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Ventilation efficiency in a low-energy dwelling setting – a parameter study for larger rooms

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SUMMARY

Mechanical balanced ventilation systems typically is applied in new and renovated dwellings in The Netherlands. The application assumes an adequate ventilation efficiency but this has not been confirmed for larger rooms (e.g. living rooms with kitchen attached). This study investigates ventilation efficiency for balanced ventilation of an L-shaped geometry, representative for a typical living room in new dwellings in The Netherlands.

Computational Fluid Dynamics (CFD) is applied for the assessment and for a parameter study into diffuser type (discharge pattern axial or radial), position, heating and cooling. Ventilation efficiency, thermal and comfort indicators are applied for the assessment. Validation, though limited, and diffuser boundary condition specification are supported by measurements.

Results show an adequate ventilation efficiency for L-shaped geometries, i.e. approaches a perfect mixing situation. However, diffusers with radial discharge pattern are advised for application in dwellings with balanced ventilation to improve the robustness of mechanical ventilation systems in dwellings.

PRACTICAL IMPLICATIONS

Use of diffusers with radial discharge pattern is advised for mechanical balanced ventilation systems as applied in Dutch dwellings to improve the ventilation efficiency in larger rooms.

KEYWORDS

Room ventilation, Local air change index, CFD, Diffuser selection, Diffuser position

1 INTRODUCTION

In the design of new dwellings energy consumption is important. In well insulated dwellings ventilation is accountable for a significant part of the total heat losses. In the Netherlands balanced (mechanical) ventilation systems have been introduced to tackle this problem as these systems generally are incorporated with heat recovery. The importance of sufficient ventilation with respect to health outcomes has been confirmed numerous times (Sundell et al. 2011). Though provision of a set air flow rate in principle is secured with balanced ventilation, the design of these systems still encounter issues with respect to noise and maintenance (Verkade et al, 2009). With respect to use generally minimum set-points for the flow rate are in place.

Though much attention has been put on the efficiency of the fans and the losses in the system, little is known about the effectiveness with which ventilation is supplied into the room. This includes the placement of diffusers. In addition to that we see that temperature differences as present in dwellings reduce due to improved insulation and the application of low temperature heating. Also air heating and cooling come into place. As such the distribution of the air in a larger room also is to be guaranteed through the momentum and the other supply characteristics. With the exception of prescribed minimum flowrates, Dutch building regulations and practical guidelines provide no or minimum information with respect to the required effectiveness (e.g. Bouwbesluit, 2012; ISSO, 2010). As main rooms (living room and kitchen) in dwellings are becoming larger in The Netherlands the air change efficiency is yet to be determined. The assumption is that a mixed situation is arrived at, but no research has been found to confirm this assumption. Questions from practice (Cremers 2014) indicate the lack of knowledge and the interest in the topic.

L-shaped geometries and alcoves influence the airflow in the room, just as the spatial layout of furniture in a room can also influence airflow patterns (Huang & Lin, 2014). Most research on balanced ventilation systems has been done using simple room shapes (e.g., Al-Sanea, Zedan, & Al-Harbi, 2012) (Hu, 2003). In this study the performance of mixing balanced ventilation systems is investigated when these systems are applied in larger, more complex geometries. For this study a single L-shaped geometry is used. The investigation includes different diffuser types and positions, heating and cooling cases, and the effect of sunlight. The research question to be answered is whether for L-shaped rooms the ventilation effectiveness resembles a mixing situation for such cases.

2 METHODS

For the analysis of the ventilation effectiveness in larger rooms the Computational Fluid Dynamics (CFD) technique is used. The mean age of air concept is applied to determine the local air change index (LACI; ε_p^a) (Mundt et al, 2004):

$$\varepsilon_p^a = {}^{\tau_n}/_{\bar{\tau}_p} \cdot 100\% \tag{1}$$

Where τ_n is the nominal time constant and $\bar{\tau}_p$ is the local mean age of the air in a small volume around point p. To assess thermal comfort the effective draft temperature (θ_{ed}; ASHRAE 2013a) and draft rating is applied (CEN, 2005).

A (limited) validation study has been performed applying measured data as came available from a climate chamber study in which LACI measurements were performed (4 positions; de Coo, 2011). The validation study approached a mixing ventilation case. After a grid dependency check on the velocity magnitude at the investigated measurement positions ($\Delta v/v_{avg}$ < 5%) the trend found for LACI was similar as for the measurements though in absolute values a shift of 5-10%-point was derived. Limitations were mainly found in the correct definition of the supply conditions. Earlier research nevertheless has shown the capabilities of CFD with respect to simulation of ventilation effectiveness indicators (Roos 1999).

To assure realistic boundary conditions for the CFD study detailed measurements have been performed to define the inlet boundary conditions. Diffusers with a radial and axial discharge pattern (indicated as ‘radial diffuser’ and ‘axial diffuser’ in the remainder of the text) as typically found in dwellings have been used in this study (Figure 1). The diffusers both require other modelling methods to be implemented correctly into the CFD-model. Measurements have been performed to determine the inlet conditions of the diffusers and develop the boundary conditions as applied in the CFD model. For the axial diffusers, in line with (Srebric and Chen, 2000), the box method has been applied. As the effective area is similar to the actual supply area for the radial diffuser the direct description method is applied.

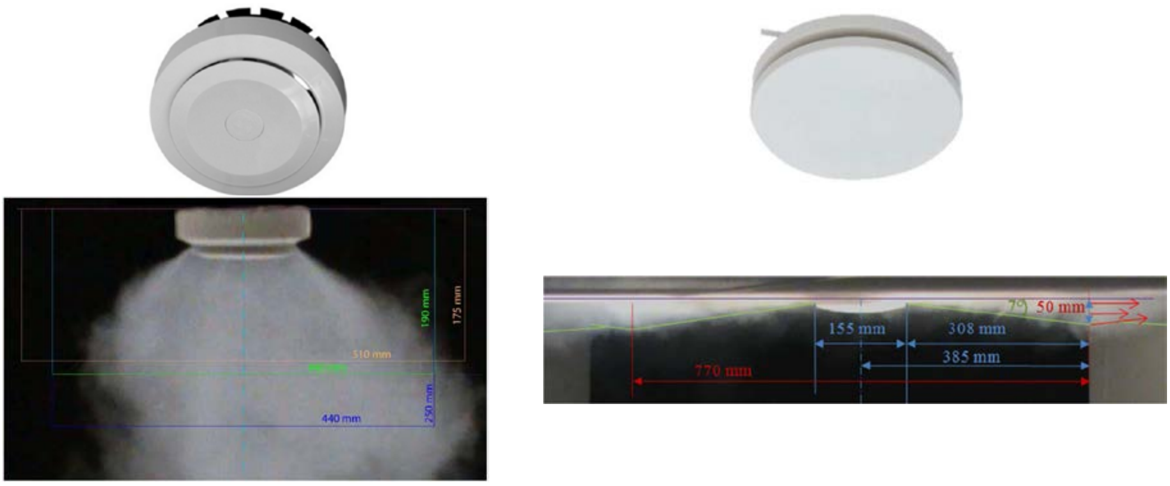


Figure 1. Axial diffuser (left), radial diffuser (right); Bottom pictures show (vertical) smoke test results.

For the axial diffuser the box size was derived to be 0.44 m wide and 0.25 m high, centrally positioned. In line with Kotani et al. (2002) a 5×5 grid was applied to define the supply conditions at the bottom side of the box. Velocity magnitude and turbulence intensity were measured with an omnidirectional hot-sphere anemometer [±0.02 m/s ±1.5% reading, min 0.05 m/s]. Validation measurements were performed to compare the modelled supply (axial diffuser: at 0.3 m from the box; radial diffuser: at 1.0 m from the supply). For this a CFD model was developed of the room in which the supply conditions were measured. Angular directions for the axial diffuser box method were calibrated based on smoke visualization and improved agreement with the validation data. Due to the complexity of the supply and the applied discretization a complete fit could not be obtained. However, assessment of the LACI distribution and absolute values for the room in which the measurements were performed indicated that sensitivity of the supply angle for this performance indicator was small. Outside the jet differences in LACI were less than 2%-point. For the radial diffusers validation showed the simulated values to be within the measured accuracy of the applied anemometer.

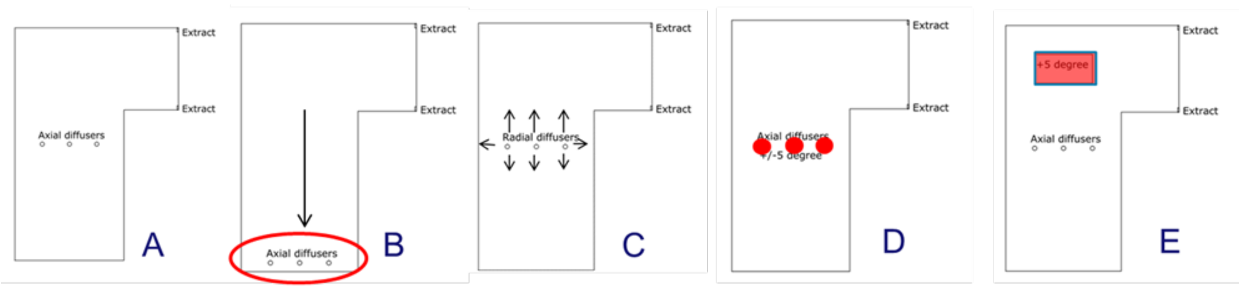


Figure 2. A. Base case, B. Envelope case, C. Radial diffuser case, D. Heating/cooling case and E. Hot patch case.

To investigate the ventilation effectiveness for a larger room a single geometry is defined. In The Netherlands terraced dwellings typically have L-shaped living rooms. The average size of these rooms is 8.5 by 6 meters (Arts, 2011), which is used as size for the simulated room model. The base case model (Figure 2.A) has dimensions 8.5×6.0×2.8 meter, as are the dimensions of all case studies (for the L-shape a volume of 5.5×2.0×2.8 meter is removed). Three supply diffusers are positioned equidistant in the core of the room, as advised by the manufacturer. Distance between the supply diffusers and the side walls is 1 meter. The base case is isothermal and assumes axial supply diffusers. The extract diffusers are positioned in the tip of the L-shape assuming the location of the kitchen in this part of the room. The air change rate is set at ~0.66 h^{-1} (3×25 m^3/h). Alternatively, supply diffusers are positioned near the envelope wall (Figure 2.B). Also in this case the distance to the side wall is set at 1.0 meter. The third case assumes the base case, but now with radial diffusers incorporated (Figure 2.C). The effect of air heating/cooling is investigated for the base case by assuming a supply temperature of 17 ºC, while walls have a temperature of 21 ºC. In heating mode 22 ºC and 19 ºC are assumed respectively, for the floor 20 ºC is applied (Figure 2.D). These temperatures resemble conditions as can be expected in the intermediate seasons for the Dutch climate, indicating limited cooling/heating capacity. For this case also the radial diffuser has been applied to investigate its effect on the draft temperature. Finally, the effect of the sun is investigated by assuming a heat flux at a small area (hot patch: 10 W/m^2, 8.6 m^2 floor area; Figure 2.E). The walls are assumed adiabatic while the supply temperature is 19 ºC. In any case an empty room is investigated. The analysis of the cases and their comparison is focussed on 13 positions in the room (Figure 3 *left*; different heights).

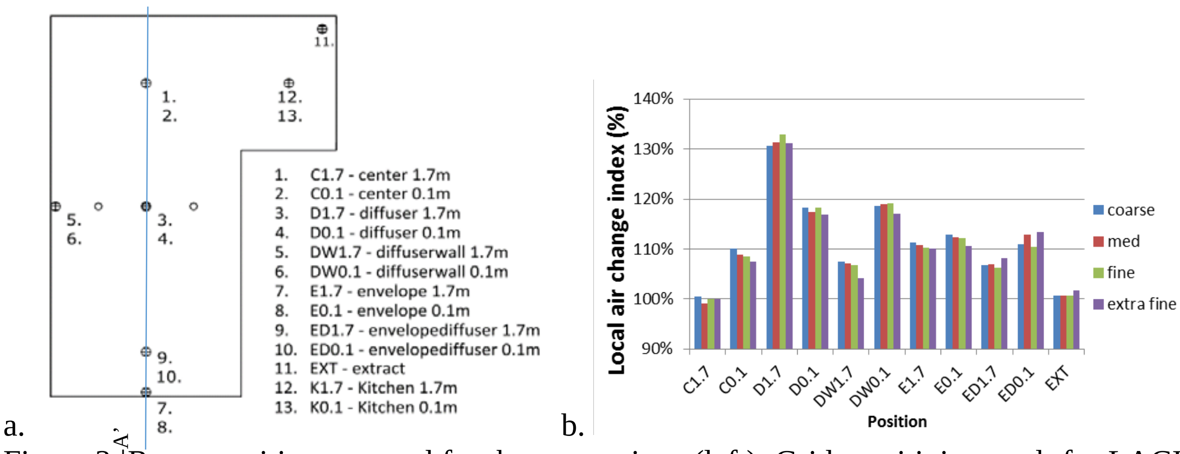


Figure 3: Room positions as used for the comparison (left); Grid sensitivity result for LACI at the room positions (right; K1.7 and K0.1 not included).

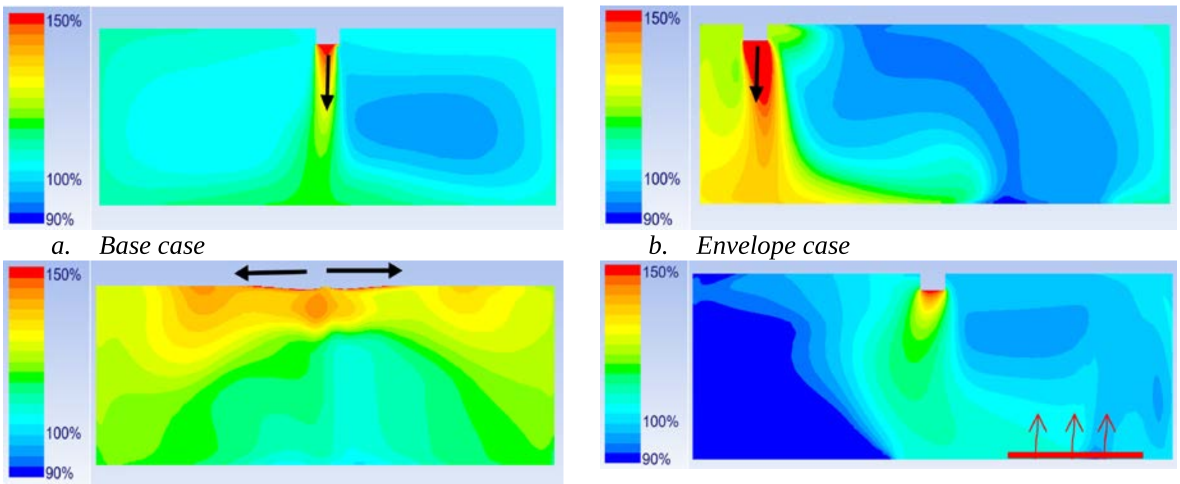
The CFD code used for all simulations is Ansys Fluent (15.0; Ansys 2013). From the validation study and the diffuser measurements and modelling, grid and settings information have been deduced. For the discretisation, structured hexahedral cells are used throughout the main parts of the room. For modelling of the supply diffusers unstructured tetrahedral cells have been applied. Grid dependency was checked on several monitoring positions in the room (see Figure 3 *left*) with four grid sizes at $\sqrt{2}$ increment starting at 750,000 cells. Grid sensitivity showed that for the LACI performance indicator difference between coarsest and finest grid was limited (Figure 3 *right*). For the velocity magnitude maximum relative difference between medium grid and finest grid was 6%. Following these results medium grid size (1.5 million cells) was used for the model. Maximum $y+$ value for this grid arrived at 5.5 where the jet impinges the floor, otherwise y+ values are between 1 and 4.

Turbulence has been modelled applying the RNG k-ε turbulence model (Zhang et al, 2007). Near wall modelling is taken care of by using the enhanced wall treatment. For non-isothermal simulations the energy equation is included. Buoyancy is modelled using the Boussinesq approximation. In the model a constant density of air of 1.225 kg/m^3 and an operating temperature of 293.15K is assumed. In none of the investigated cases radiation is included (i.e. only convective heat transfer). Mean age of air is calculated afterwards in a separate post-processing simulation, leaving flow and temperature field constant.

The solver process applied a two-step approach. An initial solution is calculated first through a steady-state simulation. In a second step the simulation is continued in transient mode. The result of the first step is used as input for that. This approach was assumed as convergence was not arrived at for the steady-state situation. Unsteady simulation was pursued for 30 minutes of simulation time. Although the nominal time constant of this model is nearly 90 minutes, the initial steady-state simulation ensured that there is a consistent airflow in the model. Monitoring points were used to monitor temperature and velocity development in the model. After 30 minutes stabilization of the flow was found, though velocity fluctuations with a standard deviation of <0.01 m/s remained. This indicates (local) unsteady flow conditions. All simulations have been performed with second order discretization.

3 RESULTS

Figure 4 and 5 present some examples of the results obtained from the analysis. Figure 4 presents the LACI distribution for section A-A' (Figure 3 *left*) of the room. Figure 5 presents results for the LACI at the defined monitor points. The values for the local air change index should be read such that LACI>100% indicates a better ventilation efficiency when compared to a (theoretical) perfect mixing situation (=100% throughout the room). Values below 100% indicate a reduced ventilation efficiency. As perfect mixing does not exist, by definition LACI will be >>100% close to the supply. Figure 5d presents the effective draught temperature. Figure 6 presents information on the velocity magnitude for case A-D. For case D Figure 6d compares the effect of the diffuser type in case of air cooling. In the discussion additional information on the results are included.



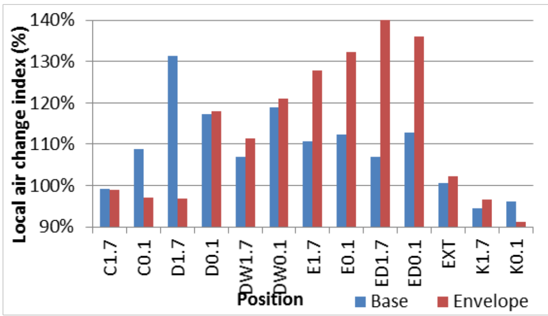
a. *Base case* b. *Envelope case*

c. *Radial diffuser* d. *Hot patch*

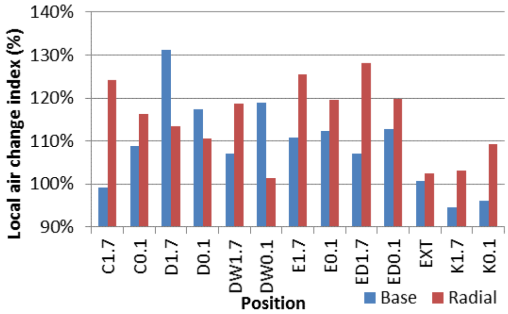
Figure 4. LACI on section A-A' (Figure 3 left) for four of the investigated cases.

4 DISCUSSION

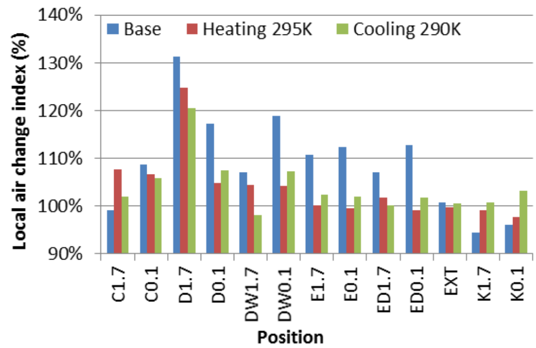
In the comparison between the Base case (A) and the Envelope case (B; Figure 4a-b, 5a and 6a), the jet area has the largest influence on both velocity and local air change index. Outside of the jet area velocities are low, and LACI is between 90% and 110% for both cases. Assuming the zone air distribution effectiveness as provided in ASHRAE (2013b) as a reference for the minimum required LACI, both cases ventilate efficiently (e.g. LACI>80% for heating cases; LACI>100% for cooling cases). The choice where to place the diffuser in the living room therefore could be based on other factors such as draft (Figure 6a). The high velocities may cause draft below the diffusers. In a cooling situation the condition can become more critical (Figure 5d, 6c).



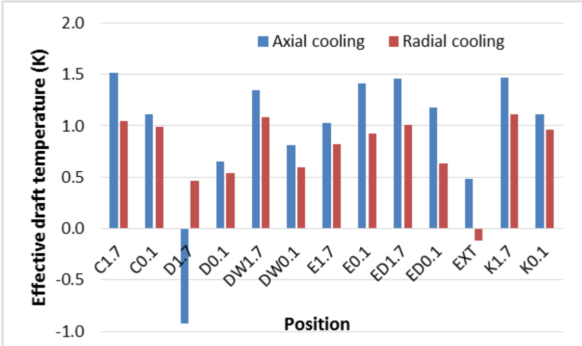
a. LACI comparison Base and Envelope case at monitored points.



b. LACI comparison Base and Radial case at monitored points.

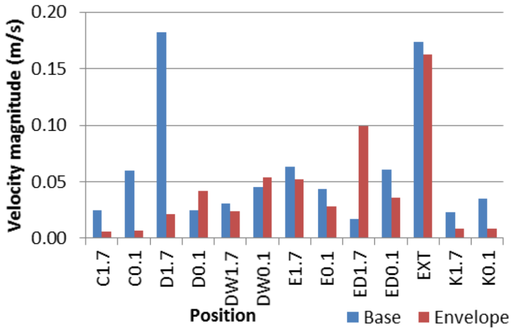


c. LACI comparison Base and Heating/Cooling case at monitored points.

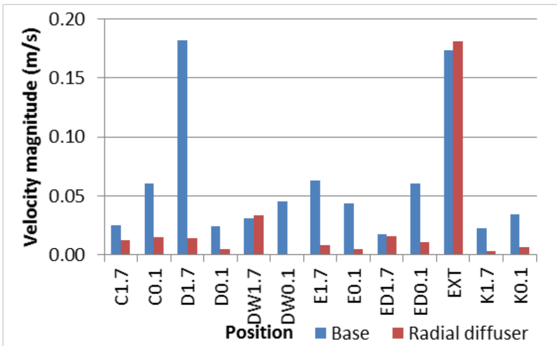


d. Effective draft temperature for Cooling case with axial diffuser and with radial diffuser.

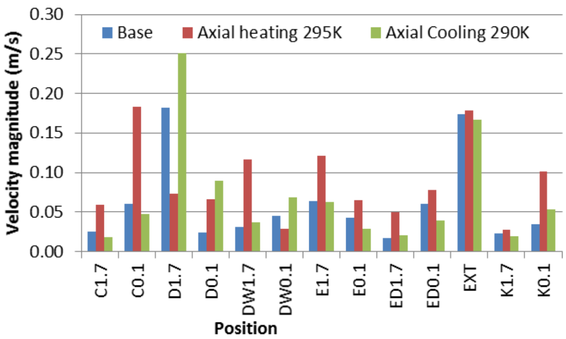
Figure 5. a-c. LACI at monitor points for four of the investigated cases. d. effective draft temperature at monitor points for cooling case for the two investigated diffusers.



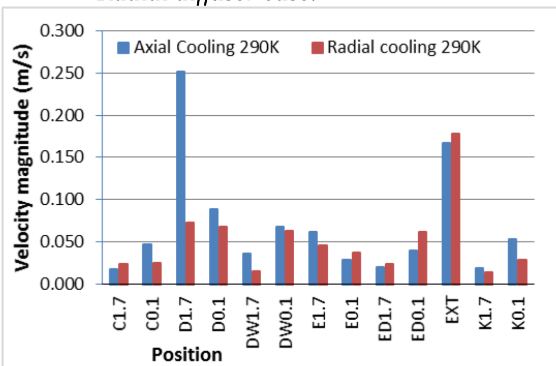
a. Velocity magnitude comparison Base and Envelope case.



b. Velocity magnitude comparison Base and Radial diffuser case.



c. Velocity magnitude comparison Base and Heating/Cooling case.



d. Velocity magnitude comparison Axial and Radial diffuser cooling case.

Figure 6. a-c. Velocity magnitude at monitor points for four of the investigated cases. d. Velocity magnitude at monitor points for cooling case for the two investigated diffusers.

The diffuser type has a large influence on both velocity magnitude and LACI in the room (Figure 4a,c; Figure 5b, Figure 6b). The axial diffuser results in a distinctive jet directed into the living zone of the room. Radial diffusers produce a jet along the ceiling. Due to this induction of air takes place outside the living zone resulting in overall lower velocities in the living zone (at the monitoring points) when compared to the axial diffuser. In addition to that, based on the comparison, radial diffusers result in a better ventilation efficiency in almost all points; averaging at 115%, compared to 100% for axial diffusers.

When the room air is heated via an axial diffuser the velocities in the room increase, except for the jet velocity which decreases. In a cooling case the jet velocity is increased while the other velocities remain similar (Figure 6c). In both cases the LACI at the monitored positions change to values more close to 100% indicating that buoyancy increases mixing in the living zone of the room (Figure 5c). With respect to draft, the use of a radial diffuser improves the effective draft temperature (Figure 5d). The axial diffuser causes a 16% predicted draft rating below the diffuser, while the radial diffuser only has a 6% draft rating at the same position.

Heating from sunlight does affect the local air change index in the non-heated part of the room negatively by causing an increased flow in the heated part of the room (Figure 4d). The local air change index was reduced by 18%-point for the non-heated part in this case study.

Overall the results indicate that for the investigated case (geometry) ventilation effectiveness in the room generally approaches a mixing condition or better. As such the position and type of diffusers applied is less problematic and other performance criteria come in place to determine its position. Draft in that context will be an important consideration. As the radial diffuser shows an increased ventilation effectiveness, in combination with a reduced draft potential its application may be preferred. When located in the center of the room, in line with current manufacturer guidelines, it can provide a more robust solution also in non-isothermal cases.

The above results are derived from a CFD model that was validated indirectly from available measurement data for a different configuration and from detailed information for setting the boundary conditions. A future study could be designed to allow direct comparison to measurements for the actually investigated geometry. Following the model development chosen, the model nevertheless is assumed fit for the presented parameter analysis.

More important limitations are found in restrictions as applied to the model, e.g., steady state boundary conditions are assumed with no occupant movement nor extra internal heat sources. Also furniture has not been included to simplify the analysis. In reality people will be present, generating heat and movement, furniture will be placed, and heat generating appliances are used. These factors will influence the airflow and ventilation efficiency (Nielsen 2004, Nielsen 2015). It is assumed that this will result in increased mixing and that the ventilation efficiency approaches a mixing situation while draft problems are not deteriorated. The assumptions for the non-isothermal case only present two examples of non-isothermal boundary conditions. From the results for the heating and cooling case mixing was found to increase, supporting the above assumption. The hot patch indicates that specific local effects can influence the outcome negatively. The example case, assuming sun, however may be regarded as a transient effect.

5 CONCLUSIONS

Based on the research presented the ventilation effectiveness for larger L-shaped rooms as found in (low-energy) dwellings in the Netherlands can still be assumed as (near) mixing when diffusers are placed and applied according to manufacturers’ guidelines. Robustness of the ventilation can be improved by applying a radial instead of an axial diffuser if other (comfort) performance indicators are taken into account as well. In all cases a correct installation of the system and alignment to the minimum (regulatory) flow rate requirements is assumed.

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REFERENCES

Al-Sanea SA, Zedan MF and Al-Harbi MB. 2012. Heat transfer characteristics in air-conditioned rooms using mixing air-distribution system under mixed convection conditions. *International Journal of Thermal Sciences*, 59, 247–259.

Ansys. 2013. ANSYS Fluent User's Guide. Release 15.0. ANSYS, Inc. Canonsburg (USA).

Arts S. 2011. De toekomst van de doorzonwoning: bewezen kwaliteit. http://www.bestaandewoningbouw.nl/de-toekomst-van-de-doorzonwoning-bewezen-kwaliteit/ (last visited 2015-12-15)

ASHRAE. 2013a. ASHRAE Handbook – Fundamentals. American Society of Heating, Refrigerating and Air-Conditioning Eng., inc. Atlanta (USA). Chapter 20.

ASHRAE. 2013b. ASHRAE Standard 62.1:2013. Ventilation for acceptable indoor air quality. American Society of Heating, Refrigerating and Air-Conditioning Eng., inc. Atlanta (USA).

Bouwbesluit 2012. Building Decree online http://www.bouwbesluitonline.nl/Inhoud/docs/wet/bb2012 (last visited 2015-12-15)

CEN. 2005. ISO 7730:2005(E) Ergonomics of the thermal environment. European Committee for Standardization, Brussels (Belgium). 52 pages.

De Coo L. 2011. Ventilation efficiency in a room with mechanical ventilation. Internal report. Eindhoven University of Technology, Eindhoven (The Netherlands). 38 pages.

Cremers B. 2014. *Personal communication*.

Hu SC. 2003. Airflow characteristics in the outlet region of a vortex room air diffuser. *Building and Environment*, 38(4), 553–561

Huang CH and Lin PY. 2014. Influence of spatial layout on airflow field and particle distribution on the workspace of a factory. *Building and Environment*, 71, 212–222.

ISSO. 2010. ISSO 62: Centrale gebalanceerde ventilatiesystemen met warmteterugwinning in woningen en woongebouwen. Kennisinstituut voor de Installatiesector, Rotterdam (The Netherlands). 215 pages.

Kotani, H., Yamanaka, T., & Momoi, Y. (2002). Cfd simulation of airflow in room with multi-cone ceiling diffuser using measured velocity and turbulent parameters in large space (pp. 117–120). Proc. 8th international conference on air distribution in rooms - ROOMVENT2002.

Mundt E, Mathisen HM, Nielsen PV and Moser A. 2004. Ventilation effectiveness. Rehva Guidebook No.2. Federation of European Heating, Ventialtion and Air-conditioning Associations .

Nielsen PV. 2004. Computational fluid dynamics and room air movement. *Indoor Air* 2004; 14 (Suppl7): 134–143

Nielsen PV. 2015. Fifty years of CFD for room air distribution. *Building and Environment*, 91, 78-90.

Roos A. 1999. On the Effectiveness of ventilation. PhD-thesis. Eindhoven University of Technology, Eindhoven (The Netherlands). 216 pages.

Srebric, J., & Chen, Q. (2000). Simplified Numerical Models for Complex Air Supply Diffusers. *HVAC&R Research*, 8(3), 277–294.

Sundell J, Levin H, Nazaroff WW, Cain WS, Fisk WJ, et al. 2011. Ventilation rates and health: multidisciplinary review of the scientific literature. *Indoor Air*, 21(3), 191–204.

Verkade A, Duim F, Merckx A, Meijer A, and ten Bolscher G. 2009. Balansventilatie: onderzoek naar de effectiviteit van de genomen verbetermaatregelen en naar de tevredenheid van de bewoners van de wijk Vathorst. GGD Midden-Nederland, Amersfoort (The Netherlands), 62 pages.

Zhang Z, Zhang W, Zhai Z and Chen Q. 2007. Evaluation of various turbulence models inpredicting airflow and turbulence in enclosed environments by CFD: Part-2: comparison with experimental data from literature, *HVAC&R Research*, 13(6), 871-886.