

## **Application of The Exergy Concept to Ventilation Using Heat Recovery from Exhaust Air**

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### **SUMMARY**

This paper describes steady-state energy and exergy analyses on an instantaneous and on a daily basis for dwelling ventilation systems and domestic hot water DHW production systems. The ventilation uses mechanical exhaust with environmental air supply without heat recovery or balanced ventilation with heat recovery. The exhaust ventilation air is used to preheat DHW, using a heat exchanger or a heat pump. Energy and exergy demands in winter days for De bilt, the Netherlands, are presented at the component level, in terms of heat and electricity, for the systems. The results indicates that for these winter days the total energy demands, but not the total exergy demands, of balanced ventilation and DHW production with preheat by the exhaust ventilation air using a heat pump are lowest. The total exergy demands are dominated by exergy of electricity input to heat recovery unit and the heat pump.

### **INTRODUCTION**

Exhaust ventilation air has been used to preheat inlet ventilation air in balanced ventilation systems and as a heat source in other heating systems. In order to recover heat from the exhaust ventilation air, electric power is required. Since electricity input is small relative to the amounts of thermal energy recovered, such systems are efficient from an energy viewpoint. One important yet often overlooked aspect, however, is the difference in ‘quality’ between the high-grade electricity input and the lower grade thermal energy recovered.

Exergy analysis provides a common basis for evaluating forms of energy (e.g. thermal and electric), considering their abilities to produce work in relation to a given environment [1-3]. Exergy recognizes that energy carried by substances can only be used ‘down’ to the level given by the environment. Unlike energy, exergy is not subject to a conservation law.

This paper presents steady-state energy and exergy analyses on an instantaneous and on a daily basis for dwelling ventilation systems and DHW production systems. The exhaust ventilation air is used to preheat DHW. Analysis results, for De Bilt, the Netherlands, are used for discussion of design options for systems.

### **SYSTEM DESCRIPTION**

The systems to be studied are divided into 2 parts: dwelling ventilation systems and domestic hot water production systems.

## Dwelling ventilation systems

Energy and exergy analysis is performed for dwelling ventilation systems, using mechanical exhaust ventilation with environmental air supply without heat recovery (Figure 1a) and balanced ventilation with heat recovery (Figure 1b). The mechanical exhaust ventilation system uses a DC fan, Model: CVE ECO-fan 2 of Itho bv. [4]. The balanced ventilation system uses a DC Heat Recovery Unit, Model: HRU ECO-fan 3 S B of Itho bv. [5], containing two DC fans and a heat exchanger with high thermal effectiveness  $\varepsilon$ . Thermal effectiveness of the heat exchanger is calculated by interpolating from manufacturer's data [5] in relation to ventilation airflow rates (see Table 1). Figure 1 presents airflow rates and temperatures of ventilation and infiltration airflows of the systems. Air infiltration is also accounted for the analysis.

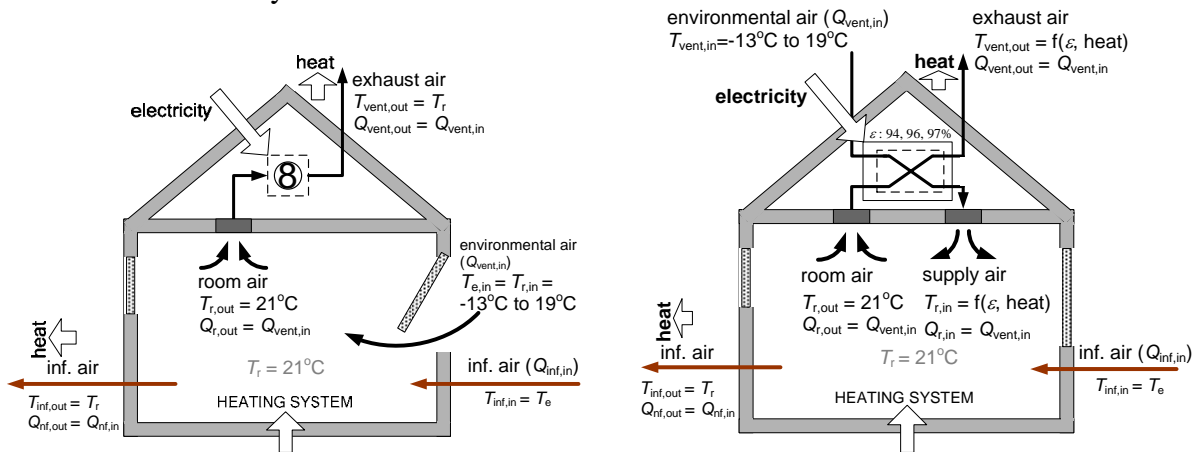


Figure 1. Dwelling ventilation systems. a) Mechanical exhaust ventilation with environmental air supply (left), b) Balanced ventilation with heat recovery (right).

The energy and exergy analysis of the ventilation systems assumes steady state operation and considers only dry air. Pressure difference between air entering and leaving the dwelling is also ignored. General calculation values are given in Table 2.

Table 1. Thermal effectiveness  $\varepsilon$  versus ventilation airflow rates  $Q_{\text{vent}}$ , for the DC HRU (left). Table 2. General calculation values (right). [6], [7]

$Q_{\text{vent}}$ [ $\text{m}^3/\text{s}$ ]	$\varepsilon$ [-]
0.063	0.94
0.042	0.96
0.028	0.97

Parameters	Values
Air density ( $\rho_{\text{air}}$ )	$1.23 \text{ kg m}^{-3}$
Spec. heat capacity of air ( $C_{p,\text{air}}$ )	$1.008 \text{ kJ kg}^{-1}\text{K}^{-1}$
Environmental air temperature ( $T_e$ )	from $-13^\circ\text{C}$ to $19^\circ\text{C}$
Room air temperature ( $T_r$ )	$21^\circ\text{C}$
Ventilation airflow rates ( $Q$ )	0.028, 0.042, 0.063 $\text{m}^3/\text{s}$

## Domestic hot water production systems

The DHW production systems use ground water at a constant temperature (assumed  $13^\circ\text{C}$ ). The DHW enters the dwelling with/without preheat by the exhaust air from the dwelling ventilation using a heat exchanger or a heat pump. Electricity is needed for DHW preheat if using the heat pump. The thermal effectiveness  $\varepsilon$  of the heat exchanger and the coefficient of performance COP of the heat pump are assumed 0.90 and 3.7 respectively. The energy efficiencies of the DHW preheat devices are assumed 1.0. The DHW devices operate when temperature of the exhaust ventilation air is higher than the maximum temperature between the inlet water and environment (if using the heat exchanger) and environment (if using the heat pump). The exhaust ventilation air is cooled until a temperature corresponding to the thermal effectiveness  $\varepsilon$  (if using the heat exchanger) and until environment  $T_e$  (if using the

heat pump). Air flow and water flow rates are assumed constant at the inlet and the outlet. Flow effects in the systems are also ignored in this study.

### Daily operation profiles of winter days in De Bilt, the Netherlands

Figure 2 shows three hourly environmental air temperature profiles, for winter days of maximum, minimum and intermediate mean daily environmental air temperatures. Climate data for De Bilt, the Netherlands are taken from the TMY2 weather data [8].

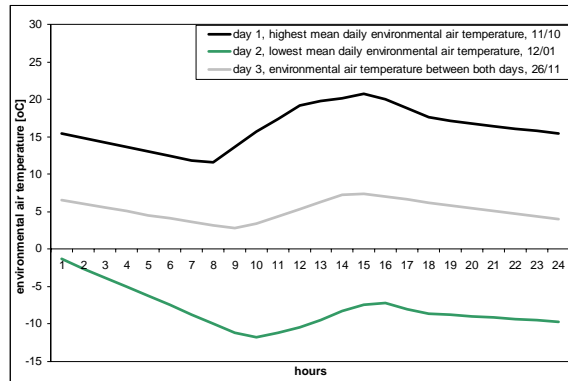


Figure 2. Hourly environmental air temperature profiles in 3 winter days in De Bilt.

The dwelling ventilation and the DHW use in the dwelling for the three days are hourly scheduled, given in Tables 3.

Table 3. Hourly operation plan: the dwelling ventilations (left); and the DHW use (right).

hour	ventilation airflow rate $Q_{vent}$ [ $m^3/s$ ]	hour	$Q_{DHW}$ [ $m^3/s$ ]	$T_{DHW}$ [ $^{\circ}C$ ]
0:00-8:00	0.028	00:00-07:00	0	0
8:00-9:00	0.042	07:00-09:00	0.00001	65
9:00-17:00	0.063	09:00-18:00	0	0
17:00-18:00	0.042	18:00-20:00	0.00001	65
18:00-24:00	0.028	20:00-24:00	0	0

## ENERGY AND EXERGY ANALYSIS

The comparison of design options for combination between dwelling ventilation and domestic hot water production is based on calculation of thermal energy and thermal exergy demands by ventilation and infiltration airflows in relation to environmental air temperature, and electricity input to ventilation unit and to DHW preheating device. The energy and exergy analysis is carried out on an instantaneous basis where  $T_e$  is between  $-13^{\circ}C$  and  $19^{\circ}C$  and on a daily basis for the representative winter days (Figure 2).

### Energy and exergy demands by infiltration airflows

Infiltration airflow rate  $Q_{inf}$  is calculated based on NEN 2867 [9]. The infiltration airflow rate relies on types of ventilation systems: mechanical exhaust ventilation  $Q_{inf,mv}$  (equation 1) and balanced ventilation  $Q_{inf,bv}$  (equation 2).

$$Q_{inf,me} = \left( \frac{q_{v10}}{500} \right) ((A + BCv + D\Delta T)q_{v10}) \quad (1)$$

$$Q_{inf,bv} = 0.8((A + BCv + D\Delta T)q_{v10}) \quad (2)$$

where  $q_{v10}$  [dm<sup>3</sup>/s] is the infiltration airflow rate expressed in a volume airflow rate  $q_v$  with pressure difference of 10 Pa,  $v$  [m/s] is the wind speed at 10 m above the ground level and  $\Delta T$  [K] is the temperature difference between the room air  $T_r$  and the environmental air  $T_e$ .  $A$ ,  $B$ ,  $C$  and  $D$  are the coefficients of the effect from turbulent airflow, shielding, partitioning of the air infiltration concerning the dwelling coating, and temperature difference respectively.

The following values are applied for the study.  $q_{v10}$  is 150 dm<sup>3</sup>/s for the dwelling with the mechanical exhaust ventilation and 80 dm<sup>3</sup>/s for the dwelling with the balanced ventilation.  $A$ ,  $B$ ,  $C$  and  $D$  are 0.02, 0.25, 0.20 and 0.004 respectively. These coefficient values are according to NEN 2867 [9] and for dwelling whose volume and height are 250 m<sup>3</sup> and 5.4 m, and having a normal shielding.  $v$ ,  $T_i$  and  $T_e$  are taken from the TMY2 weather data [8].

The infiltration airflow rate  $Q_{inf}$  is used for calculation of thermal energy  $En_{th,inf}$  and thermal exergy  $Ex_{th,inf}$  demands by infiltration airflows, using equations 3 and 4.

$$En_{th,inf} = \rho_{air} Q_{inf} C_{p,air} (T_{out} - T_{in}) \quad (3)$$

$$Ex_{th,inf} = \rho_{air} Q_{inf} C_{p,air} (T_{out} - T_{in} - T_e \ln(\frac{T_{out}}{T_{in}})) \quad (4)$$

where  $T_{out}$  and  $T_{in}$  are temperatures of the infiltration air leaving and entering the dwelling respectively. All the air temperatures are in Kelvin.

Since energy demand by the infiltration airflows accounts only the  $En_{th,inf}$ , therefore the total demands by the infiltration airflows are equal to the thermal demands for energy and exergy.

### Energy and exergy demands by ventilation airflows

Energy and exergy demands by ventilation airflows are obtained from thermal demand by ventilation airflows and electricity input to ventilation unit.

Thermal energy and exergy demands by ventilation airflows are calculated by applying equations 3 and 4. The supply air temperatures to the dwelling  $T_{r,in}$  for the balanced ventilation system is calculated by using the exchanger heat transfer effectiveness equation [10] and heat balance equation, assuming the same airflow rates though the HRU for the supply air and the exhaust air.

$$T_{r,in} = T_e + \varepsilon(T_r - T_e) \quad (5)$$

The inputs of exergy and energy for electricity are identical [11]. The electricity  $Pe$  is calculated by using the fan manufacturer data and the fan law [10] in equation 6.

$$Pe_1 = Pe_2 \left(\frac{\phi_2}{\phi_1}\right)^4 \left(\frac{Q_1}{Q_2}\right)^3 \left(\frac{\rho_1}{\rho_2}\right) \quad (6)$$

where  $\phi$  is the fan impeller diameter. The fan law is used to predict performance of the fan when test data (data with subscript 2 in equation 6) are available. The test data are used from the fan producer: Itho bv. [4], [5].

### Energy and exergy demands by domestic hot water

Thermal energy and exergy demands by DHW flow are functions of the DHW temperature required to supply to the dwelling  $T_{DHW}$  (assumed 65°C) and the preheated DHW temperature  $T_{preheat}$ . The demands are calculated by applying equations 3 and 4 and using inlet water temperature  $T_w$  as reference environment.  $T_{preheat}$  depends on types of the DHW preheat

devices and is calculated by using equation 7 (if using a heat exchanger,  $T_{\text{preheat,ex}}$ ) and equation 8 (if using a heat pump,  $T_{\text{preheat,hp}}$ ).

$$T_{\text{preheat,ex}} = \varepsilon \frac{C_{\text{min}}}{C_w} (T_{\text{vent,out}} - T_{\text{max}}) + T_w \quad (7)$$

$$T_{\text{preheat,hp}} = \left( \frac{\rho_{\text{air}} Q_{\text{air}} C_{p,\text{air}} (T_e - T_{e,\text{out}})}{\rho_w Q_w C_{p,w} (1 - \frac{1}{\text{COP}})} \right) + T_w \quad (8)$$

where  $C_{\text{min}}$  is the minimum value of the thermal capacities between the exhaust air from the dwelling ventilation and the inlet water,  $C_w$  is the thermal capacity of the inlet water and  $C_{p,w}$  is the specific heat capacity of the inlet water ( $4.19 \text{ kJkg}^{-1}\text{K}^{-1}$ ).  $T_{\text{max}}$  is the maximum temperature between inlet water and environment.  $T_w$  is the inlet water temperature and  $T_{e,\text{out}}$  is the air temperature of the exhaust ventilation air.

Electricity input to the heat pump is calculated as a different value between thermal energy supplied to heat pump by the exhaust ventilation air and thermal energy released by the heat pump to inlet water, because energy efficiency of the heat pumping process is assumed 1.0.

## RESULTS

This chapter presents energy and exergy analysis results for the dwelling ventilation and the DHW production systems on an instantaneous basis and on a daily basis.

### Total exergy demands by infiltration and ventilation airflows

Figures 3 and 4 show total exergy demands by the infiltration airflows  $Ex_{\text{inf}}$  and by the ventilation airflows  $Ex_{\text{vent}}$ , as a function of environmental air temperature  $T_e$ . In Figure 4, the mechanical exhaust ventilation with natural air supply and the balanced ventilation with heat recovery are called system A and system B respectively.

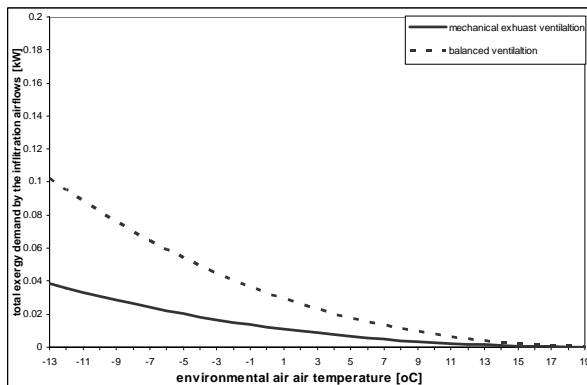


Figure 3. Total exergy demand by the infiltration airflow  $Ex_{\text{inf}}$  versus  $T_e$  (left).

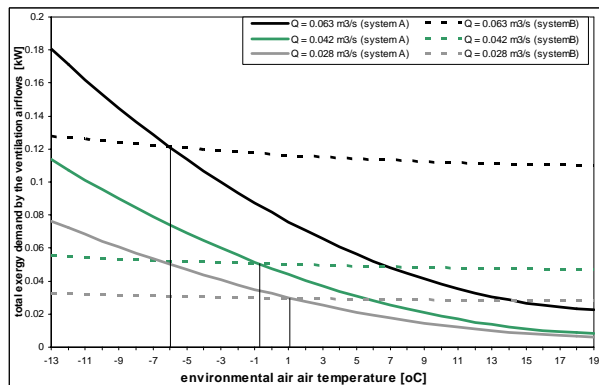


Figure 4. Total exergy demand by the ventilation airflows  $Ex_{\text{vent}}$  versus  $T_e$  (right).

In Figure 3, the  $Ex_{\text{inf}}$  lines for the ventilation systems vary nonlinearly with  $T_e$ . They rely on logarithmic term of ratio between inlet and outlet infiltration air temperatures (see equation 4), since  $Ex_{\text{inf}}$  is assumed equal to  $Ex_{\text{th,inf}}$ . In addition, the  $Ex_{\text{inf}}$  lines for the balanced ventilation with heat recovery is higher than the  $Ex_{\text{th,inf}}$  for the mechanical exhaust ventilation due to higher infiltration airflow rate in the entire range of  $T_e$ .

In Figure 4, the  $Ex_{\text{vent}}$  decreases for a given  $Q_{\text{vent}}$ , as  $T_e$  increases from  $-13^\circ\text{C}$  to  $19^\circ\text{C}$ . The  $Ex_{\text{vent}}$  lines (smooth ones) for the mechanical exhaust ventilation using the DC fan (called DC

fan line) are sharper than the  $Ex_{vent}$  lines (dashed ones) for the balanced ventilation with heat recovery using the DC HRU (called DC HRU line). The DC HRU lines are less sensitive in the range of  $T_e$  because the DC HRU requires mainly electricity, while the DC fan requires relatively more heat. The DC fan requires more exergy at lower  $T_e$ , and relatively less exergy as  $T_e$  increases towards the room air temperature  $T_r$ . electricity inputs to the DC fan and the DC HRU are given Table 4

Table 4. Electricity inputs to the DC fan and the HRU  $Pe$  versus ventilation airflow rates  $Q_{vent}$

hour	ventilation airflow rate $Q_{vent}$ [m <sup>3</sup> /s]	electricity inputs $Pe$ [W]
DC fan	0.028	6
	0.042	8
	0.063	22
DC HRU	0.028	28
	0.042	47
	0.063	110

The DC fan lines and the DC HRU lines intersect at a given environmental air temperature  $T_{e,intersect}$  for a given  $Q_{vent}$ . At  $T_e > T_{e,intersect}$ , using the mechanical exhaust ventilation with the DC fan results in less total exergy demand than using the balanced ventilation with the DC HRU at the same airflow rate.  $T_{e,intersect}$  increases when operating the ventilations at lower  $Q_{vent}$  since electricity input to the DC HRU is less.

### Total exergy demands by domestic hot water flow

Figures 5 shows the total exergy demand by DHW flow with preheat by exhaust ventilation air using the heat exchanger  $Ex_{DHW,ex}$  (Figures 5a) and the heat pump  $Ex_{DHW,hp}$  (Figures 5b) in relation to  $T_e$ , respectively. The mechanical exhaust ventilation with natural air supply and the balanced ventilation with heat recovery are called system A and system B respectively. The thin lines show the total exergy demand by DHW flow without preheat, as a reference for comparison to the  $Ex_{DHW,ex}$  and  $Ex_{DHW,hp}$ .

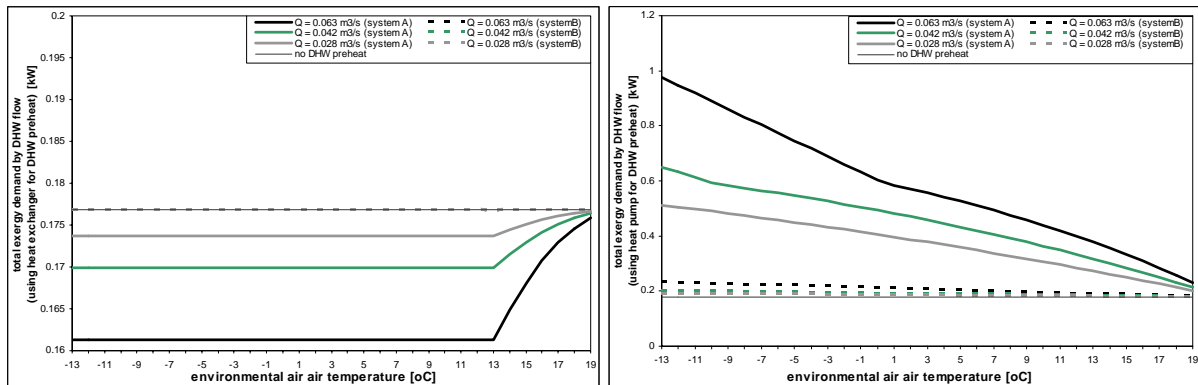


Figure 5. Total exergy demand by DHW flow with preheat by exhaust ventilation air via the heat exchanger  $Ex_{DHW,ex}$  (a, left) and via the heat pump  $Ex_{DHW,hp}$  versus  $T_e$  (b, right).

In Figure 5a, the  $Ex_{DHW,ex}$  lines for system B (dashed ones) are almost congruent to the reference line (no DHW preheat), because the  $En_{th}$  and  $Ex_{th}$  of the exhaust ventilation air are very small. Most of them are recovered in the HRU. The  $Ex_{DHW,ex}$  lines for system A (smooth ones) are constant when  $T_e < T_w$  according to the assumption that the DHW is preheated until  $T_{vent,out} = T_w$ . In addition, the  $Ex_{DHW,ex}$  depends on the  $Q_{vent}$ . For system A operating at  $Q_{vent} = 0.063, 0.042$  and  $0.028$  m<sup>3</sup>/s,  $Ex_{DHW,ex}$  are 91.24%, 96.06% and 98.24% of the  $Ex_{DHW}$  (no DHW preheat) respectively, when  $T_e < T_w$ . When  $T_e > T_w$ ,  $Ex_{DHW,ex}$  increases because the energy and exergy of the exhaust ventilation air decrease.

In Figure 5b, the  $Ex_{DHW, hp}$  decreases, as  $T_e$  increases from  $-13^{\circ}\text{C}$  to  $19^{\circ}\text{C}$  or  $Q_{vent}$  decreases. In addition, the  $Ex_{DHW, hp}$  lines for system A (smooth ones) are sharper than the  $Ex_{DHW, hp}$  lines for system B (dashed ones), because more heat from system A are used in DHW preheat. This costs more electricity, which is high-graded energy, by the heat pump.

The  $Ex_{DHW, hp}$  lines for system A (smooth ones) have more than one slope in the range of  $T_e$ . At the lower slope (in this case  $T_e > 0^{\circ}\text{C}$  for  $Q_{vent} = 0.063 \text{ m}^3/\text{s}$ ), the heat pump heats DHW until a temperature less than  $T_{DHW}$  and additional heat is then required. On the other hand, at the sharper slope, the heat pump could heat DHW to  $T_{DHW}$  only by using electricity. The  $Ex_{DHW, hp}$  lines for system B (dashed ones) have only one slope in the range of  $T_e$ , since the heat pump could heat DHW until a temperature less than  $T_{DHW}$ .

### The total energy and total exergy demands by the dwelling ventilation and the domestic hot water production systems in winter days

This item presents examples of cumulative energy and exergy demands for De Bilt (NL) for three representative winter days (Figure 2). Figure 6 shows the energy  $En$  and exergy  $Ex$  demands of each design option, in terms of heat and electricity, at the component level (noted that scales of the Figures 6a and 6b are different). The names of the design options consists of an alphabet (A: the mechanical exhaust ventilation with natural air supply; B: the balanced ventilation with heat recovery) and a number (1: no DHW preheat; 2: DHW preheat using the heat exchanger; 3: preheat using the heat pump).

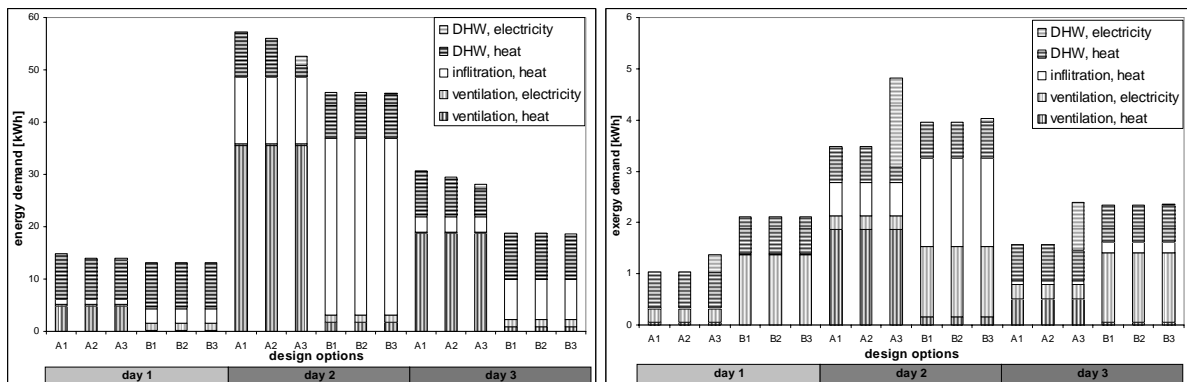


Figure 6. Energy demands (a, left) and exergy demands (a, left) of the design options.

In Figure 6a, option B3 is the most energy efficient one to use in the winter days because it requires the least energy  $En$  among the other options studied. The  $En$  of option B3 in the part of ventilation and infiltration depends on environment. The demand is highest on day 2 and lowest on day 1. Option B3 saves 0.45% of the  $En$  in the DHW production of option B1 (no DHW preheat) because of small heat of the exhaust ventilation air used in the DHW preheat.

In Figure 6b, option A2 is the most exergy efficient one because it requires the least exergy  $Ex$  among the other options studied, for the winter days. In the part of mechanical ventilation, options A1-A3 require higher thermal exergy than options B1-B3 but much lower electric exergy. In the part of infiltration, options A1-A3 require lower thermal exergy than options B1-B3. In total, options A1-A3 require lower total exergy than options B1-B3 in the parts of mechanical ventilation and infiltration. In the part of the DHW production, the  $Ex$  of option A2 are smaller than the demand of option A1 (without the DHW preheat), but for option B2 the  $Ex$  is quite similar to the  $Ex$  of option A1. However, in the same part the  $Ex$  of options A3 and B3 (with DHW preheat using the heat pump) are higher than the demand of option A1 (without DHW preheat), due to electricity input to the heat pump. This is because the heat

pump uses electricity to pump low-graded heat. Nevertheless, options A3 saves more thermal exergy than options A2. Electric exergy input to the heat pump should be reduced. Besides, options B2 and B3 save very small amounts of the thermal exergy in the DHW production.

## **CONCLUSION**

This paper presents steady-state energy and exergy analyses on an instantaneous and on a daily basis for dwelling ventilation systems and DHW production systems. Analysis results, for De Bilt, the Netherlands, are used for discussion of design options for systems.

On an instantaneous basis, the exergy demands by infiltration airflows for the balanced ventilation with heat recovery are higher than the exergy demands for the mechanical exhaust ventilation in the entire range of environmental air temperature. But this is not for the exergy demands by ventilation airflows because substantial shares of electric exergy input to ventilation unit are big and dominate the total exergy demands. The exergy demands by ventilation airflows for the balanced ventilation with heat recovery are lower than the exergy demands for the mechanical exhaust ventilation, when environmental air temperature close to room air temperature. In the DHW production part, using the heat exchanger for the DHW preheat with the exhaust air from the balanced ventilation with heat recovery saves a small amount of the exergy demands of DHW production, due to a small amount of heat from the air. Preheating the DHW with the exhaust air from the mechanical exhaust ventilation is better according to a bigger amount of the heat. DHW production with DHW preheat using the heat pump requires more exergy than DHW production without the DHW preheat, because the heat pump uses electricity to pump low-graded heat. The exergy of electricity input to the heat pump is larger than the exergy of heat recovered in the DHW preheat.

On a daily basis, for the winter days the total energy demands, but not the total exergy demands, of balanced ventilation and DHW production with preheat by the exhaust ventilation air using a heat pump are lowest. The total exergy demands are dominated by exergy of electricity input to heat recovery unit and the heat pump.

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