Analysis of convective heat transfer coefficient correlations for ventilative cooling based on reduced-scale measurements

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ABSTRACT

Ventilative cooling can be used as a passive cooling measure to reduce the cooling energy demand of buildings. It can be used during the day, directly removing excessive heat gains, or during the night (i.e. night flush), in which cold outdoor air flows through the building and cools down the indoor air volume and subsequently the thermal mass of the building. Night flushing reduces the indoor air temperatures at the beginning of the next day and the cooling demand over the day. To assess the impact of ventilative cooling on the temperatures in a building and the resulting cooling energy demand, building energy simulations (BES) can be performed. An important parameter to set in BES is the convective heat transfer coefficient (CHTC) (or CHTC correlation) for the interior surfaces to calculate convective heat transfer from these surfaces to the indoor environment and vice versa. The majority of the available CHTC correlations for internal surfaces in BES are based on natural convection, with a temperature difference as the driving force for convection. However, in case of night flush the airflow rates through the building can be quite large and mixed convection can occur due to the possible presence of relatively high indoor air velocities. A solution to this problem could be the use of convective heat transfer correlations for forced convection, or correlations for external surfaces subjected to the atmospheric boundary layer (ABL) wind flow, which generally calculate the convective heat transfer based on a reference wind speed at a certain location.
This paper presents reduced-scale experiments (ABL wind tunnel measurements) of velocities, turbulence levels, air temperatures, surface temperatures and convective heat fluxes in a generic cubical cross-ventilated enclosure. One of the walls of the enclosure is heated and has a higher temperature than the ambient air, resulting in convective heat exchange between this surface and the air inside the enclosure. The experimental results are used to calculate the values of CHTC for this mixed convection case, in which the contribution of forced convection dominates, as indicated by the calculated Richardson number. The measurement results are subsequently compared with both CHTC correlations for natural and forced convection. The results indicate that the average CHTC values from the experiments generally show a fair to good agreement with the values obtained using CHTC correlations for forced convection, while the agreement with the CHTC values from the CHTC correlations based on natural convection is, as expected, much worse (72-91% difference). This finding is in line with earlier publications in which the use of CHTC correlations for ventilative cooling assessment in BES is discussed. A proper definition of the CHTC correlations for ventilative cooling applications is required to correctly estimate the reduction in cooling energy demand and is part of a larger ongoing research effort.

KEYWORDS

Convective heat transfer coefficients, mixed convection, wind tunnel experiments, ventilative cooling

1 INTRODUCTION

An energy-efficient method to reduce the cooling demand in buildings is the use of ventilative cooling (e.g. Carrilho da Graça et al. 2002, Geros et al. 2005, Wu et al. 2006, Artmann et al. 2007, Santamouris et al. 2010). Ventilative cooling refers to the use of natural ventilation through openable elements in the building envelope to remove excessive heat from a building, and can be used either during the day (directly removing excessive heat gains) or during the night time (night flushing) when outdoor air with a lower temperature than the indoor air flows through the building and cools down the indoor air volume and subsequently the thermal mass of the building. An analysis of the effect of this passive measure with respect to the reduction of cooling energy demand can be made using building energy simulations (BES). The BES results depend heavily on the input provided by the user; more realistic, detailed and high quality input will increase the reliability and quality of the BES results. One of the input parameters that can influence the results of the simulations conducted is the convective heat transfer coefficient (CHTC) for the interior surfaces (e.g. Goethals et al. 2011, Leenknegt et al. 2013). Normally, CHTC correlations are used to predict convective heat transfer between these interior surfaces and the air in a room, but often these correlations are based on natural convection; i.e. the driving force is a temperature difference. However, in ventilative cooling applications relatively high indoor velocities due to higher volume flow rates through the building can be present, implying that the CHTC correlations based on natural convection as commonly used are not, or less, suitable and one must resort to CHTC correlations for forced or mixed convection. The disadvantage of the CHTC correlations for forced convection is that only one CHTC value is obtained for an entire wall/floor/ceiling. In addition, depending on the conditions either natural or mixed convection can be present, both of which would require other CHTC correlations.

This paper presents the results of an experimental study on convective heat transfer in a cross-ventilated generic building model. The experiments included measurements of velocities, turbulence levels, air temperatures, surface temperatures and heat fluxes and were conducted in the open-circuit wind tunnel at the University of Southampton. The objective is to generate an experimental data set which can be used for CFD model validation. The validated CFD model can subsequently be used to assess CHTC distributions in ventilative cooling cases in more detail. In addition, the experimental data can be used to analyze the suitability of interior CHTC correlations as commonly used in BES. Section 2 presents the experimental setup as used in the wind tunnel. Section 3 contains the results of the experimental campaign and a comparison between the experimentally obtained values of CHTC and the values based on existing CHTC correlations. Sections 4 and 5 present the discussion and conclusions of this paper.

2 EXPERIMENTAL SETUP

The reduced-scale experiments were performed in the open-circuit wind tunnel at the University of Southampton (UK), which has dimensions 0.9 × 0.6 × 4.5 m^3 (W × H × L). A neutral atmospheric boundary layer (ABL) was created using 427 mm high spires in combination with three different types of roughness elements (32, 16 and 7 mm high), and a carpet (see Taddei et al. 2016). The resulting mean streamwise velocity profile matched the logarithmic equation:

$$U = \frac{u^*_{ABL}}{\kappa} \cdot ln \frac{y}{y_0} \qquad (1)$$

with κ the von Karman constant (0.42), u^*_{ABL} (= 0.195 m/s) the ABL friction velocity and y_0 (= 0.0024 m) the aerodynamics roughness length. The reference velocity (U_{ref}) was equal to 1.9 m/s at building height H (= 0.15 m). The streamwise turbulence intensity I_u at the location of the building was about 10%. The air temperature of the approach-flow during the measurements was 25.5 °C (= T_{ref}). The building Reynolds number, defined as Re = $U_{ref}H/\nu$, with $\nu = 1.56 \times 10^{-5}$ m²/s the kinematic viscosity of air at an air temperature of 25.5°C, was equal to 19,000. Finally, the experiments reported were carried out for one wind direction (perpendicular to the facade with the openings).

The model of the generic building (scale 1:50) used in the experiments was a single-zone cubic building (0.15 × 0.15 × 0.15 m^3, Fig. 1a,b), with one opening in both the windward and leeward facade and a heated wall (left hand side when looking in streamwise direction; Fig. 1c). The model was made of polymethyl methacrylate (PMMA) sheets (0.01 m thick) and the window openings are 0.04 × 0.035 m^2 ($W_O \times H_O$), resulting in a facade porosity of about 6%. The wall opposite to the heated wall was equipped with a 0.75 mm thick clear polypropylene sheet to allow thermal camera measurements of the heated wall. A detailed schematic of the heated wall (brass plate) including materials and thicknesses for all layers is shown in Figure 1d. The Richardson number (Ri = Gr/Re^2) in these experiments was around 0.03, indicating that forced convection is dominant over natural convection.

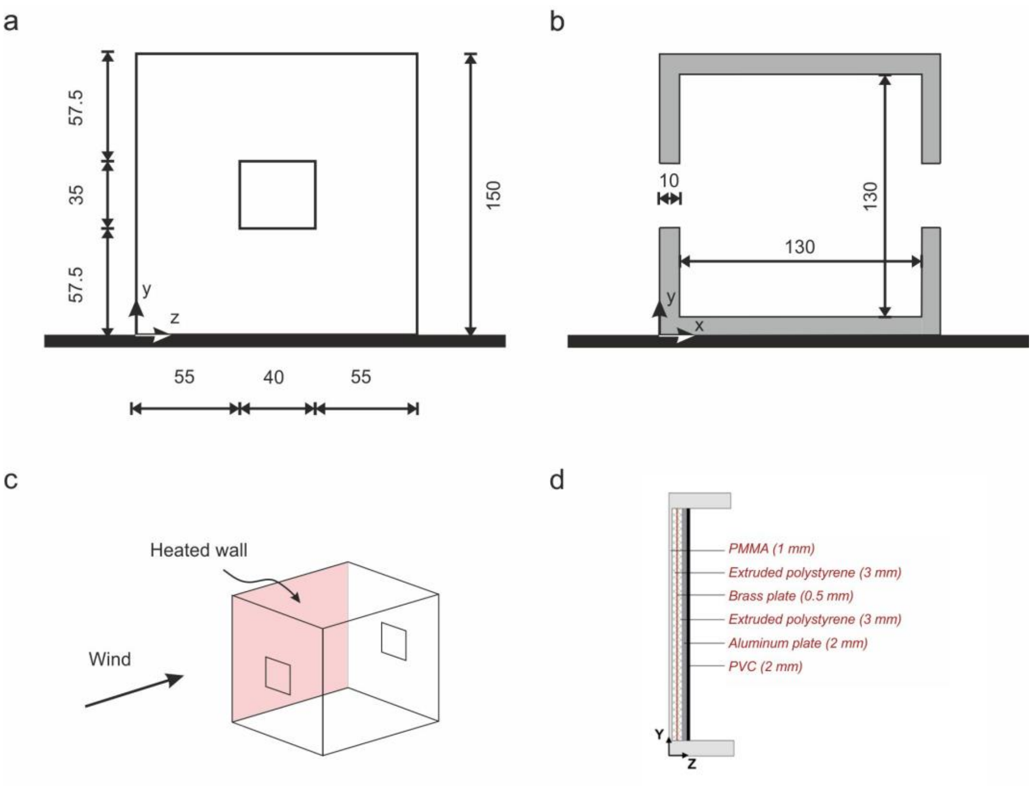

Figure 1: Geometry of the building model used in the wind-tunnel experiments (6% facade porosity). (a) Facade with window opening. (b) Vertical cross section along flow direction. All dimensions are in mm. (c) Isometric view of the building model indicating the position of the heated wall. (d) Vertical cross section of the heated wall with wall composition.

Velocities in the vertical centerplane (parallel to flow direction) were measured using a 2D particle image velocimetry (PIV) system. Two double pulse 200-15 PIV Nd:YAG lasers were used to create a laser sheet. The flow is seeded using a solution of demineralized water and glycol, and a fog machine. A charged coupled device camera (CCD) was used to capture images, and was located normal to the laser sheet. A set of 500 uncorrelated images was taken at a rate of 0.7 Hz, resulting in a total measuring time of 12 minutes. The PIV results were processed using 50% overlap and an interrogation window size of 32 × 32 pixels. The uncertainty of the measurement results is around 1-5.5%.

Measurements of surface temperatures at the heated wall were taken using a mid-range infrared camera (FLIR SC7000). NTC U-type sensors with a diameter of 2.4 mm and precision of 0.05 °C were used to measure the indoor air temperatures at 27 locations (9 along each of the three horizontal lines) (Fig. 2a). The air temperatures were measured for 120 s to obtain stationary results. Six gSKIN-Xp 26 9C heat flux sensors (greenTeg) with a size of 10 × 10 mm^2 measured the surface heat fluxes on the heated wall (see Fig. 2b) (precision of ± 3%). The measured heat fluxes are time-averaged over a period of 120 s to obtain stationary values.

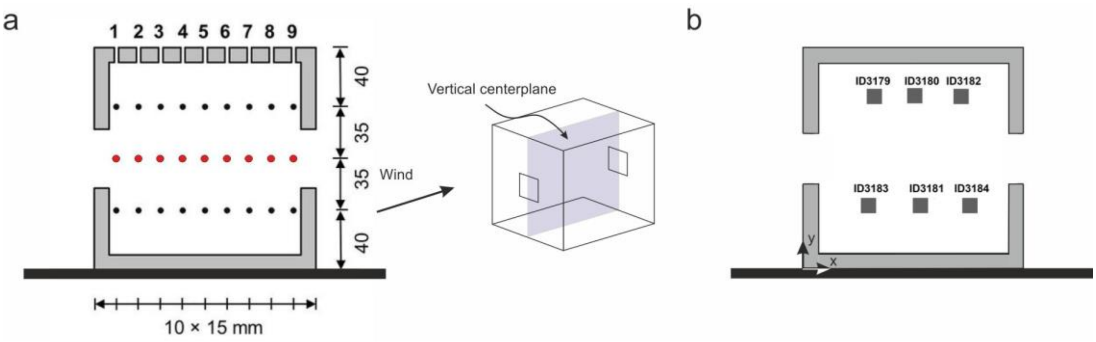

Figure 2: (a) Positions for indoor air temperature measurements in the vertical centerplane (z/H = 0.5). Only the results at mid height (red dots) are included in this paper for the sake of brevity. The probes for the air temperature measurements were inserted through closable openings in the ceiling of the building. All dimensions are in mm. (b) Positions of heat flux sensors at interior surface of heated wall.

3 RESULTS

Figure 3 provides the dimensionless time-averaged streamwise velocity (U/U_{ref}) along three vertical lines (x/H = 0.3, x/H = 0.5, x/H = 0.7) in the vertical centerplane (z/H = 0.5). A clear incoming jet is visible, with velocities higher than U/U_{ref} = 0.6 in the center of the jet (around y/H = 0.4; Fig. 3a). The jet is directed downwards and the maximum velocity decreases in downstream direction (U/U_{ref} < 0.4 at x/H = 0.7; Fig. 3c).

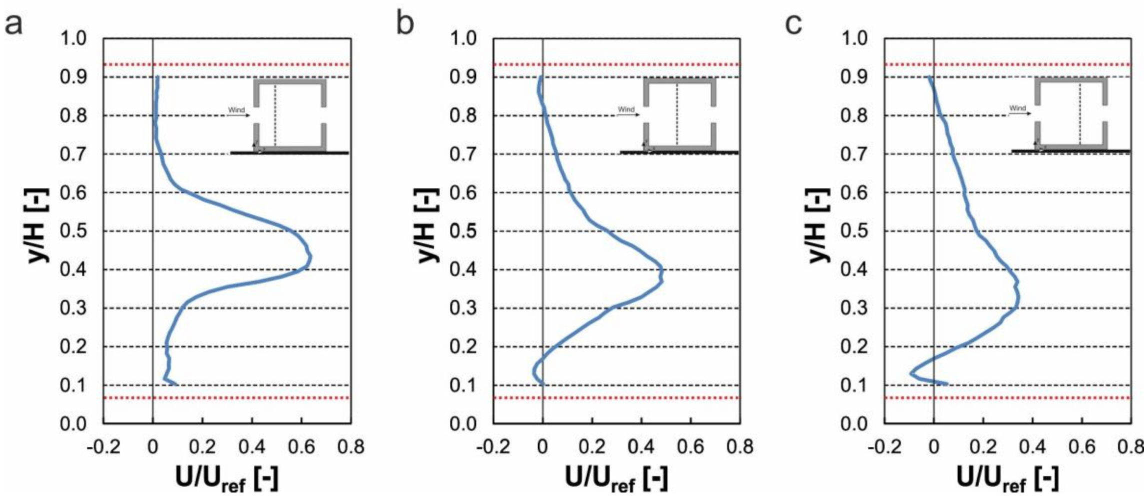

Figure 3: Dimensionless time-averaged vertical profiles of streamwise velocity component, U/U_{ref}, along three lines in the vertical centerplane (z/H = 0.5): (a) x/H = 0.3, (b) x/H = 0.5, (c) x/H = 0.7. H = 0.15 m, U_{ref} =1.9 m/s. The red dotted lines indicate the location of the floor and ceiling.

Figure 4 shows the measured dimensionless time-averaged air temperatures ($(T-T_{ref})/T_{ref}$) along a horizontal line in the vertical centerplane (y/H = 0.5, z/H = 0.5). Air temperatures are generally lowest behind the window opening, i.e. at x/H = 0.2 until x/H = 0.4, where the incoming jet provides cool air to the enclosure. Higher air temperatures are present further downstream (x/H > 0.5); the downward direction of the jet mainly provides cool air in the lower part of the enclosure (y/H < 0.5) from this point onwards (not shown here for the sake of brevity).

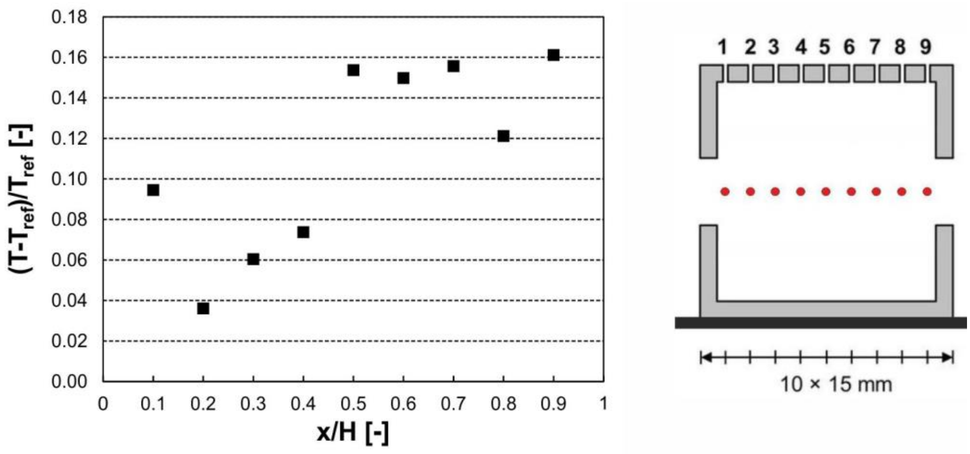

Figure 4: Time-averaged air temperatures ($(T-T_{ref}/)T_{ref}$), measured along the horizontal line at mid height of the enclosure (y/H = 0.5) in the vertical centerplane (z/H = 0.5), T_{ref} = 25.5°C. The locations are indicated on the right-hand side with red dots.

Figure 5 presents the measured heat fluxes at the six locations on the heated wall. The highest heat fluxes (500 W/m^2 < Q < 570 W/m^2) are measured at the three locations in the lower part of the enclosure, i.e. by sensor ID3181, ID3183, ID3184. These higher values can be explained by the particular flow pattern, as described above, with the downward directed jet from the windward window opening. The maximum difference between two locations (ID 3179 vs. ID3184) is 46%.

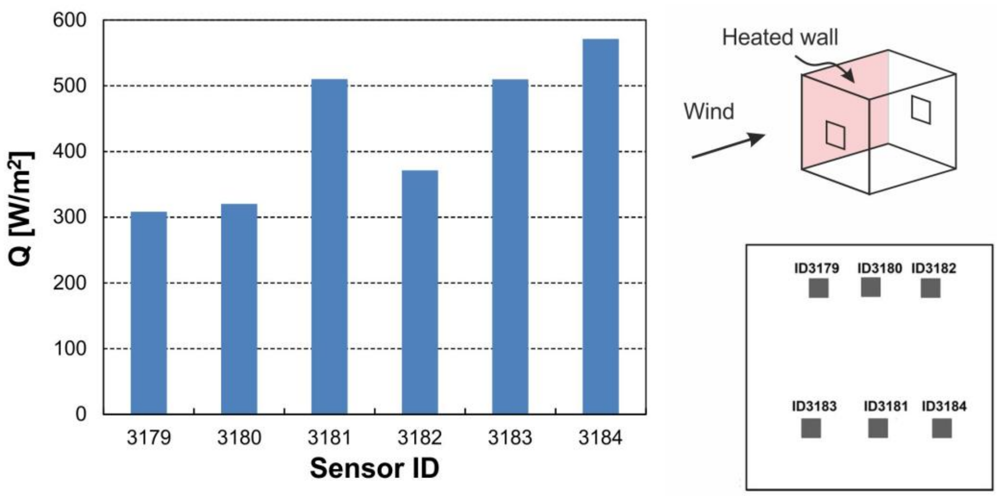

Figure 5: Measured surface heat flux (W/m^2) at the six different locations at the interior surface of the heated wall. The locations are indicated on the right-hand side.

Figure 6 presents a comparison of the CHTC values obtained from the experiments presented in this paper with CHTC values based on some of the available CHTC correlations for BES

(Table 1). The CHTC correlation for vertical surfaces which is provided by TRNSYS (Klein et al. 2010) is defined based on the temperature difference between the surface and the surrounding air, while the correlation from Khalifa (1989) uses the average room temperature in their experiments as reference temperature to calculate CHTC. Spitler et al. (1991) performed experiments to develop a CHTC correlation for situations with large volume flow rates, i.e. forced convection, as present in ventilative cooling cases. Their correlations use the velocity of the supply jet to determine CHTC and is thus not location dependent. Finally, Loveday and Taki (1996) developed a CHTC correlation for the outdoor surfaces of facades of a building based on full-scale experiments. Their correlation based on the combined measurement data for both the windward and leeward facade (see Table 1) is used in the current comparison with the data from the wind tunnel experiments described above.

Table 1: Convective heat transfer coefficient (h_c) correlations.

Source	**Correlation***	**Remarks**	**Type#**
TRNSYS (Klein et al. 2010)	$h_c = 1.5\left(T_s - T_{air}\right)^{0.25}$	Temperature of surrounding air	N
Khalifa (1989)	$h_c = 1.98\left\|\Delta T\right\|^{0.32}$	Average room temperature	N
Spitler et al. (1991)	$h_c = 1.6 + 92.7\left(\frac{\dot{m}U_0}{\rho g V_{room}}\right)^{0.5}$	Side wall inlet. High flow rates. J is outside limits (> 0.011)	F
Loveday and Taki (1996)	$h_c = 16.21V_{loc}^{0.452}$	Average for windward and leeward facade	F

*T_s and T_{air} the surface and air temperature, respectively, $\dot{m}$ the mass flow rate, U_0 the supply velocity, ρ the density, g the gravitational acceleration constant, V_{room} the volume of the enclosure, V_{loc} the local velocity. #: N and F indicate whether correlation is developed for natural or forced convection, respectively.

Figure 6 indicates the mismatch in CHTC values when one would employ the correlations based on natural convection (i.e. temperature differences) for interior CHTC in a ventilative cooling case. The underprediction of CHTC, with 72-91%, will lead to an erroneous prediction of heat transfer at the surfaces and thus an unreliable prediction of the effect of ventilative cooling on the overall building energy demand and/or thermal comfort. The CHTC obtained with the correlations by Spitler et al. (1991) (28.8 W/m^{2}K) is about 15% higher than the average CHTC from the six measurements (25.1 W/m^{2}K). However, the differences between the CHTC obtained from this correlation and the experimentally obtained local values can be as high as 62% (i.e. for ID3179). Note that the value of J in the correlation by Spitler et al. (1991), which is equal to the term in brackets in Table 1, is higher than the limits mentioned in their paper (J > 0.011), however, the correlation is included for the sake of comparison. The correlation by Loveday and Taki (1996), which is developed for exterior building surfaces, provides CHTC values close to the lowest values obtained from the measurements, i.e. around 17 W/m^{2}K. However, the difference in CHTC at locations with high measured values of CHTC is large as well for this correlation, up to 48% for ID3184, while the average difference is about 31%. Note that a disadvantage of the correlations by Spitler et al. (1991) and Loveday and Taki (1996) is that it is based on the supply or local velocities and thus results in one value for all locations on a particular surface. In addition, it is not always clear at which location to select the value of the velocity as input for the correlation.

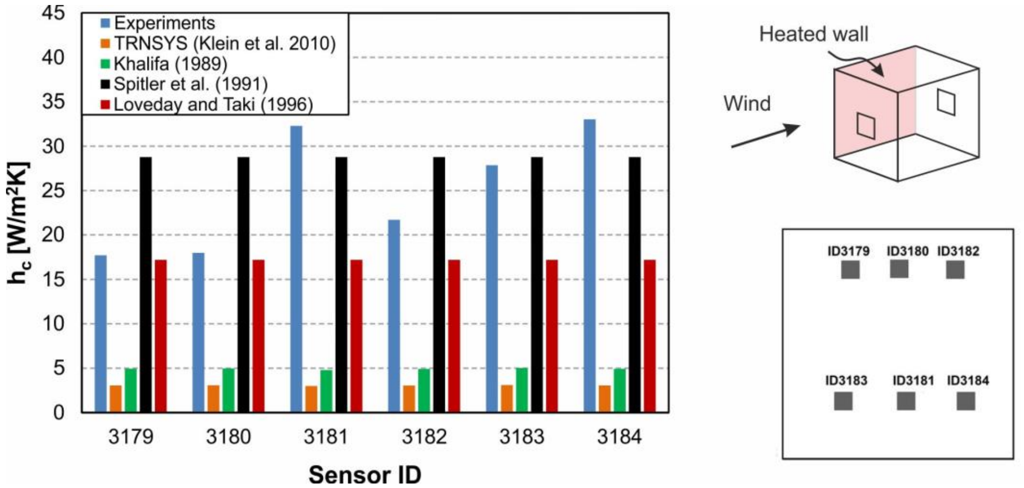

Figure 6: Convective heat transfer coefficients h_c (W/m^{2}K) based on experiments and convective heat transfer correlations at the six positions at the interior surface of the heated wall. Note that the experiments were conducted in a reduced-scale model, while the CHTC correlations were developed for real-scale cases.

4 DISCUSSION AND FUTURE WORK

The main goal of the research presented in this paper is the collection of data for detailed validation studies of non-isothermal CFD simulations of cross-ventilation. In addition, the aim is to analyze the flow and heat transfer in the enclosure and to provide a comparison with existing CHTC correlations from literature. This research is part of a larger research effort on the modeling and optimization of ventilative cooling. Future work will include CFD simulations that will be validated using the experimental data presented in this paper. Subsequently, the CFD simulations can be used to provide detailed data on CHTC distributions in buildings subjected to ventilative cooling, with the aim to develop more accurate CHTC correlations for these cases. Eventually, the new correlations could be used later in BES models to assess their influence on the calculated cooling demand and thermal comfort.

The CHTC values from the experiments indicated the large spatial differences that can occur, even in a very simple geometry subjected to cross-ventilation. Note that the experiments were conducted in a reduced-scale model, while the CHTC correlations were developed for real-scale cases. Irrespective of this difference, most correlations for forced convection provide one single value for an entire wall, floor or ceiling and the information provided is thus limited. In addition, temperature differences are often neglected. On the other hand, although the correlations for natural convection take into account temperatures and can predict local values of CHTC, their results do show large differences with the measured values and thus appear to be not suitable in case of ventilative cooling.

5 CONCLUSIONS

This paper presented results of an experimental campaign in the wind tunnel of the University of Southampton. The experiments were conducted for a cross-ventilated generic enclosure with a heated wall, and included measurements of velocities, turbulence levels, air temperatures, surface temperatures and heat fluxes. From these experiments the CHTC values on the heated wall were calculated and these values were subsequently compared with existing CHTC correlations from literature. The results indicated a clear influence of the indoor airflow pattern on air and surface temperatures, and heat fluxes. For example, the six measured heat fluxes on the heated wall differ up to 46%. The CHTC correlations based on temperature differences strongly underpredict the CHTC on the heated wall. The CHTC calculated using the correlation by Spitler et al. (1991) for enclosures with high volume flow rates shows a good agreement with the averaged values obtained from the experiments (15% difference), although local differences can still be as high as 62%. The correlation by Loveday and Taki (1996) provides a good agreement for locations with low values of CHTC, however, large difference (up to 48%) are present for other locations. On average, the difference with the correlation by Loveday and Taki (1996) is 31%. One must note that the experiments were conducted in a reduced-scale model, while the CHTC correlations were developed for real-scale cases; this difference in scale can contribute to the observed discrepancies.

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