WASTE PRESSURE ENERGY RECOVERING HYBRID SYSTEM FOR EVAPORATIVE COOLING

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ABSTRACT

To help overcoming the biggest current challenge in adopting evaporative cooling on a large scale, this study presents a novel configuration for a hybrid equipment for evaporative cooling with Tube Vortex (IEC/ VT cooler). The main challenge for the evaporative cooling systems' good performance is related to its high level dependency on the ambient air conditions, which in damp or mild climates leads to very limited cooling capacity. The main obstacle that makes it difficult the proliferation of evaporative cooling systems is represented by the minimum temperature value that can be obtained during the process, whose value depends strongly on the characteristics of the ambient air. In order to overcome this difficulty it is presented a physical configuration of a new system equipped with Vortex Tube for indirect evaporative cooling to temperatures below the adiabatic saturation temperature and, if necessary, below the dew point temperature. Numerical values of the coefficient of performance of the new evaporative cooler equipped with Vortex Tube are twice the size of COPs for equipment with vapor compression cycle, revealing a large potential for energy savings. By considering a low value for the compression ratio, and thus enabling the use of residual energy from pressurized gaseous effluents whose cost is almost zero, this study states the effective technical and economic feasibility of the evaporative cooler equipped with Vortex Tube for industrial applications.

INTRODUCTION

Indirect evaporative cooling is currently considered the best available technology to reduce the energy consumption for air conditioning in buildings. With potential growth market valued at 20% for the next two decades [1], the evaporative cooling technology is certainly an interesting bet for significant savings in energy consumption and consequent reduction of carbon dioxide emissions. The biggest current challenges in adopting evaporative cooling on a large scale are related to its high level of dependency on ambient air conditions, which in damp or mild climates leads to very limited cooling capacity. This study introduces a new equipment for overcoming this problem, whose operational mode of indirect evaporative cooling employs Vortex Tubes to reach the necessary low temperature.

Of particular interest in this work is the cooling in the range 5^0C to 20^0C, specific for maintaining thermal comfort in the built environment, and the cooling in the range 0^0C to 5^0C for electronic components conditioning in computing centers (servers' rooms) or indoor areas with a large number of functioning computers.

Aiming to contribute for the efficient use of all available energy, this paper introduces a new thermodynamic cycle based on the Vortex Tube technology for improving the energy efficiency of indirect evaporative cooling. The Vortex Tube is an easy to operate device, without moving parts, producing heating and cooling by the expansion of a gas [2-6]. The need for less expensive and cleaner technologies, with lower losses and long service life (L^4 criteria - low pollution, low cost, low loss, and long life) stimulated in recent decades a growing interest for technologies such as the Vortex Tube [7-13].

Losses of pressurized gaseous effluents, such as steam or blow off compressed air, are very common for many industrial applications. The energy associated with the high pressure of

NOMENCLATURE

c	[J/kgK]	specific heat
COP	[-]	Coefficient of performance
h	[J/kg]	Specific enthalpy
k	[-]	Adiabatic exponent
$\dot{m}$	[kg/m^3]	Mass flow rate
P	[N/m^2]	Pressure
$\dot{Q}$	[W]	Heat transfer interaction
R	[J/kg K]	Constant of gas when considered perfect gas
s	[J/kg K]	Specific entropy
T	[K]	Temperature
X	[-]	Temperature ratio

Special characters

ϕ	[-]	Relative humidity
μ_C	[-]	Ratio between the cold gas and the total mass flow rate
Π	[-]	Degree of compression
η_{II}	[-]	Relative efficiency or utilization factor
ω	[-]	Specific humidity

Subscripts

a	Air
C	Cold
H	Hot
P	Constant pressure
0	Ambient or reference

these emissions that, although it could still be reused for some useful purpose, is wasted to the environment, is called the "waste pressure energy" similarly to the "waste thermal energy" representing the energy associated with the high temperature of effluents dumped into the atmosphere. For a large number of researchers the "waste pressure energy" recently became an interesting “reclaimable form of energy” [14-16]. To emphasize the economic attractiveness of the newly proposed IEC/ VT cooler, the residual energy recovery from pressurized gaseous effluents (waste pressure energy), which is practically costless, is also considered here.

FUNDAMENTALS OF THE IEC/ VT COOLING

When employed as complementary cooling equipment to reach the necessary low temperature by the newly described IEC/ Vortex Tube cooling system, the closed loop sequence of unitary processes into the VT occurring inside the control volumes shown in Fig. 1 is as follows:

- the gas expansion into the VT from $P_1 = \Pi P_0 > P_0$ to $P_2 = P_3 = P_0$, unleashing its separation into a cold gas stream at $T_2 = T_C < T_0$ and a hot one at $T_3 = T_H > T_0$,
- the removal of heat for cooling down atmospheric air below the adiabatic saturation temperature into the cold side heat exchanger (CSHE),
- the rejection to the surroundings of the heat flux $\dot{Q}_{H,2}$ that occurs into the hot side heat exchanger (HSHE) while lowering to T_0, before the admission into the compressor, the temperature of the hot gas stream provided by the VT ($T_3 = T_H > T_0$),
- the compression up to $P_1 = \Pi P_0 > P_0$ of the two gas streams, both at the same temperature $T_4 = T_5 = T_0$ and pressure $P_4 = P_5 = P_0$, leaving the CSHE and the HSHE (when available pressurized gaseous effluents, there is no need of such compressor),
- the rejection to the surroundings, by the auxiliary cooler, of the excess energy of the hot compressed gas or pressurized gaseous effluents ($\dot{Q}_{H,3}$) in order to diminish at $T_1 = T_0$ their temperature before the admission into the VT.

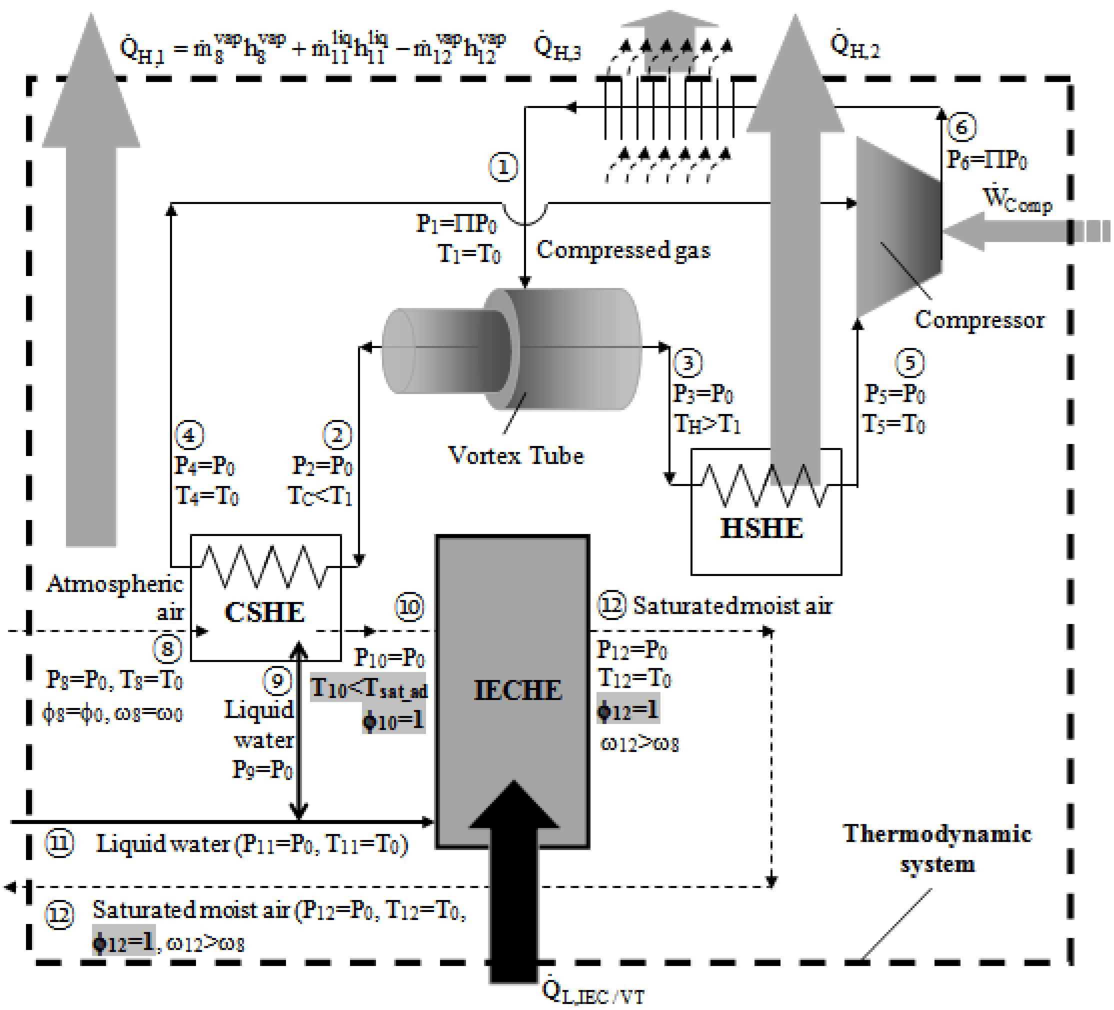

Figure 1 Physical configuration of the novel IEC/ VT cooling system functioning with minimal temperatures below the adiabatic saturation temperature of atmospheric air (the grey shadowed values indicate the fixed functional parameters).

The novel IEC/ VT cooling system diminishes the minimum temperature $T_{10'}$ into the range $T_{DewP} < T_{10'} < T_{sat\_ad}$ through a combination of simultaneous processes $11 \rightarrow 10'_w \leftarrow 8_w$ (Fig. 2a) of direct evaporative cooling and heat removal ($\dot{Q}_{L,VT}$) from the atmospheric air flowing through the CSHE. In this case the reduction of the initial temperature of the atmospheric air ($T_8 = T_0 \rightarrow T_{10'} < T_{ad\_sat}$) is carried out simultaneously by heat removal and by using a part of the internal energy of the moist air to vaporize the liquid water injected into the CSHE at T_0 and P_0 (state 9). As shown in Figure 3, the process $8 \rightarrow 10'$ is accompanied by an increasing of the water vapor content in the air $\omega_{10'} > \omega_8$. Meanwhile, the heat removal from the air crossing the CSHE is due to its thermal interaction with the cold gas supplied by the Vortex Tube at $T_C < T_{DewP}$.

As shown in Fig. 2b, when necessary to reach minimal temperatures $T_{10''}$ below the dew point temperature, the newly introduced IEC/ VT system removes heat ($\dot{Q}_{L,VT}$) from the atmospheric air flowing through the CSHE until $T_C < T_{10''} < T_{DewP}$. During such operation the water vapor content of the atmospheric air partially condenses according to the processes sequence $8_w \rightarrow DewP \rightarrow 10''_w \rightarrow 9_w$ in Figure 2b. Since the temperature of the cold gas supplied by the Vortex Tube is lower than the dew point temperature ($T_C < T_{10''} < T_{DewP}$), the mass flow control of the cold gas delivered to the CHSE would allow to reduce the temperature of the saturated moist air up to T_C. The cooling process in this case is accompanied by a reduction of the specific humidity $\omega_{10''} < \omega_8$ (Fig. 3).

After this first stage, when the working fluid (moist air) reaches the necessary minimum temperature ($T_{10'}$ or $T_{10''}$), occurs the removal of heat ($\dot{Q}_{L,IEC/VT}$) from the external body to be cooled down. This second stage develops by thermal interaction during the flow of the cold saturated moist air throughout the IECHE control volume. This is a heat exchanger that uses thermal energy simultaneously for temperature increase and for liquid water vaporization, thus changing the thermodynamic states of the working fluid from state 10 (saturated moist air at minimum temperature) to state 12 (saturated moist air at T_0). The psychrometric chart in Figure 3 explains the physical mechanism of heat removal by indirect evaporative cooling (processes $10' \rightarrow 12$ or $10'' \rightarrow 12$) from the external body at T_0 ($\dot{Q}_{L,IEC/VT}$). This mechanism is associated to the temperature differences $\Delta T'$ or $\Delta T''$ between T_0 and the minimal temperatures $T_{10'}$ or $T_{10''}$ at the CSHE's exit. The two processes $10' \rightarrow 12$ and $10'' \rightarrow 12$ are assumed constant pressure heating processes of the air - water vapor mixture along the saturation curve. As can be seen in Fig. 3, variation of the specific humidity is higher during the heating process $10'' \rightarrow 12$ ($\Delta\omega'' > \Delta\omega'$) suggesting that the lower the temperature at the CSHE's outlet greater amount of thermal energy is removed per kilogram of working fluid into the IECHE heat exchanger.

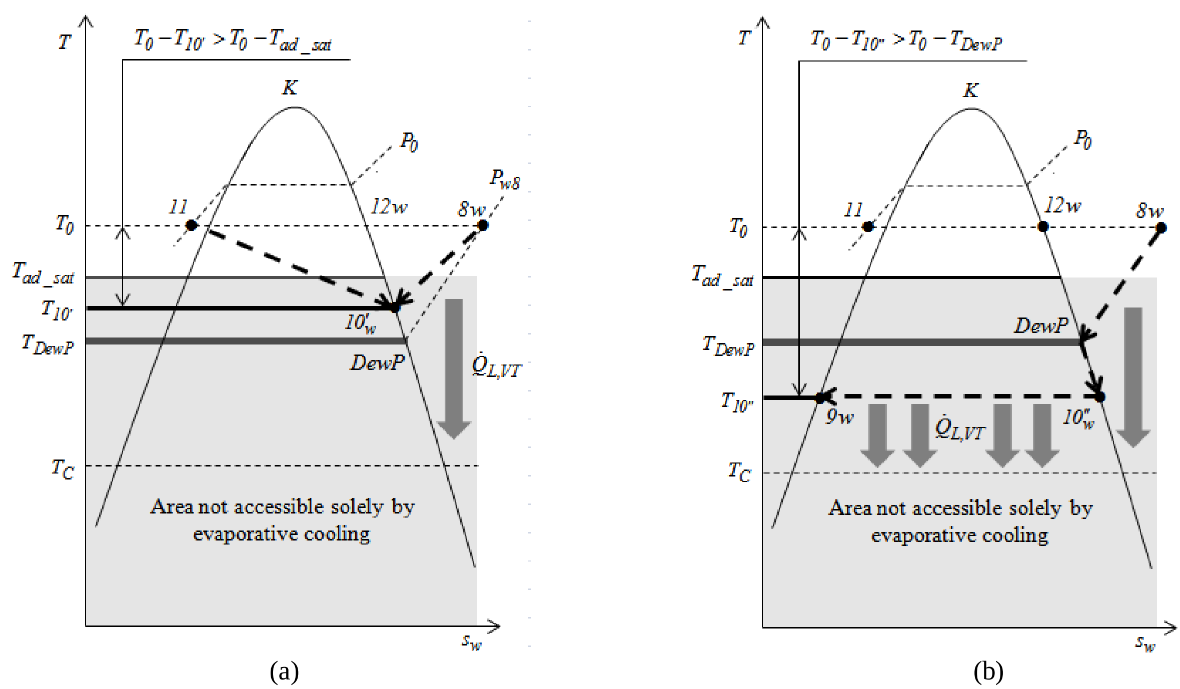

Figure 2 $T - s_w$ diagram of the water showing thermodynamic processes occurring in the IEC/ VT cooler in order to achieve minimal temperatures in the range $T_{ad\_sat} > T_{10'} > T_{DewP}$ (a) or below the dew point temperature $T_{DewP} \geq T_{10''} > T_C$ (b).

MATHEMATICAL MODELING

To study the new IEC/ VT system's functioning and for evaluating its performance, a mathematical model was developed based on the mass and energy conservation laws and on the second law of thermodynamics applied to control volumes in Figure 1.

The problem consisted in determining the main operational characteristics of the new IEC/ VT system such as the coefficient of performance $COP = \dot{Q}_{L,IEC/VT} / (-\dot{W}_{Comp})$, the minimum process temperature, the specific cooling load $q_{L,IEC/VT} = \dot{Q}_{L,IEC/VT} / \dot{m}_{a,8}$, specific consumption of water and compressed gas in the Vortex Tube, $\Delta\omega$ e $\tilde{m}_{g,1} = \dot{m}_{g,1} / \dot{m}_{a,8}$, while considering various atmospheric conditions (P_0, T_0, ϕ_0).

Monitoring of the entropy generation was performed, based on the second law of thermodynamics applied to the control volumes in Figure 1 associated to the Vortex Tube and the two heat exchangers CSHE and IECHE, for checking the realism of the results obtained. The following assumptions were also considered when writing down the algebraic equations system for mathematically modeling the IEC/ VT system's functioning:

- steady-state functioning;
- compressed dry air, assumed to be a perfect gas, is supplied at room temperature in a closed loop to the Vortex Tube;
- equipment assembly formed by the compressor and the auxiliary cooler is modelled as a reversible isothermal compressor to make possible using already published experimental data [14] on the Vortex Tube performance, whose relative efficiency η_{II} or utilization factor [17]:

$$\eta_{II} = 1 - \frac{T_0 \dot{S}_{ger}}{\dot{m}_1 R_G T_0 \ln(\Pi)} \tag{1}$$

is calculated taking into account the minimum power consumption given by

$$\left|\dot{W}_{Comp}\right| = \dot{m}_1 R_G T_0 \ln(\Pi) \tag{2}$$

that is only theoretically attainable based on the reversible isothermal compression. In the above equations $\dot{W}_{lost} = T_0 \dot{S}_{ger}$ represents the Vortex Tube rate of exergy destruction, $\dot{S}_{ger} > 0$ is the entropy generation during the Vortex Tube operation, and R_G is the constant of the compressed gas;

- all mass and energy transfer interactions with the environment occur at T_0 ;
- all flows are assumed constant pressure flows at $P_0 = 10^5\ N \cdot m^{-2}$;
- the moist air leaving the CSHE control volumes is at saturated state;

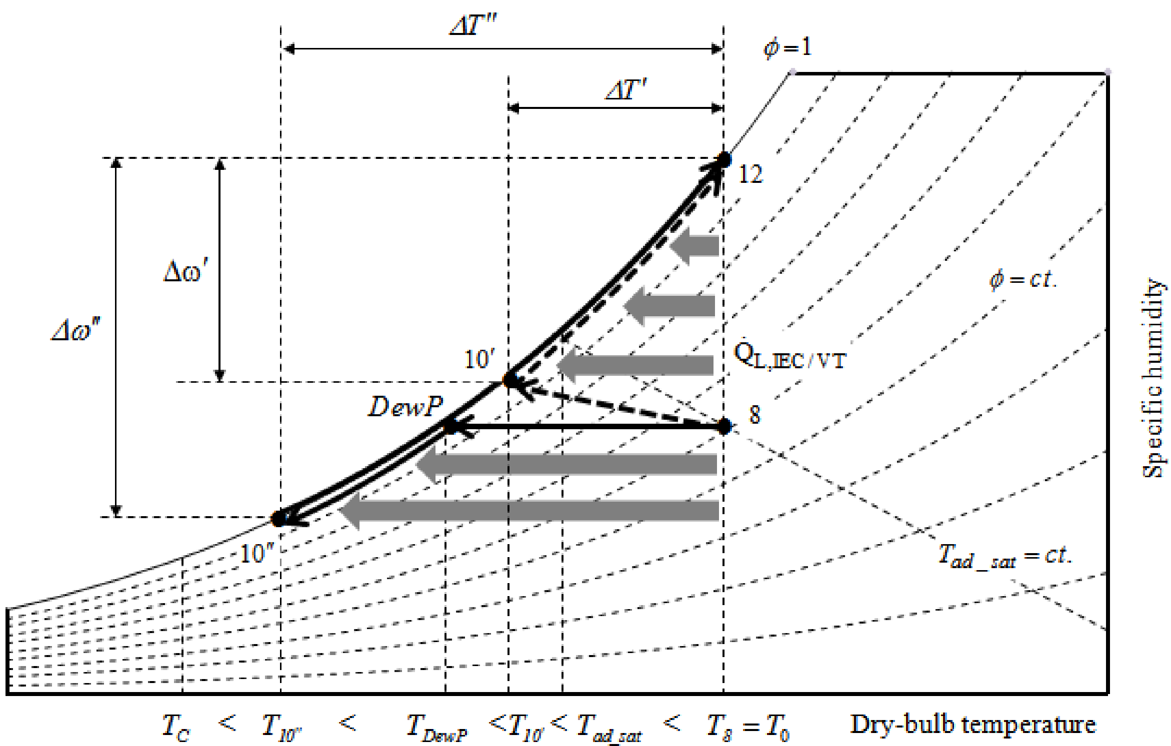

Figure 3 Psychrometric chart showing the thermodynamic processes during the indirect evaporative cooling in the IEC/ VT system running with minimum temperature between the adiabatic saturation temperature and dew point temperature (thick dotted line) and below the dew point temperature (thick solid line).

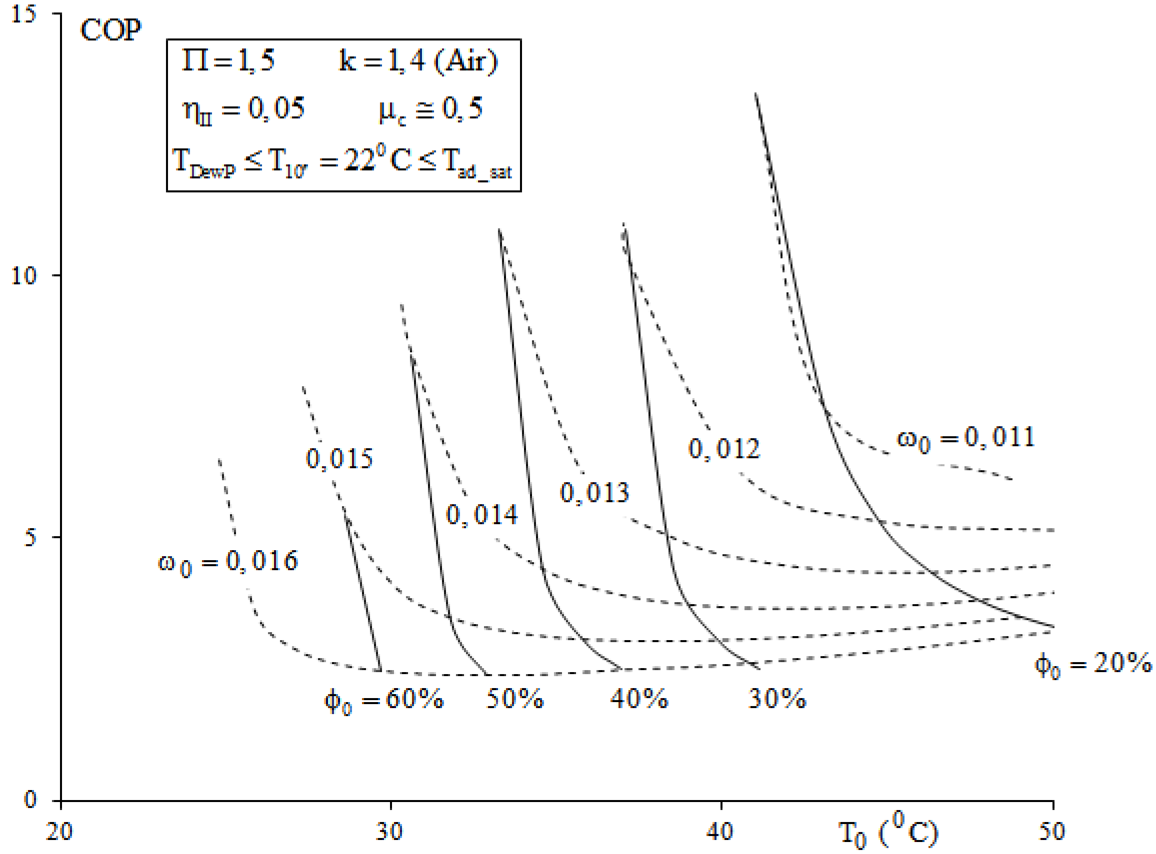

Figure 4 IEC/ VT cooling system performance for minimal temperatures $T_{10'}$ in the range $T_{DewP} < T_{10'} < T_{ad\_sat}$.

- the moist air leaving the IECHE control volumes is at saturated state;
- the saturated moist air leaves the IECHE at temperature T_0 ;
- heat losses are negligible into the CSHE;
- there is no kinetic or potential energy variations;
- Dalton model is assumed for calculating thermodynamic properties;
- instantaneous thermodynamic equilibrium between the dry air and the vaporized moisture;
- phase change processes occur depending on the water vapor partial pressure in the mixture.

NUMERICAL METHOD

The numerical solution of the algebraic system was developed considering the dimensionless mass flow rates according to $\tilde{m} = \dot{m} / \dot{m}_a$. Variations of the specific enthalpy and entropy of the dry air are calculated based on the perfect gas equations: $dh = c_p dT$ and $ds = c_p(dT/T) - R(dP/P)$, where $c_{p,a} = 1003.5\ J/(kg \cdot K)$ and $R_a = 287\ J/(kgK)$ represent the specific heat at constant pressure and the air constant. Specific enthalpy and entropy of water are determined assuming the characteristic values on the saturation lines for liquid and vapor [17].

The control variable used to define the Vortex Tube operating regime is represented by the ratio between the mass flow of gas leaving the VT cold side and the mass flow rate at the entrance of the Vortex Tube ($\mu_C = \dot{m}_{g,2} / \dot{m}_{g,1}$). The algebraic system was numerically solved for the followings unknowns: $X = T_C / T_H$, $T_2 = T_C$, $T_3 = T_H$, $\tilde{m}_{g,1}$, $\tilde{m}_{w,9}^{CSHE}$, $\tilde{m}_{w,11}^{IECHE}$, ω_{10}, ω_{12}, $q_{L,IEC/VT}$, s_{ger_VT}, s_{ger_CSHE}, s_{ger_IECHE} and COP. Typical numerical values were assumed for the compression ratio $\Pi = P_1 / P_2 = P_1 / P_3$, the adiabatic exponent k of the gas supplied to the Vortex Tube, the utilization factor η_{II}, the ratio between the flow of cold gas and the total flow of gas supplied to the Vortex Tube μ_C, and the minimum necessary temperature required for the cooling process of the external body. The residuals' higher absolute value of all equations to consider acceptable the numerical solution was 10^{-24}.

RESULTS AND DISCUSSION

By considering the thermodynamic states in Figure 1, it has been checked the feasibility of the new IEC/ VT system for air cooling at minimum temperature $T_{10'} = 22^0 C$ while the atmospheric air condition is given by $T_0 = 40^0 C$, $P_0 = 10^5\ N \cdot m^{-2}$ and $\phi_0 = 30\%$.

A limited value of the compression ratio ($\Pi = 1.5$) is considered in this study to check the feasibility of wasted pressure energy possible recovery from pressurized gaseous effluents.

The numerical results in Figure 4 show quite high numerical values for the IEC/ VT cooler's performance over a wide range of initial values. The highest values ($COP \cong 10$) are determined for the coefficient of performance when considering the functioning of the new evaporative cooler equipped with Vortex tube at minimal temperature levels that are characteristic for air conditioning processes, confirming the potential for substantial gains of energy saving in maintaining thermal comfort in built environments. Lower COP values are calculated for the IEC/ VT cooler functioning with minimal temperatures below the dew point temperature. This is a situation that is commonly classified as unfit for use of evaporative cooling. Despite the lower values of the COP in this type of application, taking into account the possibility of residual energy reuse of pressurized gaseous effluents, whose cost is almost zero, the economic advantages are obvious.

CONCLUSION

With a potential market growth valued at 20% for the next two decades, the evaporative cooling technology is certainly an interesting bet for significant savings in energy consumption and consequent reduction of carbon dioxide emissions.

To contribute to the efficient use of all the available energy, in this work is being considered the possibility of using the Vortex Tube technology for recovering the residual energy from pressurized gaseous effluents.

The study presents a new configuration of the hybrid equipment for evaporative cooling with Tube Vortex, and the analysis of its functional and performance characteristics.

Numerical values of the coefficient of performance were determined in the range $0,5 \le \mathrm{COP} \le 15$. The highest values correspond to operating regimes with minimal process temperatures between the adiabatic saturation temperature and the dew point temperature. At this temperature level are being carried out the air conditioning processes. Thus the COPs of the new evaporative cooler equipped with Vortex Tube, which is more than twice the size of COPs for equipment with vapor compression cycle, reveals a large potential for energy savings.

By considering a low value for the compression ratio, and thus enabling the use of residual energy from pressurized gaseous effluents whose cost is almost zero, this study states the effective technical and economic feasibility of the evaporative cooler equipped with Vortex Tube for industrial applications.

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