THEORETICAL ALGORITHM OF A MODIFIED CAPACITY CONTROL OF CO-PARALLEL CHILLED WATER SPRAY DEHUMIDIFIER

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ABSTRACT

A wide range control band of temperature and relative humidity at exit from a co-parallel chilled water spray dehumidifier can be achieved using a master proportional valve and a submaster three way valve. The two valves are assumed to be actuated from a microprocessor unit in order to modulate the total amount of chilled water flow rate as well as the percent of water flow rate to both parallel and counter banks. A theoretical algorithm is presented to obtain the partial sensible and total cooling capacities chart as well as a basic performance chart in order to link the dehumidifier characteristics with the nominal capacities stated at the ARI standard 441-66 conditions of air and water.

NOMENCLATURE

a_H	Heat transfer area per unit volume, m^2/m^3.
a_M	Mass transfer area per unit volume, m^2/m^3.
CC	Total cooling capacity per unit mass of air, J/kg.
C_{pm}	Specific heat of humid air, J/kg k.
C_w	Specific heat of water, J/kg K.
G	Air mass velocity, kg/s m^2.
H_a	Humidity ratio of bulk air, kg_v/kg_a.
H_i	Interfacial film humidity ratio, kg_v/kg_a.
h_a	Air side heat transfer coefficient, W/m^2 K.
h_w	Water side heat transfer coefficient, W/m^2 K.
i_a	Bulk air enthalpy, J/kg.
i_{fg}	Latent heat of evaporation, J/kg.
i_i	Interfacial film enthalpy, J/kg.
K_a	Mass transfer coefficient, kg/s m^2.
L	Water mass velocity, kg/s m^2.
NTU	Number of transfer unit, $=K_a a_M X/G$.
R.H.	Air relative humidity, %.
SC	Sensible cooling capacity per unit mass of air, J/kg.
SHF	Sensible heat factor of dehumidifier = SC/CC.
T_a	Bulk air temperature, K or °C.
T_i	Interfacial film temperature, K or °C.
T_w	Bulk water temperature, K or °C.
T_{wb}	Air wet bulb temperature, K or °C.
X	Length of air-to-water contact, m.

Subscripts

av	Available.
b	Basic.
c	Counter.
in	Inlet.
o	Outlet.
p	Parallel.
t	Total (parallel+counter).
ARI	Air Conditioning and Refrigeration Institute.

INTRODUCTION

The present progress in the modern automatic control systems using the microprocessor units with printed circuit elements leads to use easily the multi-level/multi-sensor control systems specially in air conditioning field in which both the temperature and the humidity should be adjusted [1]. The usage of the direct and conventional types of control systems may lead to wide deviation or oscillatory behaviour in the dry bulb temperature and the relative humidity as well as the slow response in partial load operation.

In the chilled water spray dehumidifier, the common present conventional control systems are to by-pass the air or modulating the total amount of water flow rate to the spray banks or combination of the two systems [2]. A two-stage combined indirect evaporative cooling was suggested by [3 and 4] to obtain accurate exit air condition from washers in general. In order to determine the characteristics and performance of the direct contact exchangers, it is recommended to analyse the system algorithm of simulation which adjusts the system number of transfer units and the air exit condition. Webb [5] presented theoretically the basic computer simulation algorithms for cooling towers, fluid coolers and evaporative condensers. He used Merkel's equations and the modifications of Mickley [6]. The method of simulation depended essentially on the experimental study of [7 and 8].

The present work modifies the control system of the chilled water spray dehumidifier which depends on modulating the total flow rate of the chilled water accompanied by modulating the percentage water rate to each bank. The system algorithm is applied on co-parallel tube banks arrangement. Also the target of this research is to enable the microprocessor designers to predict the required sensible and total cooling capacities in order to obtain a certain exit air condition of dry bulb temperature and relative humidity upon modulating the total water flow rate and further its percent to each bank of the co-parallel chilled water spray dehumidifier.

THEORETICAL ANALYSIS

A co-parallel chilled water spray dehumidifier is illustrated in Figure (1). The model consists of two opposite banks of tubes and water sprayers. The air passes through the parallel then the counter tube bank respectively. The total water flow rate L_t is divided into parallel bank water rate L_p and counter bank rate L_c.

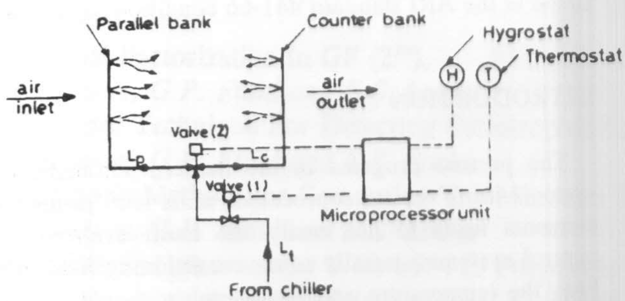

Figure 1. System arrangement.

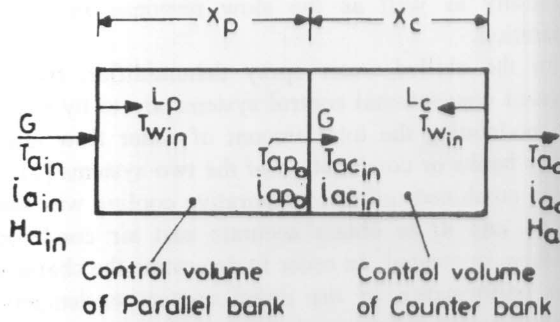

Figure 2. Calculating domain.

The parallel and counter sprays are assumed to be unmixed with each other that each spray satisfies its own transfer units. Figure (2) illustrates the system calculating domain that the energy balance equations for the parallel and counter tube banks are respectively as follows;

For total energy balance:-

$$- \int_{i_{ain}}^{i_{apo}} di_a = \frac{L_p}{G} \int_{T_{win}}^{T_{wpo}} C_w \, dT_w \tag{1.a}$$

and

$$\int_{i_{acin}}^{i_{ao}} di_a = \frac{L_c}{G} \int_{T_{wco}}^{T_{win}} C_w \, dT_w \tag{1.b}$$

Sensible heat balance:-

$$\frac{h_a a_H X_p}{G C_{pm}} = - \int_{T_{ain}}^{T_{apo}} \frac{dT_a}{T_a - T_i} \tag{2.a}$$

and

$$\frac{h_a a_H X_c}{G C_{pm}} = - \int_{T_{acin}}^{T_{ao}} \frac{dT_a}{T_a - T_i} \tag{2.b}$$

and for moisture balance:-

$$NTU_p = \frac{K_a a_M X_p}{G} = - \int_{H_{ain}}^{H_{apo}} \frac{dH_a}{H_a - H_i} \tag{3.a}$$

and

$$NTU_c = \frac{K_a a_M X_c}{G} = - \int_{H_{acin}}^{H_{ao}} \frac{dH_a}{H_a - H_i} \tag{3.b}$$

and

$$NTU_t = NTU_p + NTU_c \tag{3.c}$$

The total heat balance between the bulk air and the interface is,

$$G[C_{pm} dT_a + i_{fg} dH_a] = -[K_a a_M i_{fg} (H_a - H_i) + h_a a_H (T_a - T_i)] \tag{4}$$

Considering the Lewis relation ($K_a a_M C_{pm} / h_a a_H = 1$) and the definition of the humid air enthalpy,the total energy balance becomes,

$$G \, di_a = -K_a a_M (i_a - i_i) dX \tag{5}$$

and that for water side,

$$G \, di_a = -h_w a_H (T_i - T_w) dX \tag{6}$$

From equations (5) and (6), it is found that,

$$\frac{i_a - i_i}{T_i - T_w} = \frac{h_w a_H}{K_a a_M} \tag{7}$$

Since the air-water interface is considered a saturated air film, combination of equation (7) and ASHRAE Psychrometric algorithm [9] for saturation line, the local interfacial enthalpy and temperature as well as the humidity are obtained.

In order to relate the variation of temperature and the specific humidity in the air flow direction, equations (2) and (3) give,

$$\frac{dT_a}{dH_a} = \frac{T_a - T_i}{H_a - H_i} \tag{8}$$

The total cooling capacities per kg of dry air for parallel, counter banks, and the whole dehumidifier are calculated from,

$$CC_p = (i_{ain} - i_{apo}), \tag{9.a}$$

$$CC_c = (i_{acin} - i_{ao}) \tag{9.b}$$

and

$$CC_t = CC_p + CC_c = (i_{ain} - i_{ao}) \tag{9.c}$$

Similarly the sensible capacities are obtained from,

$$SC_t = SC_p + SC_c = C_{pm}(T_{ain} - T_{ao}) \tag{10}$$

The numerical solution is applied with $\Delta NTU_p = \Delta NTU_c = 0.0001$ with implicit finite difference calculation routine for system equations (1) through (8).

The solution is applied at first to obtain the basic sensible capacity, SC_{tb}, basic total capacity, CC_{tb}, and the basic number of transfer units NTU_b at basic nominal standard ARI 441-66, appendix (A). The results at basic condition are illustrated in Table(1).

Table 1. Results of basic condition.

SC_b KJ/Kg	CC_b KJ/Kg	SHF_b	NTU_{pb}	NTU_{cb}	NTU_{tb}
16.151	23.274	0.694	0.8297	1.9449	2.7746

The algorithm calculates the sensible and total cooling capacities as well as the air exit condition for the dehumidifier at different operating conditions. The other conditions should satisfy another value of number of transfer units called the available number of transfer unit NTU_{av}. This NTU_{av} is considered a unique characteristic of a particular direct exchanger which is normally expressed as,

$$\frac{K_G a_M X}{L} \propto \left(\frac{L}{G}\right)^n \tag{11}$$

where the average value of n ranges from -0.55 to -0.65 [7, 8 and 10]. In the present work n will be taken as -0.6,that,

$$NTU_{av} = \frac{K_G a_M X}{G} \propto \left(\frac{L}{G}\right)^{0.4} \tag{12}$$

Applying equation (12) for the parallel and counter banks separately with constant air flow rate gives,

$$\frac{NTU_{pav}}{NTU_{pb}} = \left(\frac{L_p}{L_{pb}}\right)^{0.4} \tag{13.a}$$

and

$$\frac{NTU_{cav}}{NTU_{cb}} = \left(\frac{L_c}{L_{cb}}\right)^{0.4} \tag{13.b}$$

The above two relations give the available number of transfer units corresponding to the water flow rate passing through each bank of the dehumidifier related to the basic condition.

From the previous analysis, it is obvious that the amount of flow rate of water to each bank varies the exit air conditions as well as the sensible and total capacities in two ways:- First: The system energy balance equations and Second: The available number of transfer units.

RESULTS AND DISCUSSIONS

The control map of the spray dehumidifier is illustrated in Figure (2). The proposed control system consists of a master device (valve 1) to modulate the flow rate of the chilled water coming from the water chiller to the two banks and a submaster three-way valve (valve 2) to adjust the required percentage water flow rate to each bank. The actuated signals for both valves are generated from a microprocessor unit which is programmed with the algorithm routine capable to predict the required sensible and total cooling capacities of the dehumidifier according to the signals of the supply thermostat and hygrostat.

The microprocessor unit should contain two main control charts. The first is the basic performance chart while the second is the partial capacity chart. The basic performance chart, shown in Figure(3), illustrates the variation of sensible and total cooling capacities with the total chilled water rate while the inlet air dry bulb and wet bulb temperatures as well as the inlet chilled water temperature are considered independent parameters. The chart is constructed upon the basic NTU_b for both the parallel and counter tube banks at basic condition standard ARI of air inlet condition and chilled water temperature range, appendix (A). The basic water flow rate intensity is 5.86 $m^3/hr/m^2$ per bank (2.4 gpm/ft^2) while the air velocity is 2.54 m/s (500 fpm) [11]. These values of water and air flow rates meet the ratio $L_p/G = L_c/G = 0.5$ i.e. $L_t/G = 1.0$. Figure (3) indicates that the sensible and total cooling capacities increase with increasing L_t/G ratio but the sensible heat factor SHF decreases. Also as the difference between the air inlet wet bulb and the chilled temperatures increases, the total cooling capacity increases. The sensible capacity increases with the increase of the potential between the air inlet dry bulb and the chilled water temperatures. The basic performance chart is considered a presetting chart for tuning the operating set point if the original design condition differs from the aforementioned basic condition illustrated on the figure obtained from the system algorithm.

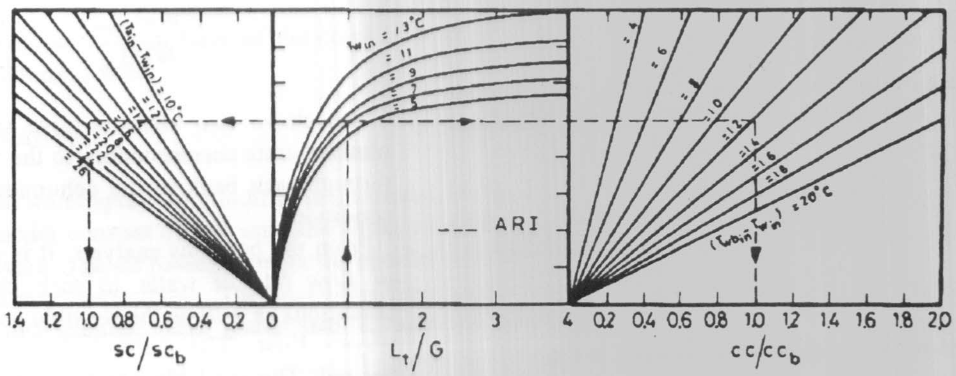

Figure 3. Basic Performance Chart.

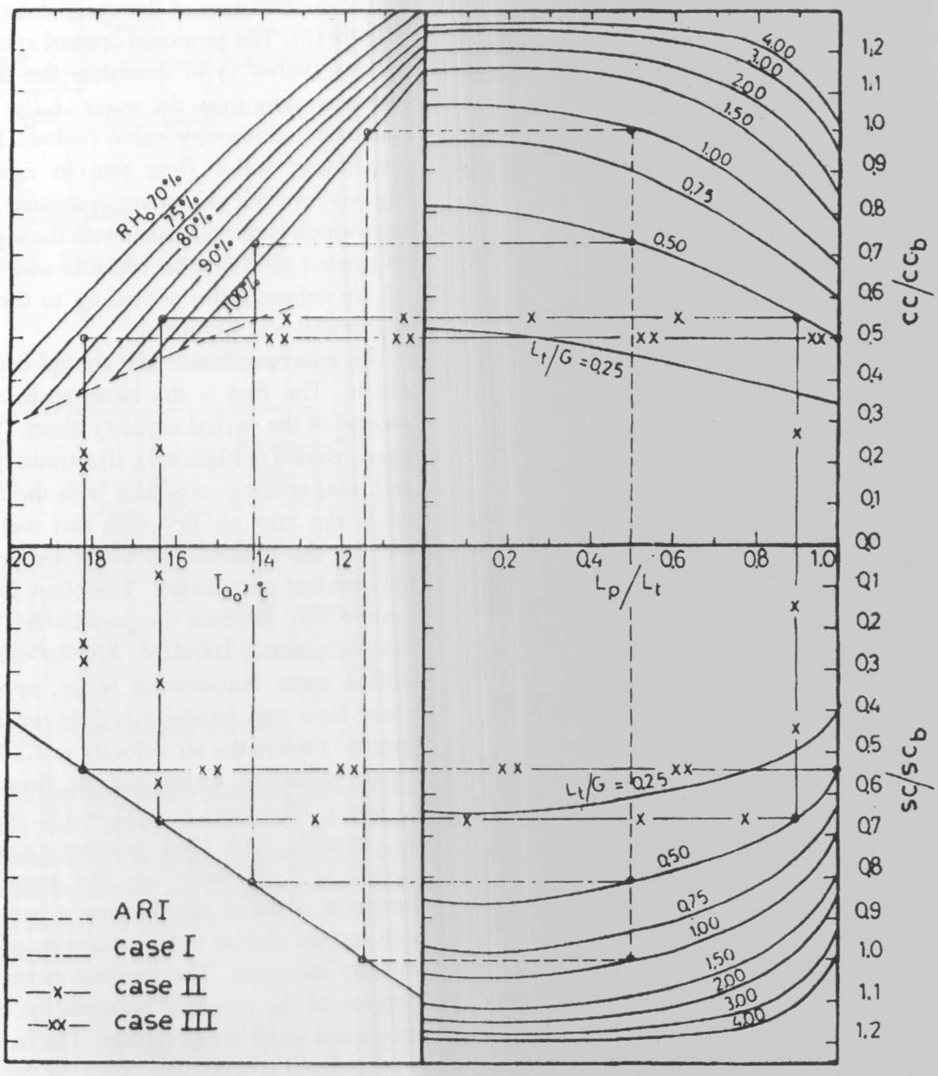

Figure 4. Partial Capacity Chart.

Figure(4) illustrates the partial capacity chart which relates the sensible and total cooling capacities as well as the air exit conditions; to the amount of total water flow rate, and the percentage water rate distribution to the parallel and counter banks. The chart is constructed by applying the algorithm with inlet air condition and water temperatures at standard ARI. Figure (4) shows that the sensible and total capacities increase with the increase of total water flow rate L_t. Also it is obvious that they decrease as the water flow rate to the parallel bank increases. Four cases including the basic nominal case are represented on the figure and their results are tabulated in table(2).

Table 2. Partial capacity performance.

Case	L_t/G	L_p/L_t	SC/SC_b	CC/CC_b	SHF/SHF_b	Exit T_{ao} $R.H._o$
ARI	1.0	0.5	1.00	1.00	1.00	11.4 °C , 98%
I	0.5	0.5	0.81	0.73	1.11	14.1 °C , 96%
II	0.5	0.9	0.66	0.55	1.20	16.4 °C , 90%
III	0.5	1.0	0.53	0.50	1.02	18.2 °C , 77%

The system simulation and performance could be detected from Table(2), that a modulation of L_t/G from 1.0 to 0.5 decreases the total capacity to 0.73 and increases the SHF by 11% (case I). Now a modulation of the ratio L_p/L_t will obtain different cases which include the required exit condition preset by the microprocessor unit. A further note is concluded from the table that dehumidifier SHF decreases again if the ratio L_p/L_t becomes greater than 0.9. Inspection of the air exit conditions concludes that the suggested control system has a wide band of the dry bulb temperature and the relative humidity control. However, due to the unlimited results which could be obtained on changing the air and water inlet conditions, it is recommended that the automatic control designer should reconstruct Figures (3) and (4) for a certain specific inlet condition different from the ARI one otherwise "Out of Valve Throttle Range" condition may occur.

CONCLUSIONS

The exit air condition from a co-parallel chilled water spray dehumidifier could be controlled by regulating the amount of the chilled water rate to each spray tube bank as well as the total water flow rate coming from the chiller. This is attained according to the concept that if the total water flow rate is constant, increasing the amount of water to the parallel bank decreases the total and the sensible cooling capacities but increases the sensible heat factor. Also if the total water flow rate decreases, the total and sensible cooling capacities decrease but the sensible heat factor increases. Therefore, wide ranges of exit dry bulb temperature and relative humidity could be obtained. A partial capacity chart and basic performance chart could be programmed in a microprocessor unit in order to modulate the water quantities according to the signals of supply or room thermostat and hygrostat.

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APPENDIX (A)

Basic ARI Standard 441-66 used in cooling and dehumidifying systems:

Inlet air dry bulb temperature	80 °F (26.67 °C).
Inlet air wet bulb temperature	67 °F (19.44 °C).
Inlet water temperature	45 °F (7.22 °C).
Outlet water temperature	55 °F (12.78 °C).