Dalarna University

This is a submitted version of a paper published in *Applied Thermal Engineering*.

Citation for the published paper:

Myhren, J. (2013)

"Performance evaluation of ventilation radiators"

Applied Thermal Engineering, 51(1-2): 315-324

URL: <http://dx.doi.org/10.1016/j.applthermaleng.2012.08.030>

Access to the published version may require subscription.

Permanent link to this version:

<http://urn.kb.se/resolve?urn=urn:nbn:se:du-11121>

<http://du.diva-portal.org>

Performance evaluation of ventilation radiators

Jonn Are Myhren, Sture Holmberg

Construction Technology, School of Technology and Business Studies,

Högskolan Dalarna, 791 88 Falun, Sweden

jam@du.se

Telephone: +46739474871

Abstract

A ventilation radiator is a combined ventilation and heat emission unit currently of interest due to its potential for increasing energy efficiency in exhaust ventilated buildings with warm water heating. This paper presents results of performance tests of several ventilation radiator models conducted under controlled laboratory conditions.

The purpose of the study was to validate results achieved by Computational Fluid Dynamics (CFD) in an earlier study and identify possible improvements in the performance of such systems. The main focus was on heat transfer from internal convection fins, but comfort and health aspects related to ventilation rates and air temperatures were also considered.

The general results from the CFD simulations were confirmed; the heat output of ventilation radiators may be improved by at least 20 % without sacrificing ventilation efficiency or thermal comfort.

Improved thermal efficiency of ventilation radiators allows a lower supply water temperature and energy savings both for heating up and distribution of warm water in heat pumps or district heating systems. A secondary benefit is that a high ventilation rate can be maintained all year around without risk for cold draught.

Keywords: ventilation radiator, convection fins, laboratory measurements, CFD simulation, exhaust ventilation, thermal comfort, energy consumption, low temperature heating

Nomenclature

A heat transferring surface area of radiator (m^2)

c_p specific heat capacity ($(\text{W}\cdot\text{s})/(\text{kg}\cdot^\circ\text{C})$)

P heat output (W)

$\dot{m}_w$ water mass flow (kg/s)

q_a ventilation airflow (l/s)

Greek letters

α_{conv} convective heat transfer coefficient ($\text{W}/(\text{m}^2\cdot\text{K})$)

Δp air pressure difference (Pa)

$\Delta\theta_m$ mean temperature difference between heated surfaces and incoming/ambient air ($^\circ\text{C}$)

$\theta_{a, \text{indoor}}$ indoor air temperature ($^\circ\text{C}$)

$\theta_{a, \text{outdoor}}$ outdoor air temperature ($^\circ\text{C}$)

$\theta_{a, \text{supply}}$ air supply temperature ($^\circ\text{C}$)

$\theta_{w, \text{return}}$ return water temperature ($^\circ\text{C}$)

$\theta_{w, \text{supply}}$ supply water temperature ($^\circ\text{C}$)

1 Background

Within the European Union (EU) there is a demand for a 20 % reduction in the total annual energy consumption before year 2020 [1]. Here the building sector can play a key role. Heating, ventilation and cooling of buildings is responsible for 40 % of the total energy consumption and a significant amount of CO₂ emission. As a consequence, there is an increased demand for efficient environmental friendly heating systems especially adapted to heat pumps and district heating systems. These heat emitters allow a lower system water temperature with the same heat output as traditional systems, which allow energy savings in heat production and distribution.

A group of European scientists (EUROVEN) has reviewed scientific papers on ventilation and the effects on health, comfort, and productivity in offices, schools and homes. The group found that the degree of ventilation was strongly linked to perceived air quality and health. A connection between ventilation rate and productivity in offices was also found. The main conclusion of the study was that a high ventilation rate improved perceived air quality, reduced risk of health problems and increased productivity. Additionally, the group concluded that more health related problems were reported in buildings with air-conditioning systems compared to natural or exhaust-ventilated buildings [2].

A ventilation radiator that combines ventilation air supply to a room with traditional heating by a hydronic radiator, satisfies both the ventilation and energy targets mentioned above while being user-friendly and simple to operate. It is more energy efficient than a traditional radiator, making it more suitable in combination with heat pumps and alternative heating, while at the same time allowing a high ventilation rate in the building. Figure 1 shows the principles.

Figure 1. Parts of a ventilation radiator system: 1) vent, 2) wall channel, 3) filter, 4) radiator with one array of internal convection fins, 5) heater and circulation pump. The symbols indicate measuring points used in the laboratory experiment. Measured variables are listed in Table 1.

Cold outdoor air enters a vent in the building wall, passes through a wall channel and a filter before being directed to a channel formed by the radiator panels. Here the air is pre-heated to room air temperature. The driving forces are partly buoyancy forces and partly pressure differences between outdoors and indoors created by the exhaust ventilation fan in the building (typically 10 Pa). The filter prevents particles in the incoming air from reaching the indoor environment. Usually each wall channel provides fresh ventilation air to a half metre of radiator length. Most ventilation radiators on the market have some sort of mixing between inlet air and room air so as to achieve comfortable temperatures where air leaves the radiator. This is illustrated by a small red arrow and referred to as “room air mix” throughout this paper.

It should be noted that very little or no extra driving force is needed with ventilation radiators compared to systems where air is brought in through conventional ventilation inlets. This means that a standard exhaust ventilation system can be used and there is no extra energy consumption for ventilation with this system.

Ventilation radiators are more thermally efficient than traditional radiators because of:

- Increased temperature difference between heated surfaces and air passing between the radiator panels
- Higher mean air velocity between the radiator panels with increased friction velocity and improved heat transfer as a result

Technical details have been discussed in two Computational Fluid Dynamics (CFD) based studies by Myhren and Holmberg [3-4]; “Design considerations with ventilation-radiators: Comparisons to traditional two-panel radiators” and “Improving the performance of ventilation radiators: the role of convection fins”. The latter paper gives practical suggestions on how to make ventilation radiators even more thermally efficient without compromising the quality of ventilation in the building. It was proposed to utilize the fan-generated under pressure in exhaust ventilated buildings to provide extra driving force pulling air through ventilation radiators, thus allowing more resistance in form of convection fins (enlarged heat transferring surface area).

The following modifications were suggested:

- Modifying the air inlet to ventilation radiators so as to avoid mixing between incoming ventilation air and room air and achieve a higher ventilation rate.
- Decreasing the distance between internal convection fins to about 6-8 mm to enlarge the area of heat transferring surfaces (Figure 3). For traditional radiators this could mean having twice as many fins. Heat output would increase and the ventilation rate would decrease slightly.
- Introducing a mixing chamber, i.e. splitting the array of fins into two vertical sections separated by mixed air (Figure 3). The purpose of a mixing chamber is to break the insulating viscous sub layer of air which always builds up close to

heated surfaces (typically only 1% of the boundary layer thickness) and to mix the air to a uniform temperature before it enters the uppermost fin array. As a result the thermal gradient very close to heated surfaces increases with increased heat transfer as a result. A slight increase in pressure drop through the system could occur depending on the height of the mixing chamber.

1.1 Objectives

The objectives of this study were to test the changes mentioned above by laboratory measurements to confirm the general results from [4]. CFD simulations always incorporate certain error margins, approximations and simplifications. This is why real life experiments are preferred to confirm results. The main factors likely to affect the accuracy in [4] were as follows:

- Geometries were simplified in the CFD model to reduce complexity and central processing unit (CPU) running time.
- A fixed radiator surface temperature was used in the CFD simulations. In real radiators the surface temperature is not even and dependent on the mass flow rate of water in the radiator circuit. The higher the mass flow rate, the more uniform the surface temperature becomes.
- Real radiators have a painted surface that slightly reduces conductive heat transfer. This was not considered in the CFD models.
- In real ventilation radiators the filter and wall channel have an effect on the degree of turbulence and the airflow profile, as supply air is led to the gap between the radiator panels. In the CFD simulations the flow was under-developed and without turbulence intensity at the entrance. This was deliberately arranged in order to compare the results with theory for channel flow.

The other main objective was to identify further possible improvements for heat transfer mechanisms and find out whether ventilation radiators could be made to be more efficient than other systems on the market in any of three possible areas: in terms of energy consumption, health aspects and thermal comfort. The specific goal was to find out whether improved ventilation radiators could provide at least the same heat output with lower supply water temperature, while the ventilation rate was maintained and the air supply temperature to the room was similar or higher.

2 Method

Five test models were proposed, based on [4] and tested inside a climate simulator through two iterative steps. The simulator was divided into one warm (indoors) and one cold chamber (outdoors) with a pressure tight wall between, each room volume being 64m^3 . The temperatures in each chamber and the pressure difference between them could be controlled to simulate the real conditions of an exhaust-ventilated building exposed to various outdoor temperatures. Outdoor air temperatures were ranged from 0.0° to -15.0°C to cover typical winter outdoor conditions in Sweden. The indoor air temperature was kept constant at 20.0°C while a pressure difference between the cold and warm sides was controlled at close to 10.0 Pa . It was partly this pressure difference and partly buoyancy forces that drove supply ventilation air from outside (cold chamber), via the ventilation radiator where heating of ventilation air occurred to indoors (warm chamber), as described in the introduction. A Micatrone pressure transmitter, MT-PD version 3 was used to monitor the pressure difference (accuracy: $\pm 0.5\%$ of total pressure range), while a Brüel og Kjær meter was used for airflow measurements (accuracy: $\pm 0.3\text{ l/s}$).

A warm water circuit was constructed for supplying and controlling warm water to the ventilation radiators. Supply temperatures were set to 55.0°C and 35.0°C , typical temperatures in radiator systems and floor heating systems in Sweden. The system incorporated a MGW Lauda K4R heater and circulation pump with accurate speed control connected to the radiators with a 13mm pipe. Thermocouples were fitted in the pipe just before and after the radiators to measure the water temperature difference (typical accuracy of temperature measurements: $\pm 0.3^\circ\text{C}$). A Danfoss MAG 3000 flow meter was connected to the circuit to measure and help control the mass flow rate at 0.01kg/s , (accuracy: $\pm 0.25\%$ of the total range). Figure 1 shows the arrangement.

As it was difficult to control the pressure difference between the rooms at exactly 10.0Pa , two test runs of about 24 minutes were conducted in each case. In one the pressure difference between cold and warm chamber was aimed at 9.5Pa and in the other 10.5Pa . A Mitec AT 40 collected data from all measuring devices every 15 seconds. Mean values from the data series made the basis for all calculations in the results part. A linear correction procedure using the least square method was used to estimate the heat output, supply air temperature and ventilation airflow at 10.0Pa . Figure 2 shows an example. After pressure correction the results were plotted in the diagrams at the average outdoor temperatures of the two test runs. Table 1 describes the accuracy of the measurements.

Figure 2. Example of the linear data correction procedure used to estimate heat output at 10.0 Pa. A ventilation radiator with traditional convection fin geometry and room air mix was used (described in Figure 3 and Table 2).

Table 1. Measurement data and deviation values for 8 variables

Controlled variables	Denotation in Figure 1	Aim set value	Accepted mean deviation	Standard deviation
Supply water temp., °C	$\theta_{w, \text{supply}}$	55.0/35.0	1.0	0.04
Water mass flow, kg/s	$\dot{m}_w$	0.01	$5.0 \cdot 10^{-4}$	$3.72 \cdot 10^{-5}$
Indoor air temp., °C	$\theta_{a, \text{indoor}}$	20.0	1.0	0.03
Outdoor air temp., °C	$\theta_{a, \text{outdoor}}$	0.0/-7.5/-15.0	1.5	0.27
Air pressure difference, Pa	Δp	9.5/10.5	1.0	0.36
Monitored variables		Comments		
Return water temp., °C	$\theta_{w, \text{return}}$	To determine heat output		0.11
Air supply temp., °C	$\theta_{a, \text{supply}}$	To evaluate comfort aspects		0.15
Ventilation airflow, l/s	q_a	To evaluate health aspects		0.38

2.1 Test models

Figure 3 and Table 2 describe characteristics of the ventilation radiator models. In all cases these were 500mm high and 600mm long, and convection fins occupied an area 400mm high. A standard filter was used and positioned as shown in Figure 1.

Figure 3. Radiator panels and internal convection fins shown in detail section from above (left), elevation (middle) and section (right), where a) denotes traditional radiator fin geometry and b) to f) denote modifications to the test models. These were: b) reduced fin-to-fin distance in central panel area, c) mixing region, i.e. removal of central part of fin array, d) convection fins added to the backside of the back radiator panel, e) front panel exchanged for solid metal plate, f) staggered fins, i.e. central part of fin array offset in relation to fins above and below.

Table 2. Description of ventilation radiator test models

Model type	a)	b)	c)	d)	e)	f)	Comments
Traditional convection fin geometry, room air mix	X						Conventional arrangement, no changes made to the radiator. Room air mixing as indicated by the small red arrow in Figure 1
Traditional convection fin geometry, modified air inlet (no room air mix)	X						Conventional arrangement, no changes made to the radiator. No mixing between supply air and room air before supply air enters the radiator
Test model 1		X					Reduced fin-to-fin distance where ventilation air passes instead of traditional fin geometry
Test model 2		X		X			The backside of the back panel was also utilized for convection fins, for possible extra heat output
Test model 3		X	X	X			Designed to check whether a mixing chamber behaves as predicted in [4]
Test model 4		X	X	X	X		The front radiator panel was substituted with a solid steel plate (no water inside)
Test model 5		X			X	X	Staggered fins (offset fins), ensuring that cold supply air comes in contact with heated surfaces inside the radiator (Figure 10)

3 Results

Laboratory measurements for heat output, supply air temperature and ventilation airflow for the ventilation radiator models tested are shown in Figure 4-8.

3.1 Validation of CFD results

Figure 9 presents CFD screenshots with radiator and ventilation radiator models representative for all CFD simulations in [4]. These were chosen as a basis for verifying the CFD (the most central mechanisms were covered by these models). Table 3 compares CFD simulation results, laboratory measurements and manufacturer's data. Observe that the CFD simulations included heat transfer only from internal surfaces of radiators, following simulation procedure and settings as detailed in [4]. Estimations had to be made for the remaining surfaces in order to compare with results for whole radiator heat output. This was done using manufacturer's data and calculations using Nusselt numbers as previously done in [3].

The results-column to the far left of Table 3 presents results from the CFD simulations in [4]. These simulations were conducted with an even radiator surface temperature of 35°C, which was equal to the supply temperature later used in the measurements (the supply and return water temperatures are shown in the "surface temperature" cells under "laboratory measurements" in Table 3). When the results from the CFD simulations in [4] were compared to measurements it could be seen that the results agreed in general, but some further observations need to be made. In the CFD simulations the buoyancy force was over predicted while the resistance in the filter was slightly too high. Additionally, for the most efficient ventilation radiator models, the total heat output in the CFD simulations became too high since the surface temperature was fixed. In reality the mean surface temperature becomes lower as the efficiency increases, which limits heat transfer. The excess in predicted heat output between CFD simulations and measurements was at most 12 % (case no 3). When new simulations were made adopting a surface temperature equal to the actual mean surface temperature of the measurements, this trend was less evident, but the total heat output and supply temperature was generally too low. Manufacturer's data with traditional radiators [5] were in better agreement with results from measurements than results from the CFD simulations.

It appeared that CFD simulations were appropriately and effectively used in the optimization process described in [4]. The findings were in general agreement with the measurements. A discussion about possible errors in the measurements is following subsequent.

Figure 4. Heat output versus outdoor air temperature in laboratory measurements, with 55 °C supply water temperature to ventilation radiators.

Figure 5. Heat output versus outdoor air temperature in laboratory measurements, with 35 °C supply water temperature to ventilation radiators.

Figure 6. Air supply temperature to the room versus outdoor air temperature in laboratory measurements, with 55 °C supply water temperature to ventilation radiators.

Figure 7. Air supply temperature to the room versus outdoor air temperature in laboratory measurements, with 35 °C supply water temperature to ventilation radiators.

Figure 8. Incoming ventilation airflow versus outdoor air temperature in laboratory measurements, mean values of 55 °C and 35 °C supply water temperature to ventilation radiators.

Table 3. Comparisons of results from CFD simulations, laboratory measurements and manufacturer's (MFR's) data. Values in brackets denote heat output from internal surfaces.

Case no	Radiator with:		CFD in [4]	Laboratory measurements	CFD (surf. temp. corr.)	MFR's data
1	Traditional convection fin geometry Natural convection Indoor air temp: 20 °C	Surface temp., °C	35.0	35.0/32.4	33.7	35.0/32.4
		Heat output, W	146 (86)	122	137	125
		Vent. airflow, l/s	-	-	-	-
		Supply air temp., °C	26.9	27.1	26.4	-
2	Traditional fin geometry Vent. rad. arrangement Outdoor air temp: 0 °C Indoor air temp: 20 °C	Surface temp., °C	35.0	35.0/27.7	31.0	35.0/27.7
		Heat output, W	301 (221)	307	268 (197)	319
		Vent. airflow, l/s	12.6	13.3	12.5	13.3
		Supply air temp., °C	17.6	18.3	15.7	18.0
3	Reduced fin-to-fin distance Vent. rad. arrangement Outdoor air temp: 0 °C Indoor air temp: 20 °C	Surface temp., °C	35.0	35.0/26.6	30.2	
		Heat output, W	394 (314)	352	346 (275)	
		Vent. airflow, l/s	12.2	12.7	12.2	
		Supply air temp., °C	24.6	22.4	22.6	
4	Mixing chamber Vent. rad. arrangement Outdoor air temp: 0 °C Indoor air temp: 20 °C	Surface temp., °C	35.0	35.0/27.0	30.5	
		Heat output, W	353 (273)	333	313 (242)	
		Vent. airflow, l/s	12.3	12.4	12.3	
		Supply air temp., °C	21.2	22.4	19.7

Figure 9. Vertical radiator section showing heat distribution in fins and air temperature sections at three heights; at 1/3 and 2/3 up the channel (AA and BB resp.) and at the outlet (CC). Fin geometry is shown by the diagram at the bottom right of each case.

3.2 Error analysis of laboratory measurements

There was potential for both systematic and statistical errors in the study. Systematic errors can be caused by inaccuracy and arrangement of the measuring devices. This could for instance be a more or less constant error in temperature readings caused by heat losses from the short water pipes where the thermocouples were built-in or by an error in calibration of the measuring devices. The most important variables to control, such as the pressure difference, supply water temperature and room temperatures, were measured by two kinds of measuring equipment to minimize systematic errors. If there were any errors these would not affect the comparison between one case and another since the same error magnitude would be reproduced in every case.

The second kind of error was statistical, that is, irregular variations in the readings such as those associated with physical phenomena such as fluctuations in air or water flow. The standard deviations of the variables are measures of the statistical errors. Of all charts shown in the results, those describing heat output were the most precise. All variables used in these calculations had low standard deviation in the readings, as previously shown in equation 1 and Table 1. As a consequence the mean R^2 -value of the graphs in Figure 3 and 4 were as low as 0.992. It should be noted, however, that each graph was made up of only three points.

$$P = \dot{m}_w \cdot c_p \cdot (\theta_{w, supply} - \theta_{w, return}) \quad (1)$$

, where P , $\dot{m}_w$, c_p , $\theta_{w, supply}$ and $\theta_{w, return}$ are the heat output, mass flow of water inside the radiator, specific heat capacity of water and temperature difference of water entering and leaving the radiator.

The supply air temperature measurements were somewhat more unreliable, while most uncertainties were related to the ventilation airflow measurements. The mean R^2 -value was 0.989 for the supply air temperature (Figures 5 and 6) and 0.938 for ventilation airflow (Figure 7). There were also larger differences in the gradients of these graphs, which were considered to be another indication of greater inaccuracy. The higher degree of errors in these results was partly due to random fluctuations in airflow (turbulence) and partly systematic periodic variations in pressure difference between the chambers (fan work influence). To minimize errors from the periodic variations, data from two full, exactly measured periods were used in every case, each of about 12 minutes duration.

4 Discussion

The overall objective of the project, of which this study was a part, was to investigate whether ventilation radiators could be made to be more efficient than other systems on the market. In this study three different aspects were investigated by laboratory measurements; heat output, air supply temperature and ventilation rate (incoming ventilation airflow). Below follows a discussion of the results.

A traditional 21 module radiator, 500 mm high and 600 mm long, positioned as normal in a room with ambient air and surroundings at 20°C typically gives out 122 W at 35°C supply water temperature (laboratory measurements). By forcing cold airflow between the panels of the same radiator the heat output markedly increases. This study shows that more than double the heat output i.e. 288 W, is to be expected in a ventilation radiator arrangement if the radiator has **traditional fin geometry and room air mixing** and the supply (outdoor) air temperature is 0.0°C. Of particular importance to note is how heat transfer responded to temperature decrease. At -7.5 °C heat output increased to 344 W and at -15.0°C heat output reached 417 W. The heat output, P , has a linear correlation with the temperature difference between heated surfaces and ambient air, as described by Equation (2). The equation also explains why convective heat output between the radiator panels is more dominant in a ventilation radiator system with 35°C supply temperature than a system with 55°C supply temperature.

$$P = \alpha_{conv} \cdot A \cdot \Delta\theta_m \quad (2)$$

, where α_{conv} is the convective heat transfer coefficient, A is the area of heat transferring surfaces and $\Delta\theta_m$ is the mean temperature difference between heated surfaces and incoming/ambient air

The traditional radiator with modified inlet had an additional 9-10% increase in heat output in relation to the radiator with mixing of room air, for both 35°C and 55°C supply water temperatures. This was because of higher flow of cold ventilation air through the system; both the heat transfer coefficient and the average temperature difference between radiator surfaces and passing air as given in Equation (2) had increased. However, as a consequence of higher airflow the supply temperature to the room became lower, the difference being up to 3°C at -15°C. A supply air temperature as low as 10°C is to be expected with this arrangement if the supply water temperature is 35°C, and this could cause discomfort for human beings in the area close to the radiator [6].

A ventilation radiator with rectangular fins has an optimum fin-to-fin distance of about 6-8 mm according to [4] ($\max(\alpha_{conv} \cdot A)$ in Equation (2)). Having twice as many fins as in a traditional radiator means this recommendation is met while the radiator panel itself can be kept unchanged and conventional welding techniques can be used for attaching the fins. **Test model 1** was based on this principle. This model had, on average, 20 % higher heat output compared to the traditional radiator with air mixing and 10 % higher than the traditional radiator with the modified inlet. Airflow decreased with 5-6 % compared to

the latter. The same trend could be seen in measurements at both 55°C and 35°C water supply temperatures.

Test model 2 had convection fins on the backside of the back panel. This gave no increase in heat output but instead increased pressure drop, which led to lower ventilation flow rate through the system. One contributing factor could have been buoyancy forces working in the opposite direction of the ventilation airflow. A large filter area on the backside of the back panel should be prioritized rather than additional convection fins.

Test model 3 consisted of less material than model 2 because the middle part of the fin array was removed to create a mixing zone. Still this model had similar performance in all aspects investigated. In other words, the positive effects of a mixing zone could be seen to be just as shown in the CFD simulations. Mixing of air half way up the channel improved heat transfer in the uppermost fin array and compensated for the reduction in fin area. The total pressure drop was similar because the extra pressure drop at the entrance to the uppermost fin array was about the same as the reduced friction losses along removed fins.

Test model 4 had the same fin configuration as model 3, the difference being that the front radiator panel was substituted with a solid metal plate. Even if this configuration lowered the total heat output to the room the supply air temperature could be maintained at a level close to that of the ventilation radiators with traditional fin geometry. It was shown that the major part of the heat transfer to the supply air generally came from the back radiator panel. This model had a higher degree of convective, heat transfer relative to radiative, heat transfer.

Colder cores of air always emerge farthest away from the heated surfaces as air rise between the “walls” formed by radiator panels and convection fins inside a radiator. **Test model 5** had staggered convection fins; parallel shifted vertical fin sections as was described under denotation f) in Figure 2. With this set up the cold cores of ventilation air were split in two when air passed from one vertical fin section to the other. Thus the coldest air came closer to heated surfaces. In addition, the insulating viscous sub layer close to heated surfaces was broken, as in a mixing chamber. Consequently the total heat transfer coefficient became higher, resulting in a higher supply air temperature at the top of the radiator where air was released to the room. These mechanisms are illustrated in Figure 10.

Figure 10. Vertical section showing flow structure around internal convection fins, temperature profiles and typical local heat transfer coefficient.

This design, however, meant that there was a considerable pressure drop every time air passed from one section to the next. It was therefore considered unsuitable for radiators relying on natural convection. To introduce staggered fins in ventilation radiators was suggested after studying the form of compact heat exchangers where comparable conditions apply [7, 8]. Staggered fins were investigated by measurements in the laboratory only, and not by CFD in [4]. The measurements showed that the ventilation airflow became about 1 l/s lower (11 %) because of increased resistance compared to having a mixing chamber, while the total heat output became around 6 % higher.

4.1 Optimal fin configuration

From this study and reference [4] it was concluded that, of all modifications tested, reduced fin-to-fin distance together with a modified air inlet had the greatest effect on heat transfer enhancement. Moreover, these two changes did not adversely affect ventilation airflow or supply air temperature. There was neither indication that such changes would increase leakage through building envelope or indoor thermal comfort. Thermal comfort with ventilation radiators and similar systems are discussed in reference [3] and [9].

Further changes to ventilation radiators may allow some savings in materials or small additional thermal efficiency improvements. It would appear that having a mixing zone is best in ventilation radiators of heights over 500mm, while in lower ventilation radiators convection fins must be present in all parts. Otherwise A in Equation (3) becomes too small and it is recommended to have staggered fins. Manglik and Bergles have developed correlations for friction and heat enhancement with staggered fins in compact heat exchangers [7]. Corberán, Cuadros and González, as well as Teruel, Nakashima and

Paglione adapted these equations to other geometries and conditions [8-10]. In the same way equations could be developed to fit the conditions and geometries of staggered fins in ventilation radiators, and similarly for mixing chambers. This would make it easier to identify the most advantageous convection fin design in ventilation radiators, depending on various optimization factors such as material cost, heat output, production methods and ventilation rate, that were outside of the scope of this paper.

Conclusions

The performance of ventilation radiators can be improved by making changes to the geometry of internal convection fins and air supply. Five experimental models were tested in a laboratory to verify findings of a previous CFD study and further improve heat transfer mechanisms. Test conditions represented that of a Swedish building in winter climate; indoor under pressure of -10 Pa and outdoor temperature reaching from -15 °C to 0 °C. The focus was on heat output, with ventilation rate and supply air temperature also investigated.

The CFD simulations were confirmed (up to a maximum deviation of 12 %) and the following conclusions could be stated. Ventilation radiator performance can be improved by avoiding room air mixing at the inlet and decreasing the distance between convection fins to 6-8 mm. This would mean about 20 % increase in heat output, a 10 % increase in ventilation rate and 3-5 °C higher air supply temperature to the room compared to an arrangement with a traditional radiator (i.e. traditional convection fin geometry and room air mixing). Having some sort of air mixing half way up the fin array may bring about additional improvements. It is proposed that radiators greater than 500 mm in height should have a mixing zone, while lower radiators should have staggered fins. Further studies are needed to investigate this.

References

- [1] European Environment Agency report No 8/2006: Energy and environment in the European Union - Tracking progress towards integration, ISBN 92-9167-877-5
- [2] P. Wargocki, et. al., Ventilation and health in non-industrial indoor environments: report from a European multidisciplinary scientific consensus meeting (EUROVEN), Indoor Air 12(2) (2002), pp.113-28
- [3] J. A. Myhren, S. Holmberg, Design considerations with ventilation radiators: Comparisons to traditional two-panel radiators, Energy and Buildings 41 (2009), pp. 92–100
- [4] J. A. Myhren, S. Holmberg, Improving the thermal performance of ventilation radiators – the role of internal convection fins, submitted to Thermal Sciences
- [5] www.purmo.com/se/ladda-hem-filer/effektsimulering.htm, Purmo Air and Purmo Compact & Ventil Compact (in Swedish)
- [6] ISO EN 7730:1994, Moderate thermal environments—determination of the PMV and PPD indices and specification of the conditions for thermal comfort, revised version, Geneva, International Organization for Standardization, 1994.
- [7] R. M. Manglik, A. E. Bergles, The Thermal-Hydraulic Design of the Rectangular Offset-Strip-Fin Compact Heat Exchanger, Compact Heat Exchangers, Hemisphere Publishing, New York, pp. 123-149 (1990)
- [8] J. M. Corberán, E. Cuadros, K. González, Pressure drop characterisation of compact heat exchanger channels, 5th European Thermal-Sciences Conference (2008)
- [9] A. Ploskić, S. Holmberg, Low-temperature baseboard heaters with integrated air supply - An analytical and numerical investigation, Building and Environment 46 (2011), pp. 176 - 186
- [10] M. H. Teruel, C. Y. Nakashima, P. Paglione, Rectangular offset strip-fin heat exchanger lumped parameters dynamic model, Brazilian Symposium on Aerospace Eng. & Applications 3rd CTA-DLR Workshop on Data Analysis & Flight Control (2009)