Convective Heat Transfer Coefficients in mechanical night ventilation: a sensitivity analysis

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ABSTRACT

Since the Energy Performance for Buildings Directive (EPBD) was accepted and implemented over the course of the last years, buildings are audited energetically to receive the necessary construction licenses. This augmented the already high attention to research on innovative (passive) energy-saving system concepts even further. Previous research suggests that, although the effect of commissioning can be significant, specific fan power is the most important factor influencing the energetic viability of mechanically driven night ventilation as an active cooling replacement. This parameter should thus be the central point of focus during the design process. In this paper, we present an analysis of the effect of detailed convective heat transfer modeling on the predicted performance, in order to determine the level of detail needed to assess feasibility of this kind of system in early design phases. Results indicate that the effect amounts to 20-50% of the predicted performance and therefore cannot be neglected. It is within the range of effect of the dominant parameter, specific fan power. In light of these results, it is suggested that detailed convective heat transfer coefficient modeling is taken into account whenever forced convection due to large volume flow is introduced.

INTRODUCTION

Since the European Energy Performance for Buildings Directive (EPBD) [1] was accepted and implemented over the course of the last years, buildings are audited energetically to receive the necessary construction licenses. This augmented the already high attention for research on innovative (passive) energy saving system concepts even further. Validation of the viability – energetical, economical, ecological, comfortwise … - of these innovative systems thus became an important issue.

Laverge and Janssens [2] assessed the energetic feasibility of mechanically driven night ventilation, combined with an earth-air heat exchanger (EAHX) and a heat recovery wheel, as an active cooling replacement through a performance evaluation of the concept in a high profile office building in Nazareth (Belgium). This indicated that the mechanically driven night ventilation seems incapable of rendering better energetic performance than the active cooling alternative for the case under consideration. Supplementary ventilator energy at night was within the range of the avoided cooling energy in both measurements and simulations. In these papers, fixed coefficients for convective heat transfer were used.

However, among others Goethals and Janssens [3] demonstrated that the influence of more detailed modeling of convective heat transfer can render significantly different results in energy simulations. Therefore, this paper presents a sensitivity analysis of the performance of this system with different assumptions on convective heat transfer.

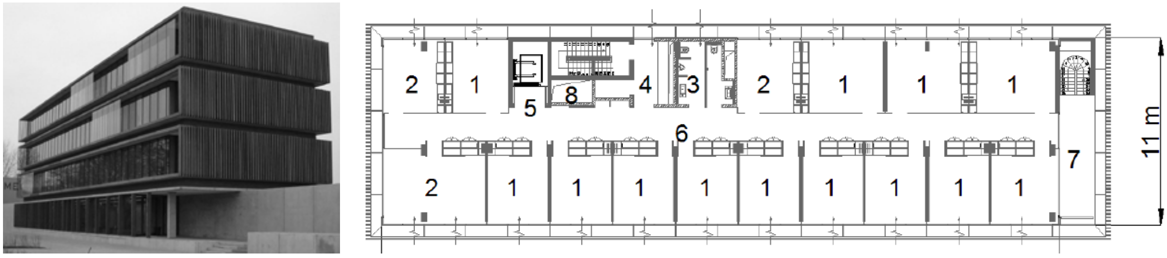
METHODS

Description of the testcase Building

As stated in the introduction, the reported research investigates the energy saving potential of (passive) systems introduced in a high profile office building in Nazareth, Belgium. It is home to the Omega Parma Belgian head quarters.

a) Geometric characteristics

The building consists of a ground floor with reception desk, three office floors ontop and a basement containing technical installations. Figure 1 a) shows the front façade, which is oriented almost exactly to the east. b) depicts the typical plan of the office floors.



a) b)

Figure 1. a) façade, b) floor plan.

b) Climatisation approach

Figure 3 is a graphical representation of how the night ventilation flows circulate trough the building. Extraction fans pull fresh air in trough electronically controlled valves in the technical floors. The air then circulates trough the offices and corridors and is extracted in the kitchenette area.

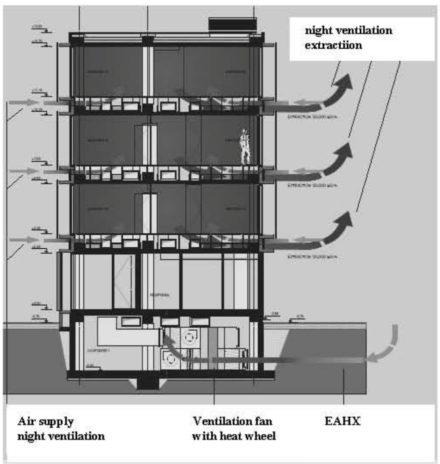


Figure 2. night ventilation flow path.

The mechanical night ventilation is activated when the following conditions are cumulatively met:

- Time control: between 22h and 6h, 7 days aweek;
- Indoor temperature > 23 °C;
- Temperature difference inside-outside > 2 °C;
- Outdoor temperature > 15 °C;
- Max. outdoor temperature over the last day > 22°C
- Windspeed < 50 km/h
- Max. indoor temperature over the last day > 23°C

The model

To assess the energy saving potential of the mechanical night ventilation, the trade off between saved active cooling and consumed electrical energy needs to be evaluated. For this purpose a TRNsys model was constructed that represents one bay of the building.

Walls separating the bay from neighboring bays and both floor and ceiling were treated as adiabatic surfaces. This makes the model representative for the middle office floor of the building. Although this strategy neglects possible energy flows between the different floors and bays, it accurately models the physically present thermal mass in the bay and with it, the thermal capacitance available for night cooling.

A few of the geometrical and numerical constraints are listed below:

- Glass surface of the bay: 9.36 m², U-value: 1,1 W/m²K, g-value: 0,26;
- Gains: 280W per office, from 8 to 22h;
- Separations between offices and corridor: single glass pane: U-value: 3,9;
- HVAC set points: heating 20 °C, cooling 26 °C;
- HVAC operation hours: 8 to 22h.

The activation algorithm respects all criteria described above. To ensure that an acceptable comfort is reached in the simulation cases, active heating and cooling is introduced in the model. These installations have unlimited maximal power and are configured to the set points indicated above. Furthermore, it is assumed that the fans for mechanical night ventilation, once activated, always run at maximum capacity and consume 130 W of electrical energy per office. With the design air flow rate, these fans have a Specific Fan Power of 811 Ws/m3 or SFP category 3 [4], which is fairly good. These data are deducted from on site electricity consumption measurements.

To assess the energy saving potential of the mechanical night ventilation under different activation criteria, a year long simulation is ran for each variant, integrating sensible heat demand for cooling and heating. These data are then compared to a base case without mechanical night ventilation. The activation time of the extraction fans is then multiplied by their nominal electric load to assess additional power consumption of the system. The ratio of the saved cooling and heating demand to this additionally required energy should not exceed the COP of a modern cooling unit for this technique to be energetically interesting.

In order to evaluate the sensitivity of the system to the setpoints of the control algorithm, the performance of the system was calculated under different assumptions for these setpoints. All possible combinations were simulated, amounting to 324 simulations in total per setup.

Four parameters were varied over the following ranges:

- Time control: daily, starting at 20, 22 and 24h and stopping at 4, 6, 8, 10h;
- Indoor temperature > 25, 23, 20 °C;
- Outdoor temperature > 20, 15, 10 °C;
- Temperature difference indoor / outdoor > 0, 2, 4 °C

Three different approaches for the modeling of the convective heat transfer coefficients were implemented. Al simulations are executed with convective heat transfer coefficients (CHTC) according to the ISO standard [4], the correlation proposed by Awbi and Hatton for natural convection [5] and the correlation for mixed convection introduced by Beausoleil – Morrisson [6]. The sensitivity of the overall performance of the system to these assumptions is assessed, together with the influence on the performance of a control strategy.

ISO 13791 has fixed CHTC for vertical surfaces, while horizontal surfaces have alternating CHTC depending on the direction of the heat transfer. Awbi and Hatton formulated the CHTC as a function of the temperature difference between the surface and the air:

$$CHTC_{natural} = C(\Delta\theta)^n, \qquad (1)$$

with C a fixed coefficient depending on the geometry of the room (hydraulic diameter) and empirically found constants. N is an empirically determined exponent.
Mixed convection was treated by Beausoleil-Morrison by the introduction of a blending function for natural and forced convection CHTC, combining correlations applicable for each of these.

$$CHTC_{mixed} = \left(CHTC^n{}_{natural} + CHTC^n{}_{forced}\right)^{1/n}, \qquad (2)$$

RESULTS

In fig. 3, the mean seasonal performance, given as seasonal performance factor (SPF) or the ratio of cooling load and electrical fan power, of the system in this case is plotted for the 3 different assumptions on convective heat transfer.

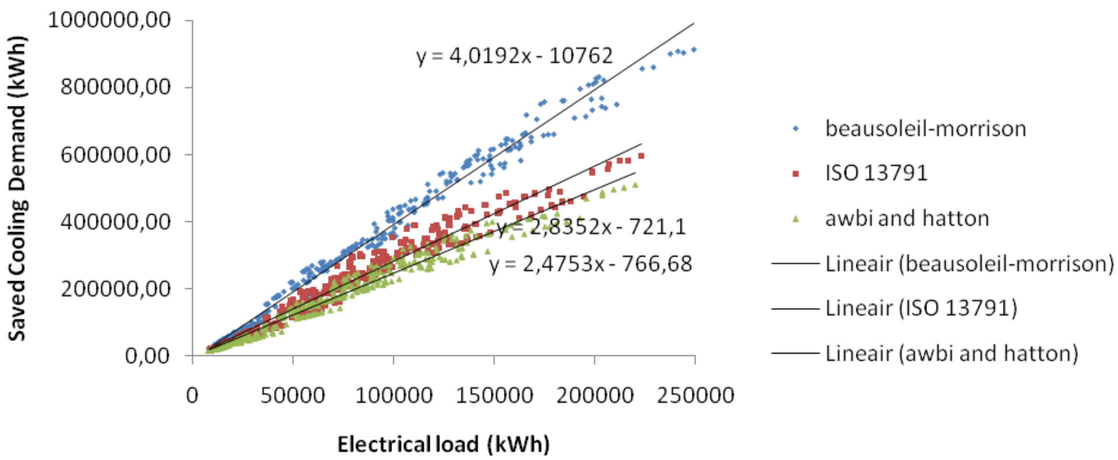


Figure 3. SPF for all simulations, according to the ISO standard, Awbi and Hatton and Beausoleil-Morrison. (x-axis: electrical load, y-axis saved active cooling load in kWh)

It can clearly be seen from this graph that for overall performance, Beausoleil-Morisson predicts significantly better performance than both others (mean SPF 4 vs. 2.5). This is due to the rather large air change rate (10 Ach) and the higher fluxes for low temperature differences predicted by Beausoleil-Morrison in these circumstances.

Fig. 4. plots the heat flux to the ceiling surface for different temperature differences between surface and room air predicted by the ISO Standard, Awbi and Hatton and Beausoleil-Morrison for a 10 Ach situation. In this graph, the discussed effect of the Beausoleil-Morrison correlation in the low temperature difference range can clearly be seen. Since Awbi and Hatton predict much smaller fluxes, the lesser performance seen in Figure 4. is to be expected.

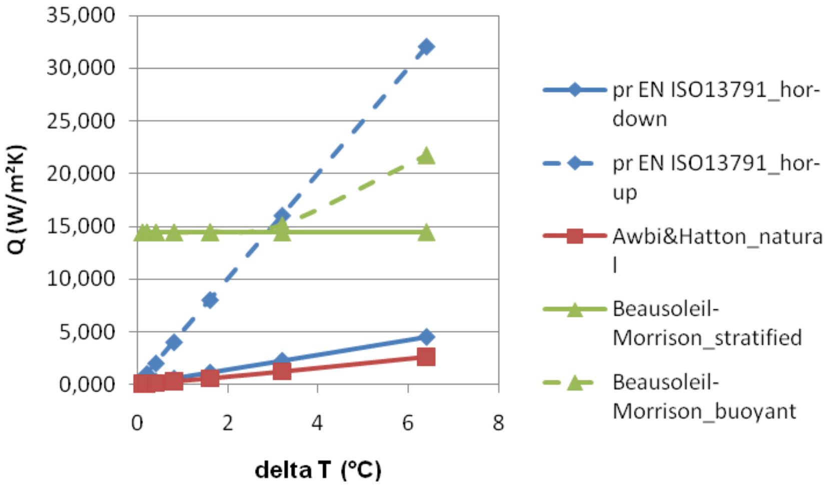


Figure 4. Flux to the ceiling surface ifo. temperature difference ceiling/air

Fig. 5. Demonstrates the influence of the choice of CHTC definition on the predicted performance of a control strategy. Beausoleil-Morrison is less influenced by the temperature difference and predicts relatively better performance for the more tolerant schemes, seen in the slightly flatter 'tale' of the results cloud.

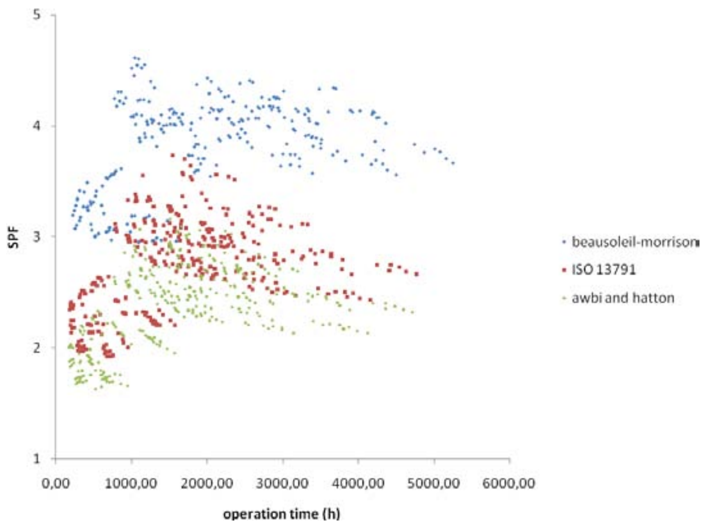


Figure 5. SPF for all simulations, according to the ISO standard, Awbi and Hatton and Beausoleil-Morrison. (x-axis: electrical load, y-axis saved active cooling load in kWh)

CONCLUSIONS

The case presented in this paper demonstrates that, to assess the feasibility of mechanical night ventilation, the choice of convective heat transfer coefficient definition can have a notable impact, due to the specific context created by mechanical night ventilation.

With the large airflow rate introduced by mechanical night ventilation, forced convection is introduced. The CHTC proposed by the ISO standard were deduced for heat transfer in to and from boundaries in the lab. Therefore, they don’t represent the heat transfer to and from the thermal mass of the case study building well, especially in the low temperature difference rage. Awbi and Hatton’s correlation for natural ventilation is affected by ill-matched context.

When forced convection is taken into account, the negative impact of tolerant control schemes on the efficiency of the system is significantly reduced. Thus, the choice of CHTC definition may have an important influence on the decisions taken during the design process.

The global performance (seasonal performance factor) of the mechanical night ventilation however, is, even with the most positive of the three assumed definitions, mediocre and within the range of conventional direct cooling (SPF 4). Whenever a mechanical night ventilation concept is considered, extreme care should be taken in minimizing specific fan power (required electrical energy per m^3 air), since this is the main influencing parameter with quasi-linear correlation of electric load and saved cooling demand

DISCUSSION

The results of this paper should not be generalized without paying attention to the context of the case. Since parameters for performance of the mechanical night ventilation are highly building specific, general conclusions on the concept can’t be deducted from 1 single case.

Moreover, the results presented clearly indicate the uncertainty of simulation results in this context. Numerical results should always be interpreted in relation to each other and never in an absolute way. The presented results nonetheless demonstrate that through sensitivity analysis, more general correlations and influences can be found, providing guidelines for the design process.

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