ASSESSING CONVECTION MODELLING IN BUILDING ENERGY SIMULATION MODELS FOR NIGHT COOLING

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ABSTRACT

A sensitivity study is conducted with TRNSYS 17 to quantify the influence of different convection correlations during night ventilation on the thermal comfort. Furthermore, three other parameters were varied as well: the air change rate at night, the internal gains, and the accessibility of the thermal mass in the ceiling. The influence of using the Richardson number as correlation selection criteria is investigated. The results show that there is a high risk in overestimating the efficiency of night ventilation, when current assumptions regarding the prevalence of mixed and forced convection regimes are used.

INTRODUCTION

Nowadays, the application of active cooling in office buildings is hardly questioned in the building practice. Also in temperate climates, it is often assumed to be a necessary condition for a good thermal comfort. However, passive techniques such as night ventilation have great promise.

Night cooling can maintain thermal comfort during the day, by pre-cooling the building at night using cool exterior air at high air change rates of typically 4 to 10 h^{-1}. In the daytime, the thermal mass buffers the heat released in the building, thereby reducing and delaying air temperature peaks. Although case studies show an effective improvement in thermal comfort (Geros 1999; Pfafferott et al. 2003), there is considerable uncertainty regarding the predicted performance (Breesch 2006).

To assess the annual energy performance of night cooling in a building, Building Energy Simulation (BES) models are used. However, the results are strongly influenced by the convective heat exchange between thermal mass and room air (Artmann et al. 2008; Breesch 2006). Moreover, surface convection is modelled in a simplified way in BES models, especially compared to the current state of modelling of radiation and conduction (Goldstein & Novoselac 2010; Peeters et al. 2011). Three simplifications are usually seen: (1) isothermal surfaces, (2) perfectly mixed zone air and (3) a simplified selection of the convective heat transfer coefficient (CHTC). When simulating a building with night ventilation, these simplifications become increasingly problematic as the convective cooling of the thermal mass is crucial for the predicted performance.

The objective of this paper is to quantify the influence of the CHTC on the predicted thermal comfort. Therefore, an office room is simulated in TRNSYS 17, while varying the CHTC at the ceiling during the period of night ventilation. Existing correlations from literature are used and the sensitivity on the thermal comfort of the room is determined and compared with other parameters such as internal gains, air change rate (ACH) and accessibility of thermal mass. Furthermore, the applicability of existing correlations is questioned, and the influence of correlation selection criteria is investigated.

LITERATURE STUDY

The problem statement of this research is founded on the combination of two factors: (1) the lack of convection modelling in BES models adapted to night ventilation and (2) the sensitivity of night ventilation to this phenomenon. Therefore, the literature study discusses these two aspects in more detail.

Convection in BES models

According to the ASHRAE Technical Committee 4.7 (ASHRAE 2004), the modelling of internal surface convection in BES-tools is ranked as one of the highest priority research topics. Fohanno & Polidori (Fohanno & Polidori 2006) and Goldstein & Novoselac (Goldstein & Novoselac 2010) both mention that the modelling of surface convection is overly simplified, certainly when compared to the current state of modelling of radiation and conduction. Also Strachan et al (Strachan et al. 2008), who made a validation of the ESP-r software, attributed a large part of the uncertainty to the modelling of the internal surface convection. Dominguez-Munoz et al (Dominguez-Muñoz et al. 2010) performed a sensitivity study with regard to the influence of input data for the determination of the peak cooling load. Even though the range used for the internal convective heat transfer coefficient (CHTC) is limited (6.25-7-9 $W/(m^2.K)$), the CHTC is still one of the top two parameters, together with the accessible internal mass per unit area.

A first problem of the simplified modelling is that BES models assume isothermal surfaces and therefore surface-averaged CHTC's. Additionally, only one or very few temperature nodes are used to describe the room air, assuming mixed air and limiting the choice of an appropriate reference temperature. Furthermore, a third problem is the selection of an appropriate (surface-averaged) CHTC-value. Crawley et al (Crawley et al. 2008) made an overview of existing BES models and their capabilities, including the state of internal convection modelling. According to this overview, most BES models calculate the internal CHTC's depending on temperature differences, hereby putting the emphasis on the use of natural convection correlations. Generally, it is possible to manually input constants or correlations in the BES model. However, most users do not have sufficient background to do this accurately or they ignore the flow characteristics: the flow pattern will determine the locally occurring convection regime. As an improvement, some tools offer a coupling between BES and CFD, such as TAS, ESP-r and EnergyPlus. Bartak et al (Bartak 2002) made an illustration of the conflation algorithm in ESP-r and its internal CFD module. Zhai & Chen (Zhai & Chen 2004) described the coupling of BES and CFD focusing on the requirements of the boundary region. The correct modelling of the convective surface heat flux (CSHF) in CFD requires a fine boundary layer mesh, as well as a turbulence model capable of modelling low turbulence flows (Loomans 1998). Therefore, a compromise must be found between calculation time and accuracy, though these are often not compatible. As a coupled simulation still requires a high computational effort as well as high user skills (Peeters et al. 2011), a more pragmatic way is needed.

Some tools allow an automatic determination of the CHTC based on the type of air flow. IES<VE> for example provides a simplified method, based on the mean room air velocity. However, currently, the most pragmatic procedure to determine CHTC based on the type of air flow is the correlation selection algorithm developed by Beausoleil-Morrison (Beausoleil-Morrison 2000), which is implemented in ESP-r and EnergyPlus. This selection algorithm distinguishes between five main flow regimes, depending on the driving force of the flow and the cause of temperature differences. At the start of the simulation, the possible convection regimes for each zone are determined, and a set of CHTC-correlations is selected from a total of 28 equations from literature. Each surface is assigned a number of possibly occurring correlations, based on its characteristics. Determination of the local regimes in each zone is done through a series of user prompts in ESP-r, inquiring about the presence and location of heating elements, fans, windows etc. Depending on the situation at each time step, a correlation is implemented at each surface. For example, when a ceiling diffuser is active, the Fisher (Fisher 1995) or Fisher-Pedersen (Fisher & Pedersen 1997) correlation for a ceiling-jet causing forced convection at ceilings is used at the ceiling. When the diffuser is inactive, either the stratified or buoyant natural convection correlation from Alamdari & Hammond (Alamdari & Hammond 1983) is used, depending on the temperature difference between ceiling surface and room air.

In EnergyPlus, the dimensionless Richardson number (Ri) is used as well to assess the flow regime. This number gives the ratio of the buoyant forces over the momentum forces and is calculated with Equation (1), in which the Grashof number Gr is calculated using the zone height as characteristic length L, whereas the Reynolds number Re is based on $\sqrt[3]{V}$. The software assumes forced convection in the zone for Ri < 0.1, natural convection for Ri > 10 and mixed convection when Ri is between these values.

$$Ri = \frac{Gr}{Re^2} = \frac{g \cdot \beta \cdot \Delta T}{u^2} \cdot \frac{L_{Gr}^3}{L_{Re}^2} \qquad (1)$$

This pragmatic approach allows a permanent evaluation of the dominant convection regimes throughout the simulation, without significantly increasing the calculation time. It can also easily be refined further through the addition of newly developed correlations. In EnergyPlus, new correlations by Fohanno & Polidori (Fohanno & Polidori 2006) and Goldstein & Novoselac (Goldstein & Novoselac 2010) were added to the original structure. The accuracy and success of the correlation selection algorithm is determined by the number of implemented correlations and an appropriate selection of a correlation for each surface at each time step. An extensive discussion of existing correlations, their derivation and applicability can be found in (Goethals et al. 2011; Peeters et al. 2011; Beausoleil-Morrison 2000).

Sensitivity of night ventilation regarding surface convection

Breesch (Breesch 2006) performed a sensitivity and uncertainty analysis on natural night ventilation. Apart from the internal gains, the air tightness and the g-value of the solar blinds, the internal CHTC caused the largest sensitivity on the thermal comfort (case with cross ventilation, south orientation). The sensitivity for the CHTC-value was for example three times higher than the sensitivity due to the air flow rate, even though only natural convection correlations were used in the analysis. A follow-up study (Breesch & Janssens 2010) investigated natural night ventilation and distinguished between buoyant and stratified horizontal surfaces with different correlations for natural convection. The research indicated that the sensitivity with regard to the CHTC is twice as important as the control strategy, given that other parameters are not changed.

Also Artmann et al (Artmann et al. 2008) performed a sensitivity study with regard to night ventilation and found a high sensitivity for the CHTC for values lower than 4 W/(m^2.K) in case of buildings with average mass, which is defined as a building with exposed concrete ceilings and gypsum walls. These conditions are expected also in realistic cases where night ventilation is applied.

Goethals et al (Goethals et al. 2011) made a detailed overview of correlations existing in literature and their application in BES. Using this information, a sensitivity study was conducted on one room with mechanical ventilation (10 ACH) using TRNSYS. A high sensitivity was found during the periods of night ventilation, also with regard to the fan operation time.

Finally, Leenknegt at al (Leenknegt et al. 2011) showed that the flow pattern during night ventilation can change drastically during the period of increased ventilation. This has large repercussions on mainly the surface convection at the ceiling. An accurate prediction of the flow evolution is therefore crucial to assign appropriate CHTC's at the ceiling.

CASE DESCRIPTION

A sensitivity study was conducted in TRNSYS 17, in order to investigate the sensitivity of the thermal comfort to different correlations. Convection and radiation are modeled with the old star temperature approach. This paragraph will focus solely on the convection at floor and ceiling, and will not comment on the selection of correlations for walls or windows.

A heated office room was simulated from January until August. The first six months are used as initialization period for the room and the thermal comfort during July and August is assessed. Office hours are assumed from 8 h to 18 h and night ventilation is used between 20 h and 6 h.

The highly insulated facade with window is oriented south and is exposed to a Belgian climate. Floor and ceiling have an identical composition: a 1 cm floor finishing, followed by 8 cm of light concrete and 20 cm of heavy concrete, with densities of respectively 650, 1200 and 2400 kg/m^3. All adjacent zones have conditions identical to the investigated zone, resulting in adiabatic internal walls. Following a common rule of thumb, the thermal capacity of the air is multiplied by 5 to take into account influence of furniture. A time step of 600 s was used. The boundary conditions are summarized in Table 1.

Four parameters are varied: the internal heat gains (IG), the air change rate at night (ACH), the CHTC at the ceiling during the increased ventilation and the accessibility of the thermal mass. Firstly, the internal heat gains vary from 12 to 30 W/m^2 with steps of 3 W/m^2. They are divided as 70 % convective and 30 % radiative. Outside office hours, 10 % of the total internal gains are assumed. The air change rate at night was varied from 4 to 12 h^{-1} with steps of 1 h^{-1}. Thirdly, eight CHTC correlations are selected for usage at night; an overview is given in Table 2. During the daytime, the default natural convection correlations from TRNSYS are used at the internal surfaces. No reference was found for these correlations, which are given in Equation (2) and (3) for buoyant and stratified flow respectively over a horizontal surface.

Finally, the simulations are made with and without a dropped ceiling panel of plasterboard, with a horizontal air gap of 5 cm, blocking direct contact between room air and the concrete slab.

$$h_c = 2\Delta T^{0.31} \qquad (2)$$

$$h_c = 1.08\Delta T^{0.31} \qquad (3)$$

This parameter variation results in a total of 1008 cases. The cases are compared using the weighted overheating hours (WHO in [Kh]), given by Equation (4). The maximum comfort temperature T_{lim} is set at 25 °C. The WOH take into account how many hours of the year the maximum comfort temperature was exceeded, but also by how many K. As such, this is a weighted evaluation of the overheating time. The impact of moisture absorption and desorption by the building construction and furnishings was ignored.

$$WOH = \sum_{i=t_0}^{t_{end}} max\left(\left(T_{a,i} - T_{lim}\right) \cdot \Delta t, 0\right) \qquad (4)$$

Table 1. Boundary conditions

Room parameters	
room dim.	W 1.8 x L 3.45 x H 2.4 m
window char.	U_{gl} = 1.1 W/(m^2.K), g = 0.39
window dim.	W 1.25 x H 1.6 m
heating setpoint	night: 10 °C, day: 20 °C
heating power	2 kW
CHTC	
internal walls	internal calculation TRNSYS 17
façade exterior	17.8 W/(m^2.K)
floor	internal calculation TRNSYS 17
ceiling – day	internal calculation TRNSYS 17
ceiling – night	**parCORR see Table 2**
Internal gains (IG)	
rad/conv. part	30/70 %
18h – 8h	10 % IG
8h – 18h (office hours)	from 8-9h: linear increase from 10 to 100 % IG 9h-17h: 100 % IG from 17-18h: linear decrease from 100 to 10 % IG
internal gains	**parIG (W/m^2) 12/15/18/21/24/27/30**
Ventilation	
T_{supply}	T_{ext}
supply opening	1 x 0.1 m
infiltration	0.2 h^{-1}
day (hygienic)	1 h^{-1}
Night (purge)	**parACH: 4/5/6/7/8/9/10/11/12 h^{-1}**

Table 2. Used correlations in parameter variation

Code	Description
TRN_NaHorStr	default calculation TRNSYS 17
AwH_ForClStr	forced, jet over heated ceiling (Awbi & Hatton 2000)
BM_MixClStr	mixed, ceiling diffuser, warm ceiling (Beausoleil-Morrison 2000)
Fi_ForClHor	forced, free horizontal jet in isothermal room (Fisher 1995)
Fi_ForClDif	forced, ceiling jet in isothermal room (Fisher 1995)
AwH_MixClStr	mixed, jet over heated ceiling (Awbi & Hatton 2000)
AwH_NaClStr	natural, stratified flow, ceiling (Awbi & Hatton 1999)
CTE-ISO	horizontal surface, upwards heat flow = 5 W/(m².K); downwards heat flow = 0.7 W/(m².K) (ISO 13791)

DISCUSSION OF RESULTS

For each case, the total WOH are summed up over the two month period. The influence of varying ACH, IG and CHTC are visualized below for the simulations with accessible thermal mass. Figure 1 show the total amount of WOH during the 2 summer months. The cases with different correlations are distinguished by different markers. They are then sorted in groups, first according to internal gains (i.e. 1-9: 12 W/m², 10-18: 15 W/m², etc.), then according to ACH, and numbered on the horizontal axis. For example, cases 1-9 all assume 12 W/m², but they have stepwise increasing ACH. For example, case 24 on Figure 1 is the case with 18 W/m² of internal gains and an ACH of 9 h^{-1}. Figure 1 shows a few clear tendencies: as the ACH increases, the overheating decreases. However, this effect starts to level out at higher ACH, seen by a decrease in slope. For example, at IG equal to 15 W/m², a change from 4 to 5 ACH results in a reduction of 22 % of WOH, whereas a change from 11 to 12 ACH results only in a reduction of 10.6 %. At 21 W/m², the reductions are respectively 21.7 % and 8.9 %, and at 27 W/m², respectively 20 % and 8 %.

The impact of different correlations can also be seen on Figure 1, though this is more clear on Figure 2, showing the change in WOH for each case compared to its reference case using the default correlations of TRNSYS, but with the same ACH and IG. The convection correlation is therefore the only parameter influencing the values plotted here. A positive value indicates an increase in WOH, and a negative value is equivalent to a decrease in WOH and therefore an improvement of the predicted thermal comfort.

Correlations AwH_NaClStr and CTE-ISO represent natural convection, respectively by Awbi & Hatton (Awbi & Hatton 1999) and from EN ISO 13791. They will result in rather low CHTC values at night, as the ceiling surface is typically warmer than the air at these times. Additionally, AwH_ForClStr refers to the forced convection correlation for a jet over a heated ceiling by Awbi & Hatton (Awbi & Hatton 2000). Its value is still rather low however, as it is based on the jet momentum number. As the supply opening is quite large, the supply air velocity is low and so is the jet momentum. Finally, AwH_MixClStr takes into account combined natural and forced convection, developed by Awbi & Hatton, and is based on correlations AwH_ForClStr and AwH_NaClStr. At low ACH, the mixed convection results agree best with the natural convection results. As the ACH increases, the forced convection based CHTC becomes higher than the natural convection based, and the forced convection part, though still low, becomes dominant in the mixed convection result. Furthermore, increasing the internal gains obviously increases the overheating, but it also reinforces the influence of the correlation on the overheating by 5 to 15 %.

Correlations BM_MixClStr and Fi_ForClDif both result in a decrease of overheating hours compared to their parallel reference cases. They show very similar behaviour throughout all cases. BM_MixClStr combines natural and forced convection, but is dominated by the forced convection part, equal to Fi_ForClDif (Fisher 1995). This explains their similar behaviour. The CHTC predicted by these correlations is rather high, as they assume air supply through ceiling diffusers. Therefore, they result in a significant decrease of the predicted overheating, with the influence going from 10 to 50 % decrease in Kh as the ACH increases.

The last correlation is Fi_ForClHor, which is the forced convection correlation for a free horizontal jet by Fisher (Fisher 1995). At low ACH, this correlation behaves similar as the natural convection correlations, but at ACH > 8 h^{-1}, it shifts towards forced convection, with rapidly decreasing overheating hours at increasing ventilation rates.

A fourth parameter was whether or not the thermal mass was made accessible. The influence of covering up the concrete slab at the ceiling is shown in Figure 3. Here, the spread on thermal comfort for all cases is shown for the different correlations. Each boxplot represents 63 cases with varying IG and ACH. On Figure 3a, the cases with accessible thermal mass are given, whereas on Figure 3b, the ones without are displayed. First of all, the thermal comfort is substantially worse if the thermal mass is covered up. The shift in WOH due to varying correlations is clear if the thermal mass can be reached. The same tendencies are seen in the cases without accessible thermal mass, though they are less obvious. In the latter case, the influence of the correlations on the average WOH of all 63 cases is between -3.5 % and + 11.5 % depending on the correlation used. The case with internal correlation from TRNSYS (TRN NaHorStr) was used as the reference.

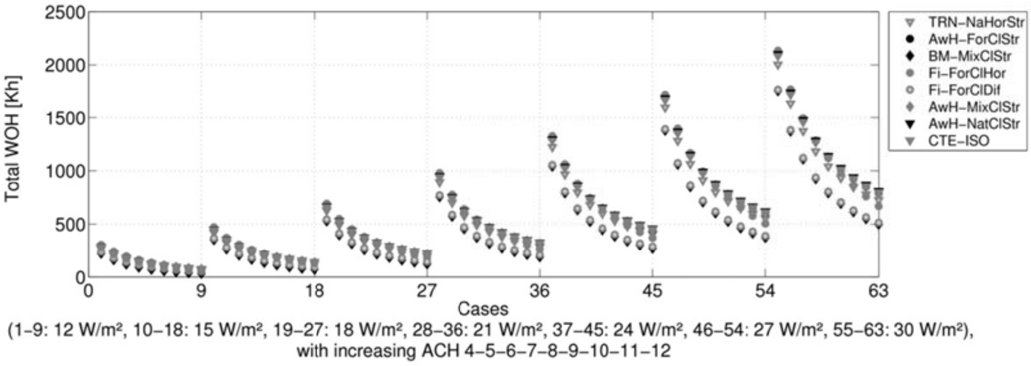


Figure 1. Total WOH for each case

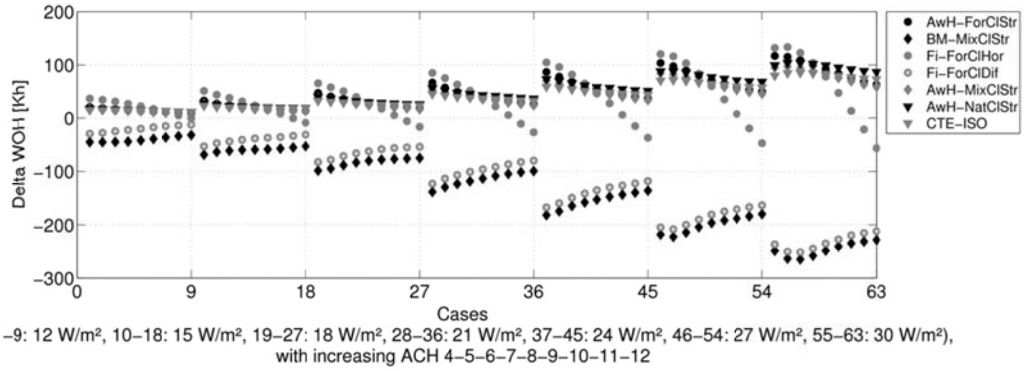
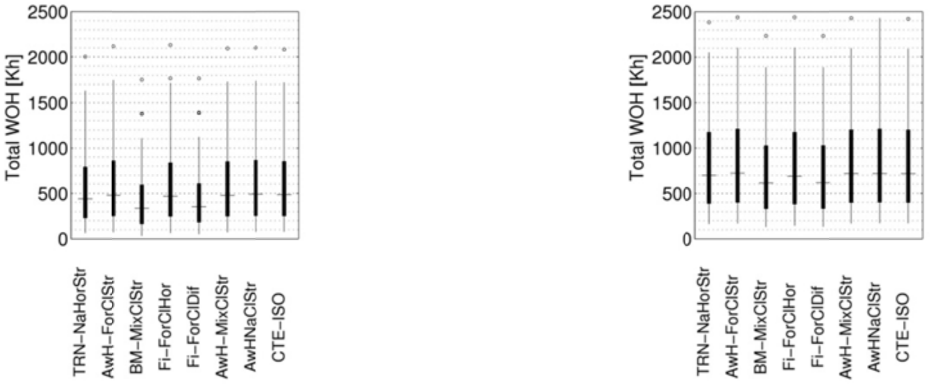


Figure 2. Influence of different correlations for cases with accessible thermal mass, legend is found in Table 2



a. with accessible thermal mass b. without accessible thermal mass

Figure 3. Spread on thermal comfort, with and without thermal mass. Comparison of correlations

In case of accessible thermal mass, this range is from -9 % to +23 %. This shows that the sensitivity to convection modeling should not be assessed separately from the specific implementation. Furthermore, this study shows that the thermal comfort is indeed sensitive to the selection of a convection correlation. The mixed and forced convection correlations based on ceiling diffusers cause a significant improvement of the thermal comfort. Although they are actually not valid for this case, they are standard implemented in ESP-r and EnergyPlus, and would most likely have resulted in an overestimation of the thermal comfort.

EVALUATION OF CORRELATION SELECTION ALGORITHM

In the literature study, the correlation selection algorithm (CSA) developed by Beausoleil-Morrison (Beausoleil-Morrison 2000) was discussed shortly. In case of night ventilation, the flow regime would fall under classification E, as there is a system that supplies air to the room (for example, an open window combined with an extraction fan), as well as large thermal gradients due to the temperature difference between the room and ambient air. Therefore, during the period in which night ventilation is active (i.e. window open), mixed convection correlations would be suggested as appropriate. Furthermore, the use of convection correlations in the CSA is based on the assumption that the convection regime remains stable in time when the boundary conditions do not change. In BES models, this transient behaviour is typically evaluated based on the (de)activation of the HVAC systems (i.e. fan on/off). However, previous research showed that the evolution of the flow pattern, initiated by, for example, increased mechanical ventilation, is not stable (Leenknegt et al. 2012). As buoyancy causes the cold air to fall down after entering, rather than adhere to the ceiling, the convection regime at the ceiling is initially stratified. Throughout the continuing flow time, the cool ambient air mixes with the room air and causes a slow decrease of the room air temperature. At a critical point, the flow pattern shifts from buoyancy dominated flow to forced convection flow, causing a much higher surface convection at the ceiling. This transient behaviour is very important to assess the actual energy performance of the night cooling.

It can be concluded that a further refinement of the current evaluation of the flow regime is required to accurately model the surface convection during night ventilation. In EnergyPlus, an additional appraisal of the convection regime is made by checking whether the Richardson number is higher than 10 or lower than 0.1. Between these values, mixed convection is assumed. The same risk of errors therefore still exists in the EnergyPlus methodology.

A solution for this problem could still lay in the use of the Richardson number to further classify the flow regime in the room. The results from the simulation suggest that a critical Ri can be defined to assess when the buoyancy dominated flow will shift towards forced flow. It must be noted that no conclusions can be made on the value of this critical Ri, based on the simulations presented in (Leenknegt et al. 2012), as the simulations are not validated by measurements and are limited to 2D. In an attempt to assess the influence of including Ri, the sensitivity study was repeated for the room with accessible thermal mass, however, mixed and forced convection correlations at the ceiling were applied only when Ri < 1, rather than during the full period of increased ventilation. The results are given in Figure 4, showing the difference in WOH compared to the reference cases with the internal calculation from TRNSYS. The influence of the correlations on the thermal comfort is greatly reduced compared to Figure 2. This is explained by Figure 5, showing the percentage of the nighttime that Ri < 1. It is seen that there is a low prevalence of sufficiently low Ri-values, required to expect mixed or forced convection at the ceiling, during most of the simulations. It is expected therefore, that applying the selection algorithm of Beausoleil-Morrison in case of night ventilation, could greatly overestimate the cooling efficiency.

To illustrate this, a detail of the ceiling surface temperature (STcl) and operative temperature (Top) evolution is given in Figure 6 for the case with 11 ACH at night, 15 W/m² internal gains and with Fi_ForClHor applied during the period of increased ventilation. Two cases are compared here, namely whether the forced convection correlation was implemented only when Ri < 1 (corrRi) or simply during the full period of increased ventilation. The figure displays on the left axis whether or not the night ventilation is active (NV on = 1) and the value of Ri. The right axis shows the temperature difference between the case with Fi_ForClHor and the reference case using the default TRNSYS natural convection correlation. A negative value indicates a lower temperature compared to the reference case.

During the first night, Ri is lower than 1, resulting in the forced convection correlation applied at the ceiling and a strong decrease of the ceiling surface temperature. However, during the third night, Ri is higher than 1, resulting in the natural convection correlation being implemented. During the second night, Ri ranges from 0.88 to 1.6, so the correlation at the ceiling is changed from forced to natural convection, which is clearly reflected in the temperature profiles.

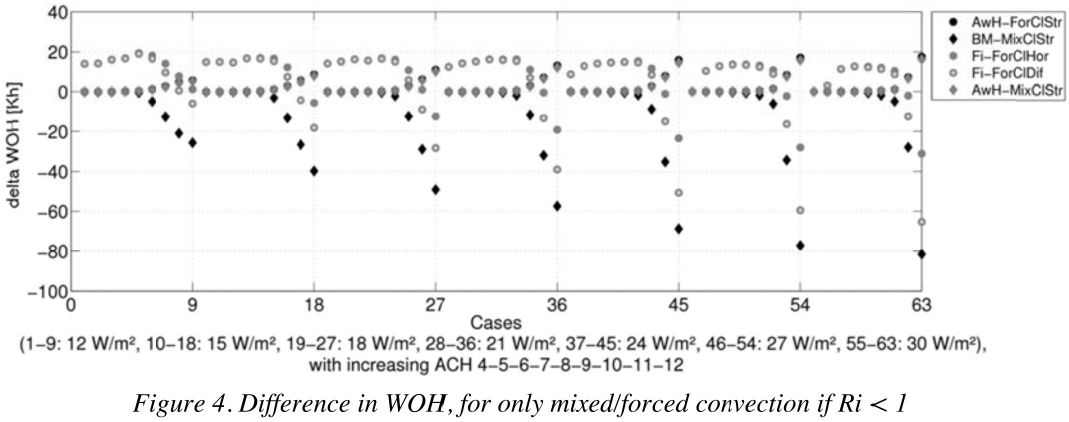


Figure 4. Difference in WOH, for only mixed/forced convection if Ri < 1

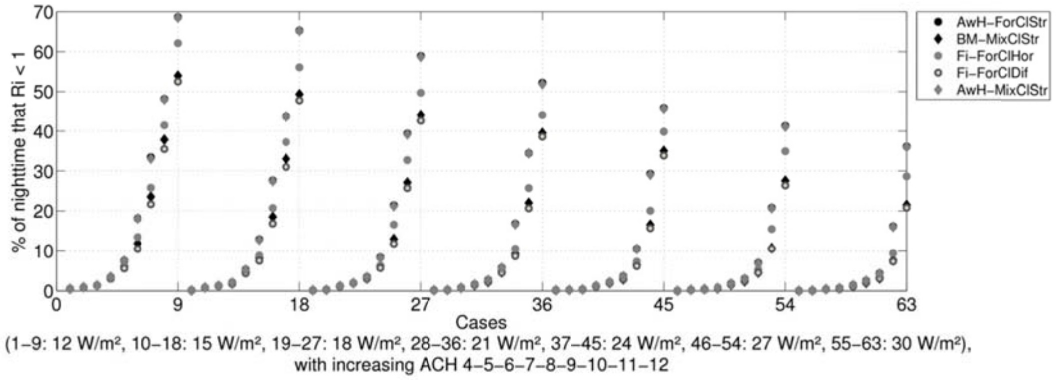


Figure 5. Percentage of night time that Ri < 1

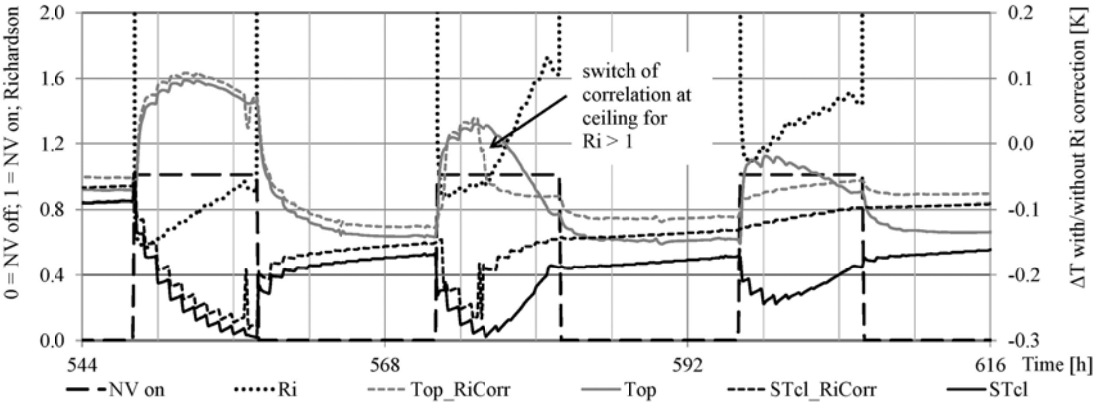


Figure 6. Illustration of influence of Ri <1 on correlation selection

CONCLUSION

The objective of this research was to investigate the current methods for selecting CHTC's in BES models, focusing on night cooling. It was shown that the usage of natural convection correlations could increase the predicted WOH at 4 and 12 ACH respectively with 18 Kh (19 %) and 105 Kh (12 %), whereas forced convection correlations could reduce this at 4 and 12 ACH respectively with 30 Kh (11 %) and 210 Kh (29 %). Although the influence of ACH and IG on the thermal comfort is higher, an appropriate choice of the CHTC during night ventilation remains important.

The sensitivity study was repeated, implementing mixed and forced convection correlations only when Ri < 1. As there is a rather low prevalence of sufficiently low Ri-values during night-time, the influence of applying mixed or forced convection correlations at the ceiling is much lower than before and is limited to a reduction of only 80 Kh in the most extreme case.

In light of this result, the correlation selection algorithm by Beausoleil-Morrison was evaluated with regard to night ventilation. It was shown that following refinements are advised in case of night ventilation.

A more refined classification algorithm is required to predict the flow pattern in the room. Currently, it is assumed that the convection regime remains constant during the (de)activation of HVAC. However, it was shown that the flow pattern is not stable during the period of increased ventilation. The Richardson number can be used to assess the flow pattern, though measurements are required to determine a critical value.

Currently, the local convection regime is classified per zone. However, the simulated cases showed simultaneously mixed or forced convection at the ceiling and natural convection at the floor. Therefore, convection regimes should be determined per surface rather than per zone, based on a prediction of the flow pattern.

NOMENCLATURE

ACH	*Air change rate [h-1]*
β	*thermal expansion coefficient of air* $[K^{-1}]$
D	*room depth [m]*
g	*gravitational acceleration* $[m/s^2]$
Gr	*Grashof number [-]*
H	*room height [m]*
IG	*internal heat gains* $[W/m^2]$
L	*characteristic length [m]*
Re	*Reynolds number [-]*
$T_{a,i}$	*operative temperature in the room at time* t_i
T_{ext}	*outdoor air temperature [°C]*
T_{lim}	*maximum comfort temperature (25 °C)*
T_{sup}	*inlet air temperature [°C]*
ΔT	*temperature difference [K]*
Δt	*simulation time step in TRNSYS (0.167 h)*
u	*air velocity [m/s]*

U_{gl}	*U-value of glazing [W/(m²K)]*
W	*room width [m]*

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REFERENCES

Alamdari, F. & Hammond, G.P., 1983. Improved data correlations for buoyancy-driven convection in rooms. *Building Services Engineering Research and Technology*, 4, pp.106–112.

Artmann, N., Manz, H. & Heiselberg, P., 2008. Parameter study on performance of building cooling by night-time ventilation. *Renewable Energy*, 33(12), pp.2589–2598.

ASHRAE, 2004. *ASHRAE Technical Committee 4.7: Energy Calculations*.

Awbi, H.B. & Hatton, A., 2000. Mixed convection from heated room surfaces. *Energy and Buildings*, 32, pp.153–166.

Awbi, H.B. & Hatton, A., 1999. Natural convection from heated room surfaces. *Energy and Buildings*, 30, pp.233–244.

Bartak, M., 2002. Integrating CFD and building simulation. *Building and Environment*, 37(8-9), pp.865–871.

Beausoleil-Morrison, I., 2000. *The adaptive coupling of heat and air flow modelling within dynamic whole-building simulation*. University of Strathclyde, Glasgow, UK.

Breesch, H., 2006. *Natural night ventilation in office buildings*. Ghent University, Belgium.

Breesch, H. & Janssens, A., 2010. Performance evaluation of passive cooling in office buildings based on uncertainty and sensitivity analysis. *Solar Energy*, 84(8), pp.1453–1467.

Crawley, D. et al., 2008. Contrasting the capabilities of building energy performance simulation programs. *Building and Environment*, 43(4), pp.661–673.

Dominguez-Muñoz, F., Cejudo-López, J.M. & Carrillo-Andrés, A., 2010. Uncertainty in peak cooling load calculations. *Energy and Buildings*, 42(7), pp.1010–1018.

Fisher, D.E., 1995. *An experimental investigation of mixed convection heat transfer in a rectangular enclosure*. University of Illinois, Urbana, Illinois, USA.

Fisher, D.E. & Pedersen, C.O., 1997. Convective heat transfer in building energy and thermal load calculations. *ASHRAE Transactions*, 103(2), pp.137–148.

Fohanno, S. & Polidori, G., 2006. Modelling of natural convective heat transfer at an internal surface. *Energy and Buildings*, 38(5), pp.548–553.

Geros, V., 1999. Experimental evaluation of night ventilation phenomena. *Energy and Buildings*, 29(2), pp.141–154.

Goethals, K., Breesch, H. & Janssens, A., 2011. Sensitivity analysis of predicted night cooling performance to internal convective heat transfer modelling. *Energy and Buildings*, 43(9), pp.2429–2441.

Goldstein, K. & Novoselac, A., 2010. Convective heat transfer in rooms with ceiling slot diffusers. *HVAC&R Research*, 16(5), pp.629–656.

Leenknegt, S. et al., 2011. Improving the modelling of surface convection during natural night ventilation in Building Energy Simulation Models. In *Building Simulation 2011*. Sydney, NSW, Australia.

Leenknegt, S. et al., 2012. Numerical sensitivity study of transient surface convection during night cooling. *Energy and buildings*, 53, pp.85–95.

Loomans, M., 1998. *The measurement and simulation of indoor air flow*. Technische Universiteit Eindhoven.

Peeters, L., Beausoleil-Morrison, I. & Novoselac, A., 2011. Internal convective heat transfer modeling: Critical review and discussion of experimentally derived correlations. *Energy and Buildings*, 43, pp.2227–2239.

Pfafferott, J., Herkel, S. & Jäschke, M., 2003. Design of passive cooling by night ventilation: evaluation of a parametric model and building simulation with measurements. *Energy and Buildings*, 35, pp.1129–1143.

Strachan, P., Kokogiannakis, G. & Macdonald, I., 2008. History and development of validation with the ESP-r simulation program. *Building and Environment*, 43(4), pp.601–609.

Zhai, Z. & Chen, Q., 2004. Numerical determination and treatment of convective heat transfer coefficient in the coupled building energy and CFD simulation. *Building and Environment*, 39, pp.1001–1009.