Thermodynamic Analysis of Adiabatic Mixing of Water Injected Into Moist Air

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The paper presents thermodynamic aspects of sprayed water humidification and cooling processes having as input data the atmospheric air temperature, important processes in design of air conditioning systems. The mathematical model is presented, as well as the obtained results in a numerical and graphical manner. The originality of the paper consists in presenting some nomographs that allow the student or designer to decide upon the humidification parameters (water temperature, respectively mass flow rate) depending on the required thermodynamic state at the humidifier exit, without other calculus. The nomographs are developed so that the user could find the searched values depending on different input data.

Keywords: humidification, humid air, nomographs; thermodynamic analysis

In human comfort and also for industrial processes, air conditioning is one of the most important issues to provide the needed thermodynamic parameters for developing activities. Thus, in technical literature, one can find many research papers in this domain, from simple calculations of air conditioning systems [1, 2] to the most developed systems, or solar air conditioning systems [3, 4]. The interest in offering easy-use tools for predicting thermodynamic properties is noticed in researches published on this subject, providing techniques for multicomponent systems, such as the ones using artificial intelligence procedures [5].

In [1] a simple supermarket energy model that takes into account different irreversibilities is presented. The analysis carried out with this model employed the energy requirement as a function of the external environmental temperature.

A simulation of the air-conditioning processes is discussed in [2], where interactive computer routines have been developed for estimating the psychrometric properties and predicting the behaviour of air undergoing constant-pressure mixing, heating, cooling, humidification and dehumidification processes. Other detailed procedures for calculating psychrometric properties are given in [6].

Thermodynamic processes of humid air are also related to other applications, not only air conditioning, as it was mentioned before. For example, humidification tower for humid air gas turbines [7, 8] is widely spread as research application; in the study and optimization of HAT (humid air turbine) cycle, the humidification of compressed air is the essential discussed issue. Also, the humidification - dehumidification process (HD) is an interesting technique adapted for water desalination when the demand is decentralized [9, 10]. In [9] the HD technique is presented for solar and geothermal desalination units, for a plant functioning by mechanical compression of humid air and for an installation functioning by absorbing water from the humid atmosphere. The paper [10] evaluates the characteristics for several layouts for the humidification dehumidification desalination process. As the author stated, the air humidification tower is the common feature among these processes, where the humidity of the ambient air is increased to saturation at the desired design temperature. Also, air humidity is an important parameter in gas drying processes by adsorption on composite materials, as seen in [11].

There are also published studies on models for performance predictions and investigation of the operational behaviour of simple solar cooling systems for small residential applications [3], or models on water-cooled air-conditioning systems which are stated to be, in general, more energy efficient than air-cooled air-conditioning systems [4]. These are only few examples where humidification process is studied and applied in a calculation model. Even if this topic seems a simple and maybe not interesting, the publications show that the subject is seriously treated in the domain. Lately, many CFD simulations have been proposed on humid air processes, as for example the drying systems optimization based on dynamic mathematical models, analysis and numerical simulations [12] where the authors present the system behavior directly influenced by the air velocity and temperature and moisture content of the air, variable in time.

The effects of air humidification on human health were also studied in some papers. In [13] the authors studied the symptoms of workers in a modern eight-floor office building in Helsinki, Finland, by comparing two zones: a humidified part of the building and a similar nonhumidified one. The conclusion of that paper was that the workers in the humidified part reported significantly less dryness of skin, throat and nose, as well as sensation of air dryness compared to those in the nonhumidified part.

Also low moisture content in cabins is known to be responsible for headache, tiredness and many other non-specific symptoms.

Concluding this introduction to the subject, the present paper has as principal aim the study of thermodynamic aspects regarding sprayed water humidification and cooling processes considering the atmospheric air temperature as input data. The mathematical model will be presented as detailed as possible, in an easy-to-understand manner. The obtained results are presented both in a numerical and graphical manner.

The originality of this paper consists in presenting nomographs that allow the student or designer to decide...
upon the humidification parameters (water temperature, respectively mass flow rate) depending on the required thermodynamic state at the humidifier exit, without other calculus, but just by reading the data. The nomographs are developed so that the user could find the searched values depending on different input data, and also on the studied situation. With respect to other papers, we present some studies for which one can choose the limit in terms of relative humidity of the outlet state that is to be reached in such a sprayed water humidification process (usually, this limit is accepted to be about 95%). We present in this paper results that could be used to decide upon the required humidification parameters (water temperature and/or spraying coefficient) in order to achieve a desired (and not necessarily 95% relative humidity) outlet state.

**General topic presentation**

The proposed subject is part of a complex problem, involving several thermodynamic processes in the analysis and design of an air conditioning system.

The treatment of moist air may involve thermodynamic processes, such as: preheating the exterior fresh air, mixing of fresh air with interior recirculated air, humidification, heating or cooling of the moist air. In figure 1 there is sketched out the complete scheme of an AC (air-conditioning) unit; it is not necessarily for the components to be used all at once. Their operation depends on the season, humid air state and desired parameters at the unit outlet, etc.

The operation scheme is chosen depending on some particular conditions (exterior and interior humid air parameters, characteristics of the conditioned space, heating/cooling source, costs, automation and control, etc). Also, depending on interior the harmful substances concentration, one may choose a scheme with or without (partial) recirculation of interior humid air [14-16].

**Winter season**

Let us focus our attention on a frequent situation. In the winter time and for a scheme with partial recirculation, the exterior humid air (denoted by state E) is passing through the filter AF and is preheated before mixing with the interior recirculated air (denoted by state I). This preheating process (E-P) is necessary in order to avoid obtaining the mixture state (M) below the saturation curve (100% relative humidity). In figure 2 there are qualitatively represented all these states (I, E, P, M, C) on a Mollier (H-x) diagram.

The two parameters needed for representing the state E on a Mollier diagram are the exterior air temperature and its relative humidity. In design procedures, these data are usually taken from meteorological data standards. For South-East Europe and winter season, for example, these parameters are about -5°C and 87% relative humidity [17-18]. As it was stated above, it follows the preheating (constant moisture content) process E-P, up to a temperature of about 2÷5°C and then the mixing process P-I-M with interior recirculated air having about 22°C and 65% relative humidity for human comfort conditions [16]. A process parameter is introduced, namely the fraction of recirculated air with respect to fresh air:

\[
f = \frac{m_r}{m_i}
\]

This fraction depends on the harmful substances concentration in the interior air and its minimum level is imposed by air quality standards. In these circumstances, the state M is defined by its enthalpy and moisture content (computed for an adiabatic mixture):

\[
H_M = H_p + f \cdot H_i; x_M = \frac{x_p + f \cdot x_i}{f + 1}
\]

Once the heat and moisture load are calculated for the studied space to be conditioned, the process direction I-C is determined:

\[
e = \frac{\dot{Q}_c}{m_{\text{air}}}
\]

and respectively, the state C of conditioned air on this direction and at an imposed temperature difference with respect to state I.
It remains to establish the processes undergone by the air from state M to state C. Let us denote by $D'$ the dew point state associated to state C, i.e. $x_{D'} = x_{C}$ and 100% relative humidity. If $x_{M} < x_{D'}$ and $H_{M} > H_{D'}$, then two processes are required for the air to pass from state M to state C, namely:

- an adiabatic humidification (mixing process of water injected into moist air) – process M-D (usually up to 95÷97% relative humidity); 
- heating (constant moisture content) process, from D to C.

If the above inequalities are not fulfilled, another preheating is required or other operating configurations are chosen.

**Summer season**

In the summer time, the values of air parameters at the inlet of the humidifying chamber (state M) are different since the exterior air is characterized by higher temperature and lower relative humidity. They could reach the values of about 38°C and 45% relative humidity. In figure 3 there are qualitatively represented all these states (I, E, M, C) on a Mollier (H-x) diagram for summer season. The processes undergone by the air in order to pass from state M to state C could be a cooling process (M-D) followed by a heating one (D-C). In some cases, the cooling process could be achieved by a humidification process.

Any other situations could be met, depending on the application and given input state (E).

In what it follows, we will focus on the adiabatic humidification process.

**Mathematical model for adiabatic mixing process of water injected into moist air**

Let us consider the common case of a mixing process of water injected into moist air. As hypothesis, the available water has a lower temperature than the moist air and higher than its corresponding dew point temperature. The schematic representation of this process is shown in figure 4.

The mathematical model relies on the mass and energy conservation laws.

The mass conservation law applied for moist air could be written as

$$\dot{m}_{x,M} = \dot{m}_{x,D} = \dot{m}_{x}$$

which leads to:

$$\dot{m}_{x} = \dot{m}_{x}(x_{D} - x_{M})$$

$x$ representing the moisture content (ratio between water vapor and dry air), and $\dot{m}_{x}$ is the mass flow rate of water injected into moist air.

The ratio between the mass flow rate of water injected into moist air and the mass flow rate of dry air represents the spraying coefficient:

$$\mu = \frac{\dot{m}_{w}}{\dot{m}_{x}} = x_{D} - x_{M} = \Delta x$$

The energy conservation law applied to this control volume, considering negligible kinetic and potential energy variation, no work and heat exchange (adiabatic insulation of the humidifying chamber), leads to:

$$\dot{m}_{x} H_{x} + \dot{m}_{w} h_{w} = \dot{m}_{x} h_{D}$$

It is to be noted that the enthalpy of moist air is expressed for unity mass of dry air, thus denoted by $H$ [J/kg of dry air], and computed for the corresponding state parameters (temperature $t$ and relative humidity $\phi$), while the specific enthalpy of water is expressed for unity mass of water, thus denoted by $h$ [J/kg] and extracted from water property tables at its temperature $t_{w}$ and spraying pressure $p_{w}$.

From eq. (7) it is obvious that:

$$\dot{m}_{w}(H_{D} - H_{M}) = \dot{m}_{w} h_{w}$$

Taking a numerical example, one can notice that the right-side term of eq. (8) could be neglected in comparison to the left-side term (for the studied case), wherefrom the name of “adiabatic” mixing of water injected into moist air. The process undergoes almost a constant enthalpy process. The available computing tools [19] allow us to consider also the right-side term in our numerical simulation, even if its value is considerably lower than the left-side term.

For avoiding cumbersome calculations and in the absence of numerical simulation tools, one could use the following rough analytical developments with quite an insignificant error. Starting from the energy balance equation (8), one could express the enthalpy of the humid air as the sum of its two components, namely dry air enthalpy and water vapor enthalpy:
In the ordinary range of temperature and pressure:

\[ m_{\text{s,}M} h_{\text{s,}M} + m_{\text{g,}M} h_{\text{g,}M} + m_{\text{w}} h_{\text{w}} = m_{\text{s,}D} h_{\text{s,}D} + m_{\text{g,}D} h_{\text{g,}D} + m_{\text{w}} h_{\text{w}} \]  

(9)

Taking into account eqs. (4) and (5) and dividing eq. (9) by \( m_w \), one could obtain:

\[ h_{\text{s,}M} + x_{\text{w}} h_{\text{g,}M} + (x_{\text{w}} - x_{\text{g,}M}) h_{\text{w}} = h_{\text{s,}D} + x_{\text{w}} h_{\text{g,}D} + x_{\text{w}} h_{\text{w}} \]  

(10)

For perfect gases (hypothesis that is valid for humid air in the ordinary range of temperature and pressure):

\[ dh = c_p dt \]  

(11)

where from:

\[ h - h_0 = c_p (t - t_0) \]  

(12)

At \( t_0 = 0^\circ\text{C} \), the enthalpy \( h_0 = 0 \) kJ/kg, so it remains that \( h = c_p t \), where the constant pressure specific heat \( c_p \) for dry air is about 1 kJ/(kgK).

The enthalpy of water vapor in the humid air \( h \) should be computed at the water vapor partial pressure and humid air temperature. For the sake of simplicity, this enthalpy could be approximated by the enthalpy of saturated vapours at the same temperature. This is shown in figure 5 on a T-s diagram for water. Numerically, the values are as follows:

- at 20°C and 60% relative humidity, the partial pressure of water vapors is 0.014 bar, while the saturation pressure is 0.02339 bar. At 20°C and 0.014 bar, the enthalpy of water vapour is 2538 kJ/kg, while the enthalpy of saturated vapors at the same temperature is 2537 kJ/kg (values computed by EES software). So the relative error of this approximation is less than 0.04%, more than acceptable. The same judgment could be applied for liquid water, so that the enthalpy of liquid water at its temperature could be approximated by the enthalpy of saturated liquid at the same temperature.

Under these circumstances, eq. (10) becomes:

\[ t_{\text{w,}M} + x_{\text{w}} t_{\text{g,}M} (t_{\text{w,}M}) + (x_{\text{w}} - x_{\text{g,}M}) t_{\text{w}} (t_{\text{w}}) = t_{\text{w,}D} + x_{\text{w}} t_{\text{g,}D} (t_{\text{w}}) \]  

(13)

in which the temperature is expressed in °C, the moisture content in kg of vapour/kg of dry air and the enthalpies in kJ/kg.

In the absence of a simulation tool, one can very easily compute the properties of state D only by knowing the temperatures and saturated states properties.

Numerical simulation setup

The parameters of state M are specified in terms of temperature \( t_{\text{w}} \), relative humidity \( \varphi_{\text{w,M}} \), atmospheric pressure \( p \) and mass flow rate of dry air \( m_w \), depending on the application (summer or winter season, or even other industrial application for which the parameters could have different values).

As an example of numerical simulation, we will firstly consider the values: \( t_{\text{w}} = 38^\circ\text{C} \), \( \varphi_{\text{w,M}} = 40\% \), \( p = 1 \) bar, \( m_w = 1 \) kg/s. The available water temperature is known and set to the value of \( t_{\text{w}} = 20^\circ\text{C} \).

The considered constrain is that the air temperature at the exit of the humidifying chamber is the same as the water temperature, namely \( t_{\text{D}} = t_{\text{w}} \). In the sensitivity studies presented below, some other cases are figured out and also other numerical values are considered too, for generalizing the results.

The unknowns of the problem are the relative humidity of moist air at the exit of humidifying chamber, \( \varphi_{\text{w,D}} \), and the mass flow rate of water to be injected, \( m_w \). As we have already mentioned in the Introduction, the limit in terms of relative humidity of the outlet state that is reached in such a sprayed water humidification process is no more imposed to be about 95%, but we show by these studies that it could have other values, too, function on some decisional parameters.

The objective is to find an easier way for establishing the temperature and mass flow rate of water that is to be injected into a quantity of moist air in order to obtain desired parameters at the exit state D \( (t_{\text{D}}, \varphi_{\text{w,D}}, x_{\text{w,D}}) \). We are going to answer this challenge by generating easy-use and user-friendly nomographs and numerical tables.

The procedure is as follows:

- the first step is to generate a set of results (constant values) for the given data \( (t_{\text{w}}, \varphi_{\text{w,M}}, p, m_w) \) and to obtain the parameters that characterize the state D \( (t_{\text{D}}, \varphi_{\text{w,D}}, p) \);

- the next level is to generate sensitivity studies for creating the nomographs, by varying some given data, for example the parameters of humid air at the inlet of the humidifying chamber or the available water temperature.

These results are presented in figures 6-15 and tables 1-4.

Numerical and graphical results

The numerical simulation was attained by the aid of EES software [19].

The obtained results are presented below.

For a first set of nomographs, we have studied different inlet states being characterized by the same temperature \( t_{\text{w}} \), but different relative humidities \( \varphi_{\text{w,M}} \) (or moisture contents \( x_{\text{w,M}} \)).

Figure 6a represents a sensitivity study with respect to available water temperature. At a given water temperature \( t_{\text{w}} \), and fixed inlet state M, this graphic gives us the value of the spraying coefficient and specifies the exit state D in terms of relative humidity \( \varphi_{\text{w,D}} \) (and temperature of course, from the condition \( t_{\text{D}} = t_{\text{w}} \) that could be reached. Or, it can be interpreted in the other sense, namely for a desired state D specified by its relative humidity \( \varphi_{\text{w,D}} \) and temperature \( t_{\text{w}} \), one could determine the amount of water per kilogram of dry air (i.e. the spraying coefficient) and the water temperature needed to reach that state. The corresponding numerical values are presented in table 1.

Figure 6b represents a similar case, but for another fixed state M, at a higher relative humidity \( \varphi_{\text{w,M}} \). Comparing the two figures, one could notice that available water temperature at 31°C, for example, would increase the
relative humidity of the air from 40% up to 69% (fig. 6b), or from 20 to 40% (fig. 6a) during the analyzed thermodynamic process, function on the inlet state M, at almost the same value of the spraying coefficient (2.993 g/kg, respectively 2.947 g/kg). These values could also be noticed in table 1 and in figure 7.

For a fixed inlet state M, the figure 8 gives us the required values for the spraying coefficient and corresponding water temperature in order to achieve a desired outlet state D (specified here by its moisture content); this figure was built for completing the information given by figure 6b.

Table 1

| CHOOSING THE WATER TEMPERATURE AND SPRAYING COEFFICIENT IN ORDER TO REACH THE OUTLET STATE D FROM A FIXED INLET STATE M (t_M=38°C) |
|---|---|---|---|---|---|---|---|---|---|---|---|
| t_M | m_w/ma | t_d | m_d | t_w | m_w | t_d | m_d | t_w | m_w | t_d | m_d |
| 21 | 7.089 | 51.97% | 60.38 | 0.00150 | 22 | 6.67 | 22.99% | 60.37 | 0.00195 | 23 | 6.267 | 23.82% | 60.54 | 0.00146 |
| 24 | 5.934 | 24.75% | 60.34 | 0.00142 | 25 | 5.441 | 25.69% | 60.52 | 0.00138 | 26 | 5.049 | 26.59% | 60.65 | 0.00134 |
| 27 | 4.651 | 27.57% | 60.72 | 0.00142 | 28 | 4.198 | 28.52% | 60.94 | 0.00126 | 29 | 3.782 | 29.48% | 61.01 | 0.00131 |
| 30 | 3.365 | 30.44% | 61.07 | 0.00117 | 31 | 2.961 | 31.40% | 61.13 | 0.00112 | 32 | 2.558 | 32.36% | 61.19 | 0.00109 |
| 33 | 2.158 | 33.31% | 62.04 | 0.00105 | 34 | 1.759 | 34.27% | 62.04 | 0.00101 | 35 | 1.360 | 35.23% | 62.04 | 0.00097 |
| 36 | 0.967 | 36.19% | 59.88 | 0.00092 | 37 | 0.564 | 37.16% | 59.83 | 0.00088 | 38 | 0.161 | 38.50% | 59.67 | 0.00084 |

Table 2

| CHOOSING THE WATER TEMPERATURE AND SPRAYING COEFFICIENT IN ORDER TO REACH THE OUTLET STATE D FROM A FIXED INLET STATE M (t_M=15°C) |
|---|---|---|---|---|---|---|---|---|---|---|---|
| t_M | m_w/ma | t_d | m_d | t_w | m_w | t_d | m_d | t_w | m_w | t_d | m_d |
| 1.067 | 69.96% | 39.57 | 0.0008 | 2.354 | 23.33% | 38.15 | 0.0008 |
| 2.85 | 8.74% | 25.18 | 0.0003 | 2.836 | 8.99% | 23.28 | 0.0006 |
| 2.44 | 9.64% | 25.07 | 0.0006 | 2.45 | 9.78% | 23.28 | 0.0005 |
| 2.08 | 10.49% | 25.07 | 0.0002 | 2.09 | 10.67% | 23.27 | 0.0003 |
| 1.72 | 11.26% | 25.07 | 0.0008 | 1.64 | 11.79% | 23.29 | 0.0009 |
| 1.36 | 12.83% | 25.04 | 0.0004 | 1.32 | 13.17% | 23.32 | 0.0005 |
| 1.01 | 13.39% | 25.02 | 0.0006 | 0.91 | 13.64% | 23.35 | 0.0009 |
| 0.66 | 14.05% | 25.01 | 0.0003 | 0.55 | 14.60% | 23.36 | 0.0004 |

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Usually, for air conditioning applications, one needs to compute the enthalpy and moisture content variations during thermodynamic processes. Thus, figures 9a and 9b gives us these variations function on the water temperature and spraying coefficient, respectively, for a fixed inlet state temperature. The inlet state relative humidity has no influence on \( \Delta H \) and \( \Delta x \) values since the enthalpy variation does not depend on \( H_M \) but on the water enthalpy and spraying coefficient, according to eq.(8).

By using figure 10 one can determine the outlet state D and the corresponding spraying coefficient function on the available water temperature \( t_w \), for different relative humidities and a constant inlet state temperature. Or, for a given state M (described by its temperature and relative humidity) and a desired state D, one can determine the water temperature and the corresponding spraying coefficient needed to reach that state, as the arrows show on figure 10.

The corresponding numerical values are also presented in table 1. One can also notice the range of possible values that could be obtained for state D.

Obviously, similar nomographs and tables could be generated for different inlet state temperatures \( t_M \). Figure 11 is just an example. The corresponding numerical values are shown in table 2.

In figure 12 one could observe the range for water temperature function on the inlet state D; a sensitivity study with respect to the inlet temperature is represented, for a fixed relative humidity. Applying this analysis for different relative humidities, table 3 is generated and could be used to find the water temperature and spraying coefficient values for a desired and physical possible outlet state D.
Another set of nomographs and tables is generated under another hypothesis, namely we no more impose the constraint $t_D = t_w$. The humidification process could undergo up to any desired temperature for the outlet state $D$, above the water temperature. The inlet state is fixed and characterized by its temperature and relative humidity (or moisture content), the available water temperature is also fixed and one could choose (fig. 13) the value of the spraying coefficient in order to reach a desired outlet state in terms of relative humidity ($O_Y$ axis) and temperature ($O_X$ axis). In order to get nomographs, other curves were added, for different inlet states (fig. 14); numerical values are presented in table 4.

Once a set of nomographs and tables is generated, it could be used for any computations involving adiabatic mixing of water injected into moist air. Their use is very simple and rapid and also very efficient in the absence of computer, specialized software or time.

As we have mentioned before, an analytical approximation could be also used. In figure 15 one could notice the relative error of using the proposed analytical simplification when computing the outlet state parameters. The error is less than 0.2% for the moisture

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**Table 3**

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**Table 4**

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**Fig. 13.** Fixed inlet state $M$; sensitivity study with respect to water temperature $t_w$ and spraying coefficient $m_w/m_a$.

**Fig. 14.** Outlet state and the corresponding spraying coefficient for a fixed inlet state temperature and available water temperature of 10°C; sensitivity study with respect to the relative humidity $\phi_M$.

**Fig. 15.** Relative error of computing the outlet state parameters $\phi_D$ and $x_D$ respectively, by using the analytical approximation (eq.(13))
content calculation and extremely close to zero when computing the enthalpy.

Conclusions
A thermodynamic study of adiabatic mixing of water injected into atmospheric (moist) air was presented. The obtained results were generated in numerical and graphical layout for an easier and quicker use.

The originality of this paper consists in presenting nomographs that allow the user to decide upon the humidification parameters (water temperature, respectively mass flow rate or spraying coefficient) depending on the required thermodynamic state at the humidifier exit, without other calculus, but just by reading the data from tables or nomographs. The nomographs are developed so that the user could find the searched values function on different input data, depending on the studied situation, different inlet state parameters or desired outlet state. As perspective, a booklet could be built with these tables, as the ones available for vapour properties.

Also a simpler thermodynamic approximation is presented for use, the relative error being indeed negligible.

Nomenclature

c_p - specific heat at constant pressure [kJ/kgK]
f - fraction of recirculated air with respect to fresh air [-]
H - enthalpy of moist air [J/kg of dry air]
\( h' \) - specific enthalpy of saturated liquid state [J/kg]
\( h'' \) - specific enthalpy of saturated vapor state [J/kg]
T - temperature [K]
m - mass flow rate [kg/s]
x - moisture content [kg vapors / kg dry air]

Greek symbols

\( \varepsilon \) - process direction
\( \phi \) - relative humidity
\( \mu \) - spraying coefficient

Subscripts

a - dry air
C - conditioned air
E - exterior humid air
I - interior recirculated air
M - adiabatic mixture
v - water vapors
w - liquid water
0 - relative to atmospheric air

References

17. *** STAS 6648/2-82 – Parametrii climatici exterior.

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