# Energetic Impact of Temperature Gradients in Heat Recovery in Ventilation in Dwellings

*Josué* Borrajo Bastero[a], *Eline* Himpe[a], *Jelle* Laverge[a]

[a] Research Group Building Physics, Department Architecture and Urban Planning, Faculty of Engineering and Architecture, Ghent University, Belgium

**Abstract.** The installation of air-to-air heat exchangers in ventilation systems can contribute to reducing the heating demand of the building by recovering the heat from the exhaust air. When there is an active heating demand, the recovered heat becomes useful to reduce that demand. However, as the heat exchanger works independently of the heating needs of the dwelling, the recovered heat may not contribute to the reduction in the heating demand. Furthermore, the indoor temperature gradients in the various rooms/zones of the dwelling can influence the heat recovered by the ventilation system and, consequently, the useful recovered heat. A suitable method to evaluate the effect is based on dynamic multi-zone building energy simulations. In this work, a residential building typology has been modelled with eleven ventilations systems. Some of them work with fixed airflows and others are commercial smart ventilation systems with variable flow rates controlled by $CO_2$, VOC and humidity sensors. Various systems do not have a heat exchanger due to their ventilation type. Still, the study was extended to those, considering that the unwasted heat as a consequence of using a smart control would be equivalent to the presence of a heat exchanger. The building was heated with two heating strategies: uniform heating in all the zones except for the attic and non-uniform heating with a specific setpoint for three different types of zones. To evaluate the impact of the heat recovery, the annual heating demand and the recovered heat are compared between scenarios with different effectivenesses for the heat exchanger and, to evaluate the effect of the temperature gradients, the results are compared between the two different heating strategies: uniform and non-uniform. The results for the case presented in this study show that centralized heat recovery systems perform better in the scenarios with uniform heating, while distributed ventilation systems perform similarly in both heating scenarios.

**Keywords.** Temperature gradients, heat recovery, heat exchanger, heating demand, ventilation.
**DOI**: https://doi.org/10.34641/clima.2022.114

## 1. Introduction

Ventilation systems are present in dwellings with the sole purpose of having an acceptable indoor air quality by removing stale air from indoors, contributing to improving people's health, comfort and productivity. The ventilation concept, however, is quite inefficient, since it requires extracting the air from inside the building to be replaced with outdoor air, which, hopefully, would have a lower concentration of pollutants. An efficient alternative would be a technology to remove the pollutants from the indoor air by filtering or capturing them. However, these alternatives are not competitive yet, especially to capture $CO_2$ [1]. The indoor air has been treated by the conditioning system to guarantee a certain level of thermal comfort for the occupants. Then, it is extracted and replaced by air that most of the time does not have the same temperature as the extracted air, which causes an extra heating demand in most heating-dominated climates. Frequently, the extracted air leaves the building without any treatment, meaning that polluted but conditioned air is released into the atmosphere.

Heating, ventilation and air conditioning (HVAC) systems can represent more than 40 % [2] of the energy needs in a new residential building, being 30 % - 50 % heat losses by ventilation and infiltration [3]. Then, one of the aspects to optimize the energy use of the building is to reduce the energy losses caused by the presence of a ventilation system, especially when considering well insulated and airtight buildings, in which transmission and infiltration heat losses are lower.

In centralized ventilation systems, the ventilation process consists of two main airflows for exhaust and supply, which can be connected to a heat exchanger to recover heat that may be useful to reduce the heating demand of the dwelling. This additional installation requires the investment for the heat exchanger, but it will reduce the extra heating demand needed to compensate for the heat losses in the exhaust with very low or null maintenance costs. The operational costs would be only associated with the increase in fan power to compensate for the extra pressure drop caused by the installation of the heat exchanger.

In decentralized ventilation systems, it is possible to have local heat exchangers in each zone. This solution has more heat exchangers and less piping. It is also more efficient compared to the centralized systems if there were different temperatures in the dwelling since each heat exchanger would recover heat to reach the temperature of the room (happening when using a perfect heat exchanger). On the contrary, in centralised ventilation systems, the maximum reachable temperature (if using a perfect heat exchanger) would be the result of a mixture of temperatures from the airflows of the different zones, which may not be sufficient for some zones and unnecessarily high for other zones.

The different temperature gradients in the dwellings may influence the effect of heat recovery in the reduction of the heating demand, especially in centralized ventilation systems. Furthermore, the configuration of the ventilation system can also have a significant impact on the heating demand. This study tries to estimate the mentioned impact on a specific case-study dwelling through simulations, using constant effectiveness heat exchangers. Some smart ventilation systems without heat exchangers were also included in the study. For those, the heat that is not extracted by the exhaust air (because of the reduced airflows defined by the smart control) is considered equivalent to the recovered heat when a heat exchanger is present. The result in both cases is heat that does not leave the building in the exhaust air.

## 2. Methodology

In this paper, the impact of the temperature gradients on the heating demand in a dwelling was evaluated using building simulations. The models were developed in Modelica language, using the commercial software Dymola with validated components from the IDEAS library [4] and extended with tailored components to include a better representation of the behaviour of the infiltration through the surfaces from a previous work. Including a detailed model for infiltration causes the simulation speed to decrease by a variable factor (depending on the case, 10 to 100 times slower compared with models with a simplified infiltration network) that makes this study to be limited to a reduced number of ventilation systems and only applied to a single dwelling.

The heat exchanger is an element that works all the time depending only on the temperature difference on both sides of its structure. The exchanged heat can be measured in simulations and in the reality, but the results do not usually give clues on how this recovered heat contributed to a reduction in heating demand. Even on very cold days, when the heating operation is intensive, the total recovered heat is not always useful heat that could reduce the heating demand. When using a heat exchanger, the air temperature in the supply is different all the time with respect to the scenario when no heat exchanger is used, even if the conditioning system is not working, affecting the operation of conditioning the spaces. Then, to know the real impact on the heating demand it is necessary to run two simulations: one with the heat exchanger as it is and another one without the heat exchanger. The difference in heating demand would be associated with the presence of a heat exchanger.

The coefficients $\eta_2$ and $\eta_4$ were defined [5] to evaluate different aspects of the impact of the ventilation on the heating demand. They are calculated based on the heating demands of three scenarios: 1) $Q$, representing the heating demand of the actual system with $Q_h$ representing the heat recovered by the heat exchanger; 2) $Q_0$, representing the heating demand when there is no heat exchanger; and 3) $Q_{nv}$, representing the heating demand when there is no ventilation system connected to the building.

**Equation 1**

$$\eta_2 = \frac{Q_0 - Q}{Q_h}$$

**Equation 2**

$$\eta_4 = \frac{Q_0 - Q}{Q_0 - Q_{nv}}$$

To calculate the coefficients, three simulations are needed. The coefficient $\eta_2$ (equation (1)) reflects the useful recovered heat that contributed to the reduction of the heating demand. The coefficient $\eta_4$ (equation (2)) represents the reduction in heating demand of the installed system with respect to the increase in heating demand caused by installing a system that does not have a heat recovery strategy. Fig. 1 shows graphically the concepts previously explained. Note that $Q_0$>$Q$>$Q_1$>$Q_{nv}$. $Q_h$ is independent and can be higher or lower than any of the heating demands.

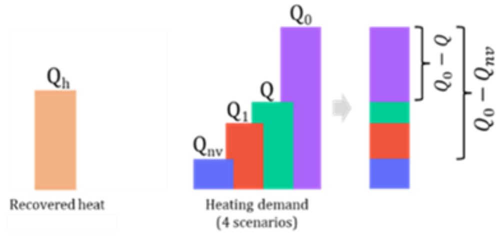


**Fig. 1** - Parameters used to calculate the coefficients.

To compare the performance of a system when using non-uniform heating (closer to reality) and uniform heating (closer to regulations) it is possible to get the ratio between $\eta_4$ in each heating scenario, which differs only in the heating strategy (uniform and non-uniform heating). Equation (3) shows the ratio. All parameters associated with the uniform heating scenario are denoted with a "u" subscript.

**Equation 3**

$$\frac{\eta_4}{\eta_{4u}} = \frac{\dfrac{Q_0 - Q}{Q_0 - Q_{nv}}}{\dfrac{Q_{0u} - Q_u}{Q_{0u} - Q_{nvu}}}$$

If the ratio is greater than 1, then the non-uniform heating scenario would be more efficient in recovering energy with respect to its corresponding $Q_0$ and $Q_{nv}$ scenarios. Also, if the resulting value is below 1, it would indicate that the uniform heating recovers more efficiently the energy than the non-uniform scenario.

The indicators are calculated using specific scenarios applicable to each ventilation system. Consequently, they cannot be used to calculate or estimate annual costs or the thermal performance of the building.

## 3. Description of the model

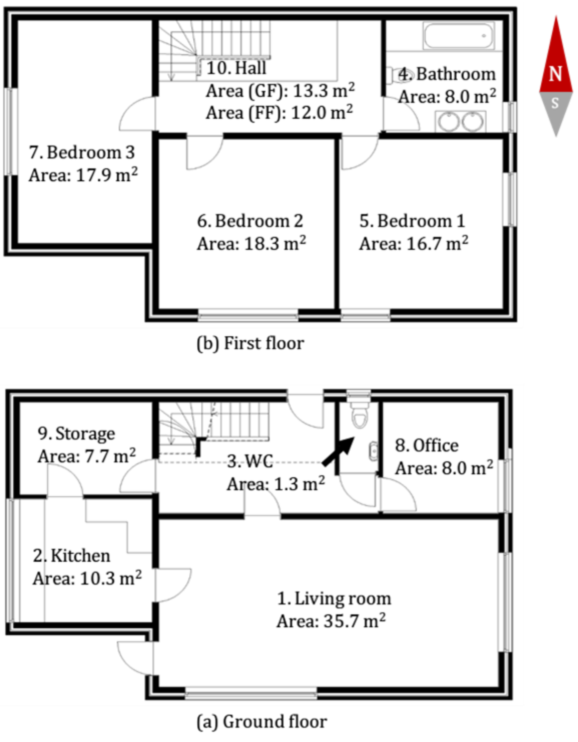


**Fig. 2** - Floorplan.

### *3.1 Building envelope*

A representative detached house of the Belgian dwelling stock was selected as the test envelope for this study. The building consists of a living room, a kitchen, an office and a toilet on the ground floor; three bedrooms and a bathroom on the first floor; and an attic on the second floor. The hall, corridor and stairs are combined into a single zone connecting the ground and the first floor. The floorplan of the dwelling is illustrated in Fig. 2, showing the net floor area of all zones. The zone named Hall includes the floor area divided by floor (GF for the ground floor and FF for the first floor). The attic (missing in Fig. 1) is included in the model as a non-conditioned and non-ventilated zone. It occupies almost the entire second floor, except for a part located on top of the Bathroom, Hall and Bedroom 3. The house has a roof with a pitch of 35º. The southern façade is located on the longest dimension of the building, on the side of the kitchen and the living room.

The height of the ground and first floor is 2.55 m, while the second floor has a representative height of 1.8 m. In total, the model has 11 zones shown in Fig. 2. The basement was not included in the model, so the ground floor is in contact with the ground through slabs.

The thermal performance of the building is characterized by the properties of the construction types listed below in Tab. 1. The total heat loss area is 349.9 m$^2$, which is the summation of all areas listed in the table.

**Tab. 1** – Performance of the constructive elements.

| Element | Area [m$^2$] | U-Value [W/m$^2$/K] | G-Value [-] |
|---|---|---|---|
| Outer Walls | 155.6 | 0.38 | - |
| Roof | 88.6 | 0.48 | - |
| Slabs (ground) | 76.6 | 0.38 | - |
| Win. Glazing | 23.9 | 1.1 | 0.589 |
| Win. Frames | 4.2 | 0.83 | - |

### *3.2 Airflow network*

The zones are represented by air volumes and they are connected to other zones and/or boundary conditions representing air or soil. Those connections can exchange energy, air and trace substances carried by the latter. The airflow network is represented by the openings (doors, windows or cavities) between two zones or a boundary condition and a zone; and by the infiltration network, which occurs through the surfaces between zones or between a zone and a boundary condition.

**Tab. 2 –** Minimum leakage area for doors and windows.

| Element | Area [cm$^2$] |
|---|---|
| Exterior doors | 1 |
| Exterior windows (with frame) | 1 |
| Interior doors | 70 |
| Interior doors (kitchen) | 140 |

As this study intends to reflect the impact of the temperature gradients in the house, all doors (interior and exterior) and windows remain fully closed all the time, to reduce the mixture of airflows inside the house that would trend to uniform the temperature in the different zones. It is assumed that the closed doors and windows have a fixed leakage area corresponding to the values shown in Tab. 2:

Doors and windows (including frames) are characterized by their minimum leakage area, while slabs are airtight, considering that on one side there is only soil. Outer walls, roofs, ceilings and floors include submodels that exchange air between both sides. The ruling airflow equation is based on pressure differences and it is proportional to the area of the constructive element. All opaque constructions have the same air permeability, defined by a coefficient that is adjusted during the initialization of the simulation, considering all outer surfaces, to get a global infiltration factor of 2 ACH50. In other words, the model calculates a factor that, when applying a pressure difference of 50 Pa between the interior of the house and the exterior, having all interior openings open (doors) and having all outer openings (windows and doors) closed, the exchanged air through the boundary elements is equal to 2 times the air volume of the house per hour.

For horizontal surfaces (ceilings/floors), the infiltration is represented by one airflow between the upper and the lower connected air volumes, while for non-horizontal surfaces (walls and roofs) there are two airflows at ¼ and ¾ of the height of the surface, used to include the stack effect, characterized by the change in air density in each of the air nodes caused by the weight of the mass of air above, which varies depending on pressure, temperature and moisture.

### *3.3 Conditioning system*

Every zone has its independent conditioning system which is quasi-ideal. It has unlimited power and it reacts almost instantaneously. It can inject or extract heat to or from the conditioned zones, individually, reaching the heating or cooling setpoints immediately. It can work all the time or when someone is present in the zone, depending on the requirements. The share of the heat transfer is 70 % for convective and 30 % for radiative heat, both in heating and cooling modes.

### *3.4 Setpoint temperatures for conditioning*

The setpoint temperatures are defined for the operative temperature (OT) of the zones. The OT is the average between the radiative temperature (the characteristic temperature of the surfaces that enclose the zone node) and the air temperature of the zone.

There are two main scenarios regarding the setpoint temperatures. On one hand, in the uniform-temperature scenario, all conditioned zones in the house are heated up to 18 ºC when heating and cooled down to 26 ºC when cooling operation takes place. For this scenario, all zones in the building are conditioned, except for the attic. In total, 10 zones are conditioned: the living room, the kitchen, the toilet, the office, the storage room, the hall, the bathroom and the three bedrooms. The conditioning system is active all the time in all zones, with or without anybody present.

On the other hand, different setpoints were defined for each type of zone (non-uniform-temperature scenario). A previous study related to thermal comfort dwellings was used to define them [6]. They are based on a reference outdoor temperature ($T_{e,ref}$) which is calculated according to equation (4).

**Equation 4**

$$T_{e,ref}[ºC] = \frac{(T_{t-0} + 0.8 \cdot T_{t-1} + 0.4 \cdot T_{t-2} + 0.2 \cdot T_{t-3})}{2.4}$$

Where $T_{e,ref}$ is the reference outdoor temperature in [ºC] for the present day, and $T_{t-i}$ ($i \in \{0,1,2,3\}$) the representative temperature of the day $i$ before the present day, which is calculated according to the equation (5):

**Equation 5**

$$T_{t-i}[ºC] = \frac{T_{\underset{t-i}{\max}} + T_{\underset{t-i}{\min}}}{2} \; ; i \in \{0,1,2,3\}$$

Where $T_{\underset{t-i}{\max}}$ and $T_{\underset{t-i}{\min}}$ are the maximum and minimum outdoor temperatures registered on the $i^{th}$ day before the present day, in Celsius degrees. For $i = 0$ the calculations are made for the present day (today).

The same study defines three types of zones for thermal comfort: bathroom, bedroom and the rest of the zones. From $T_{e,ref}$ a neutral temperature ($T_n$) is calculated for the bathroom (equation (6)), bedroom (equation (7)) and the rest of the zones (equation (8)).

**Equation 6**

$$T_n[ºC] = \begin{cases} 0.112 \cdot T_{e,ref} + 22.65 & T_{e,ref} < 11 \\ 0.306 \cdot T_{e,ref} + 20.32 & T_{e,ref} \geq 11 \end{cases}$$

**Equation 7**

$$T_n[ºC] = \begin{cases} 16 & T_{e,ref} < 0 \\ 0.23 \cdot T_{e,ref} + 20.32 & 0 \leq T_{e,ref} < 12.6 \\ 0.77 \cdot T_{e,ref} + 9.18 & 12.6 \leq T_{e,ref} \leq 21.8 \\ 26 & T_{e,ref} > 21.8 \end{cases}$$

**Equation 8**

$$T_n[ºC] = \begin{cases} 20.4 + 0.06 \cdot T_{e,ref} & T_{e,ref} < 12.5 \\ 16.63 + 0.36 \cdot T_{e,ref} & T_{e,ref} \geq 12.5 \end{cases}$$

All units are in [ºC]. $T_n$ is the temperature at which the least amount of people would not feel comfortable (5 % of dissatisfied people). Fig. 3 shows the neutral temperatures given by the equation (6), equation (7) and equation (8), in function of $T_{e,ref}$.

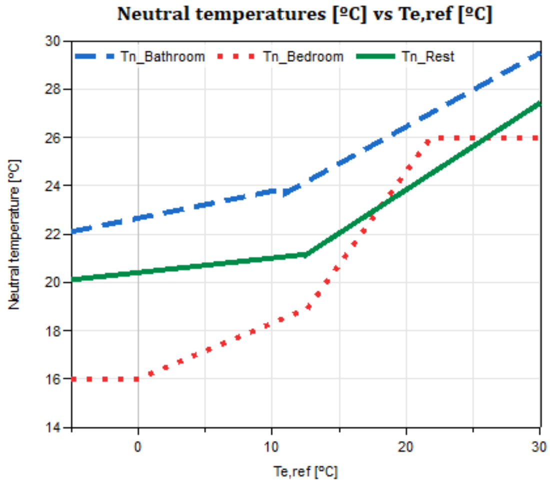


**Fig. 3** - Neutral temperatures for bedroom, bathroom and the rest of the zones.

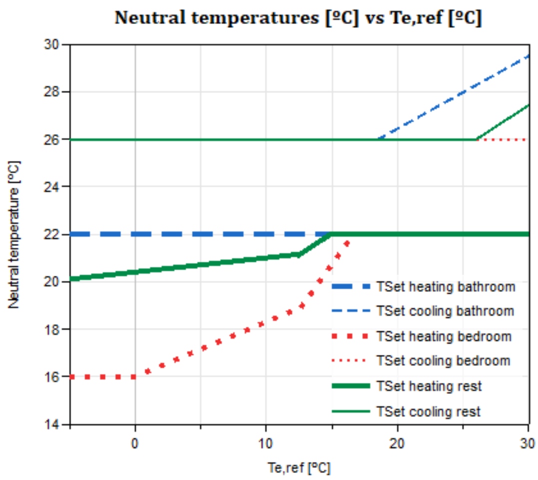


**Fig. 4** - Set temperatures for each type of zone for heating and cooling based on $T_n$.

The conditioning system can provide heating and cooling. Then, using a single setpoint temperature for each type of zone would cause the conditioning system to react to any deviation in the operative temperature around the setpoint in each conditioned zone. This situation would result in higher heating and cooling demands since heating and cooling operations would be required most of the time to compensate for the fluctuations of the operative temperature due to internal and solar gains, as the control is quasi-ideal. Furthermore, looking at the evolution of the setpoints, there are some regions in the setpoints where heating would be discouraged since they correspond to higher values for $T_{e,ref}$, associated with high values for the outdoor temperature (summer). A similar situation can be seen for cooling when looking at the setpoints for lower $T_{e,ref}$ (winter), when cooling should not be activated. To avoid this situation, the heating operation is limited to 22 ºC, while the cooling operation will not take place below 26 ºC. Fig. 3 shows the final setpoint temperatures, for heating and cooling operation, for each type of zone. Between 22 ºC and 26 ºC, the zones will not be heated or cooled.

### *3.5 Conditioning operation modes*

For the uniform-temperature scenario, the conditioning system is activated all the time, so heating or cooling takes place to keep the operative temperature always between 18 ºC and 26 ºC.

For the non-uniform-temperature scenario, the conditioning system is activated exclusively when someone is present in the zone. The conditioning system adjusts the operative temperature immediately according to the defined setpoints when someone enters the zone. Then, the conditioning system is turned off as soon as the room is unoccupied.

### *3.6 Ventilation flow rates*

The building is representative of a dwelling in the Belgian household stock. The requirements for ventilation were calculated according to the national standard NBN D 50-001. The base for the calculated airflow rates is shown in Tab. 3.

**Tab. 3 –** Airflow requirements.

| Zone type | Air flow [l/s] |
|---|---|
| Habitable spaces | max(1*Area$_{zone}$, 7) |
| Toilet | 7 |
| Bathroom | 14 |
| Kitchen | 21 |

Note that the storage room, the hall, and the attic do not have any ventilation requirements, since they are considered non-habitable spaces.

### *3.7 Ventilation systems*

The ventilation systems can be categorized in two ways. Firstly, they can be divided by the ventilation system type (VST) [7], according to the point of air extraction and supply and the way of doing it (mechanical or natural). In this study, VST3, VST4, VST5 and VST7 were investigated. VST3 consist of mechanical exhaust in the exhaust spaces and natural supply using ventilation trickles located in the habitable spaces. VST4 consists of mechanical exhaust in the habitable and exhaust spaces, while the supply consists of trickle vents located in the habitable spaces. Both VST3 and VST4 cannot have a usual heat recovery system, but it is possible to make an equivalence considering that the use of a smart control can avoid releasing heat to the exterior in the exhaust air and count it as a kind of recovered energy. Then, to calculate $Q_0$, the systems will work at nominal power, without the intervention of the sensors. VST5 consists of direct exhaust in the exhaust spaces (the bathroom, the toilet and the kitchen) and mechanical supply in the habitable spaces (the living room, the office and the three bedrooms). The airflows are centralized and a single heat exchanger can be placed to recover heat from the main exhaust airflow.

**Tab. 4** – Design nominal airflows by ventilation system type and ventilation systems. Negative values refer to exhaust.

| | Base flows [l/s] | VST3 [l/s] | VST4 [l/s] | VST5 [l/s] | VST5 [l/s] | VST5 [l/s] | VST7 [l/s] |
|---|---|---|---|---|---|---|---|
| **Ventilation systems** | **--** | **3a, 3b** | **4** | **5a_c, 5d, 5e, 5f** | **5b_c** | **5c_c** | **7a_c, 7b** |
| 1 Living room | 35.7 | 0 | -35.7 | 35.7 | 23.8 | 11.9 | -35.7 |
| 2 Kitchen | 21 | -48.4 | -21 | -48.4 | -32.2 | -16.1 | -21 |
| 3 Toilet | 7 | -16.1 | -7 | -16.1 | -10.7 | -5.4 | -7 |
| 4 Bathroom | 14 | -32.2 | -14 | -32.2 | -21.5 | -10.7 | -14 |
| 5 Bedroom 1 | 16.7 | 0 | -16.7 | 16.7 | 11.1 | 5.6 | -16.7 |
| 6 Bedroom 2 | 18.3 | 0 | -18.3 | 18.3 | 12.2 | 6.1 | -18.3 |
| 7 Bedroom 3 | 17.9 | 0 | -17.9 | 17.9 | 11.9 | 6 | -17.9 |
| 8 Studio | 8.1 | 0 | -8.1 | 8.1 | 5.4 | 2.7 | -8.1 |
| 9 Storage room | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| 10 Hall | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| 11 Attic | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| All mechanical supply | 96.7 | 0 | 0 | 96.7 | 64.5 | 32.2 | 96.7[2] |
| All mechanical exhaust | 42 | -96.7 | -138.7[1] | -96.7 | -64.5 | -32.2 | -138.7 |

[1]The maximum airflow of the real fan is 119 l/s. [2] The trickle vents have two fans: exhaust and supply.

Furthermore, the system is balanced, meaning that the summation of the exhaust airflows is the same as the summation of the supply airflows. Finally, VST7 consist of mechanical exhaust in the exhaust spaces and decentralized exhaust in the habitable spaces using special trickle vents with exhaust fans that can implement heat exchangers to recover heat from the exhaust air. VST1, VST2 and VST6 are not included in the study.

Another division would be between smart and non-smart ventilation systems. The firsts are demand control ventilation systems that use sensors that can measure $CO_2$ concentration, VOCs concentration and/or humidity located in different zones that are used by the controls to adapt the ventilation flow rates. When the detected signals are below a threshold, the ventilation systems operate at their minimum flow rate. The non-smart ventilation system involves systems that do not have sensors, so they work always at their nominal flow rates.

All smart ventilation systems included in this study are models of commercial systems that can be found in the market. The controls are specific for each system and cannot be published. However, Tab. 4 shows the nominal airflows of each system. The columns for ventilation systems 5b_c and 5c_c represent VST5 systems with reduced airflows (2/3 and 1/3 of the nominal VST5 airflows) for non-smart ventilation systems. It was decided to include these two systems to better understand the impact of the airflows in the heat recovery and to have a non-smart system comparable to smart systems in terms of annual airflows since smart systems optimize the airflows significantly.

### *3.8 Occupancy profiles*

To include human behaviour, a stochastic occupation profile was generated by a modified version of StROBe [8]. The profile is composed of two adults who are full-time workers and sleep in the same bedroom.

**Tab. 5 –** Presence of people by zone

| | presence [%] |
|---|---|
| 1 Living room | 3.9 |
| 2 Kitchen | 11.0 |
| 3 Toilet | 5.1 |
| 4 Bathroom | 1.7 |
| 5 Bedroom 1 | 47.0 |
| [6-11] Rest of the zones | 0.0 |
| House (anybody at home) | 89.4 |

The profiles appear as direct sources of convective heat, radiative heat, moisture, and $CO_2$ production for each zone in the model. They vary every 10 minutes and a moving average of a week shows that the profile is quite uniform throughout the year. They include heat and $CO_2$ production due to metabolic processes, domestic hot water, and appliances such as lights, TV or oven.

The presence in a zone is the ratio between the number of minutes a zone is being occupied by at least one person and the total minutes of the simulation. Tab. 5 shows the presence of the zones, as a percentage. Note that 89.4 % of the time there is at least one person in the house, so the level of presence is considerable for just two people. This can be explained because their working hours are not the same.

### *3.8 Weather*

The default weather included in IDEAS was used, which is representative of Brussels. The average outdoor temperature throughout the year is 10.8 ºC, which is coincident with the median. The maximum temperature is 19.4 ºC and the minimum is -7.6 ºC, while the first quartile is 5.8 ºC, and the third is 15.6 ºC.

## 4. Results and discussion

The heating demand is normalized to the net floor area of the building. The ventilation systems are named based on their ventilation system type and those that have the suffix "_c" refer to systems with constant flow rates (non-smart systems). The ventilation system named NV corresponds to the scenario where no ventilation system is installed. Fig. 5 shows the index used to determine the level of IAQ versus the heating demand for all cases, being on the left the uniform heating scenario and on the right the non-uniform heating scenario. The IAQ index is calculated as the summation in all zones of the $CO_2$ concentration multiplied by the number of people present in each zone and integrated along the year. To have a reference, the horizontal dashed lines represent the indicator for constant $CO_2$ concentrations. Values below the line of 400 ppm are not possible since this is the ambient $CO_2$ concentration in the model.

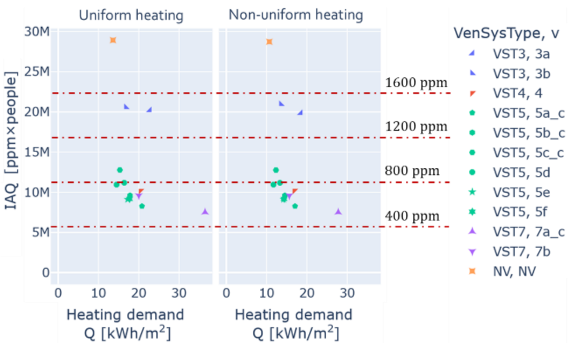


**Fig. 5** – IAQ index vs heating demand

The results are very similar for both heating strategies. For IAQ the values are almost the same, while for the heating demand, uniform heating uses around 15 % more heating energy than non-uniform heating. VST4, VST5 and VST7 have quite good behaviour both in IAQ and heating demand, compared to other cases. However, 7a_c has an extraordinary high heating demand (2.5 times the heat used by NV) combined with a comparative very good IAQ, since its exhaust airflows are quite large (see Tab. 4). The system 5d has the lowest energy heating demand (apart from NV). Both VST3 systems have the worst IAQ behaviour (1.5 to 3 times the IAQ values of the rest of the systems). The heating demand of the system 3a is the second most inefficient, while 3b uses similar heating energy as 5b_c, which is a quite low value.

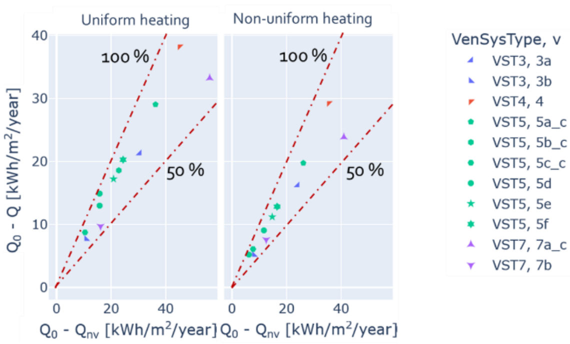


**Fig. 6** – $\eta_4$: $(Q_0 - Q)$ vs $(Q_0 - Q_{nv})$

The factor $\eta_4$, which indicates how much of the heating demand caused by ventilation is successfully reduced by the installation of a heat exchanger (or, in the case of VST3 and VST4 systems, by the installation of a smart control), is shown in Fig. 6. The systems that have lower $\eta_4$ belong to VST7, followed by VST3, while VST5 and VST4 tend to have the highest values for $\eta_4$. The values for $Q_{nv}$ are smaller for non-uniform heating, which corresponds to the lower heating demand seen before. However, the interest of this work is to look at the ratio $\eta_4/\eta_{4u}$, to see how efficient the systems are depending on the heating strategy. The graphical representation of the factor is shown in Fig. 7. The lowest value for $\eta_4/\eta_{4u}$ is for 5d (0.88), which is also the system with the lowest heating demand. The highest value corresponds to 7b (1.01), while 7a_c has a value close to 1.0.

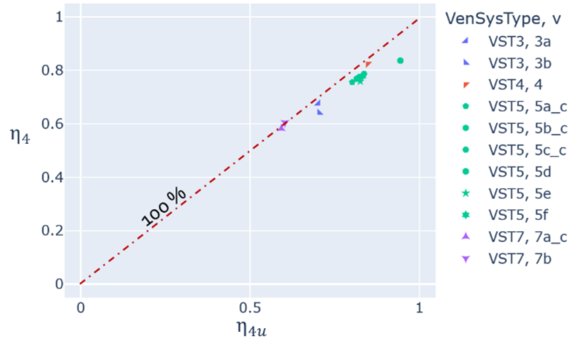


**Fig. 7** – $\eta_4$ vs $\eta_{4u}$

The previous figure shows that only the heat recovery in ventilation systems of VST7 (7a_c and 7b) have an equivalent behaviour when using non-uniform heating than when using uniform heating. The rest of the ventilation systems perform better, in regard to their contribution to their respective heating demand reduction, when using uniform heating. This can be explained because in central ventilation systems the exhaust airflows are merged into a single airflow which temperature is lower than the air in the warmer zones. The recovered heat is distributed among the habitable spaces. Some of them would have a reduction in heating demand, while others will receive the recovered heat although they do not need heating.

As VST7 are distributed systems, they work with the optimal temperatures for each room. Every zone has the potential to recover its exhaust heat, so the heat is not wasted in zones that do not need extra heating. It is important to remark that the ratio $\eta_4/\eta_{4u}$ compares two different heating strategies and the result tells how good the recovered heat of each (or unwasted heat, in VST3 and VST4) contributes to reducing the part of the heating demand that is associated with the installation of a ventilation system without any heat exchanger (or with a constant flow for VST3 and VST4). This extra demand is different for each heating strategy and, in all ventilation systems used in this study $(Q_{0u} - Q_{nvu})$ was greater than $(Q_0 - Q_{nv})$ (Fig. 6).
Looking at Fig. 5, the heating demand is 15 % higher in uniform heating scenarios than in non-uniform heating scenarios in all the cases. This situation can be tricky since higher heating demands use more recovered heat (or heat that is not unnecessarily wasted to the exterior in VST3 and VST4) to reduce the demand: more of the recovered heat (unwasted heat for VST3 and VST4) becomes useful heat that contributed to the reduction in the heating demand. This can be seen looking at $\eta_2$. Fig. 8 shows this ratio for all ventilation systems that have a heat exchanger. In all cases, $\eta_2$ is higher in the uniform heating scenario, meaning that, proportionally, the recovered heat is more useful when using uniform heating.

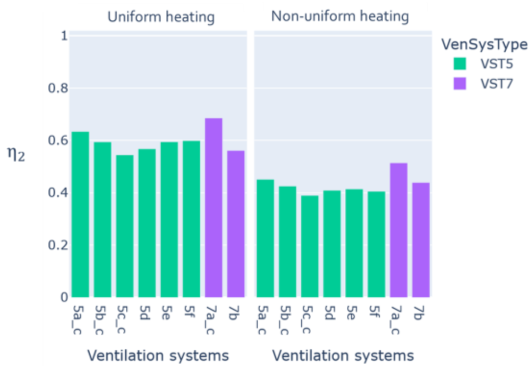


**Fig. 8** – $\eta_2$ for all ventilation systems that have one or more heat exchangers.

## 5. Conclusion

The determination of the impact of temperature gradients on heating demand can be difficult since the recovered heat (or the unwasted heat in systems without heat exchangers) affects the conditions of the zones all the time. Many factors are involved in the analysis, and it becomes difficult to come to a conclusion that could be extrapolated to other cases. For this case study, the factor $\eta_4$ is greater for uniform heating than for non-uniform heating, but this could also be affected by a higher heating demand for uniform heating. Centralized heat recovery systems perform better in the scenarios with uniform heating, while distributed ventilation systems perform similarly with both heating strategies.

The shown indicators cannot be used to determine the energy performance of the building in absolute terms. The annual heating demand is different depending on the heating strategy used and additional indicators are needed to make complementary comparisons. Furthermore, the electric power of the fans is not reflected, as the indicators are calculated using the heating demand.

This study needs to be extended with more buildings, as well as typologies, such as schools, offices, etc. to determine the effect of the temperature gradients and check if there is a relation with the results from this study and if it is possible to quantify the effect with the used ratios and get representative results.

## 6. Acknowledgement


The work presented in this paper has been developed within the frame of the AIVC project *Assessment of the effect of indoor temperature differences and zoning on the performance of energy efficient ventilation strategies for domestic buildings*. This financial support is gratefully acknowledged.


## 7. References


[1] Kim MK, Baldini L, Leibundgut H, Wurzbacher JA, Piatkowski N. A novel ventilation strategy with $CO_2$ capture device and energy saving in buildings. Energy and Buildings. 2015 Jan;87:134–41.

[2] O'Connor D, Calautit JKS, Hughes BR. A review of heat recovery technology for passive ventilation applications. Renewable and Sustainable Energy Reviews. 2016 Feb;54:1481–93.

[3] Omer AM. Renewable building energy systems and passive human comfort solutions. Renewable and Sustainable Energy Reviews. 2008 Aug;12[6]:1562–87.

[4] Jorissen F, Reynders G, Baetens R, Picard D, Saelens D, Helsen L. Implementation and verification of the IDEAS building energy simulation library. Journal of Building Performance Simulation. 2018 Nov 2;11[6]:669–88.

[5] Faes W, Monteyne H, De Paepe M, Laverge J. Useful energy transfer in air-to-air heat recovery units in partly heated low energy buildings. In: Proceedings of the 15th IBPSA Conference [Internet]. 2017 [cited 2022 Jan 13]. p. 2701–7. Available from: http://hdl.handle.net/1854/LU-8550254

[6] Peeters L, Dear R de, Hensen J, D'haeseleer W. Thermal comfort in residential buildings: Comfort values and scales for building energy simulation. Applied Energy. 2009 May;86[5]:772–80.

[7] NEN. NEN 1087: 2019 'Ventilatie van gebouwen - Bepalingsmethoden voor nieuwbouw'. 2019;

[8] Baetens R, Saelens D. Modelling uncertainty in district energy simulations by stochastic residential occupant behaviour. Journal of Building Performance Simulation. 2016 Jul 3;9[4]:431–47.


The datasets generated during the current study are not available yet because this work is a part of an ongoing project. They will be released in the following months. Contact the author for further details.