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ABSTRACT

Multiple-split air conditioning (MSAC) system has become an attractive form of air-conditioning system for small-sized commercial and residential buildings. For a common room air conditioner, the only factor affecting the performance under a fixed outdoor and indoor condition is load rate of the indoor unit. But for the MSAC system, the indoor units will have quite different working situations under the same system load rate. It has been found that when the indoor and outdoor conditions (temperature and relative humidity) keep constant, the system COP will be within a "performance domain" related to the system load rate and the indoor load unevenness index. In this paper, the part load performances of MSAC systems are studied. As a result, it is found that whether the conditions changed, the performance domain will always exist. Through further analysis, it is pointed out that the shape of performance domain is related to the conditions, and the quantitative results are presented. This study is a theoretical foundation of design optimization, energy consumption prediction and evaluation in MSAC systems.

1. INTRODUCTION

MSAC Systems is a kind of direct-expansion air-conditioning system consisting of several outdoor units (usually include one variable capacity unit and several constant capacity units), many indoor units and a lot of connecting pipes. Due to its many merits, such as the arbitrary control of every indoor unit, flexible choice of the indoor unit type, easy expansion of system, and no requirement for dedicated room to install the outdoor units, it has become more and more attractive as a form of the central air-conditioning system in the middle and large commercial and residential buildings (Hiromune, 2004).

In order to develop high performance MSAC systems, many factors were studied (Hu and Yang, 2005), which included system configuration, compressor type, and so on. Besides them, the part load performance always is a key point of MSAC system research. Hu and Yang (2005) investigated a MSAC system which had five indoor units and one outdoor unit with a digital scroll compressor. They reported the relationship between COP and LR (Load Rate of system) and presented a comparison between the inverter technology and digital scroll technology. Takashi and Hiromi (2005) studied the energy saving characteristic of MSAC system based on manufacture catalogues and pointed out that the more the parallel outdoor units are, the smaller the change of COP with LR is.

Unfortunately, all the part load researches on MSAC system only pay attention to the influence of system LR on COP and were not aware of the effect of the indoor load distribution unevenness on the performance of the system. Actually, unlike the common room air-conditioner, the indoor units will have quite different working situations under a same system LR: one of the extreme situations is some indoor units running at full capacity and the others are down; the other extreme situation is all the indoor units running at a same part load rate. According to the common sense, the later situation will have a higher COP than the former one due to a larger effective heat exchange area, which will increase the evaporating temperature of the system. The preliminary researches also confirm it. Park et al. (2001) researched a MSAC system with two indoor units, and the system performance was analyzed with variations of operating frequency of the compressor, cooling load and the cooling load fraction between rooms. As a result, it pointed out that the increment of the load difference between rooms, caused a reduction of the system performance, although the total cooling capacity was constant. Xia (2002, 2004, and 2005) investigated a three-pipe heat recovery MSAC system consisting of an outdoor unit and five indoor units for the field operation performance. Keeping indoor and outdoor temperature (tin and tout) fixed, he measured the COP of the MSAC system in different load rate under cooling only condition (Figure1). It was found that the COP varied within a large range for a constant LR. He also pointed out that the actual running performance was directly related with the control method of the MSAC system. In summary, there are previous studies concerning the load distribution of the indoor units, and people have concerned that when the total cooling capacity was constant, with the variance of load difference between each room, COP will have different values. However, there is no method or index to describe such kind of load distribution, and no analysis of load distribution of indoor units on the performance of MSAC system quantitatively.

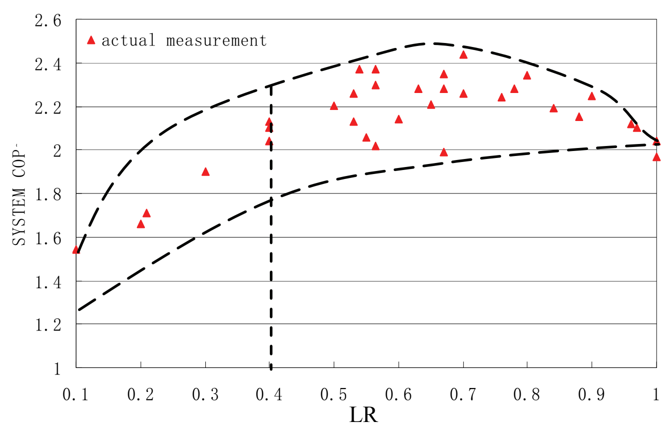


Figure 1. Laboratory test results (tin=23℃, tout=30℃)

Being aware of the effects of the different operation mode of the indoor units on the general performance of the MSAC system, the national MSAC standards of China and Japan have tried to consider it in the performance calibration. In the performance testing, both of the standards require some of the indoor units in full load mode and the others down. According to the previous analysis, the test results will be close to the worst satiation and not present the real working condition of the MSACs.

To sum up, the current researches of the influence of the load unevenness on MSAC performance is still blank or fragmented. A systematic description and research on it is quite necessary and urgent for a more deeply understanding, optimal design and control.

Therefore, an index to describe the indoor load unevenness is proposed in this paper. A typical MSAC system consisting of an outdoor unit and four indoor units is studied numerically. The influence of the indoor load unevenness on the part load performance is analyzed.

2. DESCRIPTION OF INDOOR LOAD UNEVENNESS

For a MSAC system with multiple indoor units, the load unevenness of indoor units will directly determine the utilization degree of the areas of the heat exchangers and finally influence the performance of the system. This is the essential difference between the MSAC system and the common air conditioner. It can be found that besides the control method, the factors affecting the part load performance of MSAC system mainly include: t_{out}, RH_{out} (outdoor relative humidity), t_{in}, RH_{in} (indoor relative humidity) and LR_i (rate of the cooling capacity of the i_{th} indoor unit and its rated cooling capacity, its range is 0~100%). For a MSAC system with N indoor units, the part load performance should be a function with N+4 variables. Furthermore, for a fixed MSAC system in cooling mode, when the indoor condition is fixed, the performance of MSAC system is determined by the outdoor temperature tout and the load rate of every indoor unit LR_i, i.e., N+1 variables (No dew or frost on the outdoor unit under cooling conditions, so the RHout has little effect on the performance and is neglected). The more indoor units the MSAC system has, the more complex the description of the performance is.

Actually, if the location of the indoor units is comparatively centralized or the connecting pipes have little influence on the performance of the MSAC system, the indoor units with same nominal cooling capacity will be replaceable with each other. Under this situation, it's easy to understand, if all the indoor units have same nominal capacities, the load distribution of the indoor units can be rearranged and expressed by average and square deviation, so the N LR_i can be refined to two parameters, system LR and square deviation. For the MSAC system with different capacity indoor units, besides the system LR, a square deviation with weighting efficient should be used to describe the unevenness of load distribution of the indoor units, which is defined as Unevenness Index (UI).

The variance of a group of numbers can greatly show the fluctuation of the numbers, this paper use the variance of the load rates of indoor units to express the load unevenness as bellow:

$$UI = \frac{\sum_{i=1}^{N} r_i^2 (LR_i - LR)^2}{N \cdot LR^2} / \max\left(\frac{\sum_{i=1}^{N} r_i^2 (LR_i - LR)^2}{N \cdot LR^2}\right) \quad (0 \le UI < 1) \tag{1}$$

When $LR_s \neq 0$ and $LR_{i(i \neq s)} = 0$, the function $f(LR_i)$ reaches its maximum value:

$$\max\left(\frac{\sum_{i=1}^{N} r_i^2 (LR_i - LR)^2}{N \cdot LR^2}\right) = \frac{1}{N} \cdot \left(\sum_{i=1}^{N} r_i^2 + 1 - 2r_s\right) \tag{2}$$

where, r_i is the proportion of the rated capacity of the i_{th} indoor unit in the total rated capacities; s represents the indoor unit with the smallest rated capacity and the $\max\left(\frac{\sum_{i=1}^{N} r_i^2 (LR_i - LR)^2}{N \cdot LR^2}\right)$ can be obtained when only the smallest capacity indoor unit is in full capacity operation.

In this way, three parameters, tout, system LR and UI, can completely describe the performance of the MSAC system with multiple indoor units under any situation. This idea has been validated by the numerical simulation results in section 3.3.

3. PERFORMANCE DOMAIN OF MSAC SYSTEM

3.1 Object Investigated

A MSAC system consisting of one outdoor unit with an AC inverter compressor and four indoor units is taken as the object to be investigated (Figure 2). The fundamental parameters of the system are specified in Table 1.

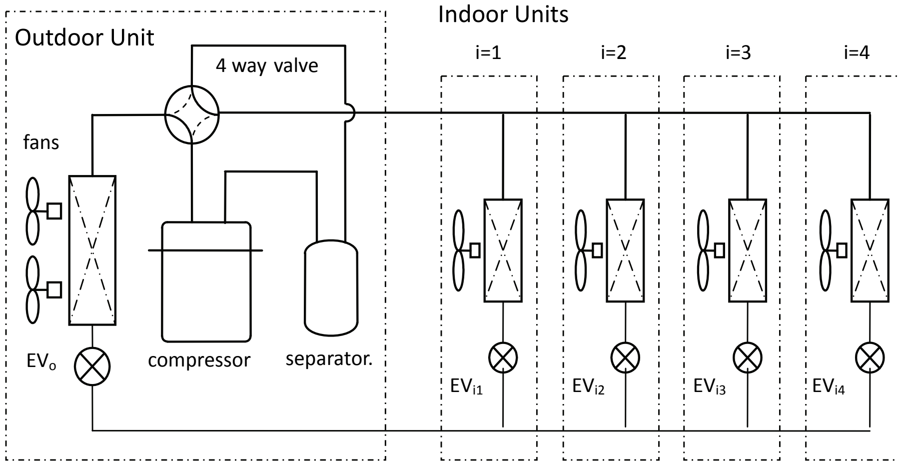


Figure 2. Schematic diagram of the MSAC system

Table 1： Fundamental parameters of the MSAC system

System description	one outdoor unit, four indoor units (same capacity)					
Rated performance	Cooling	Cooling capacity=28 kW (*f*=90HZ)				
Compressor	Swept volume: 39.0cc/rev; Frequency: 30~110 Hz; AC					
EEV	Outdoor unit	Quick open type; number: 1; Diameter: 3.2mm				
	Indoor unit	Quick open type; number: 1; Diameter: 1.8mm				
Heat exchangers	Items	Outdoor unit	Indoor unit	Items	Outdoor unit	Indoor unit
	Pipe style	rifled; copper	rifled; copper	Type of fins	Hydrophilic; Smooth; aluminum	Hydrophilic; louvered; aluminum
	Input air	10000m^3/h	1000m^3/h	Fan power	600W	--
	Refrigerant	R410A	Refrigerant			

To simplify the problem, the following assumptions had been used in this paper: (1) Ignore the effect of the refrigerant connecting pipe on the unit performance, and connecting piping resistance was not taken into account in the process of calculation; (2) Only one outdoor unit module, which included a condenser and an inverter compressor; (3) The outdoor unit fan power consumption was fixed, while the indoor unit fan power consumption was ignored.

The control method affects the system performance of MSAC directly. The control mode in this research is continuous mode and the control methods are as follows: (1) Indoor units: the indoor EEV is controlled by tin and the minimum superheat; the superheat of the evaporator outlet is controlled above 0℃; (2) Outdoor units: the outdoor EEV should be totally open, the T_{sc} of condenser outlet is controlled as 10℃ by the rotational speed of fan, and the compressor frequency f is controlled by T_{sh}.

3.2 Research toolkit and method

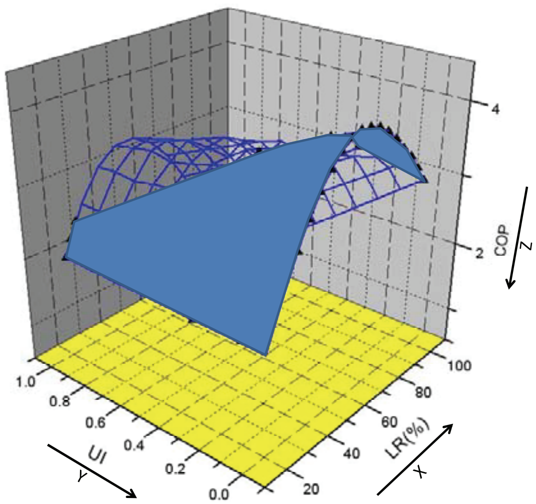
Based on the analysis of the structure of the MSAC refrigeration system, the authors (Shi, 2003) proposed a general physical model of gas-liquid two phase fluid network to describe the MSAC system. Furthermore, the authors (Shi *et al.*, 2008) proposed a thermodynamic model of the complex refrigeration system including node conservation equation, branch conservation equation, and system quality conservation equation and solving by an iterative method.

Based on this model, the influence of indoor load ratio and the load unevenness index on the COP is investigated.

3.3 Performance domain of MSAC system

Figure3 shows the relationship of COP, LR and the UI. As seen from Figure3, system COP is not only a function of LR but also obviously related to the UI. Therefore, looking along the minus Y coordinate of Figure 3, system COP doesn't locate in a line but a domain, which is named as "performance domain" of MSAC system.

Figure 4 is a vertical view of Figure3 along the Z direction, and it shows the effect of LR and UI on the COP. When UI keeps constant, COP increases firstly and then decreases with LR increasing; whereas, COP decreases when UI increase with LR constant. Figure 4 also shows the relationship between LR and the UI: when LR is fixed, there is a range $0 \leq UI \leq X$ ($X \leq 1$) for UI.



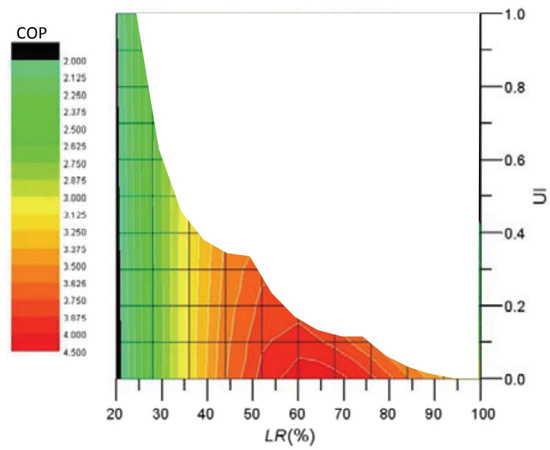


Figure3. Relationship of COP, LR and the UI Figure 4. Relationship of LR and the UI

3.3.1 Relationship between COP and LR with UI constant

Four different UI values were taken along the Y axis direction in Figure3, correspond to the different number of indoor operating unit J (J = 1 ~ 4, and LRi of each indoor unit has synchronal change), when J = 1, UI = 1; when J = 2, UI = 0.33; when J = 3, UI = 0.11; when J = 4, UI = 0. The change trend of COP with the LR was shown in Figure5. It can be seen from the simulation results of this diagram that COP changes as a parabola with LR increasing and the distribution of COP is a "performance domain" with LR and UI changing.

3.3.2 The relationship between COP and UI with LR constant

LR is fixed and the combination of indoor load ratio varied randomly, the relationship between COP and UI can be obtained. Figure6 (a) shows that when LR =50%, if UI = 0.33 (the maximum value of UI occurs when LR =50%, which can be found in Figure5), COP will be the lowest, but if UI =0, COP will reach its greatest value. When LR is 70%, the same trend can be seen in Figure6 (b). In other words, UI will directly determine the system COP when other parameters are the same. Therefore, UI is an important parameter which should be included for the analysis of system performance.

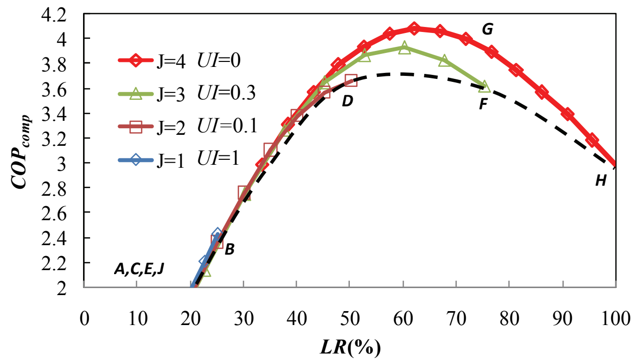
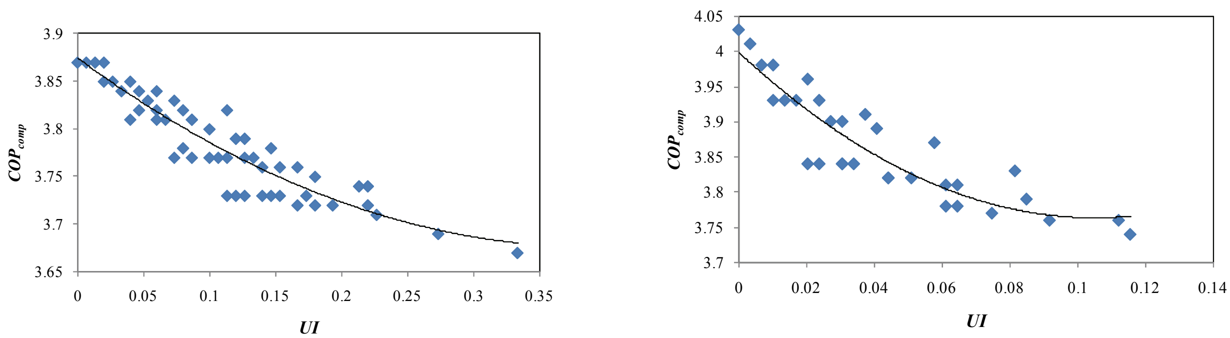


Figure5. Relationship between COP and LR when UI is known



(a) LR = 50% (b) LR = 70%

Figure 6. Relationship between COP and UI when LR is known

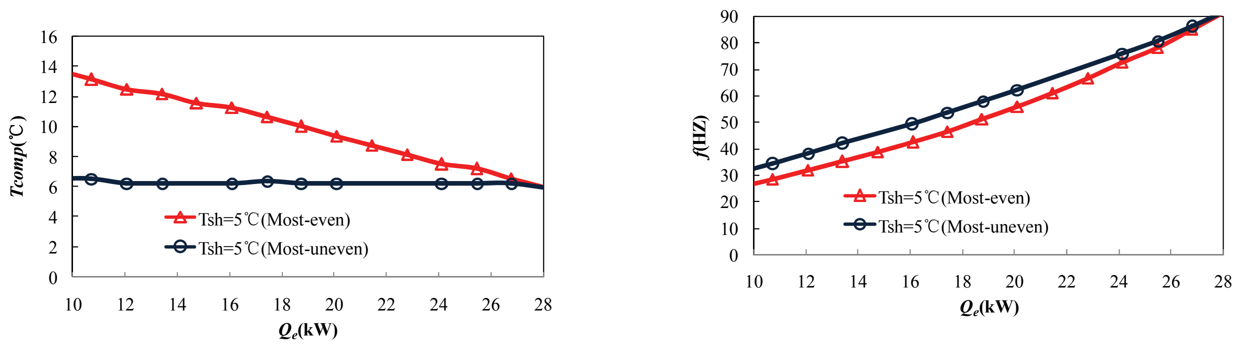
3.3.3 Analysis

Actually, the effects of UI on the system COP can also be identified by the variance of system operating parameters, such as Tcomp (the saturation temperature corresponding to suction pressure of compressor) and f (frequency of compressor). Figure7 shows the difference of Tcomp and f between the most even and the most uneven operating conditions, which can be analyzed in the following ways:

a) Under the most even operating condition, LR1= LR2=…= LRN, all LR i of the indoor units are same, UI=0. It means that all indoor loads change simultaneously (proportional rise or fall according to the rated capacity of each unit), and the heat exchanging area is most efficiently used. In this case, Tcomp is the highest and also the system COP. However, when LR keeps constant and UI varied, the utilizing ratio of the heat exchanging area under this condition is smaller compared with the best operating condition; Tcomp will be lower as well as COP, shown in Figure7 (a).

b) The influence of f on COP will be obviously weakened with the increasing LR (Figure7 (b)).

Therefore, the COP difference between the best and the worst operating conditions will be mainly determined by T_{comp}.



(a) Tcomp-Qe (b) f-Qe

Figure 7. Parameters comparison between the most-even and the most-uneven operating conditions

4. CONCLUSIONS

The load distribution of indoor units has large influence on the performance of MSAC systems. The method describing the indoor load distribution and the effect of the distribution on the part load performance of MSAC systems is important in the evaluation and improvement of control strategy in MSAC systems. Based on the mathematical model of complex refrigeration system, the effect of indoor load distribution on the part load performance of MSAC systems is investigated. The following conclusions can be drawn:

(1) The indoor load unevenness index (UI) and LR can be used to describe the load distribution of MSAC systems, and the part load performance of MSAC systems can be expressed as a function of tout, RHout, tin, RHin, LR and UI.

(2) When the continuous control method is applied, the system COP under cooling mode will be within a "performance domain" related to tout, LR, and UI. If LR and other environmental factors keep constant, COP will decrease with the increasing UI.

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