Design considerations with ventilation-radiators: Comparisons to traditional two-panel radiators

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1. Introduction

Rising energy prices and a wish to create environmentally friendly HVAC systems increase the demand for thermally efficient heating systems. Increasing efficiency of radiators allows for a lower water temperature in the radiator circuits. This may result in several positive environmental and economic aspects, such as:

- More efficient energy production by heat pumps, sun panels or similar.
- Reduced heat losses in the distribution net of district heating systems, and facilitation of alternative heating and waste heat retrieval [1].
- Improved thermal climate for occupants. Indoor thermal climate with low temperature systems is assumed better for human health than that provided by high temperature heating systems. Studies show that low temperature heating systems create more stable and uniform indoor climate, with lower air speeds and lower temperature differences [2,3].

Heat output from radiators and vertical plates with natural or forced convection have been calculated analytically and investigated by CFD simulations in several studies [4–6] at the Royal Institute of Technology, School of Technology and Health (KTH- STH) in Stockholm. The main goal has been to find ways to increase the thermal efficiency of radiators. It was found that enlarging or modifying existing radiators or adding convection fins may increase thermal efficiency. The drawback of such changes, however, has often been increased production costs. Attention therefore turned to ways of boosting heat output that might be easier and less costly, such as directing ventilation air towards heated radiator surfaces, or forcing air between radiator panels.

This paper reports on four test cases investigating how much thermal efficiency may be improved and how thermal comfort in the room can be affected by simply changing the position of the ventilation air inlet in relation to the radiator. Cases A and B used traditional radiators and different positions of the air inlet. Cases C and D used innovative ventilation-radiators with different widths between the radiator panels. All investigations were done by CFD simulations in an exhaust ventilated office room model exposed to Swedish winter conditions.

The purpose of the study was to provide guidance for manufacturers of heating and ventilation systems. A secondary
Results were evaluated with reference to recommendations in ISO 7730:1994, an international standard that specifies conditions for thermal comfort [7].

2. Theory

2.1. Heat transfer of a radiator

The correlation in a heat transfer process from warm water inside a radiator to air and room surfaces surrounding the radiator is illustrated in Fig. 1, and summarised in Eq. (1) below.

\[ \dot{m} \cdot c_p \cdot \Delta \theta = k \cdot A \cdot \Delta \theta_m \]  

(1)

Here the parameters on the left side, \( \dot{m}, c_p \) and \( \Delta \theta \), are the mass flow of water inside the radiator, specific heat capacity of water and temperature difference of water entering and leaving the radiator (\( \theta_{\text{water,in}} - \theta_{\text{water,out}} = \Delta \theta \)).

The parameters on the right side are the total heat transfer coefficient, \( k \), area of the radiator surface, \( A \), and the mean temperature difference between radiator surface and ambient air, \( \Delta \theta_m \).

The expression of \( \Delta \theta_m \) is given below.

\[ \Delta \theta_m = \frac{\theta_{\text{water,in}} - \theta_{\text{water,out}}}{\ln(\theta_{\text{air}} - \theta_{\text{water,out}} - \theta_{\text{air}})} \]  

(2)

where \( \theta_{\text{air}} \) is the mean room air temperature.

The total heat transfer coefficient, \( k \), is given by.

\[ \frac{1}{k} = \frac{1}{\alpha_{\text{inst}}} + \delta + \frac{1}{\alpha_{\text{out}}} \]  

(3)

where the terms on the right hand side represent the heat transfer from water to surface inside the radiator, conduction heat transfer through the radiator wall and heat transfer from the radiator surface to ambient air on the outside wall, respectively. The thickness of the radiator wall is given by \( \delta \) and conductivity by \( \lambda \). The limiting factor in the heat transfer, \( \alpha_{\text{out}} \), contains both a radiative and a convective part, \( \alpha_{\text{rad}} \) and \( \alpha_{\text{conv}} \). Eq. (4) describes the configuration of \( \alpha_{\text{conv}} \).

\[ \alpha_{\text{conv}} = \frac{Nu}{h} \]  

(4)

where \( \lambda \) is the conductivity of air (set to 0.025 W m\(^{-1}\)C\(^{-1}\) in this study) and \( h \) is the height of the heated vertical surface. The dimensionless Nusselt number, \( Nu \), is in a natural convection situation on a vertical single surface based on the Rayleigh number,

\begin{align*}
\theta_{\text{water,in}} &\quad \text{water inlet temperature (°C)} \\
\theta_{\text{water,out}} &\quad \text{water outlet temperature (°C)} \\
\nu &\quad \text{kinematic viscosity (m}^2\text{s}^{-1})
\end{align*}
Ra, as shown in Eq. (5a–c) below [8].

\[
Nu = 1.10(Gr \cdot Pr)^{0.17} \text{ when } 10 < Ra < 10^4 \quad (5a)
\]

\[
Nu = 0.48(Gr \cdot Pr)^{0.24} \text{ when } 10^4 < Ra < 10^8 \quad (5b)
\]

\[
Nu = 0.16(Gr \cdot Pr)^{0.32} \text{ when } 10^8 < Ra < 10^{12} \quad (5c)
\]

where \( Ra \) is given by the product of Grashof number, \( Gr \), and Prandtl number, \( Pr \). In the present study \( Pr \sim 0.71 \) for air in the temperature range of current interest.

\[
Gr = g \cdot \beta(\theta_{sur} - \theta_{air}) \frac{h^3}{v^2} \quad (6)
\]

where coefficient of expansion and kinematic viscosity, \( \beta \) and \( v \), are given the constant values \( 3.73 \times 10^{-3} \text{ C}^{-1} \) and \( 1.83 \times 10^{-5} \text{ m}^2 \text{ s}^{-1} \), respectively. Natural convection is assumed when \( Gr/Re^2 \geq 1 \), mixed convection when \( Gr/Re^2 = 1 \) and forced convection when \( Gr/Re^2 < 1 \). The Reynolds number, \( Re \), describes the degree of turbulence in the flowing medium.

\[
Re = \frac{u \cdot L}{v} \quad (7)
\]

where \( u, L \) and \( v \) stands for the characteristic velocity, characteristic length and kinematic viscosity, respectively.

Eqs. (8) and (9) give the \( Nu \) numbers on a vertical surface with mixed and forced convection, respectively [8].

\[
Nu = 0.332 \cdot Pr^{1/3} \cdot Re^{1/2} \quad (\text{vertical plate, mixed convection}) \quad (8)
\]

\[
Nu = 0.0296 \cdot Pr^{1/3} \cdot Re^{4/5} \quad (\text{vertical plate, forced convection}) \quad (9)
\]

the equations above are applicable when \( 0.6 < Pr < 60 \).

For forced convection in a relatively short channel the \( Nu \) number can be calculated as shown in Eq. (10) if the flow is laminar, and as shown in Eq. (12) with turbulent flow. The equations are based on \( Nu \) numbers by Granryd [9], and include the flow effects in the entrance region of the channels. The hydraulic diameter, \( d_h \), is used in these equations as the ducts are non-circular. Expression (13) shows the contents of \( d_h \).

\[
Nu_{ch} = \frac{0.0289Gr^{1.37}}{1 + 0.0438Gr^{2.07}} \quad \text{(channel, forced convection, laminar flow)} \quad (10)
\]

the equation above is applicable when \( Re_{ch} < 2500 \)

\[
Gz = Re_{ch} \cdot Pr \cdot \frac{d_h}{L} \quad (11)
\]

\[
Nu_{ch} = 0.407 \cdot Re_{ch}^{0.55} (d_h/L)^{0.3} \quad \text{(channel, forced convection, turbulent flow)} \quad (12)
\]

the equation above is applicable when \( 2500 < Re_{ch} < 7000 \) and \( 3 < L/d_h < 20 \)

\[
d_h = \frac{4A}{L_{per}} \quad (13)
\]

where \( L_{per} \) is the wetted perimeter.

According to Eqs. (1)–(13) the heat output increases automatically if cold air is directed to the warm radiator surfaces. The extent is influenced by both the temperature and the velocity of the air stream. The mean temperature difference, \( \Delta T_{in} \), is obviously

**Fig. 2.** Variations in convective heat transfer coefficient, \( \alpha_{\text{conv}} \), on a vertical surface due to changing ambient temperatures in a natural convection situation.

A ventilation-radiator is a combined ventilation and radiator system where cold air is brought directly from outdoors through a wall channel into the radiator where it is heated before entering the room. The temperature difference, \( \Delta T_{in} \), between the radiator and incoming air is larger than in other heating systems, as is the convective heat transfer coefficient, \( \alpha_{\text{conv}} \). This makes the ventilation-radiator more efficient than a traditional radiator of the same size. As a result more heat can be extracted from water in the radiator circuit, and water leaving ventilation-radiators may theoretically achieve temperatures similar to the room air, or even lower, depending on the mass flow rate, \( \dot{m} \). **Fig. 4** shows the principle of a ventilation-radiator.

**Fig. 3.** Convective heat transfer coefficient, \( \alpha_{\text{conv}} \), versus air velocity on a 0.6 m high vertical plate and in vertical channels 0.04 m and 0.02 m wide, respectively. Boundary conditions used in the calculations were equal to those in the numerical CFD simulations.
Fig. 4. Sketch of a ventilation-radiator. Cold air (blue arrows) enters a gap in the wall and is directed to a channel formed by radiator panels where it rises as it is pre-heated to room air temperature. The driving forces are partly pressure differences between outdoors and indoors and partly buoyancy forces. A filter in the channel between the wall and the radiator prevents particles in the incoming air from reaching the indoor environment. The major part of the pressure loss is in this filter.

A major advantage with ventilation-radiators in comparison to systems where ventilation air is brought into the building without pre-heating is the possibility to maintain a high ventilation rate even when it is cold outside. Studies indicate that a high ventilation rate with fresh air supply directly from outdoors gives better indoor climate with less Sick Building Syndrome (SBS) symptoms and increased work productivity.

2.3. Ventilation-radiators in combination with a heat pump or district heating system

The high efficiency of ventilation-radiators means that lower average water temperature is required to fulfil the heating demand compared to systems with traditional radiators. A low water temperature from a heat pump to radiators is preferable because compared to systems with traditional radiators. A low water temperature is required to fulfil the heating demand.

The theoretical coefficient of performance of the heat pump, COPC, rises as the temperature difference between condenser, $T_2$, and evaporator, $T_1$, decreases. The Carnot efficiency, $\eta_{C}$, describing the quotient between actual coefficient of performance, COP, and COPC also increases because less compressor power is needed to heat the water in relation to the heating power given by the process. See Eqs. (14) and (15).

$$\text{COP}_c = \frac{T_2}{T_2 - T_1}$$

(14)

$$\eta_C = \frac{\text{COP}}{\text{COP}_c}$$

(15)

where $T_2$ is the condenser temperature and $T_1$ is the evaporator temperature.

If, as an example, the temperature of water produced by a heat pump may be lowered from 50 to 45 °C the estimated coefficient of performance (COP) of the heat pump increases from 3.52 to 3.82, assumed a constant 0.6 Carnot efficiency and −6 °C evaporator temperature (calculations with (Eqs. (14) and (15)). This implies a reduction in energy consumption by 8.8% in the heat pump.

By a calculation with a commercial program, Vitocalc 2005 [13], a 5 °C reduction in the radiator circuit water temperature leads to an annual COP increase from 3.12 to 3.35. This implies a 7.1% energy reduction in the heat pump. For this calculation a particular single family house in Stockholm and a particular air to water heat pump was used.

Also in district heating systems energy may be saved by temperature reductions. Many kinds of heat production, such as combined heat and power, heat pumps, waste heat and flue gas condensation are favoured if water temperature is lowered [1]. The fact that the return-water temperature becomes significantly lower with ventilation-radiators than with traditional radiators is especially interesting. A low return-water temperature back to district heating stations leads to reduced heat losses, reduced pumping work and increases the capacity in the network. Both the district heating supplier and the consumer can profit by lowering the return-water temperature.

3. Method of evaluation

3.1. The room model

A CFD room model was made as a reproduction of a lab, resembling an office, used for earlier thermal climate investigations. Even building materials and heating and ventilation systems were replicated. The room had well insulated walls, one window and an exhaust ventilation system, but no furniture. A detailed description of the actual room can be found in “Thermal comfort in a room heated by different methods” by Olesen et al. [14]. The CFD model was introduced for the first time in “Flow patterns and thermal comfort in a room with panel, floor and wall heating”, by Myhren and Holmberg [5]. Fig. 5 shows a sketch of the office.

3.2. Boundary conditions

The window wall was exposed to an outdoor climate resembling a normal winter day in central Sweden. The total heat transfer coefficient, $U$ value, of this wall was set to 0.3 W m$^{-2}$ °C$^{-1}$ while the window itself had a fixed temperature of 14 °C. Other walls, floor and roof were adiabatic.

A total volume flow rate of 7 l s$^{-1}$ was used for ventilation. The fresh incoming air had a temperature of −5 °C, which was equivalent to the simulated outdoor temperature. Used air was removed through an exhaust unit located at the wall opposite of the window.

3.3. System specifications

The size and position of the ventilation inlet were varied between each case. In cases A and B typical designs for exhaust ventilated rooms were used. The idea of having a long inlet in case B was to spread the incoming air along the entire radiator length. In cases C and D the inlet was fitted between the radiator panels in the
bottom centre of the radiator. Table 1 gives air inlet specifications. Fig. 6 shows the design of the ventilation-radiator.

The two-panel radiator was 0.6 m high and 1.4 m long. A gap of 0.04 m separated the panels in all cases except case D where the gap was narrowed to 0.02 m. The radiator temperature was adjusted to give the same perceived temperature in the middle of the office where a working person normally would be seated. The heat given by the radiator was required to cover transmission losses through the walls and windows as well as heat the room air.

Water flow inside the radiators was not reproduced. Instead a fixed temperature was set for the whole radiator surface. This simplification made the CFD simulations less complicated even if a certain margin of error would occur according to theory. In reality the radiator surface temperature is not uniform. Temperature differences depend on the mass flow rate and the radiator efficiency. Efficient radiators extract more heat from passing water than inefficient ones, resulting in a larger $\Delta \theta$. With such radiators a higher system mass flow rate is required to achieve uniform surface temperature. Eq. (16) shows how radiator heat output was calculated with a simplified equation in the CFD model.

$$P = \alpha_{\text{out}} \cdot A \cdot \Delta \theta_{\text{CFD}}$$

where $\Delta \theta_{\text{CFD}}$ is the difference between the radiator surface temperature and ambient air.

### 3.4. CFD code

The code used in this study was the commercial Flovent 6.1 package. It employed a low Reynolds number $k$–$\varepsilon$ turbulence model (LVEL) specially designed for indoor-climate CFD simulations, see Agonafer et al. for a description [15]. It was easy to handle, proved to give good results and needed less computational time than other models tested. The program had automatic wall treatment functions for grid generation and a surface-to-surface radiation model where radiative conditions could be applied to individual surfaces, and large radiating surfaces could be divided into smaller sub-radiating regions.

#### 3.5. Comfort temperature illustrations

Simulated thermal comfort results are expressed as two-dimensional illustrations from the $XZ$ plane at $Y = L/2$. A window and radiator are located on the left side and exhaust outlet on the right side. See Fig. 5. A thermal index called comfort temperature, $\theta_{\text{comfort}}$, was used to describe thermal comfort in the room, see Eq. (17). Advantages of using comfort temperature instead of operative temperature have been discussed elsewhere by Myhren and Holmberg [16]. Comfort temperature is, like the more commonly used operative temperature, a variable used to obtain an understanding of the perceived thermal climate. It takes into account the balance between radiant heating or cooling and the draught-induced air temperature effects on the perceived air temperature.

$$\theta_{\text{comfort}} = \theta_{\text{sur}} + \theta_{\text{air}} \sqrt{\frac{10 \cdot u}{1 + \sqrt{10 \cdot u}}}$$

where $u$ is the air speed.

### 4. Results

#### 4.1. Results of simulations

The temperature of the radiator was in each case adjusted to give a comfort temperature of exactly 21.0 °C at 1.1 m above floor level in...
the centre of the room. The radiator temperature needed to reach this criterion was dependent on two factors—the efficiency of the radiator, and how well the radiator interacted with the environment and ventilation system. A heating and ventilation system giving high air speed levels and uneven heat distribution in the room would most likely require more energy to create a comfortable thermal climate at the given position. Fig. 7 shows the comfort temperature distributions throughout the room. Table 2 shows the radiator surface temperatures and corresponding heat output values for the cases studied. Mean heat transfer coefficients, $k$ values, for the radiators including all heat emitting surfaces were estimated on the basis of the simulation results, and were summarised in the same table.

### Table 2

<table>
<thead>
<tr>
<th>Parameters, boundary conditions and simulation data</th>
<th>Traditional radiator</th>
<th>Traditional radiator</th>
<th>Vent. radiator</th>
<th>Vent. radiator</th>
</tr>
</thead>
<tbody>
<tr>
<td>Case A</td>
<td>Case B</td>
<td>Case C</td>
<td>Case D</td>
<td></td>
</tr>
<tr>
<td>Temperature of radiator surface, °C</td>
<td>42.3</td>
<td>42.3</td>
<td>38.0</td>
<td>34.5</td>
</tr>
<tr>
<td>Channel width in ventilation-radiators, m</td>
<td>–</td>
<td>–</td>
<td>0.04</td>
<td>0.02</td>
</tr>
<tr>
<td>Air temperature at $Z = 1.1$ m in ref. line, °C</td>
<td>20.8</td>
<td>21.6</td>
<td>21.4</td>
<td>21.3</td>
</tr>
<tr>
<td>Total heat output from radiator, W</td>
<td>483</td>
<td>485</td>
<td>445</td>
<td>435</td>
</tr>
<tr>
<td>Total convection heat output from radiator, W</td>
<td>303</td>
<td>304</td>
<td>297</td>
<td>314</td>
</tr>
<tr>
<td>Total radiation heat output from radiator, W</td>
<td>180</td>
<td>181</td>
<td>148</td>
<td>121</td>
</tr>
<tr>
<td>Convection heat, front panel, facing room, W</td>
<td>74</td>
<td>71</td>
<td>43</td>
<td>38</td>
</tr>
<tr>
<td>Radiation heat, front panel, facing room, W</td>
<td>105</td>
<td>105</td>
<td>87</td>
<td>74</td>
</tr>
<tr>
<td>Convection heat, front panel, inside radiator, W</td>
<td>69</td>
<td>70</td>
<td>98</td>
<td>115</td>
</tr>
<tr>
<td>Mean heat transfer coefficient of radiator, W m$^{-2}$ °C$^{-1}$</td>
<td>6.7</td>
<td>6.8</td>
<td>7.8</td>
<td>9.6</td>
</tr>
<tr>
<td>Mean air speed in ventilation channel, m$^{-1}$</td>
<td>–</td>
<td>–</td>
<td>0.35</td>
<td>0.70</td>
</tr>
<tr>
<td>Temp. gradient, floor to ceiling along ref. line, °C</td>
<td>1.8</td>
<td>4.5</td>
<td>3.0</td>
<td>1.2</td>
</tr>
<tr>
<td>Temp. gradient, floor to $Z = 1.1$ m along ref. line, °C</td>
<td>0.4</td>
<td>3.4</td>
<td>1.7</td>
<td>0.4</td>
</tr>
</tbody>
</table>

4.2. Validation of results

The same CFD model was recently used in a well validated study [5] where results given by CFD simulations were compared to measurements from a study by Olesen et al. and values in VDI Heat Atlas [14,17]. Proof of accuracy was that similar room temperatures and air speed levels were found in simulations as in the test room with the same heating and ventilation arrangements. The arrangements used were similar to those of cases A and B, but no ventilation-radiator was used.

Several procedures confirmed the reliability of the simulations with ventilation-radiators. Grid independent checks were made, i.e. the grid density was increased and decreased to check that a
sufficient number of grid cells had been used and that the solutions were independent of the grid distribution. The grid density was especially crucial in the ventilation channel and around the radiator. The results showed that there was up to 0.3 °C deviation in the temperature distribution in the room depending on the grid density, but that the general flow patterns always were the same. This was regarded as acceptable.

Additionally, values from the simulations were compared to analytical calculations done with Eqs. (1) and (4). The main purpose was to check if the heat output and balance between convection and radiation heat were in agreement.

The calculations proved to be in good agreement with simulated heat outputs for all the ventilation-radiator surfaces including the inside of the ventilation channel. Also the balance between radiative and convective heat output was well established in the simulations. Heat output from certain radiator surfaces is shown in Table 2.

5. Discussion

In the four test cases, different radiator surface temperatures were required to reach the criterion of a comfort temperature of 21 °C in the middle of the room. There were two reasons for this. First, the efficiency of the radiators was not the same in every case, i.e. radiators had different mean heat transfer coefficients depending on the location of the air inlet. Second, the radiators distributed heat to the room in different ways. This resulted in variations in heat distribution throughout the room and influenced the energy requirement. In some cases the radiator surface temperatures had to be turned up to reach the thermal comfort criterion. Yet, all systems were inside the low temperature heating range.

5.1. Heat output

The first observation was that radiator efficiency varied. There was a clear relation between heat output and the position of the air inlet unit. Ventilation-radiators proved to perform better than the traditional radiators, in accordance with theory. In case D, when the inlet was placed inside the 0.02 m wide ventilation channel, the radiator surface temperature could be set 7.8 °C lower than in cases A and B and still provide sufficient heat to the room. This was simply because of increased heat output in the ventilation channel inside the radiator. At least 7% discussed in the theory part in heading 2.3.

The total heat transfer coefficient, \( k \), is normally considerably lower in low temperature radiator systems than in systems with medium or high water temperatures. Both the radiative and the convective part of \( k \) are strongly dependent on the mean temperature difference between the radiator and surrounding air and surface temperatures. Typical \( k \) values for high temperature radiators without convection fins are between 8 and 10 W m\(^{-2}\) °C\(^{-1}\), while low temperature radiators without convection fins have values ranging from 5 to 7 W m\(^{-2}\) °C\(^{-1}\).

The ventilation-radiator used in case D had a \( k \) value of 9.6 W m\(^{-2}\) °C\(^{-1}\), which is in the same level as high temperature radiators. In practice this means that a low temperature ventilation-radiator may replace a traditional high temperature radiator of the same size.

The two ventilation-radiators had different thermal performance. The reason for this was the width of the ventilation channels. In case C air inside the 0.04 m wide channel reached a velocity of 0.35 m s\(^{-1}\). This is relatively low and close to what would have been expected in a natural convection situation. In case D the ventilation channel between the panels was narrowed to 0.02 m. Here the air velocity reached 0.70 m s\(^{-1}\), which resulted in a 42% boost of the convective heat output in the ventilation channel.

5.2. Thermal comfort

The second observation was differing ways of heat delivery to the room. Overall, the four ventilation and radiator arrangements proved to deal with the cold inlet air and the heat loss through the window wall in a way that gave a tolerable thermal climate in the office, see Fig. 7. Thermal comfort criteria set by ISO 7730:1994 were met in all the cases studied except one. Yet there were variations between each case that were likely to be felt by most occupants.

In case A the cold inlet air came through the inlet above the window with an average air velocity of 0.70 m s\(^{-1}\) and fell towards the floor close to the window. It was heated as it came in contact with warm air rising from the radiator and prevented to reach the occupied zone before an acceptable temperature level was attained. This arrangement, which is typical in Sweden, gave a stable thermal climate in the office with a relatively low vertical temperature difference in the middle of the room. The only place which could cause discomfort was close to the window wall where air masses of large temperature differences met, leading to temperature fluctuations.

In case B, the air inlet was placed behind the radiator just beneath the window. The intention was to heat the inlet air in an early stage, so as to avoid high air speed and increased radiator heat output, but this position proved to be the worst for thermal climate. Much of the cold inlet air fell directly to the floor, slipped under the radiator and quickly spread into the occupied zone. As a consequence a cold draught at ankle height crossed the floor at the place where an office worker normally would be seated. The temperature difference between the floor and \( Z = 1.1 \) m in the middle of the room was higher than recommended by ISO 7730:1994. In addition a higher air temperature was needed to reach desired comfort temperature. This implied increased air speed levels in the room. It proved to be difficult to aim the inlet air plume in a way that prevented the inlet air to either flow over or slip under the radiator and into the occupied zone.

In case C, the ventilation-radiator with the widest opening between the radiator panels was tested. Here inlet air that passed through the ventilation slot was heated to about 8 °C before it entered the room. Because of a low velocity when leaving the radiator and a temperature lower than that of the room air, the inlet air joined the downdraft from the window and fell to the floor. This resulted in a larger vertical temperature difference in the room than in the case with the inlet placed over the window. Still the thermal climate was well within recommendations.

In case D the incoming air was heated to about 12 °C in the ventilation channel even if the air used less time to pass through the radiator. No areas in the room were found to have an unpleasant thermal climate because the cold air was pre-heated to this extent. Small temperature variations, and thus small density differences, also resulted in less air movements in the room. The stable thermal climate resulted in a smaller temperature difference along the reference line in this case than with any other arrangement. Even more important to note was that although total heat output from the radiator was much lower than in case A and B, thermal comfort criterion in the middle of the room was still achieved. This arrangement made the most of the energy that was put into the radiator.

5.3. Future work

Both theory and simulation results showed that the size of the ventilation channel through a ventilation-radiator has a big effect
on the total heat output. By narrowing the channel the air flow characteristics change. This means that the infiltration air can more easily break the isolating laminar boundary layer close to the heated surfaces and improve the heat transfer. The volume flow of incoming ventilation air, the pressure loss and the risk for draught where air enters the room above the radiator were identified as the main optimization factors. Noise in the ventilation channel is not expected to cause annoyance. This knowledge may lead to development of ventilation-radiators with design differing from that of traditional radiators. It is considered likely that new types of slim, double-panelled ventilation-radiators might be attractive both in terms of energy usage and appearance.

This study was done with radiators without convection fins, but results suggest such fins may have a role to play. Having convection fins between the radiator panels may increase the heat output by up to 70% in conventional radiators depending on the water temperature level in the radiator and the radiator height (based on data provided by manufacturers and VDI heat atlas). Adding convection fins in ventilation-radiators with relatively wide ventilation channels would probably result in at least the same increase. Further studies should investigate whether convection fins are functional in ventilation-radiators with narrow ventilation channels that are optimized for turbulent air flow. It should also be investigated whether increased roughness or small obstacles attached in the ventilation channel may increase turbulence and this way increase heat transfer by convection.

The thermal response of ventilation-radiators is stronger and faster than the thermal response of conventional radiators. These facts, and the resulting benefits, were first discussed by Myhren and Holmberg in “Energy saving and thermal comfort with ventilation-radiators—a dynamic heating and ventilation system” [18]. The same authors have also discussed whether ventilation-radiators may be functional for cooling purposes [19]. Further advantages and disadvantages regarding ventilation-radiators need to be investigated more closely. Examples are:

- How air leakages through the building façade influence the performance of ventilation-radiators (zero leakage through the wall was assumed in this study).
- What happens to heating of incoming ventilation air and the thermal indoor climate when the thermostat of ventilation-radiators is switched off or the flow of warm water to the radiators is stopped by other means.
- How the ventilation channel through the wall and filter should be designed to minimize transport of noise and pollution from outside to the indoor environment, to prevent accumulation of dust, and make cleaning or replacement of filter possible.
- How filters and ventilation-radiators should be designed to allow a wider range of volume flows and heat requirements.

6. Conclusion

It is recommended to let cold ventilation air come in contact with radiators at an early stage of fresh air entry to the room, to avoid cold drafts. The idea of introducing outside air through the radiators themselves to increase efficiency and improve thermal comfort was tested in this study and compared to traditional arrangements by CFD simulations. The room model had a ventilation rate which met Swedish recommendations even if the ventilation air came directly from outdoors at −5 °C without pre-heating.

Results from the study show that the ventilation-radiators were more thermally efficient and could give a more favourable and stable thermal climate in the room under the given conditions in comparison to the traditional arrangements. As a result the surface temperature of the ventilation-radiators could be up to 7.8 °C lower compared to the traditional radiators while still fulfilling heating requirements. In reality this means that a lower water temperature level can be used in radiator circuits with ventilation-radiators. It is thus obvious that ventilation-radiators may lead to energy savings for heating of buildings, possibly up to as much as 5–10% according to theory in 2.4.

The ventilation-radiator with the narrowest air channel between radiator panels performed best. Having a narrow air slot gave higher air velocity in the ventilation channel, which increased the heat transfer. The drawback of making the area of the ventilation channel narrower is the risk for draft close to the radiator. Consequently, the desired volume flow should be the main factor in determining ventilation channel geometry in relation to both draft risk and best filter type to use.

Further studies are needed to optimize ventilation-radiator design and overcome practical barriers. These should include studies on how modern technology can be used to create dynamic heating and ventilation systems that utilize the fast thermal response of ventilation-radiators.

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References