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Thermodynamic analysis of an innovative heat pump using indoor and outdoor air as heat source

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Heat recovery from exhaust air is an effective approach to carbon emission reduction for buildings in cold regions. This paper proposes a novel heat pump system to efficiently utilize the exhaust air. The system has two heat sources: exhaust air and ambient air. It consists of a condenser, two parallel evaporators and compressors, and two throttle valves. The exhausted indoor air is first used to heat the refrigerant in the medium-pressure evaporator. The exhaust air at a reduced temperature is then mixed with the ambient air to heat the low-pressure evaporator. The advantages of the mixed air source heat pump (MASHP) with two-stage evaporation over a conventional exhaust air heat pump (EAHP) include a larger heating capacity, higher waste heat recovery and lower carbon emission. Besides, the exhaust air can be employed for defrosting without additional power consumption. Thermodynamic performance of the system is investigated. Comparison is made with a conventional EAHP and a directly mixed air source heat pump with single-stage evaporation. The results indicate that under the conditions of exhaust air temperature of 20 ^{o}C, mass flow rate of 1.0 kg/s and ambient temperature of 0 ^{o}C, the profits for the proposed MASHP, conventional EAHP and MASHP with single-stage evaporation are about 11.5, 5.1 and 3.7 pound/day, respectively.

Keywords: heat pump, exhaust air, two-stage evaporation, heat recovery, profit

INTRODUCTION

Ventilation heat loss accounts for a large proportion of the building energy losses. Mechanical ventilation with heat recovery is able to provide the best indoor air-quality with low energy consumption. Thermal recovery from exhaust air has been widely dealt with in literature, in the form of either active or passive systems [1-3]. Among the technologies, exhaust air heat pumps (EAHP) is regarded as energy efficient heating system. It extracts heat from the exhaust air of a building which otherwise will be wasted and transfers the heat to the supply air, hot tap water and/or hydronic heating system. Extensive studies have been conducted on the EAHP, including the control algorithms [4], structural optimization [5], combination with other energy sources [6, 7], connection schemes [8], and economical performance [9]. Compared with other types of exhaust heat recovery methods, an EAHP avoids cross contamination between fresh and exhaust air, and can have a high recovery efficiency of more than 100% without any need of reheat for the fresh air [10]. The EAHPs have been commercialized. For example, new detached houses in Sweden are most often installed with an EAHP to minimize ventilation heat loss [11].

However, one challenge of EAHP is the limited heating capacity, leading to a high capital cost per kW and long payback period. The ventilation air stream for a typical house might be 30 litres per second and the maximum heat of recovery is about 720W at an ambient temperature of 0 ^{o}C. The indoor air exhausted into the environment may have a temperature of 5-10 ^{o}C higher than the ambient for the sake of an acceptable COP, thereby leading to a lower heat recovery. Between 2009 and 2013, thousands of brand new social homes were equipped with EAHPs in the UK, most owners and housing association tenants had reported increased electricity bills [12]. Moreover, there are heat losses through windows, walls and roofs. The heat recovered from the ventilation is not sufficient to meet the building energy demand. Additional heat is required to provide space heating and hot water, which may be from either electric or gas heater. This will always be accompanied by a higher operating cost or carbon emission.

In this paper, a novel heat pump system is proposed to efficiently make use of the exhaust air from the building. The exhaust air is only part of the energy sources. It is incorporated with outdoor air. The capacity of heat pump is therefore significantly larger than a conventional EAHP and the carbon emission from the building can be reduced. To the best of the authors' knowledge, it is the first time that exhaust air is mixed with ambient

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air to improve the performance of the heat pump system. The profit of the system based on the reduction in electricity consumption is analysed and compared with that a conventional EAHP system. Finally, the drawback of the novel system and future works are discussed.

SYSTEM DESCRIPTION

The heat pump is used to produce hot water for space heating (e.g., underfloor heating). The mixed air source heat pump (MASHP) with two-stage evaporators, MASHP with single-stage evaporator and conventional EAHP system are illustrated in Figs.1-3 respectively.

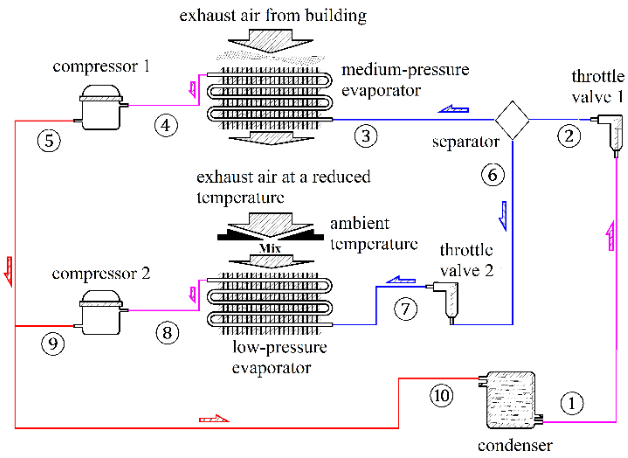

Fig.1. MASHP system with two-stage evaporators

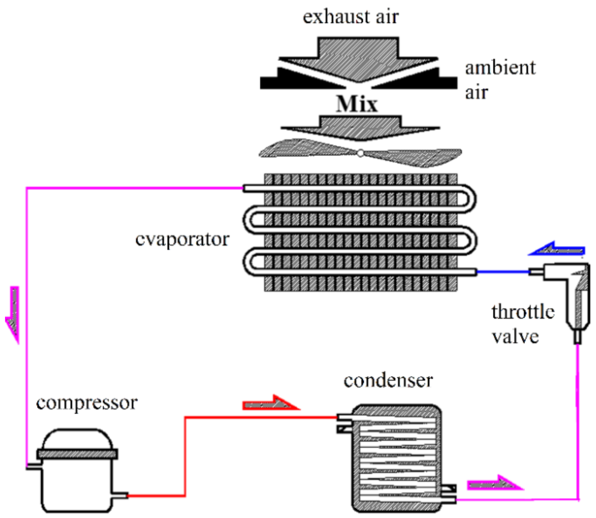

Fig.2. MASHP system with single-stage evaporator

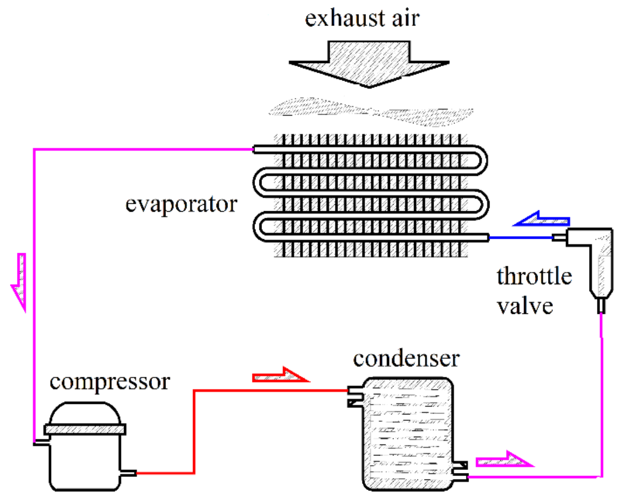

Fig.3. EAHP system

For the MASHP with two-stage evaporators, the refrigerant leaving the condenser is first throttled and is at binary phase state at the outlet of the throttle valve. Part of the liquid is further throttled and enters the low-pressure evaporator. The rest liquid and vapor flow into the medium-pressure evaporator. Vapor from the medium- and low-pressure evaporators is compressed by compressors 1 and 2, respectively. The vapor is then condensed in the condenser. The refrigerant circulates. The exhaust heat is first recovered by the medium-pressure evaporator and the exhaust air at a reduced temperature is further mixed with the ambient air. The thermodynamic states are marked in number in the figure.

The MASHP with single-stage evaporator in Fig.2 has not been reported in the literature either. It is simpler than that in Fig.1 and does not have the medium-pressure evaporator and compressor, which may lead to large thermodynamic irreversibility in the mixing process.

The EAHP in Fig.3 is commercially available. It does not have the low-pressure evaporator and compressor as compared with the MASHP with two-stage evaporators. Its heat source is just exhaust air from the building.

MATHEMATICAL MODELS

The COP of a heat pump is the ratio of the output heat to the input power. In this simulation emphasis is put on the thermodynamic cycle. The power consumption by fans and pumps are not considered. Therefore, the COP is defined by

$$COP = \frac{Q_{out}}{W_{ele}} = \frac{h_{cond,in} - h_{cond,out}}{h_{comp,out} - h_{comp,in}} \tag{1}$$

where:
$h_{cond,in}$ - specific enthalpy at the inlet of the condenser, J/kg;
$h_{cond,out}$ - specific enthalpy at the outlet of the condenser, J/kg;
$h_{comp,in}$ - specific enthalpy at the inlet of the compressor, J/kg;
$h_{comp,out}$ - specific enthalpy at the outlet of the compressor, J/kg.

The ventilation heat loss without recovery is

$$Q_{ven,l} = m_{ex} C_P (T_{ex} - T_{am}) \tag{2}$$

where:
m_{ex}- exhaust air flow rate, kg/s;
C_P – heat capacity, J/kg/^{o}C;
T_{am}- ambient temperature, °C;
T_{ex} - temperature of exhaust air leaving the building, °C.

The mixing of ambient and exhaust air for heat pump is rarely studied in the literature, and hence special attention is paid to the modelling of this process in this paper. The air temperature after mixing (i.e., air temperature at the low-pressure evaporator inlet) is calculated by

$$T_{mix,in} = \frac{m_{ex}T_{ex} + x \cdot m_{ex}T_{am}}{m_{ex} + x \cdot m_{ex}} = \frac{T_{ex} + x \cdot T_{am}}{1+x} \tag{3}$$

$$x = \frac{m_{am}}{m_{ex}} \tag{4}$$

where:
m_{am}- ambient air flow rate, kg/s

The mixed air temperature leaving the evaporator is

$$T_{mix,out} = T_{mix,in} - \frac{Q_{evap}}{m_{ex}(1+x)C_P} = T_{am} - \frac{\frac{Q_{evap}}{m_{ex}C_P} - (T_{ex} - T_{am})}{1+x} \tag{5}$$

where:
Q_{evap}- low-pressure evaporator input heat for the MASHP with two-stage evaporation or evaporator input heat for EAHP and MASHP with single-stage evaporation, W;

The temperature drop of air through the evaporator is an important parameter in the design of the heat exchanger. A smaller temperature drop can provide a higher COP but more fan power is needed owing to a larger flow rate. The optimum temperature drop is then a compromise. For a fair comparison, a constant temperature drop of 5°C is assumed for the low-pressure evaporator of the MASHP and conventional air source heat pump (ASHP).

$$\Delta T = T_{mix,out} - T_{mix,in} = 5 \tag{6}$$

$$C_P m_{ex}(1+x)\Delta T = Q_{evap} \tag{7}$$

With $C = \frac{Q_{evap}}{m_{ex}C_P(T_{ex} - T_{am})}$,

$$x = \frac{(T_{ex} - T_{am})C}{\Delta T} - 1 \tag{8}$$

$$T_{mix,out} = T_{am} - \frac{\frac{Q_{evap}}{m_{ex}C_P} - (T_{ex} - T_{am})}{1 + \frac{m_{am}}{m_{ex}}} = T_{am} - \Delta T \frac{C-1}{C} \tag{9}$$

It indicates that at a given C, the mixed air temperature leaving the evaporator is independent on the exhaust air temperature. The relative increment of $T_{mix,out}$ by that of air leaving the evaporator of a conventional ASHP is

$$\Delta T_r = T_{mix,out} - (T_{am} - \Delta T) \quad = \frac{\Delta T}{C} \tag{10}$$

The minimum temperature difference between the refrigerant and air in the evaporator shall take place at the air outlet, and is defined by

$$\Delta T_{min} = T_{mix,out} - T_{evap} \tag{11}$$

where
T_{evap} - refrigerant evaporation temperature, °C.

Profit is chosen as a key indicator in the comparison between the MASHP and conventional ASHP. It is the relative profit of the MASHP and defined by

$$Pro_{ex,heat} = C_{heat}(W_{ele}COP_{mix} - W_{ele}COP_{am}) = C_{heat}\frac{C \cdot m_{ex}C_P(T_{ex} - T_{am})}{(COP_{mix} - 1)}(COP_{mix} - COP_{am}) \tag{12}$$

where:
$Pro_{ex,heat}$- profit based on heat gain, £/s;
C_{heat} – price of heat, £/J;
W_{ele}–electricity input or consumption, W;
COP_{mix} – COP of the MASHP or EAHP system;
COP_{am} - COP of a traditional ASHP system operating at the same ambient and condensation temperature.

$Pro_{ex,heat}$ represents the net profit of heat gain by the MASHP and EAHP at the same electricity consumption. Because the MASHP and EAHP are expected to have a higher COP than a conventional ASHP the heat gain is higher at a given electricity input. Similarly, a comparison can be made based on the electricity consumption.

$$Pro_{ex,ele} = C_{ele}\left(\frac{Q_{gain}}{COP_{am}} - \frac{Q_{gain}}{COP_{mix}}\right) = C_{ele}\frac{C\cdot m_{ex}C_P(T_{ex}-T_{am})}{(COP_{mix}-1)}\frac{(COP_{mix}-COP_{am})}{COP_{am}} \quad (13)$$

where:

$Pro_{ex,ele}$- profit based on reduction in electricity consumption , £/s;

C_{ele} – price of electricity, £/J;

$Pro_{ex,ele}$ represents the net profit of electricity reduction by the MASHP and EAHP at the same heat gain. The electricity inputs of the MASHP and EAHP are lower than that of a conventional ASHP at a given heat output due to a higher COP.

RESULTS AND DISCUSSION

In the following simulation, the price of electricity of 20 p/kWh and heat of 4p/kWh, compressor efficiency of 0.8, minimum temperature difference of heat transfer in the evaporator of 4 ~ 5 ^oC, minimum temperature drop of air through the evaporator of 5 ^oC, condensation temperature of 50 ^oC, exhaust air flow rate of 1.0 kg/s and ambient temperature of 0 ^oC are assumed. The refrigerant is R134a.

Performance of the EAHP

For the EAHP, the exhaust air is the only heat source of the heat pump. Because it has a higher temperature than the outdoor air, the COP will be higher than that of a conventional ASHP, resulting in more profit. The evaporator input heat shall not be larger than the ventilation heat loss. And the heat ratio C shall be less than 1. Otherwise the exhaust air leaving the evaporator will have a lower temperature than the ambient and the utilization of exhaust air might have no advantage over a conventional ASHP. Fig.4 shows the profit variation in the range of C <1. The curves exhibit a parabola shape opening downward. As C increases from 0, the profit first increases and then reaches the maximum. Further increment in C leads to profit decrement. The C corresponding to the maximum profit is around 0.65. A larger C means a larger heat recovery from the exhaust air, but the temperature of exhaust air and the COP of the EAHP drop, as displayed in Fig.5. The maximum profit is a compromise between the waste heat recovery from the exhaust air and COP of the heat pump.

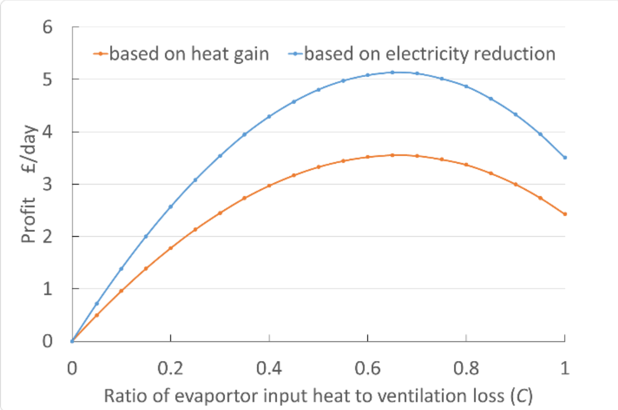

Fig.4. Variations of the profit with the input heat (C ≤1)

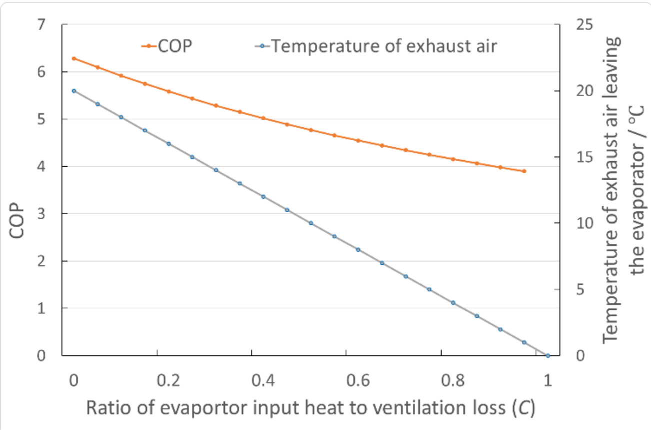

Fig.5. Variations of the temperature of exhaust air leaving the evaporator and COP with the input heat (C ≤1)

The maximum profit for the EAHP based on heat gain and reduction in electricity consumption is about 3.55 and 5.13 £/day. The results indicate that in order to maximum the profit, the heat ratio (C) is less than 100%. However, when the heat pump is used for heat supply of the entire building, the evaporator input heat at design condition is normally larger than the ventilation heat loss (i.e., C >1) because aside from the ventilation heat loss, there are heat losses through the windows, walls and roofs. To address this issue, the air ventilation can be facilitated intermittently. For example, a ventilation rate of 1 kg/s in a continuous operation mode can be adjusted to be 3 kg/s at 20 min intervals and 0 kg/s at 40 min intervals, as shown in Fig.6. For the intermittent ventilation, the daily average ventilation heat loss is unvaried but the exhaust waste heat can be recovered more efficiently.

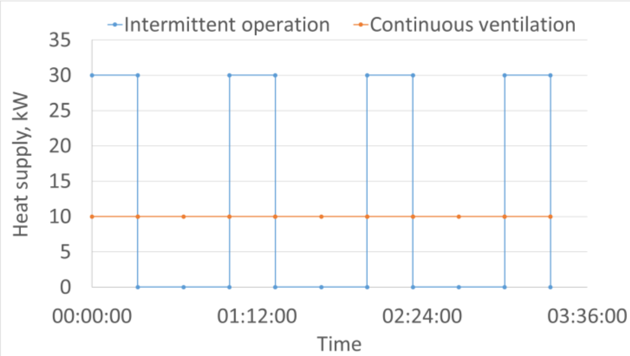

Fig.6. Control strategy for the ventilation

Performance of the MASHP with single-stage evaporator

For the MASHP with single-stage evaporator, the exhaust and ambient air are mixed directly. Although the principle is simple, analysis of this process is lacked. An insight to the mixing process is needed. Based on the established mathematical models, the following deductions can be made for the directly mixed air source heat pump when C<1, C=1 and C >1.

(1) If $Q_{evap} < m_{ex}C_P(T_{ex} - T_{am})$, then A higher x (i.e., a higher ambient air flow rate) leads to a lower $T_{mix,out}$ at a given the exhaust air flow rate. For the sake of a higher COP, there is no need to mix the exhaust air with the ambient air. This case occurs when the heat pump system is assisted by solar arrays and the heat pump provides only part of the energy required for heating. The MASHP will work in a similar way to the EAHP.

(2) If $Q_{evap} = m_{ex}C_P(T_{ex} - T_{am})$, then $T_{mix,out} = T_{am}$, which is independent on the mass flow rate of the ambient air.

(3) If $Q_{evap} > m_{ex}C_P(T_{ex} - T_{am})$, then

$$T_{ex} - T_{am} < \frac{Q_{evap}}{m_{ex}C_P} \qquad (14)$$

$$\frac{Q_{evap}}{m_{ex}C_P} - (T_{ex} - T_{am}) > 0 \qquad (15)$$

$$T_{mix,out} < T_{am} \qquad (16)$$

For C>1, the mixed air leaving the evaporator is at a temperature lower than the ambient temperature and therefore the evaporation temperature is always lower than the ambient temperature. Given the flow rate of the exhaust air, a larger flow rate of ambient temperature leads to a higher temperature of the mixed air leaving the evaporator and thus a higher evaporation temperature and COP.

The profits based on the heat gain and electricity reduction are displayed in Fig.7. The profits increase slightly with the increment in the heat ratio (C). They are about 2.53 and 3.75 £/day. The curve shows that the profit is not influenced remarkably by the evaporator input heat. The reason behind this phenomenon is that the difference between the COPs of the MASHP and conventional heat pump (i.e. $COP\text{-}COP_{ref}$) is almost reversely proportional to C, as shown in Fig.8. On the other hand, the evaporator input heat is proportional to C. Due to the trade-off between COP_{mix} - COP_{am} and C, The profits are almost constant.

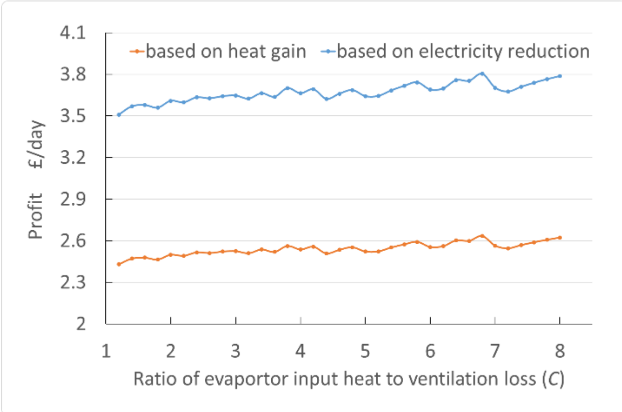

Fig.7. Variations of the profit with the input heat (C >1)

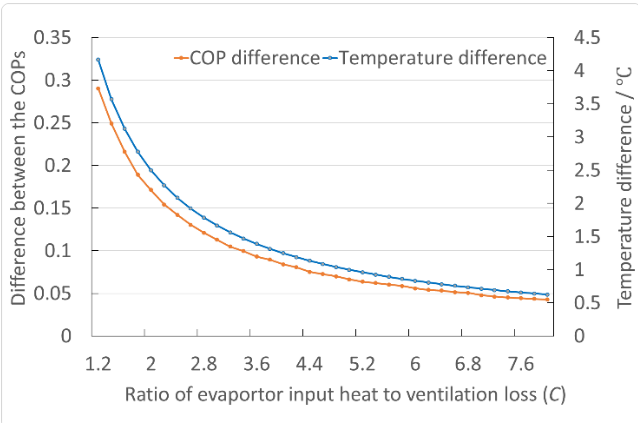

Fig.8. Variations of the COP difference and evaporation temperature difference with C

Variations of the ratio of ambient air flow rate to exhaust air flow rate (x), together with the mixed air temperature leaving the evaporator ($T_{mix,out}$), is depicted in Fig.9. As the C increases, x goes up almost linearly, while $T_{mix,out}$ drops but the decrement becomes slower at a higher C. Given a temperature drop through the evaporator of 5°C, Q_{evap} influences the mass flow ratio, outlet temperature of air leaving the evaporator and the profit.

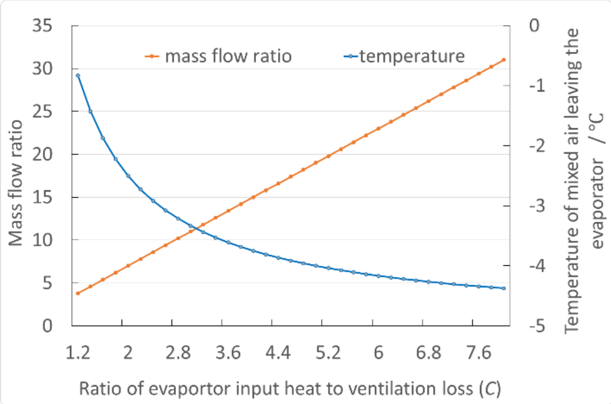

Fig.9. Variations of the mass flow ratio and mixed air temperature with the input heat (C)

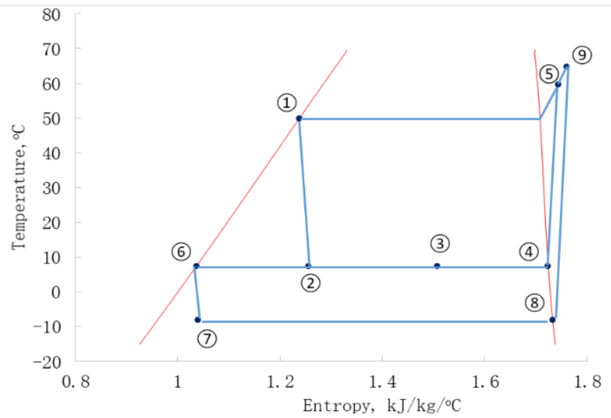

Fig.10. *T-s* diagram of the MASHP with two-stage evaporators

Performance of the MASHP with two-stage evaporators

The temperature-entropy (*T-s*) diagram of the MASHP with two-stage evaporators is illustrated in Fig.10. Similarly with those in Fig.1, the thermodynamic state points 1-10 represent the condenser outlet ①, throttle 1 outlet ②, medium-pressure evaporator inlet ③, medium-pressure evaporator outlet ④, compressor 1 outlet ⑤, throttle 2 inlet ⑥, throttle 2 outlet ⑦, low-pressure evaporator outlet ⑧ and compressor 2 outlet ⑨ and condenser inlet ⑩ (which is between ⑤ and ⑨), respectively.

Given the mass flow rate of exhaust air of 1.0 kg/s, the performance of the MASHP is influenced by the mass flow rate of the ambient air. The variations of the heat output and COP of the proposed heat pump with the mass flow ratio are displayed in Fig.11. The mass flow ratio (x) is defined by Eq.(4). The COP decreases while the heat output increases with the increment in x. The variations of the profits are depicted in Fig.12. Profits climb as x increases, which is due to an increasing heating capacity.

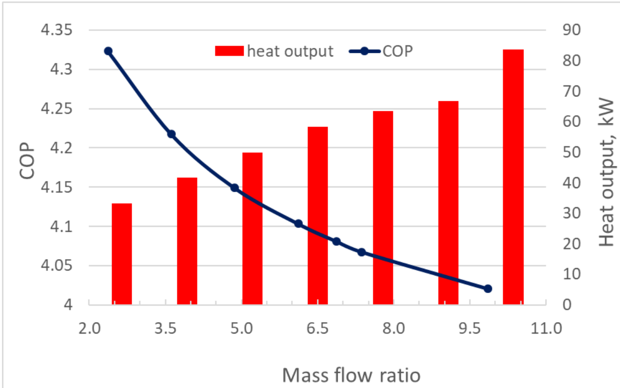

Fig.11. Variations of heat output and COP of the MASHP with the mass flow ratio

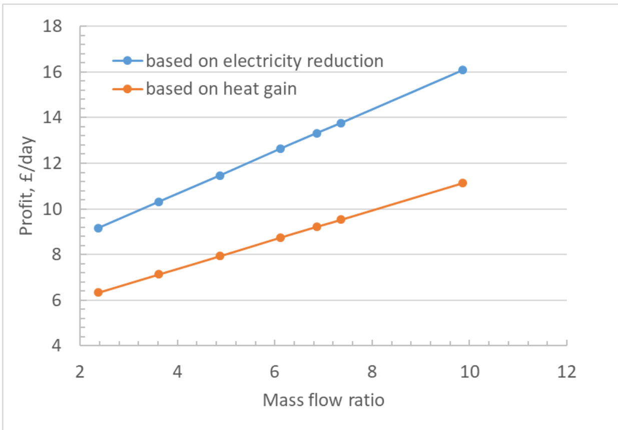

Fig.12. Variations of profits of the MASHP with the mass flow ratio

As a case study, parameter distribution of the MASHP with two-stage evaporators is exemplified in Tab.1 when the mass flow of ambient air is 4.87 (i.e., x = 4.87). The exhaust air is first used to heat the medium-pressure evaporator and then mixed with ambient air for the heating of the low-pressure evaporator. The temperature drop of exhaust air through the medium-pressure evaporator is 8.5 °C and the mixed air leaving the low-pressure evaporator is at -3.1°C. The COP and heat output of the proposed heat pump are 4.15 and 50 kW. The profit based on the reduction of electricity consumption is 11.5 £/day. Notably, the conventional ASHP operating at the same condensation temperature and ambient temperature is only 3.46.

Aside from the thermodynamic performance, the MASHP with two-stage evaporators have some technical advantages. Freezing is a serious problem of heat pumps applied in cold climate regions. A defrosting cycle is usually adopted and a considerable amount of electricity is needed, leading to a lower performance of the system. This problem can be overcome by the proposed heat pump. The exhaust air can be utilized for defrosting without additional electricity consumption.

Table 1. Parameter distribution of the MASHP with two-stage evaporators

Point	State	Temperature	Pressure	Flow rate	Quality
		°C	MPa	kg/s	%
1	liquid	50	1.32	0.30	0
2	binary	7.5	0.38	0.30	31.9
3	binary	7.5	0.38	0.14	68.4
4	vapor	7.5	0.38	0.14	100
5	vapor	59.7	1.32	0.14	100
6	liquid	7.5	0.38	0.16	0
7	binary	-8.0	0.22	0.16	10.8
8	vapor	-8.0	0.22	0.16	100
9	vapor	64.8	1.32	0.16	100
10	vapor	62.5	1.32	0.30	100

CONCLUSIONS

A direct mixing of the exhaust air and ambient air may causes significant thermodynamic irreversibility. The MASHP with single-stage evaporator has a profit of only about 3.7 £/day. The EAHP for recovery of the exhaust air without being mixed with the ambient temperature seems to be more profitable. To enable a ratio *C* of 0.65, the ventilation can be controlled. Given a compressor capacity of the heat pump, intermittent ventilation leads to a higher *C* than the continuous ventilation. The most beneficial system is the MASHP with two-stage evaporators and the profit is 11.5 £/day, which is significantly higher than that of the EAHP and MASHP with single-stage evaporator.

Despite of the superior thermodynamic performance, one more compressor is required. The capital cost of the system will be higher than a conventional ASHP and the payback period may not be shorter. To reduce the cost of the compressors, the vapor injection compressor will be employed in the future work to recover the waste heat of exhaust air.

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NOMENCLATURE

COP - Coefficient of performance;

C_{ele} - Electricity price, £/kWh (about 0.20);

C_{heat} - Heat price, £/kWh (about 0.04);

C_p - Heat capacity;

h - Enthalpy, J/kg;

m_{am} - Flow rate of ambient air to be mixed, kg/s;

m_{ex} - Exhaust air flow rate, kg/s;

$Pro_{ex,ele}$ - Profit based on electricity reduction, £/day;

$Pro_{ex,heat}$ - Profit based on heat gain, £/day;

Q_{evap} - Evaporator input heat, W;

$Q_{ven,l}$ - Ventilation heat loss without recovery, W;

x - ratio of ambient air flow rate to exhaust air flow rate, %;

T_{am} - Ambient temperature, °C;

T_{ex} - Exhaust air temperature, °C;

$T_{mix,in}$ - Mixed air temperature, °C;

$T_{mix,out}$ - Mixed air temperature leaving the evaporator, °C;

ΔT_{min} - Minimum temperature difference (about 5°C);

W_{ele} - Electricity consumption.

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