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Window ventilation efficiency in the case of buoyancy ventilation

Jörn von Grabe[a,*], Petr Svoboda[a], Armin Bäumler[b]

[a] TUM University of Technology, Chair for Timber Structures and Building Construction, Arcisstraße 21, D-80333 Munich
[b] Hydro Building Systems GmbH Söflinger Straße 70 D-89077 Ulm, Ulm

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ABSTRACT

This paper presents the results of a number of measurements that were carried out in order to quantify the ventilation efficiency of different types of windows under buoyancy ventilation. The tested window types were double vertical slide window, turn window, tilt window, awning window, horizontal pivot window and vertical pivot window. Each testing of these types comprised different opening rates, so that the performances could be compared across a wide range of window states. Performance criteria were the mass flow, the air change rate and the CO_2 removal rate, each of them normalised to a temperature difference of 1 K. The horizontal pivot window turned out to be the best performing type of window.

1. Introduction

Occupied spaces in buildings require the ventilation with fresh air in order to remove air pollutants. This fresh air supply can either be provided by mechanical ventilation systems or by natural ventilation by the use of windows. Due to a couple of reasons the latter has become more and more relevant in the building sector. One of these reasons is stated by the fact that the operation of windows and the exchange of air between internal and external space does not require the input of electric energy and therefore supports the aim to save energy and to reduce carbon dioxide emissions.

The energy that is needed to ventilate the space is provided by natural forces, i.e. either wind or buoyancy. In both cases a pressure difference between internal and external space is provoked and acts as the driving force. Due to the natural variability of these forces the ventilation by window opening is more difficult to predict than the ventilation provided by mechanical systems. Especially, the wind driven natural ventilation is subject of large and fast fluctuations which are caused by an almost permanently changing wind direction and speed. An exact prediction of the ventilation rate by wind pressure – e.g. for thermal building simulation calculations – requires knowledge about these wind characteristics, which is hardly attainable. On the other side, in order to calculate the buoyancy driven ventilation, only information about the internal and the external temperature difference is necessary. Compared to the wind characteristics, these natural forces show a far slower fluctuation rate and, additionally, are much easier to determine.

Buoyancy driven ventilation is therefore easier to handle in terms of prediction. For simulation purposes, it is also often justifiable to do without consideration of the wind driven ventilation and instead to take only buoyancy into account: During cold seasons, for which the heating demand has to be determined, the buoyancy driven ventilation is permanently high due to the high temperature difference between inside and outside. In such cases, wind may add substantial ventilation rates only in extreme situations. During hot seasons however, the temperature difference between inside and outside is low, resulting in a low buoyancy driven ventilation rate. In the majority of situations, this will exacerbate the overheating situation of the space, so that the calculation will be on the save side. Undeniable, there are situations for which this reasoning is not applicable. Nevertheless, it stays a crucial part for the performance-prediction of naturally-ventilated spaces to predict buoyancy driven ventilation reliably.

In the past, a number of studies of natural ventilation have been conducted. However, these studies have a number of shortcomings. Many studies have been conducted under natural conditions without systematic, experimental separation of the buoyancy and the wind force (e.g. [1,2]). Under these measurement conditions, buoyancy effects cannot be directly analysed. Others have only concentrated on wind induced ventilation in wind tunnel experiments and have thus excluded buoyancy effects (e.g. [3,4]). Some studies have focused on buoyancy driven ventilation by experimentally excluding wind forces. However, these studies included only one specific shape of the ventilation opening, either rectangular (e.g. [5]) or circular (e.g. [6]). Buoyancy driven ventilation for window

* Corresponding author. Tel.: +49 89 289 22047; fax: +49 89 289 23014.
E-mail address: v.Grabe@tum.de (J. von Grabe).

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Nomenclature

a	stratification factor (–)
acr	air change rate (1/h)
$acr(a)$	air change rate, depending on stratification factor (1/h)
c_{ext}	CO_2 concentration, external (–)
c_{int}	CO_2 concentration, internal (–)
$\overline{c_{int}}$	CO_2 mean concentration, internal (–)
$c_{int,outlet}$	CO_2 concentration, internal, at height of outlet (–)
c_{start}	CO_2 concentration at starting conditions
g	acceleration due to gravity (m/s^2)
Δh	height difference between inlet and outlet (m)
$\dot{m}_{measured}(a)$	measured mass flow, depending on stratification factor
$\dot{m}(a)$	mass flow, depending on stratification factor (kg/h)
$\dot{m}'(a)$	mass flow, depending on stratification factor, normalised to 1 K temperature difference (kg/h)
p	pressure (Pa)
$p_{thermal}$	thermal pressure difference (Pa)
$\dot{Q}$	ventilation rate (m^3/h)
R	gas constant, air (J/kg/K)
ρ	density (kg/m^3)
ρ_{ext}	air density, external (kg/m^3)
$\Delta\rho$	density difference (kg/m^3)
T	absolute temperature (K)
t	time (h)
V	volume (m^3)

shapes with more practical relevance – i.e. for example tilt windows and turn windows – have been studied e.g. by [7,8]. Also the buoyancy driven ventilation for double facades has been investigated (e.g. [9]). However, there are a vast number of other window types that are relevant in practice and which so far have not been studied and compared to each other. Due to the fact that natural ventilation not only depends on natural forces but also on a number of window characteristics, different window types show different ventilation efficiencies under otherwise comparable conditions. In order to quantify these differences and thus to overcome the aforementioned shortcomings, a number of measurements were carried out in a controlled environment under exclusion of wind forces. The measurement procedure, the theoretical analysis of the results and the numerical comparison between these different window types will be demonstrated and explained in this paper.

2. Method

The following subchapters describe two relevant points: First the theoretical background that is needed for the analysis of the measurement results and second the characteristics of the test chamber and the general testing procedure.

2.1. Theoretical background

2.1.1. Determination of the air change rate, the mass flow and the CO_2 removal rate

The concentration decay method was used to determine the ventilation rate for each window setting. For this method, a tracer gas is emitted into the test chamber until a defined, uniformly distributed concentration of this tracer gas has been established. At this point – the starting conditions for the measurement – the emission is being stopped and the following decay of the concentration due to ventilation with uncontaminated air is being monitored. For our measurements we have used CO_2 as a tracer gas and started the monitoring at a concentration of about 4000 ppm. However, due to the prevalent CO_2 concentration everywhere in the air, the external air cannot be considered as uncontaminated but rather as contaminated at a significant lower level. This external concentration cannot be neglected for the evaluation of the measurements.

The rate of decay depends on the rate of ventilation. The conversion into ventilation rate is based on a simplified time dependent volume balance for the tracer gas c_{int}, based on an external concentration c_{ext} and a constant ventilation rate $\dot{Q}$. V denotes the volume of the ventilated space. This balance is shown in Eq. (1).

$$V \times dc_{int}/dt = \dot{Q} \times c_{ext} - \dot{Q} \times c_{int} \quad \left[\mathrm{m^3/h}\right] \tag{1}$$

This differential equation can easily be solved and results in the following Eq. (2) for the time dependent, internal concentration:

$$c_{int}(t) = c_{ext} + [c_{start} - c_{ext}] \times e^{-\frac{\dot{Q}}{V}t}\,[-] \tag{2}$$

Solving for $\dot{Q}$ divided by V – which is the definition of the air change rate *acr* – leads to the well-known equation for the conversion arithmetic between ventilation rate and concentration at a specified point in time t:

$$acr = \frac{1}{t} \times \ln\left[\frac{c_{start} - c_{ext}}{c_{int}(t) - c_{ext}}\right] \left[\frac{1}{\mathrm{h}}\right] \tag{3}$$

If the concentration of the tracer gas outside the test chamber equals zero, Eq. (3) turns into Eq. (2) of [10].

This equation is based on the assumption that the internal CO_2 concentration is uniformly distributed in the space. In this case, the amount of exiting CO_2 and therefore the decay rate is independent of the location of the outlet because the concentration is the same everywhere. Mean and local CO_2 concentrations are identical. However, if this is not the case, Eqs. (1)–(3) are leading to wrong results. If the CO_2 concentration is e.g. stratified in such a way that the CO_2 concentration at the height of the outlet of the window is higher than the mean concentration in the space, Eq. (3) results in an air change rate that is higher than in reality. The *ventilation* efficiency of the window in question would therefore be overestimated. One could name the result in such a case CO_2-*removal efficiency* and that expression would account for two simultaneous aspects: The ventilation rate and the CO_2 stratification which is caused by the window opening in question.

Section 3 will show that different window types actually lead to different degrees of stratification, next to the different degrees of ventilation rate. Therefore, it makes sense to introduce a stratification factor a that relates the CO_2 concentration at the height of the outlet to the mean CO_2 concentration in the space $\overline{c_{int}}$:

$$a = \frac{c_{int,outlet}}{\overline{c_{int}}} \tag{4}$$

Using this factor a. Eq. (1) turns into

$$V \times d\overline{c_{int}}/dt = \dot{Q} \times c_{ext} - \dot{Q} \times \overline{c_{int}} \times a \quad \left[\mathrm{m^3/h}\right] \tag{5}$$

solving this equation leads to the following alternative arithmetic for the determination of the air change rate:

$$acr(a) = \frac{1}{a \times t} \times \ln\left[\frac{c_{start} - \frac{c_{ext}}{a}}{\overline{c_{int}}(t) - \frac{c_{ext}}{a}}\right] \tag{6}$$

For a uniform distribution of tracer gas in the space the factor a turns into 1 and thereby Eq. (6) changes into Eq. (3).

To obtain a sufficient approximation of the factor a for each measurement, three internal CO_2 sensors at different heights in the test chamber were used. One was located below the inlet, one was located at around window centre height and the third one was located above the window outlet. These three points were used to approximate the dependency of the CO_2 concentration of the height by a simple linear function. This linear function was then in turn