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Convective Heat Transfer between Rooms in Nordic Passive Houses

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Abstract

In highly-insulated buildings such as passive houses, the space-heating distribution subsystem can be simplified by reducing the number of heat emitters. In this context, the bi-directional flow through open doorways is known to be an efficient process to support the heat distribution between rooms. This process is therefore investigated using field measurements within a Norwegian passive house. The so-called *large opening approximation* proves to model fairly the mass flow rate, but also the convective heat transfer if the thermal stratification is accounted for. Furthermore, the discharge coefficient appears to be independent of the heater type and location in the room.

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Keywords: passive house; simplified heat distribution; open doors; modelling; large opening approximation

1. Introduction

The necessity to drastically reduce the space-heating (SH) needs of residential buildings in Europe have prompted the emergence of building concepts based on a highly-insulated building envelope, such as the passive house (PH) standard [1]. Basically developed for central Europe (e.g. Germany), the PH concept has been extended to Nordic countries. In particular, Norway has elaborated a national definition [2] which is expected to become the minimal performance requirement for new residential buildings in 2015. While the construction of passive envelopes is more challenging, these envelopes also offer opportunities. Given the level of insulation and the use of high-performance windows, the SH distribution system can be simplified because it is theoretically not necessary anymore to place a heat emitter in each room, or in front of windows. A well-known simplified approach is the so-

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called centralized *air heating* but one could also consider a wood stove or a limited number of low-temperature radiators. In practice, there is lack of fundamental knowledge to support the simplification of the SH distribution system in PH. In fact, the limitation of the heat emitters' number to a couple of rooms inevitably leads to temperature differences with the other "non-heated" rooms. In order to reach an acceptable thermal comfort in the entire building, the heat transfer between rooms should be promoted. If the internal doors inside the building are closed, it is rather well accepted than the heat transfer will be dominated by the conductive heat transfer through partition walls and ceilings, followed by the convective heat transfer of the cascade-flow ventilation (i.e. the unidirectional air transfer between rooms equipped with inlets air terminal devices and wet rooms). Nevertheless, there are clear evidences that the resulting heat transfer will not always be enough efficient to reach a uniform temperature inside the building. Different studies have however shown that the opening of internal doors is an efficient way to homogenize temperature in PH [1]. In fact, a large bidirectional flow will be generated through doorways with flow rates that are significantly higher than the nominal airflow rates provided by the balanced mechanical ventilation. In practice, there is a lack a measurement data in PH that confirms this. Furthermore, assessment using building performance simulation most often relies on a *ventilation network model*, e.g. the COMIS software [3]. Airflows through doors are then essentially modelled using a *large opening approximation* which introduces the concept of discharge coefficient, C_d, to tune the model to specific flow physics [4]. Hypotheses behind these models are unfortunately too crude to be directly applied to the present application [5]. Consequently, the present article reports on airflows in open doorway within a Norwegian PH located in Trondheim. Furthermore, it compares measurements with the theory and usual simulations approaches. The final objective of this research is to propose a best practice to design simplified SH distribution systems in PH, especially for cold climates.

2. Theoretical background

In detailed dynamic simulations such as TRNSYS [6], models for airflows are kept deliberately simple in order to limit the computational time and the number of input parameters. In this context, *ventilation network models* idealize the building as a network of nodes and airflows links between them. A node represents either a room or the building surrounding. The scope and limitations of the approach for ventilation prediction is well described by [5] and [7]. It essentially assumes that the air temperature in each zone is uniform and that the air momentum can be neglected (i.e. quiescent air in zones). In ventilation network models, the bidirectional flow through doors is often modeled using a *large opening approximation* [4]. The mass flow is then computed in the following way [8]:

$$m_{12} = C_d \int_0^H \sqrt{2\rho_1(z) f_{12}(z)}\, w(z) dz \,, \tag{1}$$

$$m_{21} = C_d \int_0^H \sqrt{2\rho_2(z) f_{21}(z)}\, w(z) dz \,, \tag{2}$$

where z is the vertical direction, H is the height of the doorway, the subscript 1 refers to the warm zone (here a ground floor), the subscript 2 refers to the cold zone (here an open staircase), m_{12} refers to the mass flow from zone 1 to zone 2, m_{21} refers to the mass flow from zone 2 to zone 1, $f_{12}(z)$ is equal to $p_1(z)-p_2(z)$ if positive or is equal to zero, $f_{21}(z)$ is equal to $p_2(z)-p_1(z)$ if positive or is equal to zero, w(z) is the width of the door. The pressure $p_i(z)$, with subscript i equalling 1 or 2, is evaluated based on the reservoir conditions on both sides of the doorway:

$$p_i(z) = p_i(0) - \int_0^z \rho_i(\tau)\, g\, d\tau \,, \tag{3}$$

where g is the gravity acceleration and $p_i(0)$ the reference pressure of zone i. This last value is computed to get the conservation of mass throughout the ventilation network. As the air temperature field is assumed uniform in most of the building simulation models, $T_{s,i}(z)$ and $\rho_i(z)$ are then not dependent on z which leads to:

$$p_i(z) = p_i(0) - \rho_i\, g\, z \,. \tag{4}$$

In the experimental setup, the mass flows m_{12} and m_{21} can be measured directly using the velocity and temperature measurements along the doorway:

$$m_{12}^{exp} = \int_0^H \rho_1(z)\, u_{1\rightarrow 2}(z)\, dz \,, \tag{5}$$

$$m_{21}^{exp} = \int_0^H \rho_2(z)\, u_{2\rightarrow 1}(z)\, dz \,. \tag{6}$$

Using the measured m_{12} and m_{21}, the discharge coefficient can be obtained by evaluating the right-hand side term of Eqs.(1) and (2) using the density $\rho_i(z)$ and the pressure $p_i(z)$ (in rooms on both sides of the door). This baseline coefficient is here termed $C_{d,m}$. Following the simplification of many building simulation models, the right-hand side term can also be evaluated assuming isothermal rooms on both sides of the door (i.e. considering Eq. (4)). It leads to another value of discharge coefficient, here termed $C_{d,ms}$.

For the present application, the physics of interest is the temperature difference between rooms. Consequently, the mass flows m_{12} and m_{21} are not the most important physical phenomena to capture properly. To get the correct energy balance of the different rooms in the building, it is rather imperative to evaluate the amount of energy exchanged between rooms. Using the large opening approximation along with Eqs.(1) and (2), it is evaluated in the following way:

$$h_{12} = C_d \int_0^H \sqrt{2\rho_1(z) f_{12}(z)} C_p T_{s,1}(z) w(z) dz, \tag{7}$$

$$h_{21} = C_d \int_0^H \sqrt{2\rho_2(z) f_{21}(z)} C_p T_{s,2} w(z) dz, \tag{8}$$

where h_{12} is the convective heat transfer from zone 1 to zone 2, h_{21} is the convective heat transfer from zone 2 to zone 1, $T_{s,1}(z)$ is the air temperature of the warm zone, $T_{s,2}(z)$ is the air temperature of the cold zone.

In the experimental setup, heat flows h_{12} and h_{21} can be extracted directly from the velocity and temperature measurements. As assumed in the large opening approximation, one can also assume that the fluid going from zone 1 to zone 2 has the temperature of the zone 1 (and vice versa):

$$h_{12}^{\exp} = \int_0^H \rho_1(z)\, u_{1\to 2}(z)\, C_p\, T_{s,1}(z)\, dz, \tag{9}$$

$$h_{21}^{\exp} = \int_0^H \rho_2(z)\, u_{2\to 1}(z)\, C_p\, T_{s,2}(z)\, dz. \tag{10}$$

In the IEA EBC Annex 20 [9], it has been shown that neglecting the thermal stratification (i.e. $T_{s,1}(z)$ and $T_{s,2}(z)$ independent of z) in Eqs. (7) and (8) leads to a significant error when estimating h_{12} and h_{21}. To investigate this, another discharge coefficient, $C_{d,es}$, can be obtained by equaling Eqs. (9) and (10) with Eqs. (7) and (8) where rooms are assumed isothermal in the last two equations. In conclusion, based on experiments, different discharge coefficients have been established as summarized in Table 1: either with the focus to mimic properly the mass flow or the convective heat flow, or with and without temperature stratification in rooms on both side of door.

Table 1. Different calibrations of the discharge coefficient C_d.

Name	Target	Room temperature field
$C_{d,m}$	Mass flow	Stratification
$C_{d,ms}$	Mass flow	Isothermal
$C_{d,es}$	Heat flow	Isothermal

3. Experimental setup

A PH in Trondheim has been monitored during the months of March and April 2014. This a terraced house of the *MiljøGranåsen* project, which is currently the largest PH construction project in the Nordic countries. When fully completed, it will consist of 430 dwelling units with a total heated area of 34000 m². MiljøGranåsen is developed by Heimdal Bolig and is also part of the EBLE, Concerto and Eco-city research projects. During the measurement campaign, the PH was unoccupied and not furnished. It consists of three storeys as shown in Fig. 1. Basically, the building has a large open staircase which would have potentially made the velocity measurement difficult (i.e. velocity magnitude < 0.1 m/s). It was decided to screen the staircase with thermally insulated plates to leave only a single opening of 0.9 m x 2.35 m that should represent a door. The building is constructed in wood except for the basement that is in concrete. The reference heated area is 142.5 m². The U-value of the external walls is 0.15 W/m².K while the triple-glazed windows have an overall U-value between 0.7 and 0.85 W/m².K. The roof has a thermal transmittance of 0.06 W/m².K and the equivalent U-value of the basement floor is 0.1 W/m².K while the basement walls have an equivalent U-value of 0.19 W/m².K. The balanced mechanical ventilation has an air change rate (ach) of 0.52 while the heat recovery (HR) unit has a rated temperature efficiency of 88% (EN 308). Given the ventilation layout, a net mass flow of 33 m^3/h is moving from the first to the ground floor. The SH needs are evaluated at 17.1 kWh/m².year according to Norwegian standards [2, 10].

Three different SH emission subsystems were successively applied in the ground floor while other rooms were left unheated: a convector, an airtight panel heater as well as an electric stove. The idea is to compare emission subsystems with different ratio of heat emitted by convection to heat emitted by radiation. The convector is obviously dominated by convection and can be run with or without fan, the panel heater and electric stoves are dominated by radiation. The 15 kW electric stove is an experimental device that has been created to simulate the thermal environment induced by wood stoves in PH. For the case of the convector and panel heater, the flow through the doorway has been measured for different locations of the heat emitter: in the living room (position n°1), in the kitchen (position n°2) and 1m in front of the door (position n°3). In this way, it is possible to check if this position should be taken into account in modelling or if the mixing is enough strong within the ground floor to be able to neglect this heat emitter position.

Fig. 1. *Picture of the building block: sketches of the basement (left), ground floor (centre) and first floor (right).*

Fig. 2. *Pictures of the convector, panel heaters and the electric stove, respectively.*

The house has then been instrumented to measure the bidirectional flow through the door between the ground floor and the staircase, as well as to characterize the thermal environment on both sides of this doorway. To limit the article length, the measurement equipments and the measured physical quantities are summarized in Table 2.

Table 2. List of measurement probes.

Type	*Number*	*Location*	*Precision*	*Sampling frequency*	*Measure physical quantity*
PT-100	5	Ground floor	±0.1°C	1 min	T_s, stratification
	5	Staircase	±0.1°C	1 min	T_s, stratification
	7	Walls	±1°C	1 min	Temperature surface wall
Thermocouples Type T	10	Doorway (middle)	±1% ±0.5°C	1 min 1 min	T_s, profile
Anemometer TSI 8475	10	Doorway (middle)	±3% ±0.005 m/s	1 min 1 min	Air velocity profile
Temperature logger iButton Maxim Integrated DS1922L	11	One in each room	±0.06°C	15 min	T_s, one by room
	1	Outdoor	±0.06°C	15 min	T_s, sheltered
	3	Air Handling Unit	±0.06°C	15 min	T_s, fresh air

The measurement procedure has been established to account for the slow reaction time of the PH (i.e. the relatively long characteristic timescale of the envelope). Furthermore, the boundary conditions of the building, here the outdoor temperature and the solar irradiation, continuously change with timescales shorter than the envelope dynamics. For a given emission subsystem and location, it is thus complex and time consuming to wait for steady-state thermal conditions of the entire building envelope. On the contrary, one expects the airflow to reach a steady-state regime in a much faster way than the walls. Consequently, for each emission subsystem, the heater was placed in position n°1 in the ground floor in the evening and ran overnight. The flow was measured during 1h in the morning for this emitter position. Then, the emitter was displaced to position n°2. The flow was then measured during 1h after having waited 1h for the airflow to reach a new steady-state. A same procedure is then following for the position n°3: 1h to reach a new steady-state followed by 1h measurement. Even though simplified, this procedure is a trade-off between *pseudo* steady-state airflow conditions and limited measurement time. In the case of the electric stove, the procedure was taken different because the stove emitted power cannot be controlled by the room temperature (i.e. no thermostat). Only the position n°1 was tested for constant emitted powers of 2.4 and 6 kW applied during 90 min. The regime is unsteady as those powers are well larger than the envelope thermal losses: the living room temperature field indeed increases progressively during a stove cycle. In this case, reported physical quantities are time-averaged over the 90 min cycle.

4. Results

Before proceeding any further, it worth repeating that, in field measurements, the boundary conditions of the building are continually changing while these boundary conditions could be rigorously controlled in a laboratory test chamber. Even for a same emission subsystem, these conditions may vary significantly between the different heater positions. Consequently, variables such as velocities, temperatures and their gradients should not be compared directly between test cases. Only the comparison of the C_d is here meaningful. For all test cases, the velocity profiles obtained by the large opening approximation were matching measurements pretty well. For instance, it is shown in Fig. 3 for the convector without fan and the panel heater at position n°1. Temperature profiles can also be compared: the temperature in the doorway, $T_s(z)$, compared to the room temperature on both sides of the door, $T_{s,1}(z)$ and $T_{s,2}(z)$. Measurements show that $T_s(z)$ is well bounded by $T_{s,1}(z)$ and $T_{s,1}(z)$. Nevertheless, $T_s(z)$ is not always equal to $T_{s,1}(z)$ when the flow leaves the room and not always equal to $T_{s,2}(z)$ when the flow enters the room. It is difficult to give a physical interpretation of this given the limited accuracy of the temperature measurement in the doorway (i.e. thermocouples of type T with a precision of ±0.5°C).

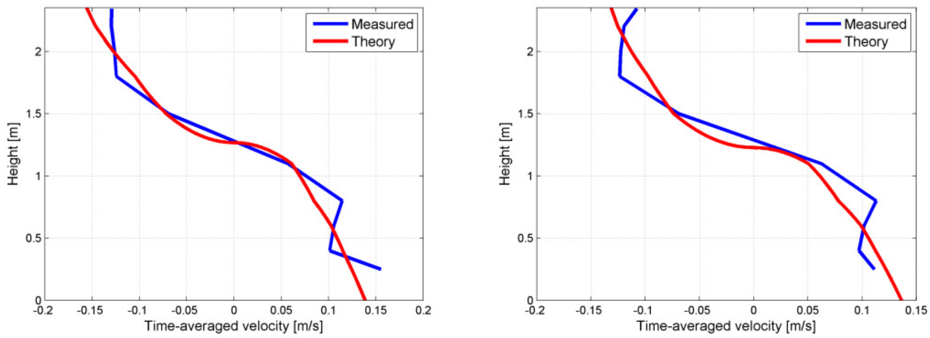

Figure 3: Time-averaged velocity profile for the convector without fan (left) and the panel heater (right) placed at position n°1.

Measurements for the different test cases are reported in Table 3. Firstly, as expected, the flow rates are one order of magnitude higher than the nominal mechanical ventilation flow rates (here 33 m^3/h). Secondly, the discharge coefficients based on the mass flow, $C_{d,m}$ and $C_{d,ms}$, are rather similar which proves that neglecting the temperature stratification on both sides of the door does not alter significantly the computed mass flow rate [9]. Furthermore, these discharge coefficients are also not strongly dependent on the heat emitter type and location. In all cases, the horizontal temperature gradient in the ground floor is limited (data not reported here). This translates that an effective temperature mixing takes place in the ground floor which is not strongly affected by the bidirectional flow through the open door. Considering the discharge coefficient based on the convective heat transfer and computed assuming isothermal rooms, $C_{d,es}$, one can notice that values are higher compared to coefficient calibrated on the mass flow. It means that neglecting the temperature stratification leads to a significant error when evaluating the convective heat transfer. In other words, the computed heat transfer will be underestimated if the discharge coefficient is based on the mass flow and the building simulation neglects the temperature stratification. This conclusion was already reported in the IEA EBC Annex 20 [9].

Table 3. Temperatures and discharge coefficients for the different test cases ($\Delta T_{1,v}$ and $\Delta T_{2,v}$ stands for the time-averaged temperature gradient of the ground floor and the staircase, respectively, while $\Delta T_{1\text{-}2}$ is the time- and space-averaged temperature difference between the two rooms).

Type	*Position [n°]*	*Flow rate [kg/h]*	$\Delta T_{1,v}$ *[°C/m]*	$\Delta T_{2,v}$ *[°C/m]*	$\Delta T_{1\text{-}2}$ *[°C]*	$C_{d,m}$ *[-]*	$C_{d,ms}$ *[-]*	$C_{d,es}$ *[-]*
Panel Heater	1	398	0.39	0.18	1.11	0.41	0.43	0.60
	2	403	0.70	0.19	1.09	0.45	0.45	0.76
	3	385	0.44	0.31	1.40	0.35	0.35	0.49
Convector (fan off)	1	439	1.26	0.34	1.85	0.38	0.36	0.62
	2	536	1.46	0.33	1.24	0.41	0.39	0.60
	3	574	0.61	0.25	3.00	0.37	0.36	0.42
Convector (fan on)	1	487	0.78	0.39	1.72	0.41	0.41	0.68
	2	494	0.84	0.29	1.94	0.43	0.41	0.63
	3	539	0.63	0.38	1.94	0.47	0.43	0.60
Stove 6kW	1	442	2.57	0.54	1.90	0.35	0.32	0.58
Stove 2.4kW	1	425	1.36	0.35	2.94	0.38	0.36	0.61

5. Conclusions and discussion

The bidirectional flow in an open doorway has been measured for a PH in Trondheim. Different SH emission subsystems have been applied in the living room while the remainder of the building was left unheated. Furthermore, different locations of the heat emitter in the living room have been compared. Measurements confirm that the mass flow rate through the door is one order of magnitude higher than the nominal hygienic ventilation flow rates. They also confirm that the discharge coefficient lies within the usual range of [0.4;0.8] reported in the literature [4, 11]. It appears independent on the emitter type and location within the room. As reported in the IEA EBC Annex 20 [9], neglecting the temperature stratification (as assumed in most of the building simulation models) does not lead to a significant error on the mass flow rate. On the contrary, it leads to a large error on the predicted convective heat transfer through the door. As guideline for the simplification of SH systems in PH, the present study suggests that the *large opening approximation* leads to a fair representation of the flow through doorways. Nevertheless, it is recommended to perform a sensitivity analysis on the discharge coefficient to investigate the robustness of the SH concept (taking for example values between 0.4 and 0.8).

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