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Experimental Quantification of Air Volume Flow by Natural Ventilation through Window Opening

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ABSTRACT

In this paper, a new mathematical model was developed experimentally to quantify the air volume flow by natural ventilation through window tilt opening. The experiment was carried out with two test facades in a laboratory building in Aachen, Germany. The test facades are equipped with windows in two different dimensions, and they are located under real weather conditions. The modelling data were determined by means of more than 70 tracer gas measurements with CO_2. The new model represents that the volume flow rate through the window opening can be estimated more reliably by the measured pressure difference between the outdoor and indoor air. The results will be used for a development of a hybrid ventilation control.

INTRODUCTION

Hybrid ventilation systems are able to provide supply air by natural (via window opening) or mechanical ventilation. Depending on ambient boundary conditions and ventilation strategies, natural ventilation by window opening could be quite suitable and even more energy efficient than mechanical ventilation to maintain the thermal comfort and a good indoor air quality (Spentzou, Cook, and Emmitt 2018). To decide, if natural ventilation is sufficient, the supplied air volume flow has to be estimated, since it directly influences the removal of thermal and contaminant loads in a room. However, the volume flow rate of the natural ventilation can be affected significantly by the window opening width, the outdoor temperature or the wind speed, which makes it challenging to estimate it accurately by a mathematical model.

Anton Maas (1995) investigated the air change rate in a test building under different meteorological conditions by a tracer gas measurement technique. A calculation model was established as a function of wind speed and temperature difference between indoor and outdoor air, aiming at estimating the volume flow rate of natural ventilation through window opening with different opening widths. The wind inflow direction was not taken into account in this function.

A wind-tunnel analysis of Golubić et al. (2020) indicated that the wind inflow direction could affect the air change rate in generic buildings by the single-sided ventilation significantly. However, a useful model for the single-sided ventilation could not be found due to the asymmetry of the system behavior.

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Monika Hall (2004) studied several models and developed a new one, which was improved by computational fluid dynamics (CFD) simulations. In this model, the volume flow rate through window tilt opening is merely calculated by the temperature difference. Using this model is limited to low wind speed conditions, cause the natural thermal convection is predominant for the assumptions made.

In this paper, based on the approach by Anton Maas (1995), we developed a new model, which takes into account the wind speed and temperature difference by using air pressure difference across the facade.

The following section 2 describes the experimental setup and procedures, while the section 3 details the data processing by the modelling. The results are presented in section 4. The paper ends with conclusions and an outlook in section 5.

METHODOLOGY

Experimental setup

Experiments were carried out on a facade system test bench in the laboratory building of the E.ON Energy Research Center at RWTH Aachen University. As shown in Figure 1, the test bench is composed of four identical test rooms facing southeast with an azimuth angle of around 126°. Each room is 5.1 m in depth, 4.7 m in width, 2.7 m in clear height, and the volume is around 65 m³. The test facade of each room can be replaced flexibly for a comparison between different systems under same external conditions. On the south side of the roof is a weather station, which is about 10 m above the ground to record local weather conditions.

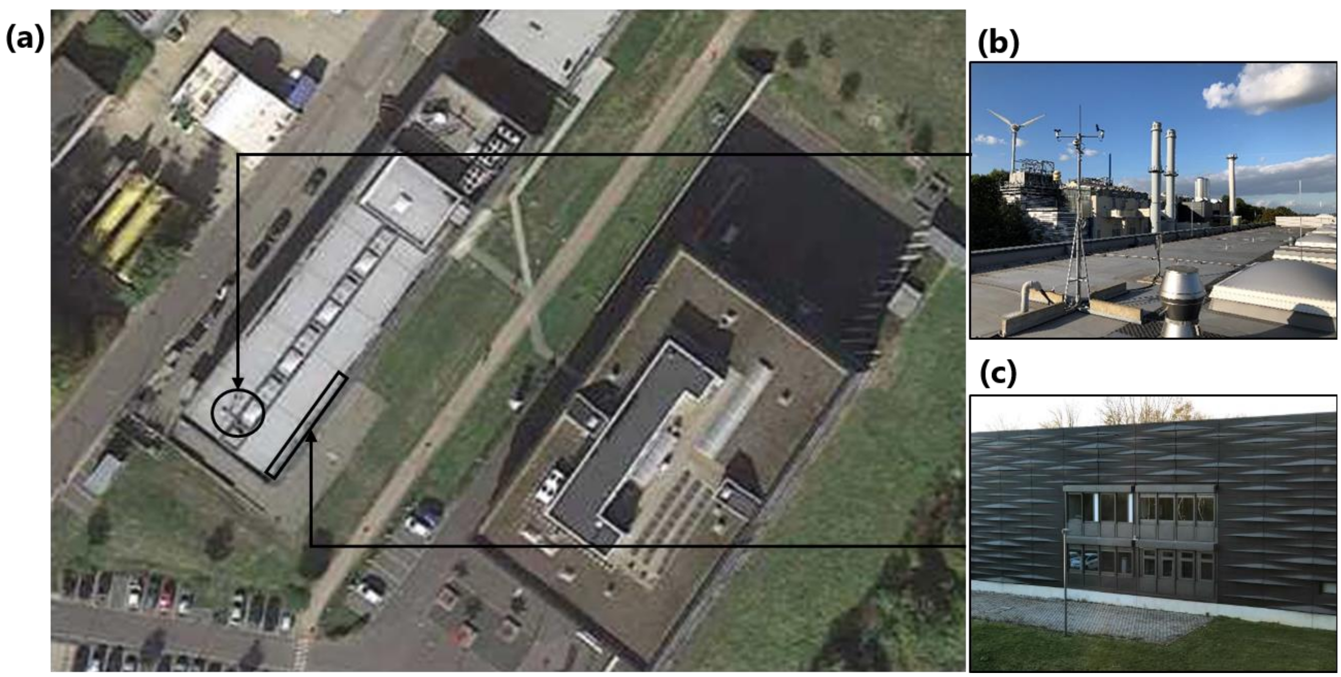

Figure 1. (a) Satellite image of the laboratory building with (b) The location of weather station and (c) Test facades

Both of the two test rooms on the ground floor were utilized in the experiment. From the interior view inside the building, there are four windows as well as four skylights in the left room. Each window is 0.81 m × 1.08 m (width × height). All of the skylights in the left room remained closed in all of the measurements. In the right room, there are four windows, each of them is 1.07 m × 1.38 m. The vertical distance from the windowsill to the floor is 0.8 m. All the windows are equipped with mechatronics fittings, which allow the sash to be opened turn or tilt via a remote control.

In order to measure the pressure difference between the outdoor and indoor air, four thin brass tubes were installed through the parapet wall underneath each window, as shown in Figure 2. The brass pipes were placed about 0.5 m above the floor, and were well sealed on both sides of the parapet. Each brass tube was connected via a polyvinyl chloride (PVC) tube to the positive terminal of a differential pressure sensor. Meanwhile, the negative terminal of the sensor was also connected with another PVC tube, whose end side was placed at the same height as the brass tubes but located inside the test facade in the test room.

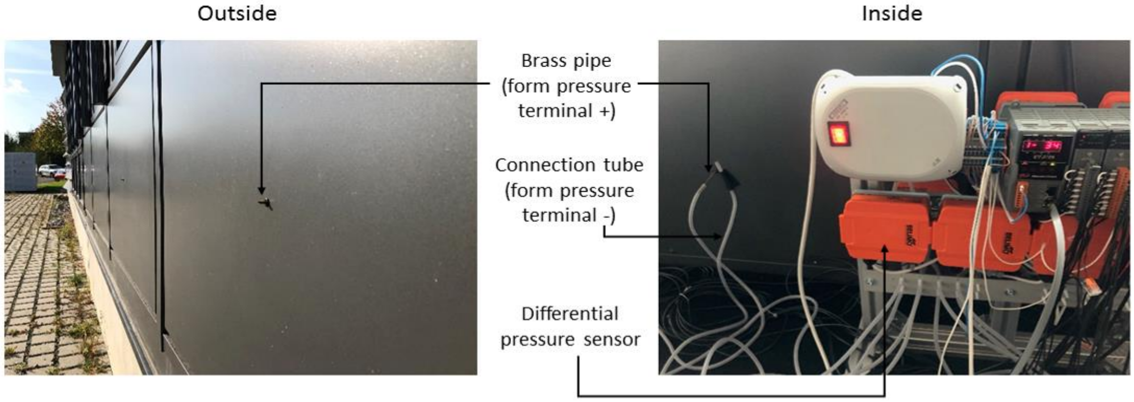

Figure 2. Schematic overview: pressure difference measurements on test facades

To determine the air volume flow by natural ventilation, the tracer gas step-down method was used to figure out the mean air change rate in two test rooms. In this study, CO_2 was utilized as a tracer gas and its concentration was measured by 16 CO_2 sensors in each test room. These sensors were placed on four tripods above the floor with different heights of 0.3 m, 1.0 m, 1.7 m and 2.4 m, respectively, as shown in Figure 3.

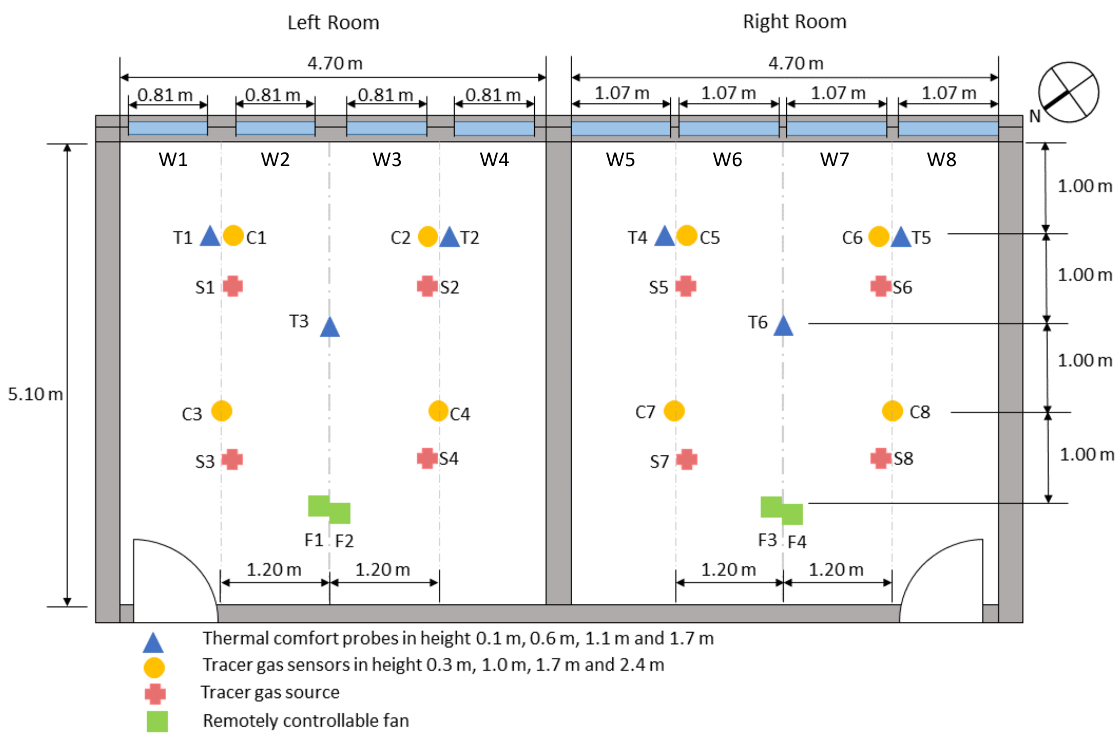

Figure 3. Schematic overview: measurements of indoor air temperatures as well as tracer gas (CO_2) concentrations

In addition to the indoor CO_2 concentration, the outdoor CO_2 concentration was measured at the beginning and at the end of every tracer gas step-down testing. The average value was considered as the constant background CO_2 concentration for each testing.

Furthermore, there were three measuring sets to evaluate the thermal comfort in each test room. Each set consisted of four hot-wire anemometers, which enabled air temperature measurements and omnidirectional air speed measurements at four different heights: 0.1 m, 0.6 m, 1.1 m and 1.7 m above the floor, respectively.

The tracer gas was injected from a central CO_2 gas cylinder through eight sources into the test rooms. Sources were located close to floor areas. With the help of four oscillating fans, the injected CO_2 gas in high concentration could be mixed up with the indoor air in a short time. These fans could be turned on and off remotely.

Experimental procedures

Each entire measurement always comprised of a preparation phase and a measuring phase.

In the preparation phase, test rooms were preconditioned by the HVAC system in the room to 23 °C at first. Once the required temperature had been reached, dampers of the supply air and the extract air in rooms were closed. Then, the CO_2 tracer gas was injected through sources into rooms, and it was mixed up by fans evenly. The injection would be stopped when the CO_2 concentration in the room reached 2000 ppm. In the entire preparation phase, all of the windows remained closed. Fans were turned off at the end of the preparation phase remotely.

At the next stage, windows were opened, therewith the step-down (decay) testing started for the measuring phase. Only one single window in each room (in this case, W3 in the left room and W7 in the right room, as shown in Figure 3) was tilt opened to the determined opening width in accordance to planned test cases, as shown in Table 1. Test cases were defined regarding the opening width to realize equal effective opening area or equal opening width for both rooms. The measurement was stopped when the average CO_2 concentration in the room sank below 600 ppm. Each combination was repeated one to three times under similar weather conditions.

Table 1. List of Test Cases

Case	Opening Width of Left Room	Opening Width of Right Room	Comment
01A	25 mm	20 mm	Equal effective opening area (A_{eff})
02L	30 mm	30 mm	Equal opening width (L)
02A	40 mm	30 mm	Equal effective opening area (A_{eff})
03A	65 mm	50 mm	Equal effective opening area (A_{eff})
04L	70 mm	70 mm	Equal opening width (L)
04A	95 mm	70 mm	Equal effective opening area (A_{eff})
05L	110 mm	110 mm	Equal opening width (L)
05A	150 mm	110 mm	Equal effective opening area (A_{eff})

DATA PROCESSING

Determining the air volume flow

The air volume flow through the window opening cannot be obtained directly by step-down testing during the tracer gas measurement, but it can be determined according to equation (1) based on DIN EN ISO 12569 (DIN 2018).

$$\dot{Q} = \frac{V}{\Delta t} ln \frac{\Delta C(t)}{\Delta C(t+\Delta t)} \tag{1}$$

Where $\dot{Q}$ is the determined volume flow rate in m³/s, V is the room volume in m³, ΔC is the concentration difference of CO_2 tracer gas in volume based (in ppm) between indoors and outdoors, Δt represents the measurement time interval between the initial and the final points in seconds.

Considering the building airtightness, the determined volume flow from equation (1) must be modified, so that the actual air volume flow through the window opening can be obtained for the further modelling. The air tightness regarding the natural infiltration was determined as 50 m³/h for the left room, and 39 m³/h for the right room by the same step-down testing without window opening. Hence, in the subsequent stage of the modelling, the difference between the determined value from equation (1) and the value of natural infiltration was regarded as the measured air volume flow through window opening ($\dot{Q}_{\mathrm{meas}}$).

Determination of the window opening area

While the air volume flow through window opening is related to the opening width and tilt angle, it is necessary to define a method to formulate these influence parameters. In this study, an effective opening area for the single tilt window A_{eff} is defined according to the study of Anton Maas (1995), see equation (2).

$$A_{\mathrm{eff}} = A_{\mathrm{s}} \cdot \Theta \tag{2}$$

Where A_{eff} represents the effective opening area in m², A_{S} is the area of the window sash in m² and Θ is the flow ratio, which depends on the opening position. Parameters A_{S} and Θ are specified by equation (3) and (4).

$$A_{\mathrm{S}} = W \cdot H \tag{3}$$

$$\Theta = \frac{A_{\mathrm{eq}}(\alpha)}{A_{\mathrm{eq}}(\alpha=90°)} \tag{4}$$

Where W is the width of the window sash in m and H is the height of the window sash in m. A_{eq} is the geometric equivalent area of the ventilation section in m², which is defined by equation (5) and (6).

$$A_{\mathrm{eq}}(\alpha) = (A_{\mathrm{S}}^{-2} + A_{\mathrm{C}}^{-2})^{-0.5} \tag{5}$$

$$A_{\mathrm{C}} = 2H\,sin(\frac{\alpha}{2}) \left[H\,cos\left(\frac{\alpha}{2}\right) + W\right] \tag{6}$$

Where A_{C} is the area of the flow cross section (in m²) formed by the window frame and the window sash, which consists of a rectangular surface at the top and two triangular surfaces on both sides. α is the tilt angle of the window opening in °.

Modelling

Overall, 80 measurements were conducted validly in both test rooms during the period from October 2020 to January 2021, resulting 35 valid measurements in the left room as well as 45 valid measurements in the right one. With these two test facades, the results from modelling were also representative for different window dimensions besides different tilt opening widths.

The new model was based on the modelling approach after Anton Maas (1995), see equation (7).

$$\dot{Q} = 3600\,\frac{1}{2} A_{\mathrm{eff}} \sqrt{(C_1 u^2 + C_2 H \Delta\vartheta + C_3)} \tag{7}$$

Where $\dot{Q}$ is the estimated volume flow rate in m³/h, A_{eff} is the effective opening area in m², u is the wind speed in m/s, H is the height of the window sash in m and $\Delta\vartheta$ is the temperature difference between indoor and outdoor air in K. The coefficients C_1, C_2 and C_3 were determined by Maas as 0.0056, 0.0037 m/(s²K) and 0.012 m²/s², respectively.

In equation (8), a new variable y is defined, which presents the square of the normalized volume flow rate derived from equation (7).

$$y = \left(\frac{2\dot{Q}}{3600 A_{\mathrm{eff}}}\right)^2 \tag{8}$$

Consequently, the variable y equals to the sum of all the terms underneath the square root in equation (7), which was defined by Maas as $C_1 u^2 + C_2 H \Delta\vartheta + C_3$. To meet our approach, the variable y was redefined as a linear function depending on the pressure difference, see equation (9).

$$y \stackrel{\text{def}}{=} C_4 \Delta P + C_5 \tag{9}$$

Where ΔP is the mean value of pressure differences (in Pa) between the outdoor and indoor air from each measurement.

The relationship between the squared normalized volume flow rate y and the mean pressure difference ΔP is represented in Figure 4. The minimal mean pressure difference is around -1 Pa, these negative values may be caused by the measurement accuracy of the differential pressure sensor, which is ±1 Pa at range <250 Pa.

The Figure 4 proved that the supposed linear relationship between parameters y and ΔP in equation (9) is plausible. However, several positive outliers occurred with the mean pressure difference from 1 Pa to 3 Pa, when the linear relationship between parameters y and ΔP was taken into account. With an evaluation of all data, all these outliers occurred for effective opening areas A_{eff} greater than 0.15 m². Finally, these outliers were excluded to determine coefficients C_4 and C_5 with a linear regression.

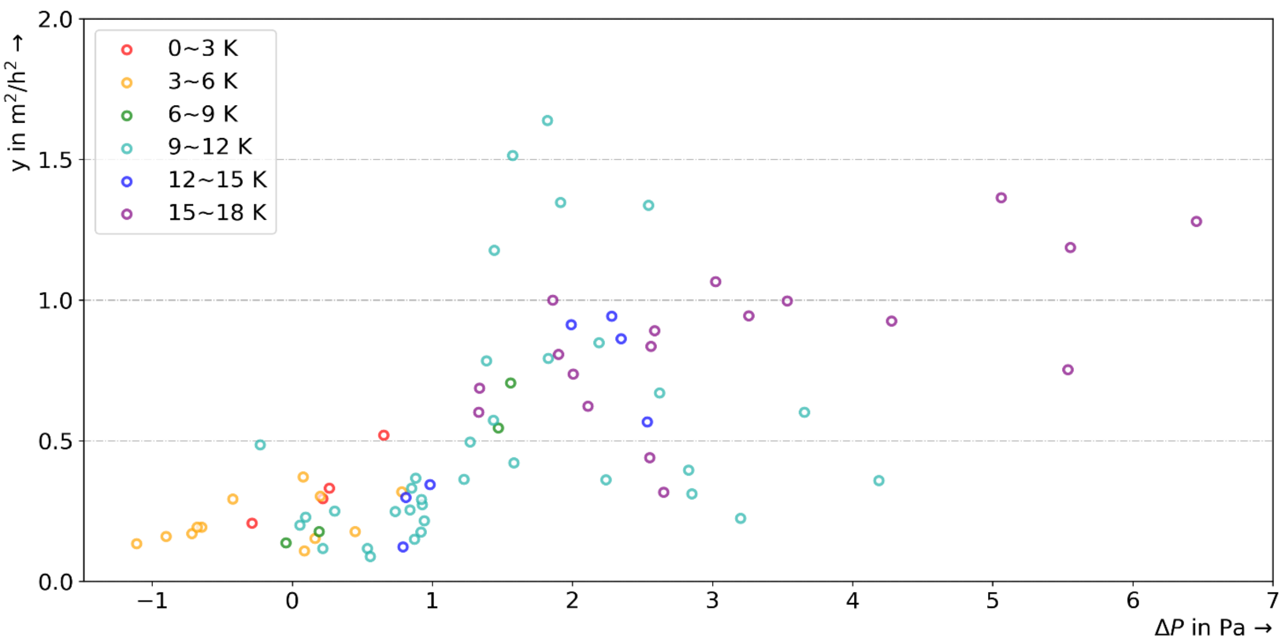

Figure 4. Relationship between the squared normalized volume flow rate and the mean pressure difference (inclusive outliers)

RESULTS

The new model developed in this study is hereby compared with the other two mathematical models from Anton Maas (1995) and Monika Hall (2004), as shown in Figure 5. On the X-axis, the parameter $\dot{Q}_{\mathrm{meas}}$ represents the measured air volume flow at the test facades, which was already modified to consider the natural infiltration of the test rooms. According to the mean measured values of the temperature difference, the wind speed and the pressure difference, three estimated air volume flows were calculated by using the three different models for each single measurement case. These values are represented as the parameter $\dot{Q}_{\text{calc}}$ on the Y-axis. The black solid line in the graph expresses an ideal accordance of the calculated and measured volume flow rates.

The model of Monika Hall (2004) shows a good linearity by using to estimate the air volume flow. However, flow rates are mostly overestimated by this model, with a mean deviation of +60.3 m³/h relating to measured values in this study. These deviations are especially significant in range of large flow rates, and the maximum relative deviation is up to +192%.

Contrarily, using the model of Anton Maas (1995), flow rates are often underestimated, with a mean deviation of -47.2 m³/h relating to measured values in this study. The validity can hereby not be proved, that whether it is plausible to integrate the wind speed as a non-directional parameter by modelling.

The new model in this study shows a good linearity, as well as a high accuracy. The mean deviation relating to measured values is +7.4 m³/h. An overview in terms of the mean deviation as well as the relative deviation between three models is represented in Table 2. All values are evaluated relating to the measured values.

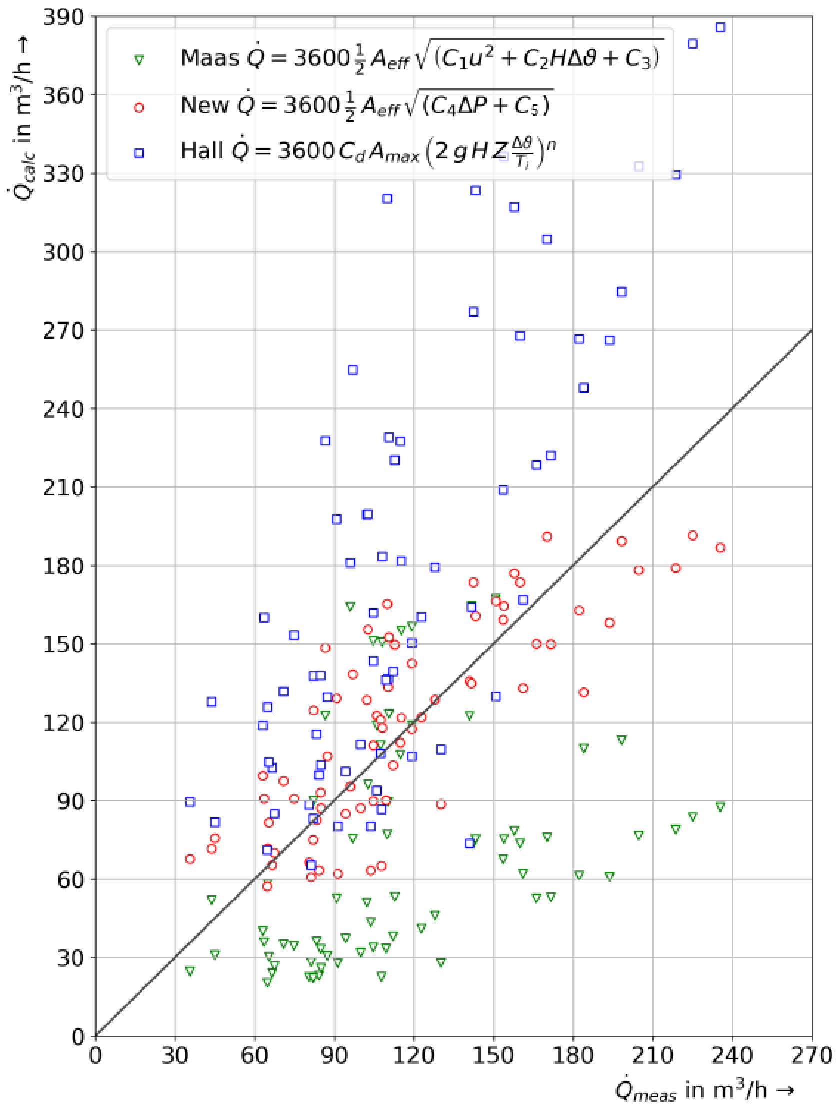

Figure 5. Comparison of the measured and calculated volume flow rates by the existing and new models

Table 2. Comparison of the Models

Model	Mean Deviation	Maximum Relative Deviation
In this study	+7.4 m³/h	From +91% to -39%
Anton Maas	-47.2 m³/h	From +71% to -79%
Monika Hall	+60.3 m³/h	From +192% to -48%

CONCLUSION AND OUTLOOK

The model developed in this study presents that the volume flow rate through window opening can be estimated by the measured pressure difference between the outdoor and indoor air directly. The pressure difference can reflect the superimposed effects of thermal inducted as well as wind driven natural ventilation more reliably, when the effective opening area is less than 0.15 m². When the window is widely open, the pressure difference between indoor and outdoor could be too small to measure precisely, as it could be more easily compensated. As a result, the calculated volume flow rate could also have a large deviation. At present, besides small opening area, this model is also limited to a simple single-sided natural ventilation. More measurements will be conducted in the future to investigate if this model is still suitable for cross ventilation cases, as the door or another window are opened in the same tested room.

This study is a pilot study in respect of a control strategy development for hybrid ventilations, which is with the aim of lowering the energy consumption in office environments. For a simple but high-performanced control strategy in a hybrid ventilation system, the controller should be able to predict the volume flow rate through window opening precisely. Using a mathematical model can minimize the complexity of these predictions, that whether the mechanical ventilation should be activated to complement the insufficient volume flow rates through natural ventilation. Hence, a reliable and generally applicable model is an essential prerequisite for the development of hybrid ventilations.

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NOMENCLATURE

A = area
C = concentration
H = height of the window
L = opening width of the window
P = pressure
$\dot{Q}$ = volume flow rate
t = time
V = volume
W = width of the window
y = squared normalized volume flow rate
α = opening angel
Θ = flow ration
ϑ = temperature

Subscripts

C = window cross section
calc = calculated
eff = effective
eq = equivalent
meas = measured
S = window sash

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