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ABSTRACT

Variable refrigerant volume (VRV) technology featured with a variable speed compressor and an electronic expansion valve (EEV) is proposed as an effective approach to help overcome the fundamental problem of capacity control for a direct-expansion (DX) variable air volume (VAV) system and improve further its energy efficiency. An experimental station of a DX VAV system with two pressure-independent VAV terminals and its complete system's dynamic simulation model are developed. The model is component-based, of partial-lumped-parameter type, and takes the dynamic characteristics of both the DX refrigeration plant and the VAV air handling system into consideration simultaneously. The model for the DX refrigeration plant includes the thermodynamic sub-models for an air-cooled condenser, an evaporator, an EEV and a variable speed compressor. The model for the VAV air handling system includes the mathematical sub-models for a supply air fan, zones, thermostats, and VAV terminals with constant-speed actuator.

1. INTRODUCTION

DX packaged air conditioning (A/C) systems, most of which are conventionally constant-air-volume operated, find wide applications in those light commercial and low rise residential buildings, such as supermarkets, schools, offices and villas. According to Department of Energy (DOE), U.S.A., packaged rooftop A/C systems, most of which are of DX type, consume about 60% of the energy use for cooling, accounting for one-third of the total heating and cooling energy use in the commercial buildings, over one quadrillion BTU per year in U.S.A (Bordick and Gilbride, 2002). A DX VAV system, a typical example of DX A/C system, integrates a VAV air handling system with a DX refrigeration plant. It is believed that such a system is high energy-efficient with significant energy saving potential and helps reduce peak demand. The evaporator in a DX refrigeration plant acts as a cooling coil in an air handler of the VAV system. Compared to a chilled-water A/C system, a DX VAV system can operate at a higher evaporator temperature for the same leaving air temperature from the coil because of the non-presence of an intermediate heat transfer medium. Meanwhile a large amount of fan energy can be saved during part load operation. Flexible individual zoning control and easy future space or zone reconfiguration can also be provided. However, it appears that there is inadequate research and development for DX VAV systems. In-depth research reports on the performance and control of DX VAV systems are limited in open literatures. In order to study in detail the system's performance and develop control methods for DX VAV systems, a multi-purpose experimental station of a DX VAV system with variable speed compressor and EEV and its complete dynamic simulation model have been developed and reported. A promising approach is investigated here to overcome the deterrent for the wide application of DX VAV systems, i.e. the difficulty to match the cooling capacity of refrigeration plant with the varying VAV thermal load.

2. CAPACITY CONTROL OF DX VAV SYSTEMS

Maintaining a constant supply air temperature is critical for a VAV A/C system to decouple or isolate its different control loops and to realize desirable zoning comfort control. If the supply air temperature changes continuously, zone air temperatures may never be stabilized. A zone air temperature sensor will not respond instantaneously to a change in supply air temperature. It will typically be several minutes (perhaps up to 20) before the VAV terminal unit damper is repositioned to alter the supply volume flow rate to the zone in a corrective manner. If the control of the air temperature downstream of a DX cooling coil is poor, the supply air temperature may probably have changed again during this period of time, and another cycle of zone and terminal unit responses will be initiated. Furthermore, such cyclic changes in supply volume flow rate will be accompanied by a cyclic variation in noise from the supply diffuser (Shepherd, 1999). It is obvious that the cooling capacity of a refrigeration plant must follow the varying VAV thermal load continuously and accurately in order to maintain a stable supply air temperature. Therefore, the essential problem for the wide application of DX VAV systems should be the capacity control of the refrigeration plant.

Capacity change for a chilled water cooling coil with three-port modulating valve can reach 0.1% of full capacity, and the ratio of maximum to minimum capacity can reach 20:1 to 50:1. Thus a stable control of supply air temperature is relatively easy to realize in a conventional VAV system with chilled water cooling coil. However, currently the capacity control for a DX A/C system with reciprocating compressors is typically by either cylinder unloading or hot gas bypass. A DX A/C system with scroll and screw-type compressors may mostly adopt methods of hot gas bypass, discharge or inlet throttling of the refrigerant flow. Medium-to-large sized DX A/C systems usually include multiple compressors to further realize staging capacity control, while a small sized DX A/C system usually makes use of frequent on/off cycling to modulate its cooling capacity, which would dramatically reduce the life and increase the noise level of the refrigeration plant. Apparently, purely staging capacity control or frequent on/off cycling will lead to the fluctuation of supply air temperature, and hot gas capacity control will result in energy inefficient operation because it imposes an artificial thermal load on the DX system.

Variable refrigerant volume (VRV) technology featured with a variable speed compressor and an electronic expansion valve (EEV) should be a promising approach to achieve precise cooling capacity matching while maximizing energy saving in a DX VAV system. The use of the variable speed compressor coupled with an EEV in a DX VAV system can modulate precisely the refrigerant flow rate and hence the system's cooling capacity, making the accurate matching between the cooling capacity and varying VAV thermal load possible. It was reported that through the modulation of compressor speed, the cooling capacity of a residential split-type DX A/C system could be varied ranging between 50 and 100% in proportion to the change in room temperature (Lida et al., 1982). Currently with the advancement of variable speed compressor and EEV technologies, inverter-aided A/C systems can vary their cooling capacity between 20% and 100% of the full load. Moreover, the use of variable speed compressor for capacity control may offer the potential for greater energy savings during part load operation. During part load operation, the condensing and evaporating pressures/temperature of a DX A/C system will respectively decrease and increase with a lower compressor speed, substantially increasing the system's COP. Yang and Lee (1991) presented an analysis of an inverter-driven variable speed air conditioning system used in a hot and humid region. The results indicated that it could provide an annual energy saving of 20%. It was reported in 1985 that an inverter-driven A/C system developed by Mitsubishi Electric Corporation in Japan could achieve 35% energy saving compared to a single speed compressor A/C system (Nakashima et al., 1985). Furthermore, it can be expected that the pull-down time needed for an A/C system to reach its air temperature setting during start-up can be reduced because a compressor can operate at its highest speed.

However, in the past variable speed compressors were generally regarded to be merely suitable for application in small-scale A/C systems, but not in medium- or large-scaled A/C systems due to the lack of insufficient development and component integration. Fortunately, in recent years medium- to large-scaled variable speed compressors have gained great improvement in various aspects and been widely used in medium-capacity multi-evaporator A/C systems, particularly in Japan (Youn et al., 2001). In order to demonstrate the benefits and feasibility of variable speed technology used in large-scale centrifugal central chiller systems, Lenarduzzi and Yap (1998) presented a demonstration installation of variable speed compressor in retrofitting a chilled water A/C system. The system was monitored for one cooling season and the results showed that variable speed drive technology could also work successfully in large-scale A/C systems. It was estimated that approximately 41% of energy saving has been achieved for this particular site and the power quality and total harmonic distortion problem induced could be neglected. From the above investigation, it can be concluded that VRV technology could be proven to be an energy efficient and practical way to realize capacity control in DX VAV systems and has many other excellent advantages such as soft start-up, rapid responding, simple system controls, easy maintenance, etc.

3. DESCRIPTION OF EXPERIMENTAL STATION

A multi-purpose DX VAV A/C experimental station has been set up in the HVAC Laboratory of Building Services Engineering (BSE) Department at the Hong Kong Polytechnic University. The schematics of the air handling system and DX refrigeration plant of the experimental station are shown in Figure 1 and Figure 2, respectively.

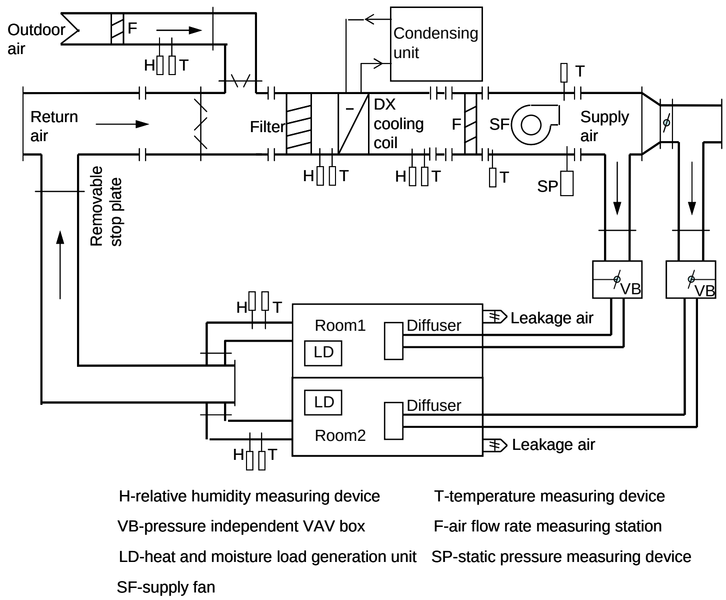

Figure 1: The schematic of the air handling system of the experimental station

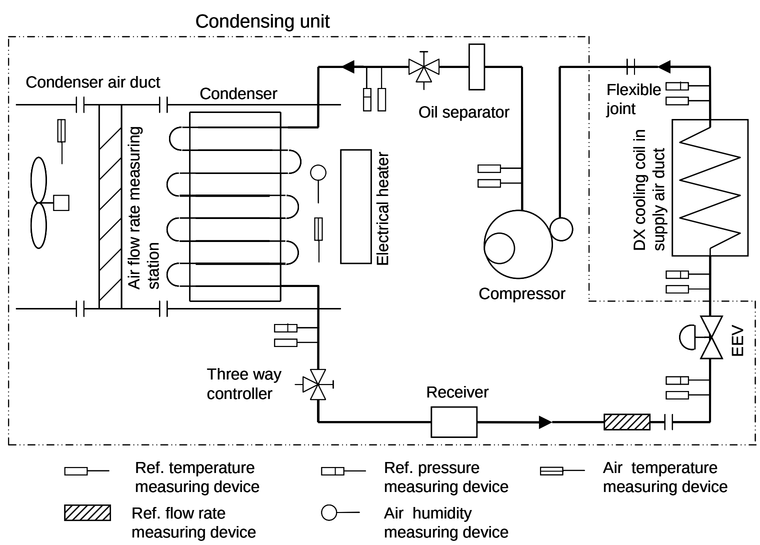

Figure 2: The schematic of the DX refrigerant plant of the experimental station

In the DX refrigeration plant, there is a condensing unit comprising of a rotor compressor driven by a variable speed drive (VFD), an air cooled condenser with its axial fan driven by a VFD. An EEV is used in the station. All other necessary accessories, such as an oil separator, a refrigerant receiver, etc., are provided to ensure the safe operation of such a DX system. The refrigerant used is R22. The nominal cooling capacity is 2.8 ton (9.9kW/90HZ) and its modulation range is 30 to120% of its normal capacity.

The air handling system includes an air handing unit (AHU), two simulated air conditioned rooms and two pressure-independent VAV terminals. Inside the AHU, the evaporator of the DX refrigeration plant is used as a DX air cooling coil. The supply fan in the AHU is of centrifugal type and VFD-driven. Its design supply air flow rate is 1200 cfm (0.57m^3/s). Two pressure-independent VAV terminals, which can feedback the damper opening signals and are controlled by direct digital control (DDC) technology, are installed in two branch supply air ducts to serve the two simulated rooms, respectively. The two identical simulated rooms of 3.7m (L) ×3.2m (W) ×2.7m (H) are separated by removable partition，so that a large simulated air conditioned room may become available when and as necessary. Inside each room is a sensible heat and moisture load generation unit (LGU). One unit is regulated with silicon control resistance (SCR) and the other with solid state relay (SSR). Sensible heat and moisture load can be varied manually or automatically with the pre-set pattern by operator's programming. In addition, for future expansion purpose, some flexible connections are reserved to facilitate future potential modification of the experimental station when required.

There are three air flow rate measuring stations (FRMS) in this experimental station for measuring the total supply air flow rate, the air flow rate passing through the condenser and outdoor air flow rate. The former two FRMSs include nozzles complying with ANSI/ASHRAE Standards 41.2, an air flow straightener and an air pressure differential measuring device with an accuracy of ±0.1% full scale. The FRMS for outdoor air includes a hot-film anemometer with an accuracy of ±0.1m/s. A refrigerant mass flow meter is available upstream the EEV with an accuracy of ±0.25% full scale. In order to ensure the measuring accuracy of air parameters, air-sampling devices for measuring dry-bulb and wet-bulb air temperatures are provided in the station. This will help minimize the influence of uneven distribution of air temperature and humidity inside the air duct. Furthermore, there are eight measuring points for refrigerant temperature and pressure. All the sensors for measuring refrigerant parameters are in direct contact with the refrigerant, which is particularly necessary for measuring transient behaviors of the parameters under investigation. All the sensors and transducers for measuring air and refrigerant parameters are of high quality, with fast response time. For example, all temperature sensors are of platinum resistance type with an accuracy of ±0.1% full scale (Pt 100/0℃-3W, Class A, SUSΦ3.2-150L). The instrumentation in the station is fully computerized, including a data acquisition system (DAS) and a desktop personal computer (PC). The sampling interval of the DAS for a complete cycle is no more than 0.2 second. All monitored parameters will be on real-time basis, curve-data displayed, recorded and processed. On the other hand, all controllers are of digital microprocessor-based PID type for the following operational parameters: the sensible heat and moisture load from the LGUs, the cooling capacity of the DX air conditioner, the refrigerant degree of superheat, condensing pressure or refrigerant sub-cooling, supply air static pressure, minimum outdoor air flow rate, VAV terminal box air flow rate and temperature of air entering the condenser. All setting of the controllers such as proportional or differential gains, integral times and set points can be reset. Furthermore, with the aid of the control software to be developed by operator's reprogramming, the PC can also act as a high-level central computerized controller to implement user-programmed to-be-developed control strategies and algorithms.

In addition to the research work on DX VAV systems, various related research work on the characteristics of individual component in a DX air conditioning system may also be carried out with this multi-purpose DX air conditioning experimental station. For example, how the configuration and geometric parameters, such as different fin types, fin pitches, fin thickness, affect a DX cooling coil's dehumidification ability may be studied. With the reserved expansion provision, this experimental station can be expected useful in studying into the operational and control characteristics of an air conditioning system having multiple DX evaporators through modeling and experimental work. Moreover, the station is sufficiently instrumented for studies related to the application of artificial intelligent control such as, neural network analysis, fuzzy control, system identification, etc. and to fault detection and diagnosis (FDD) for air conditioning systems.

4. DYNAMIC MODELING FOR A DX VAV SYSTEM

A complete dynamic simulation model representing a DX VAV system, based on thermodynamic principle and test data of the DX VAV experimental station, has been developed. The complete system's simulation model is component-based, and takes the dynamic characteristics of a DX refrigeration plant and a VAV air handling system into consideration simultaneously. The complete conceptual model which depicts the zoning of the DX VAV system is shown in Figure 3; each zone is treated as a stirred tank. The well-known A.C. Cleland Equations and air state equations recommended by ASHRAE are used to describe various thermodynamic and thermophysical properties of refrigerant and air, respectively.

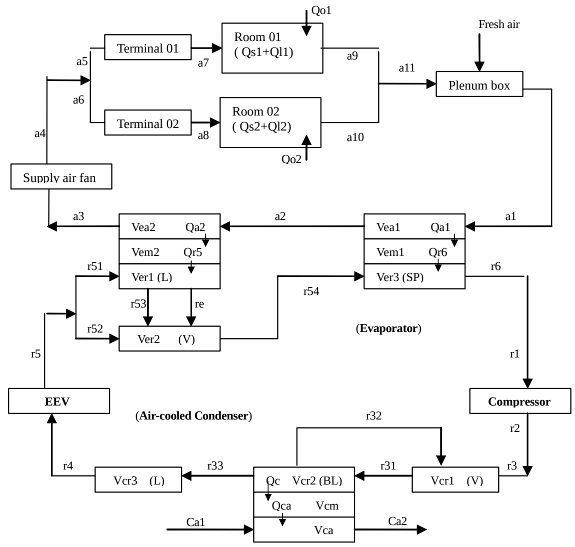

Figure 3: The conceptual model of the DX VAV system

4.1 DX Refrigeration Plant

4.1.1 Compressor and EEV: When modeling the variable-speed rotor compressor, a polytropic compression process is assumed and a quasi-steady model is established by the traditional theoretical thermodynamic approach. Volumetric and electric efficiencies are presented by empirical correlations. EEV is represented by an orifice equation, and a varying valve opening that was modulated by refrigerant superheat at evaporator exit. Its flow coefficient is determined using manufacturer's performance data with the aid of curve-fitting/regression analysis.

4.1.2 Condenser: The conceptual model of air-cooled plate-fin-tube condenser is as shown in Figure 3. Counter-flow heat exchange is assumed between the refrigerant and the air. Discharged vapor from the compressor is delivered into the vapor zone, *Vcr1*. *Vcr2* is the boundary layer where vapor from *Vcr1* is cooled down; some is condensed, and some cooled vapor will flow back to the vapor zone. The condensed liquid refrigerant is collected in zone Vcr3 in which liquid refrigerant is saturated or slightly sub-cooled. *Vca* is an assumed zone for cooling air that absorbs heat from the condensation of refrigerant. *Vcm* represents tube metal zone. Heat transfer between refrigerant and coolant is assumed to occur only in the boundary layer zone.

The mass and energy balance on vapor zone Vcr1 and liquid zone Vcr3 gives:

$$m_{r3} + m_{r32} - m_{r31} = \frac{d\left(\rho_{r31} V_{cr1}\right)}{dt} \tag{1}$$

$$m_{r3} h_{r3} + m_{r32} h_{r32} - m_{r31} h_{r31} = \frac{d\left(\rho_{r31} V_{cr1} h_{r31}\right)}{dt} \tag{2}$$

$$m_{r33} - m_{r4} = \frac{\rho_{lc} d\left(V_{cr3}\right)}{dt} \tag{3}$$

$$m_{r33} h_{r33} - m_{r4} h_{r4} = \frac{\rho_{lc} d\left(h_{lc} V_{cr3}\right)}{dt} \tag{4}$$

The energy balance equation for Vcm zone is:

$$Q_{cr} - Q_{ca} = M_{cm} c_{pcm} \frac{dT_{cm}}{dt} \tag{5}$$

The heat flow between refrigerant and metal is calculated by:

$$Q_{cr} = \alpha_{cr} A_{cr} \left(T_{lc} - T_{cm}\right) \tag{6}$$

where the heat transfer coefficient in refrigerant side is calculated by:

$$\alpha_{cr} = 0.555 r_s^{\frac{1}{4}} B_m \left(T_c - T_{cm}\right)^{-\frac{1}{4}} d_i^{-\frac{1}{4}} \tag{7}$$

Similar approach has been followed in calculating the heat flow between metal and cooling air, with its heat transfer coefficient in air side calculated by McQuistion Equation (McQuistion, 1978).

4.1.3 Evaporator: The conceptual model of the louvered-fin-tube evaporator is also included in Figure 3. The evaporator's refrigerant side is basically divided into two regions: a two-phase region and a superheated region. The former region is further divided into liquid and vapor zones. All heat transmitted into the two-phase region is assumed to be taken up by the liquid refrigerant and, therefore, heat transfer only occurs in the liquid zone. When the pressure in the evaporator decreases, some liquid will flash into vapor which flows to the vapor zone, this mass flow is denoted by m_{re} . This vapor mixes with the vapor refrigerant from the evaporator, as well as the vapor from the EEV in zone Ver2. The saturated vapor is then superheated in zone Ver3 on the suction side of the compressor. The similar approach described by Deng (2000) has been followed to model the refrigerant-side zones. The commonly-used Kandlikar equation (Kandlikar, 1987) and Petuknov-Popov equation were adopted here to calculate the refrigerant-side heat transfer coefficient in the two-phase zone and superheated zone respectively. Zivi model was used to calculate the refrigerant void fraction in two-phase zone:

$$\gamma = 1 \Big/ 1 + (1/x_r - 1)(\rho_v/\rho_l)^{0.666} \tag{8}$$

Models of refrigerant-side zones in evaporator will provide the refrigerant superheat temperature which is required by the compressor model, and will also provide the metal temperatures which are to be used in the air side models of evaporator for calculating the heat flow and moisture removal capacity.

When modeling the air side of the evaporator, the thermal capacity and mass storage of air is considered negligible and hence mass and energy balance between air and tube wall is regarded steady. In the air side of the superheated zone, *Vea1*, air dry-cooling will normally take place. While in the air side of the two-phase zone, *Vea2*, air wet-cooling will occur when the metal temperature is below the dew-point temperature of entering air; otherwise, air dry-cooling will occur. Under dry-cooling condition, the fin efficiency equation given in ARI Standard 410 was used and meanwhile the heat transfer coefficient is calculated in terms of Colburn factor correlations proposed by Wang *et al.*(1999). Under wet-cooling condition, Hong-Webb equation (Hong and Webb, 1996) which takes the influence of condensing water into consideration was applied to calculate the fin efficiency of evaporator. The sensible heat transfer coefficient for louvered-fin evaporator under wet-cooling condition is obtained from C. C. Wang correlation (Wang *et al*., 2000). The total heat transfer coefficient during dehumidification is equal to the sensible heat transfer coefficient multiplied with an amplifying factor ξ .

$$\xi = 1 + 2.46 \times (g_a - g_m)\big/(T_a - T_m) \tag{9}$$

The air process line of evaporator during dehumidifying can be determined by the state points of the entering air and the saturation air at the zone tube wall temperature. Once the air process line is determined, the moisture content of zone exiting air can be obtained with the aid of air state equations. The moisture removal capacity of evaporator, *MR*, can be evaluated by the following equation:

$$MR = m_{a2}(g_{a2} - g_{a3}) \tag{10}$$

4.2 VAV Air Handling System

4.2.1 VAV terminal: The air flow rate passing through a pressure-independent terminal is given as follows:

$$v_a = \sqrt{\frac{SP_{bi}}{R}} \tag{11}$$

Where, the flow resistance of the terminal damper is evaluated by:

$$R = \left(\rho_a / 2A_b^{\,2}\right)\left(K_d + K_{ds} - 1\right) \tag{12}$$

And the dynamic loss coefficient, K_d, could be calculated by:

$$K_d = K_{do}\left\{\frac{W_f}{\left[(1-\lambda)(1-\theta/\theta_c) + \lambda\right]^2} + \left(1 - W_f\right)\lambda^{[2(1-\theta/\theta_c)-2]}\right\} \tag{13}$$

The Proportional-plus-Integral (PI) control action of the damper and PI-controller is discretised and modeled by:

$$V_c = K_p V_e + \left(K_p / t_i\right)\left(S_{Ve}\right)(\delta t) \tag{14}$$

Where the error voltage $V_e = V_s - V_Q$; the output voltage of P-controller of the typical thermostat, which represents the desired air flow rate for the terminal, can be given by:

$$V_s = V_{\min} + (V_{\max} - V_{\min})[T_s - (T_{des} - P_B)/2]/P_B \qquad (15)$$

The feedback voltage from a linear differential pressure sensor, which represents the present air flow rate for the terminal, can be given by:

$$V_Q = B\Delta P_o + C \qquad (16)$$

In the model developed for the VAV terminal, the driving motor speed of the terminal damper is constant and equal to 0.25°/s. Hysteresis of actuator-damper linkage is 2.5° and is represented as in Hung's paper (Hung *et al*., 1999).

4.2.2 Air conditioned zone: The sensible heat balance in the air conditioned zone is given by:

$$\rho_a V_z c_{pa} \frac{dT_z}{dt} = L_f \rho_a c_{pa} v_{\max}(T_{des} - T_i) - \rho_a c_{pa} v_a (T_z - T_i) - \alpha_z A_z (T_z - T_{des}) \qquad (17)$$

The moisture content balance in the air conditioned zone is given by:

$$\rho_a V_z \frac{dg}{dt} = \rho_a v_a (g_i - g_o) + \omega \qquad (18)$$

The dynamics of the temperature sensed by the thermistor of the zone's thermostat is modeled by:

$$\frac{dT_s}{dt} = \frac{1}{t_s}(T_z - T_s) \qquad (19)$$

4.2.3 Supply fan: With the aid of the first fan law and least-square curve-fitting, the fan characteristics at different speed are developed from manufacturer's performance data at its rated speed and expressed by a set of third-order polynomial correlations. The dynamics of the fan-motor is represented by a first-order differential equation with a specific time constant.

5. CONCLUSION

The use of VRV technology should be an energy efficient and practical way to realize accurate capacity control in DX VAV systems to ensure a stable supply air temperature. The multi-purpose experimental station developed and reported here is expected to become very useful in studying DX VAV systems. The overall concept for dynamic modeling of a DX VAV system is presented here and the system's model developed will help develop control methods for a DX VAV system. Simulation results and related validation work will be reported in the future papers.

NOMENCLATURE

A	area	(m^2)
A_b	cross-sectional area at terminal inlet	(m^2)
B	slope of differential pressure sensor	(v/Pa)
B_m	refrigerant property parameter	(-)
c_p	constant pressure specific heat	(J/kg.k)
g	moisture content	(kg/kg)
h	enthalpy	(J/kg)
K_p	proportional gain of damper controller	(V/^{o}C)
L_f	load factor	(-)
m	mass flow rate	(kg/s)
M	mass	(kg)
P_B	proportional band of thermostat	(^{o}C)
Q	heat transfer rate	(W)
SP_{bi}	static pressure of upstream damper	(Pa)
S_{Ve}	summation of error voltage over time	(V)
t_i	integral time	(s)
V	volume	(m^3)
$V_{\max}$	maximum output voltage set	(V)
$V_{\min}$	minimum output voltage set	(V)
x	dryness fraction	(-)
θ	damper angel	o
δt	time step	(s)
α	heat transfer coefficient	(W/m^2K)
γ	void fraction	(-)
ρ	density	(kg/m^3)
η_f	fin efficiency	(-)
ω	moisture generation rate	(kg/s)
ΔP	pressure difference	(Pa)

Subscripts

a	air
c	condenser

T_{des}	Set point of zone air temperature	(^{o}C)	l	liquid phase
T_s	Sensed zone air temperature	(^{o}C)	m	metal
v	Air flow rate	(m^3/s)	r	refrigerant

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