How to model a wall solar chimney?

Complexity and Predictability

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Summary

Natural ventilation has gained attention in recent times and it is an interesting method for ventilating buildings. The focus of this study is therefore the application of the wall solar chimney (WSC) system which is driven by the solar irradiation and buoyancy forces and which is used to generate natural ventilation in built environment. For the design and research purposes there are different modelling approaches available. However there is stated that in practice often complex modelling approaches – for instance CFD (Computational Fluid Dynamics) – are used for this real problems which efficiently can be modelled by much simpler models. Focusing on the complexity and the predictability of different modelling approaches this study aims to develop a basis guideline which helps a designer in selecting an appropriate modelling approach when designing or studying a WSC system. Therefore different modelling approaches - BES, BES+AFN (Airflow Networks) or CFD are used to fulfil this aim. The main research question with which a lot of designers and researchers are encountering is; “What is the appropriate modelling approach which provides reliable predictions on the performance of a WSC system?” The performance of the WSC system can be described by the following quantities: the incoming solar irradiation, the massflow or the volume rate, the outlet air temperature and the surface temperatures.

To be able to answer the question above a real-sized outdoor located naturally ventilated WSC system with an aspect ratio (Height/Depth) of 44 was used as experimental set-up which was located in Molenhoek the Netherlands. Experiments were performed by the consultancy company Peutz BV. The air-temperature and the air velocity to the chimney were controlled at respectively 21 °C and 1 m/s. The supplied air to the WSC system was conditioned to a certain temperature using an air-conditioning system which was positioned in an adjacent room near the WSC system. The velocity was controlled by an actuating damper which had a velocity sensor at the inlet of the chimney’s shaft. This experimental set-up – though it wasn’t primary aimed to provide such a measured data for the validation of different computer models – was yet used as the case study. Therefore different models based on BES, BES+AFN, CFD modelling approaches were pre-processed. For BES and BES+AFN and CFD respectively the software programs ESP-r and Gambit&Fluent 6.3 were used. Moreover, the measurements at a typical winter day were used as boundary conditions to different modelling approaches. Besides, for the BES+AFN modelling approach 3 different models were generated with different discharge coefficients ($C_d=0.42$ and $C_d=0.65$) and convective heat transfer correlations (Khalifa-Marshall and Alamadari-Hammond). Also, based on the CFD approach 3 different boundary resolutions were generated, in which for the simplest model - a simple open-ended rectangular cavity of 0.25 m depth and 11 m height - two different turbulence models were compared to each other (low-Reynolds k-epsilon and standard k-omega). According to the simple model in CFD the largest boundary resolution model of 100 meter width by 60 meter height was generated. This represented the environment and this model aimed to show the impact of the choice of the boundary resolution on CFD predictions.

This master thesis shows that the WSC system has the potential to be used as a natural ventilation system which generates certain stack pressure. A thermal efficiency of 61% was measured during a typical winter day. For this certain day it provided an averaged ventilation volume rate of 1622 m$^3$/h and a heating of passing air by 1 °C per meter height. Also, there was shown that the WSC system can be modelled using different modelling approaches with
different boundary conditions. The $C_d$ values for the BES+AFN models at outlet and inlet of the WSC system showed significant effects on the predicted volume rate and outlet air temperature while the convective heat transfer correlations did not affect the latter. This study also concluded that the choice of the boundary resolutions in CFD simulations has significant effects on the prediction of different parameters. Besides, the differences in the latter predictions showed a Rayleigh number depended character. Here the Nusselt number increased with increasing Rayleigh number. Furthermore, there was shown that the Bar-Cohen and Rohsenow correlation for the calculation of the averaged Nusselt number concerning a vertical parallel plate problem suited well to this particular WSC system problem. However this conclusion was based on only the CFD result, because no convective heat-fluxes were measured.

Finally, from a empirical validation study and a simple uncertainty analysis there has been concluded that CFD model with the largest boundary resolution (the cavity plus the environment) and a low-Reynolds k-epsilon turbulence model together with implemented convective heat-fluxes from BES+AFN models with a $C_d$ of 0.42 shows the best agreement with the measured data. Nevertheless further study is required in order to develop a basis guideline for the modelling of all the possible WSC system designs. The reason is that this study encountered significant limitations on the process of measurements, modelling and as a consequence limitations in the validation study. Also, the results are based on only one design of the WSC system and the influences of other design parameters (location, orientation, aspect ratio etc.) aren’t part of the study.

There is been recommended to perform more measurements on an experimental set-up with a free-floating configuration – so without controlled air velocity as it is for current case - and measure the vertical air velocity to generate a better understanding about the air volume rate inside the chimney. Also it is been emphasized that the outcome of this study is based on one design situation, and it is recommended to used the recommended modelling approach and simulate a completely different WSC design to judge the reliability of the recommended approach. Finally, for more research oriented studies this work expected that the coupling of BES+AFN and CFD might be a more accurate simulation approach.
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This research was already in stages of development when I started to get involve with the research topic of the PHD student whose name is Ben Bronsema. Ben is designing all sorts of sustainable solutions for built environment and one of his ideas is the design of a wall solar chimney system. Therefore I had - for some time - close collaboration with Ben’s research group, and have been visiting most of his project meetings. Here I was able to go through the whole process of designing of the wall solar chimney system and its experimental set-up. I was also involved in thinking about the position of the sensors and how things could be measured. Ben I wish you the best and I hope that your sustainable design will be a success in near future.

The experimental set-up was designed in collaboration with the consultancy company Peutz BV in Molenhoek (Netherlands). Here I met Harry Bruggema – a building physic consultant – who I would like to thank in this case for his friendly collaborations and quick responds to my questions with regard to the experimental set-up.

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1. Introduction

1.1 Background

1.1.1 Application of Wall Solar Chimney

Natural ventilation has gained attention in recent times and it is an interesting method for ventilating buildings. The fundamental principles of natural ventilation are stack effects (or pressures) and wind driven ventilation (Khan et. al, 2008).

The WSC (wall solar chimney) system has been in use for centuries, particularly in the Middle East, as well as by the Romans (Wikipedia, 2009). This system is a natural draft device, which utilizes solar radiation energy into the kinetic energy of air movement to build up stack pressure. This system consist of a surface (outer pane or glass cover) with glazing oriented towards the sun, a massive wall (inner wall with absorber at the inside) which may be a solid or liquid filled reservoir, an air channel (or cavity) in between and air ventilation ports for inlet and outlet air (Figure 1).

Chimneys or channels are further used in various other applications such as heating of buildings, drying of agricultural products, and various other passive systems such as for cooling electronic components. The simplest chimney may be vertical channels without any thermal mass. In any case, in all solar chimneys for heating and ventilation, heat...
transfer is usually conjugate, by convection, conduction and radiation, which should be studied simultaneously (Nouanégué et al., 2009).

1.1.2 Performance Indicators

In the design process of a WSC system, first the behaviour of the system is studied. The behaviour is represented by several quantities. The quantities which can be considered as the performance indicators of the WSC systems are: the massflow or the volume rate (respectively $\dot{m}$ or $\dot{V}$ in kg/s or m$^3$/s), the temperature difference over the inlet and the outlet air ($\Delta T_{\text{air}}$ in Kelvin) and the surface temperatures of the absorber and the glazing.

Knowing the latter quantities is important for calculation of the dimensionless value $TE$ (Thermal Efficiency). This value makes it possible to compare the efficiency of different designs of the WSC system. This relation is derived below (Equation 1).

$$TE = \frac{Q_{\text{out}}}{Q_{\text{in}}} = \frac{\dot{m} c_p \Delta T_{\text{air}}}{H W S_2}$$

Equation 1: The TE of the solar chimney is the ratio between the buoyant energy $Q_{\text{out}}$ and the incident solar energy $Q_{\text{in}}$ at the absorber surface

$TE$ is the thermal efficiency, and it is the ratio between the $Q_{\text{out}}$ (buoyant energy or the heating power gain in Watts) and $Q_{\text{in}}$ (incident solar power in Watts). $Q_{\text{out}}$ is equal to the multiplication of the massflow ($\dot{m}$ in kg/s), the specific heat ($c_p$ in J/kgK) and the air temperature difference between inlet and outlet of the chimney ($\Delta T_{\text{air}}$ in Kelvin) (Figure 2).

The amount of incidental irradiation that is transmitted by the outer pane (transparent glazing) and subsequently hits the absorber’s surface is called $Q_{\text{in}}$. $H$ and $W$ are respectively the height and the width of the absorber in meters. $S_2$ is the incidental solar irradiation on the absorber in W/m$^2$.

1.1.3 Modelling complexity

The modelling approaches for building simulation vary in a wide range of complexity (Hensen, 2002) and capabilities (Crawley et al., 2008). The simplest model (Bansal et al., 1993) is described by a few equations (empirical or semi-empirical equations) and the more complex one is CFD (Computational Fluid Dynamics) model (Poirazis, 2006).

Briefly, beside a simple hand calculation there are at least three modelling approaches in building simulation representing different levels of complexity from simple to complex (Djunaedy et al., 2004):

1. Building energy simulation models (BES) that basically rely on guessed or estimated values of airflow,
2. Zonal airflow network (AFN) models that are based on zone mass-balance and inter-zone flow-pressure relationships,
3. Computational fluid dynamics (CFD) that is based on energy, mass and momentum conservation in minuscule cells which make up the flow domain.

Also different commercial and non-commercial software packages are available on the market in which different modelling approaches are implemented. For example, BES and AFN are programmed in ESP-r, Trnsys and IES (Crawley et al., 2008). CFD is implemented in software packages like PHOENICS, CFX, Fluent and Comsol (CFD Wiki, 2010).

Some of these programs are open source codes, which means that the end-user is able to consult the code and adjust the program algorithm if necessary. For example, this is the case for ESP-r.

1.1.4 Boundary resolution

The system’s boundaries of the WSC system (Figure 1) can be considered as follows; the exterior, the interior and the cavity. The cavity considers the air volume, the absorber surface, the glazing surface and the openings at the bottom and the top. The interior is basically meant to deal with the effects of the adjacent building on the performance of the WSC system (for example; the thermal mass of the inner-pane, the air temperature in the building and the ventilation system). Finally the exterior boundary includes all the outdoor influences on the performance of the WSC system (for example; the sun’s irradiation, the wind).

1.1.5 Literature overview

Since early 1970s, the chimney systems combining a thermal mass have been studied experimentally (e.g. Bansal et al. (2005)), analytically (e.g. Ong and Chow (2003)) and numerically (e.g. Kim et al. (1990), Burch et al. (1985) and Bilgen and Yamane (2004)). Among the numerical studies, the conjugate heat transfer by convection and conduction has been considered in Ben Yedder and Bilgen, 1991; Kim et al., 1990; Burch et al., 1985; Bilgen and Yamane, 2004, that by convection and radiation in Moshfegh and Sandberg, 1996; Bouali and Mezrhab, 2006; Cadafalch et al., 2003, and by all three modes in Lauriat and Desrayaud, 2006; Hall et al., 1999; Rao, 2007 (Nouanégué et. al, 2009). But these researches focus mainly on WSC systems which were placed in a well controlled environment (indoor experiments with controlled irradiation and incidental angle) or on chimneys with small aspect ratio (height/depth).

Furthermore, there is a broad literature overview on the advantages and disadvantages of different modelling approaches used to model double skin facades (DSF) during the design process. This work was done for IEA ANNEX 43 Task 34 (Kalyanova et. al, 2005). However - in contrast to a DSF modelling approach - the WSC system has different physics and it also contains an opaque inner-pane which might give preference to different modelling approaches.
Gan, G., 2010) also studied solar chimneys. In his work CFD simulations are used to show the effect of the boundary resolution on ventilation cooling using a WSC system. He concluded that the ventilation rate and the heat transfer coefficients in CFD simulations depend not only on the cavity size and the quantity and proportion of the heat distribution on the cavity walls but also on the boundary resolution. Although the conclusions here might be of use in the current study, the aim here was not to guide a designer (in a design process) to choose an appropriate modelling approach.

From the above literature survey one can conclude that there is been a broad and comprehensive modelling and experimental study on WSC systems. Also, there are many modelling approaches available during the design process of a WSC system each with their advantages and disadvantages. However, until now there has never been an attempt to generate a basis guideline in selecting an appropriate modelling approach during a design process of a WSC system.
1.2 Problem definition

“It can scarcely be denied that the supreme goal of all theory is to make the irreducible basic elements as simple and as few as possible without having to surrender the adequate representation of a single datum of experience”, Einstein quotes (Wikiquote, 2010). Other interpretation of what Einstein said should be the bottom line of any modelling approach, which means that models must be as simple as possible but not simpler. Nevertheless, in practice often complex modelling approaches – for instance CFD (Computational Fluid Dynamics) – are used for real problems which efficiently can be modelled by much simpler approaches (Hensen et. al, 2006).

When one wants to predict a certain Performance Indicator (PI) of a Wall Solar Chimney (WSC system) first an important question has to be answered: “what is the appropriate modelling approach to solve this real problem?” According to Djunaedy et al. this is just the challenge every time one chooses a modelling approach in order to solve a real problem (Djunaedy et. al, 2004).

The latter problem has been issued by several researchers. A well practical explanation about making a model of reality can be found in the book of Rodger T. Fenner: “choosing a suitable model for a system is a matter of making reasonable assumptions in order to simplify the real system far enough to permit it to be analyzed without an excessive amount of labor, but without at the same time simplifying it so far as to make the results of the analysis unreliable for design and other purposes” (Fenner, 2000).

Focusing on the complexity and the predictability of different modelling approaches this study aims to develop a basis guideline which helps a designer in selecting an appropriate modelling approach when designing a WSC system. The main question is: “What are the appropriate modelling approaches for each PI of WSC-system?”. This question can’t be fully answered if it is not supported by the following sub-research questions: “What is the minimum modelling complexity which is necessary to simulate a WSC system in a physically proper way?. “What are the uncertain input parameters to different modelling approaches and what are their influences on different PI’s of the WSC system?”.

To come to reasonable conclusions this work focuses on the outcome of a validation study in which the results from different modelling approaches are compared internally as well as to the results from a real sized outdoor experimental set-up.

In the next section one can read how the report is been constructed.
The content of this master thesis is constructed as follows;

In Chapter 2 some elementary heat transfer theory is explained regarding the wall solar chimney. Next, the background about the BES, BES+AFN and CFD modelling approaches is clarified.

In Chapter 3 first the experimental set-up of a real-sized outdoor WSC (Wall Solar Chimney) system is explained. Secondly, the construction, the working principle, the position of sensors is shown here using real pictures and schematic drawings. Next, the results of series of measurements on 15-12-2009 are shown in a separate section. Finally, this chapter will end up with discussions on the latter results followed by conclusions.

Chapter 4 focuses on the development of models in ESP-r (BES and BES+AFN) and in Gambit & Fluent (CFD). In this chapter different modelling approaches are generated. Next for each modelling approach the necessary information about the boundary condition and the assumptions to develop these models are shown. A combination of text, figures and schematic drawings will clarify the situation at hand. Finally, this chapter will end up with discussions on the latter models followed by conclusions.

In Chapter 5 first the results from the experiments and simulations are compared to each other, which is actually the validation study. Next, a simple uncertainty analysis on the discrepancies between the model predictions and measured data is performed and these results are also shown here. Finally, this chapter will end up with discussions on the latter models followed by conclusions.

In Chapter 6 first there is a general conclusion about whether this study could give answers to the research questions. Finally this chapter will end up with some recommendations for further study on the WSC system.
2. Theory

2.1 Heat transfer

2.1.1 Wall Solar Chimney system

The study of WSC system is a situation for which there is no forced motion, but heat transfer occurs because of convection currents that are induced by buoyancy forces, which arise from density differences caused by temperature variations in the fluid. Heat transfer by this means is referred to as free (or natural) convection.

Figure 2 shows the heat transfer behaviour of a WSC (Wall Solar Chimney) system. Air enters the chimney at the inlet with temperature \( T_{i} \) which is assumed equal to the air temperature of an adjacent environment \( T_{a} \). Warm air exits at the outlet \( T_{o} \) from the top of the chimney. Temperatures at the internal surfaces of the glazing \( T_{g} \) and wall \( T_{w} \) and mean air temperature of the flow channel \( T_{f} \) depend all on the \( S_{i} \), the incident solar irradiation which is the main driving force. The heat transfer happens consequently by convection for inside surfaces \( h_{w} \) and \( h_{g} \) and outside surfaces \( h_{\text{wind}} \), long wave radiation by inside surfaces \( h_{\text{rg}} \) and outside surfaces \( h_{\text{rs}} \), and finally by conduction through the back wall \( U_{b} \) and the glazing \( U_{t} \) (Ong, 2003). All temperatures \( T \) are indicated in Kelvin, heat transfer coefficients \( h \) in W/m\(^2\)K and all the heat fluxes \( S \) or \( U \) in W/m\(^2\).

![Figure 2: The heat transfer behaviour of a wall solar chimney (Ong, 2003)](image-url)
2.1.2 Volume rate

The air volume rate across the chimney can be expressed (Bansal et. al, 2003):

\[ \dot{V} = C_d \frac{A_o}{\sqrt{1 + \frac{A_o}{A_i}}} \sqrt{\frac{2gL(T_f - T_a)}{T_a}} \]

**Equation 2: The analytical method to calculate the volume rate through the WSC system (Bansal, 1991)**

Here is \( C_d \) the system’s discharge coefficient, \( A_o \) is the outlet area (m\(^2\)), \( A_i \) chimney’s inlet area (m\(^2\)), \( g \) is the gravitation acceleration (m/s\(^2\)), \( L \) is the chimney’s height (m), \( T_i \) is the mean temperature of air in the channel (Kelvin) which is described by the equation below and finally \( T_a \) is the inlet air temperature (Kelvin). The constant \( \gamma \) is meant for the mean temperature approximation which depends on the inlet and the outlet air temperature.

\[ T_f = \gamma T_{f,a} + (1 - \gamma)T_{f,i} \quad \text{with } \gamma = 0.74 \]

**Equation 3: The averaged flow temperature over the total height of the chimney (Bansal, 1991)**

2.1.3 Internal convective heat transfer

The table below shows the different regimes of convection for which there are semi-empirical and empirical correlations (Beausoleil-Morrison, 2001).

<table>
<thead>
<tr>
<th>convective regime</th>
<th>driving force</th>
<th>cause of driving force</th>
</tr>
</thead>
</table>
| A                 | Buoyant       | Surface-to-air temperature difference, caused by one of the following: 
|                   |               | * heat transfer through the external envelope; 
|                   |               | * solar insulation to walls or floor (i.e. sun patch); 
|                   |               | * in-floor heating; 
|                   |               | * unheated panels; 
|                   |               | * heated walls (e.g. hydronic wall panels). |
| B                 | Buoyant       | Heating device (e.g. radiator, stove) located within room |
| C                 | Mechanical    | Air handling system (central or zonal) delivering supply of heated or cooled air to room through ceiling, floor, or wall-mounted diffusers. Exhaust air mechanically extracted or infiltrated. |
| D                 | Mechanical    | Heating or cooling device with circulating fan. No intentional supply or extract of air from room. |
| E                 | Mixed flow (mechanical and buoyant) | Mechanical force caused by air handling system (central or zonal) delivering supply of heated or cooled air to room through ceiling, floor, or wall-mounted diffusers. Buoyant force caused by surface-to-air temperature differences (as described above). |

Table 1: Classification of different convective regimes (Beausoleil-Morrison, 2001).

The internal convection can be divided in different convective regimes. For the WSC system which is derived by free convection the convective regime A heated walls (by solar irradiation) is of concern. Many convective heat transfer correlations exist, but none of them are universal. Some are general in nature while the applicability of others is restricted
to specific building geometries and HVAC systems (Beausoleil-Morrison, 2001). So, there is no specific correlation which is design only for WSC systems.

In addition, there are at least 4 different correlations which are used in simulation programs like ESP-r. The Alamdari-Hammond method, the Khalifa method, Awbi-Hatton method and finally Fisher method are correlations that can estimate the \( h_w \) and the \( h_g \).

2.1.4 External convective heat transfer

Also, there are different correlations which estimate the amount of convective heat flux to the exterior. For example McAdams equation (\( h_{wind} = 5.7 + 3.8v \) with \( v \) the local wind velocity) includes the effect of wind in the convective heat loss to the exterior (Clarke, p. 245).

2.1.5 Nusselt and Rayleigh number

According to existing literature on heat transfer the physical problem of the WSC system based on only natural ventilation may be considered as a free convection problem within ‘Parallel Plate Channels’. Surface thermal conditions may be idealized as being isothermal or isoflux and symmetrical or asymmetrical (Incropera, p. 548).

Among other researchers Bar-Cohen and Rohsenow describe several empirical correlations in order to calculate the Nusselt number for such a rectangular channel problem (Lu et. al, 2010) (See Table 2).

<table>
<thead>
<tr>
<th>Researchers</th>
<th>Correlations and description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rohsenow et al. [15]</td>
<td>( \text{Nu}_{\text{min}} = 0.29 (Ra^<em>)^{1/2}, Ra^</em> \leq 5 )</td>
</tr>
<tr>
<td></td>
<td>( \text{Nu}_{\text{plate}} = \frac{h_i}{Pr} (Ra^<em>^{1/5}) ), ( 10^2 \leq Ra^</em> \leq 10^4 )</td>
</tr>
<tr>
<td></td>
<td>( \frac{h_i}{Pr} = \frac{1}{\pi} \left( \frac{6 + \sqrt{3}}{4 - \sqrt{3}} \right)^{1/4} )</td>
</tr>
<tr>
<td>Ramanathan and Kumar [16]</td>
<td>( \text{Nu} = \left[ \left( \frac{\text{Nu}<em>{\text{air}}}{\text{Nu}</em>{\text{m}}(Ra^*)^{1/3}} \right) \right]^{1/3} )</td>
</tr>
<tr>
<td></td>
<td>( m = -3.5 ) ( \leq 1 ) ( Ra^* \leq 3 \times 10^4 ), ( \text{IT} = 0.7 )</td>
</tr>
<tr>
<td>Lu [17]</td>
<td>( \text{Nu} = \frac{24}{(1 + 1.25^{Ra^<em>})^{3/4}} ) ( \frac{7.506}{(1.25)^{Ra^</em>}} ) ( \frac{1}{\sqrt{Ra^*}} )</td>
</tr>
<tr>
<td>Bar-Cohen and Rohsenow [18]</td>
<td>( \text{Nu} = \left( \frac{17}{Ra^<em>} + \frac{1.88}{(Ra^</em>)^{1/3}} \right)^{1/2} ), all ( Ra^* ) range</td>
</tr>
</tbody>
</table>

Table 2: The existing empirical correlations for calculation of the Nusselt number for parallel plate channel problems (Lu et. al, 2010)

The Nusselt number provides a measure of the convection heat transfer occurring at the surface. (Incropera, p. 365).

The convective heat transfer rate can be assessed according to the local heat transfer coefficient or the local Nusselt number which is defined as \( \text{Nu} \), where \( h_c \) (W/m²K) is the convective heat transfer coefficient, \( T_w \) (Kelvin) is the averaged temperature of the local
absorber and glazing surface temperatures, $T_a$ (Kelvin) is the local air temperature, $d$ is the depth (m) of the chimney, $q^*_{abs}$ is the convective heat flux at the absorber ($W/m^2$), $q^*_{glz}$ is the convective heat flux at the glazing ($W/m^2$) and $k$ (W/mK) is the air conductivity.

\[
Nu = \frac{h_c d}{k} = \frac{\left(\frac{q^*_{abs} + q^*_{glz}}{2}\right)}{(T_w - T_a)k}
\]

Equation 4: The local Nusselt number (Gan, 2010)

The Rayleigh number ($Ra_q$) based on the cavity width and the total heat flux indicates the relative magnitude of the buoyancy and viscous forces in the fluid. This is the product of the Grashof (Gr) and Prandtl (Pr) numbers (Incropera, p. 533). In the equation below $\beta$ is for air the volumetric thermal expansion coefficient (1/Kelvin), $\nu$ is the kinematic viscosity ($m^2/s$) and $\alpha$ is the diffusion coefficient ($m^2/s$).

\[
Ra_q = \frac{g \beta \left(\frac{q^*_{abs} + q^*_{glz}}{2}\right) d^4}{\nu \alpha k}
\]

Equation 5: The local Rayleigh number (Gan, 2010)

The above correlations (Table 2) for the Nusselt number enable the calculation of internal convective heat transfer coefficients and thereby the amount of convective heat flux which is gained ($Q_{out}$ in Equation 1) by the air moving in the WSC system.

\[
\beta \approx -\frac{1}{\rho} \frac{\Delta \rho}{\Delta T} = -\frac{1}{\rho \Delta T} \approx \frac{1}{T} - \frac{1}{T_{\infty}}
\]

Equation 6: The volumetric thermal expansion coefficient (Incropera, p. 528)

The volumetric thermal expansion coefficient shown above is a simplification using the Boussinesq approximation (Incropera, p. 528).

For more information on convection, radiation and conduction heat transfer the reader is referred to (Incropera).
2.2 BES

BES (Building Energy Balance) is a thermal energy model which is implemented in ESP-r software program. The BES in ESP-r is based on the numerical discretization and simultaneous solution on heat-balance methods. ESP-r simulates the thermal state of the WSC system by applying a finite-difference formulation based on a control-volume heat-balance to represent all relevant energy flows. More information on this method is found in (Beausoleil-Morrison, 2000). This approach is explained using Figure 3.

Figure 3: The heat transfer in two thermal zones is shown here.

Every construction and air volume will be described using representative thermal nodes. Next, these nodes are connected to each other using a heat balance approach. Simultaneously, the governing partial differential equations for every node will be solved to hold a thermal equilibrium between the zones and the surroundings (Beausoleil-Morrison, 2000).

In this dynamic modelling approach the airflow is not simulated, merely its impact is considered in the thermal simulation. It uses flow rates which are user-prescribed or estimated using simplified approaches. The impact of these air flows are implemented by assuming a fixed massflow for infiltration or ventilation. Also, the optical properties of glazing, the condition of air at the chimney's inlet and the convective heat transfer (in this case a fixed value) at different walls can be set here (Clarke).
2.3 BES +AFN

In the previous section the BES approach is explained. Yet, in ESP-r the BES approach can be combined with the so called AFN (Air Flow Networks) approach. The latter approach is based on the assumption that a building and/or plant can be considered as being composed of a number of zones or nodes (e.g. rooms, plant components) which are linked by connections (e.g. openings, cracks, ducts, pipes). Moreover a nonlinear relationship exists between the flow through a connection (air flow component) and the pressure difference across it (Hensen, 1991).

Conservation of mass for the flows into and out of each node leads to a set of simultaneous nonlinear equations which are solved (Hensen, 1991).

The figure below shows how such an approach works. According to Beausoleil-Morrison (Beausoleil-Morrison, 2000) 4 steps are involved:

1. The building is discretized by representing air volumes (usually thermal zones) by nodes (1 to 4). Nodes are also used to represent conditions external to the building (outdoor boundary nodes).
2. Components (red springs) are defined to represent leakage paths, and pressure drops (pressures losses) associated with doors, windows, supply grills, ducts, fans, etc.
3. The nodes are linked together through components to form connections (shown with double headed arrows in the figure), which establishes a flow network.
4. A mass balance is expressed for each node in the building. The resulting system of equations is solved to yield the nodal pressures and the flows through the connections.

Figure 4: Air flow network: nodes and connections (Beausoleil-Morrison, 2000). The circles are the airflow volume nodes, the red Z is the airflow components and the arrows are the connections between the airflow volume nodes.

For more information about the airflow networks the reader is referred to (Hensen, 1991). For information about the background of CFD the reader is referred to (Kan et. al, 2008).
3. Experimental set-up

3.1 Introduction

The starting point in this study is the real-sized outdoor experimental set-up of the WSC (Wall Solar Chimney) system which will be used to provide measured data on PI's (Performance Indicators) and BC's (Boundary Conditions). These data will be used to validate the results of different modelling approaches as well as to investigate the importance of some parameters on the performance of such a system.

3.1.1 Background

This research is performed in cooperation with the Eindhoven University of Technology, Technical University of Delft and the consulting company Peutz BV which is called “Earth Wind and Fire” group (Brugemma, 2009). EWF project is sponsored by SenterNovem (Agentschap NL). This organization is an agency of the Ministry of Finance in the Netherlands.

3.1.2 Aim

Primary the aim of this experimental set-up is to verify the concept of the sustainable WSC system. The measurements will be carried out by Peutz BV. The current study will further use this data for computer model validation.

The purpose of the current study is to judge the validity of different modelling approaches compared to a real situation. However the primary aim of the experimental set-up is verifying a WSC concept and not providing valuable data of BC’s and PI’s for model validation. Therefore it is obvious that there are certain experimental limitations as well as uncontrolled situations that affect the purpose of model validation. Nevertheless because no other valuable measured data was found during the literature overview, current study attempts to use this experimental set-up as its validation case.
3.2 The set-up

3.2.1 Location

The real-sized outdoor experimental set-up is located in Molenhoek in the Netherlands and its glazing construction is oriented to the south (Figure 5). This location can precisely be described by the following data (Source: Google Earth):

- Latitude: 51°45'38.27"N
- Longitude: 5°52'29.92"E
- Ground level: 22 m

3.2.2 Construction

The inner dimensions of the shaft (the volume for air) of the system are prescribed by the Width (W), Depth (D) and Height (H) of the shaft and these are respectively 2 m, 0.25 m, and 11 m (Figure 5). Besides, the ratio of the inlet and the outlet area is the unity.

Figure 5: The experimental set-up is shown here (Brugemma, 2009).

The inner-pane and the side walls are constructed from a low-emissivity aluminium absorber plate (ε=0.05) and it is well insulated (Rockwool 433; λ=0.035 W/mK; d=240 mm) (Table 7 1). The southern facade of the experimental set-up is constructed of 75% (15.7 m²) of transparent double glazing construction (U-value=1.32 W/m²K; G-value=0.7) (Appendix1: Table 7&8).

Furthermore, since the optical properties of the glazing construction are a function of the incident solar irradiation, different angular optical properties (absorbance, reflection and transmission) can be found by using the WINDOW 5 software (see WINDOW). This is a database of the glazing manufacturers. The latter information is shown in Figure 6.
Figure 6: Performance of the glazing construction used in the experimental set-up which is determined by using WINDOW 5 software.

The horizontal axis is the degree of solar incident normal to the glazing surface. The vertical axis shows the values for the visible transmission, the solar direct transmission, the solar reflection and the solar absorption of the glazing construction.

### 3.2.3 Working principle

Table 3 shows possible types of measurements. These series are meant to be carried out during a period of three years starting from October 2009. The working principle of the experimental set-up is explained using Figure 7. All the measurements are said to be performed during almost wind-still outdoor situation in order to avoid the possible influences of wind on the system.

With this experimental set-up two different flow regimes can be simulated; natural ventilation (CASE A&B) and mechanical ventilation (CASE C). For the purpose of this study only CASE B (hybrid configuration) is of concern.

<table>
<thead>
<tr>
<th>CASE A</th>
<th>CASE B</th>
<th>CASE C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type ventilation</td>
<td>Natural</td>
<td>Hybrid</td>
</tr>
<tr>
<td>Air-Control</td>
<td>Free-floating</td>
<td>Constant velocity</td>
</tr>
<tr>
<td></td>
<td></td>
<td>1 m/s</td>
</tr>
<tr>
<td>Air-conditioning</td>
<td>No</td>
<td>Yes</td>
</tr>
<tr>
<td>Chimney's Inlet air temperature</td>
<td>Ambient conditions</td>
<td>25 °C (Summer)</td>
</tr>
<tr>
<td></td>
<td>and</td>
<td>and</td>
</tr>
<tr>
<td></td>
<td>22 °C (rest of the year)</td>
<td>22 °C (rest of the year)</td>
</tr>
<tr>
<td>Outside conditions</td>
<td>Clear sky or</td>
<td>Clear sky or</td>
</tr>
<tr>
<td></td>
<td>Cloudy sky</td>
<td>Cloudy sky</td>
</tr>
</tbody>
</table>

Table 3: This is an overview of the measurement plan according to the provided report (Brugemma, 2009).
A vertical cross-section of the experimental set-up is shown in Figure 7. The inlet (point A) will divide the air over the openings B (mechanical fan) or C (hydraulic opening) depending on the measurement type (Case A, Case B or Case C). When the air is in the room, again depending on the measurement type the air will be conditioned to a certain temperature ($T_{\text{set}}$) (Table 3) (for Case B and Case C) or not conditioned as it is for Case A. Hereafter the air is ready to enter the shaft at point F, which again will raise up in direction of point H by the natural convection (in Case A&B) or forced convection (Case C). The absorber at the inside of the inner-pane will mostly warm up the entering air. Due to the stack effect of air in the shaft there is a negative pressure difference (CASE A&B) over point F and point H and as the consequence the air will flow towards the outlet openings (points I and J). In CASE C the gauge pressure of the fan (B) will be the driving force of air in the shaft.

![Figure 7: The working principle of the experimental set-up according to Table 3 is shown here.](image)

Note that all measured parameters are registered using a 96-channel datalogger coupled to a PC with maximum 32 analog channels and 64 temperature channels. Every 10 minutes the data which is registered per minute will be averaged over 10 minutes and subsequently saved in an Excel-sheet.

### 3.2.4 Measured parameters

In the shaft the $V_Y$ (the absolute vertical air velocity), $T_{\text{air}}$ (Air-Temperature), $T_{\text{abs}}$ (temperature of absorber) and $T_{\text{glz}}$ (temperature of glazing) are measured over 4 levels; on 0.5 m, 4 m, 7.5 m and 11 m (Figure 9) with the same distribution as in Figure 8.
There are 9 thermocouples for air temperature measurements (red circles Figure 8). Three thermocouples (green rectangular) are placed which are meant to measure the surface temperature of the absorber. Furthermore, there are two velocity anemometers (blue triangle) to measure the vertical air velocity. Finally the glazing temperature is measured on 2 different positions (black rectangular).

For each height there are 9 thermocouples for the air temperatures (error ± 0.5 °C) (Figure 9 E), 3 thermocouples for the inside surface temperatures of the absorber (Figure 9 G) (error ± 0.5 °C), 2 thermocouples for the surface temperature of the glazing and frame (error ± 0.5 °C) (Figure 9 F) and 2 velocity anemometers (error ± 5%) for the vertical air velocity measurements. Cylindrical and aluminium foil are used to minimize the influence of radiation on respectively the air and the surface temperature sensors.

![Figure 8: The distribution of the sensors in shaft's horizontal cross-section is shown here.](image)

Other parameters are also measured per 10 minute interval (Figure 9). The diffuse horizontal solar irradiation (Figure 9 A) and the outside and inside vertical solar irradiation (Figure 9 F) at the glazing surface on 4 meter height use solarimeters to measure the solar intensity (W/m²). Also, the wind velocity at 2 meter (Figure 9 B) and 11 meter height (Figure 9 B; together with wind direction) are measured. At the chimney’s inlet – at 0.25 meter height – the inlet air temperature to the shaft, the air humidity (also at the outlet on 11m height) and the air velocity are measured (Figure 9 H). The latter air velocity is used to control the hydraulic opening in CASE B and the fan’s revolutions per minute in CASE C.

For more detail information on the experimental set-up the reader is referred to Memo 4, versie 5 (Brugemma, 2009).
Figure 9: This is an overview of what is measured and on which levels these measurements are performed. Note that 0.5 m, 4 m, 7.5 m and 11 m height have the same distribution as shown Figure 8.
3.3 Results

The measurements results are shown for 15-12-2009 between 9:00h until 16:00h based on the CASE B working principle (3.2.3.). This period is chosen because of the quality of the measurements relative to other measurement series. For example the amount of the direct solar irradiation and its duration was better compared to other measuring days. The experiments started around 8:00h and a stationary situation was reached after approximately one hour. Therefore results are shown from 9:00h and the measurements were stopped around 16:00h.

3.3.1 Solar irradiation

$S_{Y,\text{OUT}}$ is the vertical solar irradiation at the glazing surface outside (blue line), $S_{Y,\text{IN}}$ is the vertical solar irradiation behind the glazing surface inside (black line) and $S_{\text{DIFF}}$ is the diffuse horizontal solar irradiation. The $S_{Y,\text{OUT}}$ and $S_{Y,\text{IN}}$ are measured at 4 meter height from the ground. $S_{\text{DIFF}}$ is measured on a position with a clear sky view and free of obstacles. See Figure 9 for the exact position of the sensors.

![Figure 10: Solar irradiation measured at different positions during the experiments on 15-12-2009 between 9:00h to 16:00h using solarimeter sensors. Along the horizontal axis the time (in hours) of the measurements is set against the vertical axis which is the solar irradiation (in W/m²).](image)

3.3.2 Air-Temperature

The air temperatures are measured using 9 thermocouples at 4 different heights (0.5 m, 4 m, 7.5 m and 11 m). Figure 11 shows the averaged values per height for the positions on 0.5 m, 4 m, 7.5 m and 11 m. Furthermore, $T_{\text{ambient}}$ is the outdoor temperature.
which is conditioned (Point E, Figure 7) is assumed to have the same temperature as $T_{air}$ at 0.25 m height (Point F).

Next, the air temperature distribution along the width (Figure 12) and along the depth (Figure 13) at 11 meter height is considered. Here, the averaged value of measured air temperatures in 3 points along the width or along the depth is shown. For instance, for the back the averaged value of 3 points is shown and this is called "$T_{air, 11m back}$" in Figure 13.

Figure 11: Averaged air temperatures per height (0.25 m, 4 m, 7.5 m and 11 m). Besides the air is conditioned – according the measurement plan CASE B – to approximately the same temperature as $T_{air}$ at 0.25 m (Figure 7). Also, the outdoor air temperature ($T_{ambient}$) is shown here.

Figure 12: Air temperature distribution along the width (2 m) at 11 meter height. The measured 3 point along the lines of Left, Middle and Right are averaged and shown here.
3.3.3 **Surface temperature of absorber**

Figure below shows the temperature of the absorber at 4 different heights during the day. These temperatures are measured at one point according to the position shown in graph below.

![Graph showing surface temperature of absorber](image)

**Figure 14:** Surface temperature of the absorber measured using one thermocouple approximately at Middle of the width at 4 different heights (0.5 m, 4 m, 7.5 m and 11 m).
Figure 15 shows the surface temperature distribution at 11 meter height over 3 different positions. The temperature at the absorber, the surface temperature of the side walls at the left and the right are shown here.

**Figure 15:** Surface temperature of the absorber ($T_{abs\ 11m\ Middle}$), the left wall ($T_{abs\ 11m\ left}$) and the right wall ($T_{abs\ 11m\ left}$) at 11 meter height. The exact positions are described by the red circles; dotted circles for the side walls and line circle for the sensor in the Middle.

### 3.3.4 Temperature of the glazing construction

Figure 16 shows the glazing temperature at 3 different heights (4 m, 7.5 m and 11 m) over a period of 7 hours. The position of the thermocouples is the same for every height and this is shown by the red lined circle.

**Figure 16:** Glazing temperature over 3 different heights. The values are for 4 m, 7.5 m and 11 m height and the exact position of these sensors is shown using the red lined circle.
Also the temperature difference over the glazing is shown (Figure 17). This is done for 3 different heights on the position which is indicated by the red lined circle.

![Temperature difference over the glazing at 3 different heights (4m, 7.5m and 11m). The exact position of these measurements is shown using the red lined circle.](image1)

Furthermore, the temperature of the frame of the glazing construction for 3 different heights is shown in Figure 18. The exact position of these sensors is also shown by the red lined circle.

![Surface temperature measurements of the frame of the glazing construction for 3 different heights (4 m, 7.5 m and 11 m). The exact position of these sensors is shown using the red lines circle.](image2)

### 3.3.5 Vertical air velocity

The vertical air velocity is measured by monitoring 2 points at each height (0.5 m, 4 m, 7.5 m and 11 m). Figure 19 shows the air velocity which is measured at 11 meter height on the left (V<sub>Y</sub> Left) and the right side (V<sub>Y</sub> Right) of the shaft, as indicated by the red lined circles.
3.3.6 Wind

In Figure 20 the wind velocity at 2 different heights (2 m and 11 m) is shown. Also the direction of the wind – measured at 11 m - is shown in the same graph (right axis).

At the same time the figure below shows the measured $C_p$ based on the Equation 7. $C_{id}$ (d for direction and i for surface) with the local surface pressure $P_{id}$ (N/m$^2$), air density $\rho$ (1.2 kg/m$^3$) and a reference wind speed $v_r$ (m/s) (corresponding to direction d (Clarke, p. 126).
Equation 7: This is the relation of the dimensionless surface wind pressure coefficient $C_{id}$.

\[
C_{id} = \frac{P_{id}}{\frac{1}{2} \rho V_r^2}
\]

Figure 21: Calculated wind pressure coefficients ($C_p$) for the outlet towards south (180°) as function of the wind direction (North=0 and clockwise). See Figure 7.

3.3.7 Discharge coefficient

According to Equation 2 the $C_d$ was calculated. These calculations are done using the measured inlet and outlet temperatures and the vertical air velocity (Standard Deviation = ± 14%; See Appendix 2). Furthermore the following is assumed: $g=9.81$ m/s$^2$, $L=11$ m, $\frac{A_o}{A_i}=1$. $C_d$ should be below unity (Daugherty et. al, 1965, pp. 338-349).

Figure 22: $C_d_{avg}$ based on the volume rate ($V$) calculated from the averaged velocity per time step (Figure 19) multiplied by 0.5 m$^2$ cross-sectional area. $C_d_{min}$ and $C_d_{max}$ are based on the SD of ± 14% (See Appendix 2).
3.4 Discussion

3.4.1 Thermal performance WSC

From the measured data SY_IN (Figure 10&Figure 11&Figure 19) and Equation 1 for a typical winter day an average daily thermal efficiency of 61% was found (Equation 1). The averaged air temperature is a function of the height as can been seen in Figure 11. The outlet air temperature at 11 meter height reaches a maximum value of 33.5 °C (± 0.5 °C) on 13:20h. Compared to the inlet air temperature, a temperature difference of 11.6 °C (± 0.5 °C) was created. This is approximately 1 °C / meter height for this particular situation.

3.4.2 Flow direction

The possibility to observe the direction of the flow by the existing velocity sensors wasn’t possible. Therefore a smoke test was performed; however this single smoke test can’t be conclusive for all the cases. From the measured results – especially at the start of the day between 9:00h to approximately 10:00h (Figure 11) - one can conclude that the air temperature at the top is lower than the temperature of the air entering the chimney. However this can be seen as a cooling down effect rather than a reversed flow, because as one might observe - the absorber surface temperature and the glazing/frame surface temperature are decreasing along the height (Figure 14&Figure 17&Figure 18).

3.4.3 Volume rate

As already is shown in Figure 8&Figure 19, two single points were measured to monitor the vertical air velocity. However this distribution isn’t accurate enough to be able to describe the volume rate in the chimney by using Equation 8. To be able to calculate the volume rate (V in m³/s) the area weighted average of velocity (∑vi / i in m/s) has to be multiplied to the total cross-sectional area (A_total in m²) of the shaft. The letter i indicates the number of sensors used to measure the vertical velocity distribution in the cross-section.

\[ V = \frac{\sum v_i}{i} \times A_{total} \]

Equation 8: This is the relation between the area weighted average of the measured vertical air velocity at certain height of the chimney and the volume rate.

It is obvious that additional distribution of velocity anemometers is still needed to represent an area weighted average of the velocity. Therefore extra measurements need to be carried out (Appendix 2). The vertical air velocity in the shaft was measured at two points. Based on extra measurements, average value of the vertical air velocity in the shaft was calculated according to Appendix 2 (Equation 9). It is assumed that the measured \( v_{y, right} \) (Figure 19) is 114% of the averaged value \( v_{avg} \) in reality. Thus, the minimum value of \( v_y \) is 86% of the averaged vertical air velocity, because of the SD of 14%. Although a Standard
Deviation of 14% is calculated, it must be emphasized that these values are based on different weather conditions (30-04-2010). The volume rate per hour can be now calculated by Equation 8.

\[
v_{\text{avg}} = v_{\text{y, right}} \left( \frac{1 + \frac{8e}{77}}{2} \right) \text{ m/s} (\pm 5\% \text{ m/s})
\]

Equation 9: This equation is based on the information (Appendix 2) in order to calculate the real volume rate in the shaft.

3.4.4 Wind

The experiments should be performed during a period in which wind’s influence on the system could be neglected (almost wind-still). However as one can see in Figure 20 there is wind. Unfortunately at the day of measurement (15-12-2009) – the wind direction was mainly between 40° and 120°; so information between 120° and 40° (clockwise) is missing (Figure 21). Also, the local pressure which is measured in this case is based on one single point and this point should represent the local pressure at the outlet towards south (Figure 7). At the same time the wind pressure at the outlet – towards north – is not measured. Accurate determination of the \(C_p\) as function of wind orientation - for the openings in contact with the outside world - is missing. So, for example the accidental decrease of the temperatures (Figure 11&Figure 18) at 13:50h and at the same time an increase of the absolute vertical air velocity (Figure 19) might be due to the wind. In other words at the moment there isn’t enough evidence to conclude that the temporary distortion of flow inside the shaft is because of the wind. To be able to consider the influence of wind on the performance of WSC system one needs to know exactly the \(C_p\) (wind pressure coefficient) of the inlet and the outlets of the system. Its prediction requires information on the prevailing wind, its speed, direction, vertical wind velocity profiles, the influence of local obstructions and terrain features. The two approaches to determine the surface pressure distribution are: wind tunnel test applied to scale models, and mathematical models (Clarke, p. 127). A comprehensive overview of the available tools to predict \(C_p\) is studied by (Cóstola et. al, 2009).

3.4.5 Discharge Coefficient

The discharge coefficient was calculated and it shows a time-dependent character. As the flow temperature (\(T_f\)) increased, the \(C_d\) decreases. It is known that the \(C_d\) depends on the Reynolds number; this value is decreasing for higher Reynolds number (Daugherty & Franzini, 1965, pp. 338-349). Since the averaged velocity was increasing during these experiments a minimum value of 0.44 was found at 12:30h. Due to the experimental values which were used to calculate the \(C_d\), values higher than unity were calculated by Equation 2.
3.4.6 Air temperatures

It is obvious that the differences in the air temperature over the depth (3 °C) are much larger than over the width (0.2 °C) (Figure 12&Figure 13). This is because the absorber surface has in generally has higher temperatures compared to the glazing surface; respectively at 13:20h at 11 meter height the absorber is 73 °C (± 0.5 °C) and the glazing is 39 °C (0.5 °C) (Figure 14&Figure 16). Also, based on the measured data (Figure 12) a very small temperature difference was found over the width. However - as Appendix 2 shows - the vertical air velocity difference over the width is larger than the value in the depth. These might play an important role in modelling; especially when performing CFD simulations on WSC systems and one need to choice between 2D or a 3D approach.

3.4.7 Surface temperatures

The amount of solar irradiation coming into the system depends on the sun position and the shape (obstacles) of the façade and its orientation. In Figure 15 one can see that some surfaces are heated up later on the day while the others are heated up at the beginning and gradually begin to cool down later on the day. This is due to the shading of the side walls as function of sun’s position. This may also be the reason for the larger vertical air velocity difference over width compared to the depth of the chimney. Furthermore between 9:00h to 10:00h a lower surface temperature of the glazing and the frame is observed (Figure 16&Figure 18). This might be the due to the higher radiative heat losses to the environment for the upper and the bottom side of the chimney. Here the losses are maximum while the minimum is in the middle of the chimney (Kim et. al, 1990).

3.4.8 Conduction losses

The conduction losses through the inner-pane (absorber side) was found to be neglected (well insulated construction). The conduction losses through the glazing construction can’t however be neglected (Figure 16&Figure 18). This conductive loss (U-value of 1.32 W/m²K, net glazing area of 15.7 m²) means 518.1 Watt of power at 12:30h by assuming a temperature difference of 25 °C (± 0.5 °C) over the glazing.

3.4.9 Solar irradiation

The optical properties of a WSC system – especially the direct transmission of the glazing (Figure 6) – plays an important role on the performance of the system (Figure 10). $S_{Y,IN}$ was found to be approximately half of $S_{Y,OUT}$. The vertical solar irradiation which did incident the outer surface of the glazing construction was measured to be 733 W/m² (± 5%) on 12:30h and the irradiation behind the glazing surface was at that moment 420 W/m² (± 5%). The direct transmission of the short wave radiation through the glazing is then 57%, which corresponds well to the value of direct transmission (0.55) which was calculated using WINDOW 5 (Figure 6).
3.5 Conclusion

A real-sized WSC (wall solar chimney) system with an aspect ratio (Height/Depth) of 40 was used as an experimental set-up which was placed outdoor in Molenhoek (Netherlands). Two quantities were controlled: the air velocity inside the shaft ($\approx 1$ m/s) by controlling a hydraulic opening at the inlet of the air-conditioning room and the air-temperature ($\approx 20$ °C) which went through the chimney’s inlet. These experiments were performed in one day (15-12-2009) and the results showed in this study are for 9:00h to 16:00h. A maximum of $\Delta T_{air}$ 11.6 °C and an air volume rate of 1571 m$^3$/h was generated. The maximum outside vertical solar irradiation was 736 W/m$^2$.

From Figure 14 the measured absorber temperature on 4 meter height notices an incorrect surface temperature as this temperature must increase as the height increases (Moshfegh et. al, 2005, p. 257). Therefore this sensor must be rechecked again.

The provided data from WINDOW 5 software showed good agreement with measured data concerning the amount of transmitted solar energy (Figure 6&Figure 10). To be able to increase the thermal efficiency of the WSC system it is necessary to improve this property of the glazing (Equation 1).

From the results shown in figures 15, 19 and Appendix 2 one can conclude that the problem of a WSC system is a three-dimensional physical problem. The solar irradiation and the angle of incident, the optical properties of the glazing, the vertical air velocity over the width of the chimney and the surface temperatures of the side walls all show a dynamic behaviour in time and space. Also, the discharge coefficient ($C_d$) showed a dynamic behaviour in time because it depends on the Reynolds number which in turn depends on the averaged vertical air velocity.

Normal conduction loss through the glazing was found to be high compared to the losses through the inner-pane (which is well insulated). In order to increase the efficiency of the WSC system the U-value of the glazing needs to be smaller.

Further measurements are needed to show the exact influence of the wind on the thermal performance of the WSC system. Wind tunnel test or CFD simulations might be a solution (Cóstola et. al, 2009).

Finally the results show that an outdoor real-sized WSC system has the potential to be used in the built environment for generating natural ventilation in buildings. A thermal efficiency of 61% for a winter period was calculated. However the annual system performance needs to be judged in future, and also when it is connected to a real building.

The solar irradiation, the outlet air temperature, the volume rate, the surface temperatures and the discharge coefficient are relevant data to be implemented and used for model validation. Also the design parameters will be used for modelling purposes.
4. Modelling Method

4.1 Introduction

One can use different modelling approaches for predicting certain PI (Performance Indicator) of a WSC (Wall Solar Chimney) system (1.1.3&1.1.5). This study focuses only on models which are based on BES (Building Energy Simulation), BES plus AFN (Airflow Network) and CFD (Computational Fluid Dynamics). In order to perform simulations the computer programs or codes ESP-r (for BES, BES+AFN) and Gambit & Fluent (for CFD) will be used.

For the purpose of BES and BES+AFN simulation the computer model ESP-r (Energy Simulation Performance research) is used and the reason is twofold. First, this is an open source code and it has - compared to other existing building performance simulation tools - a broad range of capabilities and validation history (Crawley et. al, 2008). Second, at the unit of building performance simulation (BPS) at Eindhoven University of Technology has experienced and essential users, who have close collaboration with the developers of ESP-r form the University of Strathclyde. For more information on ESP-r please refer to (Hensen, 2001), (Beausoleil-Morrison, 2000) and (Clarke).

Furthermore, for CFD simulations this study uses Gambit for the pre-processing and Fluent 6.3 for the solving and post-processing of the solutions (Fluent, 2006).

This chapter introduces the modelling approaches (BES, BES+AFN and CFD) which differ in their boundary conditions and other modelling assumptions.
4.2 BES (Building Energy Simulation)

4.2.1 Boundary conditions

Sets of partial differential equations (PDE’s) are solved in time as well in space (Clarke, p. 63). To be able to solve a set of PDE’s these equations need initial conditions as well as boundary conditions. These initial and boundary conditions in ESP-r are set respectively by the user (using the temporal file approaches) which is generally supported by a weather data file (ASCII format) (Appendix 6).

4.2.1.1 Weather data

The weather file consists of 24 row of information which corresponds to 24 hours per day. As already mentioned BES in ESP-r is a dynamic thermal simulation. From the list of Appendix 6, the number 1 to 4 are needed to calculate the position of sun compared to the location and the time period of the model of interest. Next, the numbers 5 to 10 are the boundary conditions to the model at hand.

4.2.1.2 Boundary resolution

The BES approach automatically considers the room, the shaft and the environment of the WSC system (See 2.2.).

The red dotted rectangular shows the boundary resolution suitable when using the BES approach. This consists of the shaft, the room and the environment (wind is not included). This approach will account for all kind of heat transfer calculation (conduction, convection and radiation) per thermal zone.

4.2.1.3 Temporal resolution

ESP-r uses a second order finite difference method based on the Crank-Nicolson method. It interpolates between the input values (per hour) - depending on the simulation time-step.
size – in order to calculate all heat transfer parameters. More about solving the governing PDE’s, is explained in all sources of numerical methods, but also in (Clarke). In order to investigate the influence of the temporal resolution on the simulation results this study will be performing a temporal sensitivity analysis. Therefore 5 different simulation time steps will be compared to make the models timestep independent.

4.2.2 Models

In order to investigate the influence of the spatial resolution on the simulation results this study will be performing a spatial sensitivity analysis. Four different models in BES (ESP-r) are generated with a different number of thermal zones. There are models which include 6, 8, 10 and 14 thermal zones (Figure 25 until Figure 28).

However with only BES a naturally ventilated WSC can’t be appropriately simulated. In reality the convective heat transfer coefficients inside the chimney are flow depended and the air moves because of the air density differences inside the chimney. In this case a fixed air volume rate must be used by the modeller which is yet the unknown quantity in the real situation of a naturally ventilated WSC system.

For the BES approach several models are generated (Figure 24). Only the frame construction and the glazing surfaces are considered as an external boundary condition and all other outdoor surfaces are considered adiabatic. Also, the distinction between different thermal zones is clear - in this example there are 7 thermal zones for the chimney. Notice that a BES modelling approach is without an airflow network (AFN). Besides all the horizontal separations between the thermal zones are considered as fictitious surfaces, which means that these surfaces are expected not influence the thermal calculation due to their high transparency and conductive properties.

4.2.3 Assumptions

It is assumed that the direct shortwave irradiation - which is transmitted by the glazing surface - will spread its solar energy into the system according to solar radiation theory (Clarke, p. 210).

The simulations in ESP-r are dynamic simulations (the sun position is changing per time) and therefore the optical performance of the glazing surface - which depends on the sun’s angle of incident – is calculated using WINDOW 5 (see WINDOW) software. The optical properties of the glazing are assumed to be the exact properties known from the manufacturer for a double glazed system (Planitherm Saint Gobian Glass Solutions). These data are shown in Figure 6 and are set in the models by using WINDOW data import in ESP-r.

Consider Figure 24 for more detailed information about the applied boundary conditions to the models based on BES. The horizontal surfaces (the separations between the thermal zones) are all considered as physical transparent and highly conductive and their influence
in thermal calculations is assumed to be small (fictitious material) (See Appendix 7). The outer surface of the inner-pane (absorber wall) is assumed to be adiabatic, since the results from the measurements showed no relevant losses. However exterior boundary conditions are assigned to the outside of the frame constructions and the glazing, as this was found to be important from the experimental results (see 3.4.8).

Figure 24: Sketch of the models in ESP-r based on BES and BES plus AFN.
4.3 BES and AFN (Air Flow Networks)

4.3.1 Boundary conditions

The BES & AFN approach will implement the same boundary conditions as in BES (see section 4.2.1).

4.3.2 Boundary resolution

Although the method presumes one-dimensional steady state flow, boundary conditions (wind, temperatures, fan operations, window openings) can vary in time. Stack effects caused by indoor-outdoor and inter-zone temperature differences are also considered. This is the main difference between BES+AFN and the BES approach.

Briefly, the environment, the air-conditioning room and the shaft are completely considered using this approach (Figure 7). This is comparable with the situation as shown in Figure 23, yet the outdoor parameters (wind, hydrostatic pressures) which influence the pressure inside the shaft are also connected here.

4.3.3 Models

For the same reason as it was explained in 4.2.2, the same spatial resolutions as in BES approach (Figure 25 until Figure 28) are used for the BES+AFN approach. This are further extended with a suitable airflow network with outdoor boundary nodes in combination with the air flow components (Figure 24). For example if there are seven thermal zones there are also seven air volumes (See Appendix 7).

In case of BES+AFN an existing model of BES is used to extend its boundaries by adding air flow network (AFN) to these models. An airflow network has several outdoor boundary nodes (influenced by outdoor conditions for example wind), airflow network components and air volume nodes (8 nodes in this example) respectively shown as a circle and a rectangular. Between two thermal zones one needs to add an air flow component. In this study the outdoor boundary nodes are the nodes 1, 2 and 3. The components a, H, I are considered as a Common Orifice Flow Component and the inner components – B until G – are considered as General Flow Conduit (Hensen, 1991).

4.3.4 Assumptions

The BES & AFN approach applies the same assumptions as in BES (see section 4.2.3). In addition to the latter the following have been assumed.

For the hybrid ventilation problem (Table 3) an airflow network is used with the accompanying components. To be able to calculate the Discharge Coefficient $C_d$, first the sum of all local losses and frictions $\sum \Delta f_i$ need to be determined. The discharge coefficient
takes into account the non-uniform distribution of inlet velocities, contraction of fluid stream, surface roughness, etc. It is an important parameter used in theoretical models to determine the massflow rate through the system (Equation 10).

Table 4 shows the input parameters to the airflow network models. The discharge coefficients \(C_d\), the local dynamic losses \(C_i\), the cross-sectional area \(A\), the hydraulic opening \(D_h\), the length \(L\), the roughness \(f\) and finally the network component are applied in the ESP-r (BES+AFN) models of the WSC system.

\[
C_d = \sqrt{\frac{1}{\sum C_i}} = \sqrt{\frac{1}{\sum f \frac{L}{D_h} + \sum C_i}}
\]

Equation 10: The discharge coefficient \(C_d\) is calculated using the sum of all local dynamic losses and friction losses \(\sum k_i\) in the system (Akbarzadeh et. al, 1982).

Values of the local dynamic losses \((C_i)\) or in some literature \(\zeta\) is found in the literature (Hensen et. al, 2005). These values have been developed in the past for velocity profiles as they occur in pipes: symmetric and having the highest velocity at the centre (Kalyanova et. al, 2005).

For comprehensive background information on airflow components which are implemented in ESP-r please refer (Hensen, 1991).

<table>
<thead>
<tr>
<th>Positions</th>
<th>(D_h)</th>
<th>(A)</th>
<th>(L)</th>
<th>(f)</th>
<th>Dynamic losses</th>
<th>Sort component used in ESP-r</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Inlet</td>
<td>(D_h=0.444) m</td>
<td>(A=0.5) m&lt;sup&gt;2&lt;/sup&gt;</td>
<td>not applicable</td>
<td>not applicable</td>
<td>(C_d=0.65) or (C_d=0.42)</td>
<td>Common Orifice Flow Component</td>
</tr>
<tr>
<td>2. Chimney inlet</td>
<td>(D_h=0.444) m</td>
<td>(A=0.5) m&lt;sup&gt;2&lt;/sup&gt;</td>
<td>0.25 m</td>
<td>0.0015 m</td>
<td>(\sum C_i=0.72)</td>
<td>General Flow Conduit</td>
</tr>
<tr>
<td>3. The shaft</td>
<td>(D_h=0.444) m</td>
<td>(A=0.5) m&lt;sup&gt;2&lt;/sup&gt;</td>
<td>11 m</td>
<td>0.0015 m</td>
<td>(\sum C_i=0)</td>
<td>General Flow Conduit</td>
</tr>
<tr>
<td>4. Outlet North</td>
<td>(D_h=0.235) m</td>
<td>(A=0.25) m&lt;sup&gt;2&lt;/sup&gt;</td>
<td>not applicable</td>
<td>not applicable</td>
<td>(C_d=0.65) or (C_d=0.42)</td>
<td>Common Orifice Flow Component</td>
</tr>
<tr>
<td>5. Outlet South</td>
<td>(D_h=0.235) m</td>
<td>(A=0.25) m&lt;sup&gt;2&lt;/sup&gt;</td>
<td>not applicable</td>
<td>not applicable</td>
<td>(C_d=0.65) or (C_d=0.42)</td>
<td>Common Orifice Flow Component</td>
</tr>
</tbody>
</table>

Table 4: Overview of the input parameters for the use of the components in the Air Flow network in ESP-r.

For exact position of the room’s inlet, the chimney’s inlet, the shaft and the outlet please consult Figure 7&Figure 9. \(D_h\) (m) is the hydraulic diameter, \(L\) (m) is the characteristic length, \(f\) (m) is the wall roughness, \(\sum C_i\) is the local dynamic loss and \(C_d\) is the discharge coefficient per component.
The maximum value of the discharge coefficients of the inlet and the outlet area were found to be 0.65 (Hensen et. al, 2005). However the minimum value of 0.42 for the discharge coefficient of the inlet and the outlet is based on the experimental data (Figure 22) and the values shown below (Table 5 & Equation 10).

<table>
<thead>
<tr>
<th>Real system</th>
<th>Physical problem</th>
<th>( L, D_h, A ) and ( f )</th>
<th>( \sum C_i )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outlet North</td>
<td>For each outlet: ( L = 0.25 , \text{m} ) ( D_h=0.235 , \text{m} ) ( A=0.25 , \text{m}^2 ) ( f = 0.0015 , \text{m} )</td>
<td>( C_{\text{T-form}} = 0.92 ) + ( C_{90^\circ \text{turn}} = 0.22 ) + ( C_{\text{Outlet}} = 1 )</td>
<td>( \sum C_i = 2.14 )</td>
</tr>
<tr>
<td>Outlet South</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Chimney’s inlet</td>
<td>For each outlet: ( L = 0.25 , \text{m} ) ( D_h=0.444 , \text{m} ) ( A=0.5 , \text{m}^2 ) ( f = 0.0015 , \text{m} )</td>
<td>( C_{90^\circ \text{turn}} = 0.22 ) + ( C_{\text{inlet}} = 0.5 )</td>
<td>( \sum C_i = 0.72 )</td>
</tr>
<tr>
<td>Rectangular Inlet</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Channel</td>
<td>For each outlet: ( L = 0.25 , \text{m} ) ( D_h=0.444 , \text{m} ) ( A=0.5 , \text{m}^2 ) ( f = 0.0015 , \text{m} )</td>
<td>No local dynamic losses</td>
<td>( \sum C_i = 0 )</td>
</tr>
</tbody>
</table>

Table 5: The values of local losses, hydraulic diameter, roughness and the concerning physical problem for the experimental set-up from chapter 2 are shown here (Source: Installatiehandboek 2004).

The calculation of the massflow using component “Common Orifice Flow Component (type 40)” one can find in (Hensen, 1991, P. 4.21). The calculation of the massflow in the air flow network based on “General flow Conduit Component (type 210)” one can find in (Hensen, 1991, P. 4.23). The equations are as follows:
- Type 40 component: 
- Type 210 component: 

Furthermore, the stack pressure difference over the shaft is caused by the natural buoyancy forces. Therefore in this study two different convective heat transfer correlations (Alamdari-Hammond and Khalifa-Marshall correlations) will be used in order to demonstrate their influence on the prediction of the massflow (Beausoleil-Morrison, 2000) (See also 2.1.3 and Appendix 13). These two correlations have shown differences in prediction of the convective heat transfer coefficients in previous building simulation studies, and therefore they might generate different results when simulating the performance of the WSC system. Appendix 13 shows for 2 different temperature differences (1 and 5 Kelvin) how the prediction of CHTC is done using these two different approaches. The Alamdari-Hammond shows a length dependent character while Khalifa-Marshall is not. This length is assumed to be the thermal zone height.

The convective heat transfer coefficients of the internal walls of the WSC system are determined using the following correlations:

1- Alamdari-Hammond correlation (Equation 11),

2- Khalifa-Marshall correlation (Equation 12).

\[
h = \left[ 1.5 \left( \frac{\Delta T}{H} \right)^{\frac{1}{4}} \right]^6 + \left[ 1.25 \Delta T \right]^{\frac{3}{2}}
\]

*Equation 11: Alamdari-Hammond convective heat transfer correlation for non-heated walls*

\[
h = 2.30 \Delta T^{0.394}
\]

*Equation 12: Khalifa-Marshall convective heat transfer correlation for non-heated walls*

Here is \( h \) (W/m²K) the convective heat transfer coefficient, \( \Delta T \) (Kelvin) is the temperature difference between a heated surface and a reference air temperature and \( H \) (m) is the characteristic length of a thermal zone. In contrast to Alamdari-Hammond correlations the Khalifa-Marshall correlations are not length depended.
4.4 CFD (Computational Fluid Dynamics)

4.4.1 Models and boundary conditions

This study considers 3 different computational domains for the CFD simulations. Models1 considers only the air volume of the shaft (Figure 29). The other models 2 and 3 are extended respectively from the inlet and the outlet until the total environment.

Figure 29: CFD Model1 is shown here.

This is the simplest model with smallest boundary resolution. It is a 2D model of a rectangular cavity with an aspect ratio of 44. It has 80 cells on the horizontal and 107 on the vertical. An inlet pressure boundary condition (□) is chosen for the inlet at the bottom of the chimney. A pressure outlet boundary condition (○) is considered for the outlet of the chimney. A constant and fixed heat-flux boundary condition (X) is used for the absorber as well for the glazing surface. Also the grid distribution is shown here in more detail. This particular distribution of grid is the result of a grid sensitivity analysis where the $Y'$ and the convective heat transfer coefficients are analyzed.

Next, Model2 (Figure 30) is generated. This has slightly broadened boundaries compared to Model1 to be able to consider the influence of a developed flow and possible reversed flow respectively on the inlet and the outlet of the system.
Figure 30: The 2D CFD Model2 was slightly broadened at the inlet and the outlet area.

For the inlet area a pressure inlet boundary condition (□) is chosen which is set to 1 meter to the right from the actual inlet of the chimney. The outlet pressure boundary condition (○) set to have 1 meter of distance from the actual outlet area. Furthermore, a constant and fixed heat-flux boundary condition is set to the absorber and the glazing surface (X).

Model3 (Figure 31) which represents the combination of the chimney with its environment was finally generated. The environment is included to this problem by choosing a domain which has a horizontal border of 50 meter (on the left and right of the chimney) and a vertical border on 50 meter from the outlet area of the chimney. A velocity inlet boundary condition was chosen 50 meters for the vertical edges on the left and the right of the domain (- - -) both directed towards the chimney. The velocity here will be set to a small value perpendicular to the edges (v=0.01 m/s). Furthermore, for the edge on the top a pressure outlet boundary condition is chosen (— —) directed upward. Also, here a constant and fixed heat-flux boundary condition is set to the absorber and the glazing surfaces (X).
Figure 31: The 2D CFD Model 3 is the most complex model in this study.

In order to create reliable models for the model validation, beforehand a sensitivity analysis is done on the temporal and the spatial resolution for the different modelling approaches. The latter study is performed by considering an arbitrary situation based on the CASE A experimental set-up (Table 3). The focus is only on BES+AFN and the CFD model since the BES only is not able to represent a natural ventilation problem.

A comprehensive solution method of the CFD models which is generally used in this study is explained in Appendix 3.
4.5 Results

4.5.1 Temporal resolution in BES+AFN

The effect of temporal resolution on the prediction of the massflow in the shaft is studied for 1 BES+AFN model with a fixed number of thermal zones of 6 zones (Figure 25). The experimental working principle CASE A was considered here (Table 3). A BES+AFN approach is used to illustrate how sensitive the simulations are due to the choice of a simulation time step.

Figure 32: The temporal sensitivity of the predicted massflow in the chimney depending on the chosen time step size is shown here. On the horizontal the time in hours is set.

As the temporal time step is increasing the predicted massflow becomes smoother over time. On the left vertical axis the massflow (kg/s) is shown. On the right vertical axis (red dotted line) the direct normal solar irradiation in W/m² can be found.

4.5.2 Spatial resolution in BES+AFN

The effect of spatial resolution on the prediction of the massflow in the shaft was studied for four models independently. These models are shown in Figure 25-28. Furthermore, the experimental working principle CASE A (Table 3) was considered here. A BES+AFN approach is used and the simulations are for a fixed simulation time step of 60 steps/hour for a summer weather condition. Finally, the Alamdari-Hammond correlations are used. In Figure 33 one can see the results of this sensitivity analysis.
Figure 33: The effect of the number of thermal zones on the massflow is shown here. These results are based on the models according to 4.2.3. On the horizontal the time in hours is set.

As the number of thermal zones increases the difference from the prediction of the previous model compared to the new model decreases. For instance the difference in prediction of the massflow between Model1 and Model2 is more than the difference in predictions between Model3 and Model4.

Figure 34 shows the effect of different CHTC correlations on the prediction of the massflow for BES+AFN Model4. The model with Khalifa-Marshall correlation predicts higher massflows with a difference of 0.01 kg/s.

4.5.3 Grid sensitivity in CFD

In order to have a reliable grid distribution for the CFD models (Figure 29&Figure 30&Figure 31), a grid sensitivity analysis is performed on Model1. Different distributions are compared to each other by calculating the CHTC (convective heat transfer coefficient) on the absorber surface. The results of this analysis are shown below. The low-Reynolds k-epsilon turbulent model was applied to all the models.
Figure 35: This graph shows the prediction of the CHTC against the height of the chimney using different grid distributions.

The value of the CHTC was calculated using the predicted local heat-flux and the local temperature difference between the absorber and perpendicular to the latter the air temperature in the middle of the depth.

The CHTC was calculated using the following equation:

\[
q_{\text{conv,y}} = h_y \Delta T = h_y (T_{\text{abs}} - T_{\text{midline}})
\]

Equation 13: The CHTC (convective heat transfer coefficient) is calculated by using this equation. 

- \( q_{\text{conv,y}} \) is the local convective heat-flux (W/m²) over the height, 
- \( h_y \) is the local CHTC (W/m²K), 
- \( T_{\text{abs}} \) is the local absorber temperature (Kelvin) and 
- \( T_{\text{midline}} \) (Kelvin) is the local air temperature which is determined by the vertical line parallel to the absorber surface on 0.125 m distance in the middle of the cavity.

Yet the value of \( Y^+ \) is also considered and this is been always approximately 0.8 for all the grid distributions (Fluent, 2006).
4.6 Discussion

4.6.1 Modelling

It seems not possible to relate the whole environment of the WSC (Wall Solar Chimney) system in the BES models, because there is no relation between the air pressures outside and inside the system (air flow is not simulated when using BES). The modeller needs to add the amount of air that is extracted from the WSC system himself while the extraction of air in reality is caused by buoyancy forces. For example there is no direct relation of wind pressure at the air inlet, the stack pressure in the shaft and the total pressure at the outlets of the system.

Since this study focuses on models in BES+AFN and 2-dimensional CFD it is not possible to consider the entering air temperature to the chimney as a temperature profile in the third dimension (width of the chimney) but rather as averaged value to the models at hand.

Even though that the chosen “Common Orifice Flow Component (type 40)” for the inlet (Table 4) in BES+AFN is not exactly representing the reality (experimental set-up CASE B) it is still useful because the CFD models in this study will also not use these kind of air flow components (component type 410). The physical behaviour of the hydraulic opening might be ascribe by “General Flow Corrector Component (Type 410)” (Hensen, 1991, p. 4.33), however a well explained literature for using this component was not found and therefore this component was left out of consideration.

4.6.2 Temporal resolution BES+AFN

From Figure 32 one can conclude that the prediction of the massflow in the system becomes stable from a time-step size which is shorter than the physical time duration of 4 minutes. Also, these results show a difference at the start of the day choosing a time-step size of 1 minute instead of 4 minutes.

The minimum possible time-step size in ESP-r was found to be 40 seconds which corresponds to 90 simulation steps per hour.

4.6.3 Spatial resolution BES+AFN

The study on BES+AFN was done using all the models shown in previous sections, but only the temporal resolution dependency of Model1 was considered. This is because the other models showed the same differences no matter how many thermal zones were used.

The number of thermal zones seems to have certain influences in the prediction of the massflow through the system. The decrease of the latter quantity – as the number of thermal zones increases - might be due to the fictitious horizontal surfaces between thermal zones (Figure 24). These surfaces hinder the direct transmittance of the short
wave radiation from one zone to another. This means that the total solar irradiation coming into the system is not distributed properly.

4.6.4 Grid sensitivity CFD

Figure 35 shows the predicted CHTC (Convective Heat Transfer Coefficients) over the height of the chimney using different discretizations in the depth. From 14X10^7 until 40X10^7 large discrepancies in the CHTC can be observed. However as the grid size becomes denser – from 40x10^7 to 80x10^4 over the depth - less differences can be seen. The grid sensitivity was performed only on Model1 (Figure 29) by using the low-Reynolds k-epsilon turbulence approach. Model2 (Figure 30) and Model3 (Figure 31) were expanded according to the grid distribution (80x10^7) as in Model1. For the latter two models no grid sensitivity analyses were performed. However, the $Y^+$ was always checked and this was always around 0.8 for all the CFD models.

Note that the calculated values of CHTC shown in Figure 35 are based on different weather conditions (boundary conditions) based on the calculated convective heat-fluxes in ESP-r for 14-06-2009. Thus, no further conclusions can be made with regard to the real performance of the WSC system.

The exact solution method used for CFD models is explained in Appendix 5.
4.7 Conclusion

The experimental set-up was translated to different modelling approaches. These models were pre-processed in ESP-r and Gambit&Fluent respectively for BES+AFN approach and CFD approach. The data from the experiments were used as input parameter to these modelling approaches (Climate file, optical properties, design parameters etc.). Four different modelling approaches based BES+AFN with differences in their temporal and spatial resolution were examined. Besides, three different modelling approaches based on CFD with differences in their boundary resolution were also compared to each other. The grid sensitivity analysis was performed on the simplest model in CFD (Model1) where the $Y^+$ and the convective heat transfer coefficient were examined. Here arbitrary boundary conditions were used, which were collected from BES+AFN.

From the previous findings one can conclude that the minimum modelling complexity which is needed to properly model a WSC system based on the natural ventilation is the BES+AFN approach, because in BES one can’t properly simulate the naturally ventilated WSC system.

What the exact influence of the fictitious surfaces are on the prediction of difference PI’s needs further study. However for the validation study Model4 with a temporal resolution of 1 minute is chosen. In this case the spatial resolution (see position of glazing Figure 9) has close agreement with the reality and because then a temporal independent situation is reached (Figure 33).

Furthermore a relevant modelling limitation in BES+AFN was found concerning the experimental set-up CASE B (hybrid ventilation). The hydraulic opening of the air-conditioning room couldn’t be properly modelled with the existing air flow components in ESP-r. However Appendix 2 (Figure 3) shows that the hydraulic opening is rarely in action. Further study needs to prove the use of General Flow Corrector Component (Type 410) for this case” (Hensen, 1991, p. 4.33).

For the validation of the CFD models a grid independent boundary resolution of 80x107 is chosen. Latter boundary resolution serves as basis for the boundary resolutions of Model2 and Model3.
5. Validation Method

5.1 Introduction

There are several validation techniques which could be used to validate the generated BES+AFN and CFD models. According to ASHRAE Standard 140 (Judkoff et al. 2006, p. 2) there are only a few ways to evaluate the accuracy of a whole-building energy simulation program (Judkoff et al. 2006) (See Table 6):

- **Empirical Validation**: in which calculated results from a program, subroutine, algorithm, or software object are compared to monitored data from a real building, test cell, or laboratory experiment.

- **Analytical Verification**: in which outputs from a program, subroutine, algorithm, or software object are compared to results from a known analytical solution or a generally accepted numerical method for isolated heat transfer under very simple, highly constrained boundary conditions.

- **Comparative Testing**: in which a program is compared to itself or to other programs.

<table>
<thead>
<tr>
<th>Technique</th>
<th>Advantages</th>
<th>Disadvantages</th>
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<tbody>
<tr>
<td>Empirical Validation</td>
<td>Approximate truth standard within experimental accuracy</td>
<td>Experimental uncertainties:</td>
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<tr>
<td>Test of model and solution process</td>
<td>Any level of complexity</td>
<td>- Incremental calibration, spatial-temporal discretization</td>
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<tr>
<td></td>
<td></td>
<td>- Imperfect knowledge/specification of experimental object (building)</td>
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<td>- Uncalibrated</td>
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<td>- High-quality detailed measurements are expensive and time consuming</td>
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<td>- Only a limited number of test conditions are practical</td>
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<tr>
<td>Analytical Verification</td>
<td>No input uncertainty</td>
<td>No test of model validity</td>
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<tr>
<td>Test of solution process</td>
<td>Exact mathematical truth standard for the given model</td>
<td>Limited to highly constrained cases for which analytical solutions can be derived</td>
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<td></td>
<td>Inexpensive</td>
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<tr>
<td>Comparative Testing</td>
<td>No input uncertainty</td>
<td>No test of model validity</td>
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<tr>
<td>Balance test of model and solution process</td>
<td>Any level of complexity</td>
<td>Limited to highly constrained cases for which analytical solutions can be derived</td>
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<td>Many diagnostic comparisons possible</td>
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<td>Inexpensive and quick</td>
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<td></td>
<td>No absolute truth standard (only statistically based acceptances are possible)</td>
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Table 6: An overview of validation techniques (Judkoff et al. 2006, p. 3)

In the context of this study the experimental set-up is used as the basis for this validation study. The data provided here will be used as input parameters to the models (weather data, heat fluxes etc.) at hand but also some data will serve as reference (outlet air temperature, absorber or glazing temperatures etc.) to be able to compare the simulation results with them (empirical validation technique). Also, different BES+AFN and CFD models will be compared to each other (comparative testing). However an analytical validation is beyond the scope of the current study.

By determining the discrepancies (magnitude of difference) between model predictions and the measured data this study attempts to provide the modellers of a WSC system the information necessary to choose a reliable modelling approach.
5.2 Empirical validation and comparative analysis

5.2.1 BES+AFN

The predicted PI’s (Performance Indicators) in ESP-r based on the BES+AFN approach - the vertical air velocity ($V_Y$), the outlet air temperature ($T_{out}$), absorber temperature ($T_{abs}$) and the glazing temperature ($T_{glz}$) - will be compared to the measured data from the experimental set-up for 15-12-2009 between 9:00h to 16:00h.

Next, for the empirical validation of the chosen BES+AFN model (Model4), the measured data - as shown in Figure 36 – will be implemented into ESP-r to predict the PI’s of the WSC system.

Figure 36: In ESP-r one can include measured data via the ‘imposed measured data’ to certain boundaries/surfaces. This approach is used during this empirical validation study. So, the data (a) set as the weather file, (b) are set to the outside glazing surfaces and the data (c) are set to the room air volume node (Figure 7 & Figure 24).

The weather file with the accompanying data was measured per 10 minute interval, and these data are on a hourly basis implemented to ESP-r modelling environment using ACSII-file’s.

Since the boundary conditions for model4 in BES+AFN must have the same boundary conditions as for the experiments on 15-12-2009, these data can be included using the ‘imposed measured data’ (Figure 36). So, the $S_{Y,OUT}$ (Figure 10) and $T_{air}$ at 0.25m height (Figure 11) will be added to the BES+AFN model. The values of $S_{Y,OUT}$ per 10 minute are set to the outside glazing surface and the values of $T_{air}$ per 10 minute are set to the volume node of the room (Figure 24).

5.2.2 CFD

The predicted PI’s - the vertical velocity profile ($V_Y$), the outlet air temperature ($T_{out}$), absorber temperature ($T_{abs}$) and the glazing temperature ($T_{glz}$) - will be compared to the measured data from the experimental set-up for 15-12-2009 between 9:00h to 16:00h.

Two different method of simulation were used for CFD models. The **method1 approach** (Figure 37) uses directly the solar irradiation which is measured during the experiments (Figure 10) as the boundary condition to the absorber and the glazing. So, 5% of $S_{Y,OUT}$ is
set to the glazing surface which corresponds to 0.05 absorbance shown in Figure 6. The measured $T_{air}$ at 0.25 m height (Figure 11) will be set as the inlet boundary and as the initial air temperature (CFD model3). The value of $S_{y,IN}$ (Figure 10) is set to the absorber surface. The simulations are done for 12 instantaneous times between 9:00h until 16:00h (See Appendix 4).

Figure 37: In CFD one can include measured data by setting the values for the chosen boundary conditions. So, the data (c) are set to the inlet of the chimney, the data (e) are set to the glazing surface and the data (f) are set to the absorber surface (Figure 29).

However the method1 approach isn't comparable to the BES+AFN models when comparing their convective heat fluxes. Therefore a method2 approach is introduced allowing a proper comparison between the CFD and BES+AFN as well as CFD and measurement, because in this way the CFD models indirectly include the effect of the thermal mass of the inner-pane using the information from BES+AFN models. The CFD method2 uses the calculated convective heat fluxes by BES+AFN model4 based on the Alamdari-Hammond correlation and a $C_d$ of 0.42 for the inlet and the outlet area (See Appendix 4).

### 5.2.3 Comparative analysis

Three different BES+AFN models are generated by considering the following configurations:

- Discharge coefficients ($C_d$) for the inlet and the outlet ($C_d=0.42$ or $C_d=0.65$),
- The convective heat transfer correlations for the vertical walls inside the chimney (Alamdari-Hammond or Khalifa-Marshall correlation).

Furthermore, 4 different CFD models will be considered during this validation study based on the following configurations:

- The choice of the boundary resolution (Model1, Model2 and Model3);
- The choice of the turbulence model. The low-Reynolds k-epsilon and standard k-omega for model1.
Note that all the CFD models above are performed twice (Appendix 4). Also, all the CFD simulations are performed without using a radiation model or a conduction model (See Appendix 5).

In addition to the comparison of the predicted and measured PI’s, the predicted convective heat transfer coefficients (CHTC) in BES+AFN (trace facility in ESP-r) will be compared to the CFD simulations. Unfortunately because no heat-flux measurements are performed this study will only show the uncertainty in the prediction of the CHTC by using different approaches.

For instantaneous times (10:30h and 12:30h) the predicted CHTC’s over the height of the chimney will be calculated using the following Equation 13. Also, the dimensionless Nusselt number (Nu) and Rayleigh number (Ra) will be used to clarify the problem at hand (2.1.5).

### 5.2.4 Uncertainty analysis

This is a very rough uncertainty analysis which shows the discrepancies between the predicted and measured total heating power gain which is calculated using the following equation: \( P_{\text{measured}} = A \cdot v_{\text{avg}} \cdot \rho \cdot c_p \cdot (T_{\text{OUT}} - T_{\text{IN}}) \) in Watt. An uncertainty band will be calculated for the measured data. For the measured values the uncertainty of each parameter is included in order to calculate the uncertainty in the total heating power gain. These uncertainties are assumed as follows:

- The minimum and maximum value for the cross-sectional area is a rough assumption of \( \pm 0.001 \text{m}^2 \) for an averaged area of \( A = 0.5 \text{ m}^2 \).
- \( v_{\text{avg}} \) is based on the vertical velocity profile measured according to Appendix 2 and estimated by Equation 9 for every simulation time. For each averaged value the maximum and the minimum uncertainty was determined by respectively including or subtracting the 5% sensor error.
- The maximum and the minimum density of air are based on respectively the minimum and maximum possible air temperatures which were measured. Therefore, the following values are used in this calculation: \( \rho_{\text{min}} = 1.067 \text{ kg/m}^3 \), \( \rho_{\text{average}} = 1.2 \text{ kg/m}^3 \), \( \rho_{\text{max}} = 1.293 \text{ kg/m}^3 \).
- Also the determination of the specific heat is done in the same manner as for density. The following values are assumed here: \( c_{p_{\text{min}}} = 1005 \text{ J/kgK} \), \( c_{p_{\text{average}}} = 1007 \text{ J/kgK} \) and \( c_{p_{\text{max}}} = 1009 \text{ J/kgK} \).
- The value of \( T_\text{IN} \) and \( T_\text{OUT} \) is based on the averaged value of respectively 3 and 9 thermocouples (Figure 9). The minimum and the maximum values are calculated by respectively subtracting and including the 0.5 °C as sensor error.

The same equation \( P_{\text{predicted}} = A \cdot v_{\text{avg}} \cdot \rho \cdot c_p \cdot (T_{\text{OUT}} - T_{\text{IN}}) \) was used to determine the total heating power gain for different modelling approaches. The models which have a calculated \( P \) in the uncertainty band of the measured heating power gain and at the same time have close agreement to other PI’s (glazing and surface temperatures) of the measured WSC system will be called an appropriate modelling approach.
5.3 Results

5.3.1 Outlet air-temperature

The predicted values of outlet air temperature ($T_{out}$), volume rate ($\dot{V}$) at 11 meter height (Figure 9) are shown below together with the measured data. CFD models with indication “Exp. BC” and “ESP-r Conv. BC” are respectively based on method1 and method2 (See 5.2 and Appendix 4). The simulations are performed for a period of 7 hours from 9:00h until 16:00h on 15-12-2009. Note that ESP-r indicated in the following graphs are the BES+AFN models.

![Outlet air temperature](image)

Figure 38: Outlet air temperature ($T_{out}$) at 11 m height according to Figure 9.

Also, the air temperature entering the chimney at 0.25 m is shown here together with the ambient air temperature ($T_{ambient}$). Note that the $T_{out}$ for all the CFD models is an averaged value of the outlet temperature profile at 11 m height.

More comprehensive data about the outlet air temperature profiles and vertical air velocity profiles one can find in Appendix 8 & 9. The measured entering air temperature ($T_{inlet}$) is been assigned to all the modelling approaches as an input parameter. Note that the volume rate for all the CFD models is based on the averaged value of the vertical air velocity profile (Appendix 8 & 9). Besides, $T_{out}$ was calculated for all the CFD models by averaging the outlet air temperature profile at 11 meter height including the absorber and glazing surface temperatures at the same height. The latter approach was also done for the experimental data.

5.3.2 The volume rate

The volume rate from the experiments is based on the averaged vertical air velocity calculated by Equation 9. Also, there is assumed that the cross-sectional area is 0.5 m²
and the converting from m$^3$/s to m$^3$/h is done by multiplying the values of m$^3$/s by 3600 seconds.

![Graph](image1.png)

**Figure 39:** This graph shows the volume rate at 11 meter height. Note that the volume rate from all the CFD models is based on the averaged value of the vertical air velocity profile at 11 meter height.

### 5.3.3 The surface temperatures

Figure 40 and Figure 41 show the surface temperatures of respectively the absorber and the glazing. These surface temperatures are the temperatures at 11 meter height inside the chimney (Figure 9). See for more detail information on surface temperatures Appendix 10 & 11.

![Graph](image2.png)

**Figure 40:** This graph show the surface temperature of the absorber at 11 m height inside the chimney.
5.3.4 Convective heat transfer coefficients

According to the predicted local surface temperature and the air temperature at 0.125 meter in depth and also by knowing the surface convective heat-fluxes (Appendix 4), the local CHTC's of the absorber ($h_{abs}$) and the glazing ($h_{glz}$) have been calculated for the ESP-r and CFD models at 12:30h. Next, these results are shown in Figure 42 & Figure 43 and these are set against the $Y^*$. $Y^*$ is the dimensionless height which can be calculated by $Y/H$ for which the total height (H) of the chimney is 11 m and this is set to the horizontal axis. On the logarithmic vertical-axis the $h_{glz}$ and $h_{abs}$ in W/m²K are set.
5.3.5 Nusselt number and Rayleigh number

Based on Equation 4, Equation 5 and Equation 6 the Rayleigh number (horizontal axis) and the averaged Nusselt number (left vertical axis) for the CFD models are calculated (Nu’s) and the results are shown here (Figure 44 and Appendix12). At the same time the \( \Delta T_{\text{air}} \) (or \( T_o \)) over the chimney for CFD, ESP-r and experiment is shown here (right vertical axis). The vertical dotted lines (blue lines) indicate different simulations. \( {\text{Figure 43: This graph shows the calculated CHTC (convective heat transfer coefficient) for the absorber surface using different modelling approaches based on the predicted } T_{\text{abs}}, T_{\text{AIR}} \text{ at 0.125m depth. Note that the heat-flux of absorber surface was not measured and its value is based on the values in Appendix 4 CFD method 2.}} \)

\( \frac{Y^*}{Y/H} \) indicates the measured and predicted values at 10:30h. From the CFD results based on CFD method2 (Appendix 4) an averaged Rayleigh number (\( R_{\text{aq}} \)) of 4.02x10^8 was calculated. \( {\text{Figure 43: This graph shows the calculated CHTC (convective heat transfer coefficient) for the absorber surface using different modelling approaches based on the predicted } T_{\text{abs}}, T_{\text{AIR}} \text{ at 0.125m depth. Note that the heat-flux of absorber surface was not measured and its value is based on the values in Appendix 4 CFD method 2.}} \)

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Furthermore, based on the Bar-Cohen and Rohsenow (Table 2) for parallel plate problem the averaged Nusselt number for 3 different averaged Rayleigh numbers – as indicated earlier here – is calculated and the results of this analyses is also shown in Figure 44.
5.3.6 Energy balance absorber surface

According to ESP-r Model4 based on the Alamdari-Hammond with a $C_d$ of 0.42 the energy balance at the absorber surface is estimated (Figure 45). Since the absorber is adiabatic at the outdoor region (Figure 24), no heat transfer takes place from or to this direction. On the horizontal axis the time is set against the vertical axis for the occurring heat-fluxes ($q^*$).

The negative sign means that the energy transfer is from the absorber surface to the air and the opposite surfaces (towards glazing in case of long wave radiation).
5.3.7 Energy balance glazing surface

According to ESP-r Model4 based on the Alamdari-Hammond with a C_d of 0.42 the energy balance at the glazing surface inside the chimney is estimated (Figure 46). Since the glazing is set as an exterior boundary condition at the outdoor region (Figure 24), exterior convection occurs at the outside surface of the glazing (Convection Outside) transfer from the glazing surface to the surroundings. On the horizontal axis the time is set against the vertical axis for the occurring heat-fluxes (q”).

![Figure 46: This graph shows the energy balance which is calculated by ESP-r (BES+AFN) Model4 with C_d of 0.42. The heat transfers occur at the glazing surface inside the chimney.](image)

In addition, the convection heat-flux which is shown here (also in Figure 45) was also used as heat-flux boundary condition to CFD models based on Method2.

5.3.8 Uncertainty analysis

In Figure 47 the equation \( P_{\text{gain}} = \rho \cdot V \cdot \Delta \text{C}_p \cdot (T_{\text{OUT}} - T_{\text{IN}}) \) is used in order to calculate the heating power gain of the air which goes through the chimney (see 5.2.4). Also the results from different models are used to calculate the predicted heating power gain with the same equation shown here.
Figure 47: This graph shows the calculated heating power gain for different modelling approaches as well as the measured data. The uncertainty band of the measured data is included here (gray area).
5.4 Discussion

5.4.1 Limitations

The following modelling limitations were of concern when validation study was performed:

1. The hydraulic opening (Figure 7) from the experimental set-up couldn’t be modelled in ESP-r because no air flow component was found which could ascribed the behaviour of this valve according to the situation at hand.

2. Also an alternative solution didn’t work out which is adding the measured vertical air velocity or static pressures to the BES+AFN models as boundary condition via the ‘imposed measured data’ tool. For these parameters no tools were available in ESP-r.

3. And finally the vertical air velocity profile which was measured is based on different outdoor conditions (Appendix 2). Standard Deviation of 14% might be different for other weather conditions. Therefore continuous measurements are needed to collect as many as possible different reference conditions for which the vertical air velocity is measured. This will provide a better understanding of the averaged vertical air velocity.

5.4.2 Modelling BES+AFN

A comprehensive discussion on the uncertain input parameters to different modelling approaches is already shown (Kalyanova et. al, 2005, p. 28). In different sources on fluid dynamics (Daugherty et. al, 1965) for different situations one can find values for the discharge coefficient, friction and the local dynamic losses. However, it is important to express that these values are based on circular cross-sectional area and are meant for fully developed flows. The WSC system of the experimental set-up has however a rectangular cross-section and a fully developed flow may not occur (Figure 42&Figure 43).

In air flow network (AFN) the massflow through a component is calculated by estimating the total pressure difference over the component and by assuming a value for the discharge coefficient or local dynamic losses and frictions. Since these values were uncertain, this study shows the influence of a low discharge coefficient \( (C_d=0.42) \) above a choice of a high discharge coefficient \( (C_d=0.65) \) for the openings at the inlet and outlet of the WSC system. Furthermore, one of the difficulties of modelling an application which has a solar driven force based on natural ventilation is the correct estimation of the CHTC’s (convective heat transfer coefficients) in order to predict the systems thermal performance. In ESP-r (BES+AFN) for the calculation of the CHTC’s the Alamdari-Hammond and Khalifa-Marshall correlations were used. However the empirical relations by which these correlations estimate the CHTC’s are based on experimental set-up’s which are different from the WSC system (Beausoleil-Morrison, 2000).

Moreover, the physical parameters of the construction material are based on the collected data from the supplier who get their material properties from different conditions other than for what these materials are applied in practice. The optical properties did show a good
agreement with the experimental data, and therefore this study expects that these data (used in ESP-r as input parameter) has good agreement with reality.

5.4.3 Results BES+AFN

The results in Figure 38 and Figure 39 show no differences between the models based on Khalifa-Marshall (K-M) and Alamdari-Hammond (A-H) with \( C_d \) of 0.42. For volume rate at 12:30h the same situation is observed. However, there are significant differences between the A-H model with \( C_d \) of 0.65 and the model with a \( C_d \) of 0.42. The maximum outlet air-temperature \( (T_{\text{outlet}}) \) was 29 °C predicted by the models with \( C_d \) of 0.42. Here the model with a \( C_d \) of 0.65 underestimates the \( T_{\text{outlet}} \) by a difference of 2.4 Kelvin. The volume rate of the models with 0.65 is always the largest namely; 3360 m³/h while the models with a \( C_d \) of 0.42 predict a volume rate of 2370 m³/h.

The results in Figure 40 and Figure 41 show a maximum absorber temperature of 55.3 °C and a maximum glazing temperature of 53.4 °C. These values are predicted by the A-H model with a \( C_d \) of 0.42. K-M and A-H have the largest differences; a maximum of 4.4 °C for the absorber temperature and a maximum of 3.4 °C for the glazing temperature.

Figure 42 and Figure 43 show more or less an independent character of convection heat transfer coefficient as function of the chimney’s height with averaged values of 3.9 W/m²K and 4.6 W/m²K respectively for the absorber and the glazing.

Figure 44 shows that for low Rayleigh numbers the effect of the \( C_d \) on the \( \Delta T_{\text{air}} \) is difficult to distinguish. However as the \( \Delta T_{\text{air}} \) increases the Rayleigh number increase and the difference between the BES+AFN models with a \( C_d \) of 0.42 and 0.65 increase as well.

Figure 45&Figure 46 show respectively the energy balance over the absorber and the glazing surface. A maximum convective heat-flux of -100 W/m² for the absorber as well as for the glazing is estimated. For the external convection at the glazing surface a maximum value of -38 W/m² is found. The long wave radiation at the glazing has its maximum – other than the convective, conductive and short radiative heat-fluxes at 12:30h – earlier on the day at 10:30h with a maximum flux of 16.5 W/m². The maximum short-wave radiation which hits the absorber surface is found to be 110 W/m², while the short-wave radiation at the glazing surface comes to a maximum of 39 W/m². Also the amount of long wave radiative heat-flux by absorber is found to be -29 W/m². Based on the latter value a convective heat loss of 660 Watt (16.5 m²) at the glazing surface can be calculated. Also, in section 3.4.8 a conductive loss of 518 Watt was predicted. However BES+AFN take also into account the heat loss due to the forced convection by wind. Therefore this 142 W/m² extra loss might be the contribution of the wind.

5.4.4 Modelling CFD

For the CFD simulations 2 different methods are applied. Method1 sets the measured vertical solar irradiation inside the chimney (SY_IN) (Figure 10) as uniform and constant
heat-flux boundary condition to the absorber. Also 5% of the incident vertical solar irradiation outside (SY_OUT) sets a uniform and constant heat-flux boundary condition to the glazing surface. As a consequence CFD Method1 (Figure 19) is predicting unrealistic absorber temperatures which exceed 100 °C. Also “It was found that, for linear wall heat flux distribution and a given heating intensity, the maximum temperature at the wall can be minimized by making the wall heat flux slightly larger in the lower sections of the channel” (Hernández et. al, 2003, pp. 806-807). Thus the way of implementing the heat-flux boundary conditions in CFD might be closer to the reality (3.5) when partial shading is taken into account and so a non-uniform heat-flux boundary condition is applied to the absorber.

Besides no radiation models were used in these CFD simulations. Some researchers state (Nouaniqué et. al, 2009) that radiation is not neglectible for low Rayleigh numbers and surfaces with high emissivity, one can state that the implementation of a radiation model to the current CFD simulations will not cause major differences in predictions. The reason for the latter statement is fourfold; first the Rayleigh numbers were found to be high $10^5$; secondly the emissivity of the surfaces in the chimney are very low (around 0.05); thirdly by using the Equation 15 in (Ong, 2003, p. 1055) an average radiative heat transfer coefficient of 0.3 W/m²K was estimated, between the absorber and the glazing and finally from the BES+AFN energy balance calculations (Figure 45&Figure 46) a very small contribution of the long wave radiation inside the chimney is shown.

However the heat transfer by conduction may play a more important role in the prediction of the thermal performance of the WSC system. The reason for this statement is that in the past some researchers (Kim et. al, 1990) showed that the maximum decrease in averaged Nusselt number due to wall conduction is 22% and these effects are more significant for low Grashof number than high Grashof number flows. Also, from the energy balance calculations (Figure 45&Figure 46) one can observe a significant amount of heat transfer by conduction (especially for the glazing surface). This might be the reason of the higher glazing temperature predictions by all the models (Figure 41). In addition, other researchers stated in the past that obstacles like square ribs in the path of the air fluid will decrease the convective heat transfer on the vertical walls (Tanda, 1997), which might interfere with the conductive heat transfer through the glazing. The heat transfer performance of especially the glazing (with its frame connections) will have convective heat transfer coefficient (CHTC) distributions which will probably fluctuate over the height of the glazing (Figure 48). The graph on the left shows a CHTC distribution for a smooth channel and on the right the distribution of a channel with on one side obstacles (or square ribs).
The presence of ribs was found to alter heat transfer considerably, causing thermally regions just upstream and downstream of each protrusion. As a consequence the CHTC value of a ribbed channel turned out to be lower than that of the corresponding smooth channel. In addition to the latter conclusion (Tanda, 1996) this might be the second reason – beside conduction through the glazing surfaces – that there are discrepancies between the predicted and measured glazing temperatures (Figure 41).

Finally, the study on the CFD models further focuses on the effects of the boundary resolution (Gan, 2009) and the difference between the two different turbulence models which have shown good predictive capacity in the past; low-Reynolds k-epsilon turbulence model (Fedorov et. al, 1997) and standard k-epsilon turbulence model. Unfortunately, the convective heat-flux was not measured, which does not allow to calculate a reference convective heat transfer coefficient for comparison with the predicted data by different modelling approaches.

### 5.4.5 Results CFD

From Figure 38 and Figure 39 a maximum $T_{\text{outlet}}$ of 41.3 °C is found for Model1 with standard k-omega turbulence model; shortly s-ko (Appendix 4, Method2, 12:30h). A minimum $T_{\text{outlet}}$ of 36.6 °C is however predicted by Model3 with low-Reynolds k-epsilon turbulence model (Method2); shortly low-Re k-e. At the same time a maximum volume rate of 2210 m$^3$/h is predicted by Model3 low-Re k-e (Method1). Finally, a minimum volume rate of 870 m$^3$/h is predicted by CFD Model2 low-Re k-e (Method2).

The results in Figure 40 and Figure 41 show a maximum $T_{\text{abs}}$ of 116 °C that is estimated by Model1 sk-o (Method1). However a minimum $T_{\text{abs}}$ of 51 °C is shown by Model3 low-Re k-e (Method2). Also the maximum $T_{\text{glz}}$ of 66.6 °C was predicted by Model1 sk-o (Method2) while the minimum of 44.6 °C was found by Model3 low-Re k-e (Method1).
The Figure 42 and Figure 43 show the local heat transfer coefficients for the absorber ($h_{abs}$) and the glazing ($h_{glz}$) are height depended. The averaged $h_{glz}$ is calculated and the values are 3.7, 3.6, 4.8 and 6.3 W/m²K for CFD Method1 and 3.7, 4, 3.8, 4.8 W/m²K for the CFD Method2. These values are respectively from Model1 low-Re k-e, Model1 sk-o, Model2 and Model3. Also the averaged $h_{abs}$ is calculated and the values are 5.6, 6.2, 7.2 and 8.6 W/m²K for CFD Method1 and 3.4, 3.9, 3.7 and 4.9 W/m²K for the CFD Method2. Again, the latter values are respectively from Model1 low-Re k-e, Model1 sk-o, Model2 and Model3.

Figure 44 shows that for asymmetric (see Appendix 4 Method2 at 10:30h, method1 at 12:30h) and symmetric (see Appendix 4, Method2 at 12h30) constant boundary heat-flux conditions different Rayleigh number (Ra) can be calculated for all the CFD models. It is obvious that Model3 predicts higher values of averaged Nusselt number (Nu) compared to other CFD models. The averaged Nu number of Model3 is always with a ratio of 1.5 larger then predictions with Model1 for approximately the same Ra number. Also for the smallest Ra number one can observe that the $\Delta T_{air}$ which is estimated by Model 2 - is closest to the measured data with an over-prediction of 2 Kelvin. Next, as the temperature difference ($\Delta T_{air}$) over the height of the chimney increases the Rayleigh number and the averaged Nu number will increase. For all models the value will increase but each with their own slope as was expected (Lartigue et. al, 2000), (Kim et. al, 1990) and (Wang et. al, 2009). The k-omega turbulence model has higher Nu number predictions compared to the same model with the low-Re k-epsilon turbulence model. For $4.02x10^8< Ra<1.24x10^9$ the standard k-omega has even higher averaged Nu number predictions than Model2 which has extended boundaries at the inlet and the outlet. For $Ra \approx 1.24x10^9$ (Appendix 4, Method2, 12:30h) Model1 with k-omega turbulence model shows the best agreement with the measured $\Delta T_{air}$. For $1.24x10^9< Ra<2.96x10^9$ Model2 has higher averaged Nu number predictions compared to low-Re k-epsilon and standard k-omega based on Model1. For $Ra \approx 2.96x10^9$ (Appendix 4, Method1, 12:30h) Model3 has the best agreement with $\Delta T_{air}$ with an over-prediction of 2.2 Kelvin. Furthermore, the Bar-Cohen and Rohsenow’s (BC&R) empirical equation shows good agreement with the calculated averaged Nu number by all the CFD simulations. At Ra $(4.02x10^9)$ BC&R show the same averaged Nu number as Model3 for low Ra numbers, however as the Ra number increases the BC&R estimates more or less the same averaged Nu number as the Model1 based on standard k-omega or low-Reynolds k-epsilon turbulence models.

5.4.6 Results uncertainty analysis

Based on the ratio between the predicted ($P_{predicted}$) and measured heating energy gain (Figure 47) ($P_{measured}$) - all the BES+AFN models and the CFD Method 2 except CFD Model3 based on method 2 (see Appendix 4) - will always underestimate the $P_{measured}$. The latter modelling approaches will stay close to the lower uncertainty band of $P_{measured}$. CFD Model3 based on Method 2 follows the evolution of the $P_{measured}$ in the lower part of the
uncertainty band except the times between 9:00h to 10:00h and after 16:00h. Besides, CFD simulations based on Method 1 will approximately overestimate the $P_{\text{measured}}$. Only Model2 shows several times good agreement with the upper level part of the uncertainty band. However the latter model doesn’t follow the evolution of the $P_{\text{measured}}$ as good as the CFD Model3 (Method2) does.

The latter scattering of data also shows the major effect of different choices of modelling boundary resolutions and turbulence models in CFD. As the boundary resolution increases for the CFD simulations the predicted heating power gain increases. This is due to the higher volume rate, which is probably the consequence of the hydrostatic pressure difference (due to the environment) which is not present in boundary resolutions of Model1 and Model2 (Fluent, 2006, pp. 7-23). At 12:30h a maximum difference (compared to averaged measured data) of 1990 Watt is found for CFD simulations based on the CFD Method2, while for the CFD simulations based on Method1 this difference is 2580 Watt. Also, no significant differences were found for $P_{\text{predicted}}$ by the two different turbulence models, namely; low-Reynolds k-epsilon and standard k-omega turbulence models. Furthermore, the CFD Model2 show lower estimation of $P$ compared to other modelling resolutions, especially for periods with higher boundary heat-fluxes (see Appendix 4). For CFD simulations based on Method1 the latter finding is always true, while for Method2 the latter statement is valid between 12:00h until 15:00h.
A validation study was performed using 2 different modelling approaches based on BES+AFN and CFD. The results from the outdoor experimental set-up (3.3) were used as input parameters (weather files) to different modelling approaches. Besides, for the BES+AFN modelling approach 3 different models were generated with different discharge coefficients ($C_d = 0.42$ and $C_d = 0.65$) and convective heat transfer correlations (Khalifa-Marshall and Alamadari-Hammond). Also, based on the CFD approach 3 different computational domains were generated, in which for the simplest model (Model1) two different turbulence models were used (low-Reynolds k-epsilon and standard k-omega).

The choice of a correct $C_d$ has shown its major impact on the predictions by the BES+AFN modelling approach. There can be concluded that more attention must be paid on the choice of $C_d$ rather than the choice of a convective heat transfer correlation when modelling the WSC system. Furthermore, BES+AFN showed more or less a constant $h_{glz}$ and $h_{abs}$ over the height which is in contrast to the estimated $h_{glz}$ and $h_{abs}$ by CFD models. The entrance effect in the CFD models diminishes rapidly with the distance from the entrance and vanishes at a distance larger than 6.6 meter. The averaged values of $h_{glz}$ and $h_{abs}$ by BES+AFN approach are respectively 4.6 W/m²K and 3.9 W/m²K, while in CFD these values are 4.3 and 5.4 W/m²K. Next, the calculated energy balance in BES+AFN showed that the heat loss via convection at the glazing surface may play an important role on the thermal performance of the WSC. The maximum amount of this external convective heat loss was 40 W/m². So, attention must be paid to the wind and the glazing system when designing such a system. With regard to the CFD models the turbulence models showed no significant differences in estimating the thermal performance of the WSC system. But the difference between these two turbulence models increases as the Rayleigh number (Ra) increases. Also, it is shown that the largest boundary resolution (Model3) always has higher estimation of Nusselt number (Nu), while the Nu number by all the CFD model increases as the system’s Rayleigh number increases. One can conclude that in general as the boundary resolution increases the CFD models show a decrease in the outlet air temperatures with at the same time an increase in the flow rate and the averaged Nu numbers. The latter differences will be more significant by increasing Ra. Furthermore, there was demonstrated that the convective heat transfer on the heated wall can be well described by a conventional correlation like Bar-Cohen and Rohsenow. Finally, based on the outcome of the simple uncertainty analysis and the predicted values of $T_{air}$ and $\bar{V}$ one can conclude that the largest computational domain CFD Model3 has the best agreement with the measured heating power gain. However more attention must be paid to the discrepancies of $T_{abs}$ and $T_{glz}$ which respectively were underestimated and overestimated by 16.5 °C.
6. Conclusions and recommendations

6.1 Conclusions

“What is the minimum modelling complexity which is necessary to simulate a WSC system in a physically proper way?”. There has been found that at least an AFN+BES approach must be used in order to model a naturally ventilated WSC (wall solar chimney) system in a physically proper way.

“What are the uncertain input parameters to different modelling approaches and what are their influences on different PI’s of the WSC system?” In BES+AFN models the discharge coefficient (C_d) of the inlet and the outlet areas are found to have significant effect on the PI’s (performance indicators) of the WSC system. In CFD the choice of the boundary resolution does affect the predicted PI’s. Also in CFD the choice of the turbulence models (low-Reynolds k-epsilon or standard k-omega) and the symmetrical or the asymmetrical distribution of the heat-fluxes does affect the PI’s significantly.

“What are the appropriate modelling approaches for each PI of WSC-system?” From this study the minimum modelling complexity - which is required to estimate the thermal performance of the hybrid ventilated WSC system properly - is found to be a combination of the convective heat-fluxes of the BES+AFN models and the CFD model with the largest boundary resolution. A good agreement is found for the outlet air temperatures and the air volume rate. Still the glazing and the absorber surface temperatures don’t show a good agreement compared to the measured data.

Even though all the sub-research questions have been answered, further study is required in order to develop a basis guideline for the modelling of all the possible WSC system designs. The reason is that this study encountered significant limitations on the process of measurements, modelling and as a consequence limitations in the validation study. Also, the results are based on only one design of the WSC system and the influences of other design parameters (location, orientation, aspect ratio etc.) aren’t part of the study.
6.2 Recommendations

In future measurement based on CASE A (Table 3) need to be done in order to avoid the limitations on the modelling of the hydraulic opening. In any case, still the vertical air velocity profile needs to be measured in order to represent the air volume rate in the shaft in a more accurate way.

Although a more accurate experimental study needs to be performed in future it is also important to indicate that the conclusions made here are based on a single WSC system with certain properties (material properties, location and orientation, aspect ratio and the weather conditions). Therefore, it is advisable to perform the same study on a totally different situation. This is certainly one point to analyze in the future because this might have influences on the range of predictions by the BES+AFN as well as the CFD approach. If it appears that the result of this parametric analysis isn’t in good agreement with the measured data, then a sensitivity analysis to all the input parameters might be the solution to check what other input parameters cause these differences on the prediction of the PI's of the WSC system.

Finally, on the basis of the findings in this study a combination of BES+AFN+CFD may provide the best solutions with regard to the reality and the reason for this is to my opinion threefold: First with BES+AFN one can easily include the environment to the models with all the corresponding dynamic quantities like solar radiation, conduction through construction and wind effects (because of the external convection on glazing surface) from and to the outdoor environment. Second, it is obvious that as the hydrostatic pressure is added to the CFD models - by enlarging the boundary resolution – it produces the best agreement with the measurements only when the convective heat-fluxes from BES+AFN models are used as boundary condition to CFD models. Third, to my opinion - as intermediate user of ESP-r (BES+AFN) and Fluent (CFD) - each modelling approach has shown their advantages but also their drawbacks for the purpose of design of the WSC system. CFD may better simulate the airflow and the convective heat transfer coefficients inside the chimney but the boundary resolution of BES+AFN can help to include the conduction and the non-uniform radiation heat-fluxes on the absorber due to shading which seemed to have significant influences on the performance of the WSC system. However this approach is still in stages of research and at the moment not the first choice for the decision maker during a design process, because it requires a lot of modelling expertise and computing time.
References


Appendix 1: Experimental set-up

The cross-sectional area (horizontal cross-section) shows how the chimney is constructed (Figure 49). There is an outer-pane which includes glazing and the frame construction. There is an inner-pane (includes also the side walls on the left and the right) which is well insulated and has an absorber with low emissivity at the inside surface.

Beside the geometrical dimensions, a list of important material properties per construction is shown below (Table 7 & Table 8).

The horizontal cross-section of the chimney’s shaft or cavity is shown here. This is constructed from an outer-pane which includes the glazing frame construction and the glazing (75% of the area). The inner-pane and the side walls (left and right) are constructed using (from outside to inside) a profiled steel sheet or the so-called absorber, Insulation and Underlayment and finally the absorber with low emissivity. See also Figure 7.

The physical properties of the inner-pane and the outer-pane are shown in tables below.

Table 7: Material properties of construction of the inner-pane are shown here.

<table>
<thead>
<tr>
<th>Inner-pane construction</th>
<th>Conductivity [W/mK]</th>
<th>Density [kg/m³]</th>
<th>Thickness [mm]</th>
<th>Absorption factor</th>
<th>Emissivity factor</th>
<th>Nett area [m²]</th>
</tr>
</thead>
<tbody>
<tr>
<td>From outside to inside</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Profiled steel</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Rockwool 433 DUO</td>
<td>0.035</td>
<td>45</td>
<td>120</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Underlayment</td>
<td>0.017</td>
<td>500 - 600</td>
<td>16</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Rockwool 433 DUO</td>
<td>0.035</td>
<td>45</td>
<td>120</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Absorber</td>
<td>-</td>
<td>-</td>
<td>0.5</td>
<td>0.95 ±0.01</td>
<td>0.05 ±0.02</td>
<td>21</td>
</tr>
</tbody>
</table>

Table 8: Material properties construction of the outer-pane are shown here. It is a double glazed outer-pane. The manufacturer of the glazing is Saint Gobian Glass Solutions.

<table>
<thead>
<tr>
<th>Outer-pane construction</th>
<th>U-value [W/m²K]</th>
<th>Density [kg/m³]</th>
<th>Thickness [mm]</th>
<th>G-value</th>
<th>Absorption factor</th>
<th>Emissivity factor</th>
<th>Nett area [m²]</th>
</tr>
</thead>
<tbody>
<tr>
<td>From outside to inside</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Diamant SGG glazing</td>
<td>-</td>
<td>-</td>
<td>6</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Cavity (10% air, 90% argon)</td>
<td>-</td>
<td>-</td>
<td>15</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Low-e coating</td>
<td>-</td>
<td>-</td>
<td>6</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Planitherm Total</td>
<td>1.32</td>
<td>27</td>
<td>0.70</td>
<td>0.60</td>
<td>-</td>
<td>-</td>
<td>15.7</td>
</tr>
</tbody>
</table>

Figure 49: The horizontal cross-section of the chimney’s shaft or cavity is shown here.
Appendix 2: Vertical Velocity Profile

There is been an attempt to measure the vertical air velocity profile in order to calculate the mean vertical air velocity. In Figure 1 one can see how this measurements are performed. These measurements are carried out on 30-04-2010. The sampling has been done by moving the sensors to or from the absorber in 10 different positions. Eight velocity sensors on the same level were used for this purpose. The maximum velocity is observed for the area on the right.

Consequently, one can calculate the volume rate since the cross-sectional area is 0.5 m$^2$. Standard deviation of 14% can be calculated from these measurements on 4 meter height. The maximum value of these measurements is always on the right side of the chimney which is measured at 4 meter height. In figure 2 the scattering of the measured air velocities is shown.

Also in figure 3 the behaviour of the hydraulic opening – which is meant to keep the air velocity in the shaft at 1 m/s – is shown.
2- Scatter of the measured vertical air velocity distribution

3- Control signal of the hydraulic opening
Appendix 3: Method of solution CFD

Assumptions

The following criteria are assumed for the simulation with the CFD models:

- Pressure-Velocity Coupling: SIMPLE
- Numerical discretization:
  - Pressure: Body Forced Weighted
  - Momentum: 2th order Upwind
  - Turbulent Kinetic Energy: 2th order Upwind
  - Turbulent Dissipation Rate: 2th order Upwind
  - Energy: 2th order Upwind
- Density: Boussinesq approximation
  - Fluid: air
  - Density: 1.1649 kg/m$^3$
  - Specific heat: 1006.5 J/kgK
  - Thermal conductivity: 0.026341 W/mK
  - Viscosity: 1.868e-5 kg/ms
  - Thermal Expansion Coefficient: 0.0033 1/Kelvin

Convergence method

1. Set the initial conditions to zero except the initial temperature which must be the same as the inlet air temperature which is an input parameter;
2. Run a steady state simulation of 10 iterations with a low Rayleigh by decreasing the gravitational force (g) by 1/10 of the actual number (thus g = - 0.981 m/s$^2$);
3. Next, change the solver to an implicit (pressure-based) second-order unsteady solver;
4. Now run the simulation with a timestep size of 0.01 seconds for duration (a physical time) of 10 seconds (15 iteration per timestep);
5. Change the Rayleigh number by setting the gravitation force to -9.81 m/s$^2$;
6. Run the simulation with a timestep size of 0.1 seconds for a physical time of 300 seconds (or 5 minutes) with 15 iteration per timestep.
## Appendix 4: Boundary condition values

<table>
<thead>
<tr>
<th>Time</th>
<th>$q''_{\text{absorber}}$ [W/m$^2$]</th>
<th>$q''_{\text{gasing}}$ [W/m$^2$]</th>
<th>$T_{\text{inst}}$ (Kelvin)</th>
<th>$T_{\text{broadband}}$ (Kelvin)</th>
<th>$q''_{\text{absorber}}$ [W/m$^2$]</th>
<th>$q''_{\text{gasing}}$ [W/m$^2$]</th>
<th>$T_{\text{inst}}$ (Kelvin)</th>
<th>$T_{\text{broadband}}$ (Kelvin)</th>
</tr>
</thead>
<tbody>
<tr>
<td>9h</td>
<td>10</td>
<td>1</td>
<td>293.04</td>
<td>293.04</td>
<td>2</td>
<td>16</td>
<td>293.04</td>
<td>293.04</td>
</tr>
<tr>
<td>10h</td>
<td>134</td>
<td>30</td>
<td>292.44</td>
<td>292.44</td>
<td>20</td>
<td>1</td>
<td>292.44</td>
<td>292.44</td>
</tr>
<tr>
<td>10h30</td>
<td>299</td>
<td>55</td>
<td>293.11</td>
<td>293.11</td>
<td>44</td>
<td>18</td>
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</tr>
<tr>
<td>11h</td>
<td>351</td>
<td>65</td>
<td>292.41</td>
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<td>82</td>
<td>66</td>
<td>292.41</td>
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</tr>
<tr>
<td>11h30</td>
<td>389</td>
<td>70</td>
<td>292.73</td>
<td>292.73</td>
<td>91</td>
<td>79</td>
<td>292.73</td>
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</tr>
<tr>
<td>12h</td>
<td>411</td>
<td>73</td>
<td>293.29</td>
<td>293.29</td>
<td>88</td>
<td>92</td>
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<tr>
<td>12h30</td>
<td>420</td>
<td>74</td>
<td>293.98</td>
<td>293.98</td>
<td>100</td>
<td>100</td>
<td>293.98</td>
<td>293.98</td>
</tr>
<tr>
<td>13h</td>
<td>413</td>
<td>72</td>
<td>294.44</td>
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<td>100</td>
<td>100</td>
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<tr>
<td>13h30</td>
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<td>294.79</td>
<td>97</td>
<td>98</td>
<td>294.79</td>
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</tr>
<tr>
<td>14h</td>
<td>352</td>
<td>62</td>
<td>292.25</td>
<td>292.25</td>
<td>94</td>
<td>97</td>
<td>292.25</td>
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<tr>
<td>15h</td>
<td>227</td>
<td>43</td>
<td>291.71</td>
<td>291.71</td>
<td>67</td>
<td>62</td>
<td>291.71</td>
<td>291.71</td>
</tr>
<tr>
<td>16h</td>
<td>42</td>
<td>5</td>
<td>293.23</td>
<td>293.23</td>
<td>10</td>
<td>8</td>
<td>293.23</td>
<td>293.23</td>
</tr>
</tbody>
</table>
Appendix 5: Journal File

Example Journal file CFD Model1

/file r-c Model1.msh
/define mo en yes no no no
/define mo vi ke-s yes n-w-t e yes
/define mat chan air , yes boussinesq 1.1649 yes , 1006.5 yes , 0.026341 yes , 1.868e-5 , , , yes 0.00330033 , ,
/define ope gra yes , -0.981
/define ope ope-te 298.29
/define bo w glazing , , , , yes heat-flux , 73 , ,
/define bo w absorber , , , , yes heat-flux , 411 , ,
/define bo p-i , , , , , , 293.29 , , no no no yes 1 0.444
/define bo p-o , , , , , , 293.29 , , no no no yes 1 0.444 ,
/solve set d-s e 1
/solve set d-s k 1
/solve set d-s m 1
/solve set d-s p 13
/solve set d-s t 1
/solve mo r conv 0.000001 0.000001 0.000001 , 0.000001 0.000001
/solve mo r plot yes
/solve mo r print yes
/solve mo r window 4
/solve ini set-d x-v 0 y-v 0
/solve ini set-d temp 293.29
/solve ini ini-f
/file w-c-d Model1_12h
/surface p-a inlet 100 0 0.5 0.25 0.5
/surface p-a outletx 100 0 10.5 0.25 10.5
/surface p-a midline 100 0.125 0 0.125 11
/solve it 10
/define mo unsteady-1 yes
/solve mon surface set-m Tout temperature (outletx) , , yes ToModel1_12h_mo.txt yes "Facet Average"
/file a-s d-f 100
/file a-s o-e-f yes
/solve set time-step 0.01
/solve dual 1000 15
/define ope gra yes , -9.81
/solve set time-step 0.1
/solve dual 2000 15
/file w-c-d Model1_12h
/plot plot , Tinlet1_12h.txt , , , temperature yes , , inlet ,
/plot plot , Vinlet1_12h.txt , , , y-velocity yes , , inlet ,
/plot plot , Toutlet1_12h.txt , , , temperature yes , , outletx ,
/plot plot , Voutlet1_12h.txt , , , y-velocity yes , , outletx ,
/plot plot , T1abs1_12h.txt , yes 0 1 , , temperature absorber ,
/plot plot , T1glz1_12h.txt , yes 0 1 , , temperature glazing ,
/plot plot , Tmidline1_12h.txt , yes 0 1 , , temperature midline ,
/exit yes
Appendix 6: ESP-r Climate File

Climate file ESP-r

*CLIMATE (15 -12-2009)
# ascii climate file from ../dbs/Model1.climate binary db,
# defined in: ../dbs/mookweer.a
# col 1: Diffuse solar on the horizontal (W/m**2)
# col 2: External dry bulb temperature   (Tenths DEG.C)
# col 3: Direct normal solar intensity   (W/m**2)
# col 4: Prevailing wind speed   (Tenths m/s)
# col 5: Wind direction     (clockwise deg from north)
# col 6: Relative humidity               (Percent)
Mook                            # site name
2009,51.45,5.52,0,   # year, latitude, long diff, global horiz rad flag
1,365    # period (julian days)
---------------------------------------------------------------
* day 15 month 12
-21,-33,0,15,55,17
-21,-34,0,16,59,19
-22,-33,0,14,64,17
-22,-34,0,20,69,18
-21,-34,0,18,66,18
-21,-36,0,19,64,18
-21,-35,0,19,63,19
-21,-35,10,18,83,19
-19,-36,31,16,65,18
-16,-37,44,18,63,18
6,-38,37,18,64,18
24,-37,93,21,57,18
28,-37,114,21,78,18
30,-37,63,23,74,17
28,-37,27,21,73,18
20,-37,9,19,72,17
5,-37,0,19,64,18
-13,-37,0,16,69,18
-16,-38,0,16,69,18
-16,-38,0,14,61,18
-18,-40,0,10,49,17
-17,-38,0,20,78,18
-17,-37,0,11,65,20
-18,-37,0,18,62,17
----------------------------------------------------------------
Appendix 7: QA Report ESP-r

ESP-r model4 (13 thermal zones)

This is a synopsis of the model 1 zone defined in Model4v1.cfg generated on Wed Jun 9 09:16:14 2010. Notes associated with the model are in Model1.log

The model is located at latitude 51.45 with a longitude difference of 5.52 from the local time meridian. The year used in simulations is 2009 and weekends occur on Saturday and Sunday.

The site exposure is typical city centre and the ground reflectance is 0.20.

Project name: Solar Chimney Test set-up
Building address: Moelenhoek Netherlands (Peutz BV)
Building city: Molenhoek
Building Postcode: not yet defined

Building owner name: not yet defined
Building owner telephone: not yet defined
Building owner address: not yet defined
Building owner city: not yet defined
Building owner Postcode: not yet defined

Simulationist name: Ehsan Baharvand
Simulationist telephone: not yet defined
Simulationist address: Technical University of Eindhoven
Simulationist city: Eindhoven
Simulationist postcode: not yet defined
The climate used is: Mook and is held in: ../dbs/Model1.climate and uses
hour centred solar data.

Temporal data is available for this model.

Temporal entities currently used...

<table>
<thead>
<tr>
<th>temporal</th>
<th>generic</th>
<th>associated</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 VerticalIRR VERTSOL Chimney2 Wall-4</td>
<td></td>
<td></td>
</tr>
<tr>
<td>2 VerticalIRR VERTSOL Chimney1 Wall-4</td>
<td></td>
<td></td>
</tr>
<tr>
<td>3 VerticalIRR VERTSOL Chimney3 Wall-4</td>
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</tr>
<tr>
<td>4 VerticalIRR VERTSOL Chimney3 Glazing</td>
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<td></td>
</tr>
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<td>5 VerticalIRR VERTSOL Chimney4 Wall-4</td>
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</tr>
<tr>
<td>6 VerticalIRR VERTSOL Chimney4 Glazing</td>
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<td></td>
</tr>
<tr>
<td>7 VerticalIRR VERTSOL Chimney5 Wall-4</td>
<td></td>
<td></td>
</tr>
<tr>
<td>8 VerticalIRR VERTSOL Chimney5 Glazing</td>
<td></td>
<td></td>
</tr>
<tr>
<td>9 VerticalIRR VERTSOL Chimney6 Wall-4</td>
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<tr>
<td>10 VerticalIRR VERTSOL Chimney6 Glazing</td>
<td></td>
<td></td>
</tr>
<tr>
<td>11 VerticalIRR VERTSOL Chimney7 Wall-4</td>
<td></td>
<td></td>
</tr>
<tr>
<td>12 VerticalIRR VERTSOL Chimney7 Glazing</td>
<td></td>
<td></td>
</tr>
<tr>
<td>13 VerticalIRR VERTSOL Chimney8 Wall-4</td>
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</tr>
<tr>
<td>14 VerticalIRR VERTSOL Chimney8 Glazing</td>
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</tr>
<tr>
<td>15 VerticalIRR VERTSOL Chimney9 Wall-4</td>
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<td>16 VerticalIRR VERTSOL Chimney9 Glazing</td>
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</tr>
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<td>17 VerticalIRR VERTSOL Chimney10 Wall-4</td>
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<td>18 VerticalIRR VERTSOL Chimney10 Glazing</td>
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<tr>
<td>19 VerticalIRR VERTSOL Chimney11 Wall-4</td>
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<tr>
<td>20 VerticalIRR VERTSOL Chimney11 Glazing</td>
<td></td>
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</tr>
<tr>
<td>21 VerticalIRR VERTSOL Chimney12 Wall-4</td>
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<tr>
<td>22 VerticalIRR VERTSOL Chimney12 Glazing</td>
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</tr>
<tr>
<td>23 VerticalIRR VERTSOL Outlet Wall-4</td>
<td></td>
<td></td>
</tr>
<tr>
<td>24 VerticalIRR VERTSOL Outlet Glazing3</td>
<td></td>
<td></td>
</tr>
<tr>
<td>25 Troom SETPTTT loop_15 ALL</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Databases associated with the model:
pressure distributions : ../dbs/Model4v2.pressuredb
materials : /home/Model4/dbs/Model1.materialdb
constructions : /home/Model4/dbs/Model1.constrdb
plant components : plantc.db1
event profiles : /usr/esru/esp-r/databases/profiles.db2.a
optical properties : /home/Model4/dbs/Model1.opticdb

The model includes ideal controls as follows:
Control description: no overall control description supplied

Zones control includes 15 functions.
no zone control description supplied

The sensor for function 1 senses dry bulb temperature in Room.
The actuator for function 1 is the air point in Room.
The function day types are Weekdays, Saturdays & Sundays
Weekday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1
periods.

<p>| | | | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Per</td>
<td>Start</td>
<td>Sensing</td>
<td>Actuating</td>
</tr>
<tr>
<td>---</td>
<td>---</td>
<td>---</td>
<td>---</td>
</tr>
<tr>
<td>1  0.00 db temp &amp; flux</td>
<td>basic control</td>
<td></td>
<td></td>
</tr>
<tr>
<td>100000.0 0.0 20.0 20.0 0.0</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
basic control: max heating capacity 100000.0W min heating capacity 0.0W
max cooling
capacity 100000.0W min cooling capacity 0.0W. Heating setpoint 20.00C
cooling setpoint 20.00C.
Saturday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1
periods.
Per|Start|Sensing |Actuating | Control law | Data
1 0.00 db temp > flux free floating
Sunday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1 periods.
Per|Start|Sensing |Actuating | Control law | Data
1 0.00 db temp > flux free floating
The sensor for function 2 senses dry bulb temperature in Chimney1.
The actuator for function 2 is the air point in Chimney1.
The function day types are Weekdays, Saturdays & Sundays
Weekday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1
periods.
Per|Start|Sensing |Actuating | Control law | Data
1 0.00 db temp > flux free floating
Saturday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1
periods.
Per|Start|Sensing |Actuating | Control law | Data
1 0.00 db temp > flux free floating
Sunday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1 periods.
Per|Start|Sensing |Actuating | Control law | Data
1 0.00 db temp > flux free floating
The sensor for function 3 senses dry bulb temperature in Chimney2.
The actuator for function 3 is the air point in Chimney2.
The function day types are Weekdays, Saturdays & Sundays
Weekday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1
periods.
Per|Start|Sensing |Actuating | Control law | Data
1 0.00 db temp > flux free floating
Saturday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1
periods.
Per|Start|Sensing |Actuating | Control law | Data
1 0.00 db temp > flux free floating
Sunday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1 periods.
Per|Start|Sensing |Actuating | Control law | Data
1 0.00 db temp > flux free floating
The sensor for function 4 senses dry bulb temperature in Chimney3.
The actuator for function 4 is the air point in Chimney3.
The function day types are Weekdays, Saturdays & Sundays
Weekday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1
periods.
Per|Start|Sensing |Actuating | Control law | Data
1 0.00 db temp > flux free floating
Saturday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1
periods.
Per|Start|Sensing |Actuating | Control law | Data
1 0.00 db temp > flux free floating
Sunday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1 periods.
Per|Start|Sensing |Actuating | Control law | Data
1 0.00 db temp > flux free floating
The sensor for function 5 senses dry bulb temperature in Chimney4.
The actuator for function 5 is the air point in Chimney4.
The function day types are Weekdays, Saturdays & Sundays
Weekday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1 periods.
Per | Start | Sensing | Actuating | Control law | Data
1  0.00 db temp > flux free floating

Saturday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1 periods.
Per | Start | Sensing | Actuating | Control law | Data
1  0.00 db temp > flux free floating

Sunday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1 periods.
Per | Start | Sensing | Actuating | Control law | Data
1  0.00 db temp > flux free floating

The sensor for function 6 senses dry bulb temperature in Chimney5.
The actuator for function 6 is the air point in Chimney5.
The function day types are Weekdays, Saturdays & Sundays

Weekday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1 periods.
Per | Start | Sensing | Actuating | Control law | Data
1  0.00 db temp > flux free floating

Saturday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1 periods.
Per | Start | Sensing | Actuating | Control law | Data
1  0.00 db temp > flux free floating

Sunday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1 periods.
Per | Start | Sensing | Actuating | Control law | Data
1  0.00 db temp > flux free floating

The sensor for function 7 senses dry bulb temperature in Chimney6.
The actuator for function 7 is the air point in Chimney6.
The function day types are Weekdays, Saturdays & Sundays

Weekday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1 periods.
Per | Start | Sensing | Actuating | Control law | Data
1  0.00 db temp > flux free floating

Saturday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1 periods.
Per | Start | Sensing | Actuating | Control law | Data
1  0.00 db temp > flux free floating

Sunday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1 periods.
Per | Start | Sensing | Actuating | Control law | Data
1  0.00 db temp > flux free floating

The sensor for function 8 senses dry bulb temperature in Chimney7.
The actuator for function 8 is the air point in Chimney7.
The function day types are Weekdays, Saturdays & Sundays

Weekday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1 periods.
Per | Start | Sensing | Actuating | Control law | Data
1  0.00 db temp > flux free floating

Saturday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1 periods.
Per | Start | Sensing | Actuating | Control law | Data
1  0.00 db temp > flux free floating

Sunday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1 periods.
Per | Start | Sensing | Actuating | Control law | Data
1  0.00 db temp > flux free floating

The sensor for function 9 senses dry bulb temperature in Chimney8.
The actuator for function 9 is the air point in Chimney8.
The function day types are Weekdays, Saturdays & Sundays
Weekday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1 periods.

<table>
<thead>
<tr>
<th>Per</th>
<th>Start</th>
<th>Sensing</th>
<th>Actuating</th>
<th>Control law</th>
<th>Data</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.00</td>
<td>db temp &gt; flux</td>
<td>free floating</td>
<td></td>
<td></td>
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</tbody>
</table>

Saturday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1 periods.

<table>
<thead>
<tr>
<th>Per</th>
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<th>Sensing</th>
<th>Actuating</th>
<th>Control law</th>
<th>Data</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.00</td>
<td>db temp &gt; flux</td>
<td>free floating</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Sunday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1 periods.

<table>
<thead>
<tr>
<th>Per</th>
<th>Start</th>
<th>Sensing</th>
<th>Actuating</th>
<th>Control law</th>
<th>Data</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.00</td>
<td>db temp &gt; flux</td>
<td>free floating</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

The sensor for function 10 senses dry bulb temperature in Chimney9. The actuator for function 10 is the air point in Chimney9. The function day types are Weekdays, Saturdays & Sundays.

Weekday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1 periods.

<table>
<thead>
<tr>
<th>Per</th>
<th>Start</th>
<th>Sensing</th>
<th>Actuating</th>
<th>Control law</th>
<th>Data</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.00</td>
<td>db temp &gt; flux</td>
<td>free floating</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Saturday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1 periods.

<table>
<thead>
<tr>
<th>Per</th>
<th>Start</th>
<th>Sensing</th>
<th>Actuating</th>
<th>Control law</th>
<th>Data</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.00</td>
<td>db temp &gt; flux</td>
<td>free floating</td>
<td></td>
<td></td>
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</tbody>
</table>

Sunday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1 periods.

<table>
<thead>
<tr>
<th>Per</th>
<th>Start</th>
<th>Sensing</th>
<th>Actuating</th>
<th>Control law</th>
<th>Data</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.00</td>
<td>db temp &gt; flux</td>
<td>free floating</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

The sensor for function 11 senses dry bulb temperature in Chimney10. The actuator for function 11 is the air point in Chimney10. The function day types are Weekdays, Saturdays & Sundays.

Weekday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1 periods.

<table>
<thead>
<tr>
<th>Per</th>
<th>Start</th>
<th>Sensing</th>
<th>Actuating</th>
<th>Control law</th>
<th>Data</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.00</td>
<td>db temp &gt; flux</td>
<td>free floating</td>
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<td></td>
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</tbody>
</table>

Saturday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1 periods.

<table>
<thead>
<tr>
<th>Per</th>
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<th>Sensing</th>
<th>Actuating</th>
<th>Control law</th>
<th>Data</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.00</td>
<td>db temp &gt; flux</td>
<td>free floating</td>
<td></td>
<td></td>
</tr>
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</table>

Sunday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1 periods.

<table>
<thead>
<tr>
<th>Per</th>
<th>Start</th>
<th>Sensing</th>
<th>Actuating</th>
<th>Control law</th>
<th>Data</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.00</td>
<td>db temp &gt; flux</td>
<td>free floating</td>
<td></td>
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</tbody>
</table>

The sensor for function 12 senses dry bulb temperature in Chimney11. The actuator for function 12 is the air point in Chimney11. The function day types are Weekdays, Saturdays & Sundays.

Weekday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1 periods.

<table>
<thead>
<tr>
<th>Per</th>
<th>Start</th>
<th>Sensing</th>
<th>Actuating</th>
<th>Control law</th>
<th>Data</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.00</td>
<td>db temp &gt; flux</td>
<td>free floating</td>
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Saturday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1 periods.

<table>
<thead>
<tr>
<th>Per</th>
<th>Start</th>
<th>Sensing</th>
<th>Actuating</th>
<th>Control law</th>
<th>Data</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.00</td>
<td>db temp &gt; flux</td>
<td>free floating</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Sunday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1 periods.

<table>
<thead>
<tr>
<th>Per</th>
<th>Start</th>
<th>Sensing</th>
<th>Actuating</th>
<th>Control law</th>
<th>Data</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.00</td>
<td>db temp &gt; flux</td>
<td>free floating</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

The sensor for function 13 senses dry bulb temperature in Chimney12. The actuator for function 13 is the air point in Chimney12. The function day types are Weekdays, Saturdays & Sundays.
Weekday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1 periods.

<table>
<thead>
<tr>
<th>Per</th>
<th>Start</th>
<th>Sensing</th>
<th>Actuating</th>
<th>Control law</th>
<th>Data</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.00</td>
<td>db temp</td>
<td>&gt; flux</td>
<td>free floating</td>
<td></td>
</tr>
</tbody>
</table>

Saturday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1 periods.

<table>
<thead>
<tr>
<th>Per</th>
<th>Start</th>
<th>Sensing</th>
<th>Actuating</th>
<th>Control law</th>
<th>Data</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.00</td>
<td>db temp</td>
<td>&gt; flux</td>
<td>free floating</td>
<td></td>
</tr>
</tbody>
</table>

Sunday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1 periods.

<table>
<thead>
<tr>
<th>Per</th>
<th>Start</th>
<th>Sensing</th>
<th>Actuating</th>
<th>Control law</th>
<th>Data</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.00</td>
<td>db temp</td>
<td>&gt; flux</td>
<td>free floating</td>
<td></td>
</tr>
</tbody>
</table>

The sensor for function 14 senses dry bulb temperature in Outlet.
The actuator for function 14 is the air point in Outlet.
The function day types are Weekdays, Saturdays & Sundays
Weekday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1 periods.

<table>
<thead>
<tr>
<th>Per</th>
<th>Start</th>
<th>Sensing</th>
<th>Actuating</th>
<th>Control law</th>
<th>Data</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.00</td>
<td>db temp</td>
<td>&gt; flux</td>
<td>free floating</td>
<td></td>
</tr>
</tbody>
</table>

Saturday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1 periods.

<table>
<thead>
<tr>
<th>Per</th>
<th>Start</th>
<th>Sensing</th>
<th>Actuating</th>
<th>Control law</th>
<th>Data</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.00</td>
<td>db temp</td>
<td>&gt; flux</td>
<td>free floating</td>
<td></td>
</tr>
</tbody>
</table>

Sunday control is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1 periods.

<table>
<thead>
<tr>
<th>Per</th>
<th>Start</th>
<th>Sensing</th>
<th>Actuating</th>
<th>Control law</th>
<th>Data</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.00</td>
<td>db temp</td>
<td>&gt; flux</td>
<td>free floating</td>
<td></td>
</tr>
</tbody>
</table>

The sensor for function 15 senses dry bulb temperature in Room.
The actuator for function 15 is the air point in Room.
There have been 1 day types defined.

Day type 1 is valid Thu-01-Jan to Thu-31-Dec, 2009 with 1 periods.

<table>
<thead>
<tr>
<th>Per</th>
<th>Start</th>
<th>Sensing</th>
<th>Actuating</th>
<th>Control law</th>
<th>Data</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.00</td>
<td>db temp</td>
<td>&gt; flux</td>
<td>reads from temporal 500000.0 0.0 500000.0 0.0 1.0 3.0 -5.0 25.0 0.0</td>
<td></td>
</tr>
</tbody>
</table>

match temperature (ideal): max heat cp 500000.W min heat cp 0.W max cool cp 500000.W min
heat cp 0.W Aux sensors 1. mean value @reads temporal item.25 scale 1.00 offset 0.00

Zone to control loop linkages:
zone ( 1) Room            << control 15
zone ( 2) Chimney1        << control 2
zone ( 3) Chimney2        << control 3
zone ( 4) Chimney3        << control 4
zone ( 5) Chimney4        << control 5
zone ( 6) Chimney5        << control 6
zone ( 7) Chimney6        << control 7
zone ( 8) Chimney7        << control 8
zone ( 9) Chimney8        << control 9
zone (10) Chimney9        << control 10
zone (11) Chimney10       << control 11
zone (12) Chimney11       << control 12
zone (13) Chimney12       << control 13
zone (14) Outlet          << control 14

The model includes an air flow network.
Flow network description.

17 nodes, 16 components, 16 connections; wind reduction = 1.000
<table>
<thead>
<tr>
<th>#</th>
<th>Node</th>
<th>Fluid</th>
<th>Node Type</th>
<th>Height</th>
<th>Temperature</th>
<th>Data_1</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Room</td>
<td>air</td>
<td>internal &amp; unknown</td>
<td>1.6250</td>
<td>20.000</td>
<td>(-)</td>
</tr>
<tr>
<td>2</td>
<td>Chimney1</td>
<td>air</td>
<td>internal &amp; unknown</td>
<td>0.1250</td>
<td>20.000</td>
<td>(-)</td>
</tr>
<tr>
<td>3</td>
<td>Chimney2</td>
<td>air</td>
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Component Type C+ L+ Description

**Inlet_Room**
- 40 3 0 Common orifice flow component \( \text{m} = \rho \cdot f(C_d, A, \rho, \Delta P) \)

**Fluid**
- 1.0 opening area \((m^2)\) 0.500 discharge factor (-) 0.650

**Room_ch1**
- 210 6 0 General flow conduit component \( \text{m} = \rho \cdot f(D, A, L, k, SC_i) \)

**Fluid**
- Fluid, hydr diam, x-sect, conduit ln, roughness, loss fac.
- 1.0 0.444 0.500 0.250 0.000 0.720

**ch1_ch2**
- 210 6 0 General flow conduit component \( \text{m} = \rho \cdot f(D, A, L, k, SC_i) \)

**Fluid**
- Fluid, hydr diam, x-sect, conduit ln, roughness, loss fac.
- 1.0 0.444 0.500 0.250 0.000 0.100

**ch2_ch3**
- 210 6 0 General flow conduit component \( \text{m} = \rho \cdot f(D, A, L, k, SC_i) \)

**Fluid**
- Fluid, hydr diam, x-sect, conduit ln, roughness, loss fac.
- 1.0 0.444 0.500 0.635 0.000 0.100

**ch3_ch4**
- 210 6 0 General flow conduit component \( \text{m} = \rho \cdot f(D, A, L, k, SC_i) \)

**Fluid**
- Fluid, hydr diam, x-sect, conduit ln, roughness, loss fac.
### General Flow Conduit Component

1. **ch4_ch5**
   - **Component**: General flow conduit component
   - **Equation**: $m = \rho \cdot f(D, A, L, k, SCi)$
   - **Details**: Fluid, hydr diam, x-sect, conduit ln, roughness, loss fac.
   - Parameters: $1.0 \ 0.444 \ 0.500 \ 1.020 \ 0.000 \ 0.100$

2. **ch5_ch6**
   - **Component**: General flow conduit component
   - **Equation**: $m = \rho \cdot f(D, A, L, k, SCi)$
   - **Details**: Fluid, hydr diam, x-sect, conduit ln, roughness, loss fac.
   - Parameters: $1.0 \ 0.444 \ 0.500 \ 1.020 \ 0.000 \ 0.100$

3. **ch6_ch7**
   - **Component**: General flow conduit component
   - **Equation**: $m = \rho \cdot f(D, A, L, k, SCi)$
   - **Details**: Fluid, hydr diam, x-sect, conduit ln, roughness, loss fac.
   - Parameters: $1.0 \ 0.444 \ 0.500 \ 1.020 \ 0.000 \ 0.100$

4. **ch7_ch8**
   - **Component**: General flow conduit component
   - **Equation**: $m = \rho \cdot f(D, A, L, k, SCi)$
   - **Details**: Fluid, hydr diam, x-sect, conduit ln, roughness, loss fac.
   - Parameters: $1.0 \ 0.444 \ 0.500 \ 1.020 \ 0.000 \ 0.100$

5. **ch8_ch9**
   - **Component**: General flow conduit component
   - **Equation**: $m = \rho \cdot f(D, A, L, k, SCi)$
   - **Details**: Fluid, hydr diam, x-sect, conduit ln, roughness, loss fac.
   - Parameters: $1.0 \ 0.444 \ 0.500 \ 1.020 \ 0.000 \ 0.100$

6. **ch9_ch10**
   - **Component**: General flow conduit component
   - **Equation**: $m = \rho \cdot f(D, A, L, k, SCi)$
   - **Details**: Fluid, hydr diam, x-sect, conduit ln, roughness, loss fac.
   - Parameters: $1.0 \ 0.444 \ 0.500 \ 1.020 \ 0.000 \ 0.100$

7. **ch10_ch11**
   - **Component**: General flow conduit component
   - **Equation**: $m = \rho \cdot f(D, A, L, k, SCi)$
   - **Details**: Fluid, hydr diam, x-sect, conduit ln, roughness, loss fac.
   - Parameters: $1.0 \ 0.444 \ 0.500 \ 1.020 \ 0.000 \ 0.100$

8. **ch11_ch12**
   - **Component**: General flow conduit component
   - **Equation**: $m = \rho \cdot f(D, A, L, k, SCi)$
   - **Details**: Fluid, hydr diam, x-sect, conduit ln, roughness, loss fac.
   - Parameters: $1.0 \ 0.444 \ 0.500 \ 1.020 \ 0.000 \ 0.100$

9. **Ch12_Outlet**
   - **Component**: General flow conduit component
   - **Equation**: $m = \rho \cdot f(D, A, L, k, SCi)$
   - **Details**: Fluid, hydr diam, x-sect, conduit ln, roughness, loss fac.
   - Parameters: $1.0 \ 0.444 \ 0.500 \ 0.685 \ 0.000 \ 0.100$

### Common Orifice Flow Component

1. **Outlet_N**
   - **Component**: Common orifice flow component
   - **Equation**: $m = \rho \cdot f(Cd, A, \rho, dP)$
   - **Details**: Fluid, 1.0 opening area ($m^2$), 0.250 discharge factor (-), 0.650
   - Parameters: $40 \ 3 \ 0$

2. **Outlet_S**
   - **Component**: Common orifice flow component
   - **Equation**: $m = \rho \cdot f(Cd, A, \rho, dP)$
   - **Details**: Fluid, 1.0 opening area ($m^2$), 0.250 discharge factor (-), 0.650
   - Parameters: $40 \ 3 \ 0$

### Node Details

<table>
<thead>
<tr>
<th>+Node</th>
<th>dHght</th>
<th>-Node</th>
<th>dHght</th>
<th>Component</th>
<th>Z @+</th>
<th>Z @-</th>
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<td>BC_Inlet_N</td>
<td>0.000</td>
<td>Room</td>
<td>-0.270</td>
<td>Inlet_Room</td>
<td>1.350</td>
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<td>1.355</td>
<td>Room</td>
<td>-1.495</td>
<td>Chimney1</td>
<td>0.000</td>
<td>Room_ch1</td>
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</table>
thermal zone to air flow node mapping:
thermal zone -> air flow node
Room -> Room
Chimney1 -> Chimney1
Chimney2 -> Chimney2
Chimney3 -> Chimney3
Chimney4 -> Chimney4
Chimney5 -> Chimney5
Chimney6 -> Chimney6
Chimney7 -> Chimney7
Chimney8 -> Chimney8
Chimney9 -> Chimney9
Chimney10 -> Chimney10
Chimney11 -> Chimney11
Chimney12 -> Chimney12
Outlet -> Outlet

Multi-layer constructions used:

Details of opaque construction: Absorber

<table>
<thead>
<tr>
<th>Layer</th>
<th>Prim</th>
<th>Thick</th>
<th>Conduct-</th>
<th>Density</th>
<th>Specif</th>
<th>IR</th>
<th>Solr</th>
<th>Diffu</th>
<th>R</th>
<th>Descr</th>
</tr>
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<td>30</td>
<td>0.5</td>
<td>237.000</td>
<td>2702.</td>
<td>903.</td>
<td>0.05</td>
<td>0.95</td>
<td>100.</td>
<td>0.00</td>
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<tr>
<td>mirotherm : Mirotherm</td>
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<tr>
<td>Int</td>
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<td>2702.</td>
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<tr>
<td>rockwool : Rockwool</td>
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</table>
| ISO 6946 U values (horiz/upward/downward heat flow)= 0.278 0.280 0.275 (partition) 0.271
Total area of Absorber is 27.50

Details of transparent construction: Glazing with Planitherm optics.

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<th>Cond</th>
<th>Density</th>
<th>Specif</th>
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<th>Solr</th>
<th>Diffu</th>
<th>R</th>
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<td>1.</td>
<td>1000.</td>
<td>0.00</td>
<td>0.00</td>
<td>1.</td>
<td>0.83 argon : Argon</td>
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<tr>
<td>Int</td>
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For solar chimney: with id of: Planitherm with 3 layers [including air gaps] and visible trn: 0.77
Direct transmission @ 0, 40, 55, 70, 80 deg
0.517 0.491 0.437 0.299 0.141
Layer吸收 @ 0, 40, 55, 70, 80 deg
1 0.042 0.047 0.050 0.055 0.056
2 0.001 0.001 0.001 0.001 0.001
3 0.179 0.195 0.197 0.192 0.128
Total area of Glazing is 15.67

Details of transparent construction: Fictive with SC_fictit optics.

<table>
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<tr>
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<th>Cond</th>
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<th>IR</th>
<th>Solr</th>
<th>Diffu</th>
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<tr>
<td>1</td>
<td>41</td>
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<td>0.01</td>
<td>19200.0</td>
<td>0.00 fict : fictitious material (almost not there)</td>
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<td>ISO 6946 U values (horiz/upward/downward heat flow)= 5.881 7.140 4.761 (partition) 3.845</td>
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: with id of: SC_fictit
with 1 layers [including air gaps] and visible trn: 0.33
Direct transmission @ 0, 40, 55, 70, 80 deg
0.087 0.068 0.053 0.035 0.018
Layer吸收 @ 0, 40, 55, 70, 80 deg
1 0.869 0.883 0.873 0.789 0.590
Total area of Fictive is 13.99

Details of opaque construction: Frame

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<th>Cond</th>
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<th>Diffu</th>
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<tr>
<td>ISO 6946 U values (horiz/upward/downward heat flow)= 0.226 0.227 0.224 (partition) 0.221</td>
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</table>
Total area of Frame is 7.33

Details of opaque construction: Room_Cons

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<th>Density</th>
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-89- | PAGE
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<th>heat</th>
<th>emis</th>
<th>abs</th>
<th>resis</th>
<th>m^2K/W</th>
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<td>1200.</td>
<td>0.50</td>
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<tr>
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<td>550.</td>
<td>1200.</td>
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</table>

ISO 6946 U values (horiz/upward/downward heat flow)= 0.209 0.211 0.208 (partition) 0.206

Total area of Room_Cons is 44.30
Appendix 8: Measured and predicted Outlet Air-Temperature

Note only CFD Method 2 is shown here!
15:00h (Tin = 293 K ± 1K)

Outlet temperature distribution (Kelvin)

Depth (m)

16:00h (Tin = 293 K ± 1K)

Outlet temperature distribution (Kelvin)

Depth (m)
Appendix 9: Measured and predicted Outlet Air-Velocity

Note only CFD Method 2 is shown here!
15:00h

Y-Velocity (m/s) vs Depth (m)

- Experiment
- CFD Model 1 Low-Re K-Epsilon
- CFD Model 1 K-Omega
- CFD Model 2 Low-Re K-Epsilon
- CFD Model 3 Low-Re K-Epsilon
- ESP - r Cd=0.65 Alamdari-Hammond
- ESP - r Cd=0.42 Alamdari-Hammond
- ESP - r Cd=0.42 Khalifa-Marshall

16:00h

Y-Velocity (m/s) vs Depth (m)

- Experiment
- CFD Model 1 Low-Re K-Epsilon
- CFD Model 1 K-Omega
- CFD Model 2 Low-Re K-Epsilon
- CFD Model 3 Low-Re K-Epsilon
- ESP - r Cd=0.65 Alamdari-Hammond
- ESP - r Cd=0.42 Alamdari-Hammond
- ESP - r Cd=0.42 Khalifa-Marshall
Appendix 10: Measured and predicted absorber temperature

Note only CFD Method 2 is shown here!

---

**9:00h**

[Graph showing absorber temperature over height at 9:00h]

- Experiment
- CFD Model1 low-Re K-Epsilon
- CFD Model1 K-Omega
- CFD Model2 low-Re K-Epsilon
- CFD Model3 low-Re K-Epsilon
- ESP-r Cd=0.42 Alamdari-Hammond
- ESP-r Cd=0.65 Alamdari-Hammond
- ESP-r Cd=0.42 Khalifa-Marshall

---

**10:00h**

[Graph showing absorber temperature over height at 10:00h]

- Experiment
- CFD Model1 low-Re K-Epsilon
- CFD Model1 K-Omega
- CFD Model2 low-Re K-Epsilon
- CFD Model3 low-Re K-Epsilon
- ESP-r Cd=0.42 Alamdari-Hammond
- ESP-r Cd=0.65 Alamdari-Hammond
- ESP-r Cd=0.42 Khalifa-Marshall
Appendix 11: Measured and predicted glass temperature

Note only CFD Method 2 is shown here!
15:00h

Glas temperature (Kelvin)

Height (m)

- Experiment
- CFD Model 1 low-Re K-Epsilon
- CFD Model 1 K-Omega

+ CFD Model 2 low-Re K-Epsilon
- CFD Model 3 low-Re K-Epsilon
- ESP-r Cd=0.65 Alamdari-Hammoud
- ESP-r Cd=0.42 Khalifa-Marshall
- Ambient

16:00h

Glas temperature (Kelvin)

Height (m)

- Experiment
- CFD Model 1 low-Re K-Epsilon
- CFD Model 1 K-Omega

+ CFD Model 2 low-Re K-Epsilon
- CFD Model 3 low-Re K-Epsilon
- ESP-r Cd=0.65 Alamdari-Hammoud
- ESP-r Cd=0.42 Khalifa-Marshall
- Ambient
Appendix 12: Estimated local Nusselt number against the Rayleigh Number

Note that the very maximum Nusselt number was calculated at the inlet area (at 0.25 m) and the very minimum Nusselt number was calculated at the outlet area (at 11 m). Rayleigh number of 3E09 are for CFD Method 1!