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DEVELOPMENT AND ANALYSIS OF AN ECONOMIZER CONTROL STRATEGY ALGORITHM TO PROMOTE AN OPPORTUNITY FOR ENERGY SAVINGS IN AIR CONDITIONING INSTALLATIONS

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Abstract. *This work presents an algorithm control strategy denominated enthalpic economizer. The objective of this algorithm strategy is to determine the adequate fractions of outside and return air flowrates entering a cooling coil based on the analysis of the outside, return and supply air enthalpies, rather than on the analysis of the dry bulb temperatures. The proposed algorithm predicts the actual opening position of the outside and return air dampers in order to provide the lower mixing air enthalpy. First, the psychometrics properties of the outside and return air are calculated from actual measurements of the dry and wet bulb temperatures. Then, three distinct cases are analyzed: (i) the enthalpy of the outside air is lower than the enthalpy of the supply air (free cooling); (ii) the enthalpy of the outside air is higher than the enthalpy of the return air; (iii) the enthalpy of the outside air is lower than the enthalpy of the return air and higher than the temperature of the supply air. Different outside air conditions were selected in order to represent typical weather data of Brazilians cities, as well as typical return air conditions. It was found that the enthalpy control strategy could promote an opportunity for energy savings mainly during mild nights and wintertime periods as well as during warm afternoons and summertime periods, depending on the outside air relative humidity. The proposed algorithm works well and can be integrated in some commercial automation software to reduce energy consumption and electricity demand.*

Keys-word: Economizer, Air Conditioning, HVAC, control

1. INTRODUCTION

Electricity consumption in commercial buildings plays a significant role in the scenery of the global energy consumption of a country. For example, in United States approximately one third of the total electricity consumption comes from the air conditioning installations of commercial buildings, while about seventy five percent of the peak demand experimented by the electricity utilities occurs during summer time.

In tropical weather, such as in Brazil, electricity consumption in commercial buildings is certainly representative and tends to raise rapidly, due to requirements for comfort, health and productivity of their occupants.

The Brazilian metropolis buildings are certainly responsible for most of the commercial electricity consumption, mainly due to the high electricity consumption rates required by their air conditioning installations. Additionally, most of the air conditioning systems installed in the large commercial buildings (e.g., shopping centers, high rise buildings, hotels, hospitals) are *''two pipes fan-coil system"* type. Basically, this system does not have sophisticated controls to promote energy savings as well as to promote a better indoor air quality.

During the last years, software developers of the HVAC area have launched in the market commercial automation programs to make the supervisory and controlling of the commercial buildings systems. This fact created a real opportunity to developing control strategies that can promote both energy savings and improvement of the indoor air quality. Also, the availability of new hardware for air conditioning control has contributed to the implementation of the control strategies for air conditioning systems.

This paper is concerned with the potential energy savings that can be carried out when appropriate flow rates of outside and return air are selected to enter the cooling coil. This control strategy is denominated *economizer*. Honeywell (1991) and Kreider (1994) described some basic control strategies to determine the proper outside and return air flow rates. The following three strategies can be used: *(i)* to adjust the opening positions of the outside and return air dampers to a fixed value, based on values given by rules of thumb and standards; *(ii)* to allow the dampers to adjust their opening positions automatically in response to the changes in the outside and return air dry bulb temperatures; (iii) to allow the dampers to adjust their opening positions automatically in response to the changes in the enthalpies of the return and outside air. The most common strategy that has been used in the air conditioning installations is *the ''fixed opening damper position''*, due to its simplicity and low cost. Following is the *''dry bulb control opening damper position''* strategy.

In this paper we present an algorithm called *''enthalpic economizer''*, based on enthalpy comparison rather than temperature comparison. This strategy consists in develop a control algorithm to determine the best opening position of the outside and return air dampers, in order to provide the lowest energy consumption in the air conditioning systems.

2. ENTHALPIC ECONOMIZER MODEL

The input variables required to this model are: (i) the dry and wet bulb temperatures of the outside air, return air and supply air (i.e., the air entering the space); (ii) the required fraction of outside air for ventilation; (iii) the local altitude.

The following steps are used to determine the best control strategy:

1. Calculation of the atmospheric pressure in function of the local altitude.
2. Calculation of the enthalpies of the outside air, return air and supply air using psychometric equations.
3. **Free cooling Mode**: The enthalpy of the outside air is lower or equal than the enthalpy of the supply air. In this case, the cooling equipment can be disabled. The model determines the fraction of outside air to the total supply air, and the fraction of the return air to the total supply air, in order to maintain the enthalpy of the mixing air equal to the enthalpy of the supply air.
3.1 The calculated fraction of outside air is adopted if it is higher than the fraction of outside air required for ventilation.
3.2 The fraction of outside air required for ventilation is used if it is higher or equal than the calculated air outside air fraction.
3.3 The new temperature of the mixing air is calculated based on the fraction of the outside air determined in items 3.1 or 3.2.
4. **100% outside air Mode**: If the enthalpy of the outside air is higher than the enthalpy of the supply air and lower than the enthalpy of the return air, the outside air fraction is 100%.
5. **Minimum outside air Mode**: If the enthalpy of the outside air is higher than the enthalpy of the return air, the outside air fraction is the minimum air fraction required for ventilation.

Figure 1 shows the algorithm for the *enthalpic economizer*. The mentioned variables are as follows:

- $T_{db,OA}$: Dry bulb temperature of the outside air, (°C)
- $T_{wb,OA}$: Wet bulb temperature of the outside air, (°C)
- $T_{db,RA}$: Dry bulb temperature of the return air, (°C)
- $T_{wb,RA}$: Wet bulb temperature of the return air, (°C)
- $T_{db,SA}$: Dry bulb temperature of the supply air, (°C)
- $T_{wb,SA}$: Wet bulb temperature of the supply air, (°C)
- h_{OA} : Enthalpy of the outside air, KJ/Kg$_{da}$
- h_{RA} : Enthalpy of the return air; KJ/Kg$_{da}$
- h_{SA} : Enthalpy of the supply air, KJ/Kg$_{da}$

- OA_{req} : Volumetric fraction of outside air required for ventilation, %
- OA_{calc} : Volumetric fraction of outside air calculated, %

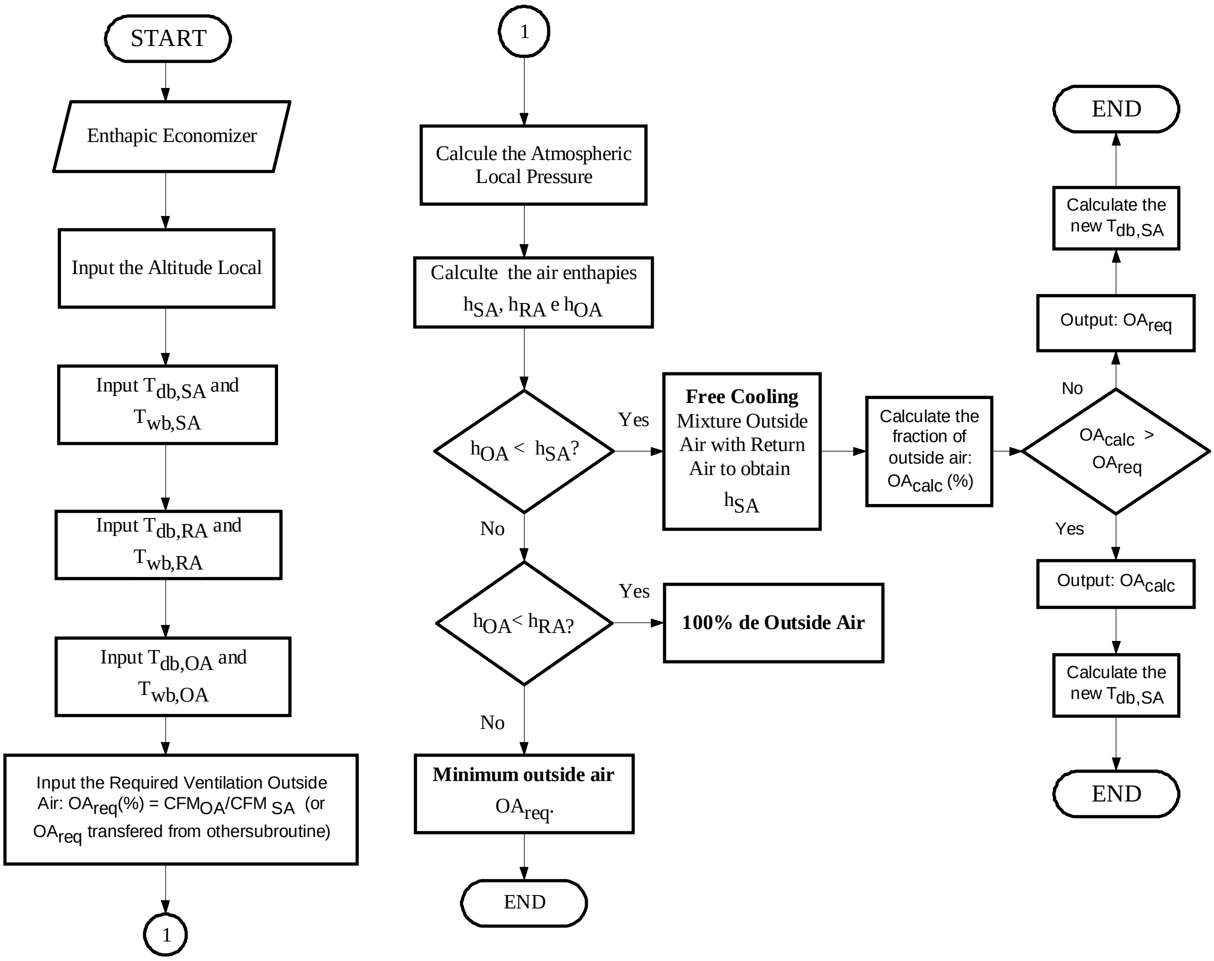

Figure 1 - Algorithm for the enthalpic economizer.

3. PSYCROMETRIC FORMULATION

The enthalpies of the outside air, return air, and mixing air are determined from equation (1):

$$h = c_{p,da} \times T_{db} + \left(\omega \times \left(h_{fg} + c_{p,w} \times T_{db}\right)\right) \tag{1}$$

where:

$c_{p,da}$ is the specific heat of the dry air, [KJ/Kg$_{da}$ /°C]

T_{db} is the dry bulb temperature [°C]

ω is the humidity ratio [Kg$_w$/Kg$_{da}$/°C]

$c_{p,w}$ is the specific heat of the water vapor, [KJ/Kg$_w$ /°C]

h_{fg} is the latent heat of the water, at the reference temperature of 0 °C, [KJ/Kg$_w$]

The humidity ratio, ω, is calculated from equation (2):

$$\omega = 0.622 \times \left(\frac{\phi \times P_{sat,\,db}}{P_{atm} - \phi \times P_{sat,\,db}} \right) \tag{2}$$

where:

ϕ, is the relative humidity, [%]

$P_{sat,db}$ is the saturation pressure at the dry bulb temperature, [MPa]

P_{atm} is the local atmospheric pressure, [MPa]

The relative humidity, ϕ, is determined using the algorithm described by Pallady (1989) and mentioned in (Kreider, 1994). The relative humidity is given in function of the saturation pressure at the dry bulb temperature, $P_{sat,db}$, the saturation pressure at the wet bulb temperature, $P_{sat,wb}$, and the average pressure, P_m . The average pressure is used to take into account the depression pressure caused by the web bulb temperature. The relative humidity is given by equation (3):

$$\phi = \frac{P_{sat,\,wb} - P_m}{P_{sat,\,db}} \tag{3}$$

where

$P_{sat,db}$ is the saturation pressure of the dry bulb temperature, [MPa]

$P_{sat,wb}$ is the saturation pressure of the wet bulb temperature, [MPa]

P_m is the average pressure due to the web bulb depression, [MPa]

The saturation pressures, $P_{sat,db}$ and $P_{sat,wb}$, depend on the saturation temperatures of the dry and web bulb temperatures. Equations (4) and (5) are based on regression analysis.

$$P_{sat,\,db} = PC \times 10^{\left(K_{db} \times \left(1 - \frac{TC}{T_{db}} \right) \right)} \tag{4}$$

$$P_{sat,\,wb} = PC \times 10^{\left(K_{wb} \times \left(1 - \frac{TC}{T_{wb}} \right) \right)} \tag{5}$$

where:

PC is the critical pressure of the water, [MPa]

TC is the critical temperature of the water, [K]

T_{db} is the dry bulb temperature of the air, [K]

T_{wb} is the wet bulb temperature of the air, [K]

The constants K_{db} and K_{wb} are given by equations (6) and (7), respectively:

$$K_{db} = 4.39558 - 6.2442 \times \left(\frac{T_{db}}{1000} \right) + 9.53 \times \left(\frac{T_{db}}{1000} \right)^2 - 5.151 \times \left(\frac{T_{db}}{1000} \right)^3 \tag{6}$$

$$K_{wb} = 4.39558 - 6.2442 \times \left(\frac{T_{wb}}{1000}\right) + 9.53 \times \left(\frac{T_{wb}}{1000}\right)^2 - 5.151 \times \left(\frac{T_{wb}}{1000}\right)^3 \qquad (7)$$

The average pressure, P_m , is given by equation (8):

$$P_m = P_{atm} \times \left(\left(\left(\frac{T_{db} - T_{wb}}{1514}\right) \times \left(1 + \left(\frac{T_{wb}}{873}\right)\right)\right)\right) \qquad (8)$$

The atmospheric local pressure, P_{atm} , is given in function of local altitude, as

$$P_{atm} = 1.01323 \times E^{-1}\left(1 - \left(\frac{6.5 \times H_{local}}{288000}\right)\right)^{5.25} \qquad (9)$$

where:

P_{atm} , is given in [MPa]

H_{local} is the local altitude, [m]

The constants described above have the following values:

PC = 22.1 MPa

TC = 647.3 K

$c_{p,da}$ = 1.003 KJ/Kg_{da} /°C

$c_{p,w}$ = 1.86 KJ/Kg_w /°C]

h_{fg}, = 2501.3 KJ/Kg_w]

The opening position of the outside air and return dampers depend on the volumetric fraction rather than on the mass fraction. Therefore, it is necessary to convert the calculated outside air mass flow in volumetric basis. The volumetric flow of the outside air is determined from the following equation:

$$\dot{V}_{oa} = \dot{M}_{oa} \times v_{oa} \qquad (10)$$

where:

$\dot{V}_{oa}$ is the volumetric flow rate of the outside air, [m^3/s]

$\dot{M}_{oa}$ is the mass flow rate of the outside air, [Kg/s]

v_{oa} is specific volume of the outside air, [m^3/Kg_{da}]

The specific volume of the outside air is determined from the ideal gas equation:

$$v_{oa} = \frac{R_{da} \times T_{db}}{\left(P_{atm} - \phi \times P_{sat,db}\right)} \qquad (11)$$

where:
R_{da} is the constant of the ideal dry air, [0,287 kJ/kg K]

4. VALIDATION OF THE PSYCROMETRICS

Table 1 shows the relative humidity, specific volume, enthalpy, and absolute humidity of the outside air obtained from the equations shown above and from the psycrometric chart at sea level for different outside air dry bulb and wet bulb temperatures.

Table 1. Comparison between the humid air properties obtained from the psycrometric chart and calculated from the equations.

T_{db} ºC	T_{wb} ºC	ϕ_{chart} %	ϕ_{calc} %	v chart m^3/kgda	v calc. m^3/kgda	h_{chart} KJ/Kgda	$h_{calc.}$ KJ/Kgda	w chart gw/kda	w calc. gw/kda
10	8.33	80.00	81.03	0.8095	0.8108	25.00	27.24	6.125	6.830
11	8.37	70.00	71.56	0.8121	0.8132	26.00	27.28	5.738	6.440
12	8.15	60.00	60.50	0.8140	0.8152	25.30	26.71	5.250	5.820
13	8.06	50.00	51.76	0.8160	0.8174	25.20	26.45	4.625	5.310
13	9.17	60.00	61.89	0.8176	0.8188	27.80	29.10	5.625	6.360
14	7.67	40.00	41.43	0.8180	0.8193	24.40	25.49	4.000	4.530
15	8.00	35.60	37.83	0.8210	0.8220	26.40	26.20	4.000	4.410
15	9.83	50.00	52.63	0.8229	0.8243	28.40	30.61	5.000	6.160
16	8.00	30.00	32.12	0.8233	0.8243	24.40	26.15	3.375	3.990
16	9.00	37.22	39.59	0.8245	0.8255	26.68	28.52	4.250	4.930
17	9.00	30.00	33.99	0.8270	0.8278	27.00	28.41	4.000	4.510
18	10.00	35.00	35.76	0.8300	0.8314	29.70	30.86	4.500	5.150
19	10.00	30.00	30.80	0.8330	0.8337	29.70	30.81	4.000	4.630
20	10.00	25.00	26.33	0.8350	0.8360	29.70	30.75	3.500	4.210
21	11.50	30.00	31.13	0.8400	0.8403	33.00	34.54	4.700	5.300
22	17.19	62.30	62.94	0.8502	0.8515	48.20	51.34	10.313	11.520
23	17.25	56.65	57.43	0.8521	0.8539	48.40	51.46	10.000	11.160
24	17.06	50.00	50.82	0.8540	0.8559	48.00	50.75	9.312	10.480
24	18.63	60.00	60.82	0.8570	0.8587	53.00	56.11	11.250	12.580
24	20.06	70.00	70.47	0.8590	0.8615	57.52	61.31	13.125	14.620
25	16.19	40.00	40.94	0.8550	0.8567	46.50	47.85	7.875	8.940
26	17.00	40.00	41.12	0.8586	0.8603	52.44	50.39	10.500	9.540
27	19.55	50.00	50.84	0.8650	0.8674	55.28	59.15	11.187	12.570
28	18.50	40.00	40.80	0.8660	0.8676	52.60	55.31	9.500	10.660
29	20.00	44.00	44.45	0.8707	0.8728	57.00	60.63	11.000	12.340
30	20.00	39.80	40.51	0.8730	0.8751	56.80	60.54	10.500	11.910

A statistical analysis based on a linear regression was conducted to evaluate the calculated properties of the humid air. Table 2 shows the square of the Pearson product moment correlation ‘‘R^2’’ and the standard error ‘‘SD’’ for the relative humidity, specific volume, enthalpy, and absolute humidity. Table 2 shows that the calculated properties are in good accordance with the same properties found by the psycrometric chart.

Table 2. Statistical Analysis of the calculated properties of the humid air.

STATISTICAL	ϕ (%)	v (m^3/kg_{da})	h (KJ/Kg_{da})	w (gw/Kg_{da})
R^2	0.996667	0.9996606	0.9931566	0.982652
SD	0.856515	0.0003838	1.1001074	0.4193898

5. RESULTS

Table 3 shows a sample of the input and output data obtained from the proposed algorithm. These results are for locations at sea level. The fraction of volumetric outside air required for ventilation is ‘‘**Req**’’, while the calculated fraction of volumetric outside air recommended for energy savings is ‘‘**Calc**’’. There are three operation modes: Type ‘‘**A**’’ is the free cooling strategy; type ‘‘**B**’’ is the 100% outside air strategy, and type ‘‘**C**’’ is the minimum outside air strategy. In case ‘‘**A**’’, the algorithm outputs a new supply air temperature, **‘‘calc. supply air**’’, necessary to maintain constant the required supply air enthalpy ‘‘h_{sa}’’.

Table 3. Results of the economizer control.

Input Parameters						**Output Parameters**						
Outside Air (ºC)		**Return Air (ºC)**		**Supply Air (ºC)**		**Enthalpy KJ/Kgda**			**Outside air fraction (%)**		**Calc. Supply Air (ºC)**	**Mode**
Tdb	Twb	Tdb	Twb	Tdb	Twb	hoa	hra	hsa	Req	Cal	Tdb	Type
10	8.3	24	17	13	8.1	27.24	50.55	26.45	20	100	13.0	B
11	8.4	24	17	13	8.1	27.28	50.55	26.45	20	100	13.0	B
12	8.2	24	17	13	8.1	26.71	50.55	26.45	20	100	13.0	B
13	8.0	24	17	13	8.1	26.31	50.55	26.45	20	99.4	13.1	A
14	7.7	24	17	13	9.2	25.49	50.55	29.10	20	85.7	15.4	A
15	8.0	24	17	13	9.2	26.20	50.55	29.10	20	88.5	16.1	A
15	9.8	24	17	13	9.2	30.61	50.55	29.10	20	100	13.0	B
16	8.0	24	17	13	9.2	26.15	50.55	29.10	20	88.5	17.0	A
16	9.0	24	17	13	9.2	28.52	50.55	29.10	20	98.2	16.2	A
17	9.0	24	17	13	9.2	27.00	50.55	29.10	20	97.1	17.20	A
18	9.0	24	17	13	9.	29.7	50.5	29.	20	100	13.0	B

					2	0	5	10				
19	9.0	24	17	13	9.2	29.70	50.55	29.10	20	100	13.0	B
20	9.0	24	17	13	9.2	29.70	50.55	29.10	20	100	13.0	B
21	9.0	24	17	13	9.2	29.70	50.55	29.10	20	100	13.0	B
22	17.2	24	17.1	13	8.1	51.34	50.75	26.45	20	20	13.0	C
23	17.3	24	17.1	13	8.1	51.46	50.75	26.45	20	20	13.0	C
25	16.2	24	17.1	13	8.1	47.85	50.75	26.45	20	100	13.0	B
26	17.0	24	18.6	13	8.1	50.39	56.11	26.45	20	100	13.0	B
27	19.6	24	20.1	13	8.1	59.15	61.31	26.45	20	100	13.0	B
28	18.5	24	20.1	13	8.1	55.31	61.31	26.45	20	100	13.0	B
29	20	24	20.1	13	8.1	60.63	61.31	26.45	20	100	13.0	B
30	20	24	20.1	13	8.1	60.54	61.31	26.45	20	100	13.0	B

Figure 2 shows the volumetric fraction of outside air for different outside air dry bulb temperatures, when two strategies control are used: the dry bulb temperature strategy and the economizer strategy, respectively.

Figure 2 is based on the data shown in Table 3. The results show that the dry bulb strategy can carry out erroneous outputs for the outside air damper position, since it analyzes only the dry bulb temperature rather than the dry and wet bulb temperatures simultaneously. Therefore, the enthalpy strategy based on the analysis of the dry and wet bulb temperatures of the outside air are most reliable and can promote higher energy savings than the dry bulb strategy.

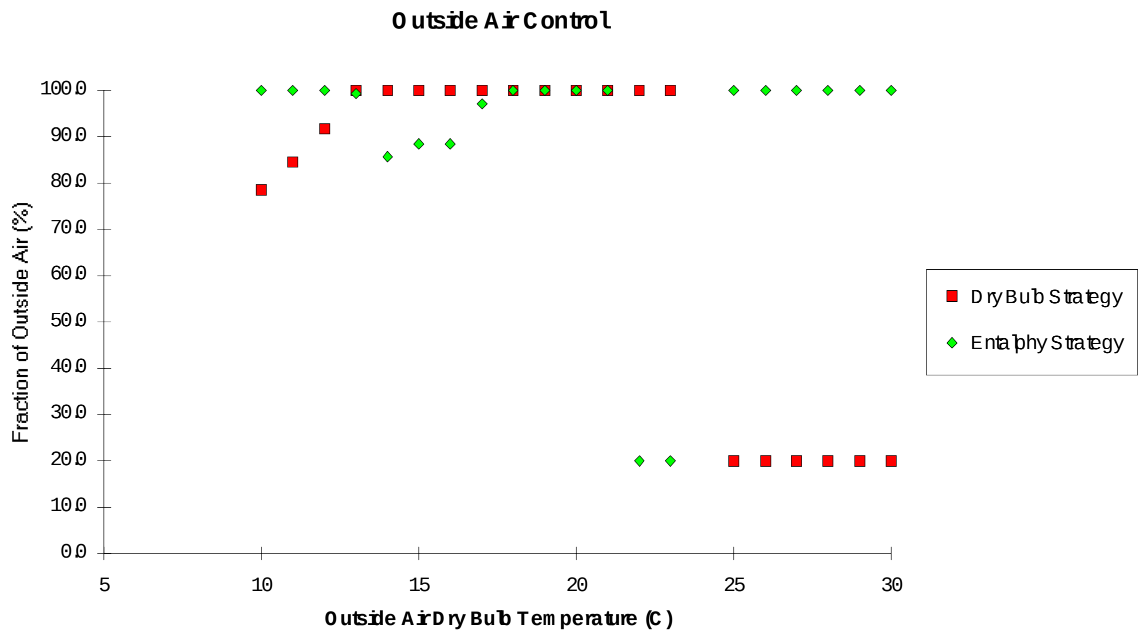

Figure 2 - Volumetric fraction of outside air in function of the outside air dry bulb temperature for the dry bulb temperature strategy and for the enthalpy strategy.

A analysis of Figure 2 and Table 2 carries out the following observations:

1. The economizer takes advantage of the *free cooling* if the outside air temperature drops below the supply air temperature and its relative humidity is less than 50%.
2. The economizer allows 100% of outside air, if the outside air temperature is lower than the supply air temperature but its humidity is higher than 50%, or if its temperature is higher than the return air but its relative humidity is less than 50%.
3. The economizer will restrict the outside air fraction to the minimum fraction required for ventilation if the outside air temperature and humility are higher than the return air temperature and humidity, respectively.
4. The economizer strategy can promote a better control than the dry bulb strategy mainly during the low temperature period (wintertime and mild nights) and high temperature period (summertime or warm afternoons) depending on the relative humidity of the outside air.

6. CONCLUSIONS

This paper presents a control strategy denominated "*enthalpic economizer*". This strategy can be used to determine the optimal opening of the outside air damper, in order to supply an adequate outside air flow necessary to achieve the lowest energy consumption, while maintaining the requirements for the indoor air quality. This strategy is based on the comparison of the outside air enthalpy with the return air enthalpy and can operate in three different modes: *free cooling*, *100% outside air*, and *minimum outside air*.

The psycrometric air properties obtained from the proposed control algorithm were validated against air humid properties obtained from the psycrometric chart, for various outside air conditions. The results show excellent accordance.

The algorithm was tested with typical outside air conditions in order to determine the best operation mode, and the results show consistency. It was found that the *enthapic economizer* strategy can predict the outside air flow with more accuracy and reliability than the dry bulb strategy.

The *enthalpic economizer algorithm* can be useful to control the appropriate outside air flow rate, due to the daily and seasonal time variation of the outside air temperature.

The *enthalpy economizer* strategy can be integrated in commercial automation software, contributing significantly to the reduction of the energy consumption in air conditioning installations of commercial buildings.

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