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Simulation study on a three-evaporator air conditioning system for improved humidity control

Huaxia YAN[a], Shiming Deng[a,*]

[a]*Department of Building Services Engineering, The Hong Kong Polytechnic University, Kowloon, Hong Kong SAR, China*

Abstract

Multi-evaporator air conditioning (MEAC) may be regarded as one of the energy conscious air conditioning technologies. However, most MEAC systems currently focus on temperature control only, so that the potential of MEAC technology in achieving energy saving has not yet been fully explored. Therefore, in this paper, based on previous extensive related research by the investigators on modeling and controlling both single evaporator air conditioning (SEAC) systems and MEAC systems, a novel controller for a three evaporator air conditioning system will be developed. The novel controller was developed by integrating two previous control algorithms for a dual-evaporator air conditioning system for temperature control and for a SEAC system for improved indoor humidity control. Simulative controllability test results showed that, with the novel controller, improved control over indoor humidity levels could be obtained.

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Key words: TEAC system, simulation, humidity control, controllibility tests

1. Introduction

Multi-evaporator air conditioning (MEAC) systems featuring variable refrigerant flow, with a three-evaporator air conditioning (TEAC) system being a typical example, have recently become attractive. This is because MEAC based air conditioning (A/C) has a number of advantages such as installation convenience, high design flexibility and higher energy efficiency [1]. Using an MEAC system to correctly control both indoor air temperature and humidity is a difficult task. This is because in an MEAC system, a number of indoor units (IUs), possibly of different cooling capacities, are connected to a common condensing unit. Hence, changes in the operating condition of one particular IU are very much likely to influence the conditions of other IUs. On the other hand, particularly in buildings located in hot

* Corresponding author. Tel.: +852 27665859; fax: +852 27657198.
E-mail address: besmd@polyu.edu.hk.

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and humid subtropics such as Hong Kong, controlling indoor humidity at an appropriate level is critically important since this directly affects building occupant thermal comfort, indoor air quality and the operating efficiency of the building A/C installations [2].

Developments in capacity controllers for MEAC systems have also been reported, for controlling indoor air dry-bulb temperature [3-7], evaporating temperature [6, 8 and 9] or refrigerant degree of superheat (DS) [8-10]. The control algorithms developed for MEAC systems mentioned above are mainly based on the system simulation without experimental validation. One exception is the experimental work by Xu et al. [11], where a novel control algorithm for a dual-evaporator air conditioning system, which imitated On-Off control of a compressor in a single-evaporator air conditioning (SEAC) system, was developed for temperature control. Controllability tests validated the control algorithm in term of control accuracy and robustness. However, indoor air humidity control was not considered in the study. This leads to the advantages of MEAC systems not being fully realized, as higher indoor humidity impacts negatively on indoor thermal comfort and energy efficiency. A High-Low control strategy was proposed by Xu et al. [12] for an SEAC system to enable both compressor and supply air fan to operate at high speeds when the indoor air dry-bulb temperature setting was not satisfied and at low speeds otherwise. Indoor air humidity can be directly controlled under the simple controller.

Although MEAC systems are being extensively used and related experimental and simulation research work has been carried out, current efforts have focused on controlling indoor air temperature only without paying attention to indoor humidity control. In this paper, therefore, the development of such a novel capacity controller for a TEAC system for improved indoor humidity control, is reported.

Nomenclature		**Abbreviation**	
E	control signal	A/C	air conditioning
m_r	refrigerant mass flow rate, kg/s	DS	degree of superheat, ºC
n	compressor speed, rpm	EEV	electronic expansion valve
t	time, s	IS	indoor space
T	indoor air dry-bulb temperature, ºC	IU	indoor unit
ΔT	dead-band for temperature control, ºC	MEAC	multi-evaporator air conditioning
x_i	the ratio of refrigerant mass flow rate at a L-period to that at an H-period	RH	relative humidity, %
Subscripts		SEAC	single-evaporator air conditioning
i	i^{th} $(i = 1, 2, 3)$ indoor unit	TEAC	three-evaporator air conditioning
set	setting point		

2. Control algorithm for improved indoor humidity control

The novel controller development will be built on the two previous developed controllers [11 and 12]. Similar to the High-Low controller previously developed [12], the novel capacity controller reported in this chapter was not directly to control indoor air humidity, but to address two issues causing possible high indoor air humidity for an On-Off temperature only controlled TEAC system. These were the absence of dehumidification and the re-evaporation of condensate on the surface of an IU while its matching fan may still operate during an Off-period when the indoor air temperature setting was satisfied and the refrigerant supply to the IU was completely stopped. This was done by supplying less refrigerant, instead of completely stopping its supply, and at the same time, running the supply fan at a lower speed. Consequently, improved humidity control, as compared to that under On-Off temperature only control, can be achieved.

In a TEAC system, for the i^{th} $(i = 1, 2, 3)$ IU serving the i^{th} indoor space (IS), with its indoor air temperature setting, T_{set_i}, a control signal at a time point t, $E_i(t)$, can be defined as:,

$$\begin{array}{ll} If\ \ T_i(t) \geq T_{set\_i} + \Delta T_i & E_i(t) = 1 \\ If\ \ T_{set\_i} - \Delta T_i < T_i(t) < T_{set\_i} + \Delta T_i & E_i(t) = E_i(t-1) \\ If\ \ T_i(t) \leq T_{set\_i} - \Delta T_i & E_i(t) = 0 \end{array} \quad (1)$$

where $T_i(t)$ is the actual space air temperature at t time point, $t-1$ the last time point and ΔT_i the temperature control dead-band.

The period when $E_i(t)=1$ was referred to as an H (high)-period when the compressor and fan are operated at higher speeds and that when $E_i(t)=0$, L (low)-period at lower speeds. At an H-period, the supply refrigerant mass flow rate to the i^{th} IU was m_{ri}, or the design refrigerant mass flow rate. At a L-period, instead of stopping supplying refrigerant to the IU, less refrigerant was supplied at $x_i \times m_{ri}\ (0 < x_i < 1)$, where $x_i\ (i=1,2,3)$ is the ratio of refrigerant mass flow rate at a L-period to that at an H-period.

The refrigerant mass flow rate entering an evaporator or an IU was controlled by its matching electronic expansion valve (EEV). When $E_i(t)=1$, the EEV functioned to control the refrigerant DS and the supply fan operated at its full speed. On the other hand, when $E_i(t)=0$, the EEV functioned as a modulating valve to reduce the refrigerant mass flow rate to supply less refrigerant to an IU. At the same time, the supply fan speed was also reduced, resulting in a smaller supply air flow rate passing through the IU. When the indoor air temperature was within the dead-band, as represented by Equation (1), the operational status of both EEV and the supply fan at time t were the same as they were at time $t-1$. Consequently, totally there were eight different operating IU combinations for a TEAC system at any time, evaluated by: $\sum_{i=0}^{3} C_3^i = 8$. For example, if IU 1 and IU 2 were operated at H-periods and IU 3 at a L-period, then EEV 1 and EEV 2 functioned to control the DS, and the refrigerant mass flow rates to IU1 and IU1 were m_{r1} and m_{r2}. EEV 3, however, functioned as a modulation valve to reduce m_{r3} to $x_3 \times m_{r3}\ (0 < x_3 < 1)$, and the compressor speed was determined by $n = f(m_{r1} + m_{r2} + x_3 \times m_{r3})$.

In practice, indoor fan speed cannot be set too low to adversely affect the indoor air flow pattern. Since the impact of varying fan speed on the total output cooling capacity of an IU was much less significant than that of varying compressor speed [14], only a 60% fan speed reduction was utilized in this study, i.e., at a L-period, an indoor fan was operated at 60% of the fan speed at an H-period.

On the other hand, to determine the exact extent of refrigerant flow reduction at a L-period, or the exact value of $x_i\ (i=1,2,3)$, there were two points of consideration, avoiding space overcooling, and providing a non-stop dehumidification ability without requiring a precise control over indoor air humidity. In the current study, values of $x_i = 0.5\ \ (i=1,2,3)$ were adopted.

Simulative tests using the experimentally validated dynamic mathematical TEAC model were carried out to ascertain the controller's correctness and robustness prior to its experimental validation. The results of simulation controllability tests are reported following.

3. Controllability tests results and discussions

A dynamic mathematical model for an experimental TEAC was developed and experimentally validated [13]. The model was made of sub-models for major system components, such as compressor and evaporators, etc. The experimental validation suggested that the model could simulate a real TEAC system in both transient and steady-state operations, with a prediction error of not greater than 6%.

Table 1 Details of the two simulation-based controllability tests

Test	IS	Dimension of each IS (m)	Indoor air T_{set} (°C)	sensible load (W)	latent load (W)
1	1	1.5×3.7×3.3	26	1150	450
	2	1.85×3.7×3.3	26	1250	650
	3	1.85×3.7×3.3	26	1350	550
2	1	1.5×3.7×3.3	27	step change from 1300 to 1400 W at t = 2400 s	step change from 500 to 700 W at t = 2400 s
	2	1.85×3.7×3.3	26	1300	500
	3	1.85×3.7×3.3	25	1300	500

Using the validated TEAC model, extensive simulation-based controllability tests were carried out and two examples of test results are given in this section. The testing conditions for the two tests are shown in Table 1. In order to prevent frequent speed change of the compressor, a dead-band of ±0.35 ºC was utilized for indoor air temperature control in all the ISs. The air dry-bulb temperature in the outdoor space was maintained at 35 °C. In all the tests, indoor air RH was not controlled directly but allowed to fluctuate, depending on the actual latent output cooling capacity from each IU.

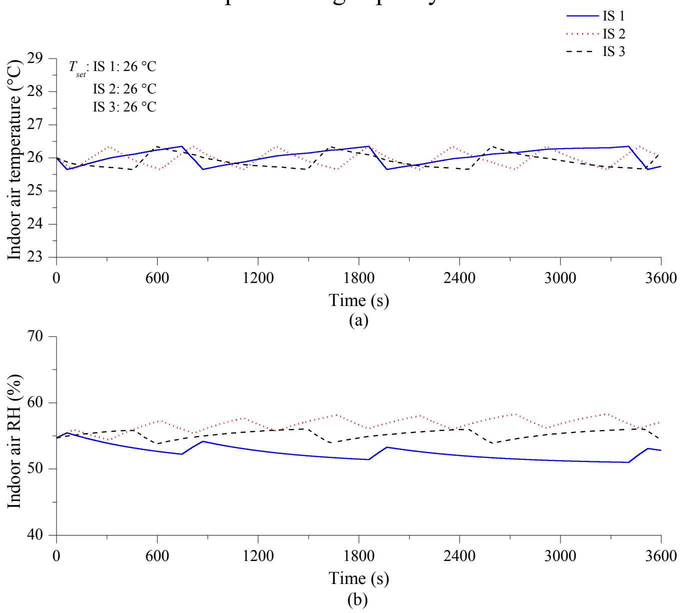

Fig. 1 Variations of the simulated indoor air temperature and RH in all ISs (Test 1)

Fig. 1 shows the simulation results of controllability Test 1. As seen in Fig. 1a, indoor air temperatures in all ISs fluctuated around the same temperature setting point (26°C) in a repeatable pattern. However, the variation patterns for indoor air temperature in the three ISs were different, because of the different indoor sensible and latent cooling loads. For example, indoor air temperature in IS 1, T_1, reached its low boundary of the ±0.35°C dead-band earlier than that in IS 2 and IS 3. When T_1 reached its low boundary of the dead-band, i.e., $T_1 < T_{set\_1} - \Delta T_1$, the speed of the supply fan in IU 1 was reduced to run at a L-period and compressor speed was also reduced to supply less refrigerant to IU 1.

It should be mentioned that with the novel controller, the matching of the output cooling capacity of an IU in a TEAC system to the cooling load in an IS was not realized by accurately controlling the refrigerant mass flow rate at a specific value, but rather by the different operating durations at H and L periods. While indoor air dry-bulb temperature was controlled within a dead-band, indoor air humidity was indirectly controlled without a specific setting point. Therefore, with the enhanced dehumidification ability due to low speed operation, in general, a lower indoor RH can be resulted in. For instance, the latent load in IS 2 was much higher than that in other ISs, its RH was only slightly higher than those in the other two ISs.

Fig. 2 shows the simulation results of controllability Test 2 where there were step increases in space sensible and latent loads at t = 2400 s in IS 1, as detailed in Table 1. As seen in Fig. 2, air temperatures in all spaces were stabilized at their respective set points, within ±0.35 °C before the change introduced at t = 2400 s. In IS 1, where the step increases in thermal loads were introduced at t = 2400 s, the controller responded by operating the IU 1 at a longer H-period. This, together with an increase in the latent load in IS 1, led to an increase in the indoor air RH in IS 1. However, the RH levels in the other two ISs were slightly reduced.

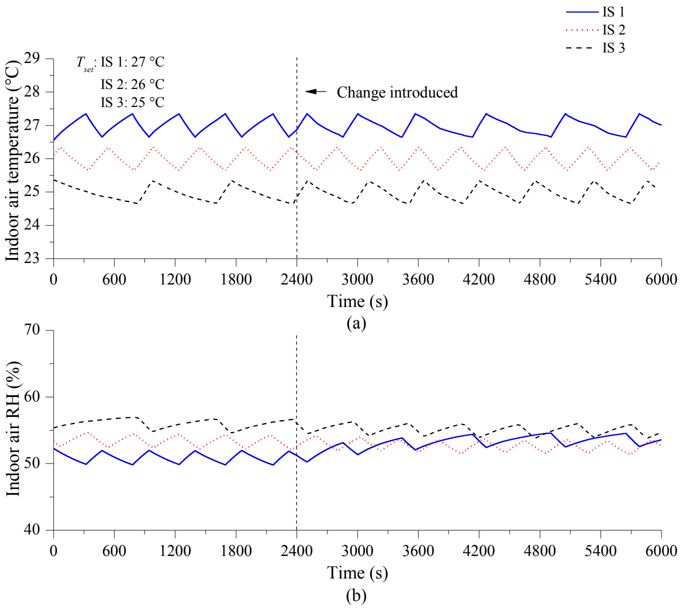

Fig. 2 Variations of the simulated indoor air temperature and RH in all ISs (Test 2)

Fig. 2b shows the simulated results of indoor air RH in the three ISs when the TEAC system was controlled by the novel controller. It can be seen that the RH fluctuation ranges in all the three ISs were small, at around 1.6%, 1.3% and 0.9%, respectively. The indoor air RH level in all ISs was increased at their respective H-periods but reduced at their respective L-periods, suggesting that the dehumidifying ability of the TEAC system during L-periods was better than that during H-periods.

The examples of simulative controllability test results shown in Figs. 1 and 2 clearly demonstrated that the novel controller for the TEAC system for improved indoor humidity control appeared operational, being able to maintain the required indoor air temperature and stable indoor RH.

4. Conclusions

The development of a novel capacity controller for a TEAC system for improved indoor humidity control is simulated and reported in this paper. Two examples of simulative controllability test results clearly demonstrated that the novel controller for the TEAC system for improved indoor humidity control appeared operational, being able to maintain the required indoor air temperature and stable indoor RH. However, further experimental evidences were desired, and the experimental controllability test will be conducted in the future.

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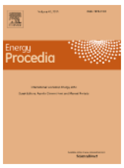

Biography

Prof. Deng Shiming is currently a professor and associate head of the Department of Buildin Services Engineering at The Hong Kong Polytechnic University. He obtained his PhD from South Bank Polytechnic, London in 1991 and His research focuses on the modelling and control of refrigeration and air conditioning systems.