Ventilation effectiveness of alternating façade-integrated ventilation devices

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Abstract

Domestic ventilation units are usually dimensioned solely based on the provided air flow rate instead of their ability to renew the room air volume. This can be quantified by performing a tracer gas analysis. In the presented research project, a total of 15 tests are carried out in a climatic chamber representing a single room equipped with two alternating façade-integrated ventilation devices with regenerative heat recovery. The tests include summer, winter and isothermal supply air conditions. Further investigations are dedicated to the effect of thermal convection due to human heat dissipation on the room air flow. In dependence on these boundary conditions, the determined air exchange efficiency varies, lagging the expected range $0.5 < \varepsilon^a < 1$ in almost all cases.

Keywords: decentralised ventilation, ventilation effectiveness, air exchange efficiency

1 Ventilation Effectiveness

The primary goal of ventilation is to provide fresh air while simultaneously removing internally produced CO_2 and humidity loads. For cases in which no or just fragmentary information on the used air source is available the evaluation of air renewal assessment parameters is suitable. In order to do so it is necessary to evaluate the age of air τ (in s) introduced by Sandberg (1981) as an instrument to analyse ventilation efficiency. The local air age $\bar{\tau}_{P,i}$ (in s) of each coordinate in the room can be evaluated by the methods described by Sandberg (1981) and DIN ISO 16000-8 (2008). The spacial average of air age $\langle\bar{\tau}\rangle$ (in s) is given by equation (1-1) and the reciprocal of the air exchange rate $\bar{n}$. This approach has been used instead of the commonly applied definition like in Mundt et al. (2004, p. 27) based on concentration measurements in the exhaust air ducts. This approach has been chosen since of the alternating behaviour of the duct concentration of the evaluated ventilation devices.

$$\langle\bar{\tau}\rangle = \frac{\sum_{i=1}^{N} \bar{\tau}_{P,i}}{N} = \frac{1}{\bar{n}} \tag{1-1}$$

To evaluate the global absolute air exchange efficiency ε^a (in -) the definition according to Skaaret (1986, pp. 420–421) will be applied. The equation (1-2) following this definition is already widely accepted and includes as well the statement of Novoselac and Srebric (2003, p. 2).

$$\varepsilon^a = \frac{\tau_{nom}}{2} \cdot \frac{1}{\langle\bar{\tau}\rangle} = \frac{1}{2} \cdot \frac{\bar{n}}{n_{nom}} \tag{1-2}$$

For the evaluation of the situation at an arbitrary point i both Novoselac and Srebric (2003, p. 3) as well as Mundt et al. (2004, p. 24) mention the local air exchange index ε_i^a (in -). However, since ε_i^a does not relate to the actual spacial average room age of air $\langle\bar{\tau}\rangle$, it is not unambiguously evident whether zones of the room are ventilated above or below spatial average. Therefore, the definition by Skaaret (1986, p. 421) of the local air exchange indicator $\varepsilon_{N,j}^a$ (in -) according to equation (1-3) is preferably used in this paper. In case of ideal mixing, the local age of air $\bar{\tau}_P$ is the same throughout the room. This results in a local air exchange indicator $\varepsilon_{N,j}^a = 1\ \forall j \in \mathbb{N}$ and a global air exchange of $\varepsilon^a = 0.5$. If the local age of air at any point j in the room ($\bar{\tau}_{P,j}$) is different than the average of all local ages of air in the room $\langle\bar{\tau}\rangle$, the resulting air exchange indicators are $\varepsilon_{N,j}^a \neq 1\ \forall j \in \mathbb{N}$. Values $\varepsilon_{N,j}^a > 1$ indicate better ventilated areas with local ages of air $\bar{\tau}_P$ below the spatial average age of air $\langle\bar{\tau}\rangle$.

$$\varepsilon^a_{N,j} = \frac{\langle \bar{\tau} \rangle}{\bar{\tau}_{P,j}} \quad (1\text{-}3)$$

2 Methods

The aim of this research project is to investigate whether or not the currently available measurement and evaluation methods to characterize the ventilation efficiency are applicable. The units investigated are alternating façade-integrated ventilation devices with regenerative heat recovery (push-pull devices). An axial fan inside the unit delivers a maximum airflow of 15 $m^3 \cdot h^{-1}$ resp. 30 $m^3 \cdot h^{-1}$ and reverses the flow direction every 70 seconds. During the exhaust air phase, the regenerative heat exchanger is charged with thermal energy of the room air in order to subsequently heat the incoming supply air.

The investigations are carried out in a full-scale inner climate chamber (ICC) (2.77 m x 2.70 m x 5.00 m) connected to an external climate chamber (ECC). A passive-house facade with one window and two push-pull devices each to the right and left of it (clear distance 1.35 m) is integrated into the 4 m^2 opening in between ICC and ECC. Initially, the air of the ICC corresponds to the conditions of the surrounding laboratory. In the following the influence of the various thermal conditions such as, unaffected ($\vartheta_{ECC} \approx \vartheta_{ICC}$), summer and winter in the ECC on the ICC air distribution can thus be examined. In order to reproduce these scenarios, the supply air to the ECC is heated with fan heaters in the summer case, while cold air from the vicinity of the laboratory building is used to create winter conditions. All conditions achieved during the measurements are summarized in Table 1. In a further step, a dummy (300 W) was placed in the middle of the room. A detailed summary of the methodology and all considered measurement uncertainties can be found in Auerswald (2020) und Hörberg (2019).

Table 1. Hygrothermal conditions

Condition	*Temperature* $\bar{\vartheta}_{ECC}$ *in °C*	*ϑ-range* R_ϑ *in °C*	*Abs. humidity* $\bar{X}_{ECC}$ *in* $g(H_2O)\cdot kg(da)^{-1}$	*X-range* R_X *in* $g(H_2O)\cdot kg(da)^{-1}$
unaffected	22.1	3.2	4.9	3.6
summer	36.9	10.5	4.2	2.9
winter	8.3	7.6	4.0	1.8

3 Results

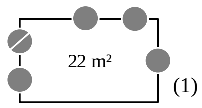
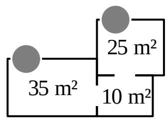
For the global air exchange efficiency ε^a, strong variances depending on the supply air temperature could be detected as can be seen in Table 2. It is also noticeable that the values in Table 2 for the nominal volumetric air flow acc. DIN 1946-6 (2018) $\dot{V} = 28\ m^3 \cdot h^{-1}$ are worse than those for the lower $\dot{V} = 15\ m^3 \cdot h^{-1}$. The values $\varepsilon^a < 0.5$ determined for almost all cases indicate noticeable short-circuit currents. Solely in case of additional thermal convection the global air exchange improves up to $\varepsilon^a = 0.60$.

Table 2. Global air exchange efficiencies ε^a of various experimental setups

Supply air	$\dot{V} = 15\ m^3 \cdot h^{-1}$	$\dot{V} = 28\ m^3 \cdot h^{-1}$
isothermal	0.38 ± 0.11	0.34 ± 0.07
warm ($\bar{\vartheta}_{ECC}$ = 35.0 °C)	0.50 ± 0.14	0.43 ± 0.09
cold ($\bar{\vartheta}_{ECC}$ = 8.3 °C)	0.43 ± 0.11	0.35 ± 0.07
isothermal + dummy	0.60 ± 0.15	0.41 ± 0.07

Similar scientific work on rooms equipped with push-pull devices, summarized in Table 3, indicate that the air exchange efficiency ε^a is close to ideal mixing and often indistinguishable given the measurement uncertainty. The values obtained in the present work fall short of this expectation.

Table 3. *Literature review of air exchange efficiencies ε^a in push-pull devices equipped rooms*

Author	*System*	*Boundary conditions*		*Method*	*ε^a*
Röder		$\dot V$ t_{cyc} $\Delta\bar\vartheta$	$\approx 43\ m^3 \cdot h^{-1}$ (2),(3) no value = -13 K	CFD	0.49 … 0.55
Merzkirch		$\dot V$ t_{cyc} $\Delta\bar\vartheta$	$\approx 30\ m^3 \cdot h^{-1}$ no value no value	CFD + Measure.	0.45 … 0.55
Merckx et al.	n.a.	Unit $\dot V$ t_{cyc} $\Delta\bar\vartheta$	Renson Endura Twist $= 30\ m^3 \cdot h^{-1}$ = 30s = -20 K	Measure.	0.53
Merckx et al.	n.a.	Unit $\dot V$ t_{cyc} $\Delta\bar\vartheta$	O.ERRE Tempero eco 150 ceram = n.a. = 70s = -20 K	Measure.	0.51

(1) Various combinations of balanced push-pull devices
(2) nominal ventilation according to DIN 1946-6 (2018)
(3) Twin-devices: two push-pull devices in one casing

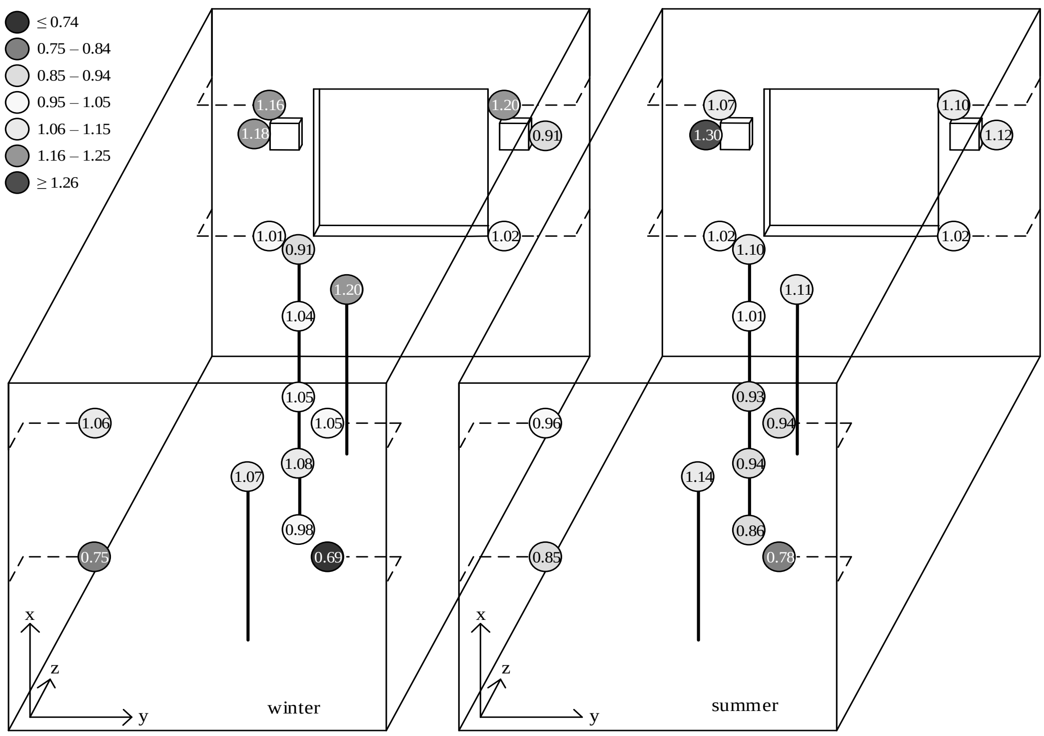


Figure 1. *Local air exchange indices $\varepsilon^a_{N,j}$ (2-3) at $\dot V = 15\ m^3 \cdot h^{-1}$ at winter (left) and summer (right) supply air conditions as well as isothermal supply air conditions*

The considerations presented below are focusing on the case of the smaller airflow $\dot{V} = 15\ m^3 \cdot h^{-1}$. Almost ideal mixing ventilation, as often described in scientific literature for push-pull devices, cannot be confirmed. The local air exchange indices $\varepsilon^a_{N,j}$ vary within a range of $0.91 \le \varepsilon^a_{N,j} \le 1.34$ for the isothermal baseline measurement. The range of all cases studied is $0.69 \le \varepsilon^a_{N,j} \le 1.34$ (nominal air exchange indices $0.53 \le \varepsilon^a_i \le 1.59$). The specific values for the investigated setups are graphically illustrated in

Figure ***1*** for a better spatial understanding.

4 Conclusion

The results show that the sole indication of the volumetric air flow rate conveyed by push-pull devices is not enough for evaluation of the air renewal in the room. The flow behaviour changes due to varying boundary conditions. Whilst the isothermal baseline measurement falls significantly short of the expected range $0.5 \le \varepsilon^a \le 1$, the air exchange efficiency increases to a maximum of $\varepsilon^a = 0.6$ with additional thermal convection. Other than initially assumed, no mixing ventilation occurs, but a spatially inhomogeneous distribution with pronounced short-circuiting currents. The installation of both ventilation units in the same façade side as specified in the climate chamber does not seem optimal. Further distance between the communicating ventilation units is assumed to improve air exchange efficiency. Further research is therefore also required on the placement of domestic ventilation units in the room.

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