https://doi.org/10.1051/e3sconf/201913601047

Performance Diagnosis and Energy-Saving Optimization of the Heat Recovery Unit in An Office Building in Sichuan

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Abstract: In recent years, the number and volume of public buildings have been increasing sharply, which has led to an increase in the working capacity of HVAC systems in public buildings. This paper analyzes and tests a heat recovery unit in an office building in Sichuan Province, China. The fresh air input temperature, the fresh air output temperature and the exhaust air temperature of the heat recovery unit in January in 2018 are analyzed. The positive and negative benefits are calculated, and thus the energy saved by the system during the test is obtained.

1 Introduction

After entering the 21st century, as people's requirements for living environment and energy demand gradually increase, the consumption and demand of energy have increased sharply, which has led to serious energy waste and environmental pollution crisis. Therefore, energy conservation and environmental protection have become an urgent research topic in all fields of today's society [1]. Based on this situation, heat recovery units are widely used in HVAC system. A lot of researches have been done by domestic and foreign scholars for energy saving of heat recovery devices. Lin Xiyun [2] qualitatively analysed the influencing factors of the heat recovery device, and proposed a new heat recovery energy evaluation method, and verified the feasibility of the method through practical engineering. Xu Huqing [3] calculated the energy saved by the heat recovery device according to the outdoor fresh air hourly temperature and humidity through engineering examples. The difference of the energy consumption between setting and without the heat recovery device was calculated by simulating the building load, and it was found that the energy saving effect of the heat recovery unit was obvious. Literature [4]-[5] studied the using conditions of heat recovery units and outdoor meteorological parameters for actual projects, and produced energy recovery graphs, which can estimate the recovery energy according to indoor and outdoor conditions, thus the performance of the heat recovery unit is evaluated.

2 Experimental background and arrangement

2.1Experimental background

This article takes an office building in Chengdu, Sichuan Province as an example. The total area of building is 86,545 m^2. The height of the building is 69.30 m.

The air-conditioning area of the project is about 50,000m^2, the summer design cold load is 5042 kW, and the winter design heat load is 2661kW. In order to flexibly adapt to the changes of the load, two large and one small water chilling units operate in summer. Two centrifugal chillers with rated cooling capacity of 1934KW and one of 1260kW were designed as the cold source in summer. The supply and return temperature of chilled water are 7/12℃.

3 full heat recovery units were set on the roof as heat source. Due to the problems such as control and operation management, the operation efficiency of the units is low. Therefore, the total heat recovery unit is tested.

2.2 Experimental arrangement

The experiment was conducted from January 29 to 31, 2018. The daily recording time was 9:00-21:00.

The inlet and outlet temperature and humidity of the fresh air runner, the inlet temperature and humidity of the exhaust air, the speed of the fresh and exhaust air and the current of the motor in the runner were tested. The pressure difference between the inlet and outlet of the pumps, the current of each pump and the input power of the pump were also tested.

2.3 Experimental data

The test data of the pumps are shown in the table 1

Table 1. Measured parameters of water pump

Total flow of the water system	Head of each pump	curr ent	Input power
340m3/h	17m	25A	16.4kW

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The fresh air pipe and exhaust air pipe of the unit are in the same size. The speed of the wind in the fresh air pipe is 6.5m/s and in the exhaust air pipe is 5.9m/s, and the fresh air volume is 28080m^3/h, the exhaust air volume is 25488m^3/h. The speed and the size of the pipes are shown in Table 2.

Table 2. Wind speed and pipe size

	Wind speed (m/s)	Pipe size (mm)
Fresh air side	6.5	1500*800
Exhaust air side	5.9	1500*800

The inlet and outlet temperature and humidity of the fresh air runner, the inlet temperature and humidity of the exhaust air will be illustrated in the graphs.

3 Analysis of experimental data

3.1Analysis of test data of heat recovery unit

The efficiency of the total heat exchange unit is an important indicator to measure its energy saving effect. According to the measured data, the temperature and humidity of the indoor exhaust air are stable. Considering the energy-saving reason, indoor humidity is not controlled in the office building during the heating season, that is, the office building does not have any additional humidification device, and only the sensible heat recovery efficiency can be analyzed for the project.

$$\eta_R = \frac{G_f（t_{i2}-t_{i1}）}{G_p（t_0-t_{i1}）} \quad (1)$$

η_R is sensible heat efficiency; t_0 is indoor exhaust air temperature ℃; t_{i1} is outdoor fresh air temperature ℃; t_{i2} is fresh air temperature after the rotor ℃; G_f is fresh air volume m³/h

Figure 1, 2, and 3 are the graphs of the test results on January 29, 2018. During the operation period of the unit, the temperature change of the fresh air entering and leaving the runner and the temperature change before the exhaust wind wheel.

Because the indoor temperature fluctuation is small, the lowest fresh air temperature is 5.7℃ and the highest is 8.1℃ in the whole day (9:00-21:00). The temperature of the outdoor fresh air has a slightly increase and reaches the peak at around 11:00, and then falls. It's clear from the figures below that the temperature difference between the fresh air entering and leaving the rotor increases as the temperature difference between the exhaust air and the fresh air growing. For example, the temperature difference between exhaust air inlet and fresh air inlet is 15.28℃ at 9:00 am, the temperature difference between fresh air inlet and outlet is 4.87℃. And in 17:45, these two temperature are 17.08℃ and 5.49℃. Hence, the greater the difference between the fresh air inlet temperature and the exhaust inlet temperature is, the higher the temperature of the fresh air increased by the runner.

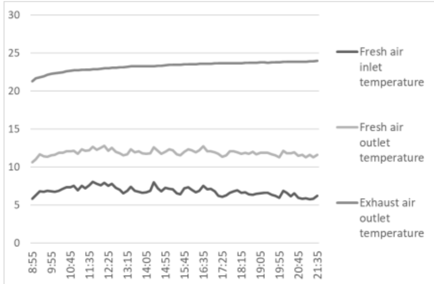

Fig 1. Fresh air inlet and outlet temperature of 1.29

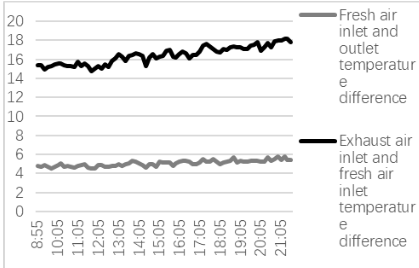

Fig 2. Fresh air inlet and outlet temperature difference - exhaust air inlet and fresh air inlet temperature difference

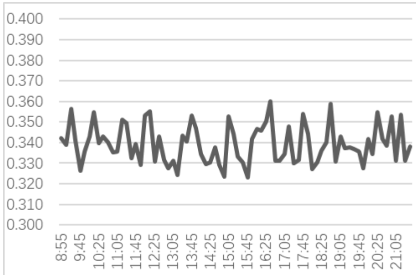

Fig 3. Sensible heat efficiency of 1.29

Figures 4-6 are graphs of the test results for January 30&31, 2018.

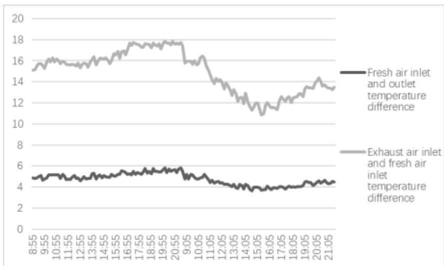

Fig 4. Fresh air inlet and outlet temperature difference-exhaust air inlet and fresh air inlet temperature difference 1.31.

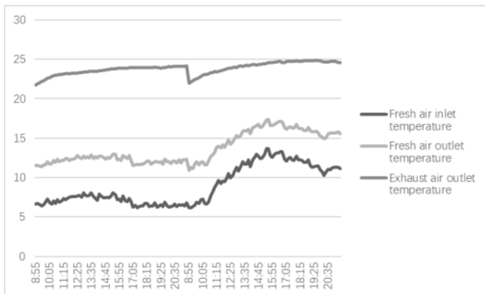

Fig 5. Fresh air inlet and outlet temperature of 1.30

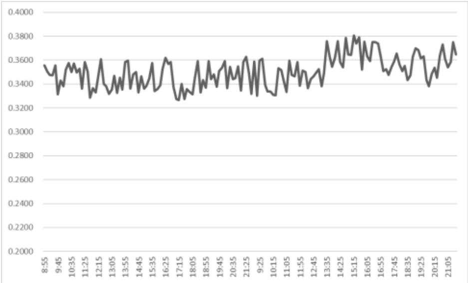

Fig 6. Sensible heat efficiency of 1.30&31

As can be seen from Figure 5, the outdoor temperature fluctuation is not large, but there is a decrease around 17:00. This leads to a reduction in sensible heat efficiency.

3.3 Calculation of energy recovered by the unit

Only the sensible heat is concerned in this passage. When calculating the amount of the hour-by-hour sensible heat recovered by the total heat recovery unit, the indoor and outdoor air temperature is recorded and calculated. Then the total energy saved by the unit for the whole time can be calculated according to that.

Hourly recovering heat Q_r can be calculated as follows:

$$Q_r = G_f c_p \rho (t_0 - t_i) \eta_R \quad (2)$$

Where G_f is the fresh air volume, ρ is the density of the air, t_i is the dry bulb temperature of the outdoor air, t_0 is the hourly indoor exhaust air temperature, t_i is hourly outdoor air temperature, η_R is hourly sensible heat efficiency.

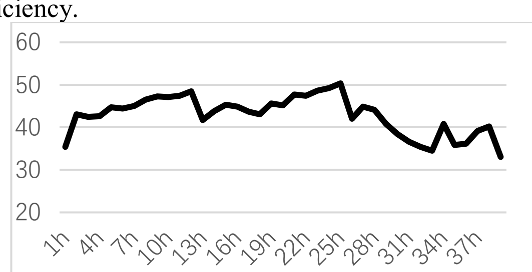

Fig 7. Hourly energy recovered

Calculating the hourly sensible heat efficiency and the electricity consumption is shown in the Fig.12. During the test period, the unit operated for 39 hours, and the amount of the heat recovered fluctuates between 33kWh and 50 kWh. Thus the total energy is the sum of the hourly energy and is approximate 1667kWh.

① Positive benefits the heat recovery unit brings to the system

(1) Reduction of energy consumption in cold and heat source systems

The energy reduced in the boiler in winter can be calculated as follows:

$$N_{hs,i} = \frac{Q_r}{\eta_b \eta_1 \eta_2} \quad (3)$$

$N_{hs,i}$ is the electricity reduced in the boiler in winter, kWh. The heat efficiency of the boiler η_b is 75%, the heat value of the nature gas η_1 is 50839kj/kg. The conversion coefficient η_2 of natural gas consumption and electricity consumption is 0.101kg/(kWh). Hence the final amount of the electricity is 1783.47kWh.

② Negative benefits the heat recovery unit brings to the system

(1) Increased energy consumption of the fan due to the resistance of the heat recovery unit.

According to the speed and the size of the pipes, the volume of the fresh air is 28080 m³/h while the exhaust air is 25488 m³/h. The increased energy consumption caused by the heat recovery unit in both fresh air and exhaust air side ϕ_{fan} can be calculated as follows:

$$N_{f,i} = \left(\frac{L_s \Delta P_s}{3600 \times 1000 \eta_{CD} \eta_F} + \frac{L_e \Delta P_e}{3600 \times 1000 \eta_{CD} \eta_F}\right) * T \quad (4)$$

$N_{f,i}$ is the increased delivery energy consumption in the fans, kWh. ΔP_s, ΔP_e are the additional resistance of the fresh air and exhaust air side, and is 200Pa and 300Pa respectively. L_s, L_e are the volume of the fresh air and the exhaust air, m³/h. η_{CD} is the efficiency of fan motor and transmission, which, according to the GB 50189-2015 Design Standard for Energy Efficiency of Public Buildings[6], is 0.855, and η_F is the total efficiency of the fan.

It's calculated that the increased energy consumption of the fresh air side is 2.61kw, the energy consumption increased by the exhaust side is 3.55kw, totaling 6.16kw, during the test period the time T is 39h.Therefore the total increased energy consumption of the delivering system caused by the adding of the heat recovery unit is 240.24kWh.

It's calculated that the additional delivery consumption in fresh air and exhaust air are 2.61kW and 3.55 kW. The T of the test period is 36h thus the total energy increment is 221.76kWh.

(2) Energy consumption increased by the additional motors for the runner itself operates

The value of the current of the fresh air fan and the exhaust air fan and the runner motor is recorded during the test period. The power of the rotor motor is 0.47kW, the total energy consumption of the electricity during the test period is 16.2 kW.

Table 3. Current data of heat recovery unit

fresh air fan	44.5	42.3	42.1
exhaust air fan	32.5	31.6	32.7
runner motor	0.94	0.91	0.95

4 Conclusion

1) The energy saved by the heat recovery unit during the test is converted into electricity. And the positive income minus the negative income is 1305.27kWh.

2) Through the analysis of the influencing factors and test data, it can be seen that the energy recovered by the total heat recovery unit is related to the outdoor fresh air temperature when the indoor exhaust air temperature is constant. The lower the fresh air inlet temperature is, the higher the heat recovered by the runner.

3) It can be seen that the energy saving effect is still obvious even in the case where the sensible heat efficiency is low.

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