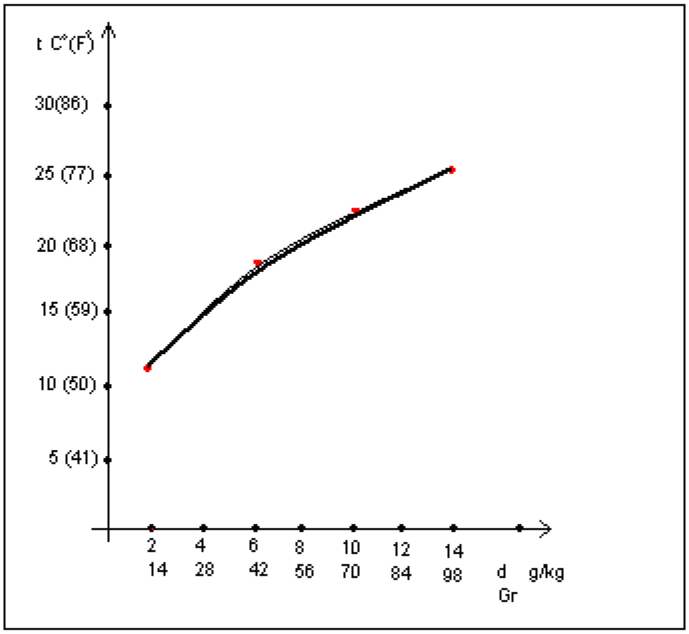

Fig.3

Let's remember, that the Energy Efficiency Factor (ξ) of a particular indirect evaporative cooler depends on its design and its operating mode. Table-3 below presents parameters of the summer design conditions of the ambient air versus the Energy Efficiency values of the indirect evaporative air cooling means, operating in the optimal mode for several selected countries and cities.

Analysis data from Table-3 shows that the indirect evaporative air cooling is applicable for the vast regions of the Earth with significant energy benefits because the indicated Energy Efficiency Ratios for all listed counties and cities significantly exceed the ones of the conventional refrigeration means, used in these areas for cooling and air conditioning.

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.

Table-3

Some Countries and Cities	Design Summer parameters of ambient air		Energy Efficiency Ratio, ξ	Notes
	Dry bulb temperature, ^{o}C	Total Moisture content, g/kg		
Ukraine				
Kiev	28.7	10.4	10.	
Odessa	28.6	13	8.	
Kharkov	29.4	10	10.5	
Dnepropetrovsk	31	9.8	11.1	
Lugansk	31.	10.	10.4	
Simferopol	31.8	12.0	9.	
Russia				
Volgograd	33	9.5	11.4	
Voronezh	28.9	9.8	11.1	
Krasnodar	30.6	12.8	8.9	
Samara	29.7	9.7	11.0	
Saratov	30.5	9.6	10.9	
SIS Countries				
Alma-Ata	30.2	9	11.8	Kazakhstan
Ashkhabad	39	9	11.8	Turkmenistan
Yerevan	34.6	11.1	9.8	Armenia
Karaganda	31	6.4	15.0	Kazakhstan
USA				
Salt Lake City	36,1	3,5	19.5	
Phoenix	42,8	7,5	13.5	
Los Angeles	33,9	10,2	10.7	
The Near/Middle East				
Kabul	35	8	13	
Jerusalemм	35.6	11	9.7	
Bagdad	45	7	14	
Saudi Arabia	43.8	5	17	

General results achieved in development of RIEACs':

Conducted theoretical and experimental studies of the Regenerative Indirect -Evaporative Air Cooling Method have resulted in development of high-performance Indirect Evaporative Air Coolers for a wide variety of applications.

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.

Some examples of RIEAC applications

Example #1:

RIEACs for cooling residential/commercial buildings

Photo 1 – RIEAC install on McDonalds (Yalta, Ukraine)

Table-4 below presents the basic performance characteristics of the RIEAC equipped with the single regenerative energy exchange module.

Table-4

Parameter	Value
Nominal flow rate of the "useful" main cooled supply air	1300 m^3/hr
Nominal flow rate of the auxiliary exhaust air	1150 m^3/hr
Nominal volume of total main (outdoor) airstream entering the Unit	2450 m^3/hr
Max power draw of the operating unit	0.3kW
Total cooling capacity of the unit at outdoor air temperature of t_{osa} = 32°C and moisture content of d_{osa} = 10 g/kg	5kW
Winter time heat recovery rate of the energy transfer module at outdoor & indoor air temperatures t_{osa} = -10°C & t_{in} = 18°C	3.6kW
Overall dimensions of the RIEAC Unit	520(W)x870(L)x800(H) mm
Dry weight of the RIEAC Unit	65kg
Operating weight of the RIEAC Unit	115kg

Notes:

- A number of the regenerative energy exchange modules within the RIEAC other than indicated capacities in Table 3, as well the RIEAC design configuration should be determined for specific application and design conditions.
- In cases, when the localized building's exhaust air is available and its wet bulb temperature is lower than the wet bulb temperature of the RIEAC', that air could be introduced into the wet channels of the energy exchanger instead of the regular auxiliary airstream. This would result in increased efficiency of the RIEAC.

Photo 2 - RIEAC install on McDonalds (Yalta, Ukraine)

Example #2:

The RIEAC applied for cabin cooling of the grain combine.

The cabin of a grain combine protects the operator from the influence of excessive dust, heat, and direct solar radiation. The cabin is usually equipped with the conventional refrigeration air conditioner. The high cost, high energy (fuel) consumption, and low -reliability (due to harsh operational conditions) of the refrigeration air conditioner dictates necessity of using some alternative cooling solutions, devoid of these deficiencies. In response to that demand we have developed a special Regenerative Indirect Evaporative Air Cooler for the grain combine's cabin to replace the original refrigeration air conditioner (see attached Picture). The main performance characteristics of the mentioned special Regenerative Indirect Evaporative Air Cooler for the grain combine are presented in Table-5.

Table-5

Parameter	Value
Nominal flow rate of the main cooled supply air	350 m^3/hr
Total cooling capacity at outdoor air of t_{osa} = 35°C and moisture content d_{osa} = 10 g/kg	1.6 кW
Max power draw	0.3 kW
Water usage	2.5, kg/hr
Over all dimensions, mm	1240x840x240mm
Total unit's operating weight	60kg

The new special IREAC was installed on a real operating grain combine and field tested during the harvesting campaign. During the test time period, the new special IREAC was able to maintain the cabin's inside comfort air conditions such as air temperature within 24-26^{0}C and respective relative humidity up to 60%. While the new special IREAC was turned off, the cabin's air temperature had easily exceeded 40^{0}C. That type of the IREAC could also be successfully used for the cooling of the cabins of variety of construction equipment, semi trucks, motor homes, trailer homes, etc. In this case, the fuel usage by the vehicle would be significantly lower in comparison to the conventional refrigeration air conditioners.

Photo 3 – RIEAC install on grain combine

Example #3:

The RIEAC for cooling of cabins of small - size cars/vehicles

The attached photo shows a streamlined profile RIEAC mounted on the roof of a small-size car. This unit was tested on the car for two cooling seasons, and the tests confirmed that the unit was able to provide comfortable environmental conditions inside the car's cabin during an entire summer season in one city (Odessa, Ukraine). During the test, the car was loaded usually with 4, and, sometimes, even with 5 passengers.

Some physical & technical data of the car-mounted RIEAC unit

Physical dimensions	31.5(L) x 31.5(W) x 10(H) inches
Total intake airflow rate	294 cfm
Useful (cooling) supply airflow rate	177 cfm
Auxiliary airflow rate	117 cfm

The local outdoor summer ambient design conditions

Ambient air design dry bulb temperature	28.7^0C (83.7^0F)
Ambient air design moisture content	10.4 g/kg (72.8 gr/lb)

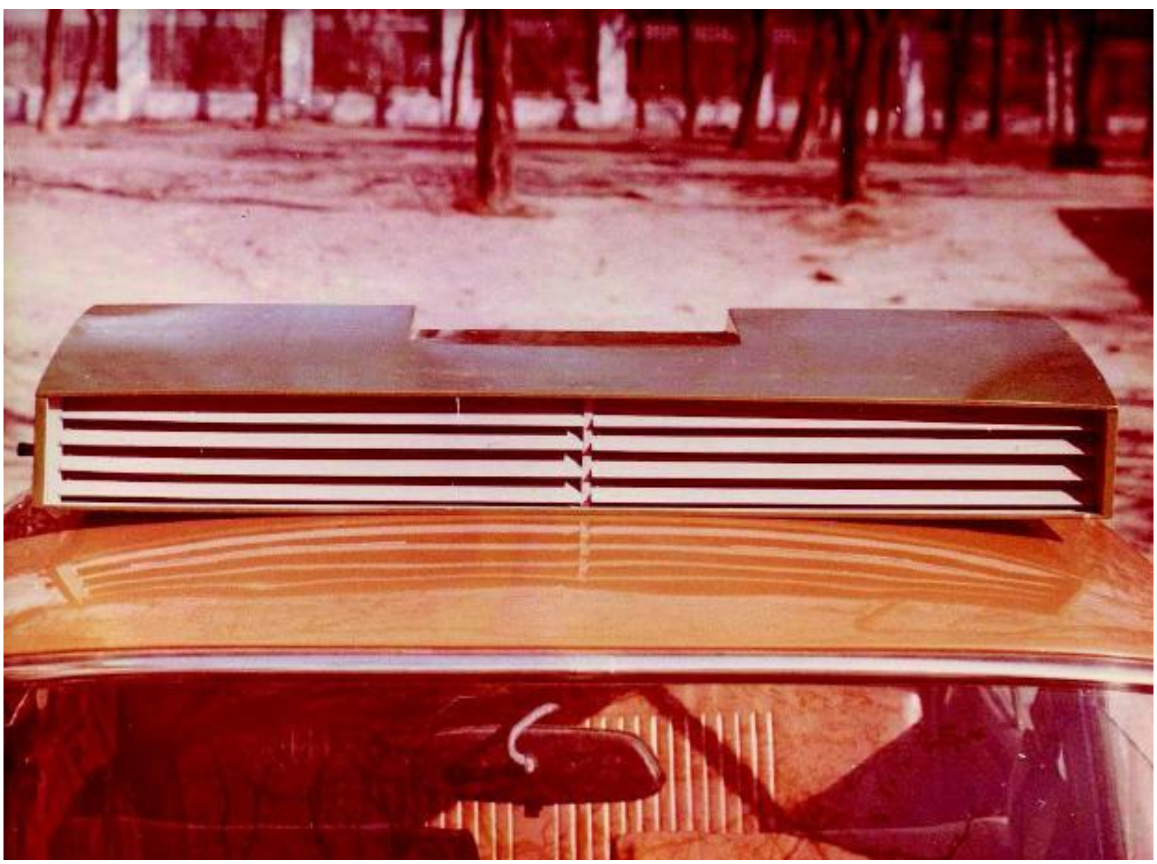

Photo 4 – RIEAC installed on car

The air conditions in the car's cabin (with operating RIEAC unit) during the tests

An average discharge temperature of “useful” (cooled) air 17-18^{0}C (66.2-64.4^{0}F)
The cabin’s air dry bulb temperature range 26-27^{0}C (78.8 – 80.6^{0}F)
The cabin’s air RH range……………………….55-60%

Conclusions:

1. Extensive R&D resulted in development of a new type of an advanced air cooler – the Regenerative Indirect Evaporative Air Cooler, which provides maximum evaporative air cooling effect at a minimum power and water inputs.
2. Extensive studies and 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).
3. The Thermo-physical Lab had patented, designed, developed, and fabricated experimental RIEACs of different capacities and configurations for different 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.
4. The technology of the RIEACs is mature and ready for Global use.

Notes:
1. Continued and extensive R&D has resulted in development of the newest approaches to the Indirect Evaporative Air Technology and Equipment (with and without the air drying means,) which significantly outperforms all known equipment of that kind on the market. Information on these developments is not presented in this article.
2. Currently the TT - Group (Ukraine) has established manufacturing and installations of various (customized) types of the RIEACs.

Some major advantages of the RIEACs are:

- Pollution-free operation.
- The RIEACs supply 100% of “fresh” cooled air into the servicing space, while the conventional refrigeration air conditioners usually re-circulate at least 90% of the indoor air. Therefore, it provides much healthier indoor environmental conditions (no indoor micro-flora, mildew, mold, odors, etc.).
- The RIEACs are highly-economical: at significantly lower power consumption. Their discharge air temperature is just slightly higher that the one from the conventional refrigeration air conditioners.
- The RIEACs’ simple single-fan modular design translates into relatively low manufacturing cost at mass production, low operating and maintenance costs, as well as low-labor requirements at units’ factory assembly and/or their field modification. The RIEACs could also be easily integrated into the existing air conditioning and ventilating systems, and significantly improve their overall performance efficiency.
- The RIEACs could be successfully used for either stationary or mobile applications for year-round operations, providing summer cooling/pre-cooling and/or winter pre-heating of the makeup air by means of recovering heat from the warmer building or process exhaust.

List of publications and Patents:

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