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Cooling performance of direct expansion system using multiple stages and adjustable heat exchange areas for fresh air handling

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Abstract. Providing fresh air to rooms can ensure indoor air quality. The direct expansion fresh air handling units are widely used, and most of them are single-stage treatment units, which are less energy-efficient under variable operating conditions. To overcome the shortcomings of existing systems, a direct expansion system using multiple stages and adjustable heat exchange areas for fresh air handling is proposed in this study. The fresh air is handled by coils stage by stage to achieve high efficiency, and the dampers of the idle coils are opened when some of the coils are working, which results in the performance improvement under partial load conditions. The simulation model is established, and operating conditions at load ratio of 95.70%, 45.38%, 31.76% and 0 are selected as typical operating conditions. The proposed system performances under different conditions are compared with traditional system. The results show that: (1) For four conditions, as the load rate decreases, the proposed system can save energy by 6.2%, 26.1%, 46.5%, and 25.0% compared to the traditional system respectively; (2) The proposed system can bypass the filter when the outdoor air is good, which can further save energy by 1.3%, 1.8%, 3.4%, and 25.9% under four conditions, respectively.

1 Introduction

Providing fresh air to rooms can ensure indoor air quality. However, the proportion of fresh air load in the air conditioning system load can reach 30%~50% [1]. Compared with the chilled and hot water fresh air system, the direct expansion fresh air system presents a simple structure, and can reduce the heat exchange loop in the system.

Market products are mainly fresh-air direct-expansion single-stage treatment units, which are less energy-efficient under variable operating conditions due to variations in load rates [2]. In some researches, multi-stage coils with different evaporation temperatures or condensation temperatures are used to treat fresh air can better meet air treatment needs and improve system energy efficiency [3-8]. Liu [3] proposed a large temperature difference fresh air treatment unit, which uses two-stage coils and corresponding two heat pump systems to treat the air step by step. Under cooling conditions, the first stage is a high-temperature stage evaporator, and the second stage is a low-temperature stage evaporator. When the outdoor temperature is low, the cooling load is small, and only the high-temperature stage heat pump is turned on. Compared to two-stage operation, the system COP can be increased by 10.3%. Li et al. [4] proposed a three-stage coil and corresponding three-stage heat pump system to better achieve humidity control requirement and energy saving. The performance of the system was simulated, and the system can reduce annual air conditioning energy consumption by 15.6%. Two-stage compression heat pumps and corresponding two-stage coils are used to increase the energy efficiency of air handling by 26% at rated operating conditions compared to single-stage system [5].In existing research, the multi-stage fresh air treatment system can achieve high performance improvement in design conditions, but its performance is poor in variable operating conditions; when some coils are used for air treatment, other idle coils will generate resistance, which limits the performance improvement [6-9]. Variable capacity heat pumps are used to improve the performance of heat pumps under part load conditions. The system performance of the air conditioner with variable volume and compression ratio by using three-cylinder two-stage rotary compressor is obtained by experiment, which shows that system performance can be improved by 10.4% at 53% PLR [10]. However, the cost of this compressor is high, and it is difficult to promote the application because it needs to be customized for different scenarios.

To solve the problems in existing direct expansion fresh air units, a direct expansion system using multiple stages and adjustable heat exchange areas for fresh air handling is proposed in this study. The fresh air is handled by coils stage by stage to achieve high efficiency, and the

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dampers of the idle coils are opened when some of the coils are working, which results in the performance improvement under partial load conditions.
The cooling performance of the proposed system with three stages is analyzed in this paper. Firstly, the model of compressor, heat exchangers, and fans in the proposed system are built based on the experimental data. The simulation model of the proposed system is then established. The air treatment process and system performance of the proposed system under different working conditions are analyzed and compared with traditional system.

2 Proposed system

In this section, the system forms of traditional direct expansion fresh air system and multi-stage fresh air handling system are given, and the components of these systems are introduced.
The traditional direct expansion fresh air system is shown in Fig. 1, which includes a fresh air duct and a single direct expansion unit. In this system, fresh air is only sent into the room through a single stage coil for cooling or heating treatment after passing through a filter. In other words, only one temperature grade of refrigerant is used to treat the air.

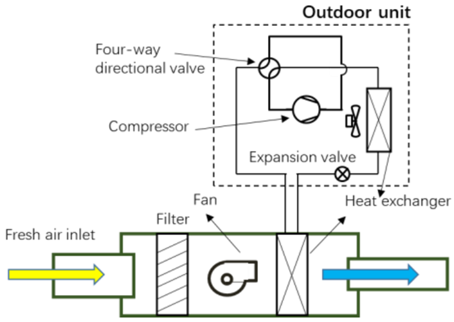


Fig. 1. Traditional direct expansion fresh air system

For the shortcomings of traditional systems, a direct expansion system using multiple stages and adjustable heat exchange areas for fresh air handling is proposed as shown in Fig. 2, which includes a fresh air duct and multiple direct expansion units. The bypass air valve is located next to each coil, and a filter is installed at the fresh air inlet for fresh air purification. Outdoor fresh air passes through filter and multi-stage coils, and then is sent indoors. Each coil is connected to the corresponding outdoor unit. Various grades of refrigerants are used in this system to treat air. The temperature of the refrigerant from the first to the Nth stage increases or decreases in sequence, and outdoor fresh air is treated step by step. Due to the significantly lower refrigerant grade of some heat pump cycles compared to traditional systems, the energy efficiency of the whole system is significantly improved.

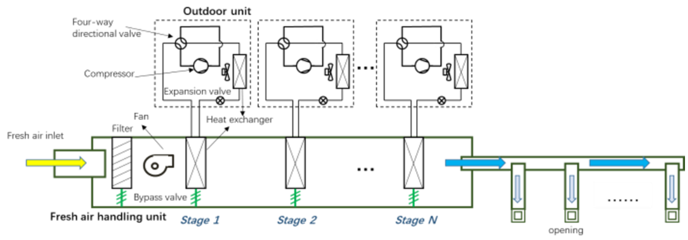


Fig. 2 Multi-stage fresh air handling system

When the proposed system enters partial load operating conditions, some coils are used for fresh air treatment, and the dampers of the idle coils are opened. The load ratio of the operating units increase, thus achieving energy conservation.

3 Methodology

The energy consumption of the direct expansion fresh air system includes the energy consumption of outdoor unit corresponding to each coil and the fan in the fresh air duct. In this section, the energy consumption calculation method of outdoor units and fans in fresh air ducts are proposed, and the total energy consumptions of the traditional system and proposed system are obtained.

3.1 Energy consumption of outdoor units

The heat exchanger in outdoor unit, compressor, expansion valve, and indoor unit heat exchanger form a vapor compression heat pump cycle. Both the outdoor unit heat exchanger and the heat exchanger in the fresh air duct adopt ε− NTU method solution, number of heat transfer units (NTU) and efficiency (ε) are calculated by Eq. (1) and Eq. (2), respectively. Based on the parameters of the sample, the heat transfer coefficient ($K_{r,a}$) is calculated by Eq. (3). Finally, the heat transfer capacity ($Q_{r,a}$) of the heat exchanger is obtained by Eq. (4) [11].

$$NTU = \frac{K_{r,a}A_{r,a}}{C_{min}} \tag{1}$$

$$\varepsilon = 1 - exp(-NTU) \tag{2}$$

$$K_{r,a} = 50v_a^{0.75} \tag{3}$$

$$Q_{r,a} = K_{r,a}A_{r,a}\varepsilon(T_r - T_{ain}) \tag{4}$$

where NTU is the number of heat transfer units; ε is the heat transfer efficiency; C_{min} is the heat capacity of air; $K_{r,a}$ and $A_{r,a}$ are the heat transfer coefficient and area of the heat exchanger, respectively; T_r and T_{ain} are the temperature of refrigerant and air, respectively; v_a is the air velocity; $Q_{r,a}$ is the heat exchange capacity. The heat exchange area and coefficient of the heat exchanger on the outdoor units and fresh air unit are determined by the sample.
The calculation of the rated power of the fans in the outdoor units is shown in Eq. (5), and the power calculation under variable air volume is shown in Eq. (6).

$$W_{ouf\_rated,i} = \frac{m_{a\_rated}P_{ouf,i}}{1000\rho_a\eta_{fan}} \tag{5}$$

$$\frac{W_{ouf,i}}{W_{ouf\_rated,i}} = \left(\frac{m_a}{m_{a\_rated}}\right)^3 \tag{6}$$

where the $W_{ouf\_rated,i}$ and $W_{ouf,i}$ are the fan power under rated and variable operating conditions; m_{a_rated} and m_a are the mass flow rate of air under rated and variable operating conditions; $P_{ouf,i}$ is the pressure head of outdoor unit fan; ρ_a is the air density; η_{fan} is the fan efficiency, which is 0.6.

In this study, the compressor adopts a Carnot efficiency load ratio correction model [11]. Taking the compressor in the X7 outdoor unit as an example to establish the mathematical model. The compressor is a scroll compressor and the refrigerant is R410A. The COP calculation formula of the unit is shown in Eq. (7), and the partial load energy efficiency correction coefficient and load ratio calculation method are shown in Eq. (8) and Eq. (9), respectively. The calculation of compressor power is shown in Eq. (10), and the relationship between heat exchange capacity of evaporator, heat exchange capacity of condenser and compressor power is shown in Eq. (11).

$$COP_{com,i} = COP_{RCC}\eta_{carnot}\varphi = \frac{T_c + 273.15}{T_c - T_e}\eta_{carnot}\varphi \quad (7)$$

$$\varphi = a \cdot LR^2 + b \cdot LR + c \quad (8)$$

$$LR = \frac{Q_{i\_c}}{Q_{capacity,i\_c}} \quad (9)$$

$$W_{com,i} = \frac{Q_{i\_c}}{COP_{com,i}} \quad (10)$$

$$Q_{i\_c} = W_{com,i} + Q_{i\_e} \quad (11)$$

where the $COP_{com,i}$ and COP_{RCC} are the actual efficiency of the compressor and the efficiency of the ideal cycle; η_{carnot} is the carnot efficiency, which is 0.5; φ is the load rate correction factor; LR is the load ratio; a, b, c are obtained by fitting the unit samples; Q_{i_c}, Q_{i_e} and $W_{com.i}$ are the heat exchange capacity of the condenser, evaporator, and the compressor power; $Q_{capacity,i\_c}$ is the unit rated capacity.

Based on the fresh air load of the condenser, the condensation temperature is calculated using Eq. (4), and then assuming the evaporation temperature, the calculated heat exchange capacity of the evaporator can be obtained from the perspectives of the condenser and compressor using Eq. (7), (10), and (11). The calculated heat exchange capacity of the evaporator can be obtained using Eq. (4). When the predicted value is equal to the calculated value, the evaporation temperature at this time can be obtained, and then the energy consumption of the outdoor unit can be calculated.

3.2 Energy consumption of Fan in fresh air unit

The calculation of fan energy consumption for the fresh air unit is shown in Eq. (12). The resistance in the fresh air duct includes the resistance of the filtering device and coils at all stages, and requires external residual pressure for air supply. The calculation of the pressure head of the fan is shown in Eq. (13).

$$W_{faf} = \frac{m_{a\_fa}P_{faf}}{1000\rho_a\eta_{fan}} \quad (12)$$

$$P_{faf} = P_{filter} + \sum_1^i P_{coil} + P_{out} \quad (13)$$

where W_{faf} is fan power consumption; m_{a_fa} is mass flow rate of air under rated condition; P_{filter} is the resistance of the filter, which is 70 Pa; P_{coil} is the resistance of each stage of the coil, which is 30Pa; P_{out} is external residual pressure, which is 200Pa.

3.3 Total energy consumption of fresh air systems

In traditional systems, the total energy consumption of the system is the sum of the energy consumption of the outdoor unit and the fresh air unit fan. The energy consumption and COP of the traditional system are shown in Eq. (14) and Eq. (15), respectively. For the proposed system, the energy consumption of each stage is from the compressor and fan in outdoor unit, and the total energy consumption is the sum of energy consumption of outdoor units in all stages and energy consumption of fan in the fresh air duct. The energy consumption and COP of each stage are shown in Eq. (16) and Eq. (17), and the energy consumption and COP of the proposed system are shown in Eq. (18) and Eq. (19), respectively.

$$W_{total} = W_{com} + W_{ouf} \quad (14)$$

$$COP_{total} = \frac{Q_{total}}{W_{total}} \quad (15)$$

$$W_i = W_{com,i} + W_{ouf,i} \quad (16)$$

$$COP_i = \frac{Q_i}{W_i} \quad (17)$$

$$W_{total\_m} = W_{faf} + \sum_{i=1}^{m} W_i \quad (18)$$

$$COP_{total\_m} = \frac{Q_{total}}{W_{total\_m}} \quad (19)$$

where the W_i and Q_i are the power consumption and load of each stage, respectively; W_{total} and W_{total_m} are the total power consumption in traditional system and proposed system, respectively; Q_{total} is the total load; COP_i , COP_{total} and COP_{total_m} are the energy efficiency of each stage, traditional system and proposed system, respectively.

The energy-saving rate (E_{saving}) and energy efficiency improvement ($COP_{improved}$) of the proposed system compared to the original system are shown in Eq. (20) and Eq. (21).

$$E_{saving} = \frac{W_{total} - W_{total\_m}}{W_{total}} \times 100\% \quad (20)$$

$$COP_{improved} = \frac{COP_{total\_m} - COP_{total}}{COP_{total}} \times 100\% \quad (21)$$

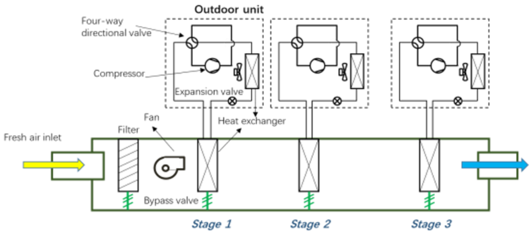
3.4 Case study

In this section, for the traditional direct expansion fresh air system, the outdoor unit is RUXYQ20BA, and the fresh air fan is FMQ50PG30. The rated power of the system is 20 HP, and the rated cooling capacity is 56.5kW. For the proposed system, three-stage coils are used to treat fresh air, and three outdoor units are connected to corresponding coils. The rated power of each outdoor unit is 7 HP.
The total heat exchange area and total unit capacity of the traditional system and the proposed system are equal, and each coil in a multi-stage system handles the same temperature difference of the fresh air.
During cooling season, based on the outdoor temperature and humidity, the rated design condition, partial load condition, low load condition, and ventilation condition are selected. The relevant parameters under each working condition are shown in Table 1.

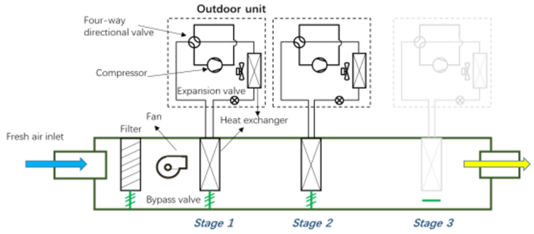
Table 1 System parameters under typical conditions

Working condition	Fresh air temperature/°C	Relative humidity	Fresh air load /kW	Load ratio
Rated design condition	33	69%	54.07	95.70%
Partial load condition	29	61%	25.64	45.38%
Low load condition	25	74%	17.94	31.76%
Ventilation condition	18	62%	0	0

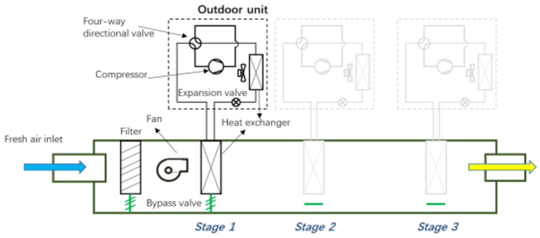
The system operation modes corresponding to the above working conditions are shown in Fig. 3. Under the rated design condition, three stage coils are used to process fresh air step by step. Under the partial load condition, when the system load ratio is low, two-stage coils are used to handle fresh air, and the outdoor unit corresponding to the third-stage coil is switched off, and the bypass valve is opened. Under the low load condition, one-stage coil is used to handle fresh air, the outdoor units corresponding to the other two stages are closed, and the corresponding bypass valves are opened. Under the ventilation condition, all the units are shut down, and all the bypass valves are opened. Compared with traditional systems, the coils of this system are bypassed, and the energy consumption of fan is reduced significantly.



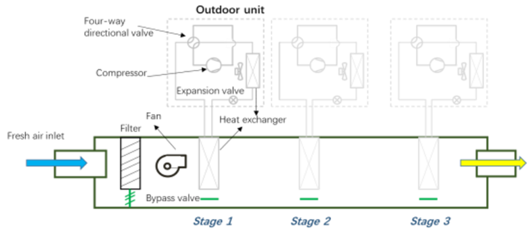
a) Rated design condition



b) Partial load condition



c) Low load condition



d) Ventilation condition

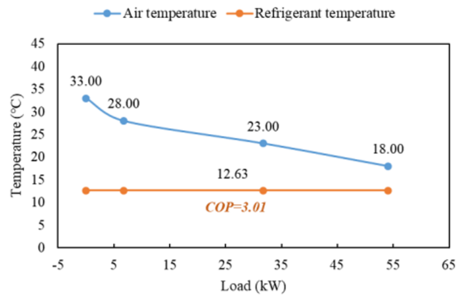
Fig. 3 Diagram of the operating modes under typical cooling operating conditions

4 Results and discussion

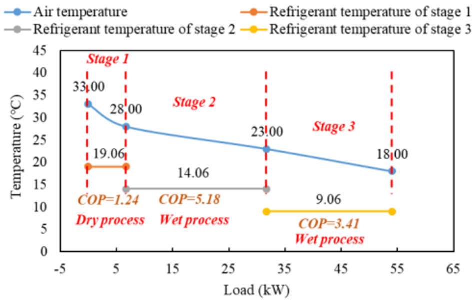
In this section, the air treatment processes of the traditional system and the proposed system are first introduced, and then the energy consumption of the two types of systems under typical operating conditions is analyzed.

4.1 Air treatment process analysis

The air handling processes of the two types of systems under rated design conditions are shown in Fig. 4. Only one temperature of refrigerant is used in the single stage system to cool the fresh air, while three different temperatures of refrigerant in the multi-stage system to cool the air step by step. Compared to the traditional single stage system, the refrigerant evaporation temperature of the first and second stages of the multi-stage system is higher. The first stage coil operates under dry condition, and the load handled in this stage is low, so the energy efficiency is not high. The second and third stages work under wet condition, and the energy efficiency of each stage is higher than traditional system.



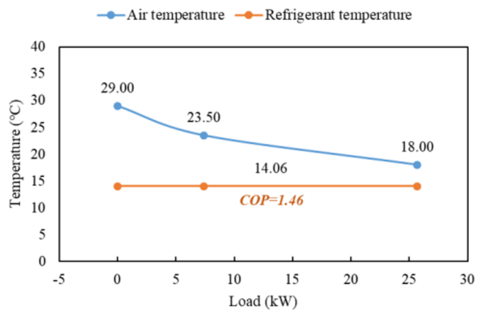
a) Traditional system



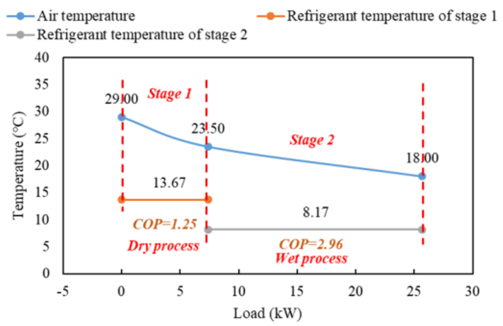
b) Proposed system

Fig. 4 Air handling process under rated design condition

Under the partial load condition, the air treatment processes of the two types of systems are shown in Fig. 5. The systems are operated under partial load in this condition, and two stages are used to cool the fresh air in the proposed system. For the traditional system, single stage treatment results in lower energy efficiency due to a decrease in load ratio. For the proposed system, the first stage is lower energy efficiency due to dry condition, while the second stage improves energy efficiency due to an increase in load ratio of the unit.



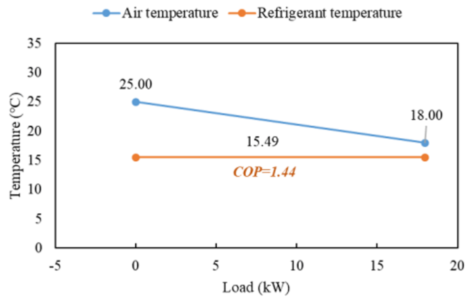
a) Traditional system



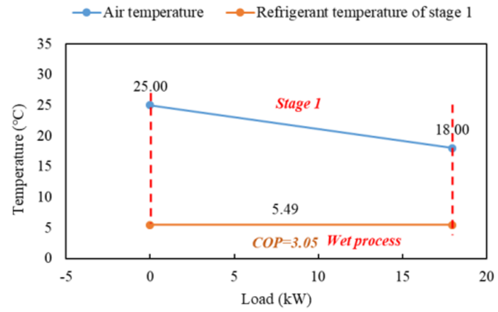
b) Proposed system

Fig. 5 Air handling process under partial load condition

Under low load conditions, the air treatment processes of the two types of systems are shown in Fig. 6. The system load ratio is 31.76%, and the multi-stage system can meet the fresh air treatment demand by using one stage. Compared with traditional system, the load ratio of the operating unit has been significantly improved, resulting in an increase in system energy efficiency.



a) Traditional system



b) Proposed system

Fig. 6 Air handling process under low load condition

4.2 System performance analysis

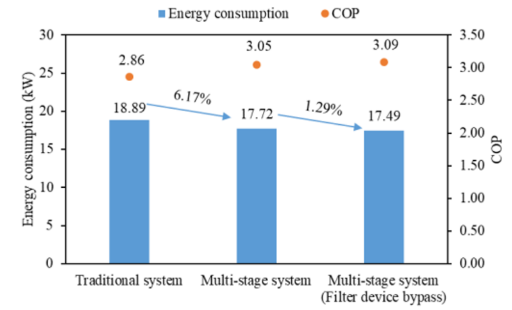
Fig. 7 a) shows the system performance under rated design condition. The first and second stages have higher evaporation temperatures compared to traditional single stage systems, and the evaporation temperature of the third stage is lower. Therefore, the total energy consumption of the multi-stage system is reduced by 6.17% compared to single stage systems.

The system performance under partial load condition is shown in Fig. 7 b). The multi-stage system can meet the processing requirements of fresh air with only two stages, thereby increasing the load ratio of each stage and significantly improving the energy efficiency by 26.09% compared to the traditional system.

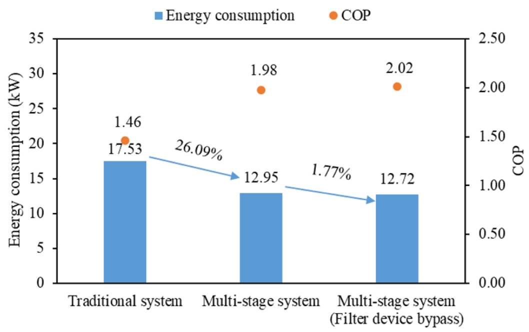
Fig. 7 c) shows the system performance under low load condition. At this time, the unit of the traditional single stage system is operating at low load condition, significantly reducing energy efficiency. Compared with traditional single stage systems, proposed system can reduce energy consumption by 46.52%.

The system performance under ventilation condition is shown in Fig. 7 d). In this mode, the resistance of all coils is bypassed. Compared to traditional single stage system, the proposed system can reduce energy consumption by 25.00%.

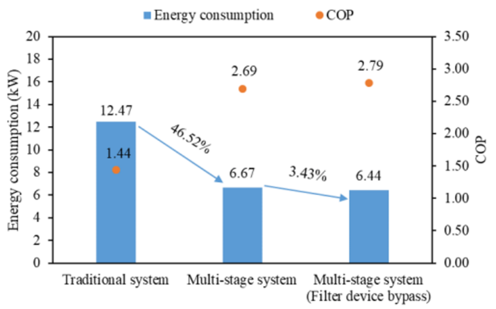
When outdoor air quality is good, bypassing the filter can further reduce energy consumption by 1.29%, 1.77%, 3.43% and 25.93% in four typical conditions, respectively.



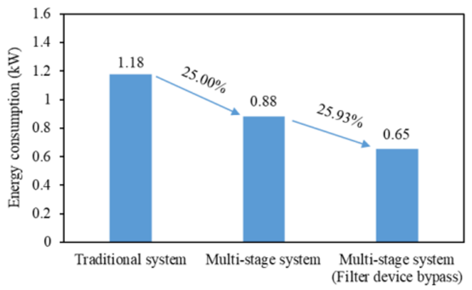
a) Rated design condition



b) Partial load condition



c) Low load condition



d) Ventilation condition

Fig. 7 System performance in typical conditions

Compared to traditional system, the major initial investment of the proposed system is only four air dampers and the bypass air duct. Replacing the 20 HP unit with 3 small capacity units of 7 HP increases investment, but the initial cost increase is not significant. The proposed system can realize good energy-saving under part-load working condition in summer, and can also be used for fresh air treatment in winter and transition season to realize energy-saving, so the payback period will be shorter and the economy of the proposed system is promising.

5 Conclusion

In this paper, a direct expansion system using multiple stages and adjustable heat exchange areas for fresh air handling is proposed. Based on the experimental data of fresh air unit, the simulation model of the proposed system is established. The cooling performance of the proposed system with three units is analyzed in this study. The system performance of the proposed system under different working conditions are analyzed and compared with traditional system. The main conclusions are shown below:

(1) For the proposed system, the fresh air is handled by coils stage by stage to achieve high efficiency, and the dampers of the idle coils are opened when some of the coils are working, which results in the performance improvement under partial load conditions.

(2) For four typical operating conditions, as the load rate decreases, the proposed system can save energy by 6.2%, 26.1%, 46.5%, and 25.0% compared to the traditional system, respectively.

(3) The proposed system can bypass the filter when the outdoor air quality is good, which can further save energy by 1.3%, 1.8%, 3.4%, and 25.9% under four typical operating conditions, respectively.

Acknowledgements

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References

1. C. Liang, X. Li, W. Shi, B. Wang. Energy and Buildings, 238 (2021)
2. G. Zheng, X. Li. International Journal of Refrigeration, 131,10-19 (2021)
3. J. Liu, *Theoretical and experimental research on large temperature difference fresh air treatment units*. (Master thesis, 2011)
4. L. Han, L.W.L, J. Jie. Energy conversion & management, 121(Aug.):162-173 (2016)
5. Z Zhang, X Cao, Z Yang, et al. Applied Energy, 242:1600–12 (2019).
6. L. Wang, G.Ma, A.Ma, et al. International Journal of Refrigeration, 108:26-36 (2019)
7. X. Jia, G.Ma, F. Zhou, S.Liu, G. Wu, Q. Sui. Journal of Building Engineering, 103468 (2022)
8. L. Wang, G. Ma, A. Ma, et al. Energy Conversion and Management, 176(NOV.):384-392 (2018)
9. L. Wang, G. Ma , F. Zhou. Energy and Buildings, 223:110149 (2020)
10. B Xiao, T Huang, L He, et al. International Journal of Refrigeration, 100: 343-353 (2019).
11. W. Wang, C. Liang, F. Zha, et al. Energy and Buildings, 113073 (2023)