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Impact of air leakages and short circuits in ventilation units with heat recovery on ventilation efficiency and energy requirements for heating

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Abstract

The impact of unintentional air flows on the performance of ventilation units with heat recovery is discussed on the basis of single room ventilation units. Assuming an external short circuit (outdoor) and internal (inside the ventilation unit) air leakages, which lead to internal short circuits, a model is developed and characteristic numbers for ventilation efficiency, efficiency of heating load reduction and effectiveness of electrical energy use are derived. Differences between supply and extract air flow rates, resulting in increased air flows through cracks in the building envelope, are taken into consideration too. The use of tracer gas techniques to measure air leakage rates from ventilation units is described briefly. It is shown by numerical examples that unintentional air flows can considerably reduce the performance of ventilation units in terms of ventilation efficiency and, in combination with unintentional heat flows through the casing, energy savings. Therefore, these flows should be avoided or at least reduced to an acceptable level by an appropriate construction, manufacturing process and installation of the units.

Keywords: Mechanical ventilation; Single room unit; Air leakage; Ventilation efficiency; Heat recovery

1. Introduction

For various reasons, mechanical ventilation systems with heat recovery are virtually mandatory for low-energy buildings in regions with cold winters and, as a result, are attracting increasing interest. With these systems fresh air is preheated in a heat exchanger by extract air. Unfortunately, depending on the construction, manufacturing process and installation of the units and ducts, air leakages can occur which reduce ventilation system performance or even render ventilation ineffective. These air leakages, for example through cracks between metal sheets or seals between the extract and supply air flow, are induced by local pressure differences. The existence of air leakages and short circuits is well known in centralized mechanical ventilation systems [1–3] as well as in single room ventilation units [4–6].

The European standard [7] for heat exchanger testing demands that internal and external air leakage rates are less than 3% of nominal air flow rate. The test procedure can only be performed if this condition is satisfied. The analysis of measured data does not take air leakages into account. Most single room ventilation units have difficulty fulfilling this criterion. Hence a method for determining characteristic numbers for ventilation efficiency, efficiency of heating load reduction and effectiveness of electrical energy use that takes account of the impact of these unintentional air flows is suggested. The method is discussed with reference to single room ventilation units, but can be applied in principle to other types of ventilation systems too.

2. Objects and methods

Single room units have certain specific properties compared with centralized ventilation systems. These properties and the performance criteria are discussed in [8,9]. Above all, they are potentially highly suitable for building retrofits, because they do not need any ducting within the dwelling. The performance criteria for these units include acoustic aspects such as sound transmission through the units from the exterior to the interior and sound emission, thermal comfort and ventilation efficiency in the room which is supplied with fresh air, as well as aspects of heat recovery and efficient use of electrical energy. Performance also has to be guaranteed if thermal or wind-induced pressure differences occur between indoors and outdoors. These variable pressure boundary conditions are discussed in [10].

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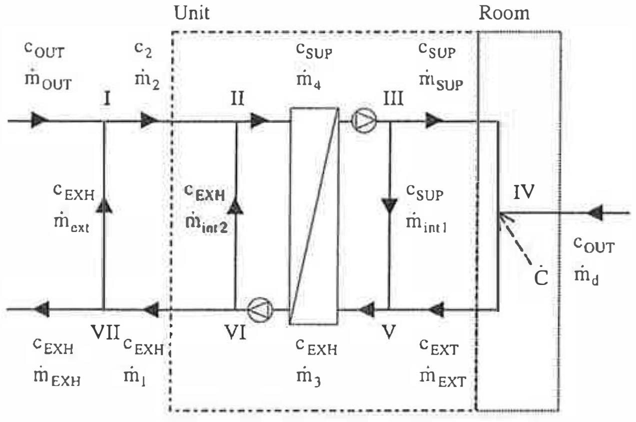

Fig. 1. Main and unintentional air flows in a system consisting of a ventilation unit, room and outdoor space (internal air flow directions I and $\dot{m}_{\mathrm{EXH}} \geq \dot{m}_{\mathrm{OUT}}$).

Two types of unintentional flow that reduce the performance of a single room ventilation unit can occur: unintentional heat flows and unintentional air flows. An unintentional heat flow occurs because the mean temperature in the unit is lower than room temperature during a heating period and, therefore, a flow from the room air surrounding the unit into the unit itself is induced. This heat flow reduces the heat recovery effect and should be kept small by an appropriate unit construction and particularly by appropriate unit insulation [8,9].

Additionally, as already mentioned, air leakages and short circuits can occur. In accordance with Heidt et al. [4–6] three unintentional air flows are taken into consideration (Fig. 1). First, an external flow $\dot{m}_{\mathrm{ext}}$ from the exhaust air into the outdoor air intake. This external air flow is dependent on the construction and positions of the outlet and intake, but also on the velocity and direction of the wind and the difference in temperature between the outdoor air and the exhaust air. Second, an internal air flow $\dot{m}_{\mathrm{int1}}$ which mixes supply air into the extract air, and third, an internal air flow $\dot{m}_{\mathrm{int2}}$ which mixes extract air into the supply air. Internal leakage paths can occur in front of, inside or behind the heat exchanger. The precise leakage distribution is usually unknown. Fig. 1 therefore shows a simplified situation. It is assumed that there are no air flows through the casing. The direction and magnitude of unintentional air flows inside the unit depend on local pressure differences by the leaks and leak properties. Pressure differences inside the unit are mainly determined by the positions of fans. For example, Fig. 1 shows a case where the supply air fan is positioned between nodes II and III and the exhaust fan between nodes V and VI. If supply and extract air flows are not equal, a differential air flow $\dot{m}_{\mathrm{d}}$ occurs through cracks in the building envelope. If the building envelope were absolutely airtight, supply and extract air flows would automatically become equal. However, real building envelopes do have cracks and, therefore, this differential mass flow can occur. Flow resistance of the building envelope – which in practice depends on the building construction – is not taken into consideration here. Further assumptions in the model described here are that specific heat is not temperature dependent and that room air is mixed completely.

In an experimental set-up (Figs. 2 and 3), tracer gas was injected at a rate of $\dot{C}$ into the extract air, representing indoor contamination. All tracer gas concentrations c_{OUT}, c_{SUP}, c_{EXT}, c_{EXH}, c_2, air flow rates $\dot{m}_1$ and $\dot{m}_2$, air temperatures T_{OUT}, T_{SUP}, T_{EXT}, T_{EXH} and electrical power P_{el} were measured (Fig. 1) and used as input data for analysis (Chapters. 3 and 4). In a real building, $\dot{m}_1$ and $\dot{m}_2$ cross the building envelope within two ducts. These two air flows could easily be measured in the experimental set-up and were taken as nominal rates. The pressure difference between 'indoor' and 'outdoor' was kept at zero. Air velocities in the 'outdoor' space of the experimental set-up were low at the time that data for further analysis were taken, corresponding to a no-wind situation.

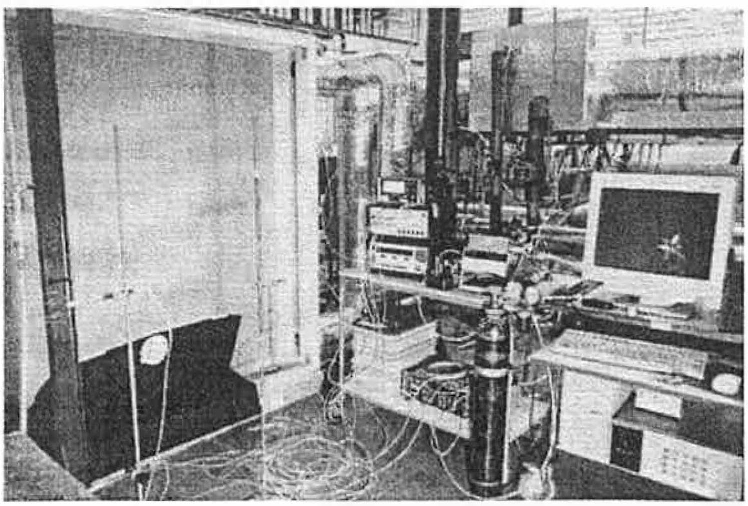

Fig. 2. Experimental set-up with outdoor air intake and exhaust air outlet (round grill) and wooden box, which contains the room with the ventilation unit (left). On the right tracer gas and data acquisition equipments are visible.

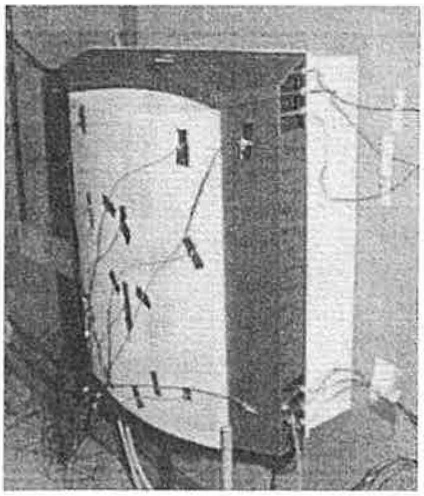

Fig. 3. Single room ventilation unit inside the wooden box (Fig. 2). Temperature sensors are visible in the extract air (at the top of the unit), in the supply air (at the bottom of the unit) and on the surface of the casing. In the supply air, tracer gas concentration is measured too.

In principle, four different situations of internal air flow (leakage) directions are possible, but only the two most frequently encountered in single room ventilation units are dealt with here. Two subcases are discussed for each of the two cases, depending on the direction of the air flow through the building envelope.

3. Modeling

3.1. Ventilation efficiency of the units

3.1.1. Internal air flow directions I and $\dot{m}_{EXH} \geq \dot{m}_{OUT}$

Eleven quantities were not measured in the experiment (Fig. 1): air flow rates $\dot{m}_{OUT}$, $\dot{m}_{SUP}$, $\dot{m}_{EXT}$, $\dot{m}_{EXH}$, $\dot{m}_{d}$, $\dot{m}_3$, $\dot{m}_4$, $\dot{m}_{ext}$, $\dot{m}_{int1}$, $\dot{m}_{int2}$ and injection rate $\dot{C}$. Applying the mass conservation law for air and tracer gas to nodes I to VII, eleven equations are obtained:

$$\dot{m}_{OUT} + \dot{m}_{ext} - \dot{m}_2 = 0 \quad (1)$$

$$\dot{m}_{OUT}c_{OUT} + \dot{m}_{ext}c_{EXH} - \dot{m}_2 c_2 = 0 \quad (2)$$

$$\dot{m}_2 + \dot{m}_{int2} - \dot{m}_4 = 0 \quad (3)$$

$$\dot{m}_2 c_2 + \dot{m}_{int2}c_{EXH} - \dot{m}_4 c_{SUP} = 0 \quad (4)$$

$$\dot{m}_4 - \dot{m}_{int1} - \dot{m}_{SUP} = 0 \quad (5)$$

$$\dot{m}_{SUP} - \dot{m}_{EXT} + \dot{m}_{d} = 0 \quad (6)$$

$$\dot{m}_{SUP}c_{SUP} - \dot{m}_{EXT}c_{EXT} + \dot{m}_{d}c_{OUT} + \dot{C} = 0 \quad (7)$$

$$\dot{m}_{EXT} + \dot{m}_{int1} - \dot{m}_3 = 0 \quad (8)$$

$$\dot{m}_{EXT}c_{EXT} + \dot{m}_{int1}c_{SUP} - \dot{m}_3 c_{EXH} = 0 \quad (9)$$

$$\dot{m}_3 - \dot{m}_{int2} - \dot{m}_1 = 0 \quad (10)$$

$$\dot{m}_1 - \dot{m}_{ext} - \dot{m}_{EXH} = 0 \quad (11)$$

This system of eleven linear equations can easily be solved using a computer program for symbolic computation [11], for example. Eqs. (12)–(17) show how the unknown air flow rates can be calculated using known air flow rates $\dot{m}_1$, $\dot{m}_2$, and tracer gas concentrations c_{OUT}, c_{SUP}, c_{EXT}, c_{EXH} and c_2. Only the quantities that will be used to calculate ventilation efficiency are shown.

$$\dot{m}_{ext} = \dot{m}_2 \frac{c_2 - c_{OUT}}{c_{EXH} - c_{OUT}} \quad (12)$$

$$\dot{m}_{int1} = \dot{m}_1 \left(1 - \frac{c_{EXH} - c_{SUP}}{c_{EXT} - c_{SUP}}\right) + \dot{m}_2 \left(\frac{c_{SUP} - c_2}{c_{EXH} - c_{SUP}} - \frac{c_{SUP} - c_2}{c_{EXT} - c_{SUP}}\right) \quad (13)$$

$$\dot{m}_{int2} = \dot{m}_2 \frac{c_{SUP} - c_2}{c_{EXH} - c_{SUP}} \quad (14)$$

$$\dot{m}_3 = \dot{m}_1 + \dot{m}_2 \frac{c_{SUP} - c_2}{c_{EXH} - c_{SUP}} \quad (15)$$

$$\dot{m}_4 = \dot{m}_2 \frac{c_{EXH} - c_2}{c_{EXH} - c_{SUP}} \quad (16)$$

$$\dot{m}_{EXH} = \dot{m}_1 - \dot{m}_2 \frac{c_2 - c_{OUT}}{c_{EXH} - c_{OUT}} \quad (17)$$

External and internal leakage ratios are defined as follows:

$$\varepsilon_{ext} = \frac{\dot{m}_{ext}}{\dot{m}_2} \quad (18)$$

$$\varepsilon_{int1} = \frac{\dot{m}_{int1}}{\dot{m}_3} \quad (19)$$

$$\varepsilon_{int2} = \frac{\dot{m}_{int2}}{\dot{m}_4} \quad (20)$$

Using Eqs. (12)–(16) and Eqs. (18)–(20), the following is obtained:

$$\varepsilon_{ext} = \frac{c_2 - c_{OUT}}{c_{EXH} - c_{OUT}} \quad (21)$$

$$\varepsilon_{int1} = \frac{c_{EXT} - c_{EXH}}{c_{EXT} - c_{SUP}} \quad (22)$$

$$\varepsilon_{int2} = \frac{c_{SUP} - c_2}{c_{EXH} - c_2} \quad (23)$$

Leakage ratios can be calculated with Eqs. (21)–(23) and measured quantities.

In the following the ventilation efficiency of the unit, η_C, will be derived. This describes how effectively contaminants are removed from a room and is characteristic of the unit and not of the air flow pattern in the room. The case where

$$\dot{m}_{EXH} \geq \dot{m}_{OUT} \quad (24)$$

is considered (Fig. 1).

Starting from a system that includes the room, unit and external leakage flow, a fictitious air flow rate $\dot{m}_C$, which is relevant to the removal of contaminants, is obtained:

$$\dot{m}_C(c_{EXT} - c_{OUT}) = \dot{m}_{EXH}c_{EXH} - \dot{m}_{OUT}c_{OUT} - (\dot{m}_{EXH} - \dot{m}_{OUT})c_{OUT} = \dot{C} \quad (25)$$

and

$$\dot{m}_C = \dot{m}_{EXH}\frac{c_{EXH} - c_{OUT}}{c_{EXT} - c_{OUT}} \tag{26}$$

The ventilation efficiency of the unit, η_C, is defined as

$$\eta_C = \frac{\dot{m}_C}{\dot{m}_1} \tag{27}$$

In the case of $\dot{m}_{EXH} \geq \dot{m}_{OUT}$, air flow rate $\dot{m}_1$ is used as the reference value because it is the higher of the two non-leakage air flows through the building envelope. With Eqs. (17), (26) and (27), the following is obtained:

$$\eta_C = \frac{c_{EXH} - c_{OUT}}{c_{EXT} - c_{OUT}} - \frac{\dot{m}_2}{\dot{m}_1}\frac{c_2 - c_{OUT}}{c_{EXT} - c_{OUT}} \tag{28}$$

In the special case of no leakages with $c_{EXH} = c_{EXT}$ and $c_2 = c_{OUT}$, the ventilation efficiency becomes $\eta_C = 1$. On the other hand, if all the exhaust air is sucked into the intake ($c_2 = c_{EXH}$, $\dot{m}_2/\dot{m}_1 = 1$) or outdoor air does not reach the room ($c_2 = c_{EXH} = c_{OUT}$), ventilation efficiency is $\eta_C = 0$.

3.1.2. *Internal air flow directions I and $\dot{m}_{EXH} \leq \dot{m}_{OUT}$*

With the same internal air flow directions as in Fig. 1 but

$$\dot{m}_{EXH} \leq \dot{m}_{OUT}, \tag{29}$$

$\dot{m}_d$ changes now its direction and eleven mass conservation equations can be formulated and solved again. As a result, leakage ratios according to Eqs. (21)–(23) are obtained. Starting from a system including the room, unit and external short circuit, the fictitious air flow rate $\dot{m}_C$, which is relevant to the removal of contaminants, is obtained:

$$\dot{m}_C(c_{EXT} - c_{OUT}) = \dot{m}_{EXH}c_{EXH} - \dot{m}_{OUT}c_{OUT} + (\dot{m}_{OUT} - \dot{m}_{EXH})c_{EXT} = \dot{C} \tag{30}$$

and

$$\dot{m}_C = \dot{m}_{OUT} - \dot{m}_{EXH}\frac{c_{EXT} - c_{EXH}}{c_{EXT} - c_{OUT}} \tag{31}$$

The ventilation efficiency of the unit, η_C, is defined:

$$\eta_C = \frac{\dot{m}_C}{\dot{m}_2} \tag{32}$$

In the case of $\dot{m}_{EXH} \leq \dot{m}_{OUT}$, air flow rate $\dot{m}_2$ is used as a reference value, because it is the higher of the two non-leakage air flows through the building envelope. It is obtained thus:

$$\eta_C = \frac{c_{EXT} - c_2}{c_{EXT} - c_{OUT}} - \frac{\dot{m}_1}{\dot{m}_2}\frac{c_{EXT} - c_{EXH}}{c_{EXT} - c_{OUT}} \tag{33}$$

In the special case of no leakages with $c_{EXH} = c_{EXT}$ and $c_2 = c_{OUT}$, ventilation efficiency becomes $\eta_C = 1$. On the other hand, if $\dot{m}_2/\dot{m}_1 = 1$ and all the exhaust air is sucked into the intake ($c_2 = c_{EXH}$) or outdoor air does not reach the room ($c_2 = c_{EXH} = c_{OUT}$), ventilation efficiency is $\eta_C = 0$. In the case of $\dot{m}_{EXH} = \dot{m}_{OUT}$, the mass flow ratio is $\dot{m}_2/\dot{m}_1 = 1$ and the ventilation efficiencies according to Eqs. (28) and (33) become identical.

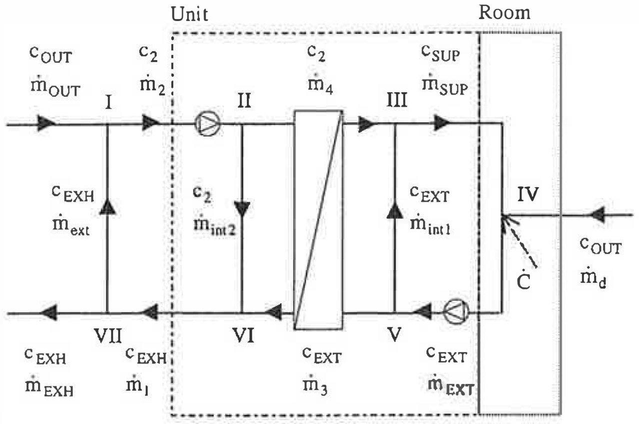

Fig. 4. Main and unintentional air flows in a system consisting of ventilation unit, room and outdoor space (internal air flow directions II and $\dot{m}_{EXH} \geq \dot{m}_{OUT}$).

3.1.3. *Internal air flow directions II and air flow rates $\dot{m}_{EXH} \geq \dot{m}_{OUT}$*

Fig. 4 shows a second typical fan position and internal air flow direction scenario which is found in single room ventilation units. Here again, eleven mass conservation equations can be formulated and solved. External and internal leakage ratios can be defined and calculated:

$$\varepsilon_{ext} = \frac{\dot{m}_{ext}}{\dot{m}_2} \tag{34}$$

$$\varepsilon_{int1} = \frac{\dot{m}_{int1}}{\dot{m}_{SUP}} \tag{35}$$

$$\varepsilon_{int2} = \frac{\dot{m}_{int2}}{\dot{m}_1} \tag{36}$$

$$\varepsilon_{ext} = \frac{c_2 - c_{OUT}}{c_{EXH} - c_{OUT}} \tag{37}$$

$$\varepsilon_{int1} = \frac{c_{SUP} - c_2}{c_{EXT} - c_2} \tag{38}$$

$$\varepsilon_{int2} = \frac{c_{EXT} - c_{EXH}}{c_{EXT} - c_2} \tag{39}$$

The ventilation efficiency of the unit, η_C, which is defined according to Eq. (27), can also be calculated with Eq. (28).

3.1.4. *Internal air flow directions II and air flow rates $\dot{m}_{EXH} \leq \dot{m}_{OUT}$*

With the same air flow directions as in Fig. 4, but with $\dot{m}_{EXH} \leq \dot{m}_{OUT}$, again, eleven mass conservation equations can be formulated and solved. As a result, leakage ratios ε_{ext}, ε_{int1} and ε_{int2} according to Eqs. (37)–(39) are obtained. Using definition 32, the ventilation efficiency of the unit can be calculated according to Eq. (33).

3.2. *Efficiency of heating load reduction*

3.2.1. *Air flow rates $\dot{m}_{EXH} \geq \dot{m}_{OUT}$*

To quantify the effect of the heat recovery in a ventilation unit on the heating load during wintertime and, as a con-

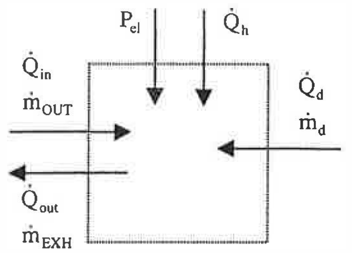

Fig. 5. Energy and mass flows into and out of a system including room, unit and external short-circuit ($\dot{m}_{EXH} \geq \dot{m}_{OUT}$).

sequence, on energy savings, a system is considered (Fig. 5), which includes a room, unit and external short circuit. The heat flow from the room air through the casing into the unit, as described in chap. 2, takes place within the boundaries of this system and is therefore not taken into consideration here. It was assumed that the electrical energy used to power the fans is converted completely into heat within the system boundaries.

Neglecting heat storage effects in the building, conservation of energy demands:

$$\dot{Q}_h + \dot{Q}_{in} - \dot{Q}_{out} + \dot{Q}_d + P_{el} = 0 \qquad (40)$$

Transmission and infiltration heat losses, solar gains and internal gains from people and appliances are not taken into consideration here because they are not relevant to the heat recovery effect. Using Eq. (40), heating load in the case of mechanical ventilation with heat recovery is:

$$\dot{Q}_h = \dot{m}_{EXH} c_p (T_{EXH} - T_{OUT}) - P_{el} \qquad (41)$$

Heating load in the case of ventilation with a fictitious air flow rate $\dot{m}_C$ and without heat recovery is taken as a reference case:

$$\dot{Q}_{h,0} = \dot{m}_C c_p (T_{EXT} - T_{OUT}) \qquad (42)$$

The ratio between the reduction in heating load due to heat recovery and the heating load in the reference case is defined as the efficiency of heating load reduction:

$$\eta = \frac{\dot{Q}_{h,0} - \dot{Q}_h}{\dot{Q}_{h,0}} \qquad (43)$$

Using Eqs. (17), (21) and (27) and Eqs. (41)–(43), and the definition of temperature efficiency in relation to the exhaust air

$$\eta_{T,EXH} = \frac{T_{EXT} - T_{EXH}}{T_{EXT} - T_{OUT}}, \qquad (44)$$

which also includes temperature differences from the waste heat of the fans, the following is obtained:

$$\eta = 1 - \frac{1}{\eta_C}\left[\left(1 - \frac{\dot{m}_2}{\dot{m}_1}\varepsilon_{ext}\right)(1 - \eta_{T,EXH}) - \frac{P_{el}}{\dot{m}_1 c_p (T_{EXT} - T_{OUT})}\right] \qquad (45)$$

When the characteristic numbers ε_{ext}, η_C, $\eta_{T,EXH}$ and air flow rates $\dot{m}_1$ and $\dot{m}_2$, as well as electrical power consumption P_{el} are known, Eq. (45) can be used to calculate the efficiency of heating load reduction at given temperature boundary conditions. Eq. (45) is valid for both internal air flow directions. In the special case of $P_{el} = 0$ and ideal ventilation properties $\varepsilon_{ext} = 0$ and $\eta_C = 1$, the well-known temperature efficiency is obtained from Eq. (45):

$$\eta = \eta_{T,EXH} \qquad (46)$$

The efficiency of heating load reduction according to Eq. (45) but without electrical power, is defined as

$$\eta_{P_{el}=0} = \eta(P_{el} = 0) \qquad (47)$$

3.2.2. Air flow rates $\dot{m}_{EXH} \leq \dot{m}_{OUT}$

If $\dot{m}_{EXH} \leq \dot{m}_{OUT}$, it is obtained in analogy to Eq. (40):

$$\dot{Q}_h + \dot{Q}_{in} - \dot{Q}_{out} - \dot{Q}_d + P_{el} = 0 \qquad (48)$$

In the case of mechanical ventilation with heat recovery, heating load is:

$$\dot{Q}_h = \dot{m}_{EXH} c_p (T_{EXH} - T_{EXT}) + \dot{m}_{OUT} c_p (T_{EXT} - T_{OUT}) - P_{el} \qquad (49)$$

Using Eqs. (21), (32), (42), (43), (44), (49) and the air flow rates $\dot{m}_{EXH}$ and $\dot{m}_{OUT}$, which were found as solutions to the eleven mass conservation equations, the efficiency of heating load reduction becomes:

$$\eta = 1 - \frac{1}{\eta_C}\left[1 - \varepsilon_{ext} + \eta_{T,EXH}\left(\varepsilon_{ext} - \frac{\dot{m}_1}{\dot{m}_2}\right) - \frac{P_{el}}{\dot{m}_2 c_p (T_{EXT} - T_{OUT})}\right] \qquad (50)$$

Where ε_{ext}, η_C, $\dot{m}_1$, $\dot{m}_2$, $\eta_{T,EXH}$ and P_{el} are known, the efficiency of heating load reduction can be calculated with Eq. (50) at given temperature boundary conditions. Eq. (50) is valid for both internal air flow directions. The efficiency of heating load reduction without electrical power can again be defined according to Eq. (47). In the case of a ventilation unit with $P_{el} = 0$ and ideal ventilation properties $\varepsilon_{ext} = 0$, $\dot{m}_1/\dot{m}_2 = 1$ and $\eta_C = 1$ the well-known temperature efficiency is obtained from Eq. (50):

$$\eta = \eta_{T,EXH} \qquad (51)$$

In the case of $\dot{m}_{EXH} = \dot{m}_{OUT}$, the mass flow ratio is $\dot{m}_2/\dot{m}_1 = 1$ and the efficiencies of heating load reduction according to Eqs. (45) and (50) become identical.

3.3. Electrical power

3.3.1. Air flow rates $\dot{m}_{EXH} \geq \dot{m}_{OUT}$

The ratio between the heating load reduction due to mechanical ventilation with heat recovery and the electrical power consumed by the fans is sometimes named electro-

Table 1
Examples of measured data

Parameter	Ventilation unit		
	A	B	C
Tracer gas concentration in outdoor air c_{OUT}	5.4×10^{-4}	7.2×10^{-4}	6.5×10^{-4}
Tracer gas concentration in supply air c_{SUP}	10.0×10^{-4}	7.7×10^{-4}	10.5×10^{-4}
Tracer gas concentration in extract air c_{EXT}	18.8×10^{-4}	15.8×10^{-4}	17.4×10^{-4}
Tracer gas concentration in exhaust air c_{EXH}	16.7×10^{-4}	14.4×10^{-4}	16.5×10^{-4}
Tracer gas concentration in intake air c_2	6.6×10^{-4}	7.3×10^{-4}	7.5×10^{-4}
Outdoor air temperture T_{OUT} (°C)	−2.2	4.0	9.2
Supply air temperature T_{SUP} (°C)	12.7	16.1	17.0
Extract air temperature T_{EXT} (°C)	20.8	21.0	21.0
Exhaust air temperature T_{EXH} (°C)	12.6	10.7	17.4
Air flow rate in outlet $\dot{m}_1$ (kg/h)	35.4	35.5	32.6
Air flow rate in intake $\dot{m}_2$ (kg/h)	34.0	36.2	28.7
Electrical power P_{el} (W)	18.6	8.4	6.8

thermal amplification ETA. Taking account of air leakages and external short circuits, it is defined as:

$$\mathrm{ETA} = \frac{\dot{Q}_{h,0} - \dot{Q}_h}{P_{el}} \tag{52}$$

Eq. (52) is defined so that the fans' waste heat contributes to the heating load reduction. With Eqs. (27), (42), (43) and (52), an electro-thermal amplification ETA that also includes the ventilation efficiency of the unit is obtained:

$$\mathrm{ETA} = \frac{\eta\eta_C\dot{m}_1 c_p(T_{EXT} - T_{OUT})}{P_{el}} \tag{53}$$

In the case of $\eta = \eta_{T,EXH}$ and $\eta_C = 1$, the following is obtained from Eq. (53):

$$\mathrm{ETA} = \frac{\dot{m}_1 c_p(T_{EXT} - T_{EXH})}{P_{el}} \tag{54}$$

3.3.2. *Air flow rates $\dot{m}_{EXH} \leq \dot{m}_{OUT}$*

In the case of $\dot{m}_{EXH} \leq \dot{m}_{OUT}$ and using Eqs. (32), (42), (43) and (52), the following is obtained:

$$\mathrm{ETA} = \frac{\eta\eta_C\dot{m}_2 c_p(T_{EXT} - T_{OUT})}{P_{el}} \tag{55}$$

In the case of $\dot{m}_{EXH} \leq \dot{m}_{OUT}$, it follows that $\dot{m}_1 = \dot{m}_2$ and electro-thermal amplification according to Eqs. (53) and (55) become identical. In the case of $\eta = \eta_{T,EXH}$ and $\eta_C = 1$, the following is obtained from Eq. (55):

$$\mathrm{ETA} = \frac{\dot{m}_2 c_p(T_{EXT} - T_{EXH})}{P_{el}} \tag{56}$$

4. Numerical examples

The model presented here was used to analyse data derived from measurements on three single room ventilation units. Two of them are commercially available and the third is a prototype unit. Units A and C have a very similar construction. The measured data include tracer gas concentrations c_{OUT}, c_{SUP}, c_{EXT}, c_{EXH} and c_2, air temperatures T_{OUT}, T_{SUP}, T_{EXT} and T_{EXH}, air flow rates $\dot{m}_1$ and $\dot{m}_2$, and electrical power P_{el} (Table 1). In units A and C air flow directions I and $\dot{m}_{EXH} > \dot{m}_{OUT}$ and in unit B air flow directions II and $\dot{m}_{EXH} < \dot{m}_{OUT}$ were found. Table 2 shows data which were calculated using the model presented here. Temperature efficiency in relation to the supply air was also calculated:

$$\eta_{T,SUP} = \frac{T_{SUP} - T_{OUT}}{T_{EXT} - T_{OUT}} \tag{57}$$

Table 2
Characteristic numbers calculated with data taken from Table 1

Characteristic number	Ventilation unit		
	A	B	C
External leakage ratio ϵ_{ext}	0.10	0.02	0.10
Internal leakage ratio 1 ϵ_{int1}	0.24	0.05	0.14
Internal leakage ratio 2 ϵ_{int2}	0.34	0.17	0.34
Ventilation efficiency of the unit η_C	0.76	0.83	0.83
Temperature efficiency in relation to the supply air $\eta_{T,SUP}$	0.65	0.71	0.66
Temperature efficiency in relation to the exhaust air $\eta_{T,EXH}$	0.36	0.61	0.31
Efficiency of heating load reduction including electricity η	0.35	0.58	0.31
Efficiency of heating load reduction excluding electricity $\eta_{P_{el}=0}$	0.24	0.52	0.24
Electro-thermal amplification ETA	3.21	9.72	4.10

It can be seen in Table 2 that considerable external short circuits and internal air leakages occurred in these single room ventilation units. As a result, the ventilation efficiencies of the units, η_C, are reduced. The significant differences between the temperature efficiencies in relation to the supply air $\eta_{T,SUP}$ and the temperature efficiencies in relation to the exhaust air $\eta_{T,EXH}$ are due to the heat flow from the room into the unit and air leakages. Taking ventilation unit A as an example, it can be seen that using temperature efficiency $\eta_{T,EXH}$ instead of efficiencies η or $\eta_{P_{el}=0}$, the heat recovery effect will be overestimated by 3 or 33%. Taking $\eta_{T,SUP}$ instead of η or $\eta_{P_{el}=0}$, the heat recovery effect will be overestimated by as much as 46 or 63%. Using nominal air flow rates, contaminant removal will be overestimated by 24% for the same unit.

5. Conclusions

It was shown by means of measured data, a model and numerical examples, that external short circuits, internal air leakages and heat flows through the casing can reduce the performance of single room ventilation units considerably. A ventilation efficiency of the unit, η_C, was derived in order to quantify the effectiveness of contaminant removal from a room. The found efficiency of heating load reduction η depends on temperature efficiency $\eta_{T,EXH}$, but also on the ventilation efficiency of the unit, η_C, and additional parameters. An electro-thermal amplification, ETA, that also depends on the ventilation efficiency of the unit, η_C, and the efficiency of heating load reduction, η, was derived. Temperature efficiency in relation to the supply air $\eta_{T,SUP}$, temperature efficiency in relation to the exhaust air $\eta_{T,EXH}$, efficiency of heating load reduction including electricity η and efficiency of heating load reduction excluding electricity $\eta_{P_{el}=0}$ can differ substantially. Hence, if significant unintentional air flows occur in a ventilation unit, the use of nominal air flow rates to estimate real contaminant removal from a room has to be regarded as a rough estimate only. In the case of significant heat flows from the room into the unit and unintentional air flows, temperature efficiency in relation to the supply air $\eta_{T,SUP}$ is much higher than the efficiency of heating load reduction and should not be used to calculate energy savings.

External short circuits and air leakages should be taken into consideration, particularly if they exceed a certain degree, when characterizing ventilation units and quantifying their ventilation efficiency or the benefits of heat recovery in terms of heating load reductions and, therefore, energy savings. In addition to ensuing that systems are properly installed, manufacturers of ventilation units should try to avoid leaks and external short circuits or at least reduce them to an acceptable level by an appropriate construction and manufacturing process. This will increase the user benefits derived from ventilation units because contaminants are removed more effectively from the room and more energy will be saved by heat recovery.

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