APPLICATION OF THE EXERGY CONCEPT TO DESIGN EFFICIENT MECHANICAL EXHAUST VENTILATION SYSTEMS

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ABSTRACT

This work presents energy and exergy comparison of several design options for combination between dwelling ventilation and domestic hot water production. The dwelling ventilation uses mechanical exhaust with natural air supply (without heat recovery) or balanced ventilation with heat recovery. The outlet ventilation air is used as a heat source for domestic hot water, by using a heat exchanger or a heat pump. Energy and exergy demands for DeBilt, the Netherlands, are presented at the component level, in terms of heat and electricity, for the dwelling ventilation systems and for the domestic hot water production systems. Energy and exergy results are presented in seasonal and annual. Exergy uses for the energy (heat and electricity) productions are presented as well for the system components, to determine whether the main exergy use comes from the energy supply side or from the energy demand side. Recommendations for design are finally made.

KEYWORDS

Exergy, Mechanical exhaust ventilation system

INTRODUCTION

Exhaust ventilation air has a potential level of energy and exergy, while it might be regarded as (local) waste from buildings. At present, exhaust ventilation air has been used to preheat inlet ventilation air in balanced ventilation systems and as a heat source for 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 different forms of energy (e.g. thermal and electric), considering their different abilities to produce work in relation to a given environment (Boelman 2002, Asada and Boelman 2004, Wall 1990, Rosen and Dincer 2001, Ala-Juusela (ed.) 2004). Exergy recognizes that the energy that is carried by substances can only be used 'down' to the level that is given by the environment. Unlike energy, exergy is not subject to a conservation law.

This paper presents steady-state energy and exergy analyses for dwelling ventilation systems and domestic hot water systems. The exhaust ventilation air is used to preheat domestic hot water. Energy and exergy demands of the systems for DeBilt, the Netherlands, are presented in seasonal and annual, and discussed design options for dwelling ventilation systems and domestic hot water production 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 1) and balanced ventilation with heat recovery (Figure 2). Air infiltration is also accounted for the analysis.

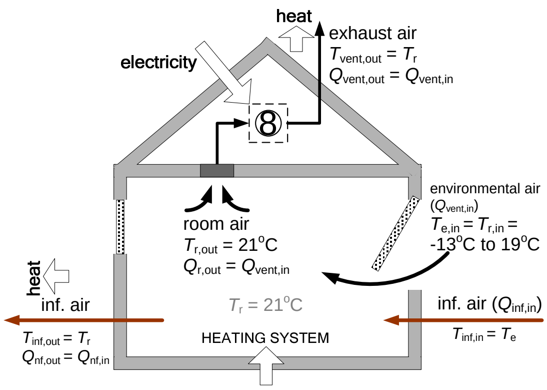

Figure 1 Mechanical exhaust ventilation with environmental air supply

The mechanical exhaust ventilation system uses a DC fan, Model: CVE ECO-fan 2 of Itho bv. (2005a). Natural air enters the dwelling by infiltration at $Q_{inf,in}$ and via openings at $Q_{vent,in}$. $Q_{inf,in}$ depends on indoor and outdoor climate conditions (see equation 1 in the next chapter). $Q_{vent,in}$ depends on dwelling occupation conditions. It is considered in 3 values for the analysis (see Table 1). A part of the air leaves the dwelling by infiltration $Q_{inf,out}=Q_{inf,in}$. The rest goes to the DC fan and leaves the DC fan at the same airflow rate $Q_{r,out}=Q_{vent,out}=Q_{vent,in}$. Temperatures of supply air are equal to environment for infiltration and ventilation $T_{inf,in}=T_{vent,in}=T_e$. The infiltration air goes out the dwelling at room air temperature $T_{inf,out}=T_r$. The ventilation air goes through the exhaust fans at room air temperature $T_{r,out}=T_{vent,out}=T_r$.

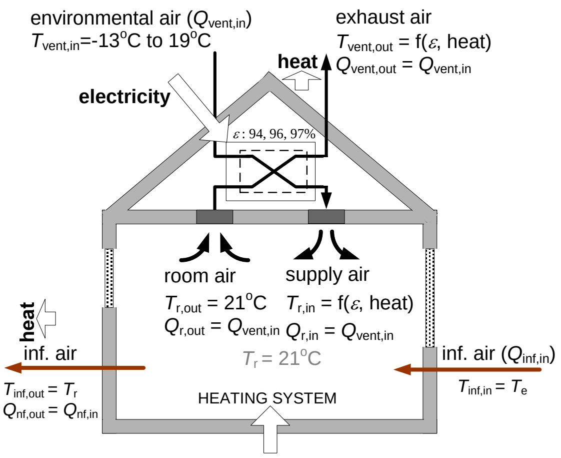

Figure 2 Balanced ventilation with heat recovery

The balanced ventilation system uses a DC Heat Recovery Unit, Model: HRU ECO-fan 3 S B of Itho bv. (2005b), containing two DC fans and a heat exchanger with high thermal effectiveness ε. Thermal effectiveness of the heat exchanger and electricity input to the fans are calculated by interpolating from manufacturer's data (Itho bv. 2005b) in relation to ventilation airflow rates (see Table 1). Environmental air enters the dwelling by infiltration at $Q_{inf,in}$ and via the HRU at $Q_{r,in}=Q_{vent,in}$. $Q_{inf,in}$ is calculated according to equation 2 (see the next chapter). Values of the $Q_{vent,in}$ for the analysis are the same ones used in the mechanical exhaust ventilation. The infiltration air leaves the dwelling at $Q_{inf,out}=Q_{inf,in}$. The ventilation air leaves the dwelling via the HRU at the same airflow rate $Q_{r,out}=Q_{vent,out}=Q_{vent,in}$. The environmental air to the HRU gains heat from room air entering the heat exchanger in the HRU at room air temperature $T_{r,out}=T_r$ and leaving the dwelling at temperature $T_{vent,out}$. Temperatures of the infiltration air entering the dwelling is equal to environment $T_{inf,in}=T_e$ and leaving the dwelling is equal to room air $T_{inf,out}=T_r$.

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

Table 1 Thermal effectiveness ε versus ventilation airflow rates Q, for the DC HRU

Q [m^3/s]	**ε [–]**
0.063	0.94
0.042	0.96
0.028	0.97

Table 2 General calculation values [CEN 2000, CEN 2005]

PARAMETERS	**VALUES**
Air density (ρ_{air})	1.23 kg m^{-3}
Spec. heat capacity of air ($C_{p,air}$)	1.008 kJ kg^{-1}K^{-1}
Environmental air temperature (T_e)	from -13°C to 19°C
Room air temperature (T_r)	21°C
Ventilation airflow rates (Q_{vent})	0.028, 0.042, 0.063 m^3/s

The ventilation systems operate in different levels of the ventilation airflow rates, for a day. The ventilation airflow rates are hourly scheduled, given in Table 3.

Table 3 Hourly operation plan of the dwelling ventilations.

HOUR	**VENTILATION AIRFLOW RATE [M^3/S]**
0:00-8:00	0.028
8:00-9:00	0.042
9:00-17:00	0.063
17:00-18:00	0.042
18:00-24:00	0.028

Heating system for the ventilations uses a gas-fired boiler and uses natural gas as fuel. Energy efficiency of the heating supply system is assumed at one: there is no thermal energy loss of the system.

Domestic hot water production systems

Domestic hot water DHW is supplied to the dwelling as an hourly operation plan, given in Table 4. The DHW production uses ground water at a constant temperature (assumed 13°C; as a temperature of tap water in the Netherlands in a general case) and the same gas-fired boiler as one for the heating system for the ventilations.

The DHW enters the dwelling with/without preheat by the exhaust air from the dwelling ventilation using either a heat exchanger or a heat pump. The thermal effectiveness ε 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 constant at one: there is no thermal energy loss of the devices.

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 ε (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.

Table 4 Hourly operation plan for the DHW use

HOUR	Q_{DHW} [m^3/s]	T_{DHW} [^{o}C]
00:00-07:00	0	0
07:00-09:00	0.00001	65
09:00-18:00	0	0
18:00-20:00	0.00001	65
20:00-24:00	0	0

Characteristics of the climate in De Bilt, the Netherlands

In this paper, the climate data for De Bilt, the Netherlands are used. They are taken from the TMY2 weather data (NREL 1995). In Figure 3, the gray-colored areas and the white-colored area represent environmental air temperatures in the heating and non-heating seasons respectively. The non-heating season is defined as the period from 1 May to 30 September of the TMY (3672 hours) and the heating season is defined as the remaining period of the TMY (5088 hours).

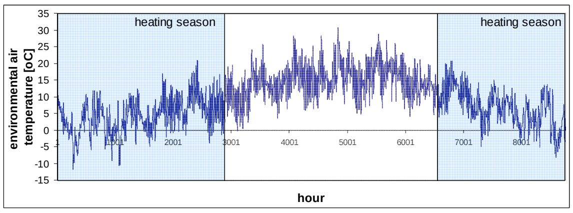

Figure 3. Hourly environmental air temperature profile for a year in De Bilt, the Netherlands (data from TMY2 data: NREL 1995)

The environmental air temperatures are between -11.77^{o}C and 20.78^{o}C in the heating season and between -0.31^{o}C and 30.71^{o}C in the non-heating season. The average air temperature is 5.31^{o}C with the standard deviation of 5.28 in the heating season and 14.99^{o}C with the standard deviation of 4.84 in the non-heating season.

ENERGY AND EXERGY ANALYSIS

The comparison of design options for combination between dwelling ventilation and domestic hot water production is based on the following method to calculate 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 for two defined periods: the heating and non-heating seasons. The dwelling is heated only in the heating season, and thus the DHW is preheated only in the season. The DHW production and the air ventilation operate for the whole year, scheduled according to Table 3 and Table 4.

Energy and exergy demands by infiltration airflows

Infiltration airflow rate Q_{inf} is calculated based on NEN 2867 (1989). 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).

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

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

where q_{v10} [dm^3/s] is the infiltration airflow rate expressed in a volume airflow rate q_v with pressure difference of 10 Pa, ν [m/s] is the wind speed at 10 m above the ground level and ΔT [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 coating of the dwelling, and temperature difference respectively.

The following values are applied for the study. q_{v10} is equal to 150 dm^3/s for the dwelling equipped with the mechanical exhaust ventilation and 80 dm^3/s for the dwelling equipped 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 (1989) and for dwelling whose volume and height are 250 m^3 and 5.4 m, and having a shielding in a normal circumstance. ν , T_i and T_e are taken from the TMY2 weather data (NREL 1995).

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.

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

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

where ρ_{air} is the air density (1.23 kg m^{-3}), $C_{p,air}$ is the specific heat capacity of air (1.008 kJ kg^{-1}K^{-1}), T_e is environmental air temperature and T_{out} and T_{in} are temperatures of the infiltration air leaving and entering the dwelling ($T_{inf,in}$=T_e and $T_{inf,out}$=T_r) respectively. All the air temperatures are in Kelvin.

Since there is only thermal energy used for treatment of the infiltration air, therefore the total demands 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 $T_{\mathrm{r,in}}$ is between -13$^{\mathrm{o}}$C and 19$^{\mathrm{o}}$C if using the mechanical exhaust ventilation. If using the balanced ventilation system with heat recovery, $T_{\mathrm{r,in}}$ is calculated by using the exchanger heat transfer effectiveness equation (ASHRAE 1993) and heat balance equation, assuming the same airflow rates though the HRU for the supply air and the exhaust air.

$$T_{\mathrm{r,in}} = T_{\mathrm{e}} + \varepsilon(T_{\mathrm{r}} - T_{\mathrm{e}}) \qquad (5)$$

The inputs of exergy and energy for electricity are identical because electric energy can in theory be totally converted into mechanical work. The electricity Pe is calculated by using the fan manufacturer data (a relation between the total pressure in the fan, an airflow rate through the fan, and electricity demand by the fan) and the fan law in equation 6.

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

where ϕ 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 come from the data at the intersection points where the system line intercepts the working line of the fans, in the graph of fan static pressure versus airflow rate given by the fan producer: used Itho bv. (2005a, 2005b).

Energy and exergy demands by domestic hot water flow

Thermal energy and exergy demands by domestic hot water DHW flow are functions of the DHW temperature supplied to the dwelling T_{DHW} (assumed 65$^{\mathrm{o}}$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 (heat exchanger and heat pump). It is calculated by using equation 7 (if using a heat exchanger, $T_{\mathrm{preheat,ex}}$) and equation 8 (if using a heat pump, $T_{\mathrm{preheat,hp}}$).

$$T_{\mathrm{preheat,ex}} = \varepsilon\frac{C_{\mathrm{min}}}{C_{\mathrm{w}}}\left(T_{\mathrm{vent,out}} - T_{\mathrm{max}}\right) + T_{\mathrm{w}} \qquad (7)$$

$$T_{\mathrm{preheat,hp}} = \left(\frac{\rho_{\mathrm{air}}Q_{\mathrm{air}}C_{\mathrm{p,air}}(T_{\mathrm{e}} - T_{\mathrm{vent,out}})}{\rho_{\mathrm{w}}Q_{\mathrm{w}}C_{\mathrm{p,w}}(1 - \frac{1}{\mathrm{COP}})}\right) + T_{\mathrm{w}} \qquad (8)$$

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

Electricity input to the heat pump is calculated as a different value between the thermal energy supplied to the heat pump by the exhaust ventilation air and the thermal energy released by the heat pump to the inlet water. There is no energy loss in the heat pumping process, since the energy efficiency of the process is assumed constant at one.

Exergy loss of energy supply system

In this paper, the energy (heat and electricity) supply system is considered from primary energy transformation to energy delivery processes. Exergy loss of the energy supply system is a different value between primary exergy input and exergy demand. The primary exergy input Ex_{pri} is calculated by using equation 9.

$$Ex_{\mathrm{pri}} = En_{\mathrm{supply}}F_{\mathrm{p}}F_{\mathrm{q,s}} \qquad (9)$$

where En_{supply} is the energy supply to the dwelling, F_{p} is primary energy factor and $F_{\mathrm{q,s}}$ is quality factor of source. It is assumed that energy supply is equal to energy demand: i.e. there is no energy loss during energy delivery process. F_{p} and $F_{\mathrm{q,s}}$ are assumed 1.1 and 0.9 for heat production (Schmidt 2004) and 2.58 and 0.9 for electricity production (using these values from the French mixture electricity production, Schmidt 2004). The fuel for the energy production is natural gas.

RESULTS AND DISCUSSIONS

The energy and exergy demands of the dwelling ventilation systems and the domestic hot water production systems for De Bilt, the Netherlands, are presented in Table 5 and Table 6 respectively, at the system component level in terms of heat and electricity. The demands are calculated in an hourly basis and repeatedly accumulated for the seasonal and yearly periods.

In terms of energy, option B3 is the most energy-efficient one to use in the heating season because it requires the least energy among the other options studied.

Table 5 Energy demands of the different ventilation and domestic hot water production systems in De Bilt, the Netherlands

	HEATING SEASON	NON-HEATING SEASON	YEAR
mechanical ventilation			
system A: mechanical exhaust			
energy for heat [kWh]	3867.03		3867.03
energy for electricity [kWh]	58.28	42.06	100.34
system B: balance ventilation			
energy for heat [kWh]	171.74		171.74
energy for electricity [kWh]	287.27	207.32	494.58
infiltration			
system A: mechanical exhaust			
energy for heat [kWh]	1205.98		1205.98
system B: balance ventilation			
energy for heat [kWh]	1715.17		1715.17
DHW production			
system A: mechanical exhaust			
option 1: no preheat + boiler			
energy for heat [kWh]	1847.62	1333.43	3181.05
option 2: heat recovery + boiler			
energy for heat [kWh]	1625.96	1333.43	2959.38
option 3: heat pump + boiler			
energy for heat [kWh]	1487.43	1333.43	2820.85
energy for electricity [kWh]	97.35	0.00	97.35
system B: balance ventilation			
option 1: no preheat + boiler			
energy for heat [kWh]	1847.62	1333.43	3181.05
option 2: heat recovery + boiler			
energy for heat [kWh]	1843.81	1333.43	3177.24
option 3: heat pump + boiler			
energy for heat [kWh]	1840.97	1333.43	3174.40
energy for electricity [kWh]	1.80	0.00	1.80
Total energy [kWh]			
option A1	6978.91	1375.49	8354.40
option A2	6757.24	1375.49	8132.73
option A3	6716.06	1375.49	8091.55
option B1	4021.80	1540.74	5562.54
option B2	4017.99	1540.74	5558.74
option B3	4016.95	1540.74	5557.69

Table 6 Exergy demands of the different ventilation and domestic hot water production systems in De Bilt, the Netherlands

	HEATING SEASON	NON-HEATING SEASON	YEAR
mechanical ventilation			
system A: mechanical exhaust			
exergy for heat [kWh]	114.63		114.63
exergy for electricity [kWh]	58.28	42.06	100.34
system B: balance ventilation			
exergy for heat [kWh]	9.47		9.47
exergy for electricity [kWh]	287.27	207.32	494.58
infiltration			
system A: mechanical exhaust			
exergy for heat [kWh]	37.35		37.35
system B: balance ventilation			
exergy for heat [kWh]	53.11		53.11
DHW production			
system A: mechanical exhaust			
option 1: no preheat + boiler			
exergy for heat [kWh]	149.96	108.23	258.19
option 2: heat recovery + boiler			
exergy for heat [kWh]	147.56	108.23	255.79
option 3: heat pump + boiler			
exergy for heat [kWh]	143.51	108.23	251.74
exergy for electricity [kWh]	97.35	0.00	97.35
system B: balance ventilation			
option 1: no preheat + boiler			
exergy for heat [kWh]	149.96	108.23	258.19
option 2: heat recovery + boiler			
exergy for heat [kWh]	149.94	108.23	258.17
option 3: heat pump + boiler			
exergy for heat [kWh]	149.91	108.23	258.13
exergy for electricity [kWh]	1.80	0.00	1.80
Total exergy [kWh]			
option A1	360.22	150.29	510.51
option A2	357.82	150.29	508.11
option A3	451.12	150.29	601.41
option B1	499.81	315.55	815.36
option B2	499.80	315.55	815.34
option B3	501.56	315.55	817.10

Option B3 is, however, not the best one to use in the non-heating season because in the season the option requires more energy than the other due to highly significance of electricity input to ventilation unit. In addition, there is no DHW preheat and no heat required by ventilation and infiltration airflows in the non-heating season.

The total energy demands of options B1-B3 are lower than options A1-A3 in the heating season, because large amounts of thermal energy are saved by using the HRU. In the non-heating season, the total energy demands of options B1-B3 are, however, higher than options A1-A3, due to energy inputs to the HRU. For the year, the total energy demands of options A1-A3 are higher, because the large amounts of thermal energy saved by using the HRU in the heating season are much higher than the energy input to the HRU in the non-heating season.

Using option A2 and option A3, instead of option A1, save 6.97% and 8.26% of the total energy demand in the part of the DHW production respectively. Using option B2 and option B3, instead of option B1, save only 0.12% and 0.15% of the total energy demand in the part of the DHW production respectively. This is because the exhaust air from the balanced ventilation has smaller heat due to heat recovery in the ventilation and temperature of the heat close to environment.

In terms of exergy, option A2 is the most exergy-efficient one because it requires the least total exergy among the other options studied, for the seasons and the year. 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 mechanical ventilation, ratios between thermal exergy and thermal energy demands Ex_{th}/En_{th} of options A1-A3 (114.63/3867.03=0.030) are lower than of options B1-B3 (9.47/171.74=0.055). This implies that for mechanical ventilation options B1-B3 are more suitable to use higher-graded thermal energy than options A1-A3. In the part of infiltration, the ratios of options A1-A3 and options B1-B3 are similar.

The annual total exergy demands in the part of the DHW production of options A2 and B2 (with DHW preheat using the heat exchanger) are smaller than the demand of option A1 (without the DHW preheat): 1.600% for option A2 and 0.012% for option B2. However, the annual total exergy demands in the part of the DHW production of options A3 and B3 (with DHW preheat using the heat pump) are higher than the demand of option A1 (without the DHW preheat): 60.61% for option A3 and 1.16% for option B3. This is because the heat pump uses electricity to pump low-graded heat. Nevertheless, options A3 and B3 save more thermal exergy than options A2 and B2. Exergy of electricity input to the heat pump should be reduced.

Exergy losses $Ex_{supply,loss}$ and primary exergy inputs Ex_{pri} of the energy (heat and electricity) supply system (equation 9) are given in Table 7.

Table 7 Exergy losses $Ex_{supply,loss}$ and primary exergy inputs Ex_{pri} in the energy supply system in De Bilt, the Netherlands

		$Ex_{supply,loss}$		Ex_{pri} [kWh]
		[kWh]	[% of Ex_{pri}]	
mechanical ventilation				
system A: mechanical exhaust	heat	3713.72	97.01%	3828.36
	electricity	132.6495	56.93%	232.9895
system B: balance ventilation	heat	160.55	94.42%	170.03
	electricity	653.8348	56.93%	1148.415
infiltration				
system A: mechanical exhaust	heat	1156.57	96.87%	1193.92
system B: balance ventilation	heat	1644.90	96.87%	1698.02
DHW production				
system A: mechanical exhaust				
option 1: no preheat + boiler	heat	2891.05	91.80%	3149.24
option 2: heat recovery + boiler	heat	2674.00	91.27%	2929.79
option 3: heat pump + boiler	heat	2540.91	90.99%	2792.64
	electricity	128.6967	56.93%	226.0467
system B: balance ventilation				
option 1: no preheat + boiler	heat	2891.05	91.80%	3149.24
option 2: heat recovery + boiler	heat	2887.30	91.79%	3145.47
option 3: heat pump + boiler	heat	2884.52	91.79%	3142.65
	electricity	2.3796	56.93%	4.1796
Total Thermal exergy				
option A1		7761.34	94.98%	8171.51
option A2		7544.30	94.87%	7952.07
option A3		7411.20	94.83%	7814.92
option B1		4696.51	93.61%	5017.28
option B2		4692.75	93.60%	5013.51
option B3		4689.98	93.60%	5010.70
Total electric exergy				
option A1		132.6495	56.93%	232.9895
option A2		132.6495	56.93%	232.9895
option A3		261.3462	56.93%	459.0362
option B1		653.8348	56.93%	1148.415
option B2		653.8348	56.93%	1148.415
option B3		656.2144	56.93%	1152.594

The results show that the total $Ex_{th,supply,loss}$ for the design options are all more than 93.60% of the total $Ex_{th,pri}$. Using balanced ventilation causes relatively smaller $Ex_{th,supply,loss}$ than using mechanical exhaust ventilation. In addition, DHW preheat could reduce the exergy, less than 0.15%. The total $Ex_{pe,supply,loss}$ for the design options are constant at 56.93% of the total $Ex_{pe,pri}$, because of the assumption that there is no energy loss during energy delivery process and also energy and exergy of electicity are identical: i.e. energy and exergy of electricity supply to the dwelling are similar. The total $Ex_{pe,supply,loss}$ therefore only depends on primary energy factor and quality factor of source, which are assumed constant for this study.

CONCLUSION

This paper presents a steady-state energy and exergy analyses for dwelling ventilation systems in combination with domestic hot water systems. The exhaust ventilation air is used to preheat domestic hot water. Energy and exergy demands of the systems for DeBilt, the Netherlands, are presented and discussed on a two seasonal and an annual basis.

In terms of energy, the most energy-efficient design option is option B3 (balanced ventilation coupled to DHW production using the ventilation exhaust air and a heat pump for (pre)heating of DHW) for the heating season and option A1 (mechanical exhaust ventilation coupled to DHW production without (pre)heating of DHW) for the non-heating season. In the heating season, option B3 could save 0.15% from the total energy demand of the DHW production in option A1. In the non-heating season, option A1 is recommended because it requires the smallest total energy demand: the demand only depends on the electricity input to the ventilation unit and there is no DHW (pre)heating (see Table 5).

In terms of exergy the amounts of electricity input to the ventilation units and the DHW preheating devices are much more significant than in terms of energy, because electricity has a higher exergy value than thermal energy. Option A2 (mechanical exhaust ventilation coupled to DHW production using the ventilation exhaust air and a heat exchanger for (pre)heating of DHW) is the most exergy-efficient one because it requires the least amount of exergy input, for both seasons and the whole year. However, this option requires more total energy demands than option B3, and therefore option A2 is not the one recommended. As a result of the exergy analysis, the electricity input to the ventilation unit and to the heat pump should be reduced in both seasons for option B3.

The design option B3 is the best one among the options considered over the whole year, because it needs less energy. Furthermore, to reduce energy demand of this option further, it could make sense to use the HRU only in the heating season and to bypass the HRU by the ventilation air in the non-heating season to reduce the electricity input to the ventilation unit. To reduce exergy demand of the option, using a heat recovery unit and a heat pump which consume low electricity should be considered as the first priority.

Exergy losses ($Ex_{th,loss}$ and $Ex_{pe,loss}$) of the design options studied mainly come from the energy supply side. The total $Ex_{th,supply,loss}$ for the design options are all more than 93.60% of the total $Ex_{th,pri}$ and the total $Ex_{pe,supply,loss}$ for the design options are constant at 56.93% of the total $Ex_{pe,pri}$. Using the balanced ventilation causes relatively smaller $Ex_{th,supply,loss}$ than using the mechanical exhaust ventilation. In the case of DHW production, preheating of DHW could reduce $Ex_{supply,loss}$ of the DHW production in relatively small percentage of Ex_{pri} to the case without preheating of DHW, less than 0.15%.

Furthermore, the waste DHW stream could be a potential source for preheating fresh DHW or, eventually, could be used for (partially) heating of ventilation air. This study does not consider the use of the waste DHW stream yet, but this is an interesting topic to study in the near future.

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NOMENCLATURE

A	Coefficient of the effect from turbulent airflow
B	Coefficient of the effect from shielding
C	Coefficient of the effect from partitioning of the air infiltration concerning the coating of the dwelling
COP	Coefficient of performance
c_p	Specific heat capacity
D	Coefficient of the effect from temperature difference
DHW	Domestic hot water
En	Energy
Ex	Exergy
F_p	Primary energy factor
$F_{q,s}$	Quality factor of source
HRU	Heat recovery unit
Pe	Electricity
Q	Airflow rate; or water flow rate
q_v	Volume airflow rate
q_{v10}	Infiltration airflow rate expressed in a volume airflow rate qv with pressure difference of 10 Pa
T	Temperature
TMY	Typical meteorological year

Greek letters

ε	Thermal effectiveness
ϕ	Fan impeller diameter
ρ	Density
ν	Wind speed

Subscripts

air	Air
bv	Balanced ventilation
DHW	Domestic hot water
e	Environment
ex	Heat exchanger
hp	Heat pump
in	Inlet
inf	Infiltration
min	Minimum
mv	Mechanical exhaust ventilation
out	Outlet
pe	Electricity
preheat	Preheated air
pri	Primary energy
r	Room air
supply	Supply
th	Thermal
vent	Ventilation
w	Water

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