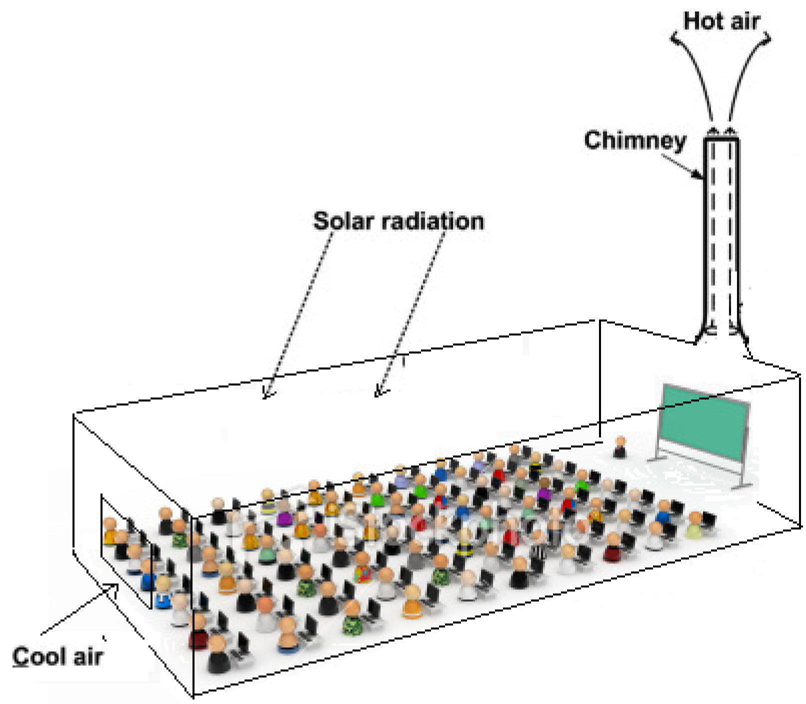
Solar Chimney Analysis

Analysis of an integrated solar chimney / air conditioning system applied to an auditorium at highland tropical climate using simulation



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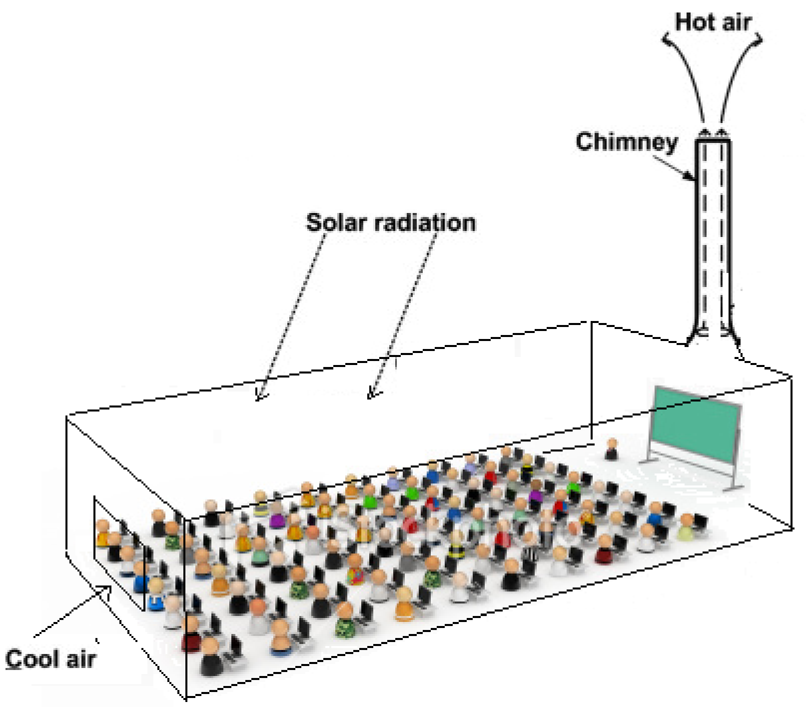

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Foreword

The report you are holding in your hands is the result of my graduation project conducted in Brazil; an amazing six months in my life. I always wanted to study in a foreign country and almost went to Indonesia for my first internship or to South-Korea for an exchange program. Both of these options did not succeed at the time. Thanks to professor Johannes Bruinsma through whom the amazing opportunity arose to graduate in a Brazilian university. I found a suitable project matching my system engineering background and interests. I conducted this project in the city of Viçosa in the state of Minas Gerais at a relatively new research center. In this research center called NEMOS, I worked along with 25 other undergraduate students who did research on the subject of "simulation of Air Conditioning and Natural Ventilation Systems". I was the only one who worked full-time in there. As being also necessary of my own research, I developed a more general model which many students will use in the future for their work. I especially want to thank my coordinator Prof. Dr. Eng. Álvaro M. Bigonha Tibiriçá who gave me this study opportunity and hugely contributed to expanding my vision on research and modeling. During my stay in Brazil I learned how to speak Portuguese up to a decent level, learned a lot about the Latin American culture and made friends with people from all over the world.

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Summary

This report examines the possibilities of using a solar chimney in a highland tropical climate to promote thermal comfort, air-quality and energy consumption. The study is carried out using modeling and simulation techniques and is conducted at the Federal University of Viçosa in Brazil.

A model is built to simulate room temperature, humidity and CO_2 concentration as being functions of several parameters. The methodology chapter describes the mathematical models of the individual parameters as well as the integration into the overall model. The modeled parameters influencing temperature, humidity and CO_2 concentration are: the spatial characteristics of the room; the amount of people seated in the room; the amount of solar radiation received on the external surfaces; the weather; the ventilation and the air-conditioner usage. The type of ventilation that is used in this model is natural ventilation, caused by a solar chimney. The solar chimney working principle is based on a physical phenomenon: hot air rises. The ventilation rate is a function of the spatial characteristics of the room and of the difference between the inside and outside temperature. The solar chimney potential is being tested by compairing four scenarios: 1st using neither the solar chimney nor the air-conditioner; 2nd using only the air-conditioner; 3rd using only the solar chimney; 4th using the solar chimney and air-conditioner in combination. The simulation is ran for the four scenarios and for different seasons in the year. The thermal comfort is discussed by the measure of PMV, a indicator for the level of thermal satisfaction. The air-quality is discussed on the basis of CO_2 concentration.

Simulation results show that solar chimney ventilation can reduce air-conditioner electricity expenditure to less than 80% of the initial expenditure for a summer day. The air-conditioner electricity expenditure is reduced to only 10% of the initial usage for a winter day. The solar chimney guaranties high indoor air-quality whenever the outside temperature is below 25°C.

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1. Introduction

This report is the result of a graduation project for the Bachelor of Mechanical Engineering degree at the Hanze University of Applied Sciences. The project took place at the Federal University of Viçosa (UFV), at the Department of Mechanical and Production Engineering (DEP) under the supervision of professor Álvaro M. Bigonha Tibiriçá. This project is conducted within the research group NEMOS (within DEP) that focuses on simulation of Air Conditioning and Natural Ventilation Systems.

It is an ongoing challenge to lower the energy consumption of climate control equipment and simultaneously improve the indoor conditions. Important indoor conditions are thermal comfort and indoor air quality (IAQ), which can both be measured and controlled in various ways. Dounis et al (1995) shows the potential for controlling the thermal comfort on a measure called the Predicted Mean Vote (PMV) instead of temperature (1). PMV is a measure to display the average satisfaction with the thermal situation as a function of many variables. The environment in that study was fuzzy logic controlled and uses window area and air-conditioner power as the control variables. Dounis et al (1996) also succeeded to maintain acceptable IAQ by controlling a window opening (2).

This study evaluates the IAQ and thermal comfort achieved when integrating a solar chimney within an air-conditioner controlled environment. This is done with modeling and simulation techniques for a highland tropical climate applied on an auditorium as a reference room. Can the solar chimney contribute to lowering the air-conditioner electricity expenditure? What is the effect on the IAQ and thermal comfort during the different seasons of the year?

2 Methodology

The block scheme of Figure 1 shows the model used to simulate the temperature, humidity and CO_2 concentration in a room. The model's various parameters (number of people in the room, room dimensions, geographical situation, etc.) can easily be adjusted in this program. The solar chimney and air-conditioner can be used simultaneously or with only one of the individual systems activated.

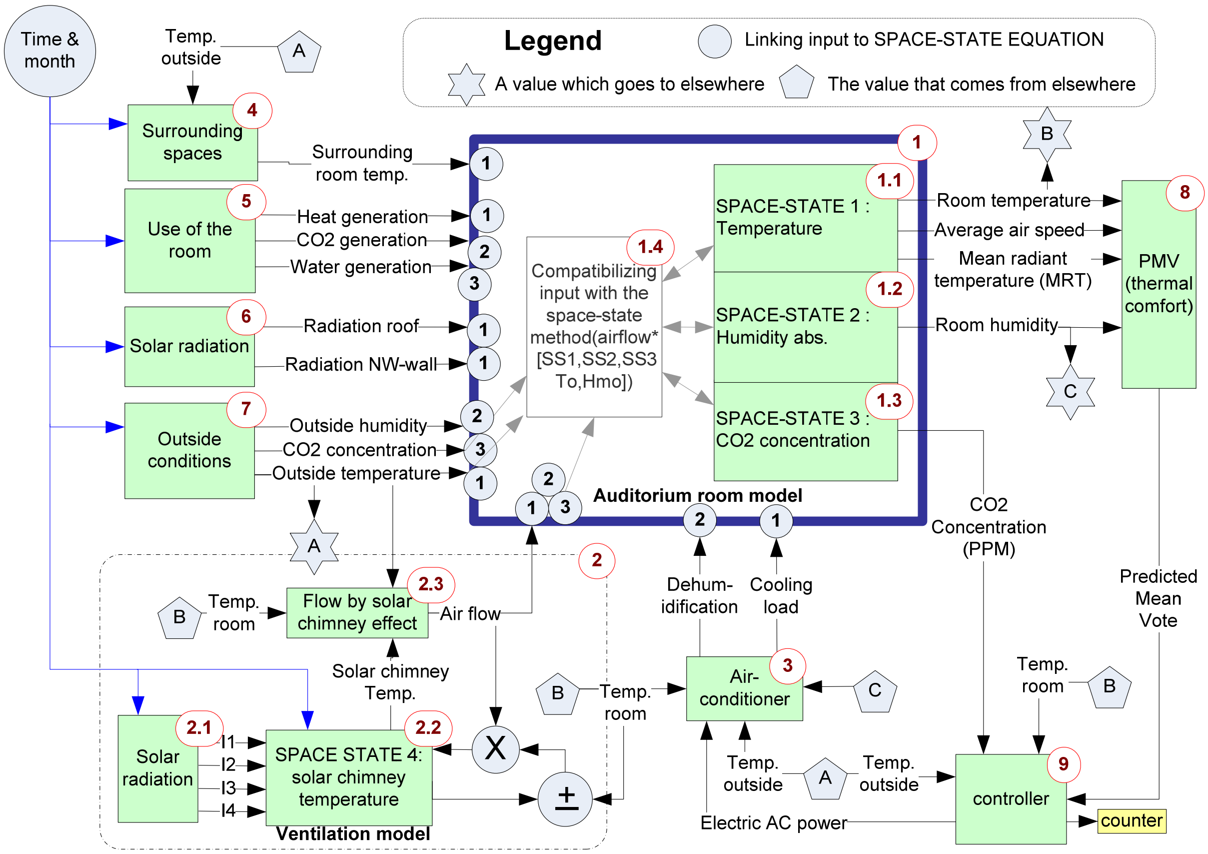


Figure 1: VISIO block scheme displaying the Matlab model

The auditorium room model (block1) and its SPACE-STATE models are described in section 2.2. The ventilation model (block2) is described in section 2.2.1-2.2.3. The air-conditioner model (block 3) is described in section 2.2.4. Blocks 4-7 are described in section 2.1, the PMV (block 8) in section 2.3.2, and the controller (block9) in section 2.6.

2.1 Reference room

The auditorium of the Department of Mechanical and Production Engineering (DEP) at the Federal University of Viçosa (UFV) serves as a reference room and data source for the performed simulation. The auditorium is used for seminars and undergraduate classes and seats 40 people. The dimensions are: 9.75 m (length) x 5.10 m (width) x 2.80 m (height). The left wall in Figure 2 is facing the exterior and the others are internal. The back wall has windows (as shown in Figure 3) along its entire length which are facing the interior of the building and do not receive direct sunlight. The ceiling is plaster and has a span of 70 cm and the slab is covered with metal tiles (3). The simulation is calculated for 30 people in the auditorium and a beamer, computer and 5 double TL-lights switched on during the class hours.

Figure 2: Reference room: Auditorium DEP/UFV

Figure 3:Windows in the back of the auditorium

2.2 Room modeling

The temperature, humidity and the CO_2 concentration in the room are being influenced by dozens of parameters. The way these input parameters are considered to influence the room variables, using a numerical method, are explained in the next paragraphs. An elucidation on the SPACE-STATE method is given in section 2.2.1.2.

2.2.1 Room temperature

The model considers the internal temperature to be a function of the heat flux through the inner walls, the outer walls and the roof, the effect of the ventilation, air-conditioner and evaporative cooling systems as well as the heat generated inside the room by humans, the computer and beamer.

2.2.1.1 Differential equation

The behavior of the average room temperature can be described by the following first order differential equation:

$$\boldsymbol{\rho_{air} * c_{air} * V_r * \frac{dT_r}{dt} = \dot{Q}wi + \dot{Q}wo + \dot{Q}r + \dot{Q}gi + \dot{Q}ac + \dot{Q}vent} \qquad \text{Eq. 1}$$

Where:

ρ_{air}	density of air $[kg/m^3]$
c_{air}	specific heat of the air $[J/(kg * K)]$
V_r	volume of the room $[m^3]$
T_r	average room temperature $[°C]$
$\dot{Q}wi$	heat flux through internal walls, door and window[W]

$\dot{Q}wo$	heat flux through outer walls [W]
$\dot{Q}r$	heat flux through the roof [W]
$\dot{Q}ig$	sum of internally generated heat by people, lights, computer and beamer [W]
$\dot{Q}ac$	cooling power air conditioning [W]
$\dot{Q}vent$	cooling by ventilation [W]

The terms in Eq. 1 are described by Eq. 2 to Eq. 7.

$$\dot{Q}wi = (U_w * \sum A_{wi} + A_{door} * U_{door} + A_{window} * U_{window}) * (T_r - T_{sr}) \qquad \text{Eq. 2}$$

$$\dot{Q}_{wo} = Uw * A_{wo} * (I_w * abs_{wall} * R_{se} + (T_o - T_r)) \qquad \text{Eq. 3}$$

$$\dot{Q}_{roof} = U_{roof} * A_{roof} * \left(I_{roof} * abs_{roof} * R_{sit}(T_o - T_r)\right) \qquad \text{Eq. 4}$$

$$\dot{Q}_{gi} = Q_{people} * n * occupation \qquad \text{Eq. 5}$$

$$\dot{Q}ac = COP_{AC} * electric \qquad \text{Eq. 6}$$

$$\dot{Q}vent = \dot{V} * (To - Tr) * \rho_{air} * c_{air} \qquad \text{Eq. 7}$$

Where

U_w	thermal transmittance of the walls [2.28 W/(m^2K)]
A_{wi}	Surface area of inner walls $[m^2]$
A_{door}	Area of the door (1.7015m^2)
U_{door}	Thermal transmittance of the door [2.8W/(m^2K)]

A_{window}	Window area[3.6m^2]
U_{window}	Thermal transmittance of the window [5.7W/(m^2K)]
T_{sr}	Temperature surrounding rooms[$°C$]
A_{wo}	Surface area of outer walls[m^2]
I_w	Radiation on the wall[W/m^2]

Abs_{wall}	Radiation absorbance wall [0.2]
R_{se}	Convection resistance of outer surface of the roof and external wall [0.04m^2K/W]
T_o	Outside temperature[$°C$]
U_{roof}	thermal transmittance of the roof [1.83 W/(m^2K)]
A_{roof}	Area of the roof (50m^2)
I_{roof}	Radiation on the roof $[W/m^2]$
abs_{roof}	Radiation absorbance wall [0.6]
R_{sit}	Confection resistance of the ceiling and the inner surface of the walls[0.17m^2K/W]
Q_{people}	Heat disposal by people [130W]
n	Amount of people in the room[30]
$occupation$	Room is occupied or not [class hours]
COP_{AC}	Coefficient of performance air-conditioner
$electric$	Electric power supply to facility[W]
$\dot{V}$	Volume flow $[m^3/s]$

Eq. 1 - Eq. 7 are transformed into Eq. 8 -Eq. 19. The constants C1-C11 depend on the modeled environment's dimensions, geographic location and other characteristics (which in this simulation can easily be changed).

$$C1 * \frac{dT_r}{dt} + C2 * T_r = \frac{C3}{C1} * Tsr + \frac{C4}{C1} * Iw + \frac{C5}{C1} * Ir + \frac{C6}{C1} * To + \frac{C7}{C1} Qgi + \frac{C8}{C1} Qac + \frac{C9}{C1} * (Qvent * To) + \frac{C10}{C1} * (Qvent * Tr) \quad \text{Eq. 8}$$

Where C1 t/m C15 are constants:

$$C1 = \rho_{air} * Vr * c_{air}\ (J/K) \quad \text{Eq. 9}$$

$$C2 = (AW_{NW} + AW_{NE} + AW_{SE} + AW_{SW}) * U_w + A_{roof} * U_{roof} + A_{window} * U_{window} + A_{door} * U_{door}\ (W/K) \quad \text{Eq. 10}$$

$$C3 = (AW_{NE} + AW_{SE} + AW_{SW}) * U_w + A_{door} * U_{door} + A_{window} * U_{window};\ (W/K) \quad \text{Eq. 11}$$

$$C4 = U_w * R_{se} * abs_{wall} * AW_{NW} (W/W) \quad \text{Eq. 12}$$

$$C5 = U_{roof} * R_{sit} * abs_{roof} * A_{roof}\ (W/W) \quad \text{Eq. 13}$$

$$C6 = AWnw * U_w + Aroof * Uroof\ (W/K) \quad \text{Eq. 14}$$

$$C7 = C8 = 1 \quad \text{Eq. 15}$$

$$C9 = C10 = \rho_{air} * c_{air} (J/(m3 * K)) \quad \text{Eq. 16}$$

$$C14 = (A_{tot} - U_{gem} * A_{tot} * R_{se}/A_{tot}) \quad \text{Eq. 17}$$

$$C15 = R_{se}/A_{tot} \quad \text{Eq. 18}$$

$$Ugem = (U_{roof} * A_{roof} + U_{wall} * (AW_{NW} * 2 + AW_{NE} * 2))/A_{tot} \quad \text{Eq. 19}$$

Where:

AW_{NW}	Area north-west wall [27.3m^2]
AW_{NE}	Area north-east wall [14.3m^2]
AW_{SE}	Area south-east wall [25.6m^2]
AW_{SW}	Area south-west wall [10.7m^2]
A_{tot}	Total of surfaces for radiation calculation without the floor $[m^2]$
U_{gem}	Parameter for mean radiant temperature calculations [W/(m^2K)]

2.2.1.2 SPACE-STATE representation

These constants can all be rewritten in the SPACE-STATE form: This results are Eq. 20 and Eq. 21.

$$\boxed{\begin{aligned} x' &= A * x + B * u \\ y &= C * x + D * u \end{aligned}}$$

$$x' = \frac{-C2}{C1} * Tr + \begin{pmatrix} \frac{C3}{C1} & \frac{C4}{C1} & \frac{C5}{C1} & \frac{C6}{C1} & \frac{C7}{C1} & \frac{C8}{C1} & \frac{C9}{C1} & \frac{-C10}{C1} \end{pmatrix} * \begin{pmatrix} \mathrm{Tsr} \\ \mathrm{Iw} \\ \mathrm{Ir} \\ \mathrm{To} \\ \dot{\mathrm{Q}}\mathrm{gi} \\ \dot{\mathrm{Q}}\mathrm{ac} \\ \dot{V} * To \\ \dot{V} * Tr \end{pmatrix} \quad \text{Eq. 20}$$

$$y = \begin{pmatrix} C14 \\ -C2 \\ 0 \end{pmatrix} * Tr + \begin{pmatrix} \frac{C15}{C3} & \frac{C15}{C4} & \frac{C15}{C5} & \frac{C15}{C6} & 0 & 0 & 0 & 0 \\ C3 & C4 & C5 & C6 & C7 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & C8 & C9 & -C10 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \end{pmatrix} * \begin{pmatrix} \mathrm{Tsr} \\ \mathrm{Iw} \\ \mathrm{Ir} \\ \mathrm{To} \\ \dot{\mathrm{Q}}\mathrm{gi} \\ \dot{\mathrm{Q}}\mathrm{ac} \\ \dot{V} * To \\ \dot{V} * Tr \end{pmatrix} \quad \text{Eq. 21}$$

2.2.2 Room humidity

The model considers room humidity to be a function of moisture evaporated by human, condensed moisture in the air-conditioner and exchanges with outside air by ventilation.

2.2.2.1 Differential equation

The behavior of the average room humidity can be described by the following first order differential equation:

$$\boldsymbol{\rho_{air} * V_r * \frac{dH_r}{dt} = H_{people} + H_{AC} + \dot{V} * \rho_{air} * (H_o - H_r)} \quad \text{Eq. 22}$$

Where:

H_r	Average humidity in the room [g/kg]
H_{people}	Moisture added to the atmosphere by breathing and sweating [g/s]
H_{AC}	Condensed moisture in the air-conditioner [g/s]

H_o	Absolute outside humidity outside[g/kg]
H_r	Absolute humidity in the room in [g/kg]

2.2.2.2 SPACE-STATE representation

These above differential equation can be rewritten in the SPACE-STATE form as shown in Eq. 23 and Eq. 24:

$$x' = (0) * H_r + \begin{pmatrix} \frac{1}{C12} & \frac{1}{C12} & \frac{\rho_{air}}{C12} & -\frac{\rho_{air}}{C12} \end{pmatrix} * \begin{pmatrix} H_{people} \\ H_{AC} \\ \dot{V} * H_o \\ \dot{V} * H_r \end{pmatrix} \quad \text{Eq. 23}$$

$$y = (1) * H_r + \begin{pmatrix} 0 & 0 & 0 & 0 \end{pmatrix} * \begin{pmatrix} H_{people} \\ H_{AC} \\ \dot{V} * H_o \\ \dot{V} * H_r \end{pmatrix} \quad \text{Eq. 24}$$

Where:

$$C12 = V_{r*}\rho_{air} \quad \text{Eq. 25}$$

2.2.3 Room CO_2 concentration

The room CO_2 concentration model considers human emissions and the effect of air-exchanges with the outside using the solar chimney.

2.2.3.1 Differential equation

The behavior of the average room CO_2 concentration can be described by the following first order differential equation:

$$\boldsymbol{V_r * \frac{dC_r}{dt} = \dot{V} * (C_r - C_o) + C_{people} * ppm_{people}} \quad \text{Eq. 26}$$

Where:

C_r	CO_2 concentration in the room in parts per million [ppm]
C_o	CO_2 concentration outside [ppm]

C_{people}	CO_2 emission per person *[m³/s]*
ppm_{people}	CO_2 concentration of $C_{people} = 10^6 [ppm]$

2.2.3.2 SPACE-STATE representation

The differential equation can be rewritten in the SPACE-STATE form as shown in Eq. 27 and Eq. 28:

$$x' = (0) * C_r + \begin{pmatrix} \frac{1}{V_r} & -\frac{1}{V_r} & \frac{1}{V_r} \end{pmatrix} * \begin{pmatrix} V * C_o \\ \dot{V} * C_r \\ C_{people} \end{pmatrix} \quad \text{Eq. 27}$$

$$y = (1) * C_r + \begin{pmatrix} 0 & 0 & 0 \end{pmatrix} * \begin{pmatrix} V * C_o \\ \dot{V} * C_r \\ C_{people} \end{pmatrix} \quad \text{Eq. 28}$$

2.3 External parameters

2.3.1 Solar radiation

The incoming heat through the outer wall and the roof are heavily influenced by the received solar radiation. The received radiation can be calculated by the 'solar intensity' and the solar 'angle of incidence'. The model considers the shading effect of the atmosphere but does not consider clouds at all. The solar angles of incident are calculated according to the methods of *Renewable energy recources 2005* by Weir, John Twidell and Tony. The solar intensity is modeled accoring to *2009 ASHRAE Handbook – Fundamentals* (4).

2.3.1.1 Solar intensity

The solar intensity is a function of the solar altitude(θ_z *in* Figure 5) and two other factors (A and B) which show monthly averages and can be found in the database attached to the 2009 ASHRAE handbook (5). Factor A represents the solar intensity outside the atmosphere and number B is a dimensionless ratio dependent on atmospheric thickness and other variables. The altitude is a function (Eq. 31) of the hour angle (ψ), latitude(ϕ), and declination (δ_0) which are explained in section 2.3.1.3.

$$A = [1230\ 1215\ 1186\ 1136\ 1104\ 1088\ 1085\ 1107\ 1151\ 1192\ 1221\ 1233] \quad \text{Eq. 29}$$

$$B = [0.142\ 0.144\ 0.156\ 0.180\ 0.196\ 0.205\ 0.207\ 0.201\ 0.177\ 0.160\ 0.149\ 0.142] \quad \text{Eq. 30}$$

$$Altitude = sind(\delta_0) * sind(\phi) + cosd(\delta_0) * cosd(\phi) * cosd(\psi) \quad \text{Eq. 31}$$

The solar intensity (SI) can be calculated using Eq. 32:

$$SI = \frac{A(m)}{\exp\left(\frac{B(m)}{\sin(Altitude)}\right)} \quad \text{Eq. 32}$$

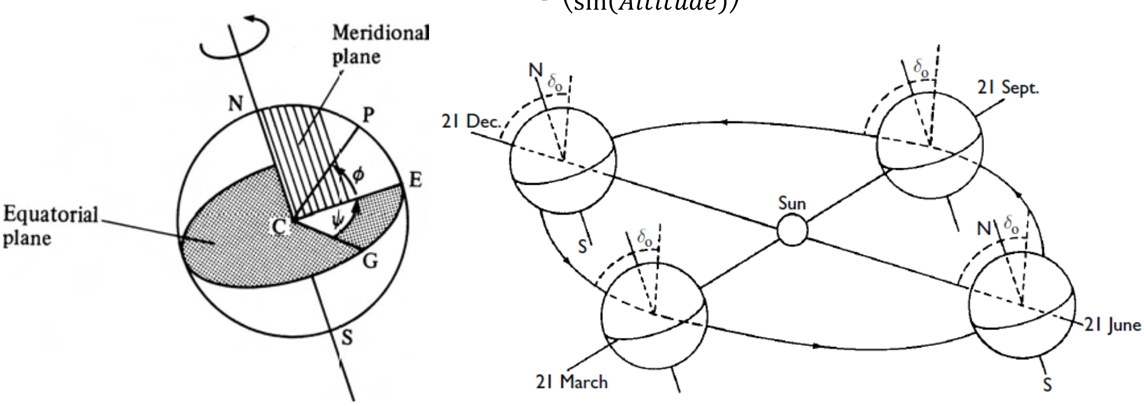


Figure 4: source renewable energy resources

2.3.1.2 Direct radiation

The angle of incidence (θ in Figure 5) is a function of five main variables:

- The hour angle (ψ in Figure 4) as a function of the time of the day. It describes the angle through which the Earth has rotated since solar noon. The Earth rotates 360 degrees in 24 hours, therefore the hour angle is given by Eq. 33.

- The declination of the earth (δ_0 in Figure 4) is a sine wave with a frequency of 1 Hz per year and goes from $+23.45$ degrees in midsummer in the northern hemisphere, to -23.45 degrees in northern midwinter. The declination is described by Eq. 34 where *n* is the day in the year (*n* = 1 at the 1st of January).

- The inclination of the surface (β in Figure 5) is 90 degrees for the wall because it is vertical and the reference is horizontal. Therefore the roof inclination is of 0 degrees.

- The latitude (ϕ in Figure 5) of Viçosa is -20.75.

- The surface Azimuth (in Figure 5) for the wall is -135 degrees because the wall is orientated North-West which is 135 degrees orientated away from the south.

$$Hour\ angle = \ (15 * hr)(tzone - 12\ hr) + (\psi - \psi zone) \quad \text{Eq. 33}$$

$$Declination = 23.5 * \ sin[360 * (284 + n)/365] \quad \text{Eq. 34}$$

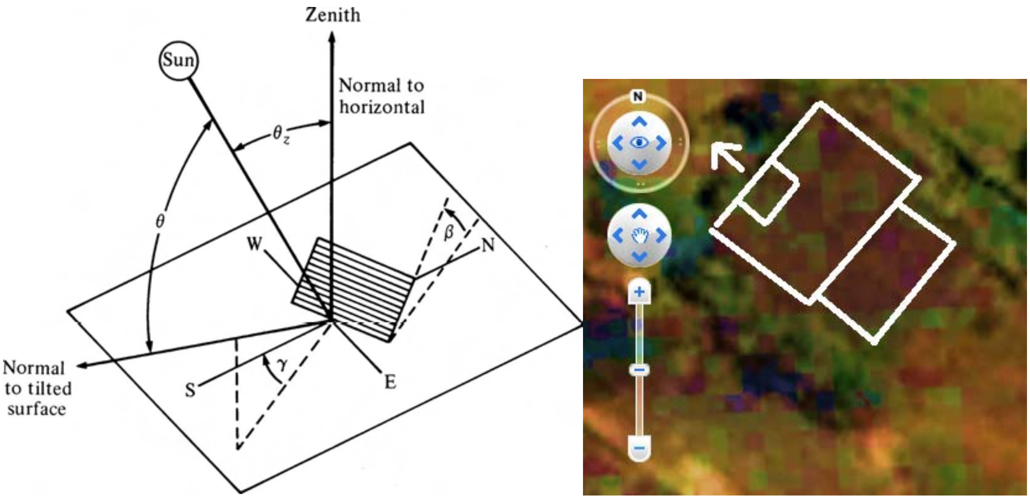


Figure 5: Zenith angle θ_z, angle of incidence θ, slope $\beta$and azimuth angle γ for a tilted surface. (Note: for this easterly facing surface $\gamma < 0$.) After Duffie and Beckman 1991.

To get the solar angles of incidence the five variables mentioned above must be substituted into Eq. 35 to Eq. 39 which in turn have to be substituted into Eq. 40.

$$A = \sin(\phi) * \cos(\beta) \quad \text{Eq. 35}$$
$$B = \cos(\phi) * \sin(\beta) * \cos(\gamma) \quad \text{Eq. 36}$$
$$C = \sin(\beta) * \sin(\gamma) \quad \text{Eq. 37}$$
$$D = \cos(\phi) * \cos(\beta) \quad \text{Eq. 38}$$
$$E = \sin(\phi) * \sin(\beta) * \cos(\gamma) \quad \text{Eq. 39}$$
$$\cos(\theta) = (A - B) * \sin(\delta_0) + [C * \sin(\psi) + (D + E) * \cos(\psi)] * \cos(\delta_0) \quad \text{Eq. 40}$$
$$Ib = SI * \cos(\theta) \quad \text{Eq. 41}$$

Where:

ϕ	Latitude
β	Inclination
γ	Surface azimuth
δ_0	Declination
ψ	Hour angle
$\cos(\theta)$	Cosines of the angle of incidence
Ib	Beam solar radiation

2.3.1.3 *Diffuse radiation*

The diffuse solar radiation(Id) is calculated with Eq. 41 - Eq. 44.

$$Fs = 0.5 * (1 + cosd(\beta)) \quad \text{Eq. 41}$$

$$Fg = 0.5 * (1 - cosd(\beta)) \quad \text{Eq. 42}$$

$$C = [0.058\ 0.060\ 0.071\ 0.097\ 0.121\ 1.134\ 0.136\ 0.122\ 0.092\ 0.073\ 0.063\ 0.057] \quad \text{Eq. 43}$$

$$Id = C(m) * SI * Fs + SRfl * SI * [C(m) + sind(Altitude)] * Fg \quad \text{Eq. 44}$$

Where:

Fs	Factor from defuse radiation from the sky
Fg	Factor from defuse radiation from other surfaces
$SRfl$	Average reflectivity of the environment

2.3.1.4 *Total radiation*

The effective radiation (It) is the sum of the beam and diffuse radiation as given in Eq. 45.

$$It = (Ib + Id)[W/m^2] \quad \text{Eq. 45}$$

Running the simulation for the 15th of July gives the radiation curves of Figure 6.

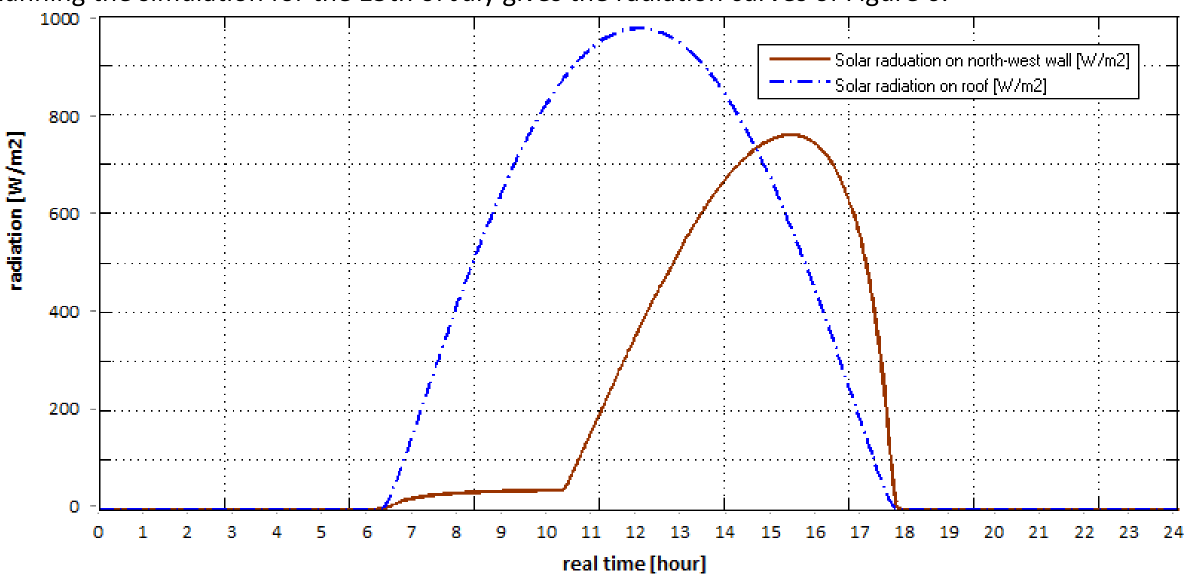


Figure 6: Radiation curves on the roof and wall of the auditorium for the 15th of July for midnight to midnight (without considering clouds)

2.3.2 Internal heat generation

The model assumes that 30 people are inside the room during all the possible class hours. When these people are inside the room they are each considered to add 130W of heat to the environment. And whenever people are in the room it is assumed they are using 1 computer with a beamer and have the lights on which add 700W of heat to the environment. Outside the class hours there are no people in the room and no heat is generated inside. The class hours are as following: 7- until 7.50am, 8 until 9.40am, 10 until 11.40am, 1pm until 1.50pm, 2 until 3.40pm, 4 until 5.40pm, 6.40 until 8.20pm and occasionally from 8.40 until 10pm as shown in Figure 7.

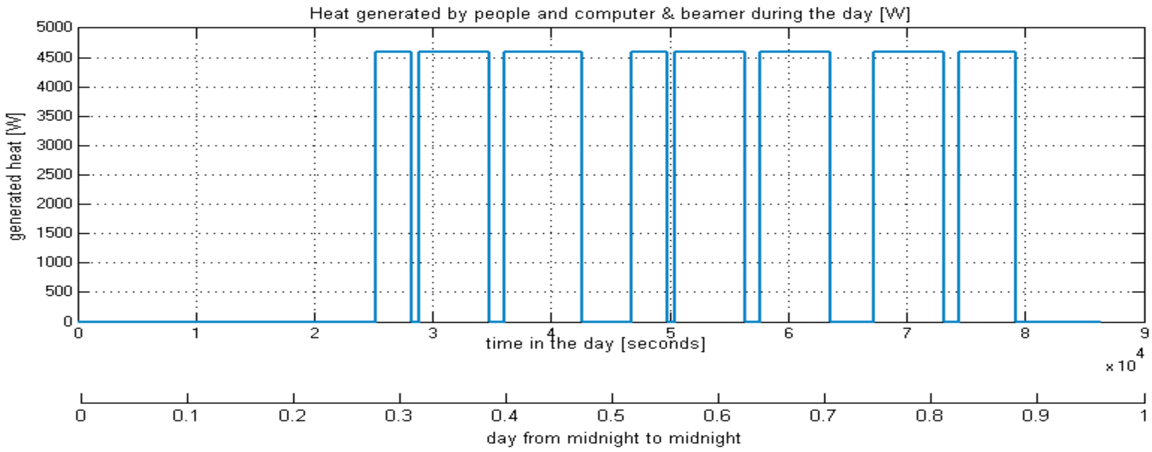


Figure 7: Heat added to the room by 30 people, lamps and one computer during the day

2.3.3 CO_2 emission from people

Figure 8 shows the carbon dioxide emission per person for different levels of activity (6).

Activity	Respiration per Person *(m^3/h)*	Carbon Dioxide Emission per person *(m^3/h)*
Sleep	0.3	0.013
Low activity work	0.5	0.02
Normal work	2 - 3	0.08 - 0.13
Hard work	7 - 8	0.33 - 0.38

Figure 8: The carbon dioxide emission from people is a function of their activity. Source: http://www.engineeringtoolbox.com/co2-persons-d_691.html

The people in this study are each assumed to produce 0.01 cubic meters per hour.

2.3.4 Moisture emission from people

The moisture added to the atmosphere by breathing and sweating (H_{people}) is assumed to be $\frac{60\ gram}{hour*person}$ (7).

2.3.5 Outdoors temperature and humidity

The temperature and humidity data is taken for each hour of the day for the 15th day of every month from the database available on the website of the LabEEE (Laboratório de Eficiência Energética de Edificações (8)). The temperature during an hour is assumed to be constant. The monthly data is from different years (2006-2009) chosen in a way that the data is most representative for the specific month in a random year.

2.3.6 Temperature surrounding spaces

The temperature in the surrounding rooms in this model is assumed to be the same as the temperature outside with an delay of two hours.

2.4 Environmental control equipment

2.4.1 Solar chimney

The model works with natural ventilation produced by a solar chimney. A solar chimney uses natural air flow caused by thermal forces and if building internal resistance is not to high the air flow caused by this stack effect can be expressed by Eq. 46 (9):

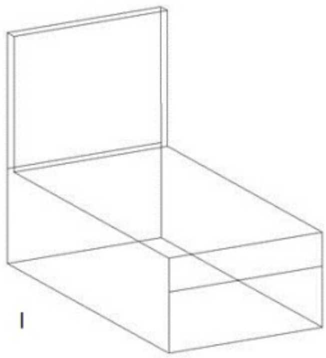
$$\dot{V} = C_D * A * \sqrt{2g * \Delta H_{NPL} \frac{Ti - To}{Ti}} \qquad \text{Eq. 46}$$

Where:

C_D	Discharge coefficient for opening
A	Cross-sectional area of opening, m^2
ΔH_{NPL}	Height from midpoint of lower opening to NPL [m]

Ti	indoor temperature, [K]
To	outdoor temperature [K]
g	acceleration of gravity [9,807 m/s²]

The solar chimney stimulates the air flow caused by increasing the ***ΔHNPL*** and the ***ΔT*** according to Eq. 46. The ***ΔHNPL*** depends on the height of the chimney and the ***ΔT*** depends on the average indoor temperature which is a function of both the room temperate and the temperature of the air inside the chimney Eq. 55. It is preferable to keep the room temperature low to stimulate thermal comfort and try to get the temperature in the chimney high using the natural resource of solar radiation. A solar chimney average temperature SPACE-STATE equation is introduced according to the Eq. 47- Eq. 54.



$$\boldsymbol{\rho_{air} * c_{air} * V_{SC} * \frac{dT_{sc}}{dt} = \dot{Q}vent + \sum \dot{Q}_{SCsurfaces}} \qquad \text{Eq. 47}$$

Eq. 54 can be rewritten in the SPACE-STATE form resulting in Eq. 48 and Eq. 49.

$$x' = ((-\frac{U_{SC}*SA_{tot}}{C19}) * T_{SC} + \begin{pmatrix} \frac{C20*SA_{NW}}{C19} & \frac{C20*SA_{NE}}{C19} & \frac{C20*SA_{SE}}{C19} & \frac{C20*SA_{SW}}{C19} & \frac{U_{SC}*SA_{tot}}{C19} & -\frac{c_{air}*\rho_{air}}{C19} \end{pmatrix} * \begin{pmatrix} I_{NW} \\ I_{NE} \\ I_{SE} \\ I_{SW} \\ T_o \\ (T_{SC}-T_R)*\dot{V} \end{pmatrix} \qquad \text{Eq. 48}$$

$$y = \begin{pmatrix} 1 \\ -U_{SC} * SA_{tot} \end{pmatrix} * T_{SC} + \begin{pmatrix} 0 & 0 & 0 & 0 & 0 & 0 \\ C20 * SA_{NW} & C20 * SA_{NE} & C20 * SA_{SE} & C20 * SA_{SW} & U_{SC} * SA_{tot} & 0 \end{pmatrix} * \begin{pmatrix} I_{NW} \\ I_{NE} \\ I_{SE} \\ I_{SW} \\ T_o \\ (T_{SC}-T_R)*\dot{V} \end{pmatrix} \qquad \text{Eq. 49}$$

Where:

$$C19 = c_{air} * \rho_{air} * V_{SC} \qquad \text{Eq. 50}$$

$$SA_{tot} = SA_{NW} + SA_{NE} + SA_{SE} + SA_{SW} \qquad \text{Eq. 51}$$

$$C20 = \tau_{cov} * abs_{SC} \qquad \text{Eq. 52}$$

Where:

SA_{tot}	Surface area total of the solar chimney [m^2]			chimney [°C]
$SA_{NW..}$	Surface area north-west face of the solar chimney[m^2]		$\dot{Q}_{SCsurfaces}$	Heat load trough solar chimney surface [kJ]
U_{SC}	Thermal transmittance solar chimney walls of 5 $[W/(K*m^2)]$		U_{sc}	Thermal transmittance of the solar chimney [5W/K*m^2]
τ_{cov}	Transmittance factor of 0.7		$I_{surface}$	Radiation on solar chimney surface [W/m^2]
abs_{SC}	Absorbance factor of 0.7		τ_{cov}	Transmittance of wind blocking and insulation material [0.7]
I_{NW}	Radiation on north-west face [W/m^2]			
T_{sc}	Average temperature of the solar		V_{SC}	Volume of the solar chimney [m^3]

$\dot{Q}vent$ is the cooling of this air due to ventilation and $\dot{Q}_{SCsurfaces}$ is the heat that is conducted through the solar chimney walls.

$$\dot{Q}_{vent} = (T_{SC} - T_r) * \dot{V} * \rho_{air} * c_{air} \quad \text{Eq. 53}$$

$$\sum \dot{Q}_{SCsurfaces} = \sum_{i=surface\ 1}^{surface\ 4} (U_{sc} * A_{surface} * \left(I_{surface} * abs_{SC} * \tau_{cov} + (T_r - T_{sc})\right)) \quad \text{Eq. 54}$$

The model calculates the radiation on the different surfaces according to the methods described in section 2.3.1.

$$Ti = \frac{Tsc * V_{SC} + T_r * V_r}{V_{SC} + V_r} \quad \text{Eq. 55}$$

In order to determine the solar chimney strength behavior researchers at the UFV have been using computed fluid dynamics (CFD) software to simulate an environment with the exact same dimensions. The approach (10) is summarized here:

Draw the room in SolidWorks, convert to ANSYS and determine the heated surface(s), in this case only one surface of the solar chimney. Describe the openings, being the entrance and exit for the air. During these simulations the outside temperature is constantly 25$°C$. The mesh size is shown in Figure 9.

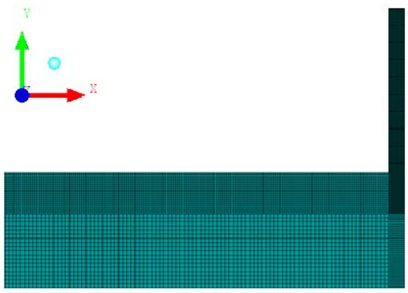


Figure 9: Mesh hexane dominant "type D", with elements of mesh size 0.1 m, 0.05 m and 0.025 m.

The CFD simulations run for three configurations with different solar chimney heights, respectively 1, 2 and 4 meter high. Figure 10 shows the room with chimney (4 meter high) and a box around it that contains atmosphere. The heated surface is highlighted. The air speed in the room is established as shown in the right picture of Figure 10.

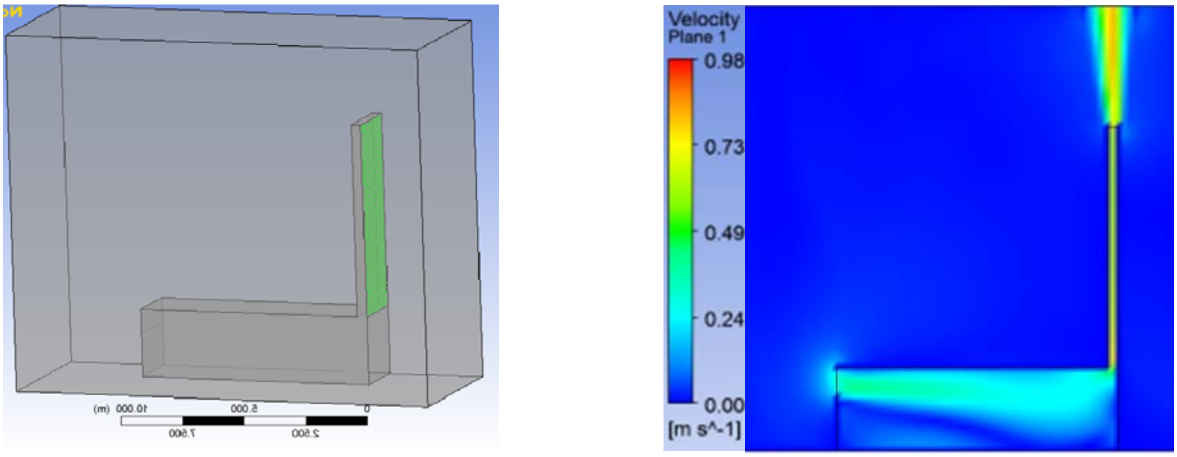


Figure 10: The model and the flow rate distribution which is established in the auditorium

Next the simulation runs with different temperatures of the heated surface of the solar chimney. The average inside temperature and the air flow generated are calculated by the software. Plotting this gives the graph of Figure 11 and a correlation that can be used in the Matlab simulation.

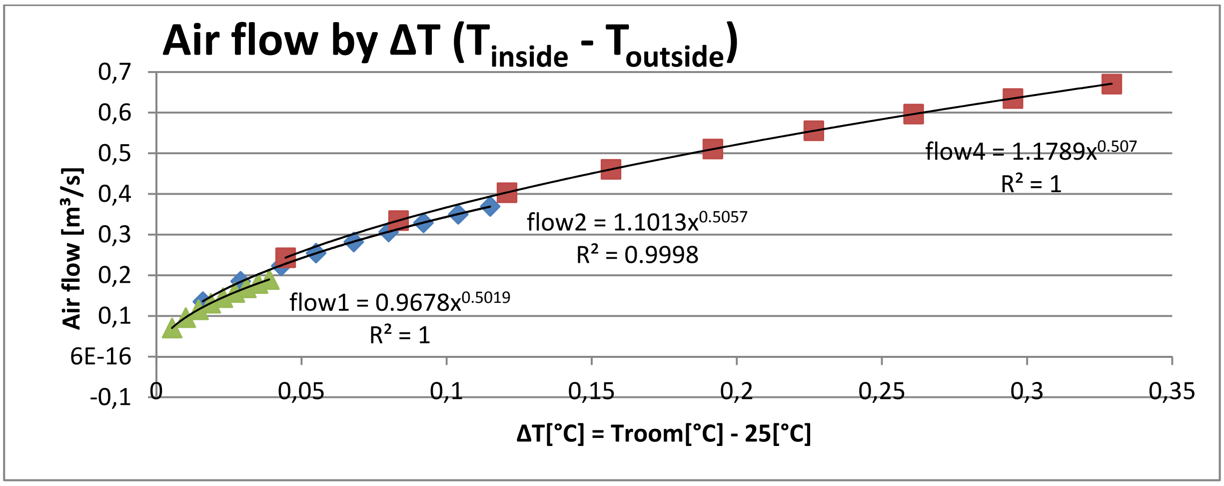


Figure 11: Correlations ΔT(outside temp - avg. inside temp) against air flow, for the configurations with a solar chimney of 1 meter high(flow1), 2 meter high(flow2) and 4 meter high(flow4).

The correlations displayed in Figure 11 show a great similarity to Eq. 46 which claims that the airflow is calculated by constants multiplied by $\sqrt{(T_r - T_o)/T_o}$ where all temperatures are given in Kelvin. The temperature range in which the formula is being used allows it to write the correlation as a constant multiplied by $\sqrt{T_r - T_o}$.

2.4.2 Air-conditioner

The assumptions for the model: the air-conditioner works in a steady state, the outlet temperature of condensed water is equal to the air temperature at the outlet of the air-conditioner, there is no work performed, the reference temperature is considered zero degrees Celsius, the heat exchange is sensitive until the saturation point after that it is only latent heat loss. The saturation temperature for any particular humidity can be found with Eq. 56, fitted to points from the psychometric chart, Figure 12. And the amount of enthalpy drop it would require to come to that point is described by formula Eq. 57.

$$T_{sat} = 15.66 * \ln(1000 * w_e) - 21{,}695 \qquad \text{Eq. 56}$$
$$Q_{ssat} = c_p * (T_{sat} - Taer) + w_e * 1.82 * (T_{sat} - T_{aer}) \qquad \text{Eq. 57}$$
$$\dot{Q}_{AC} = Q_{electric} * COP \qquad \text{Eq. 58}$$

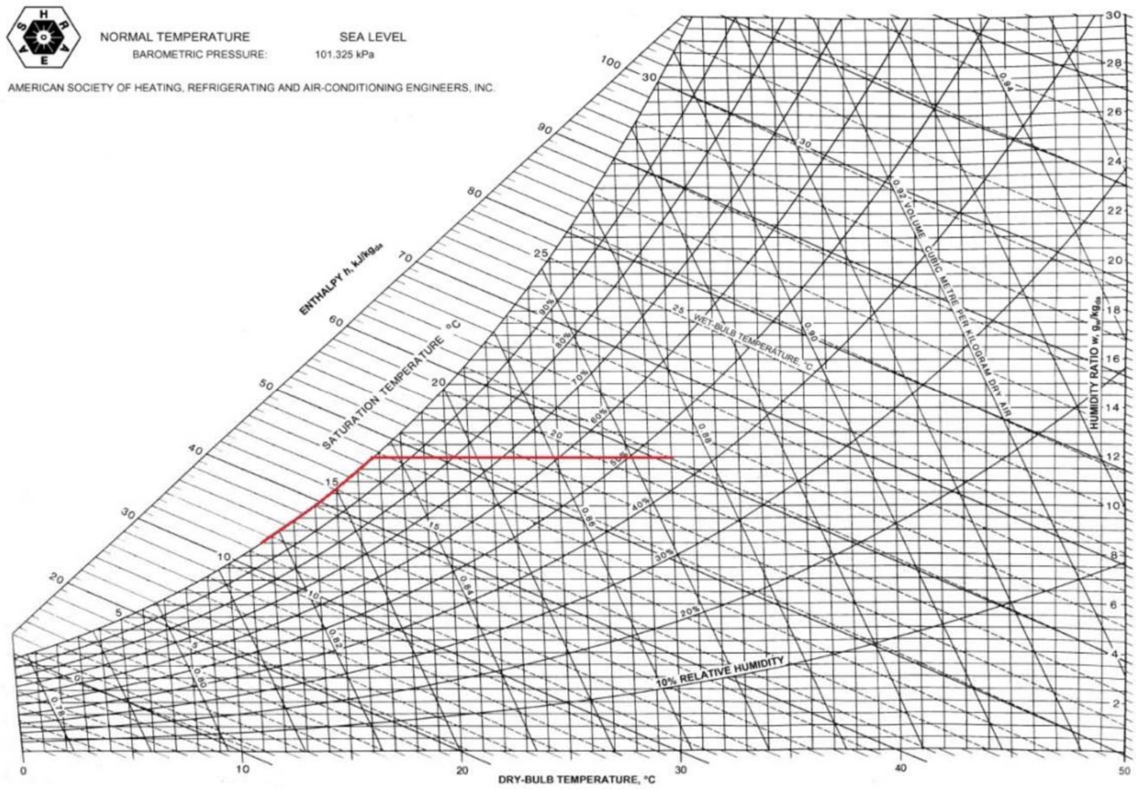


Figure 12: Psychometric chart. Adapted from ASHRAE.

If $(Qac/ma < Qssat)$ there is only sensible heat loss and the absolute humidity at the air-conditioner exit is the same as at the inlet. The temperature of the outlet air is described in Eq. 59.

$$Tsar = (Q_{AC}/ma + Taer * (cp + 1.82 * we))/(cp + 1.82 * we) \qquad \text{Eq. 59}$$

Where:

$Tsat$	Saturation temperature of the atmospheric air [°C]
w_e	Absolute humidity at the inlet of the air-conditioner [kg/kg]
$Qssat$	Enthalpy differential to reach the saturation line [kJ/kg]
C_p	Specific heat [J/(kg*K)]
$Taer$	Air temperature at the entrance of air-conditioning [°C]
$Tsar$	Air temperature at the outlet of the air-conditioner (° C)
$\dot{Q}ac$	Effective cooling charge of the air-conditioner (W)
ma	Air mass flow rate through the air-conditioner(kg / s)

If $(Q_{AC}/ma > Qssat)$ then a mass balance is performed for the air-conditioner:

$$\sum inlet\ mass - \sum outlet\ mass = 0 \qquad \text{Eq. 60}$$

Showing that

$$m_l = (w_e - w_s) * m_a \qquad \text{Eq. 61}$$

For energy balance, we have that:

$$\sum inlet\ energy = \sum outlet\ energy - Qac \qquad \text{Eq. 62}$$

Making mathematical manipulations to isolate the term referring to the cooling capacity of the air-conditioner Q_{AC} makes Eq. 63 and Eq. 64.

$$\frac{Q_{ac}}{m_a} = (h_s - h_e) + (w_e - w_s)(h_{ls} - h_{lsat}) \quad \text{Eq. 63}$$

$$\frac{Q_{ac}}{m_a} = (h_{as} - w_s h_{gs}) + \left(h_{ae} - w_{e*h_{ge}}\right) + (w_e - w_s)(h_{ls} - h_{lsat}) \quad \text{Eq. 64}$$

Where:

m_l	Mass flow of condensed water (kg / s)
Ws	Absolute humidity at the outlet of the air-conditioner (kg / kg)
hs	Enthalpy of the out coming air from air-conditioner (kJ / kg)
he	Enthalpy of the incoming air from air-conditioner (kJ / kg)
Hls	Enthalpy of compressed liquid at the outlet of the air-conditioner (kJ / kg)
$Hlsat$	Enthalpy for the latent heat of condensation (kJ / kg)

Has	Enthalpy of dry air at the outlet of the air-conditioner (kJ / kg)
Hgs	Enthalpy of steam at the outlet of the air-conditioner (kJ / kg)
Hae	Enthalpy of dry air at the entrance of the air-conditioner (kJ / kg)
Hge	Enthalpy of the steam at the entrance of the air-conditioner (kJ / kg)

Curve-fitting is performed on thermodynamic tables to create expressions for h_{ls}, h_{lsat} and T_{sar}. These expressions are used to rewrite the mass balance as a function of w_s.

$$h_{ls} = 4.1855 * T_{sar} + 0.1306 \quad \text{Eq. 65}$$

$$h_{lsat} = 2500.9 - 1.82 * T_{sar} \quad \text{Eq. 66}$$

$$T_{sar} = 1243.3 * w_s - 1.2333 \quad \text{Eq. 67}$$

Thus, the mass balance is

$$\frac{Qac}{ma} = c_p.T_{sar} + w_s * (2500{,}9 - 1{,}82.T_{sar}) - c_p.T_{ear} - w_e\,(2500{,}9 - 1{,}82.T_{sar})(w_e -)(h_{ls} - h_{lsat}) \quad \text{Eq. 68}$$

Inserting Eq. 67 into Eq. 68 yields a second degree quadratic equation with w_s as an independent variable. The equation has the form $A * {w_s}^2 + B * w_s + C = 0$, and the coefficients are:

$$A = -2941.026 * \left[\frac{kJ}{kg}\right] \quad \text{Eq. 69}$$

$$B = 6214.622 + 5203.83 * we + 28.5012 * \log(1000 * we) \left[\frac{kJ}{kg}\right] \quad \text{Eq. 70}$$

$$C = -\left[we * (5006.832 + 1{,}82 * Tear + 1.82 * Tsat) + 1.239 + 1.005 * Tear + \frac{Q}{ma}\right]\left[\frac{kJ}{kg}\right] \quad \text{Eq. 71}$$

Solving this equation gives two answers; the right answer must be positive and in the case that both are positive it is the smaller number.

The amount of condensed moisture in the air-conditioner (H_{AC}) is given by Eq. 72

$$H_{AC} = ma * (we - ws) * 10^3 \left[\frac{g}{s}\right] \quad \text{Eq. 72}$$

2.5 Indoor air quality and thermal comfort index

2.5.1 Temperature range

The CSA Standard CAN/CSA Z412-00 (R2011) and (ASHRAE) Standard 55 – 2010 give acceptable ranges of temperature and relative humidity for offices. The recommended temperature ranges have been found to meet the needs of at least 80% of individuals. Some people may feel uncomfortable even if these values are met (5) (11) (12).

Conditions	Relative humidity	Acceptable operating temperatures
Summer (light clothing)	If 30%, then If 60%, then	24.5 - 28[°C] 23 - 25.5[°C]
Winter (warm clothing)	If 30%, then If 60%, then	20.5 - 25.5[°C] 20 - 24[°C]

Figure 13: Acceptable room temperatures for different circumstances. Adapted from ASHRAE 55-2010.

A simple air flow (e.g. generated by a ceiling fan) compared to a situation without air flow will allow you to raise the thermostat setting about 2.5°C with no reduction in comfort (13).

2.5.2 Predicted mean vote (PMV)

The PMV model is based on the International Standard 7730 which describes the ergonomics of the thermal environment. The PMV is an index that predicts the mean value of the votes of a large group of persons on the 7-point thermal sensation scale (Figure 14), based on the heat balance of the human body. Thermal balance is obtained when the internal heat production in the body is equal to the loss of heat to the environment. In a moderate environment, the human thermoregulatory system will automatically attempt to modify skin temperature and sweat secretion to maintain heat balance (14).

PMV	Sensation
+ 3	Hot
+ 2	Warm
+ 1	Slightly warm
0	Neutral
-1	Slightly cool
-2	Cool
-3	Cold

Figure 14: Seven point thermal sensation scale. Adapted from ISO 7730.

The PMV can be calculated with the following set of equations:

$$PMV = \left[0{,}303 \cdot \exp(-0{,}036 \cdot M) + 0{,}028\right] \cdot \left\{ \begin{array}{l} (M-W) - 3{,}05 \cdot 10^{-3} \cdot \left[5\,733 - 6{,}99 \cdot (M-W) - p_a\right] - 0{,}42 \cdot \left[(M-W) - 58{,}15\right] \\ -1{,}7 \cdot 10^{-5} \cdot M \cdot (5\,867 - p_a) - 0{,}0014 \cdot M \cdot (34 - t_a) \\ -3{,}96 \cdot 10^{-8} \cdot f_{cl} \cdot \left[(t_{cl}+273)^4 - (\bar{t}_r + 273)^4\right] - f_{cl} \cdot h_c \cdot (t_{cl} - t_a) \end{array} \right\}$$

$$t_{cl} = 35{,}7 - 0{,}028 \cdot (M-W) - I_{cl} \cdot \left\{3{,}96 \cdot 10^{-8} \cdot f_{cl} \cdot \left[(t_{cl}+273)^4 - (\bar{t}_r+273)^4\right] + f_{cl} \cdot h_c \cdot (t_{cl} - t_a)\right\}$$

$$h_c = \begin{cases} 2{,}38 \cdot \left|t_{cl} - t_a\right|^{0{,}25} & \text{for} \quad 2{,}38 \cdot \left|t_{cl} - t_a\right|^{0{,}25} > 12{,}1 \cdot \sqrt{v_{ar}} \\ 12{,}1 \cdot \sqrt{v_{ar}} & \text{for} \quad 2{,}38 \cdot \left|t_{cl} - t_a\right|^{0{,}25} < 12{,}1 \cdot \sqrt{v_{ar}} \end{cases}$$

$$f_{cl} = \begin{cases} 1{,}00 + 1{,}290\, l_{cl} & \text{for } l_{cl} \leqslant 0{,}078 \text{ m}^2 \cdot \text{K/W} \\ 1{,}05 + 0{,}645\, l_{cl} & \text{for } l_{cl} > 0{,}078 \text{ m}^2 \cdot \text{K/W} \end{cases}$$

Where:

M	The metabolic rate $[W/m^2]$	v_{ar}	The relative air velocity $[m/s]$
W	The effective mechanical power $[W/m^2]$	p_a	The water vapor partial pressure [Pa]
I_{cl}	Clothing insulation $[m^2K/W]$	h_c	The convective heat transfer coefficient $[W/(m^2 * K)]$
f_{cl}	The clothing surface area [factor]	t_{cl}	The clothing surface temperature $[°C]$
t_a	The air temperature $[°C]$		
t_r	The mean radiant temperature $[°C]$		

NOTE 1: metabolic unit = 1 met = 58,2 W/m2; 1 clothing unit = 1 clo = 0,155 m2 ·�°C/W.

Using range

PMV may be calculated for different combinations of metabolic rate, clothing insulation, air temperature, mean radiant temperature, air velocity and air humidity (see ISO 7726). The equations for t_{cl} and h_c may be solved by iteration.

The PMV index is derived for steady-state conditions but can be applied as a good approximation during minor fluctuations of one or more of the variables, provided that time-weighted averages of the variables during the previous 1 hour period are applied.

The index should be used only for values of PMV between -2 and +2 and when the six main parameters are within the following intervals: Metabolic rate of 46-232 $[W/m^2]$ (0,8 met to 4 met); Clothing insulation of 0-0,310 m^2K/W (0 clo to 2 clo); Air temperature of 10-30 [°C]; Mean radiant temperature of 10-40 [°C] and an air velocity of 0-1 [m/s].

2.5.2.1 Mean radiant temperature

The radiant temperature of each surface is obtained using the simple model of Eq. 73. Next the mean (average) radiant temperature is calculated using Eq. 74 (15). These T_{MRT} equations are implemented in the room temperature SPACE-STATE function. The floor temperature is not included in the equation.

$$T_{ri} = (T_r + 273) + \frac{\dot{Q}_{roof}}{A_{roof}} * R_{se} \;\; [K] \qquad \text{Eq. 73}$$

$$T_{MRT} = \frac{T_{ri} * A_{roof} + T_{wo} * A_{wo} + T_{wi} * A_{wi}}{A_{tot}} - 273[°C] \qquad \text{Eq. 74}$$

Where:

T_{ri}	Internal surface temperature roof [K]	A_{wo}	Total area of external walls $[m^2]$
$\dot{Q}_r$	Heat load through the roof [W]	T_{wi}	Internal surface temperature inner walls [K]
T_{MRT}	Mean radiant temperature in the room$[°C]$	A_{wi}	Total area of internal walls $[m^2]$
T_{wo}	Internal surface temp. external wall [K]		

2.5.2.2 Predicted percentage of dissatisfaction (PPD)

The PPD is a function of only the PMV as shown in Eq. 75 and helps with the interpretation of the PMV. As the name already explains, PPD displays the percentage of people who are dissatisfied with the combination of parameters that influences their thermal comfort (14).

$$PPD \;=\; 100 - 95 * exp(-0.03353 * PMV^4 - 0.2179 * PMV^2) \qquad \text{Eq. 75}$$

2.5.3 Humidity

If humidity is too high this will cause discomfort (excessive perspiration, exacerbation of the effects of high temperature, feelings of 'closeness', etc) and if it is too low it can cause respiratory problems. Optimum humidity levels are between 40% and 60% - but in any case they should be kept between

30% and 70%. Humidity levels below 40% will begin to cause problems for workers with conditions such as sinusitis (16) (5).

2.5.4 CO_2 concentration

CO_2 concentration is a important measure of air quality. A high CO_2 concentration negatively affects the performance of the room's occupants. According to The Engineering Toolbox the effects of increased CO_2 levels on adults at good health can be summarized as following (17):

Normal CO_2 Levels

- normal outdoor level (18): *350 - 450 ppm*
- acceptable levels: *< 600 ppm*
- complaints of stiffness and odors: *600 - 1000 ppm*
- ASHRAE and OSHA standards: *1000 ppm*
- general drowsiness: *1000 - 2500 ppm*
- adverse health effects expected: *2500 - 5000 ppm*
- maximum allowed concentration within a 8 hour working period: *5000 ppm*

The levels above are quite normal and maximum levels may occasionally happen from time to time.

Extreme and Dangerous CO_2 Levels

- slightly intoxicating, breathing and pulse rate increase, nausea: *30,000 ppm*
- above plus headaches and sight impairment: *50,000 ppm*
- unconscious, further exposure death: *100.000 ppm*

Carbon Dioxide Standard Levels

The recommendation in ASHRAE standard 62-1989 are

- classrooms and conference rooms *7.1 liter/second* per occupant (corresponds to *1000 ppm CO_2)*
- office space and restaurants *9.5 liter/second* per occupant (corresponds to *800 ppm CO_2)*
- hospitals *12 liter/second* per occupant

2.6 The simulation strategies

The reference room is simulated using four control strategies, being:

1. Without ventilation or air-conditioner.
2. Natural ventilation using a two meter high solar chimney which has the natural ventilation characteristics obtained by the CFD simulations.
3. Air-conditioner controlled environment by a very strong proportional controller which has its set-point at 25 degrees Celsius. The electric capacity is limited to 5kW assuming a constant coefficient of performance (COP) of 3.
4. Using strategy 1 and 2 together. The solar chimney is assumed to have only a one way flow direction and the airflow there is zero when the room is colder than the outside.

3 Results

The results of control strategies 1-3 (as described in section 2.6) are shown in section 3.1. The solar chimney behavior is displayed in section 3.2. The results of control strategy 4 are shown in section 3.3 and the results are discussed in section 3.4.

3.1 Strategies 1,2 and 3

This paragraph shows graphs and discussion for strategies 1, 2 and 3 regarding the thermal comfort and indoor air quality.

3.1.1 Thermal comfort

The extremes 'summer and winter' indicate the thermal comfort ranges using strategies 1, 2 and 3 during the year.

3.1.1.1 Summer

The temperature and predicted mean vote graphs for the summer (Figure 16) show that the solar chimney configuration can keep the room temperature close to the outside temperature. The AC-controlled room keeps the room temperature steady at 25 Celsius, with a good resultant PMV. The temperature in the room without ventilation and air-conditioner goes up to unacceptable levels.

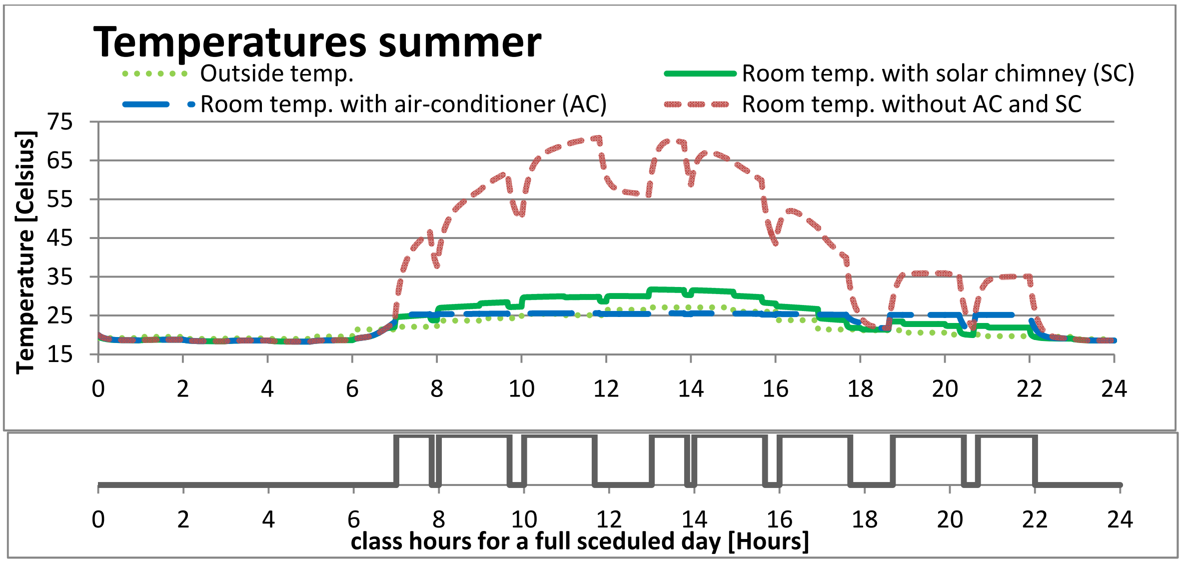


Figure 15:Above shows the room temperature in the summer (15th of January) for case 1, 2, 4 and the outside temperature.

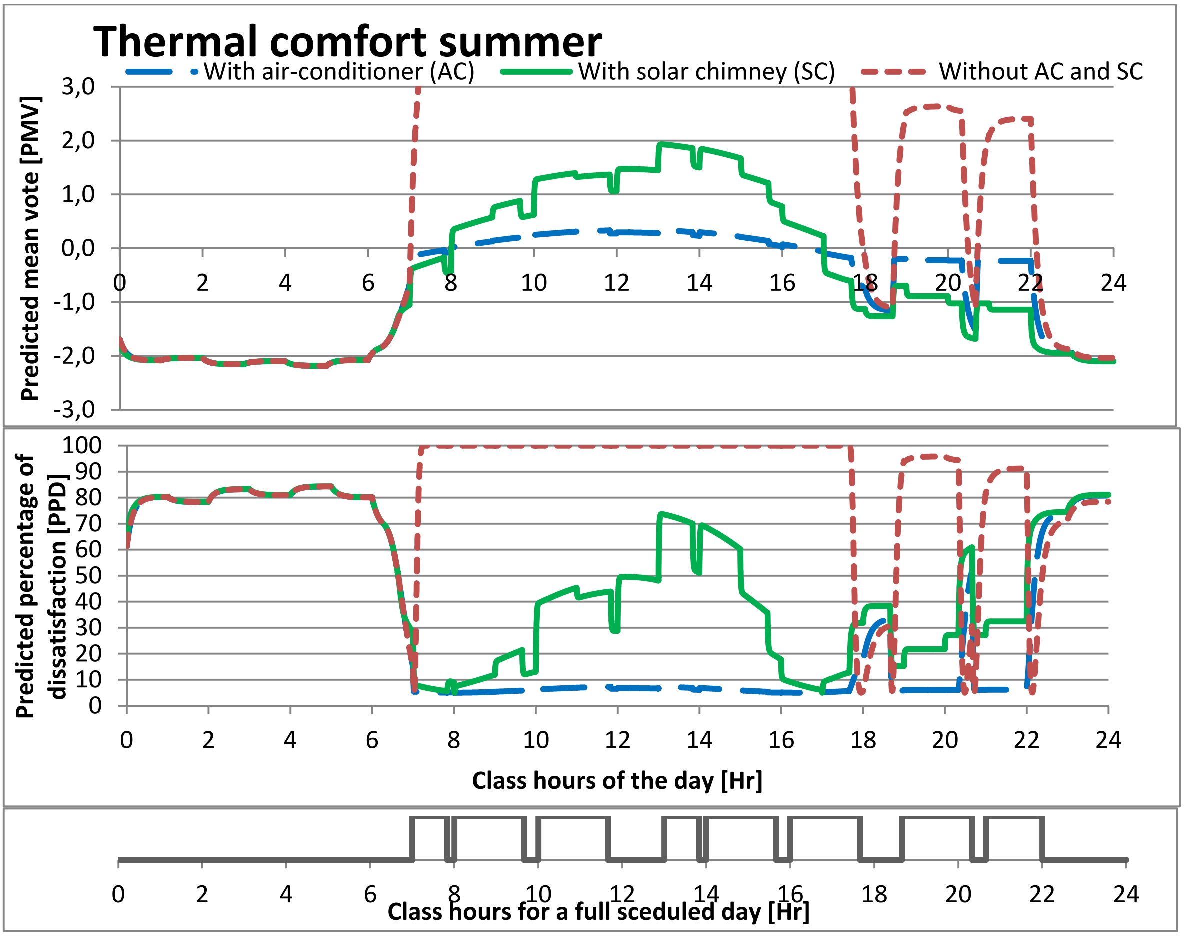


Figure 16: The thermal comfort in summer (15th of January)) for case 1, 2 and 4 and below that the hours where the people are scheduled to be in the room.

As long as the outside temperature is above the inside temperature the ventilation can be used in combination with the air-conditioner. According to Figure 16 this means that the ventilation should only be used between 12pm and 4pm.

It is interesting to note how the PMV for the air-conditioner is not constant at 0 while the temperature at most places is constant; the cause is the mean radiant temperature. The mean radiant temperature goes up to 28.5 degrees Celsius in the middle of the day and 24.6 during the night classes.

With controlled ventilation (controlled with a damper or by window angle control) all conditions between the 'SC' and 'without SC and AC' line can be obtained. The highest thermal comfort therefore can be obtained using a combination of air-conditioner and controlled ventilation.

3.1.1.2 Winter

Figure 18 and Figure 18 show that the solar chimney is able to keep the temperatures below 26 degrees Celsius in the winter. Early and late in the day the solar chimney without controller contributes to a temperature lower than what is comfortable for most people. This also shows the need for a control system on the solar chimney. The solar chimney in that case would be able to promote the same thermal comfort as the air-conditioner. Without ventilation and air-conditioner it is again too hot in the room.

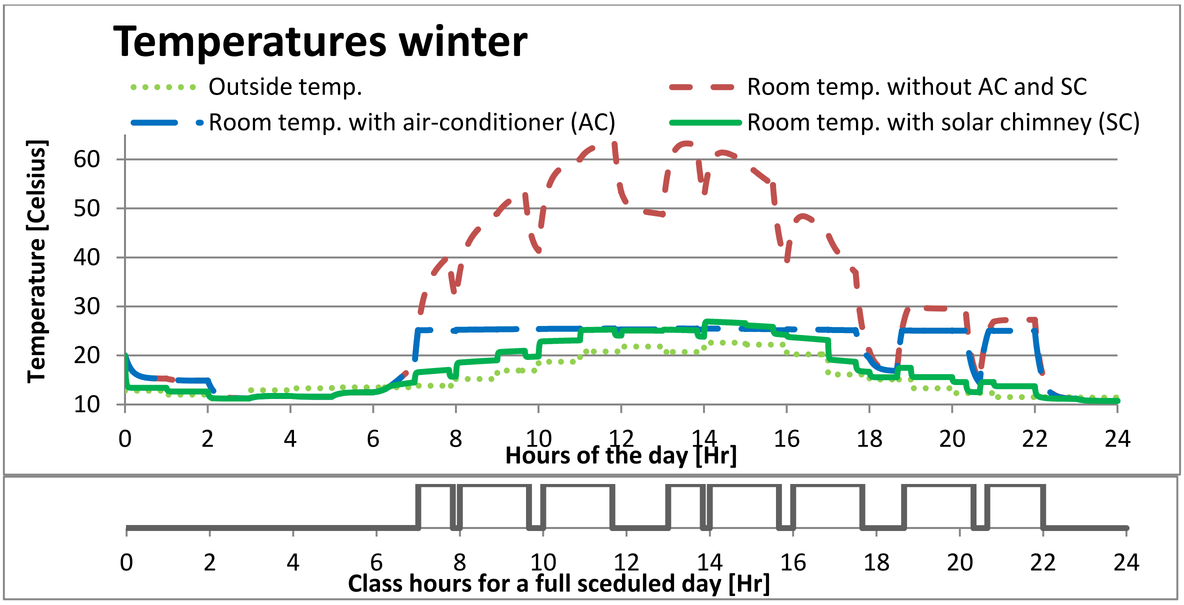


Figure 17: The room temperatures in the winter (15th of July)) for case 1, 2, 4 and the outside temperature.

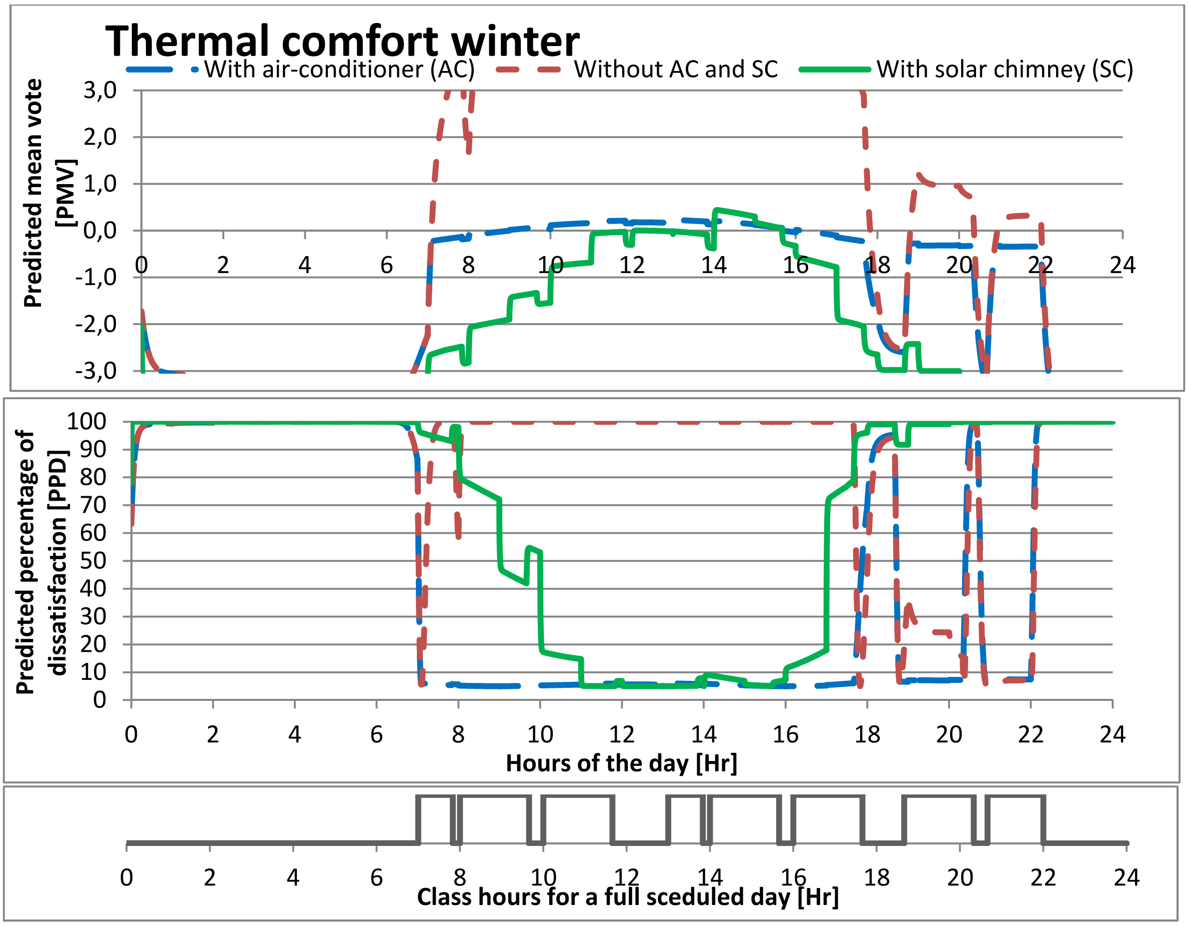


Figure 18: Thermal comfort in the winter (15th of July) for case 1, 2 and 4.

3.1.2 Indoor air quality

Thermal comfort is not the only parameter of importance. The air quality can be measured in humidity and CO_2 concentration. In the case that there is no ventilation the CO_2 concentration would reach highly dangerous levels within one day; 30 thousand PPM is considered slightly intoxicating, increasing breathing and pulse rate and creating nausea (17). This can all be prevented by using a solar chimney, where the CO_2 concentration will be very close to the outside concentration of 400ppm and very far below the ASHRAE standard of 1000ppm.

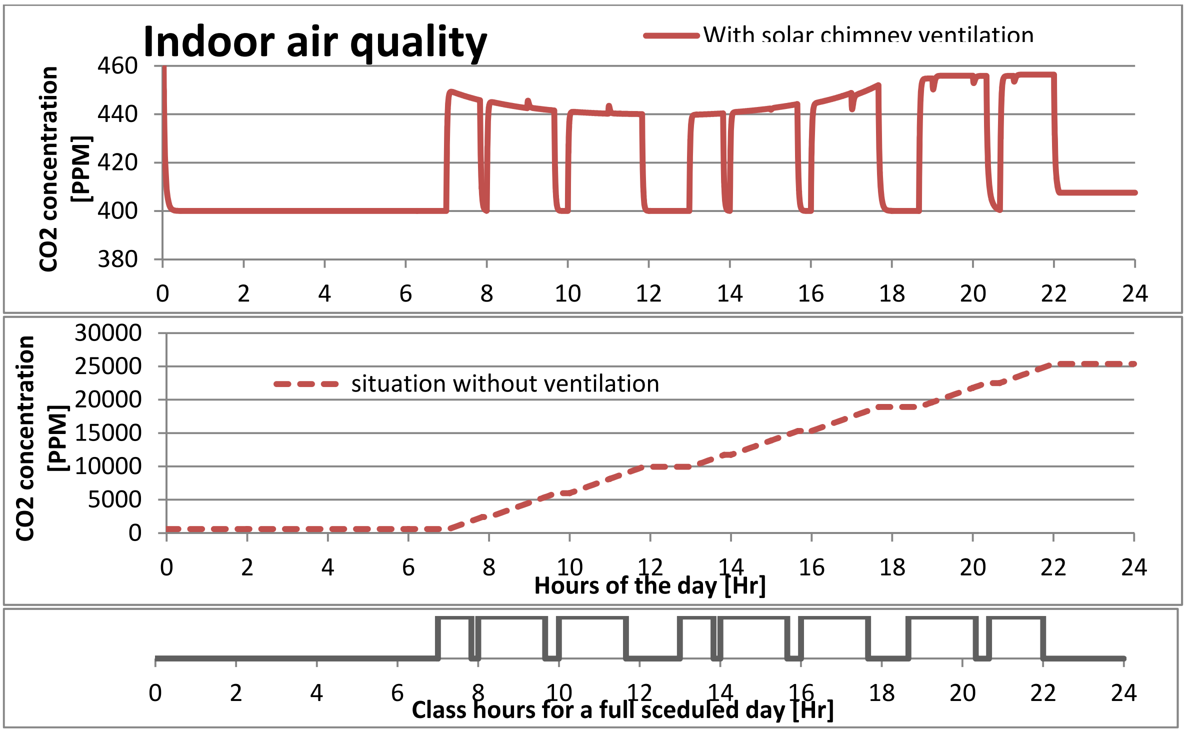


Figure 19: Above the CO_2 concentration when using the solar chimney (without AC) in February, promoting a high air quality. Below the CO_2 concentration without any ventilation, going up to unappreciated levels within one day

3.2 Solar chimney behavior

Figure 20 and Figure 21 show the temperature and PMV for days of all four seasons.

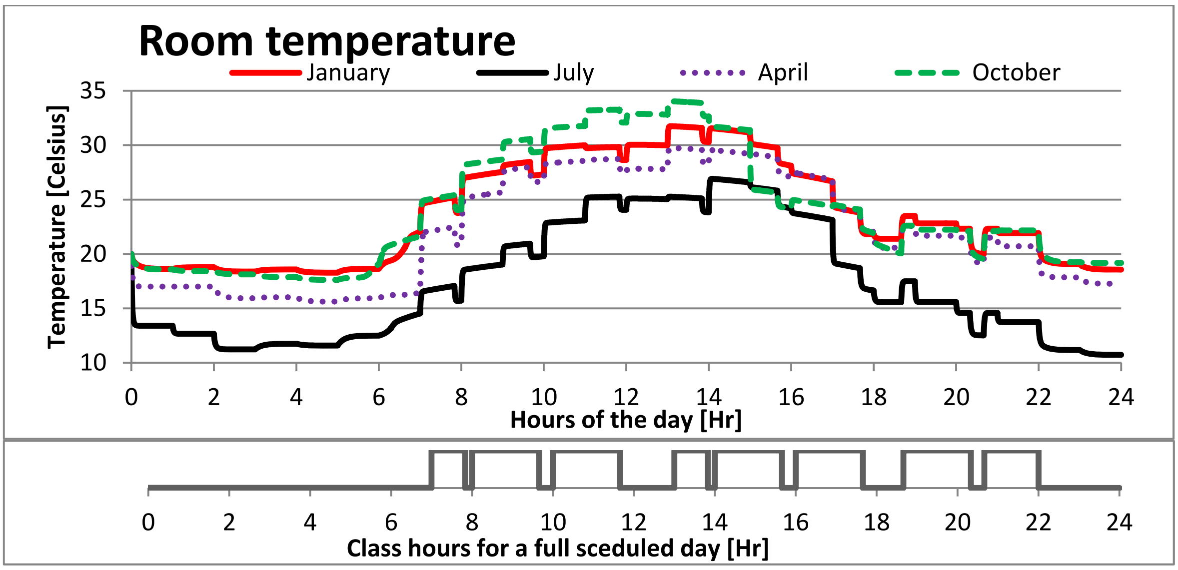


Figure 20: Room temperature using the solar chimney and without using the air-conditioner.

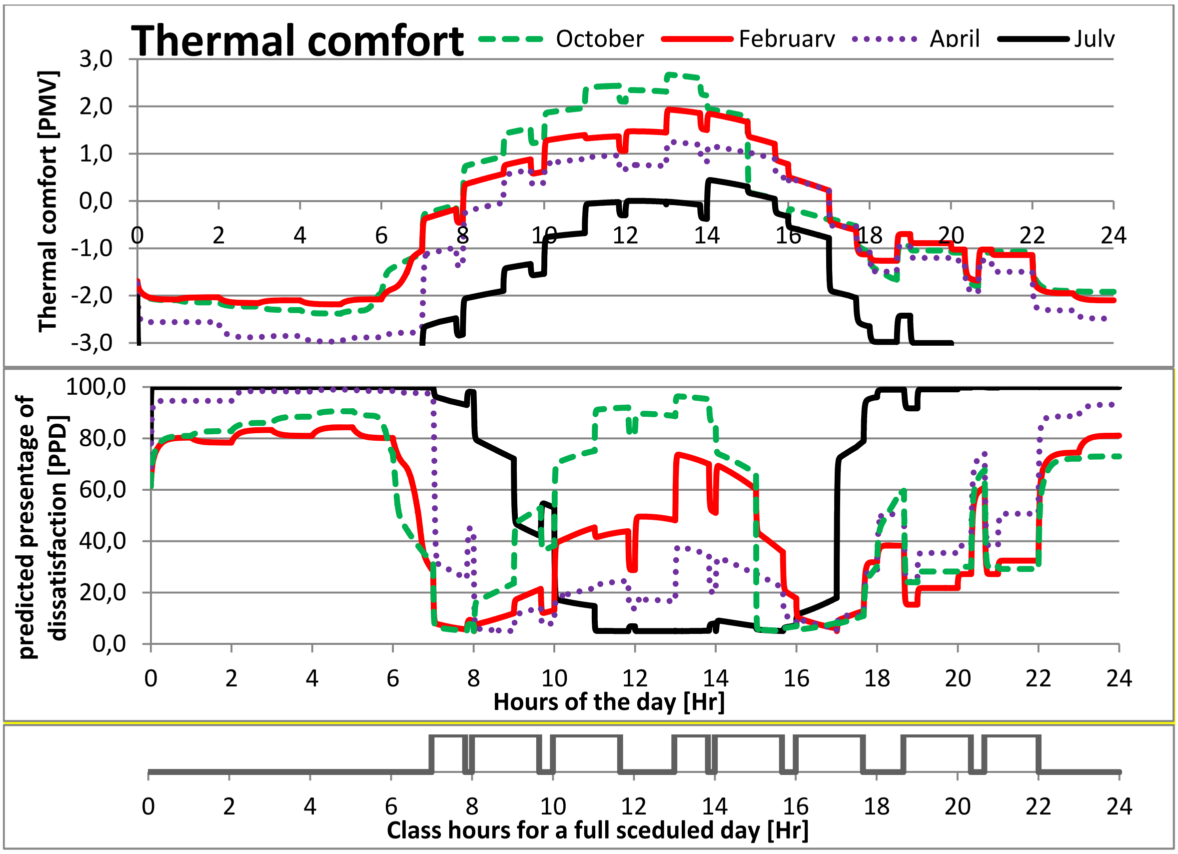


Figure 21: Thermal comfort in different seasons while using the solar chimney without air-conditioner. The lower graph predicted percentage of dissatisfaction as a function of only the PMV using Eq. 75.

The notable fact that the room temperature in October (spring) is higher than in January (summer) is due to the high outside temperature in October. The latitude of Viçosa does not show significant solar radiation differences between the seasons of spring, summer and fall. And besides that the outer wall faces north-west, which means it receives most radiation during spring and fall in the afternoon. The graphs indicate that the solar cannot promote sufficient comfort for most of the year and needs the air-conditioner in addition.

The flow generated by the solar chimney during all seasons while not using an air-conditioner is shown in Figure 22.

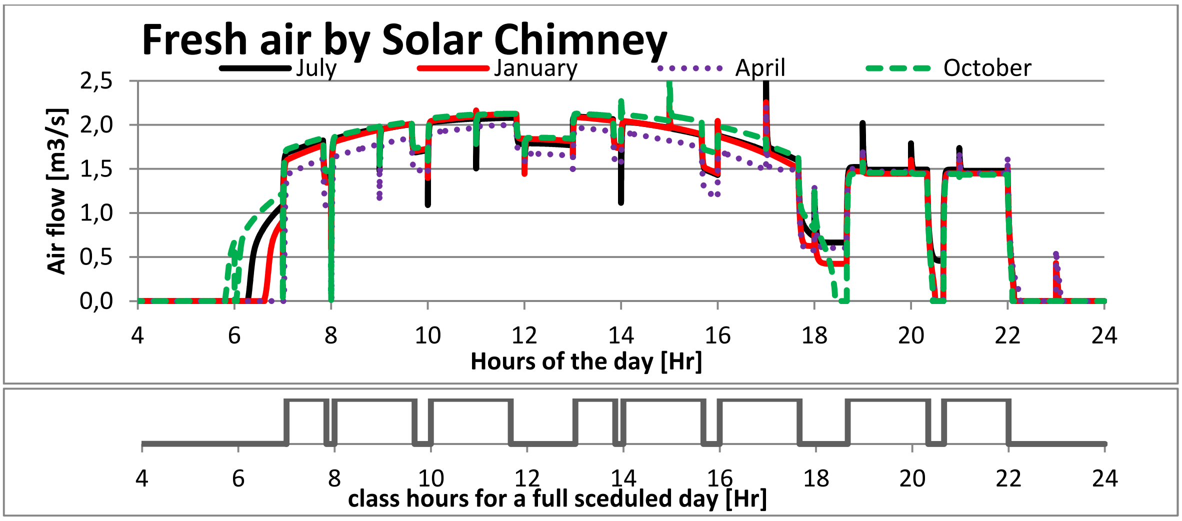


Figure 22: Air flow during the day for the four seasons caused by the solar chimney, while not using an air-conditioning.

The air flows generated during the year in the case that there is no air-conditioner used are pretty much the same. And because air-flow is a function of the temperature difference between inside and outside the room, there is a strong correlation between air flow and room occupation. The airflow in the window area goes up to $1\ m/s$ and the average air-speed in the room is $0.14\ m/s$.

3.3 Combined Solar chimney and air-conditioner

Control strategy 4 concerns the solar chimney and air-conditioner system combined. The air-conditioner controls the temperature at $25.3°C$. When the outside temperature is higher than the room temperature, the airflow is considered zero.

3.3.1 Thermal comfort

Figure 23 and Figure 24 show the room temperature and thermal comfort for control strategy 4.

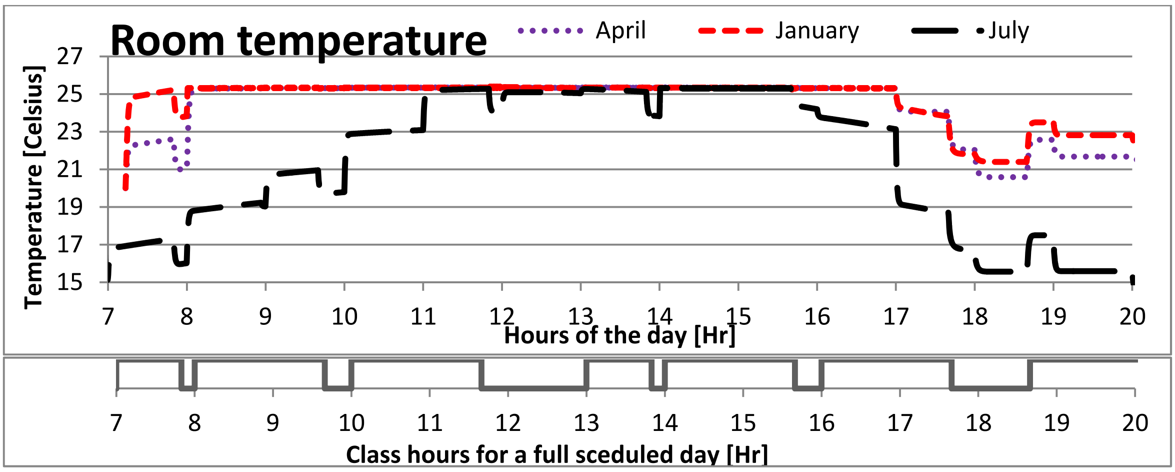


Figure 23: Thermal comfort for solar chimney integrated in an air-conditioned environment

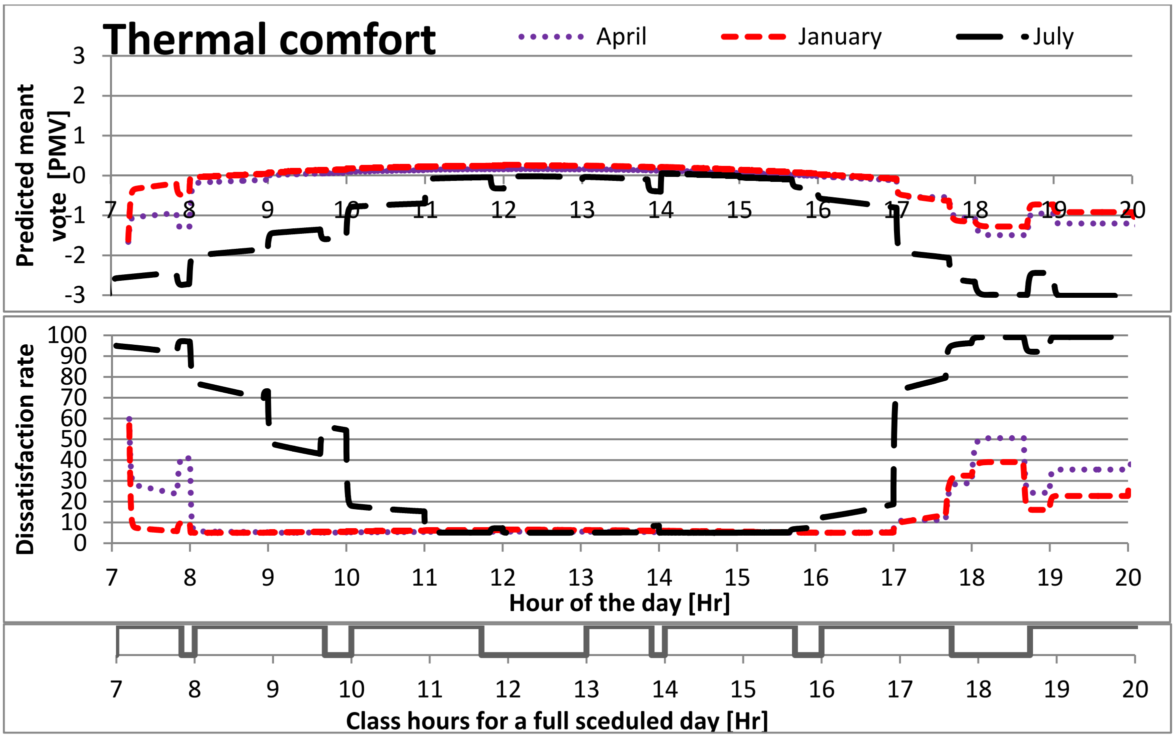


Figure 24: Thermal comfort over the course of a day in January, April and July as well as the predicted percentage of dissatisfied people within this thermal situation.

An incredible thermal comfort is achieved during January and April. The room temperature graphs are very similar to the only air-conditioner graphs whenever the room temperature is above 25°C and look similar to the solar chimney graphs whenever it is below.

3.3.2 Indoor air quality

Figure 25 shows the airflow and Figure 26 the CO_2 concentration for control strategy 4.

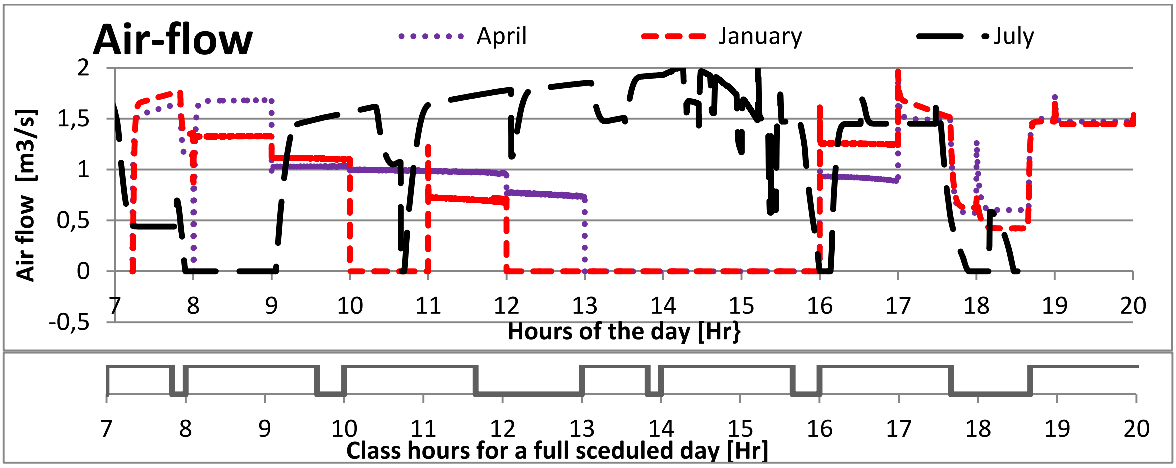


Figure 25: Air-flow generated by the solar chimney when the air-conditioner is active all day to keep the temperature below 25.3 degrees Celsius.

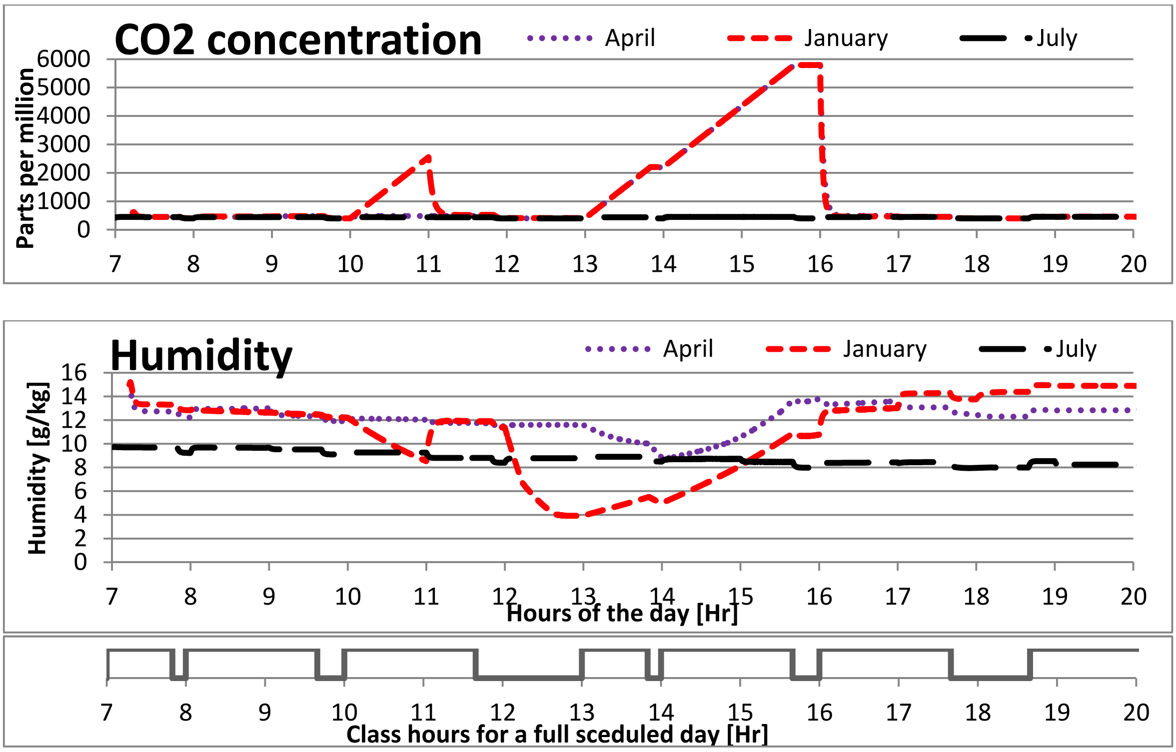


Figure 26: Air-flow and the air-quality measures CO_2 concentration and humidity for the course of a day in January, April and July

The air flow during January (summer) remains 0 during the hottest part of the day because the outside temperature exceeds the room temperature. This results in high CO_2 concentrations which go beyond the ASHRAE standard. However the solar chimney performs excellently in July and in April when it comes to air quality during the course of the day.

3.4 Discussion

Case 1 and 2 are shown not to be able to promote thermal comfort within acceptable ranges while case 3 and 4 perform well. Therefore only case 3 and 4 will be discussed. Although CO_2 concentrations as shown in Figure 19 are not likely to occur, the air quality promoted by the solar chimney is a big improvement. Figure 27 shows that strategy 4 can reduce electricity expenditure of the air-conditioner during every season of the year.

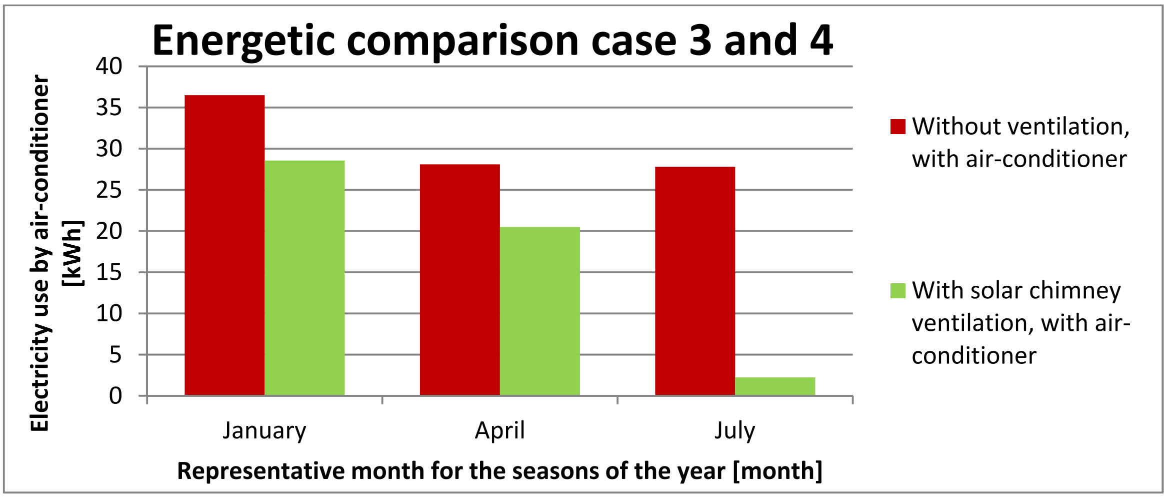


Figure 27:Electricity usage for control strategy 3 and 4 for various months

Figure 28 shows the data on which Figure 27 is based plus the percentage energy saved when comparing case 3 with case 4.

Electric energy savings air-conditioner	January (summer)	April (fall)	July (winter)
Case 3. Without any ventilation, with air-conditioner	36.5 kWh	28.1 kWh	27.8 kWh
Case 4. Solar chimney ventilation, with air-conditioner	28.6 kWh	20.5 kWh	2.2 kWh
Energy saved due to ventilation (case 3 – case 4)	7.9 kWh	7.6 kWh	25.6 kWh
Energy saved due to ventilation, in percentage	21.7%	27.1%	91.9%

Figure 28: Energy consumption of the air-conditioner given for two scenarios for the months of January, April and July.

Comparing Figure 16 Figure 18 with Figure 21 shows that strategy 4 can reduces the amount of hours the AC is required from 13 to 1 hours for a winter day and from 14 to 9 hours for the summer day.

Every modeled external parameters introduces small deviations into the model. One of them is the human sweat rate which is taken to be constant and in fact is heavily influenced by the thermal sensation. Clouds are not taken into consideration in the solar model and the temperature and humidity are assumed to be the same everywhere in the room. Refinement of the model would lead to more accurate simulation results. Simulating more months would serve the same purpose. Nevertheless this model appears to be refined enough to compare different control strategies with one another.

4 Conclusion and future work

Recall that the simulation of the model was performed for a highland tropical climate where air-conditioner is normally used in all the seasons. When integrating the solar chimney system, the electrical consumption of the air-conditioner can be reduced by well over 20% in the summer season, compared to a situation without ventilation. When using a solar chimney in the winter the air-conditioner almost does not have to be used anymore. The air-quality promoted by the solar chimney is remarkably better during the winter than in the summer. The solar chimney has the highest cooling potentials for climates where the outside temperature stays below the desired room temperature for most of the day. In the case of our experiment, the air-flow is mainly influenced by the heat generated by people or extracted by air-conditioner instead of solar radiation on the solar chimney. Considering air quality, the correlation between the quantity of people in a room and the ventilation is desirable. The use of an air-conditioner nullifies this effect and therefore introduces the necessity for more complex control strategies to promote IAQ.

There are many future works imaginable using the current model in the basis. For example: How much comfort can be achieved with a PMV controlled air-conditioner? What additional mechanical ventilation strategy can improve IAQ without excessively increasing air-conditioner electricity expenditure? What is the potential of an evaporative cooling system for this particular building and climate?

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