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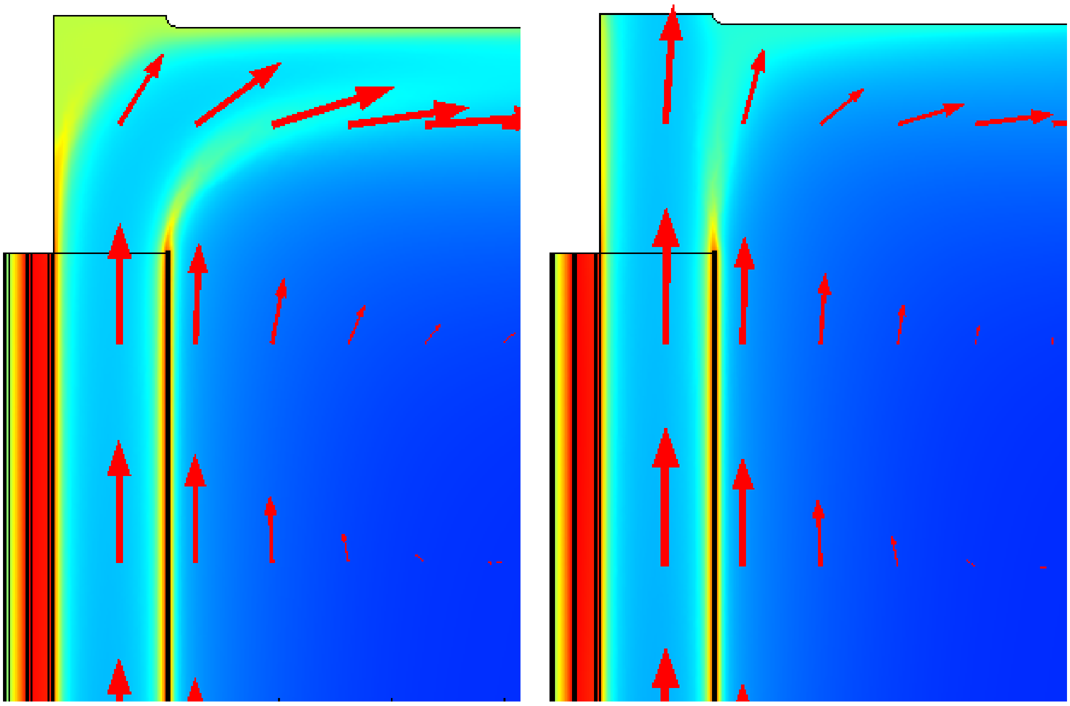

Decreasing g-value by Combining Internal Solar Screen and Exhaust Ventilation

A Numerical Study on what Parameters Affect the Possible Reduction

Master of Science Thesis in the Master's Programme Structural Engineering and Building Technology

EMIL GUSTAFSSON
FREDRIK SÄFBLAD

Department of Civil and Environmental Engineering
Division of Building Technology
CHALMERS UNIVERSITY OF TECHNOLOGY
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Master's Thesis 2014:56

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Department of Civil and Environmental Engineering
Division of Building Technology
Chalmers University of Technology
SE-412 96 Göteborg
Sweden
Telephone: + 46 (0)31-772 1000

Cover:
Illustrations of temperature and velocity fields for the top of the cavity between window and internal solar screen. Picture to the left show a standard case, picture to the right show a ventilated case.

Department of Civil and Environmental Engineering Göteborg, Sweden 2014

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ABSTRACT

In many modern buildings, using large glazed areas in the building envelope, the solar heat load needs to be limited. To do this without disturbing the architectural display of the building, internal solar shading is often the only alternative. However, using internal solar shading means a great part of the incoming solar load still ends up inside the building, causing an often large heat load.

To decrease this problem, one solution is to put exhaust ventilation close to the windows or even connected to the cavities between solar screen and window. The purpose of this is to extract the heated air before it mixes with the air further inside the room.

The purpose of this study is to investigate the possibilities to reduce the g-value of a window and solar screen by using the exhaust ventilation. The aim is to find a reduction factor of the g-value, applicable for these ventilated windows. By changing many design parameters, the aim is to see what parameters affect the efficiency of the system.

The study is performed by creating a numerical model in COMSOL Multiphysics. The heat transfer and air movements around the screen and window are simulated.

The study shows it is difficult to find a general reduction to the g-value by applying ventilation to the cavity. The efficiency of the system depends on numerous parameters. It is first and foremost dependent on exhaust flow but other parameters matter as well. Such parameters are: window height, distance between screen and window, location of exhaust device and which type of window or screen is used.

With a correct and still reasonable design, results suggest that a reduction of the g-value by -45 % should be possible.

Key words: Solar load, g-value, COMSOL, windows, shading, heat transfer, convection

Reducering av g-värde genom kombination av invändig solavskärmning och frånluftsventilation

En numerisk studie över vilka parametrar som påverkar den möjliga minskningen

Examensarbete inom *Structural Engineering and Building Technology*

EMIL GUSTAFSSON

FREDRIK SÄFBLAD

Institutionen för bygg- och miljöteknik

Avdelningen för Byggnadsteknologi

Chalmers tekniska högskola

SAMMANFATTNING

I byggnader, med stora glasade ytor i klimatskalet, behöver den värmelast som solinstrålning orsakar ofta begränsas. Att göra det utan att ändra byggnadens arkitektoniska uttryck kräver ofta att solskärmar sätts på insidan fönster och glasfasader. Användande av invändiga solskydd innebär dock att den stor del av sollasten når in i rummet och skapar en stor värmelast.

En metod för att minska detta problem är att dra fram ventilationskanaler så att byggnadens frånluft tas ut nära fönstren, eller i anslutningen mellan fönster och invändigt solskydd. Syftet med detta är att föra bort den uppvärmda luften för att undvika att den når in i rummet.

Syftet med denna studie är att undersöka möjligheterna att reducera g-värdet för ett fönster ihop med invändig solskärm genom att använda frånluft. En parameterstudie genomförs för att se vilka parametrar som har inverkan på effektiviteten med ett sådant system.

En numerisk modell skapas i COMSOL Multiphysics. Värmeöverföring ihop med luftrörelser runt fönster och skärm simuleras i denna modell.

Studien visar att det är svårt att ta fram ett generellt värde för hur mycket g-värdet reduceras när frånluftsventilation används vid spalt mellan fönster och solskärm. Effektiviteten av systemet varierar med en mängd olika parametrar. Främst frånluftsflöde har inverkan men andra parametrar selar också in. Sådana parameterar är: fönsterhöjd, avstånd mellan skärm och fönster, placering av frånluftsdon, och vilken typ av fönster eller skärm som används.

Resultaten antyder att, vid en korrekt men ändå rimlig design, en reducering av g-värde med -45 % bör vara möjlig.

Nyckelord: Solinstrålning, g-värde, COMSOL, fönster, solskydd, värmeöverföring, konvektion

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Preface

In this thesis project, the effects of combining exhaust ventilation with internal solar shading have been studied in a numerical model, using COMSOL Multiphysics. The work was carried out at the Division of Building Technology, Departement of Civil and Environmental Engineering at Chalmers University of Technology, in Gothenburg, Sweden. The work was carried out in some collaboration with Max Tillberg at Bengt Dahlgren AB.

The examiner of the thesis was Angela Sasic, and the project was supervised by Axel Berge. We want to thank both of them for their guidance and supervising during the work. We also want to send special thanks to Tommie Månsson, at the Division of Building Technology, who many times helped us with modelling in COMSOL, and who kindly lent us his powerful computer.

Finally, we want to send thanks to Max Tillberg, for his help and guidance regarding the aims of this project and for sharing his knowledge about solar shadings combined with exhaust ventilation.

Göteborg May 2014

Emil Gustafsson

Fredrik Säfblad

Notations

Roman case letters

A	Area	[m^2]
Ar	Aspect ratio	[-]
c_p	Specific heat capacity	[J/kg·K]
d	Thickness	[m]
$F_{1,2}$	View factor between two surfaces	[-]
g	Gravitational constant	[m/s^2]
H	Height of cavity	[m]
h	Thermal conductance	[W/m^{2}K]
h_{room}	Height of the test room	[m]
J	Radiosity	[W/m^2]
$\widehat{M}$	Molar mass	[g/mol]
Nu	Nusselt number	[-]
p	Pressure	[Pa]
q	Heat flux	[W/m^2]
qc	Conductive heat flux	[W/m^2]
qr	Radiative heat flux	[W/m^2]
$\mathcal{R}$	Universal gas constant	[J/mol·K]
Ra	Rayleigh number	[-]
r	Reflectance	[-]
S	Absorbed solar radiation	[W/m^2]
T	Temperature	[K]
$\bar{T}$	Average temperature of two surfaces	[K]

Greek case letters

β	Thermal expansion coefficient	[1/K]
ε	Emissivity	[-]
λ	Thermal conductivity	[W/m·K]
λ_w	Wavelength	[nm]
μ	Dynamic viscosity	[Pa·s]
ρ	Density	[kg/m^3]
σ	Stefan-Boltzmann constants	[W/m^{2}K^4]
τ	Transmittance	[-]

Subscripts

ai	Air
b	Back
ca	Cavity
cv	Conduction and convection
ex	External
f	Front
i	Index for cavity or layer
int	Internal

la	Layer
m	Mean temperature in a cavity
n	Number of layers
rm	Radiation mean

1 Introduction

A large part of the heating loads in modern buildings, often using large glazed areas, occur due to solar radiation. In order to decrease the cooling demand of buildings, this load needs to be limited. By decreasing the g-value, which is the part of the solar load hitting a glazed surface that passes through, the heating load will decrease.

A convenient method to decrease g-values of windows and glazed facades is to use internal solar screens. These have no effect on the exterior architecture of the building and are protected from the weather outside, making for less wear and service demand. However, using internal shades means that most of the heat passing through the window still ends up inside the building and heating it. Compared to using external shades there is therefore a much lower reduction of the solar heat load.

One solution to decrease this problem is to extract the heated air between window and sunscreen by using the ventilation system in the building. This solution has for long been used in buildings with ventilated facades (Carlson, 2005), and prevents the heated air to affect the indoor climate. Thereby, the required power to cool the building is decreased.

In the tools available today for calculating the solar transmission through windows with solar screen, forced air flow of the open air cavity between window and screen cannot be taken into account. According to Tillberg (2014) there is a need to be able to verify the effects of ventilated air cavities to customers, to motivate implementing of the solution.

1.1 Aim

The purpose of this study is to investigate the possibilities to reduce the g-value of a window using an internal solar screen, by ventilating the air cavity between the screen and the window. The aim is to present the possible reduction as a reduction factor of the g-value compared to when the cavity is not ventilated. Further, the aim is to identify what parameters will affect the efficiency of using a ventilated cavity, find the most influential parameters and describe how the reduction factor is affected by these parameters.

The study will thereby answer the following questions.

- What reduction of g-value for a window with internal solar screen can be achieved by using exhaust ventilation to extract heated air around the window?
- What parameters is the system sensitive to?
- How do these parameters affect the efficiency of the system?

1.2 Methodology

To evaluate the efficiency of the system, a room with a window using the ventilated cavity between screen and window surface was modelled. The model computed the air and heat flow pattern of the system. Both the power heating the room and the heat exhausted through the ventilation were investigated in the model. To find the reduction of g-value the system has on the window, g-values for a ventilated cavity were compared to cases where the ventilation was shut off.

To solve for the air flow and heat fluxes, finite element method was used. The FEM-model was created in COMSOL Multiphysics, where both the CFD-module and the heat transfer module were used. This was necessary in order to simulate all the physical processes affecting the heat transfer.

In order to compute some of the optical data required as input to the model in COMSOL, the program WINDOW 7.2, calculating according to ISO 15099, was used as well.

The model was step by step verified against ISO standard 15099. This was to ensure the g-values calculated in the model were correct. To assess the quality of the CFD modulations, simplified cases with lots of empirical data will be simulated to make comparisons possible.

In order to see what parameters affect the efficiency of the system, different parameters such as location of exhaust device, screen properties and air flow, were changed and g-values calculated for each case.

Due to difficulty to accurately model a fine perforated surface in COMSOL, all screens were modelled as solid, with no air permeability. To see if the result of this study also is applicable on perforated screens, and in what way the result would be affected in that case, some perforated screens were modelled roughly by adding holes to the solid screens modelled.

1.3 Limitations

The windows modelled in COMSOL are modelled as 2D, with plane perpendicular to the window pane surfaces. As mentioned above, perforations of the screens are only roughly modelled. When making the comparison between a system with forced air flow in the cavity between window and solar screen and a system relying on natural convection no focus will be placed upon the economic or technical aspect of this solution. The conclusions of this thesis are strictly regarding the g-value of the system.

2 Theory

In this chapter, it is described how the different heat transfer mechanisms, conduction, convection and radiation affect the thermal performance of windows.

2.1 Conduction

Conduction is the mechanism for heat transfer within a material, due to vibrations of the molecules within the material (Hagentoft, 2001). In a window, this mainly takes place within the panes and in the frame. For a homogeneous and isotropic material the conductive heat transfer, $\boldsymbol{q}$, is a linear process described by equation (2.1). Assuming isotropic and homogeneous material is possible for common building materials, within small temperature differences.

$$\boldsymbol{q} = -\boldsymbol{k}\nabla\boldsymbol{T} \tag{2.1}$$

Where $\boldsymbol{k}$ [W/(m·K)] is a material property and $\boldsymbol{T}$ is the temperature vector.

For heat transfer in one direction the equation is simplified.

$$q = -k\frac{dT}{dx} \tag{2.2}$$

2.2 Convection

Convection is the heat transfer from one place to another due to movements of a fluid where the heat is carried by the heat capacity of the fluid (Hagentoft, 2001). The convection can be forced or natural. Forced convection occurs when the movement of the fluid is a result of applied force, for example from a fan or from wind. Natural convection occurs due to the variations of density within a fluid, caused by temperature and pressure differences. In a multiple pane window there is natural convection in the gas between the panes and both forced and natural convection on both in and outside of window.

2.3 Radiation

The dominant heat transfer mechanism in a window is radiation (REHVA, 2011). In difference to conduction and convection, heat transfer by radiation between surfaces can occur in vacuum, since the heat is transferred by electromagnetic waves (Hagentoft, 2001). The heat transfer between two surfaces of different temperature is dependent of the temperature difference, the surface emissivity and of the view factor between the surfaces. The view factor is dependent on the geometry and states the fraction of radiation leaving one surface that is intercepted by the other surface. The equation for calculating radiative heat flux from one surface to another, when $T_1 \approx T_2$, is stated below.

$$q = \frac{4\sigma\bar{T}^3(T_1 - T_2)}{\frac{1-\varepsilon_1}{\varepsilon_1} + \frac{1}{F_{1,2}} + \frac{1-\varepsilon_2}{\varepsilon_2}\frac{A_1}{A_2}} \tag{2.3}$$

Where.

$$\bar{T} = \frac{T_1 + T_2}{2} \tag{2.4}$$

In a window, heat transfer by radiation is more complex than the other two mechanisms because the energy reaching the glass surface will be divided in three parts (ES-SO, 2012). One part is reflected, one is absorbed and one is transmitted, see Figure 2.1 below.

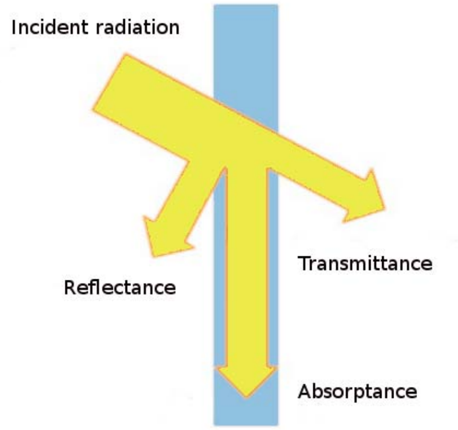

Figure 2.1 Division of energy from incident radiation.

The transmittance is different depending on the wavelength of the electromagnetic radiation, where some wavelengths can transmit trough the material and some cannot. For wavelengths below 300 nm and above 4000 nm, glass is completely opaque (Pilkington, 2012b). This mean most of the long wave radiation emitted from surfaces indoors or even panes themselves will not pass through a window.

Sunlight consists of a spectrum from 280 nm to 2500 nm where the visible spectrum of light is between 380 nm and 780 nm and contains 42 % of the energy from the sun irradiated (or emitted) to the ground (ES-SO, 2012). Therefore, the visible light transmitted through a window will affect the indoor climate, since it is impossible to let light in without letting heat in. Most of the remaining energy from the solar spectrum is in the short wave infrared spectrum (55 %). Since this part of the solar spectrum does not give any contribution to the level of light indoors, different coatings of the panes in the windows can be used in order to minimize the transmittance of it. Thereby the solar load is limited without decreasing the light transmittance. By limiting the transmittance for infrared radiation the U-value of the window can also be decreased.

2.4 g-value

The g-value is the factor of incoming energy from solar radiation that is transmitted into the room for a window or a window system. An example of a window system is a window in combination with a shading device. In Figure 2.2 an example of g-value is illustrated. If 80 % of incoming solar energy is transmitted through window, the g-value is 0.8.

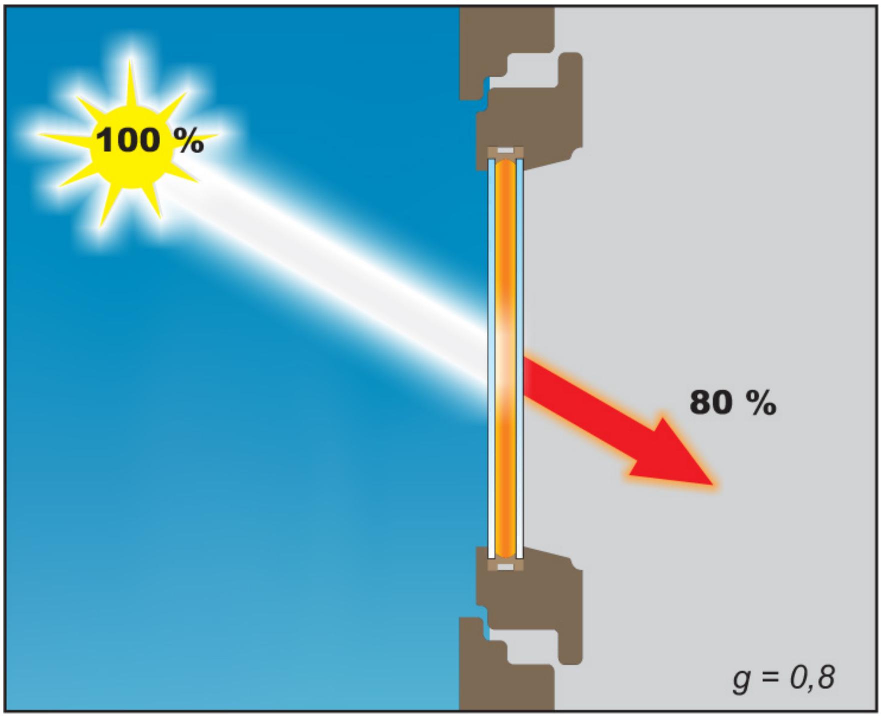

Figure 2.2 Explanation of the term g-value.

2.5 U-value

The U-value of a window or a window system is the heat transfer coefficient and is measured in the unit $W/(m^2 \cdot K)$. It is a measurement of how much heat is lost for a given temperature difference. U-value is not affected by the sun and is calculated without solar radiation.

3 Internal shades with exhaust ventilation

This chapter briefly explains how a window system with internal shades and exhaust ventilation works. Furthermore the different characteristics of windows and shades used in this thesis are described.

3.1 Basic concept

Figure 3.1 and Figure 3.2 illustrates heat gains from solar irradiation in a window with and without internal shading. Figure 3.2 also shows that convective heat between shade and window is a quite significant part of the total heat gain. When using internal shades, all the convective heat between shade and window ends up inside the building. By installing exhaust air devices above the solar shading part of this heat is transferred away, resulting in an even further decreased g-value.

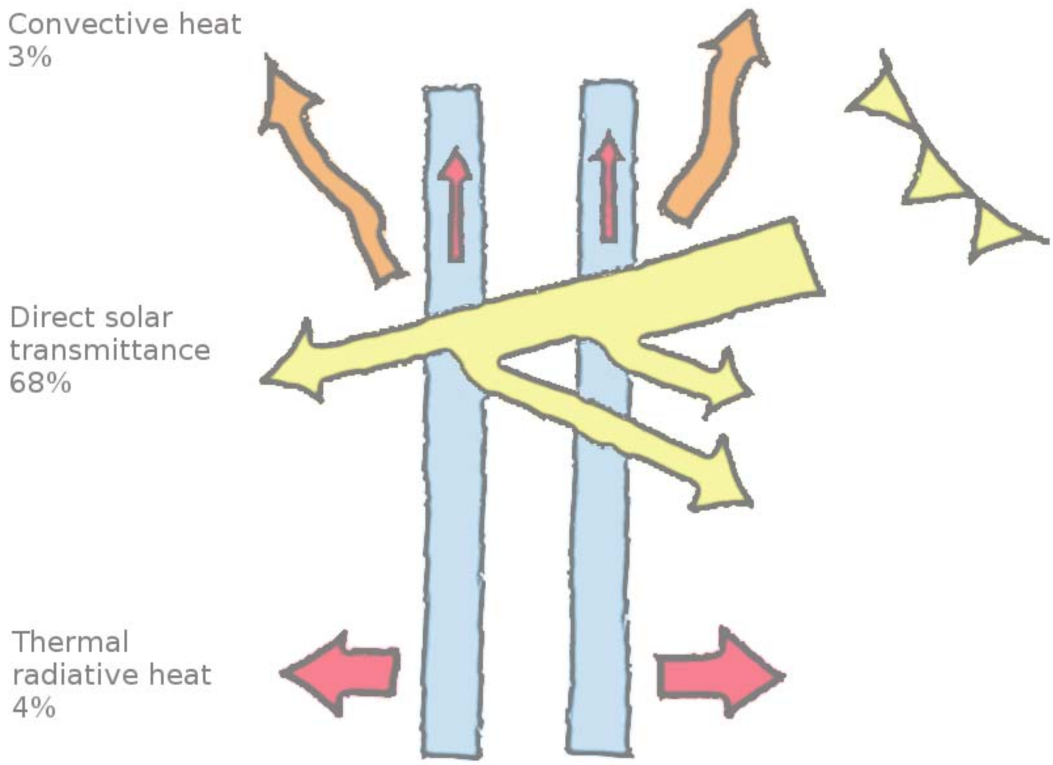

Figure 3.1 Heat gains from solar irradiation through a window without internal shades.

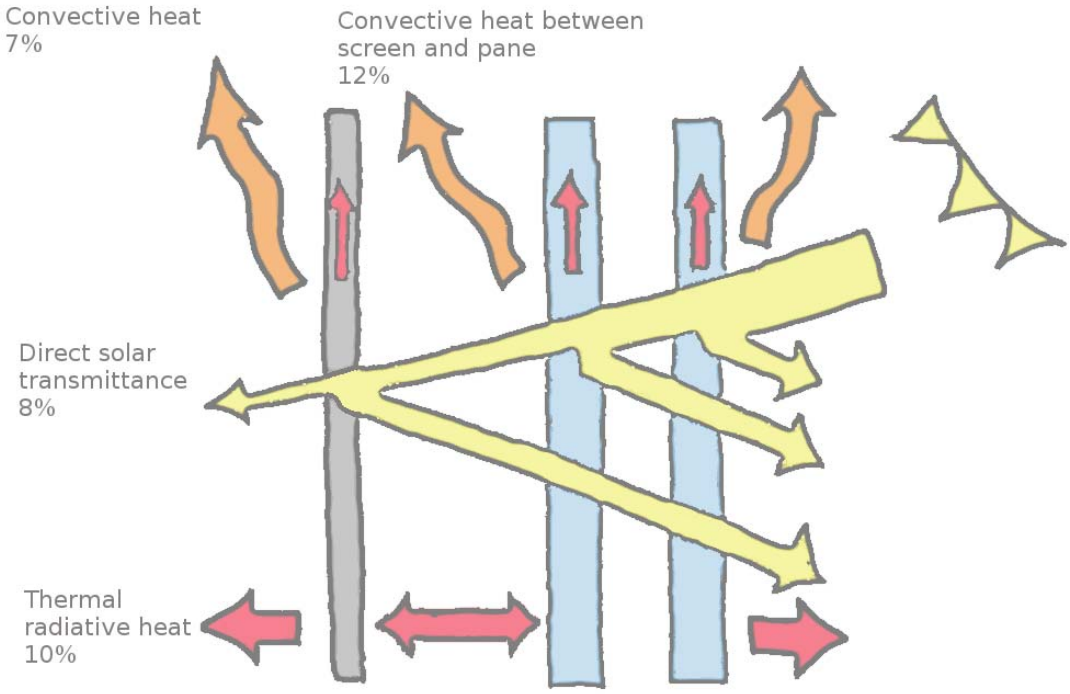

Figure 3.2 Heat gains from solar irradiation through a window with internal shades.

The values in figures above are taken from simulations made in software WIS (2014). Summarising the heat gains gives the g-value for the vision part of the window. This means windows in Figure *3.1* and Figure *3.2* has a g-value off 0.75 and 0.37 respectively. Theoretically, if all the convective heat between window and screen could be exhausted by ventilation, the g-value could be decreased from 0.37 to 0.25.

3.2 Characteristics of different windows

The characteristics of windows depend on many different factors, such as number of panes and the gas in between them and type of coatings. Manufacturers of windows often denote their products by three main properties. These being g-value, LT-value and U-value. LT-value is a factor of how much of the visible light hitting the fenestration product that is transmitted through. In this thesis the g-value is the most relevant.

Two ways to decrease U-value is to increase the numbers of panes or apply low-emissivity coatings. A side effect of this is less transmittance of both solar radiation and visible light. Simplified, the general pattern is that low U-value often means low g- and LT-value. A high LT value is of course preferable in close to all situations, since it is usually preferred to allow a lot of daylight indoors. However the selection of a window with a high or low g-value is more dependent on the situation. In this thesis three different kinds of windows with varying properties will be part of the investigation. Below the general properties of these different windows are described more in detail.

It should be stressed that U-, g- and LT-value could be used both to describe the entire window, meaning frame included, as well as just the part consisting of glass. In this thesis all window properties will always be regarding vision part of the window, meaning the effect of the frame is excluded.

3.2.1 Clear glass windows

Clear glass panes are basically glass that has not been refined to achieve different properties. Clear glass is the foundation to glass panes in both energy saving windows and solar control windows. When compared to energy saving windows and solar control windows with equal geometric properties, clear glass windows have high U-, g- and LT-values. In short this means that it lets in more visible light and more solar radiation but is less thermally insulated (Pilkington, 2012).

A clear glass window is exclusively made up out of clear glass panes.

3.2.2 Energy saving windows

To enhance the thermal insulation properties a low-emissive coating is applied to one side of the clear glass panes. This procedure can potentially reduce the emissivity from 0.84 for an uncoated glass surface to as little as 0.03 on that specific side. Since a dominant part of heat transport through a window consists of radiative heat transfer, the low-emissive coating roughly reduces the U-value by 50%. A side effect of the low-emissive layer is a slightly reduced g- and LT-value. These types of windows are suitable in for example passive houses where good thermal insulation and the possibility to use solar gains are desired (REHVA, 2011).

Energy saving windows generally consist of clear glass panes and panes with low-emissive coating.

3.2.3 Solar control windows

Solar control windows are used in buildings which have an excess of heat. While some of these windows also shuts out light, the most common concept and purpose behind solar control windows is to let visible light in and shut out solar heat gain. In other words, find a combination between a low g-value without decreasing LT-value too drastically. A typical area of use for solar control windows would be office buildings (REHVA, 2011) (Pilkington, 2012).

Solar control characteristics are obtained in many different ways. Just as the energy saving windows, solar control windows often consist of clear glass panes as well. It should be noted that solar control windows may also have low-emissive coatings, allowing them to be similarly insulated as energy saving windows. The main difference is that solar control windows have a drastically lower g-value and a slightly lower LT-value.

3.2.4 Comparison and final word about different windows

In Table 3.1 the three types of windows discussed in previous section are compared. They are actual existing products from Pilkington and for comparison they are of more or less identical geometric properties. All these windows consist of three panes 4-6 mm and 16 mm argon in the spaces between them.

Table 3.1 Properties of three different types of three-panes windows from Pilkington.

Type	Name	U-value	g-value	LT-value
Clear glass	Optifloat clear	1.7	0.66	0.74
Energy saving	K-glass	0.8	0.58	0.63
Solar control	Suncool 70/40	0.6	0.38	0.64

The three windows in Table 3.1 are not the windows used in the final simulations. They are however used in early verification processes.

3.3 Characteristics of shadings (screen fabrics)

Although there are many different kinds of shading systems, the solutions studied in this thesis consists of screen fabrics. This is a woven screen that comes in many different varieties. Although they exist in total opaque variants these screens are normally a bit transparent, allowing some vision towards the outside. The properties of interest in these simulations are:

- Emissivity on both sides
- Transmittance
- Openness factor

Note that emissivity and transmittance are blunt values when it comes to screen properties. In reality and in the model these values are wavelength-dependent, meaning they will be different regarding solar and long wave radiation. However when investigating the screens, this will be the significant way of describing how the screens differ from each other.

In technical data, from manufacturers, screen fabrics are commonly denoted with an openness factor. This data is not regarding which percent of surface area that is open but rather describes the amount of UV light transmitted through the screen (Hunter Douglas, 2006). An openness factor of 0 % would be similar to an opaque wall while 100 % would be the equivalent of a window pane.

In this thesis openness factor is interesting from the standpoint of air leakage through the screen. This means when discussing openness factor in future chapters, the term denotes the percent of the surface area that is perforated.

4 ISO 15099

ISO 15099 is a standard used when calculating thermal transmittance, solar energy transmittance and visible light transmittance, in other words denoted the U-, g- and LT-values, of window and shading devices.

The calculations in ISO 15099 are performed as one-dimensional (1D) heat transfer networks. Radiative heat transfer, as well as conductive heat transfer through solid layers is treated with first principle calculations. In non-ventilated cavities, e.g. the gas layer between panes, the heat transfer by convection and conduction is approximated by empirical models. ISO 15099 also has calculation models for ventilated cavities, which is the case between panes and the internal shading devices studied in this thesis. This part of the window system is however not modelled accordingly with ISO 15099's 1D calculations but rather as 2D heat transfer with CFD. Therefore this part of the standard is not explained further in this report.

4.1 Definitions

As mentioned, the ISO model consists of layers, i.e. glass panes or shading devices, and cavities in between those layers. Each layer and cavity has an index, *i*, which denotes the position of this layer or cavity. In addition each layer also has a front and a back side denoted as *f* and *b,* where the front side is the side towards the exterior (see Figure 4.1).

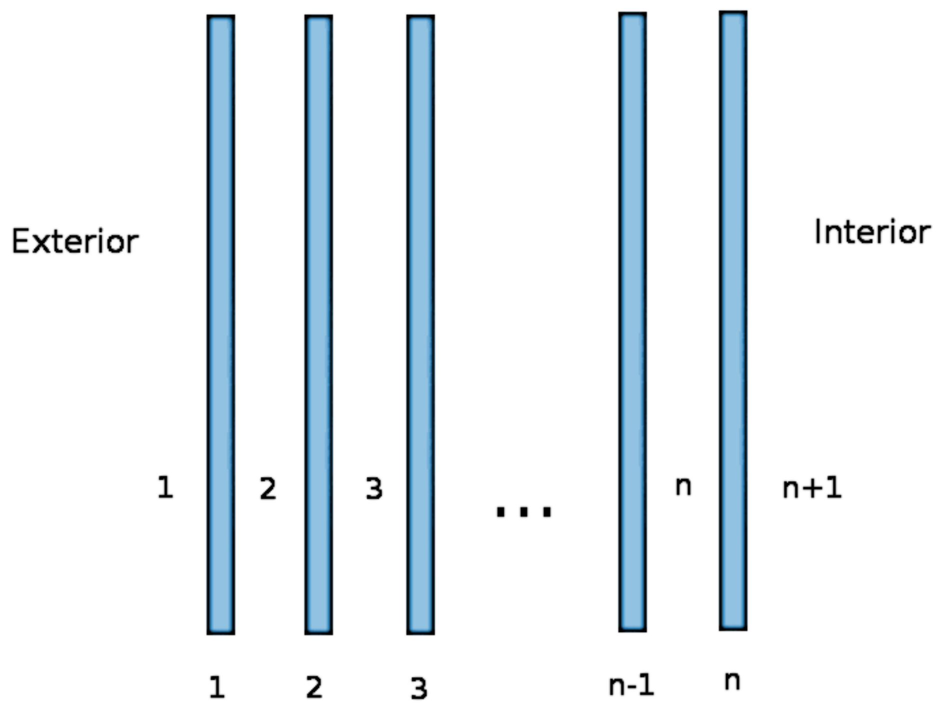

Figure 4.1 Index system of panes and layers in ISO 15099.

4.2 Heat transfer calculations

One basic assumption that ISO 15099 builds upon is that solar radiation and thermal radiation consists of different wavelengths. This allows the two to be handled separately. Transmitted and absorbed solar radiation in each layer is determined first and is assumed to be independent of other heat transfer processes (Collins & Wright, 2006). Figure 4.2 briefly illustrates the concept behind ISO 15099's calculations. Absorbed solar radiation is denoted as S_i, radiosity J_i and heat flux as q_i.

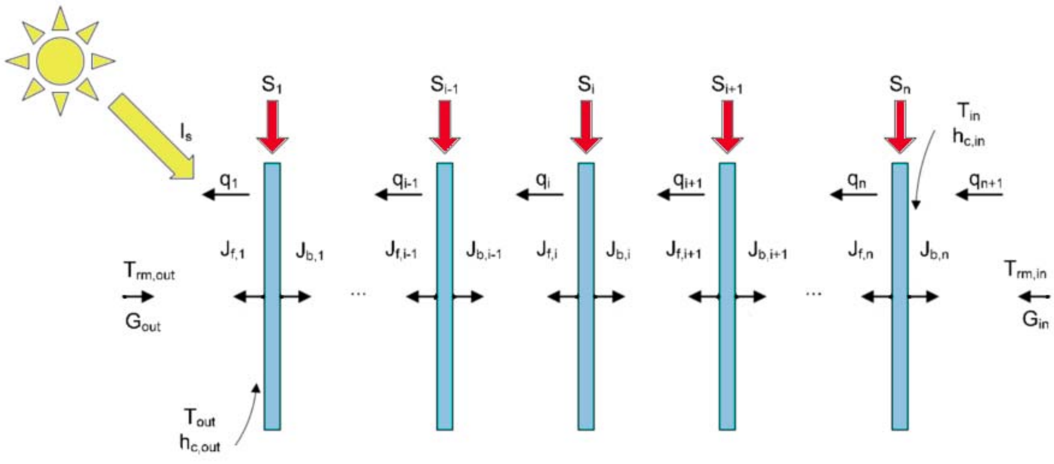

Figure 4.2 Heat transfer processes and their index in ISO 15099.

4.2.1 Absorbed and transmitted solar radiation

In a window without solar shading, for each layer, there are three necessary properties to determine absorption and transmittance of solar irradiation. The spectral reflectance for front and back side, $r_{f,i}(\lambda_w)$ and $r_{b,i}(\lambda_w)$, as well as the spectral transmittance, $\tau_i(\lambda_w)$.

When solar shading is added the complexity increases. This is because solar irradiation may change direction when being reflected or transmitted by the solar shading device. In short direct solar irradiation might be transmitted or reflected as direct or diffuse radiation. The effect of these characteristics is that each the required properties, for solar shading devices, are split into three. A direct-direct, a direct-diffuse and a diffuse-diffuse part (ISO 15099, 2003).

This step produces the values τ and S_i which is the transmittance through the entire window system and absorption in layer i respectively.

The procedure to determinate absorbed and transmitted solar irradiation is described fully in the international standard ISO 15099. This thesis however, will not cover this part any further.

4.2.2 Radiative heat transfer

ISO 15099 calculates radiative heat transfer by radiosity. In short radiosity is the sum of all radiation leaving a surface in the form of transmitted, reflected and emitted heat as seen in Figure 4.2.

The radiative heat flux qr through cavity i is therefore expressed as in equation (4.1):

$$qr_i = J_{f,i} - J_{b,i-1} \tag{4.1}$$

For full set of equations regarding radiosity see ISO 15099. Since the model in this thesis builds upon the assumption that no thermal radiation is transmitted through glass panes, a simplified thermal radiation model is used, see Section 5.3.3.2.

4.2.3 Conductive heat transfer in layers

The thermal network for the conductive heat flux in layers is split into two. This allows the absorbed effect from the solar irradiation S_i to be applied as a heat source (see Figure 4.3). Heat flux through half of a layer is the expressed as in equation (4.2):

$$qc_{la,f,i} = \frac{2\lambda_{la,i}}{d_{la,i}} \cdot (T_{m,i} - T_{f,i}) \tag{4.2}$$

Where $\lambda_{la,i}$ and $d_{la,i}$ is the thermal conductivity and thickness of the layer respectively.

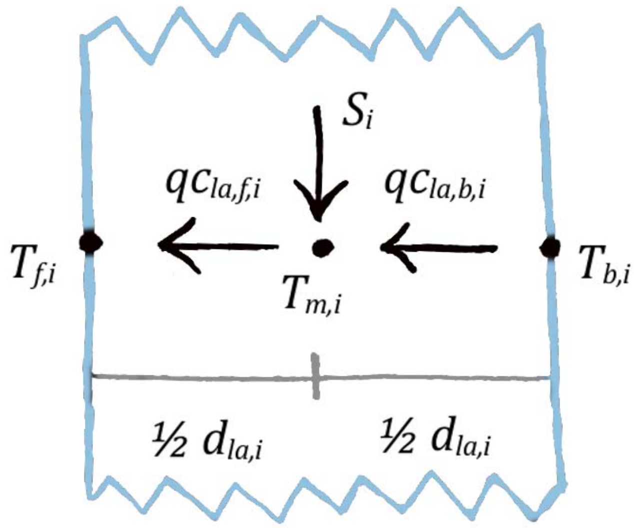

Figure 4.3 Conductive heat transfer in layers.

4.2.4 Conductive and convective heat transfer in closed cavities

The heat flux due to convection and conduction, in cavities, are treated with one combined heat transfer coefficient for each cavity. As previously mentioned this heat transfer in gas cavities is calculated by correlations based on experimental data from measurements in air layers. These heat transfer coefficients $h_{cv,i}$ are determined with the dimensionless Nusselt numbers, Nu_i.

$$h_{cv,i} = Nu_i \cdot \left(\frac{\lambda_{ca,i}}{d_{ca,i}}\right) \tag{4.3}$$

Where $\lambda_{ca,i}$ and $d_{ca,i}$ is the thermal conductivity of the fill gas and width of the cavity respectively.

The Nusselt number in turn is a function of the Rayleigh number, Ra, the cavity aspect ratio, $Ar_{ca,i}$, and the actual slope of the cavity. In this thesis only vertical cavities are considered.

Rayleigh number is expressed below, in equation (4.4). Note that subscripts *i* and *ca* are removed for convenience.

$$Ra = \frac{\rho^2 d^3 g \beta c_p \Delta T}{\mu} \tag{4.4}$$

The fill gas in layer *i* is treated as a perfect gas and the thermal expansion coefficient of the fill gas, β, is expressed as as in equation (4.5):

$$\beta = \frac{1}{T_{m,i}} \tag{4.5}$$

Where $T_{m,i}$ is the average temperature inside the cavity.

The values for c_p, μ and also λ are calculated using linear functions of temperature in the form $y = a + bT_m$. Coefficients a and b as well as molecular mass for different gasses are found in ISO 15099.

The density of the gas is calculated with the perfect gas law setting pressure to 101 300 Pa and temperature to 293 K.

$$\rho = \frac{101300 \cdot \widehat{M}}{\mathcal{R} \cdot 293} \tag{4.6}$$

The Nusselt number for vertical cavities is evaluated when the Rayleigh number is obtained in equation (4.7):

$$Nu_i = \left(Nu_{1,i}, Nu_{2,i}\right)_{max} \tag{4.7}$$

Where.

$$Nu_{1,i} = 0.0673838 \cdot Ra^{1/3} \qquad 5 \cdot 10^4 < Ra \tag{4.8}$$

$$Nu_{1,i} = 0.028154 \cdot Ra^{0.4134} \qquad 10^4 < Ra \leq 5 \cdot 10^4 \tag{4.9}$$

$$Nu_{1,i} = 1 + 1.7596678 \cdot 10^{-10} Ra^{2.2984755} \qquad Ra \leq 10^4 \tag{4.10}$$

$$Nu_{2,i} = 0.242 \cdot \left(\frac{Ra}{Ar_{g,i}}\right)^{0.272} \tag{4.11}$$

The aspect ratio of the cavity i is expressed as in equation (4.2):

$$Ar_{ca,i} = \frac{H_{ca}}{d_{ca,i}} \tag{4.12}$$

4.2.5 Energy balance

The energy balance for the system seen in Figure 4.2 is set up accordingly.

$$q_i = h_{cv,i} \cdot (T_{f,i} - T_{b,i-1}) + J_{f,i} - J_{b,i-1} \tag{4.13}$$

The solution is generated by using four different equations at each layer, i.

It is an iterative process. By approximating initial temperatures on each layer, calculate heat transfer coefficients and evaluate a new temperature distribution a new solution is found. This is repeated until convergence has reached an acceptable tolerance (Carli, Inc, 2006).

4.2.6 Interaction with environment and boundary conditions

Temperatures on the inside and the exterior, n number of layers, are assigned accordingly.

$$T_{f,n+1} = T_{ai,int} \tag{4.14}$$

$$T_{b,0} = T_{ai,ex} \tag{4.15}$$

Long-wave irradiance on internal and external surfaces.

$$J_{f,n+1} = \sigma T_{rm,int}^4 \tag{4.16}$$

$$J_{b,0} = \sigma T_{rm,ex}^4 \tag{4.17}$$

And the convective heat transfer coefficients.

$$h_{cv,n+1} = h_{cv,int} \tag{4.18}$$

$$h_{cv,1} = h_{cv,ex} \tag{4.19}$$

ISO 15099 uses the boundary conditions and input data seen in Table 4.1 below.

Table 4.1 Input data and boundary conditions (ISO 15099, 2003).

	Summer conditions	Winter conditions
$T_{ai,int}$	25 °C	20 °C
$T_{ai,ex}$	30 °C	0 °C
$h_{cv,int}$	2.5 W/m^{2}K	3.6 W/m^{2}K
$h_{cv,ex}$	8 W/m^{2}K	20 W/m^{2}K
$T_{rm,int}/T_{rm,ex}$	$T_{ai,int}/T_{ai,ex}$	
I_s	500 W/m^2	300 W/m^2

4.2.7 Calculation of g-value

g-value of the window system is then calculated as in equation (4.20):

$$g\text{–}value = \frac{q_{int}+\tau \cdot I_s - q_{int}(I_s=0)}{I_s} \tag{4.20}$$

Where τ is the transmittance of solar irradiation and q_{int} is the heat flux to the inner cavity, which is the room itself. $q_{int}(I_s = 0)$ is the heat flux through the window without solar irradiation, i.e. only due to temperature difference between the interior and the exterior.

5 Implementation of ISO 15099 in COMSOL Multiphysics

In this chapter the process of creating a COMSOL model for the vision part of a window with solar shading is described. The intention is that this model, as far as possible, shall resemble the ISO 15099 standard. In the first step the entire window system is analysed using more or less the 1D strategy explained in previous chapter. Even though the simulations in COMSOL are 2D, they will mostly perform equivalently to 1D heat transfer (see Section 5.3).

In a second step, the model is expanded by adding CFD modelling for the air in the cavity between glass pane and solar shading (see Section 5.4).

Verifications of models are performed both in the first and second step. Results from simulations are compared to calculations made in WINDOW 7.2, a software built upon the ISO 15099 model. Furthermore a mesh dependency analysis for the CFD domain is performed to additionally verify the model (see Section 5.5).

Three different window systems are modelled for the verifying process, see Appendix A. All these follow the geometry seen in Figure 5.1 and differ compared to each other only in the variation of panes. Note that cavity between inner pane and screen is considered closed, meaning a separate air volume shut off from the rest of the room. This is of course not a case existing in reality but rather to simplify the verification. See Appendix A for more information about pane and screen properties.

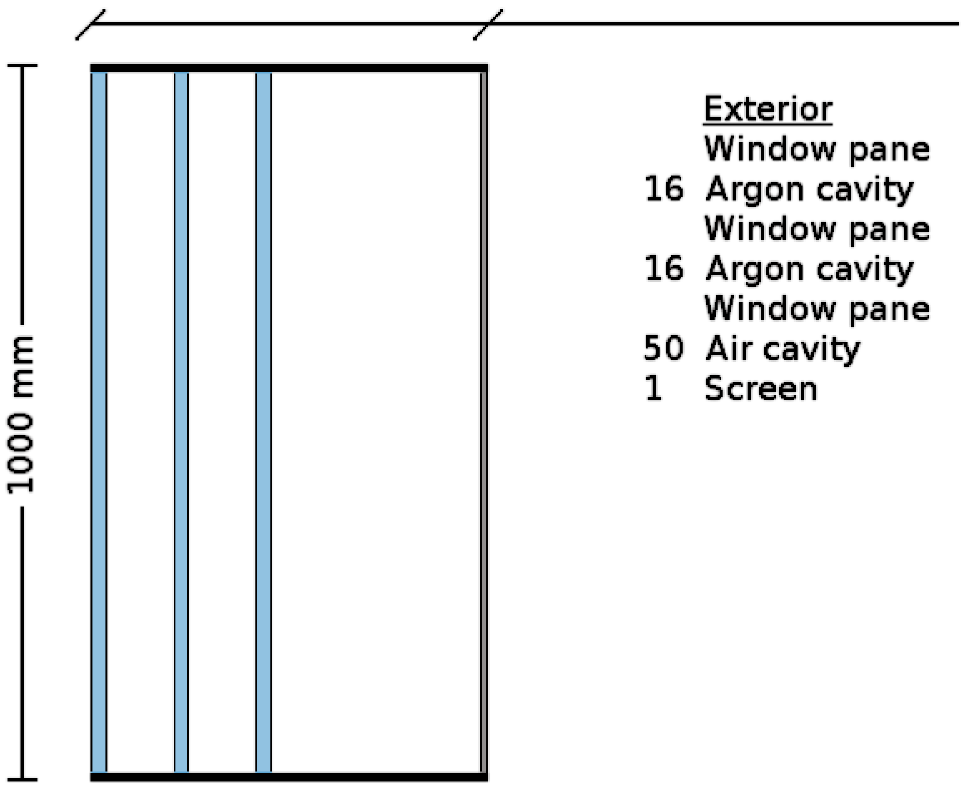

Figure 5.1 Geometry of window with screen modelled in COMSOL for verification.

5.1 COMSOL Multiphysics 4.3b

COMSOL Multiphysics 4.3b is the software and version used in these simulations. It is a finite element analysis solver and simulation software. COMSOL Multiphysics uses modules to simulate different physical processes. These modules can be combined in one interface to simulate coupled systems of physical phenomena (COMSOL, 2014a).

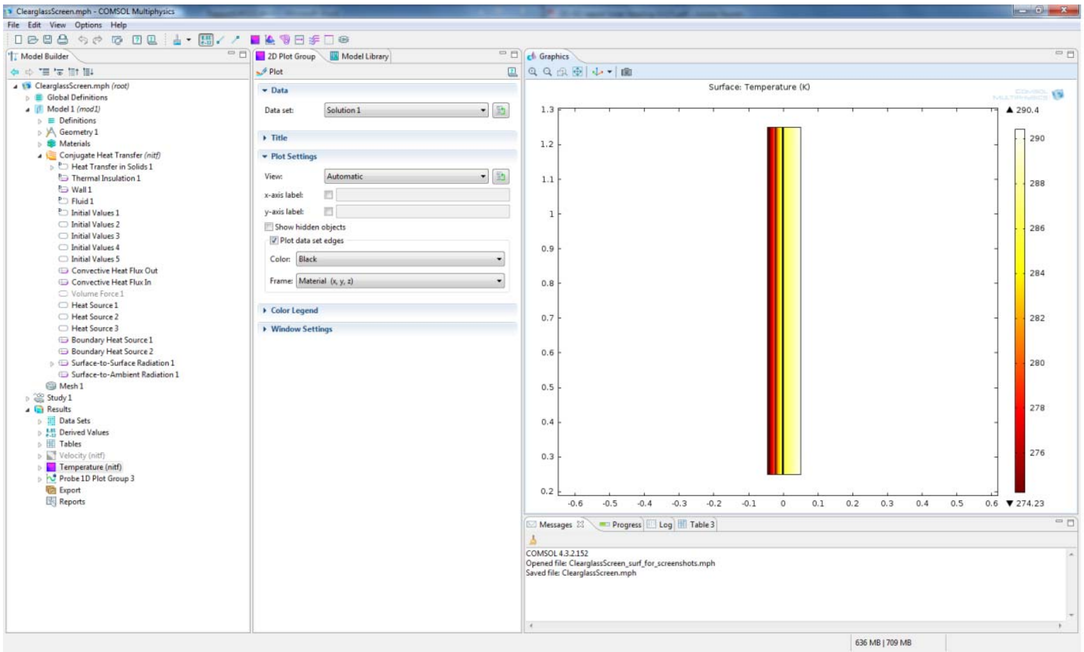

Figure 5.2 COMSOL Multiphysics 4.3b.

5.2 WINDOW 7.2

WINDOW 7.2 is a program developed at the Lawrence Berkeley National Laboratory, for calculating thermal and optical performance of windows and shading devices (Windows, 2014). The program is connected to a database where data of properties for many commercially available glass panes and solar shades are available.

WINDOW 7.2 is calculating according to the ISO15099 standard and is therefore used for verification of the results from COMSOL in this study. It is also used to calculate the absorption for window panes and shading layer and for calculating the transmittance of visible light for windows, both used as input to the models in COMSOL.

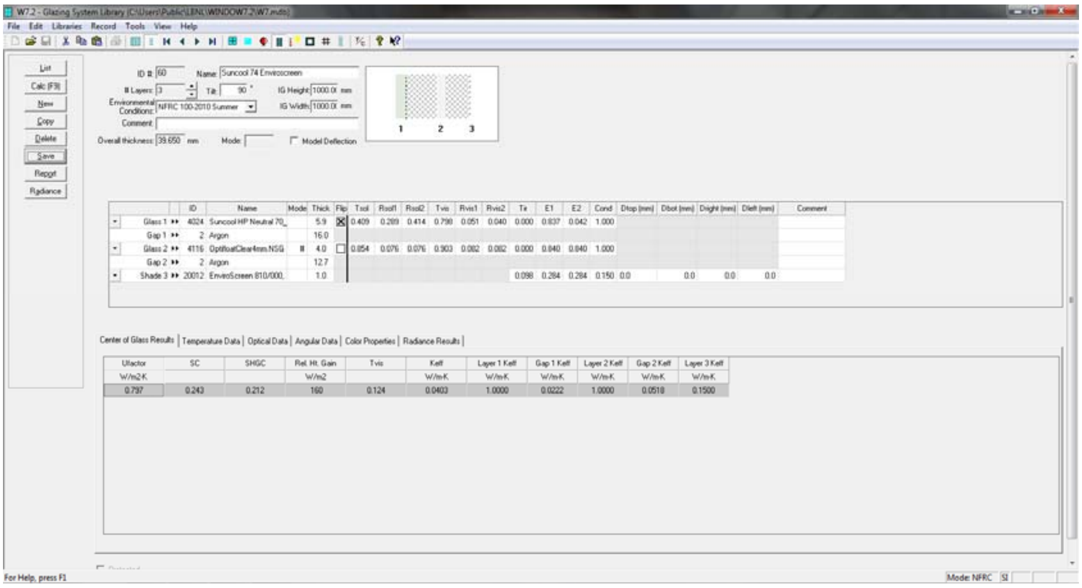

Figure 5.3 WINDOW 7.2.

5.3 Creating a model equivalent to ISO 15099 in COMSOL

The model uses the physics interface *Conjugate Heat Transfer*. This includes both the CFD- and the heat transfer–module and is suitable for simulating heat transfer in both solids and fluids. As mentioned before the model is set in the 2D-plane and it uses a stationary solver (steady-state).

This particular model is representing a three glass window with interior shading. The cavity separating the interior glass pane from the screen is considered to be a closed air volume.

5.3.1 Geometry and Domain Point Probes

The geometry consists of simple adjacent rectangles representing both the solid layers and the cavities. In the model the exterior is on the left hand side. Note that each layer is split up into two domains (see Figure 5.4). This will be explained further in Section 5.3.3.1.

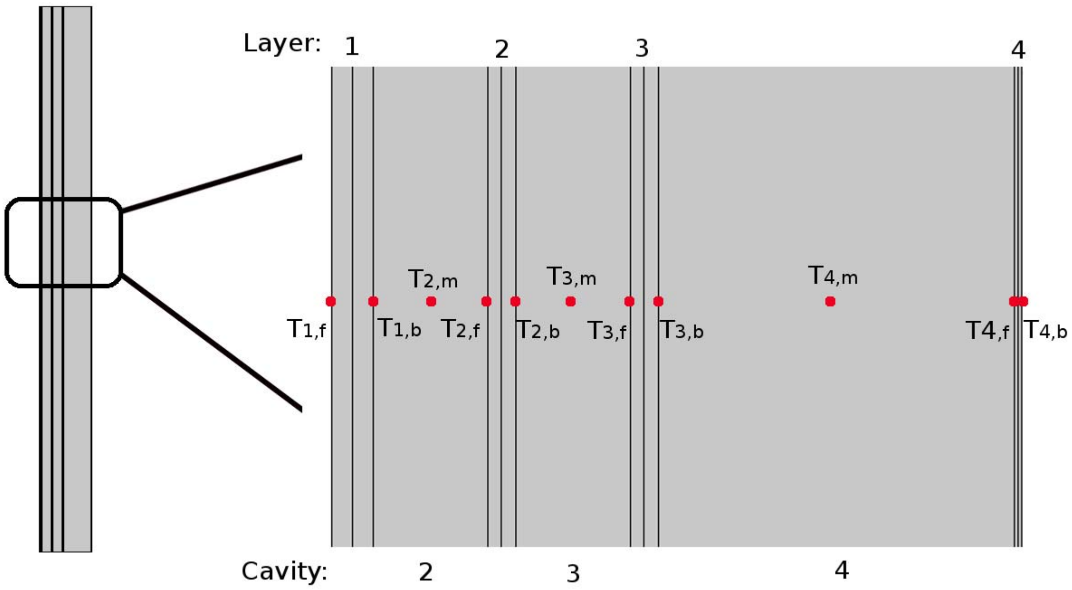

Figure 5.4 Geometry of entire model (left) and magnified view with probe placement (right).

Domain Point Probes are placed in the centre of the window from a vertical perspective. These probes measure the temperature between each iteration and are placed on the surface of each layer, as well as in the centre of each cavity. Probes are named according to which temperature they would correspond to in ISO 15099. Domains (cavities and layers in ISO 15099) are also named as their counterparts in the ISO standard.

5.3.2 Materials

The materials in the model are assigned in the manner that can be seen in Table 5.1.

Table 5.1 Material properties and where they are assigned in the model. Only underlined material properties has significance in a steady-state simulation.

Material in reality	Domain in model assigned to	Material properties
Glass	Layer 1, 2 & 3	<u>Thermal conductivity</u>: 1.0 [W/(m·K)] Density: 2595 [kg/m^3] Heat capacity at constant pr.: 750[J/(kg·K)]
	Layer 3 (back surface)	<u>Surface emissivity</u>: *Window dependent*
Argon	Cavity 2	<u>Thermal conductivity</u>: $\lambda_{r,2} + \lambda_{cv,2}$ [W/(m·K)][1] Density: 1 [kg/m^3] Heat capacity at constant pr.: 1000 [J/(kg·K)]
	Cavity 3	<u>Thermal conductivity</u>: $\lambda_{r,3} + \lambda_{cv,3}$ [W/(m·K)][1] Density: 1 [kg/m^3] Heat capacity at constant pr.: 1000 [J/(kg·K)]
Air	Cavity 4	<u>Thermal conductivity</u>: $\lambda_{cv,4}$ [W/(m·K)][1] Density: 1 [kg/m^3] Heat capacity at constant pr.: 1000 [J/(kg·K)]
Screen	Layer 4	<u>Thermal conductivity</u>: 1.0 [W/(m·K)] Density: 100 [kg/m^3] Heat capacity at constant pr.: 2000[J/(kg·K)] Dynamic viscosity: 16.7·10^{-6} [Pa·s] Ratio of specific heat: 1.4 [-]
	Layer 4 (front surface)	<u>Surface emissivity</u>: 0.706 [-]
	Layer 4 (back surface)	<u>Surface emissivity</u>: 0.592 [-]

[1]See Section 5.3.3.2 & Section 5.3.3.4 for details regarding $\lambda_{r,i}$ & $\lambda_{cv,i}$.

- All panes are assigned as one material
- Screen is assigned as one material
- Each cavity is assigned as a unique material
- Interior surface of pane 3, back and front surface of the screen are each assigned as a unique material

As previously mentioned, the heat transfer through cavities are dependent on temperature on adjacent surfaces and the mean temperature inside the cavity itself. This is the reason each cavity is assigned as a unique material.

It should also be said that even though this is a steady state case, COMSOL demands input for material properties not underlined in Table 5.1 above.

5.3.3 Conjugate heat transfer

As mentioned the physics interface *Conjugate Heat Transfer* is used to create this model. In the folowing subsections it is explained how, and by witch tools ISO 15099 is implemented into COMSOL.

5.3.3.1 Absorbed and transmitted solar radiation

The absorbed solar radiation in each layer, S_i, and the transmitted solar radiation, τ, is calculated using the software WINDOW 7.2. The absorbed solar radiation is then used as input for *boundary heat sources* placed in each layer in COMSOL. This is the reason layers are split into two domains. Using a *domain heat source*, as oppose to a *boundary heat source*, when implementing the absorbed solar radiation leads to numerical errors when integrating heat flux over surfaces. Something that is necessary in order to acquire the g- and U-value of the window system.

5.3.3.2 Radiative heat transfer

In cavity 2 and cavity 3 radiative heat transfer is treated by the tool *Heat Transfer in Solids*. This is done by setting the material property for thermal conductivity as a variable dependent on probe temperatures on adjacent surfaces. A new conductivity is thereby calculated, between each iteration, by the following equation:

$$\lambda_{r,i} = 4\varepsilon_{12,i}\sigma\bar{T}_i^3 d_{ca,i} \tag{5.1}$$

Where.

$$\bar{T}_i = \frac{T_{b,i-1} + T_{f,i}}{2} \tag{5.2}$$

$$\varepsilon_{12,i} = \frac{1}{\frac{1}{\varepsilon_{b,i-1}} + \frac{1}{\varepsilon_{f,i}} - 1} \tag{5.3}$$

Equation (5.1) builds upon two assumptions. First one being that $T_{b,i-1}$ and $T_{f,i}$ is of the same order. According to Hagentoft (2001) this is “a reasonable approximation in building physics applications”. Secondly the surfaces of the panes are considered to be two infinitely large parallel surfaces.

This way of evaluating radiative heat transfer as a thermal conductivity is a simplification to be able to simulate cavity 2 and cavity 3 as solid material. The cavity between inner glass pane and screen (cavity 4) is in step two simulated with CFD. To allow for easy addition of a CFD module the domain here is defined as *fluid*. For the same reason the radiative heat transfer is not simulated using thermal conductivity in material but instead by using the existing tool *Radiative Heat Transfer* in COMSOL. This is also the reason why interior pane and surfaces of the screen must be assigned a material. It is necessary to be able to use the built in tool for surface radiation.

A problem with using the built in tools for thermal radiation is that there will also exist a heat exchange with the ambient surroundings. This will occur in the bottom and top of the cavity where no screen exists. As mentioned the ISO 15099 is a 1D model and does not take this into account. To decrease the errors from this ambient heat transfer, the ambient radiation temperature is set to an average of the temperatures at the surface of the screen and the surface of the window. The values of these temperatures are from the probes located centrally in the vertical direction of the panes.

Another problem with the previously mentioned ways to simulate radiative heat flow is that they build upon the assumption that layers are opaque to long wave radiation. This is a fair assumption for glass panes but not for the screen (Collins & Wright, 2006). In Figure 5.5 below illustrates how the thermal network changes when one layer transmits long-wave radiation.

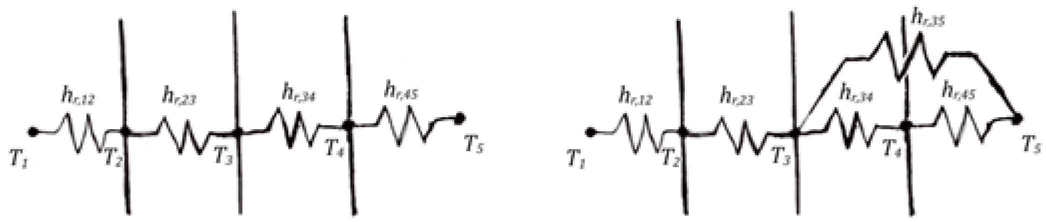

Figure 5.5 Thermal networks where all three layers are opaque to long wave radiation (left), and where one layer is transparent to long wave radiation (right).

This is solved by a domain heat source in model denoted as *qr_through*. The heat source is assigned to the right domain in the layer representing the inner glass pane and is calculated as follows:

$$qr_{through} = \tau_{screen,long-wave}\varepsilon_{b,n-1} \cdot \sigma \cdot (T_{f,n+1}^4 - T_{b,n-1}^4) \tag{5.4}$$

Where $\tau_{screen,long-wave}$ is a property specific to the screen in use.

5.3.3.3 Conductive heat transfer in layers

This is simply simulated with *heat transfer in solids* in COMSOL. A material property for thermal conductivity is assigned to the glass layers as well as the screen.

5.3.3.4 Conductive and convective heat transfer in closed cavities

In this first step all conductive and convective heat transfer, for all cavities, are treated as a material property of thermal conductivity. Just as for the radiative heat transfer in cavity 2 and cavity 3, conductive and convection is simulated as heat transfer in a solid. The only difference in cavity 4 is that this domain is a fluid, but with no force applied to it, it behaves like the solids.

The thermal conductivity due to convective and conductive heat transfer is evaluated equivalently to how the term $h_{cv,i}$ is calculated in ISO 15099 (see Section 4.2.4). Note that while $h_{cv,i}$ is a thermal conductance, the value inserted in material properties is thermal conductivity. $h_{cv,i}$ is simply multiplied by the thickness of the cavity to obtain this thermal conductivity, $\lambda_{cv,i}$.

5.3.3.5 Interaction with environment and boundary conditions

Convective heat flux for exterior and interior are set as the boundary conditions for temperature and thermal conductance due to convection seen in Section 4.2.6. For the interior boundary the radiative heat flux is treated with the built in tool *Surface-to-ambient radiation*. For the exterior convective heat flux, a radiative thermal conductance is instead added. This is calculated for each iteration, by the following equation:

$$h_{r,ex} = 4\varepsilon_{f,1}\sigma\bar{T}_{ex}^{3} \tag{5.5}$$

To obtain this equation it is once again assumed that $T_{f,1}$and $T_{b,0}$ is of the same order, see Section 5.3.3.2. It is also assumed that $\varepsilon_{b,0}$ is equal to 1. This is a simplification of the boundary condition in ISO 15099 seen in Section 4.2.6, equation (4.17).

5.3.4 Calculation of g-value

The g-value is finally evaluated by using the same equation as in Section 4.2.7. The value of q_{int} is given by integrating the heat flux over the screen border facing the interior. The solar transmittance τ_s is taken from simulations with software WINDOW 7.2.

To receive the value of $q_{int}(I_s = 0)$ the simulation is run with the heat sources simulating solar absorption turned off.

5.4 Addition of CFD in cavity between window and screen

As mentioned, the model described in Section 5.3 is verified by comparing it to calculations by WINDOW 7.2. When agreement between COMSOL model and calculations in WINDOW 7.2 are sufficient, CFD is added in the cavity between window and screen.

5.4.1 Changes of model

The material representing the cavity between window and screen (cavity 4) is changed to the, in COMSOL, built in material *Air*. Properties for this material are defined so that dynamic viscosity, thermal conductivity and density are dependent on temperature of the material itself. Density is also dependent on pressure.

Furthermore a volume force is set in the domain representing the aforementioned cavity to simulate the effect of gravity and difference in density upon the air volume. A *pressure point constraint* is also added since it is a closed air volume and no pressure level can be defined in boundary conditions.

The domain representing the screen is changed from the real value of 1 mm to 3 mm. To compensate for decreased conductance the conductivity is increased by a factor of three. A thin domain, in this case, is meshed by COMSOL with an unnecessary amount of elements and at the same time increases the growth rate of the mesh. Increasing the domain size in the manner explained solves this problem.

5.4.2 Simulating turbulent flow

The dimensionless Reynolds number is often used as an indicator if the flow is in the laminar or turbulent region. A set boundary for at which Reynolds number flow is turbulent does not exist. Adding the fact that the geometry studied in this thesis is irregular it could be very difficult, even with low calculated Reynolds numbers, to exclude the possibility of local turbulence in certain parts of the model. In his report Davidson (2011) claims that "almost all fluid flow which we encounter in daily life is turbulent". Davidson continuous to give air movements in a room as an example of one of these turbulent flows. In this thesis the flow is assumed to be turbulent at least in some regions of the geometry modelled.

When dealing with fluid simulations two approaches can be used. Either using what is called direct numerical simulations (DNS), or by using a turbulence model. In DNS, the Navier-Stokes equation is solved directly to acquire one realization of the flow-field. In a turbulence model on the other hand equations are solved for mean quantities to determine a statistical solution to the problem. Theoretically DNS would give the most accurate solution, but due to computer power limitations this is only practically applicable for flows in the lower Reynolds number regions. The computational time required increases proportionally with the Reynolds number by the power of three (Pope 2000). In short, adding a turbulence model to simulate turbulent flow is not done in order to increase accuracy but rather to get a solution at all. In this thesis Reynolds Avarage Navier-Stokes (RANS) turbulence models are used. More about which specific RANS-model is described in the next section.

5.4.3 Selection of turbulence model

In turbulent flow there are fluctuations in the flow pattern where the speed and direction of the fluid varies continuously differing from laminar flow, where the flow is continuous (Turbulent Flow, 2014). To compute fluid flow in COMSOL a turbulence model, prescribing the turbulent oscillations in the fluid, needs to be selected.

At a surface where a fluid meets a solid, non-moving material the velocity of the fluid is always zero (Grundmann, 2009). This surface is denoted as the wall. With increased distance from the wall, the velocity increases. This mean there is a zone close to the wall, where the flow is affected by the solid surface and hence prescribed differently from the unaffected zones. To solve the flow in this region numerically is complex and requires a lot of computational power compared to a free zone, where the flow is not affected by surrounding walls (COMSOL, 2014b). Hence, approximations are often used to describe the flow close to walls. These approximations, called wall functions, differ between the different turbulence models available in COMSOL. There are also some turbulence models not using wall functions and instead solving the flow field at all points in the fluid close to the wall.

These models require a very fine mesh close to wall boundaries, to solve the flow in the boundary layer adequately. Models without wall functions take longer time to compute but do increase the accuracy of the results, especially when dealing with heat flux between wall and fluid (Davidson, 2011).

The turbulence model selected is called the *Low Reynolds number k-ε model*, which is an extension of the *standard k-ε model*. This is selected because it describes turbulent flow without using wall functions approximating the flow close to wall regions (COMSOL, 2014b).

However, this initial model is made for verification of how well CFD can be combined with a similar 1D model as seen in ISO 15099. When simulations where made with the *Low Reynolds number k-ε model* it was hard to make the model converge, at least without switching to a transient solver and thereby severely increase computational time. To be able to perform more simulations, and thereby more verification, the *standard k-ε model* was chosen. This is only for the initial verifications and not for the final model. In the final model the *Low Reynolds number k-ε model* is used. This can be done since the model uses a transient solver, meaning it is not a steady state model.

5.5 Verification of models implemented in COMSOL

In order to prove the models created in COMSOL give reliable results they need to be verified. This is done in several steps. First a mesh dependence study is performed. Secondly the simulations both with and without CFD are compared to values from WINDOWS 7.2. As mentioned before, WINDOW 7.2 builds upon the ISO 15099 standard.

5.5.1 Mesh dependence

To make sure the model in COMSOL gives a correct result without numerical errors, the derived values must be independent of the mesh. Hence, a parametric study is performed where the same simulation is run for different mesh qualities. Heat fluxes and different surface temperatures are measured and compared for calculations using the different meshes.

The mesh is generated using the automatic size function in COMSOL, where the fineness of the mesh is set on a scale in a number of steps from *extremely fine* to *extremely coarse*. The settings affect a number of parameters regarding the mesh, such as the maximum element size of the mesh and the number and thickness of the boundary layers, which are located in fluids regions close to solid surfaces. Boundary layers are the regions close to the wall where the mesh consists of thin rectangular elements, see Figure 5.6 below.

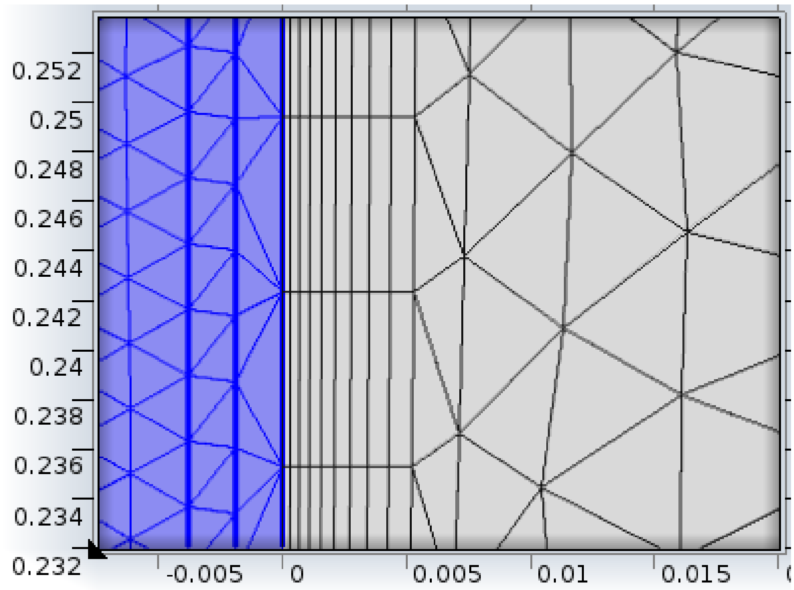

Figure 5.6 Illustration of boundary layer where fluid meets solid (darker area).

It is observed that in all materials, including the gas filled cavities modelled according to ISO 15099, the resolution of the mesh do not affect the result of the calculations when changing from *extremely fine* to *extra coarse*. For fluid dynamics and the heat transfer in fluids the results show more dependence of the mesh quality. To find a mesh fine enough, without causing unnecessarily long calculation time, a number of mesh sizes are tested.

The average growth rate and the number of boundary layers for elements in the fluid layer are shown in Table 5.2 below. These parameters also affect the total number of elements and nodes and therefore affect the computational time. Number of elements is also shown in table below. In the table the heat flux trough the screen is shown. This result is selected for comparison between different meshes since the screen is adjacent to the fluid and the flux trough it is likely to be highly affected by the fluid.

Table 5.2 Mesh study for the closed cavity model.

Mesh Size	Number of elements	Number of boundary layers	Average growth rate fluid	Heat flux trough screen [W/m^2]	Temp. inside screen [°C]
Coarse	26 484	5	1.368	172.49	53.17
Normal	34 680	5	1.315	172.49	53.17
Fine	68 546	5	1.295	172.49	53.17
Finer	157 166	5	1.272	172.67	52.67
Extra Fine	201 480	5	1.223	172.67	52.67

It is observed that the heat flux trough the screen is nearly independent of the mesh size in the investigated span. However if velocity and temperature plots are studied, some deviations occur at certain points for the coarse mesh. The solution is considered independent of mesh size for mesh quality of normal or better. The changes in the result for finer meshes are very small and not worth the increased computational time they cause. Hence, the mesh setting Normal is chosen.

5.5.2 Verifying using ISO

To verify the results of the numerical model created in COMSOL a comparison is performed between the numerical model and calculations according to ISO 15099. The calculations according to the ISO-standard are performed in the software WINDOW 7.2, which calculates window properties according to the ISO 15099. To make sure the numerical model always performs according to the ISO-standard, results are compared for different types of windows and both for a summer and a winter case.

The reason different types of windows and variations of climate are used is because climate and type of window affect the temperatures of the panes in the window and the properties of the cavities vary with these temperatures. Both the g-value and U-value are compared, as well as surface temperatures on a specific window pane.

The verification is performed in two steps. First using the model with all cavities modelled according to the ISO 15099, see Section 5.3. Secondly the cavity between interior window pane and screen is modelled as a real fluid in COMSOL. This two-step method is chosen in order to identify the possible errors strictly bound to the usage of the CFD-module of COMSOL and what errors occur when implementing a model equivalent to ISO 15099 in COMSOL.

The COMSOL model without CFD shows almost perfect agreement with the ISO-standard for the three different types of windows and the temperature spans investigated. When CFD is introduced for the cavity between screen and window, some deviations occur. These mainly occur for the U-value of the window. The g-value follows very well, with deviations less than 1%. The results of the convergence study are shown in the tables below. The tables show the temperature on the interior side of the middle pane, in Figure 5.4 denoted as $T_{2,b}$.

Table 5.3 U-values, g-values and temperature $T_{2,b}$ calculated in WINDOW 7.2.

Window & Seasonn	g-value	U-value	$T_{2,b}$
Optifloat clear (Summer)	0.584	1.351	51.4
Optifloat clear (Winter)	0.571	1.302	19.7
K-glass (Summer)	0.598	0.760	60.3
K-glass (Winter)	0.582	0.729	26.3
Suncool (Summer)	0.394	0.559	56.6
Suncool (Winter)	0.379	0.554	23.8

Table 5.4 U-values, g-values and temperature $T_{2,b}$ calculated in COMSOL, without CFD, including deviation from Table 5.3.

Window & Season	g-value	Deviation	U-value	Deviation	$T_{2,b}$	Deviation
Optifloat (S)	0.583	-0.2 %	1.334	-0.5 %	51.5	0.2%
Optifloat (W)	0.570	-0.2 %	1.296	-0.5 %	19.6	0.5%
K-glass (S)	0.597	-0.2 %	0.754	-0.8 %	60.4	0.2%
K-glass (W)	0.581	-0.2 %	0.726	-0.4 %	26.3	0%
Suncool (S)	0.395	0.3 %	0.565	1.1 %	56.7	0.2%
Suncool (W)	0.380	0.3%	0.560	-1.1 %	23.9	0.4%

Table 5.5 U-values, g-values and temperature $T_{2,b}$ calculated in COMSOL, using CFD in the cavity including deviation from Table 5.3.

Window & Season	g-value	Deviation	U-value	Deviation	$T_{2,b}$	Deviation
Optifloat (S)	0.583	-0.2 %	1.334	-1.3 %	51.5	0.2%
Optifloat (W)	0.569	-0.4 %	1.268	-2.6 %	19.6	-0.5%
K-glass (S)	0.593	-0.8 %	0.733	-3.6 %	61.2	1.5%
K-glass (W)	0.577	-0.9 %	0.702	-3.7 %	26.6	1.1%
Suncool (S)	0.394	0.0 %	0.561	0.4 %	57.2	1.1%
Suncool (W)	0.378	-0.3%	0.554	-0.2 %	23.8	1.3%

From the calculations it is concluded that the implementation of ISO 15099 into COMSOL has worked. The small deviations are from numerical errors, and can probably be decreased even further if the tolerances are decreased and mesh made extremely fine. However, the price to pay for this is longer computational time, and the result above is considered accurate enough.

When CFD is introduced to the cavity between screen and window the accuracy is still considered good. The deviations are believed to mainly occur due to the turbulence model and 2D effects. By introduction of CFD, the model is no longer 1D since the air flow in the cavity varies in both x- and y-direction. Compared to when all cavities are modelled according to ISO 15099, a temperature gradient is prevailing in y-direction. This will cause changes to the u- and g-values. The main changes are believed to come from the heat transfer at the top and bottom of the cavity, where the air flow changes direction. For a 1D model this change of direction is not modelled.

6 Combining ISO Model with Test Room in COMSOL Multiphysics

In this chapter the procedure of combining the window built according to ISO15099 with an air volume simulating an indoor room is described.

6.1 The test room

To find the performance of a system combining internal solar screen with exhaust air ventilation a test room is modelled. For this room the main criterion is that it must be modelled in a way that makes it possible to measure what part of the solar load can be evacuated through the ventilation and what part will enter the room. Further the conditions in the room must not affect the performance of the window. Hence the air movements not resulting from the solar load heating the air in and around the cavity must be small and the temperature in the room must be constant for varying solar loads. The model and its components are illustrated in Figure 6.1 below.

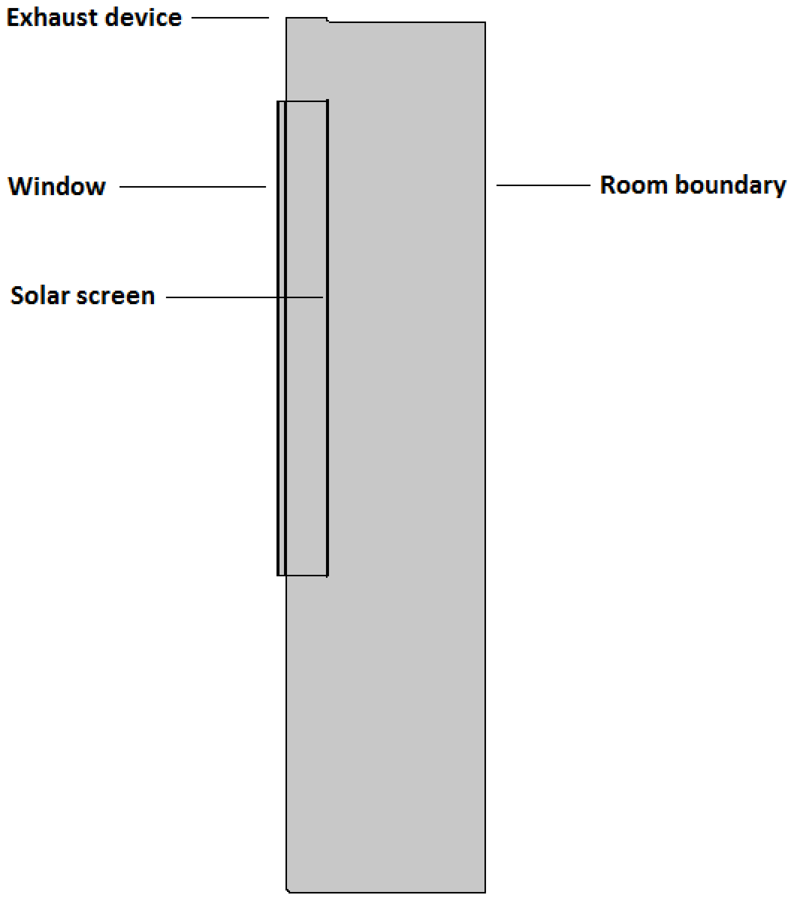

Figure 6.1 Geometry for room model in COMSOL.

The test room is a box of 0.5 m depth from the window, with a boundary representing a large room (see section 6.2) at the side opposite the window. This boundary is denoted the *room boundary*. The net heat flux through this boundary is considered be the heat load on the room.

6.2 Boundary conditions

The boundary conditions are set up according to Figure 6.2 below and further explained in this chapter.

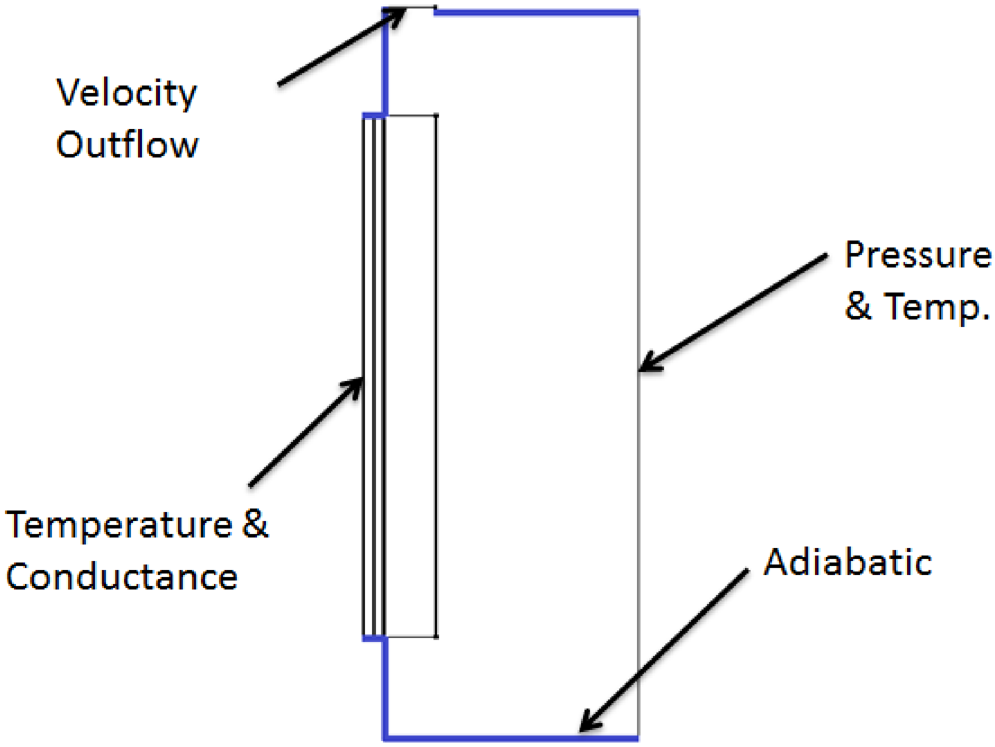

Figure 6.2 Boundary conditions in the COMSOL model.

The test room is modelled as an insulated room in order to limit the parameters affecting the solutions. The floor, ceiling and exterior walls are therefore adiabatic.

As for the model without the test room, the outside of the window is modelled according to ISO 15099 (see section 5.3.3.5).

The ventilation is simply modelled as a velocity outlet, i.e. a line where the velocity of the fluid is set.

To create a model representing a large room, the room boundary needs to be open for fluid flow and heat flux in all directions. This is achieved by setting an outlet boundary condition, where the static pressure is set and the viscous stress is zero. No viscous stress means there is no shear stress between the liquid and the boundary. This way of setting up the boundary will not cause any acceleration of the flow at the boundary and hence not disturb the velocity pattern caused by the solar radiation. The pressure over this boundary is the static pressure, only depending on the gravity and density of the air. This gives a pressure over the boundary according to the equation below.

$$p = \rho g(h_{room} - y) \tag{6.1}$$

Where h_{room} is the height of the test room.

The pressure is the deviation from the reference pressure, set to 1 atm. Since the testing room only is a few meters tall at maximum, the total pressure in the room will be close to 1atm.

At the room boundary the temperature is set to the indoor temperature, at 25°C. This is to enable conductive heat flux into the room and to make sure air entering the room holds correct temperature. On the outside of the window the temperature is set to 25°C, differing from the 30°C used when calculating g-values according to ISO15099. The 25°C outdoor temperature is selected to limit simulation time, since putting indoor and outdoor temperatures the same gives $q_{int}(I_s = 0) = 0$, see section 4.2.7, and the g-value can be calculated with only one simulation.

6.3 Calculation g-value in room model

As mentioned in Section 6.1, the net heat flux through the *room boundary* is considered to be the heat load on the room. This load is measured by integrating the heat flux over the mentioned boundary. However, this is not the entire heat load entering the room. Other fluxes occur, but cannot be measured by this integration. The ambient radiation from the screen, the infrared-transmittance through the screen and the direct transmitted radiation all contribute to the heat load on the room. All fluxes are illustrated in the figure below.

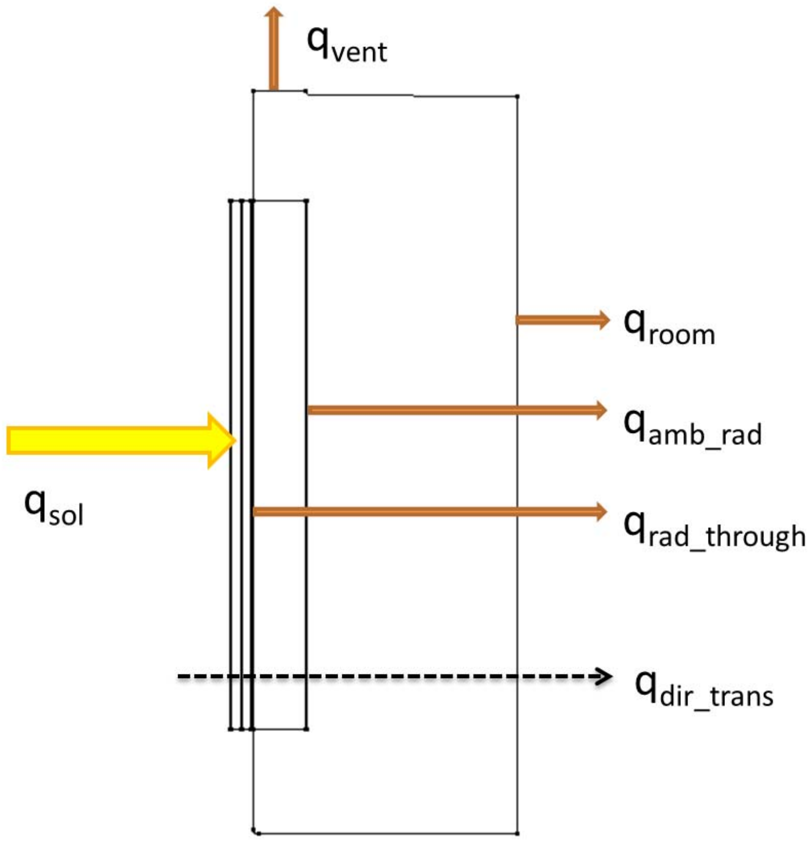

Figure 6.3 Heat fluxes in the 'room model'.

The direct transmission is kept outside of the COMSOL model completely, see Section 4.2.1. By adding all fluxes reaching the room together and dividing by the total applied solar radiation, the g-value is found.

$$g\text{-}value = \frac{q_{room} + q_{amb\_rad} + q_{rad\_through} + q_{dir\_trans}}{q_{sol}} \tag{6.2}$$

The ambient radiation from the screen to the room is found by integrating radiative heat flux on the inside of the screen, and the heat flux on the outside of the window is found by integrating the het flux in x-direction on the outside of the exterior pane

The heat evacuated trough the ventilation is found by applying the following integration on the outlet.

$$q_{vent} = \int v\rho c_p(T - T_{in})\,dx \tag{6.3}$$

This will only measure the part of the heat due to the solar load that is evacuated trough the ventilation, since it measures power due to an air temperature above the indoor temperature.

6.4 Time dependent solver

To find a solution that converges with the stationary solver in COMSOL proved to be difficult. The solution to the problem is to run a transient study, instead of a stationary as in the models without the room, and running it until the solution is constant from one time step to the next and all heat fluxes are in equilibrium. By running all calculations for a simulated time of four hours, this is achieved. The selection of time is done by studying how temperatures and heat fluxes develop over time in the model. Simulating longer times than four hours is found to not have any effect on the results.

In order for the solution to converge at all, some modifications need to be done for the initial time steps, to avoid steep gradients that cause convergence issues. By setting the outlet velocity and the heat sources in the panes and screen to zero at the initial time step (zero seconds) and linearly ramp them up to their supposed values over the first minute, the gradients are manageable for the solver, and the solution converges.

6.5 Verification of room model

In this section the mesh dependence of the room model is studied and the performance of the model is investigated by comparing g-values calculated in COMSOL to ones calculated according to ISO15099.

6.5.1 Mesh dependence

As for the simple model, with a closed cavity, the model for the entire room needs to be independent of the mesh. The study is performed in the same way as before, see section 5.5.1, but due to difficulties to measure the flux through the screen when it is adjacent to a fluid domain on both sides, the flux through the open boundary is measured and compared instead. Results are shown in Table 6.1 below.

Table 6.1 Mesh study for the room model.

Mesh Size	Number of elements	Number of boundary layers	Average growth rate fluid	Heat flux to room [W]
Coarse	27 290	9	1.401	63.5
Normal	38 575	9	1.375	63.7
Fine	69 526	9	1.304	63.4
Finer	95 700	9	1.243	63.5

The result show very low dependency of mesh quality in the investigated span. As before, the quality of velocity and temperature plots is poor for the coarse mesh. The *Normal mesh* quality is selected, since it gives accurate results, good plot quality and do not cause long computational time, compared to the finer meshes.

6.5.2 Comparison with WINDOW 7.2

In WINDOW 7.2 it is possible to calculate g-values for windows with an open cavity between window and internal solar screen, i.e. where air flow will occur between the cavity and the room. This g-value should agree with g-values derived from calculations in COMSOL, when the exhaust ventilation is turned off and the flow in the cavity is not restricted. For six different combinations of window and screen, g-values are compared. The results, shown in Table 6.2 show good agreement which proves the model performs accurate when a room is modelled together with the window. Note that screens used in WINDOW 7.2 have been altered so the openness factor is 0. This means that, just as in the COMSOL model, no air can penetrate the screen.

Table 6.2 Comparison of g-values from WINDOW 7.2 and COMSOL.

Window & screen	g-value WINDOW 7.2	g-value COMSOL	Deviation
Optifloat clear Conventional screen	0.693	0.685	-1.2 %
K-glass Conventional screen	0.540	0.552	2.2 %
Suncool Conventional screen	0.378	0.379	0.3 %
Optifloat clear Low emissivity screen	0.429	0.425	-1.0 %
K-glass Low emissivity screen	0.407	0.405	-0.5 %
Suncool Low emissivity screen	0.234	0.240	2.5 %

6.5.3 Global power equilibrium

To validate the model when the ventilation is turned on would require comparison to measured values for a real case. This is not done in this study. To verify the model, the global power equilibrium, meaning the heat absorbed in panes and screen equals power leaving the model, is studied. If the model contains numerical errors or errors due to the turbulence model, they would probably lead to errors in the heat transfer, changing the heat balance and generating a solution not in equilibrium. If the solution reaches power equilibrium, it is assumed to be correct.

When the g-values are calculated, for each case, the power equilibrium is also calculated. The difference is at most 0.6 %, of the total absorbed power. Hence, the model is considered to work accurately and perform like a real case would.

7 Parametric Study

In the study, seven different parameters of the model were investigated. These parameters, and the different values of them, are listed in Table 7.1 below.

Table 7.1 Initial parametric study.

Parameter	Values
Exhaust velocity	• 0 m/s • 0.18 m/s • 0.36 m/s
Exhaust location[1]	• At ceiling • Connected • Connected with opening • Above inner ceiling
Window type[2]	• 2-glass Clear glass • 3-glass Energy saving • 2-glass Solar control
Screen emissivity[2]	• 0.815 (Conventional) • 0.284 (Enviroscreen)
Window height	• 1.2 m • 2.5 m • 8 m
Cavity width	• 5 cm • 10 cm • 20 cm
Solar radiation	• 300 W/m^2 • 500 W/m^2 • 800 W/m^2

[1]See Section 7.2 for more details
[2]See Appendix A for more details

When the combinations of the different parameters were investigated a standard case was set up according to Table 7.2. The standard parameters were selected in order to generate a case with common geometry and without any precautions taken to limit the solar load. Hence, the Clear glass in combination with a conventional screen was selected, since this combination has the highest g-value of the possible combinations. In future sections, when investigating different combinations, only the parameters that differ from the standard case in Table 7.2 will be displayed.

Table 7.2 The standard case.

Exhaust velocity	0.18m/s
Exhaust location	At ceiling
Window type	2-glass Clear glass
Screen emissivity	0.815 (Conventional)
Window height	2.5 m
Cavity width	10 cm
Solar radiation	500 W/m^2

7.1 Reasoning behind exhaust velocity

The term *exhaust velocity* denotes the velocity of the outlet in the COMSOL model. It is just the velocity that corresponds to a certain exhaust flow in reality, when COMSOL model has a certain outlet size.

The exhaust velocities tested are selected to give reasonable flow for a realistic, 3D, case. A velocity of 0.18 m/s when the outlet is 100 mm wide is considered for the standard, most reasonable, exhaust velocity for the 2D COMSOL model. A doubled exhaust velocity of 0.36 m/s is also tested for comparison. This is however considered to be a little too high for most cases. The conditions, for a realistic case, these two exhaust velocities would correspond to can be seen in Table 7.3.

Table 7.3 Conditions for the real case that corresponds to the exhaust velocities chosen in COMSOL model.

	0.18 m/s (Standard)	0.36 m/s (Doubled)
Width of window	1.35 m	1.35 m
Duct diameter	100 mm	125 mm
Flow velocity in duct	3 m/s	4 m/s

These exhaust velocities corresponds to a real case when each window has an exhaust flow of about 25 l/s for the standard velocity, and 50 l/s for the doubled velocity.

In a realistic case the reasonable amount of exhaust air for each window could also be dependent on the buildings total exhaust air demand. It is not desirable that the windows exhaust flow is larger than the total exhaust air demand the buildings. This would result in a situation where it costs energy to save energy, due to an unnecessarily high ventilation rate. An in-depth analysis regarding this is outside the scope of this thesis. It can however be said that, for a building with measurements

50x30 m and 80 % of the horizontal space covered with windows, the standard exhaust velocity would mean a total exhaust flow of 1.5 l/s per square meter. This is 0.2 l/s/m^2 more than recommended flow for office buildings in Sweden, according to Sveby (2013). For the same outlet size the exhaust velocity and the exhaust flow has linear correlation. Therefore, when future text refers to correlation between exhaust flow and reduction of g-value, the same correlations are true for exhaust velocity.

7.2 Reasoning behind exhaust location

The different exhaust locations modelled in COMSOL are illustrated in Figure 7.1 below. The exhaust location *at ceiling* is to simulate the case when exhaust outlet is not at all in connection with solar screen. The *connected* exhaust location is created to represent a case when the exhaust device is connected to the top of the cavity. Exhaust location *connected with opening* is similarly modelled as the previously mentioned but with a 3 cm opening, simulating a not perfectly sealed connection. It is assumed that in reality, it will be difficult to perfectly seal these connections.

One way of creating a connection between the exhaust outlet and solar screen is to not mount the inner ceiling all the way out to the windows. The top of the screen is placed above the inner ceiling and the ventilation outlet is mounted vertically (Tillberg 2014). This design is the exhaust location denoted *above inner ceiling* and is visualised in Figure 7.1 below.

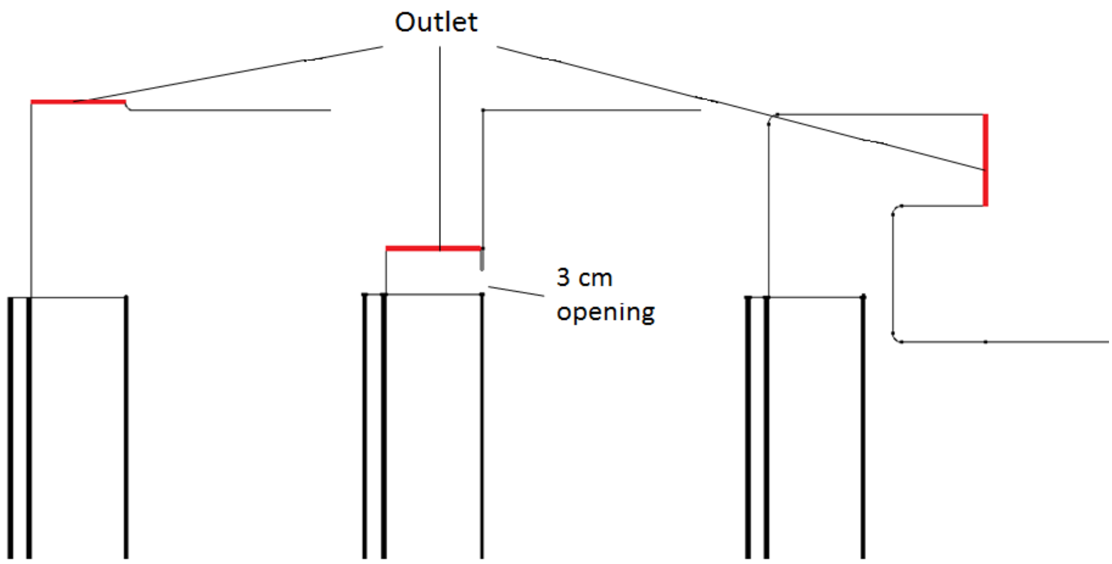

Figure 7.1 Model geometry in COMSOL for 'at ceiling' (left), 'connected with opening' (middle) and 'above inner ceiling' (right). Geometry for 'connected' case is identical to the middle image, except for the opening.

7.3 Reduction of g-value

The main result sought after in this parameter study is the possible reduction of g-value for a window with internal solar screen by adding ventilation. This is expressed in percent and is thereby calculated with the following expression.

$$possible\ reduction = \frac{(g-value\ with\ vent.) - (g-value\ open\ cavity\ without\ vent.)}{(g-value\ open\ cavity\ without\ vent.)}$$

In order to investigate the effects implementation of a ventilated cavity has on a window with internal solar screen, g-values needs to be simulated in models with and without ventilation applied. However, for some exhaust locations, the g-value of the window will be changed even when the outlet is shut off, since some outlet locations effect the natural flow in the cavity between window and screen. For example an outlet connected to the cavity will completely stop the flow through the cavity when turned off. Hence, the reduction of the g-value is always compared to a case where the cavity is open.

Every correlation described here is regarding a non-permeable screen. The influence of a permeable screen is investigated and described in Chapter 8.

7.4 Different exhaust locations with different exhaust velocities

Four different exhaust locations where investigated in combination with the three different exhaust velocities. This means a total of twelve different combinations where investigated in this section. In Figure 7.2 the results of these simulations are shown.

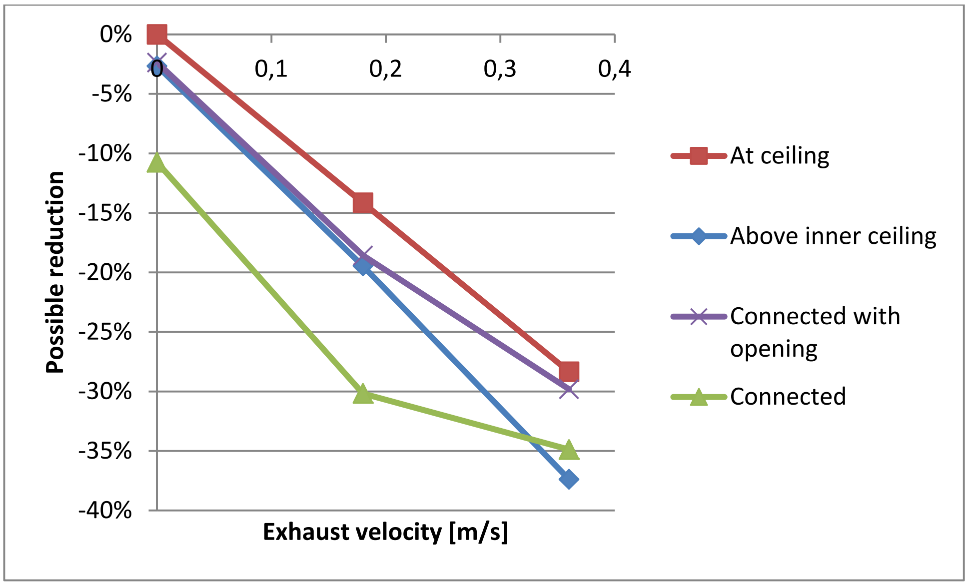

Figure 7.2 Reduction of g-value for different exhaust locations in combinations with different exhaust velocities.

It appears that the way different exhaust locations perform under different exhaust velocities could be considered as a spectrum, where one end would be the fully connected exhaust location and the other side would be the exhaust located at the ceiling. In between these two extremes, the *above inner ceiling* and *connected with opening* would be found.

Within the range of exhaust velocities investigated, the exhaust located at ceiling shows a linear correlation between the exhaust flow and the reduction of g-value. For the connected case however the larger changes in reduction of g-value occurs in the lower exhaust flow regions. As mentioned in Section 7.3 the reduction for zero flow is equal to zero for the open *at ceiling* case while the *connected* case has a significant reduction already. This happens because the perfectly connected case basically creates a closed cavity as opposed to an open and thereby decreases the g-value.

In Figure 7.2 it can be seen that the possible reduction for *above inner ceiling* is larger than for the *connected* case once exhaust flow is high enough. This is not surprising and both the *at ceiling* and the *connected with opening* exhaust locations would have a higher possible reduction as the exhaust flow increases. This is because the *connected* case only has the potential to extract the convective heat inside the cavity, while the three other cases also extracts the convective heat from the inside of the screen. As previously mentioned this is when assuming that the screen is non-permeable.

For convenience, the term convection factor is introduced. This denotes the ratio of the g-value that consists of convective heat. The convection factor is also the highest potential reduction of g-value possible since this would occur when all the convective heat is extracted through the exhaust device. For these particular window/screen combinations this is around 55%.

When studying the graphs in Figure 7.2 there seems to be a linear correlation, between the exhaust flow and the reduction of g-value, for the *at ceiling* and *above inner ceiling* exhaust locations. Note that this linearity might not continue outside the investigated range of flow. It is actually more likely that the derivate for these two curves also gradually will approach zero as exhaust flow increases.

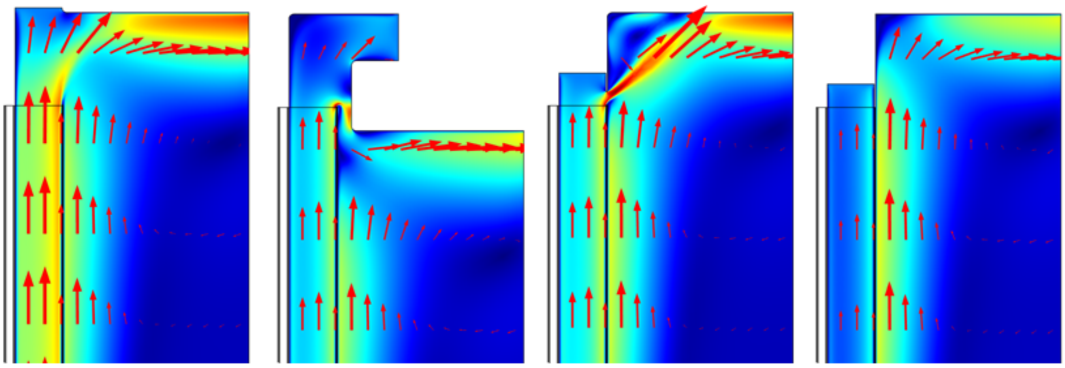

Figure 7.3 Velocity plots for 'at ceiling', 'above inner ceiling', 'connected with opening' and 'connected' exhaust locations. Arrows show direction of flow. Colours and size of arrows show velocity of flow. Larger arrows or red colour means higher velocity.

In Figure 7.3 above it can clearly be seen that some leakage of air exists in the exhaust locations *above inner ceiling* and *connected with opening*. This makes the reduction less effective compared to the *connected* case.

7.5 Window types with different exhaust velocities

The three window types where investigated in combination with the three different exhaust velocities. These simulations where performed for both the *at ceiling* and the *connected* exhaust location cases. The results of these two cases are shown in Figure 7.4 and Figure 7.5 below.

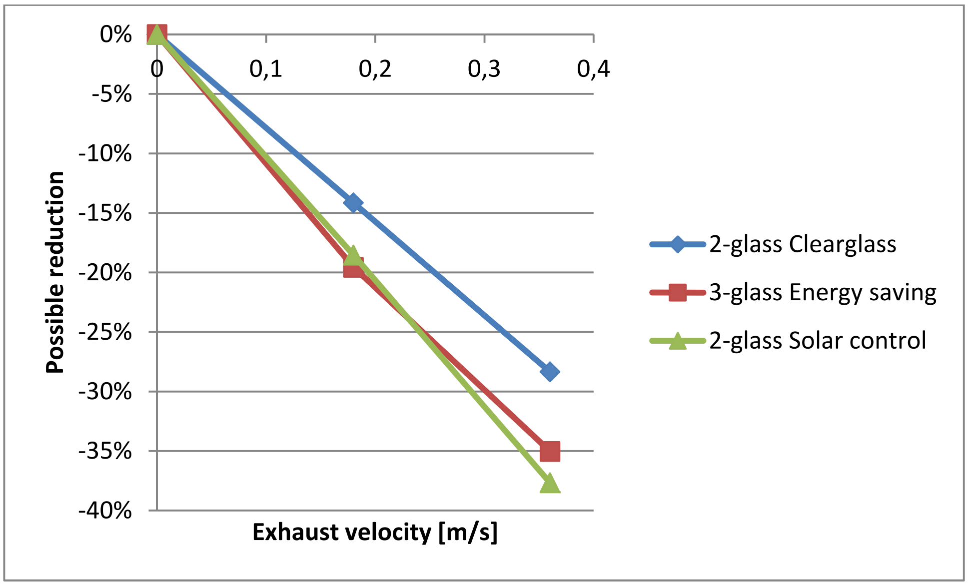

Figure 7.4 Reduction of g-value for different window types in combinations with different exhaust velocities. Values are for the 'at ceiling' exhaust location case.

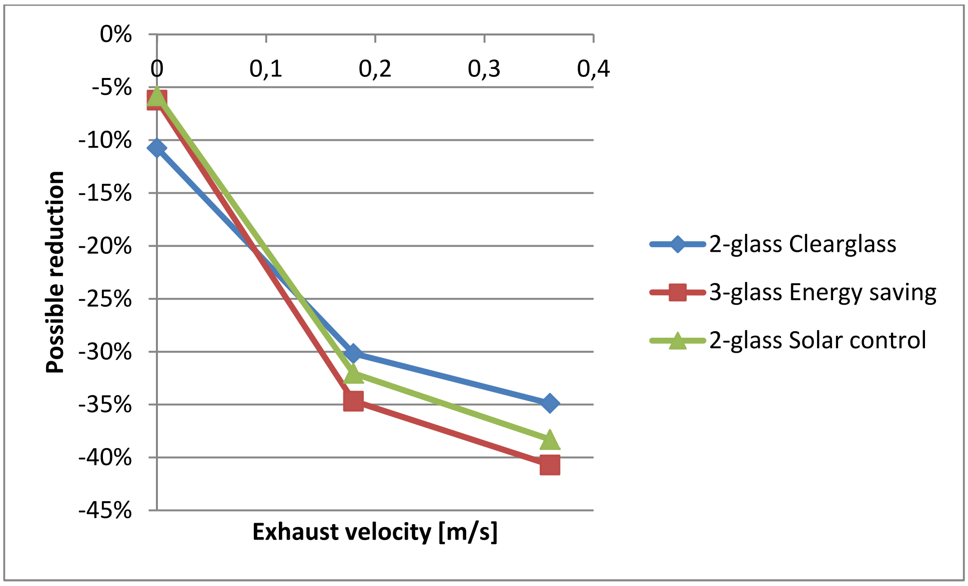

Figure 7.5 Reduction of g-value for different window types in combinations with different exhaust velocities. Values are for the 'connected' exhaust location case.

It seems correlations between window type and exhaust flow depends on multiple factors. One of these factors is likely to be the convection factor. In theory, the larger the exhaust flow the closer to the maximum reduction value the system will come. As can be seen in Table 7.4, the convection factors are very similar for these three windows. If this was the only deciding factor the graphs for Clear glass and the Solar control window would be almost identical in Figure 7.4. A plausible additional factor could be the g-value itself. In Table 7.4 it is shown that the g-value for the Solar control window with screen is considerably lower compared to that of the Clear glass window. This means that the 55% convective heat from a Clear glass is also considerably larger when measuring in absolute convective heat. Therefore a larger flow might be necessary to extract an equally large part of the convective flow for the Clear glass window.

In general the trend seems to be that a window system with a high convection factor in combination with a low total g-value gives the best overall reduction when applying ventilation. This when accounting for reasonable exhaust flows (See Section 7.1).

However when looking in Figure 7.5 the opposite seems to be true for low exhaust flows when cavity is connected. When the exhaust flow is close to zero, the Clear glass shows a larger possible reduction, compared to the other windows. An explanation to this could be that the larger g-value of the Clear glass, compared to the other window systems, means a larger natural flow velocity in cavity between window and screen. When the exhaust device is connected to the screen, the velocity in the exhaust device controls the flow between the cavity and the screen completely. This means the flow in a Clear glass window is more prevented compared to its natural state then a window system with lower g-value. This would increase the possible reduction for the g-value as seen in Figure 7.5.

Table 7.4 Simulated g-value and convection factor for different types of windows with conventional screen when no exhaust ventilation is present.

Window type	g-value	Convection factor [%]
2-glass Clear glass	0.69	55.5
3-glass Energy saving	0.55	59.1
2-glass Solar controll	0.38	55.2

7.6 Screen emissivity

Two screens with different surface emissivity where investigated. These simulations where performed for both the *at ceiling* and the *connected* exhaust location. The results of these two cases are shown in Table 7.5 below. The difference in reduction between 0.815 and 0.284 emissivity is also shown.

Table 7.5 Reduction of g-value for different emissivity's for both 'at ceiling' and 'connected' exhaust location, compared to a system with the same properties but with ventilation turned off.

	At ceiling / Connected		
	2-glass Clear glass	3-glass Energy saving	2-glass Solar control
Reduction of g-value with screen emissivity 0.815	-14% / -30%	-20% / -35%	-19% / -32%
Reduction of g-value with screen emissivity 0.284	-17% / -33%	-27% / -43%	-23% / -28%
Difference in reduction between the two screens.	3% / 3%	7% / 8%	4% / -4%

In most cases the possible reduction is increased when changing to a low emissivity screen. Note that this is the reduction compared to a system with the same properties but with ventilation turned off. This means changing the screen does not only lower the g-value, but in most cases also increases the possible reduction of the g-value compared to when using a higher emissivity screen. This holds true for the exhaust flow range investigated. When comparing data from Table 7.4 and Table 7.6 it is shown that the g-value decreases whilst the convection factor increases when changing to low emissive screen. From Section 7.5 it seems as both these changes, in general, increase the possible reduction for the g-value. With a few exceptions, the results in this section further strengthen these trends.

In Table 7.5 it can be seen that the possible reduction, for the Solar control window with *connected* exhaust, is actually less for the low emissive screen. It seems that, as also seen in section 7.5, exceptions to this trend exist when exhaust outlet is connected to the screen. In this case however, the exceptions in data appear when exhaust velocity is 0.18 m/s. A possible explanation could be the relatively low g-value of 0.24 of the Solar control window with low emissive screen. A lower g-value means a lower natural flow in the cavity between screen and window. Because of the low g-value of this window system, the natural flow might be lower than the exhaust velocity of 0.18 m/s. If this is the case the exhaust outlet will raise the flow in the cavity and thereby increase the g-value, by increasing the convective heat transfer.

Table 7.6 Simulated g-value and convection factor for different types of windows with low emissivity screen when no exhaust ventilation is present.

Window type	g-value	Convection factor [%]
2-glass Clear glass	0.42	55.3
3-glass Energy saving	0.40	64.4
2-glass Solar controll	0.24	56.6

7.7 Window height with different exhaust velocities

Three window heights where investigated in combination with the three different exhaust velocities. These simulations where performed for both the *at ceiling* and the *connected* exhaust location cases. The results of these two cases are shown in Figure 7.6 and Figure 7.7 below.

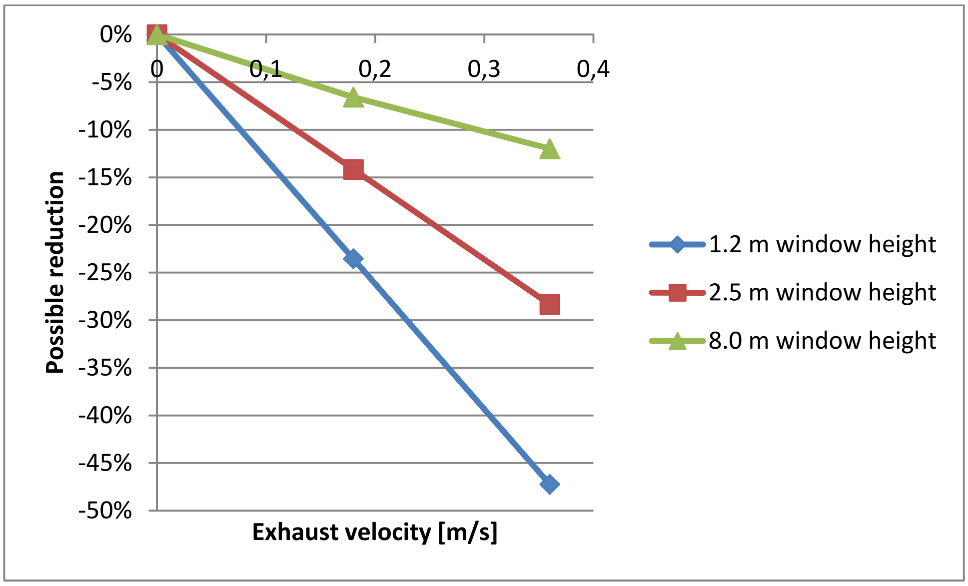

Figure 7.6 Reduction of g-value for different window heights in combinations with different equivalent exhaust velocities. Values are for the 'at ceiling' exhaust location.

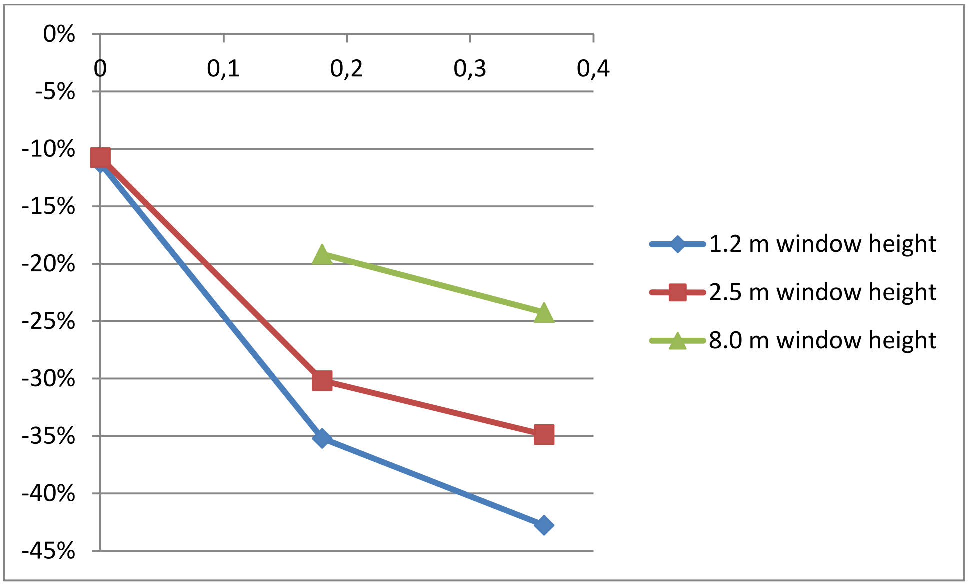

Figure 7.7 Reduction of g-value for different window heights in combinations with different equivalent exhaust velocities. Values are for the 'connected' exhaust location.

The value for the 8.0 m window with 0 exhaust velocity did not converge. This value will however most likely be around the -10 % mark.

For both the *at ceiling* and the *connected* exhaust location the reduction is better for lower window heights. The g-value for the three window heights without ventilation is very similar but the convection factor differ more (See Table 7.7). Once again a high convection factor seems to suggest a better reduction of g-value.

Table 7.7 Simulated g-value and convection factor for different types of window heights when no exhaust ventilation is present.

Window height	g-value	Convection factor [%]
1.2 meter	0.69	66.6
2.5 meter	0.69	55.6
8.0 meter	0.67	53.0

Note that possible reduction is as low as 7 % for the 8.0 meter window. An eight meter window would in many cases correspond to an entire façade. It could be discussed if it is reasonable to compare this with the same flow as the other window heights.

7.8 Cavity width with different exhaust velocities

Cavity width is the distance between the window pane and the solar screen. Three cavity widths where investigated in combination with the three different exhaust velocities. These simulations where performed for both the *at ceiling* and the *connected* exhaust location cases. The results of these two cases are shown in Figure 7.8 and Figure 7.9 below. Note that while 0.5 cm cavity width might not be that common in reality it is still investigated in this parameter study. This is to perceive trends and correlations between the reduction of g-value and the width of the cavity.

By default the size of the exhaust outlet is the same as the size of the cavity width. When changing the cavity width in simulations, the width of the outlet is changed to match this. If the same outlet velocity would be kept, the exhaust flow would be twice as big when changing from 10 cm to 20 cm outlet. In a real case it is of course the amount of exhausted air that is of interest, rather than the velocity in the exhaust air device. To compensate for this fact the exhaust outlet velocity is doubled when cavity width is 5 cm and halved when cavity width is 20 cm. For purpose of comparison the term equivalent exhaust velocity is used in this section. Equivalent exhaust velocity of 0.18 m/s mean the exhaust velocity is set to match the exhaust flow a 10 cm outlet would have if simulated at 0.18 m/s.

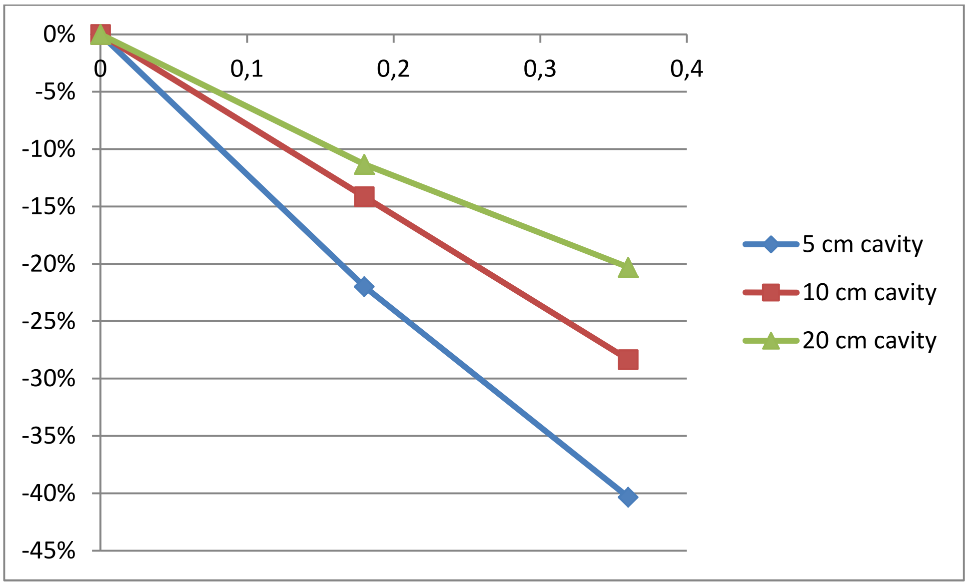

Figure 7.8 Reduction of g-value for different cavity widths in combinations with different equivalent exhaust velocities. Values are for the 'at ceiling' exhaust location.

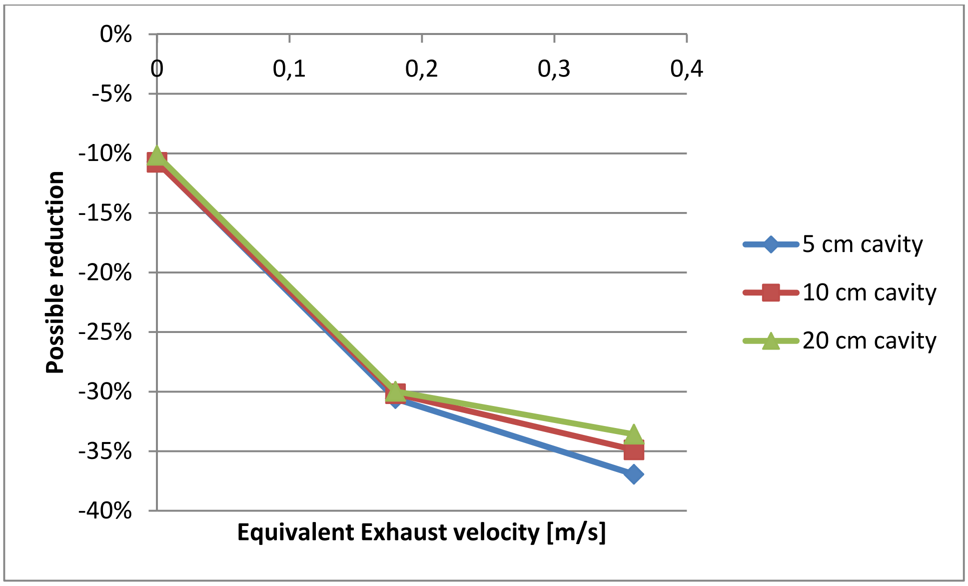

Figure 7.9 Reduction of g-value for different cavity widths in combinations with different equivalent exhaust velocities. Values are for the 'connected' exhaust location.

In Figure 7.8 it can be seen that smaller cavity width seems to result in an increased reduction of g-value within the flow range investigated. As opposed to Section 7.5 and 7.6 the convection factor and the g-value are very similar for the three investigated cavity widths. It seems the difference in reduction of g-value depends on other factors than those named in previous sections. Table 7.8 shows air temperature, air velocity and air flow in the top of the cavities investigated. As can be seen here, the 5 cm cavity has a significantly higher air temperature and a lower air flow. The heating effect these flows would have on 25 °C room air is around 300 W, for all cavity widths investigated. This means the different flows evacuate the same amount of heat from the cavity, but with a narrower cavity the heat is concentrated to a smaller volume of air. According to the authors this would be a reasonable explanation to the increased reduction possibility with a narrower cavity. For the same equivalent exhaust velocity the amount of extracted heat is larger if it is concentrated to a smaller volume. This seems to only be true for the *at ceiling* exhaust location case. The reason for this could be that a heat flow concentrated to a smaller volume of air mixes less with the air inside the room. In the *connected* case no mixing of the air in the cavity and the air inside the room occurs anyway, why there is no significant difference between the different cavity widths, as seen in Figure 7.9.

In Section 7.5 it seemed that the more the natural flow in the cavity is prevented, the more the g-value is reduced. In Table 7.8 the average temperature, velocity and flow is shown, for the different cavity widths when the ventilation is shut off and the cavity is open at the top and bottom why the air flow only occur due to natural convection. As can be seen in the table, the natural flow is highest in the 20 cm case. This would suggest the 20 cm *connected* case would show a better possible reduction at lower equivalent flows compared to the other cavity widths investigated. However, according to Figure 7.9 the three cavity widths shows almost identical possible reduction for low equivalent exhaust velocities. This is unexpected and no explanation to this has been found.

Table 7.8 Average air temp, velocity and flow in top of cavities for different cavity widths. Values are for the 'at ceiling' exhaust location.

	5 cm cavity	10 cm cavity	20 cm cavity
Average air temp [°C]	35.9	32.2	30.4
Average velocity [m/s]	0.47	0.36	0.23
Flow [l/s]	24	36	46

Two simulations where also made when a 20 cm wide cavity where compared for equivalent exhaust velocities of 0 m/s, 0.18 m/s and 0.36 m/s with both 10 cm and 20 cm exhaust size. The one with 10 cm showed almost complete agreement with the 20 cm exhaust size (See Figure 7.10). It seems that the assumptions with equivalent exhaust velocities are valid, meaning the reduction of g-value depends only on the equivalent exhaust velocity, i.e. the exhaust flow and not the actual exhaust velocity.

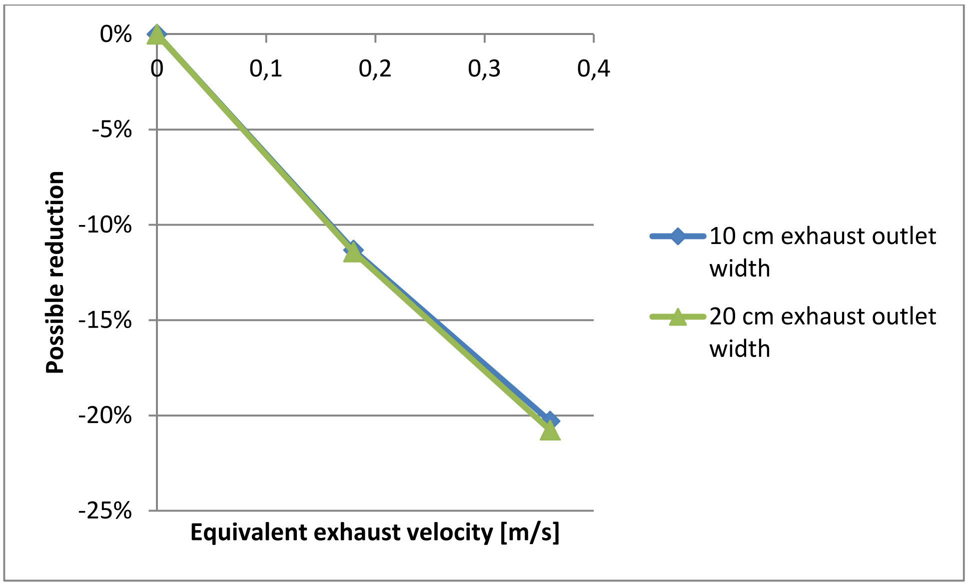

Figure 7.10 Reduction of g-value for different equivalent exhaust velocities when exhaust outlet width is changed. Values are for the 'at ceiling' exhaust location case.

7.9 Solar radiation with different exhaust velocities

The effect of solar radiation in combination with the three different exhaust velocities where investigated. The results of these simulations are shown in Figure 7.11 below.

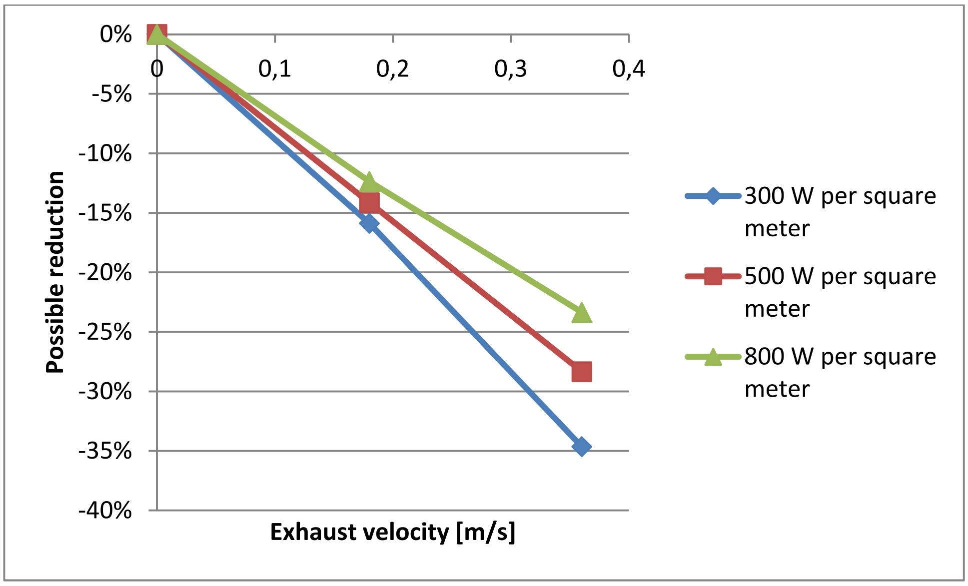

Figure 7.11 Reduction of g-value for different window types in combinations with different exhaust velocities. Values are for the 'at ceiling' exhaust location case.

The investigation shows solar radiation can have a significant affect upon the possible reduction of g-value. A higher solar radiation means a larger convective heat flow and also a larger natural airflow in the cavity between window and screen. In turn this leads to, that for the same ventilation flow, the possible reduction is better with a lower solar radiation.

Although the graphs seem to be linear in the investigated exhaust flow span, this will most likely not be the case for higher flows. The theoretically highest reduction, obtained by dividing the convective part of the g-value with the actual g-value, should not be significantly changed by changing the solar radiation and the graphs in Figure 7.11 should flatten out for higher exhaust flows.

7.10 Window type, screen type and exhaust locations effect on final g-value

To give some perspective regarding if it is worth implementing a solution with a ventilated cavity, Figure 7.12 show g-values for some of the different design options when using this system. The parameters not mentioned in the figure are set according to the standard case, prescribed in Table 7.2. The possible reductions calculated previously in this chapter are values for reduction of g-value when adding ventilation to a case when the screen is already in place. These g-values are within the grey circle in the figure below. For comparison, g-values for windows without screen or any other precaution taken to limit g-value are shown as well.

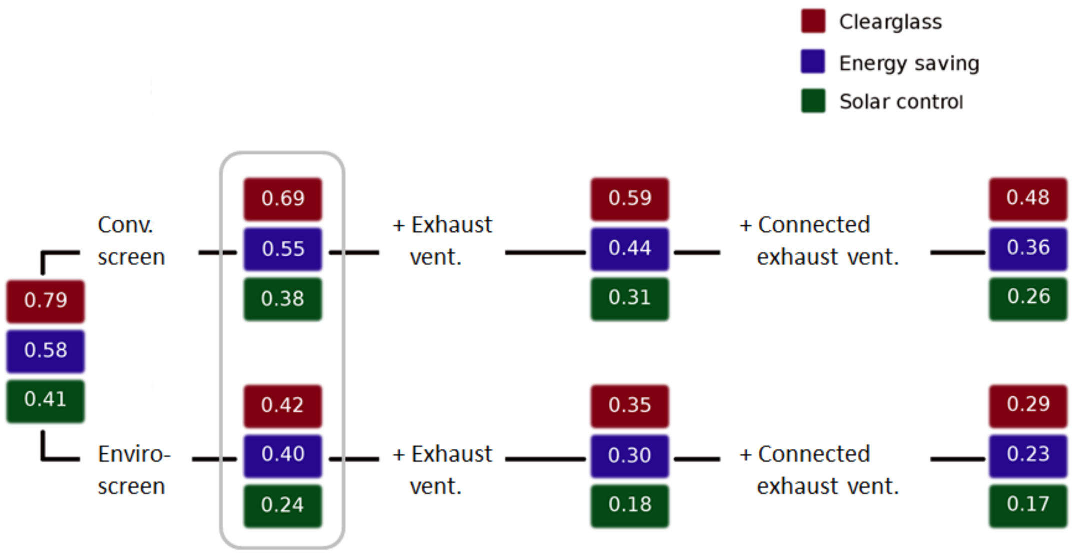

Figure 7.12 g-values for different design options: only windows, adding different screens and ventilating the cavity with different connections.

8 Analysis of How a Perforated Screen Affects the Result

This study is limited to usage of solid screens, due to the difficulty to accurately model perforated screens in COMSOL. However, in reality screens with perforations are often used, to enable visibility through the window even when the screen is in use (see section 3.3). These perforations might enable air to pass through the screen, which will change the air flow pattern around the screen and window. In turn this can affect the convective heat transfer around the window and thereby change the efficiency of using a ventilated cavity, compared to when a solid screen is used. Hence, it is needed to investigate the possible effects a perforated screen will cause. The aim is not to find exact results, but to see if the perforations have any impact at all and, if possible, see if the result of this study also is applicable to perforated screens, and what changes it might cause the result.

8.1 Method for analyzing impact of perforations in screen

The perforated screens are modelled by adding openings to the screen of previous models. Size and amount of openings are varied to simulate different openness factors of the screen, where the openness factor is the ratio between open and total area of the screen. This factor is varied from 0 to 0.25. Since the openings in the screen causes a decreased screen area compared to a solid screen, the heat source representing the absorption of the screen is increased, making the amount of heat absorbed the same for solid and perforated screen. The decreased area of the screen also causes changes in radiative heat flux. The impact from the radiative heat flux is difficult to measure for the perforated screens in COMSOL, why the simulations are run with the radiation turned off and thereby all heat transfer occur by conduction and convection.

Many parameters are varied to see in what cases the perforations have effect. The different parameters are listed in Table 8.1below.

Table 8.1 Different parameters investigated for the analysis of the perforated screens.

Location of exhaust device	At cavity / 25 cm above cavity /
Exhaust air flow	0 to 0.36 m/s
Type of screen	Conventional / Low emissivity
Width of cavity	5 cm / 15 cm
Height of window	1.0 m / 1.5 m / 2.0 m

The different outflow velocities are selected to investigate the flow around the natural flow, caused by temperature differences, in the cavity. This is because the perforations are believed to affect the flow around and trough the screen differently if the outlet flow is higher or lower than the natural flow. The other parameters are selected on the same basis as for the main study, see chapter 7.

8.2 Results

For the cases investigated, the perforations will cause an air flow through the screen. This flow will affect the heat load in the room compared to a solid screen. The flow through the screen changes the conductive and convective heat fluxes around the screen and will change the amount of heat entering the room and evacuated through the ventilation. How the heat fluxes changes with the openness factor of the screen, for different parameters, is shown in diagrams below.

It is observed that perforations in the screen lead to increased velocity due to natural convection in the cavity. Hence, more heat is evacuated through the cavity by convection, compared to a solid screen. Thereby, there is a potential to evacuate more heat through the ventilation.

To compare how the perforated screens perform compared to a solid one, the ratio of heat flux trough the room boundary and the total absorbed heat, in panes and screen, is studied. In the plots in this chapter, the difference in ratio between a perforated and solid screen is shown, why a negative value in the graphs means perforations cause a decreased g-value. Note the difference is shown in percentage, not percent. It should be stressed that the reductions in the diagram do not directly correspond to a change of the total g-value of the window, due to simplifications made in the model, such as turning off radiation. Further, the direct transmission needs to be added for proper g-value calculations. The convective heat flux entering the room represents about 30 – 50 % of the total g-value. For cases when ventilation is turned on this drops even further.

For all cases investigated where the ventilation is not connected to the cavity, the perforations will improve the efficiency of using a ventilated cavity or the effects will be very small. How the convective heat load on the room changes for screens of different perforations compared to a solid screen is shown in Figure 8.1 below. The result is the same for all investigated parameters.

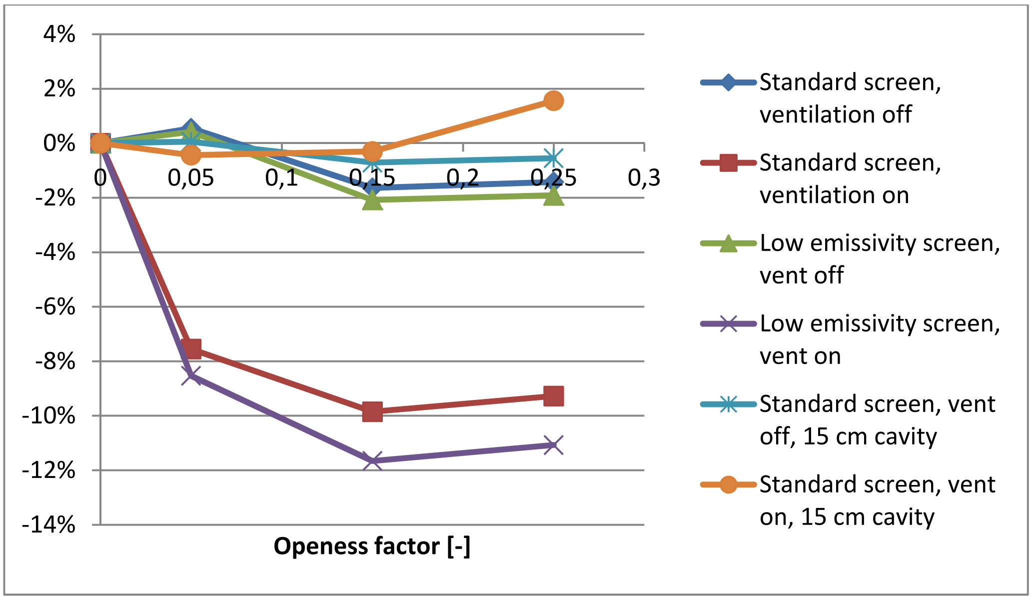

Figure 8.1 Change of ratio of heat flow through room boundary and total absorbed heat, for different openness factor of the screen, investigated for different types of screen. The cavity is 5 cm wide, if not stated elsewise.

The reduction of heat load on the room occurs because the perforations cause air flow through the screen, towards the window. This air flow will erase the convective and conductive heat transfer from the screen towards the room, by pushing the heated air towards the cavity. How this affects the temperature around the screen is shown in Figure 8.2 below. The air flow towards the window pushes the heated air towards the ventilation why more heat will be evacuated. Due to the decreased convective and conductive heat transfer towards the room, a perforated screen will lead to decreased heat load on the room even when the ventilation is turned off. However, the effect of using a perforated screen is small in this case.

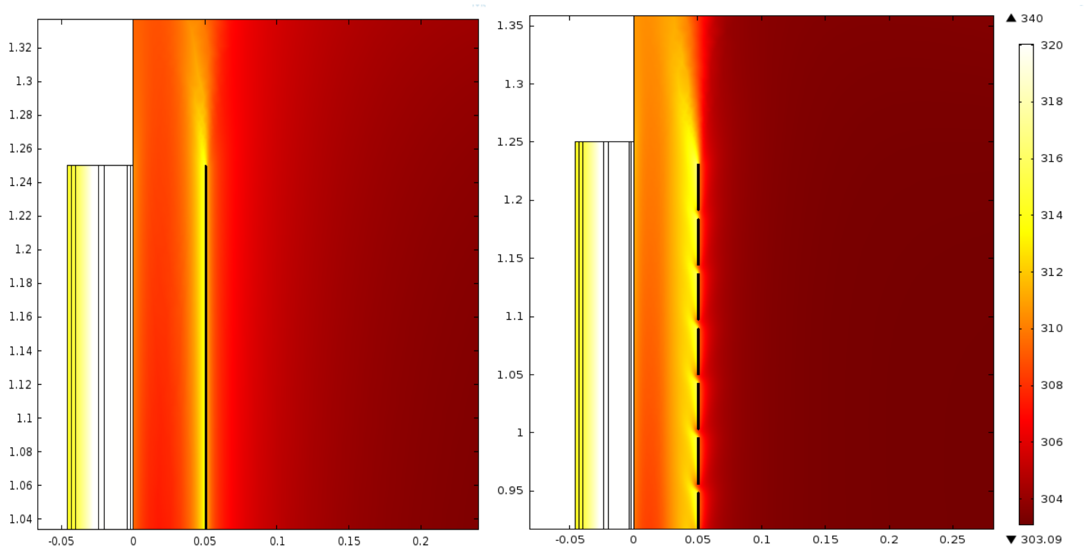

Figure 8.2 Temperature [K] around the solid screen (left) and perforated screen (right). Note the difference of heated air on the right side of the screen.

This tendency is the same as long as the flow through the cavity is allowed to flow freely. If the top of the cavity is closed, the heated air in the top of the cavity will instead press through the screen towards the room and the heat evacuated trough the ventilation will decrease. The different air flow patterns are illustrated in Figure 8.3 below.

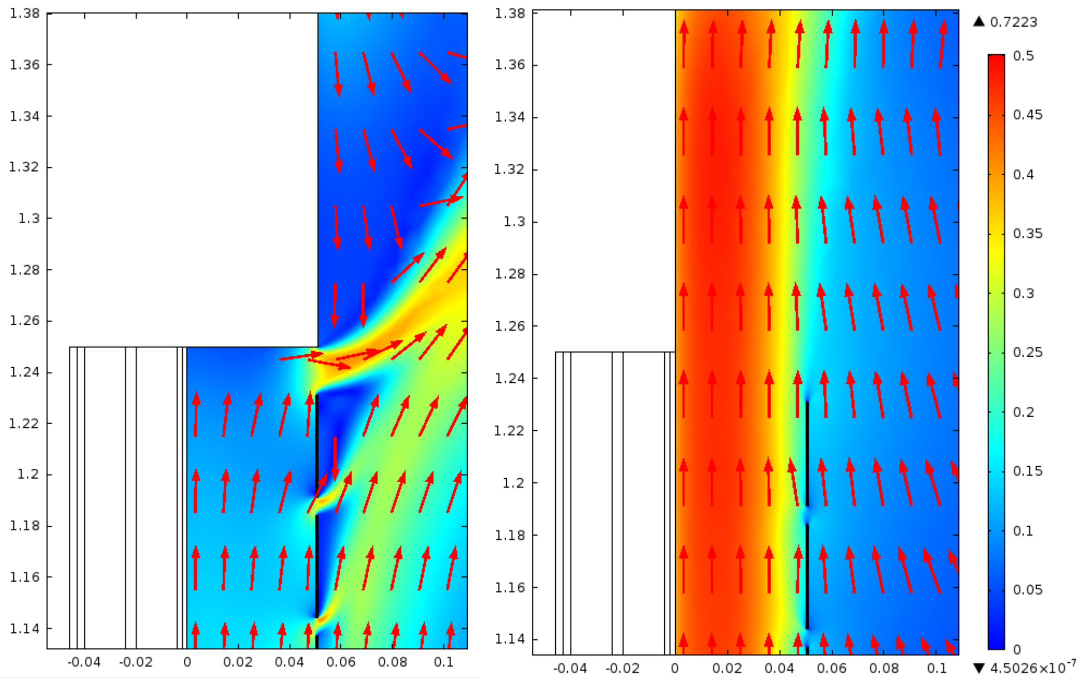

Figure 8.3 Air velocity [m/s] around the top of the screen when the cavity is closed (left) and open (right) at the top. Note flow direction through the screen. Surface indicates flow speed, arrows direction.

If the outlet is placed at the top of the cavity, but with a low flow rate, the effect is the same, but the flow towards the room decreases when outflow increases. The critical flow, where the perforations instead causes a decreased heat load on the room, is observed to be a bit less than the flow caused by natural convection in the cavity, for a completely open cavity. If the flow is higher than or the same as the critical flow, the effects will be the same as for the case when the ventilation outlet is located at the ceiling, i.e. the perforations always improves the system. An example of how the flow changes the heat load on the room for different exhaust flows, when the outlet is connected to the cavity, is shown in Figure 8.4. Note that around 0.2 m/s the effect of the perforations goes from increasing the heat load to decreasing it meaning, in the case below, an outlet velocity of 0.2 m/s is the critical flow.

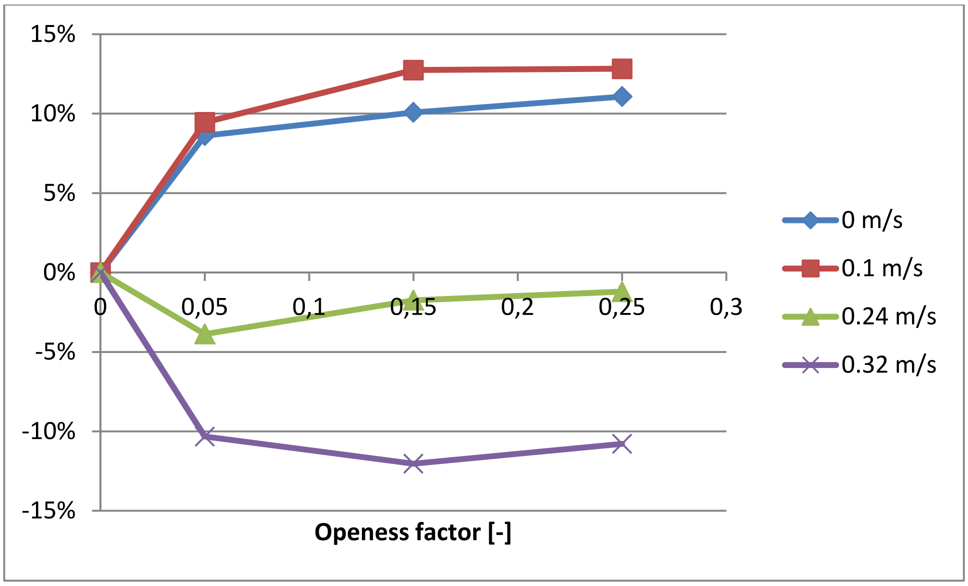

Figure 8.4 *How the ratio of heat flow through room boundary and total absorbed heat changes when perforated screens are used, compared to a solid, when the outlet is connected to the cavity.*

The critical flow varies with different parameters, such as window height, solar load or cavity width. In the figures below, it can be seen that these parameters all affect the critical flow, since changing them cause changes to the effect of the perforations. The exhaust is connected to the cavity and the equivalent exhaust velocity (see section 7.8) is kept constant in all cases, to 0.24 m/s. Low emissivity screen is used in all cases.

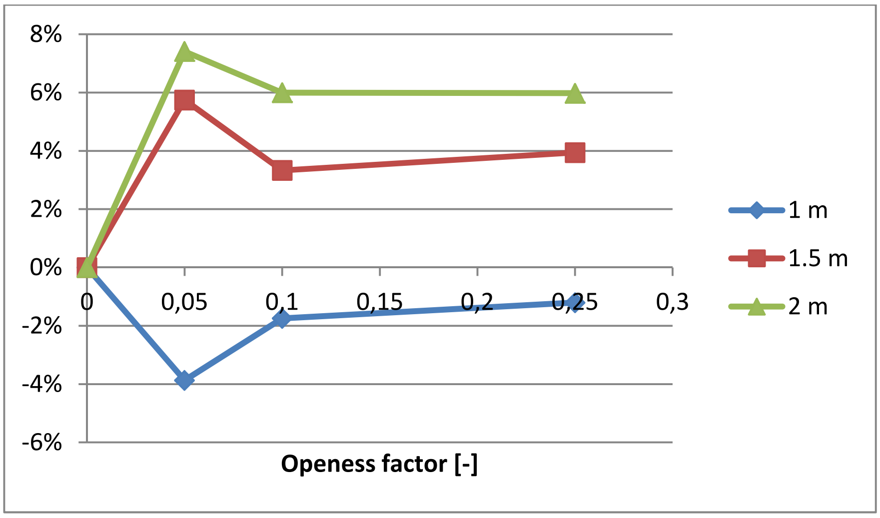

Figure 8.5 *How the effect of perforations changes the ratio of heat flow through room boundary and total absorbed heat, for different window height.*

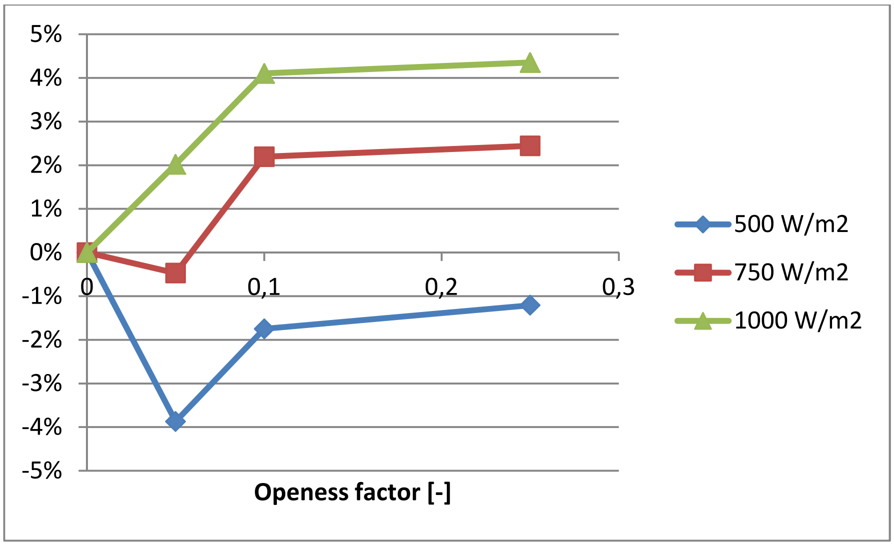

Figure 8.6 How the effect of perforations changes the ratio of heat flow through room boundary and total absorbed heat, for different solar load.

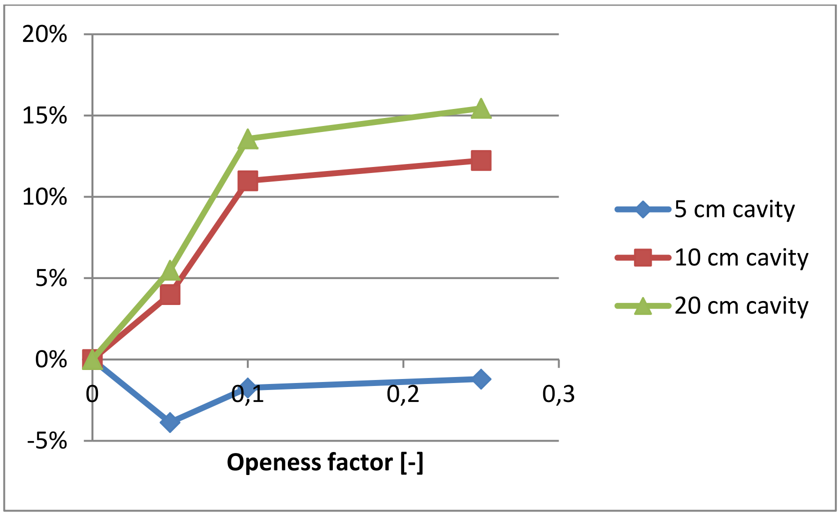

Figure 8.7 How the effect of perforations changes the ratio of heat flow through room boundary and total absorbed heat, for different cavity widths.

8.3 Effect of perforation for connected standard cases

The effects of the perforations are also investigated for the standard case, see Table 7.2, but with the outlet connected to the cavity. Simulations are run for all three kinds of windows. This is to find if main parts of the results from Chapter 7 are on the conservative side or not. The results are shown in Figure 8.8. From this it is found that addition of perforations will increase the heat flow to the rom, and thereby decrease the possible reduction of g-values for the cases with the *connected* exhaust location.

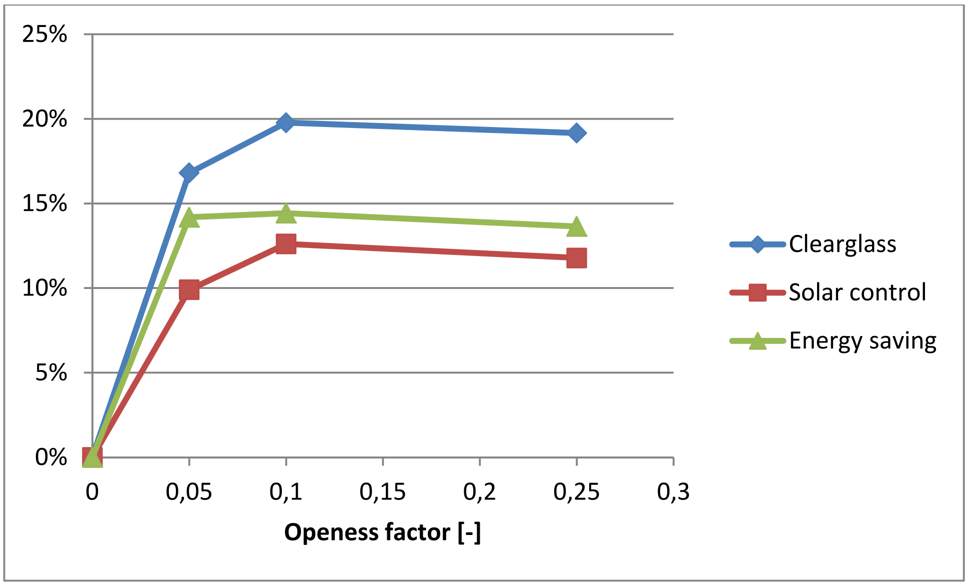

Figure 8.8 Change of ratio of heat flow through room boundary and total absorbed heat for standard case, but with connected outlet and all three window types.

8.4 Conclusion and Discussion regarding perforated screens

The effects of using a perforated screen, compared to a solid one, depend mainly on exhaust flow and exhaust location. Three different scenarios can be identified, depending on these variables.

For an outlet located at the ceiling, the usage of a perforated screen will improve the efficiency of using a ventilated cavity for all the cases investigated. Similar result is valid as long as the flow in the cavity between window and screen is unrestricted or when it is dominated by the outlet flow. This is because the perforations allow for air flow through the screen towards the cavity. This decreases the conductive and convective heat flow towards the room and also pushes more heat towards the ventilation outlet. If the flow in the cavity is blocked or limited by a low flow outlet connected to the cavity, the effect is the opposite and the heat load in the room increase with increased openness of the screen.

In short, three different cases occur, with different result regarding the heat load on the room compared to a solid screen, see Figure 8.9.

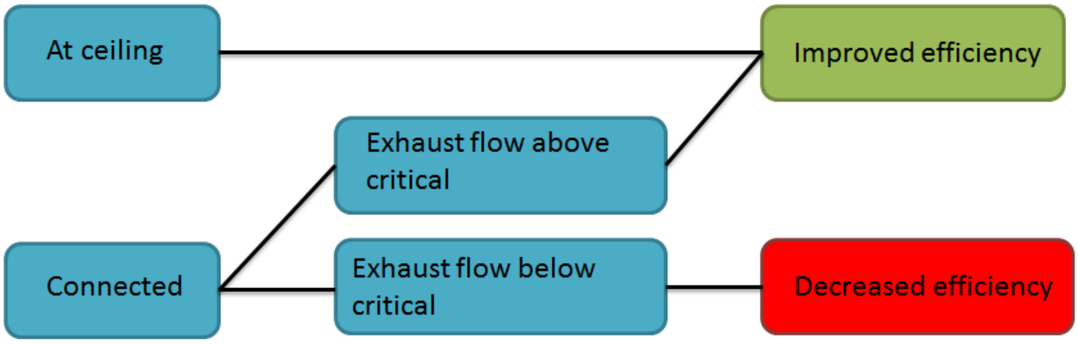

Figure 8.9 Flowchart of perforation effect for the 'at ceiling' and the 'connected' exhaust location case.

The critical flow is difficult to prescribe in a general manner and therefore needs to be decided for every individual case. For the cases investigated, a few variables increasing the critical flow are identified. These are window height, solar load and cavity width, which all increase the critical flow if increased themselves.

For windows of the standard height of 2.5 m, with an outlet connected to the cavity and the standard exhaust flow, the perforations will cause a higher g-value than for a solid screen.

The simplifications necessary for modeling a perforated screen are believed to cause some differences, compared to a real perforated screen. The sizes of the perforations in the screen of the COMSOL model, ranging from 1.6 mm to 17 mm, are larger than they would be for a real perforated screen. This is believed to cause a decreased resistance to flow, compared to a real screen. Further, the turned off radiation cause the perforated screens to be hotter than they would be in a real case. This cause more heat transfer by convection around the screen. These both simplifications are believed to exaggerate the effects of the perforations.

Due to the difficulty to model a screen with perforations fine enough to correspond to a real screen, this result only show in what direction the performance will be affected if the screen is air permeable. This investigation shows that to apply the results of this study on a perforated screen is on the safe side if the flux is not restricted at the top of the cavity. If the flow is limited, the improvement of g-value calculated in this study will be decreased if a perforated screen is used.

9 Conclusion

The g-value for a window with an internal screen will be decreased when using exhaust ventilation to extract the heated air from the cavity between window and screen. Results from Chapter 7 shows that the reduction of g-value is dependent on multiple parameters in combination with each other. This means that a general value for reduction of the g-value when exhaust ventilation is applied cannot be given. Hence, the purpose of this thesis must be answered by giving case specific reduction of g-value, rather than a general number.

The possible reduction of g-value by adding exhaust ventilation is very dependent on exhaust flow, where increased flow gives more reduction of the g-value. In fact, for most cases when the exhaust device is not sealed to the cavity, the reduction is doubled when the exhaust flow is doubled.

Further this study shows the system behaves very differently depending on the location of the exhaust air device. To mount the exhaust device sealed to the cavity seems to be the preferable solution compared to having the non-sealed exhaust (in this text denoted *connected* and *at ceiling* respectively). In reality many cases would be a compromise between a sealed and a non-sealed exhaust device. According to the results, this case would produce values for possible reduction somewhere in between the sealed and the non-sealed case, at least when flow is between zero and standard.

To answer the question, how big possible reduction that can be expected, somewhat of a reasonable standard case must be decided. In this thesis the authors has found it reasonable to account for a flow of 25 l/s for a 1.35 meter wide window. The rest of the set parameters can be seen in Table 7.2.

It should also be said that the values for possible reduction presented in this conclusion and in Chapter 7 are valid for the standard case described. Large deviations in design will affect the possible reduction according to the trends seen in Chapter 7. A taller window or an increased solar load will cause less reduction of the g-value. When the outlet is not connected to the cavity, a narrow cavity gives better reduction of the g-value than a wider. When the outlet is connected to the cavity, changes in cavity width do not cause changes to the results.

When using a low emissivity screen (surface emissivity 0.284 in this study) with a sealed exhaust device and the standard parameters described above, results show that a possible reduction of the g-value between -28 to -45 % is possible. This depending on which of the three investigated window types are used. Same window systems with a non-sealed exhaust device and results drop to between -17 and -27%. Changing the screen to a Conventional (surface emissivity 0.815) and the possible reduction are between -30 to -35 % and -14 to -20 %, for sealed and non-sealed case respectively.

If no limit exists on the exhaust flow however, the possible reduction for each window should theoretically be as high as its convection factor. In one of the investigated window systems this is as high as 66 %. It appears as for, a high enough flow, the non-sealed exhaust device is preferable to the sealed. This is however in flow regions that, according to the authors, are unreasonably high.

All the data covering possible reduction in this thesis are without having accounted for perforation in the screen surface. Chapter 8 shows probable scenarios and how effects of perforations affect the possible reduction under different conditions. How big the effects from using a perforated screen are, is however hard to tell from this data. The following correlations seem to hold true for perforation:

- When exhaust device is not connected to cavity; increased perforation means increased or no difference in possible reduction, i.e. a lowered g-value.
- When exhaust device is connected to cavity and exhaust flow is lower than a for the window system specific critical flow; increased perforation means decreased possible reduction, i.e. less reduction of the g-value.
- When exhaust device is connected to cavity and exhaust flow is higher than a for the window system specific critical flow; increased perforation means increased possible reduction.

According to data in Section 8.3 it seems as the 25 l/s per window is lower than the critical flow for the set standard window system. As mentioned this seems to indicate that any values for possible reduction acquired under these conditions are probably on the non-conservative side, when the exhaust device is sealed to cavity. To summarize, the values for possible reduction regarding a sealed screen will most likely not be better for a real case with a perforated screen. For the non-sealed case however, if perforation has any effect at all, it should only improve the possible reduction.

9.1 Design recommendations

These design recommendations are focused on how to obtain a window system for best possible reduction of g-value (See Section 7.3). The final g-value is however also considered as a secondary objective. In Section 7.1 exhaust air flow of 25 l/s for a 1.35 meter wide window, where decided as reasonable, according to the authors. These recommendations are formed with the assumption that this flow is used and is referred to as reasonable flow in text below.

- **Keep window height low**

Although window height perhaps is not a design parameter that can be influenced, it should be noted that higher windows decrease the possible reduction when same exhaust flow is used, why using exhaust air to ventilate the cavity are less effective with increased window height.

- **Keep cavity width between screen and window low**

Especially when the exhaust location is placed so it is not sealed towards the cavity the cavity width seems to matter. When choosing cavity width in the range of 5 – 20 cm, a lower cavity width means the possible reduction increases and a lower g-value can be obtained.

- **Seal connection between cavity and exhaust device**

The general recommendation is to keep the exhaust device and the screen as sealed as possible, meaning the system should aim to mimic the design denoted as *connected* in Chapter 7. For most combinations, this increases the possible reduction both for an unventilated window and for ventilation flow at least up to 25 l/s per window.

- **Use Energy saving window or Solar control window rather than Clear glass**

As a general guideline, the possible reduction is better and the final g-value lower for Energy saving and Solar control windows compared to Clear glass windows. Generally the possible reduction is best for the Energy saving windows but the final g-value is lower for the Solar control windows. Note that the statements above are valid for the windows used in Chapter 8, described in Section 3.2.4. It is however the opinion of the authors that these general guidelines most likely are valid for similar window products as well.

- **Use screen with low surface emissivity**

A screen with lower emissivity increases the possible reduction as well as lowers the g-value for close to all combinations investigated in this thesis. From Figure 7.12 the importance of using a low emissivity screen is clearly shown. Only changing to a low emissive screen has about the same effect as when using exhaust ventilation connected to the cavity for a conventional screen. It is the authors opinion that connecting exhaust ventilation to the cavity is a far more complicated and costly operation than changing to a low emissivity screen.

10 Discussion

As can be seen in Chapter 7, the exhaust flow has a vital effect on the possible reduction of g-value and it is important to connect the exhaust device to the cavity. Installing this system would in most cases demand longer and more ducts for the exhaust air, compared to normal locations of exhaust air devices. This would cause an extra economic cost and a small extra energy usage due to transporting the exhaust air longer distance. If no consideration is taken to these facts one might argue that as long as there is a demand to exhaust air, the air might as well be taken from the windows. Then the exhaust flow per window would be decided by the total exhaust demand and the amount of windows. Since an economic analysis is outside the scope of this thesis, only conclusion that can be drawn regarding this is that available exhaust flow can vary a lot depending on situation, why the efficiency of the system also will vary a lot.

In Section 6.5.2 it can be seen that results from the COMSOL model show very good agreement with g-values from WINDOW 7.2 when window and screen without exhaust ventilation is simulated. It should however be said that this is for a case where 2D-simulations should be very suitable compared to other cases. For example the screen cavity is considered perfectly air tight at the right and left side. This is a situation that would rarely occur in reality. Further the COMSOL model simulates the exhaust device as a 1.35 meter long unit, stretching all along the window width. In reality this would most likely be placed near the middle of the widow width and not be covering the entire width of the window. Which effects these 3D-properties would have on the 2D-model is difficult to approximate.

10.1 Further studies

The model created in this study was created in 2D. To make it more similar to a real case, it would be of interest to build the model in 3D and compare the results. This would show side effects, such as effects of air flow in the cavity between window and solar screen at the sides of the window. It would also show the effects of having an exhaust device not covering the entire width of the window. Creating a 3D model working similar to the 2D model of this study is possible in COMSOL, however it will require a lot of computational power to reach reasonable computational time.

In the study, computed g-values were validated by comparing to ISO 15099 and showed good agreement. However, this comparison could not be done for cases when the ventilation was turned on. Since the global power equilibrium in the model always was good, it could be assumed that it was accurate even when ventilation is on, but for a final verification it would be interesting to compare to measured values from a real model. By using both solid and perforated screens, the effects of the perforations can be investigated as well, giving accurate results instead of the approximations in this study.

This study strictly considered the reduction of g-values the exhaust ventilation cause a window with internal solar screen. The results from this study can be used in a wider study, considering economic and environmental analysis. Analyzing the economic benefit of using the solutions is probably necessary before implementing them in real buildings. Things to consider are, for example, cost of installing and running the ventilation of the cavities, energy savings due to the decreased solar load and possible usage of the heat in the exhaust air.

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12 Appendices

Appendix A

Values for windows used when testing ISO model implementation

Manufacturer	Name of product	Geometry[1]	U	g	LT
Pilkington	Optifloat clear	4-16Ar-4-16Ar-4	1.7	0.66	0.74
	K-glass	4K-16Ar-4-16Ar-K4	0.8	0.58	0.63
	Suncool 70/40	6C(74)-16Ar-4-16Ar-S(3)4	0.6	0.38	0.64

Values for windows used in full model whit room

Manufacturer	Name of product	Geometry[1]	U	g	LT
Pilkington	Optifloat clear	4-16-4	2.7	0.78	0.82
	K-glass	4K-16Ar-4-16Ar-K4	0.8	0.58	0.63
	Suncool 70/40	6C(74)-16Ar-4	1.1	0.43	0.71

Values for window panes and screens used in all models

Layer	WINDOW 7.2 ID[2]	Thickness	λ	τ_{ir}	ε_f	ε_f
6	4118	5.9	1.00	0.000	0.840	0.840
4	4116	4.0			0.840	0.840
4K	4140	3.8			0.837	0.173
6(C)74	4024	5.9			0.837	0.042
S(3)4	4031	3.8			0.063	0.837
Screen for validation of COMSOL model[3]	12000	1.0	0.15	0.166	0.706	0.592
Conventional screen[3]	7002	1.0	0.15	0.093	0.815	0.815
Enviroscreen[3]	20001	1.0	0.15	0.098	0.284	0.284

[1] Numbers outside brackets denotes thickness of layer and cavity. Denotation Ar means cavity is filled with argon.

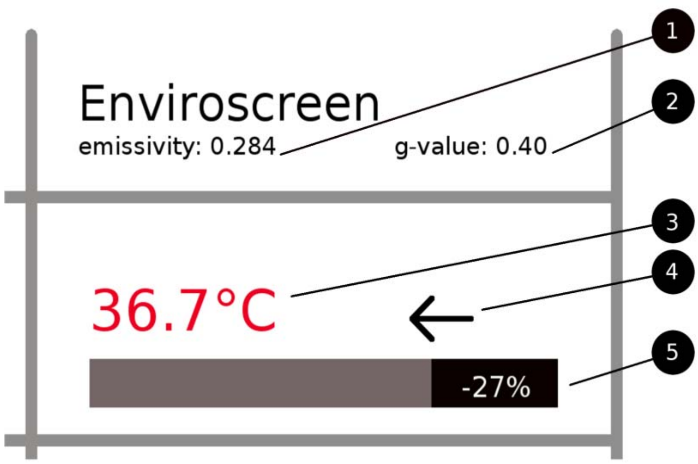
[2] ID in database W7-CGDB-2.0

[3] Screen is a modified version of these ID's. Openness factor is changed to 0.0

Appendix B

On the following page values of the possible reduction are presented for three common window types in combination with two different screens and two exhaust locations. These values are valid for design parameters and conditions similar to the ones presented on the right. Below the signs and values presented are explained.

Design parameter	Value
Exhaust flow per window	25 l/s
Window width	1.35 m
Window height	2.5 m
Distance betw. screen and window	0.1 m
Solar radiation	500 W/m^2



1. Surface emissivity of named screen. In these cases this denotes both sides. The significant side however is the one facing the window.
2. g-value of the named window in combination with the named screen before exhaust air is applied. This is for a case where air can flow freely through the cavity between screen and window.
3. Mean temperature of the screen-surface facing the room.
4. The values presented are from a model where screen is considered perfectly impenetrable by air. The arrow indicates probable effect on reduction value a perforated (real) screen would have. Left indicates that reduction value would probably be higher in real case.
5. Possible reduction of g-value for the named window/screen system when exhaust ventilation is applied.

2-glass Clearglass
(g-value: 0.78)

	Conventional screen emissivity: 0.815 g-value: 0.69	Enviroscreen emissivity: 0.284 g-value: 0.42
At ceiling	47.0°C ← -14%	39.4°C ← -17%
Connected	48.9°C → -30%	40.9°C → -33%

3-glass Energy saving
(g-value: 0.58)

	Conventional screen emissivity: 0.815 g-value: 0.55	Enviroscreen emissivity: 0.284 g-value: 0.40
At ceiling	41.6°C ← -20%	36.7°C ← -27%
Connected	43.7°C → -35%	38.7°C → -43%

2-glass Solar control
(g-value: 0.43)

	Conventional screen emissivity: 0.815 g-value: 0.38	Enviroscreen emissivity: 0.284 g-value: 0.24
At ceiling	38.0°C ← -19%	33.7°C ← -23%
Connected	39.0°C → -32%	32.1°C → -28%