ANALYTICAL ASSESSMENT OF THERMAL PERFORMANCE OF A VENTILATED GLAZED FACADE SYSTEM

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ABSTRACT

This paper outlines a methodology for determining the thermal performance of a complex ventilated glazed facade system through desktop analysis and computer simulation.

There is a limited amount of published work on the overall thermal performance values of complex doubled skin facades with a ventilated cavity between the facade layers. A well designed double skin facade should significantly improve the thermal performance of the overall construction as the heat build up within the cavity can be dissipated through stack effects and natural ventilation.

The ISO 15099:2003 [1] standard provides a detailed calculation methodology for determining the thermal properties of complex glazing systems, including those with a ventilated cavity. A study has been undertaken of a particular assembly proposed for a commercial building in Brisbane. The study has used the ISO 15099 approach to determine the thermal properties of the glazing system. The assembly has also been modelled through dynamic thermal modelling and computational fluid dynamics (CFD) to compare its performance to the analytically derived solution using the ISO standard. The results show that there is good agreement between the analytical solution and the numerical analysis and that the approach presented herein can be used for determining the thermal performance of these types of facades where no manufacturer data exists.

INTRODUCTION

The Building Code of Australia (BCA) Section J deals with the energy efficiency provisions for new buildings. Section J2 in particular addresses the glazing requirements within new buildings for Deemed to Satisfy compliance. This requires calculations to be undertaken based upon window area, glazing system U-values and solar heat gain coefficient (SHGC) as well as shading provisions across the building to determine if the average air conditioning energy value attributable to the glazing is below the prescribed allowance.

The design for a commercial building project in Brisbane includes a large double height entrance space served off a major city centre street. The external facade of this space is fully glazed to provide a high degree of transparency externally to the Brisbane CBD. This high glazing to wall ratio causes issues for compliance with Section J.

Different shading combinations have been considered to comply and the design team have arrived at a solution which provides a secondary glazing system, similar to the primary facade, separated by a ventilated cavity. To demonstrate compliance with the glazing requirements, it was necessary to determine the thermal performance of this double skinned facade arrangement.

The usual method for assessment of glazing under Section J is to use certified properties of the glazing assembly from the glazing manufacturer. The WINDOW software program published by Lawrence Berkeley National Laboratories (LBNL) is widely accepted in industry as the method for determining glazing system properties including U-value and SHGC. This utilises the ISO 15099:2003 (ISO15099,2003) standard as the basis for the calculations. One limitation with the software is that it is restricted to sealed glazing assemblies and therefore cannot be applied to a double skinned facade with a ventilated cavity. The ISO 15099:2003 standard does provide guidance and calculations for a ventilated cavity arrangement, relevant to this particular case.

The SHGC for this glazed construction has been determined through a desktop analysis utilising first principles and data provided by the glazing manufacturer for each glass laminate layer. This has been supplemented by guidance from ISO 15099:2003 and ISO 9505:2003 (ISO9059,2003). The results from this have been validated through calculations as well as dynamic thermal modelling. A steady state computational fluid dynamics has also been carried out to provide further verification of this approach to determine SHGC.

Nomenclature

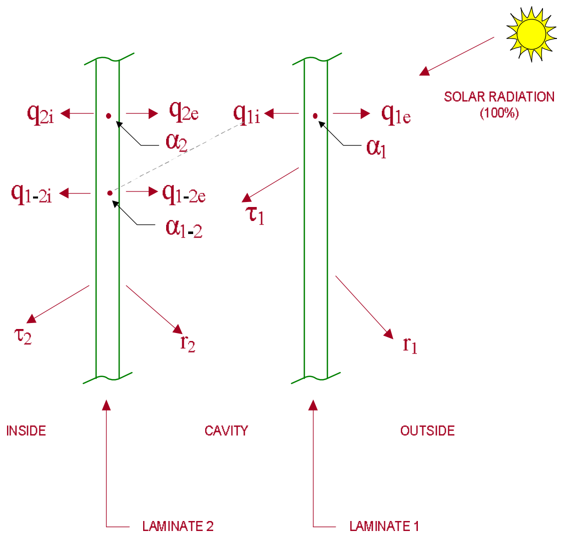

Figure 1 – Heat transfer diagram for the ventilated glazing system

Table 1 Nomenclature

Terms	Description
α	Absorptance – The fraction of solar energy which is absorbed by the glass
τ	Transmittance – The fraction of solar energy which passes directly through the glass
r	Reflectance – The fraction of solar energy which is reflected by the glass
q	Secondary heat transfer factor comprising of a convective and radiative component
ε	Emissivity – the fraction of absorbed energy which is radiated from a surface
Subscripts	Description
1	Relating to laminate 1
2	Relating to laminate 2
1-2	Relating to energy transfer from laminate 1 to laminate 2
i	Internal
e	External
T	Total
K	Counter
rad	Radiation – the radiated component of absorbed energy
Cv	Convective – the convective component of absorbed energy

Solar Transmission Through Glass

When solar radiation strikes a window the energy is broken up into three components, the energy which is transmitted directly through the window (Transmittance), the energy which is reflected off the window (Reflectance) and the energy that is absorbed by the window (Absorptance).

These three components sum to the total incident solar radiation, such that:

$$\alpha + \tau + r = 1 \quad (1)$$

The absorbed component of energy is directed towards the exterior or the interior via radiation and convection.

The SHGC is a ratio of the energy transferred through the window into the internal space compared to the solar radiation that the window is exposed to. The SHGC can be determined by calculating the transmittance through the glass and the fraction of absorptance which enters the space through convection and radiation.

DESKTOP ANALYSIS

Both glazing layers proposed for the foyer are a laminated glass comprising of two panes of glass with an intermediate interlayer bonded together to form a single laminated layer. The properties of a single laminate layer have been calculated using the WINDOW program and provided by the manufacturer. The properties are outlined in table 1. These values are used to determine the SHGC for the total glazing system assuming each layer has these properties.

Table 2 Glass Properties

	τ	r	α	e_e	e_i
Glass	0.32	0.06	0.62	0.84	0.2
	U Value		SHGC		
Glass	3.44		0.46		

It is noted that SHGC changes with incident solar angle and this is taken into account using the ISO 15099 method by the WINDOW calculator. However, for this desktop analysis solar angle and spectrum have been ignored and the overall SHGC is used.

The proportion split of the incident solar radiation across the various means of energy transfer previously described have been determined from the properties of the glass. For simplicity these calculations are not repeated here as this is not the focus of this study.

The cavity is completely open at the bottom and has a continuous open slot of 200mm at the top. The width of the cavity (i.e the separation between the glazing layers) is 1800mm and the whole surface is exposed to the environment. This arrangement has a reasonable level of exposure and suggests that the convective components from the glass surfaces facing the cavity could transfer to the air cavity volume and dissipate through natural ventilation or stack effect. This will be dependent on the cavity temperature and size of the boundary layer established.

The energy which the second laminate is exposed to as a result of the performance of the first laminate is:

τ_1 =0.32 – where $\tau 1$ is the transmittance through laminate 1

$q_{rad,1i}$ = 0.028 – where $q_{rad,1i}$ is the inward radiated component of the absorbed energy

The final component to consider is the radiant energy reflected between the two laminate surfaces bounding the cavity.

From undertaking further analysis of this energy transfer utilising the glass properties, the amount entering into the conditioned space through this path is calculated as 0.15%.

There will be a further decreasing interaction across the surfaces. However as this calculated component is already small, further interactions have not been considered as they are not significant.

Therefore, the total energy emitted into the space is the total SHGC.

$$\begin{aligned} SHGC &= \tau_T + q_{2i} + q_{1\text{-}2i} + q_{2\text{-}l\text{-}2i} \\ &= 0.1024 + 0.0447 + 0.006328 + 0.0015 \\ &= 0.155 \end{aligned} \quad (2)$$

In summary, the significant fractions of energy that are transferred into the space are the transmitted solar load through the second pane (τ_T), the inward flow of absorbed heat from the second pane (q_{2i}), small fractions of absorbed heat that radiate off the front laminate and flow through the second laminate ($q_{1\text{-}2i}$) and the radiation from the front of the second laminate that is reflected off the back of the first laminate and absorbed again by the second laminate.

The cavity is key to this reduction in SHGC. The ability for the cavity to accept and dissipate the convected components of energy from the two glazed surfaces bounding it is critical and this assumption made in the desktop analysis will now be tested further.

Validation

To validate the analytical calculation, a non ventilated cavity case was also calculated and compared to the SHGC output from Window 5 (i.e. sealed double glazed unit).

The SHGC for the non ventilated case will be as for the ventilated case with the additional included convective energy previously considered to have dissipated through natural ventilation.

In the non ventilated case these convective components are not exhausted through natural ventilation and are contained within the cavity. The flow of energy from the air cavity will be partly to outside through the first laminate and partly to inside through the second laminate. The distribution will be based on the thermal resistance of the laminates, cavity and air films.

By using a similar methodology and using a simplified heat transfer model provided by ASHRAE Fundamentals Chapter 15, Eq 19 (ASHRAE,2009). This provides a calculated SHGC of 0.319.

This compares well to the value of 0.32 calculated by the WINDOW software. This suggests that the desktop approach taken to calculate the SHGC for both the ventilated and non-ventilated cases is valid.

In order to determine that the convective components from the glass surfaces facing the ventilated cavity are effectively dissipated through natural ventilation within the cavity, determination of the properties within the cavity, primarily temperatures and air velocities, are necessary.

There are two methods for evaluating this. The first utilises the heat convected into the space to determine conditions within the cavity. The second utilises the pressure drop for the natural ventilation due to the physical configuration of the cavity to determine the cavity conditions.

Method 1 – Utilising the Convected Heat into the Cavity to determine Cavity Conditions

To do this, we will use the methodology in ISO15099:2003 Section 7.4.2.3 – Ventilated Gap. Equation 112 in section 7.2.3 provides the following equation for determining the heat transfer to the gap by ventilation:

$$q_{vl,i} = \frac{\rho_i \; x \; C_p \; x \; \emptyset_{vl,i} \; (T_{gap,i} - T_{gap,o})}{H_i \; x \; L_i} \quad (3)$$

Where:
$q_{vl,i}$ - Heat transfer to the gap by ventilation, in W/m^2.
ρ_i - Density of air in cavity. (kg/m^3)
c_p - Specific heat capacity of air in J/kgK.
$\emptyset_{vl,I}$ - Air flow rate in the cavity in m^3/s.
$T_{gap,i}$ - Temperature of air inlet into the cavity (K).
$T_{gap,o}$ - Temperature at the outlet of the gap (K).
L_i - Length of the cavity (m). For simplicity a unitary width has been assumed.
H_i - Height of the cavity (m).

In addition we also know that $q_{vl,i}$ is equal to the sum of the convective components outlined above multiplied by the total solar irradiance.

$$q_{vl,i} = (q_{cv,li} + q_{cv,2e}) \; x \; Q_{solar} \quad (4)$$

Where Q_{solar} is the total (direct and diffuse) solar irradiance in W/m^2.

From the analysis, the following were established for $q_{cv,li}$ and $q_{cv,2e}$:

$q_{cv,li}$ = 0.112
$q_{cv,2e}$ = 0.028

Other factors in the equations are as follows:

ρ_i = 1.164kg/m^3
c_p = 1006.2 J/kgK
H_i = 5.75m
L_i = 1m (Assume a 1m width of façade)
Q_{solar} = 620W/m^2.

The Q_{solar} value is taken from CIBSE Guide A (CIBSE,1998) Table A.2.35(f) 25oN with appropriate correction factors applied for the southern hemisphere. The maximum SE aspect direct solar irradiance is 540W/m2, occurring at 8am on December 21 with a corresponding diffuse radiation of 80W/m2. Brisbane's latitude is 27^{o}29', and the application of 25^{o}S has been taken as worst case.

Q_{solar} = 540 + 80 = 620W/m^2
$q_{vl,I}$ = 620 x (0.112 + 0.028) = 86.8 W

Therefore,

$$q_{vl,I} = 86.8 = \frac{1.164 \; x \; 1006.2 \; x \; \emptyset_{vl,i} \; x \; (T_{gap,i} - T_{gap,o})}{5.75 \; x \; 1}$$

$$86.8 = 203.7\ \emptyset_{vi,I}\ x\ (T_{gap,i} - T_{gap,o})$$

$$0.426 = \emptyset_{vi,I}\ x\ (T_{gap,i} - T_{gap,o}) \qquad (5)$$

This provides a simple relationship for the facade between the flow rate through the cavity and the temperature difference between the top and bottom of the cavity. The flowrate through the cavity will largely be driven by the stack effect. This will depend on the driving pressure difference, the temperature difference across the cavity and the resistance at the top of the canopy.

Assuming a $T_{gap,i}$ of 27^{o}C or 300K, which is considered reflective of a peak load scenario for a SE façade (i.e early summer morning), and estimating an outlet temperature, $T_{gap,o}$ of 30^{o}C or 303K gives:

$\emptyset_{vi,I}$ = 0.142m^3/s

Average velocity in cavity $= \frac{0.122}{1.8}$ = 0.0789m/s.
Where the depth of the cavity is 1.8m.

This equates to a 0.71m/s exit velocity through the 200mm opening at the top of the cavity. This is considered to be a realistic figure and is validated by the computer simulation presented later.

Method 2 - Utilising the Air Pressure Drop through the Cavity to Determine Cavity Conditions

This method determines the conditions within the cavity through evaluation of the physical parameters of the cavity which contribute to the pressure drop and the temperature difference across the cavity that drives the stack effect.

From ISO 15099:2003 Section 7.4.4.3 Equation 121:

$$\Delta P_{T,i} = \rho_o T_o g H_o \cos\gamma \; x \; \frac{T_{gap,i} - T_{gap,o}}{T_{gap,i} \; x \; T_{gap,o}} \qquad (6)$$

Where:
$\Delta P_{T,i}$ - The driving pressure difference between the cavity and outside (Pa).
ρ_0 - Density of air at temperature T_0 (1.164 kg/m^3)
T_0 - Reference temperature (say, 27^{o}C = 300K).
g - Acceleration due to gravity (9.81 m/s^2).
γ - tilt angle of cavity in degree from vertical (0^o)
H_i - Height of the cavity (5.75m).
$T_{gap,i}$ - mean temperature of the air in the cavity (K)
$T_{gap,o}$ - mean temperature of the external environment (outside) (K)

Therefore,

$$\Delta P_{T,i} = 1.164 \; x \; 300 \; x \; 9.81 \; x \; 5.75 \; x \cos 0 \; x \; \frac{T_{gap,i} - T_{gap,o}}{T_{gap,i} \; x \; T_{gap,o}}$$

$$\Delta P_{T,i} = 19{,}697.5 \; x \; \frac{T_{gap,i} - T_{gap,o}}{T_{gap,i} \; x \; T_{gap,o}}$$

The flow within the cavity is described as pipe flow and a number of pressure factors also need to be taken into consideration:

$$\Delta P_{T,i} = \Delta P_{B,i} + \Delta P_{HP,i} + \Delta P_{Z,i} \qquad (7)$$

and :

$$\Delta P_{B,i} = \frac{1}{2} \; x \; \rho_o \; x \; V_i^2 \qquad (8)$$

$$\Delta P_{HP,i} = 12 \; x \; \mu_i \; x \; \frac{H_i}{b_i^2} \; x \; V_i \qquad (9)$$

$$\Delta P_{Z,i} = \frac{1}{2} \; x \; \rho_o \; x \; V_i^2 \; x \; (Z_{inl,i} + Z_{out,i}) \qquad (10)$$

Where
$\Delta P_{B,i}$ -The Bernouilli pressure loss in the cavity,i (Pa).
ρ_i - Density of air at the cavity air temperature $T_{gap,i}$ (1.2 kg/m^3)
V_i - Mean velocity of air (m/s)
$\Delta P_{HP,i}$ - The Hagen-Pouiseuille pressure loss in the cavity (Pa).
H_i - Height of the cavity (5.75m)
μ_i - Dynamic viscosity of air at cavity air temperature $T_{gap,I}$ (1.872 x 10^{-05} Ns/m^2)
b_i - width of the cavity (1.8m)
Z_i - pressure loss factors of the cavity inlet and outlet given by the following equations:

$$Z_{inl} = \left(\left(\frac{A_{s,i}}{0.6 \; x \; A_{eq,inl,i}}\right) - 1\right)^2 \qquad (11)$$

$$Z_{out} = \left(\left(\frac{A_{s,i}}{0.6 \; x \; A_{eq,out,i}}\right) - 1\right)^2 \qquad (12)$$

Where
$A_{s,i}$ - the cross sectional area of the cavity (m^2)
$A_{eq,int,i}$ - equivalent inlet opening area of cavity (m^2)
$A_{eq,out,i}$ - equivalent outlet opening area of cavity (m^2)

Solving for Z_{inl} and Z_{out} first:

$A_{s,I}$ = 1.8m x 1m = 1.8m^2.
This is based on a unitary length of the cavity of 1m and the width of the cavity being 1.8m.

$A_{eq,int,i}$ = 1.8m^2
The bottom opening to the cavity is the same as the main cavity section

$A_{eq,out,I}$ = 0.2 x 1 = 0.2m^2. The opening at the top of the cavity is around 285mm but is interrupted by mullions at 1800mm centres. A constant 200mm slot width at the top of the cavity has therefore been assumed.

Therefore, $Z_{inl} = \left(\left(\frac{1.8}{0.6\,x\,1.8}\right) - 1\right)^2$

$Z_{inl} = 0.444$

And, $Z_{out} = \left(\left(\frac{1.8}{0.6\,x\,0.2}\right) - 1\right)^2$

$Z_{out} = 196$

Continuing to solve the for the three pressure components:

$$\Delta P_{T,i} = \Delta P_{B,i} + \Delta P_{HP,i} + \Delta P_{Z,i}$$

$$\Delta P_{T,i} = \left(\frac{1.2}{2}V_i^2\right) + \left(12\,x\,1.872E-05\,x\left(\frac{5}{1}\right)x\,V_i\right) + \left(\frac{1.2}{2}x\,V_i^2\,x\,(0.444+196)\right)$$

$$\Delta P_{T,i} = (118.5V_i^2) + (1.123E-03V_i)$$

Merging the two equations we have for $\Delta P_{T,I}$ gives

$$19{,}697.5\,x\,\frac{T_{gap,i} - T_{gap,o}}{T_{gap,i}\,x\,T_{gap,o}} = (118.5V_i^2) + (1.123E-03V_i)$$

Using the values for V_i and T_{amb} used in Method 1 previously of 0.0789m/s and 300K respectively gives:

$$19{,}697.5\,x\,\frac{T_{gap,i} - 300}{T_{gap,i}\,x\,300} = \left(118.5\,x\,0.0789^2\right) + (1.123E-03\,x\,0.0789)$$

$$\frac{T_{gap,i} - 300}{T_{gap,i}\,x\,300} = (3.74E-05)$$

$$T_{gap,i} - 300 = \left(0.0112\,x\;\;T_{gap,i}\right)$$

Finally,

$$T_{gap,i} = 303.4K$$

This suggests a 3.4K difference between inlet and outlet temperatures would be required to drive the flows required to dissipate the convective heat calculated in Method 1.

This correlates with the values obtained within Method 1 using the peak convective gains where a mean cavity temperature of 3K above ambient was estimated. This suggests that given the physical constraints within the cavity, the air flows predicted are capable of accepting the convective loads assuming temperature driven flows only – that is heat will not get "trapped" within the cavity.

It should be noted that this is a "worst case" scenario, and predicts that during peak solar load on the SE façade and utilising temperature driven flow calculations only (i.e no wind driven flow) the maximum cavity temperature will be around 3K warmer than the ambient external temperatures.

DYNAMIC THERMAL MODELLING

To investigate and validate the desktop approach further, a simple three dimensional dynamic thermal model of the facade was constructed using Integrated Environmental Solutions' Virtual Environment software package version 6.2 which has passed the BESTEST validation test. The Brisbane 1986 weather file has been designated as the Test Reference Year (TRY) for Brisbane by the Commonwealth Scientific and Industrial Research Organisation (CSIRO) and used for this simulation exercise. The TRY is a typical year with no extreme or unusual temperature conditions. This was used as we are largely interested in temperature distribution and air flow rather than energy and as a comparative study the weather file is largely irrelevant.

This modelling was undertaken to provide a time based solution as well as providing some outputs which could then be used as boundary conditions for a computational fluid dynamics (CFD) study. An image of the model constructed is shown below in Figure 2. The outer and inner glazed elements are the same size and vertical glazed returns are included.

In the model the cavity was split into two horizontal sections and three vertical sections. The central vertical section is 1m wide and provides consistency with the desktop analysis and provides some isolation from any localised effects at the edges of the cavity.

Glazing parameters as outlined in Table 2 were applied to the glazing elements within the model. Internal conditions in the internal spaces were set to controlled to between 19 °C and 23 °C to represent the specified building comfort conditions.

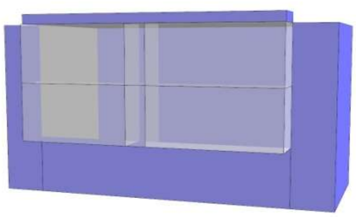

Figure 2 – Image of the simple IES model

As with the desktop analysis the effect of the framing and structural support for the glazing in the model has been ignored. Wind effects were isolated in the model such that the air flows predicted within the cavity were temperature driven only.

The SHGC for the system was determined from the model output by extracting the solar gain into the conditioned space and dividing this by the total incident solar radiation on the outer pane on the outer glazing at each time step.

The results for the SHGC from the model for the peak incident solar radiation on a South East facing facade (19 November, from the weather file) are shown in Figure 3 with the desktop calculation result included as a comparison.

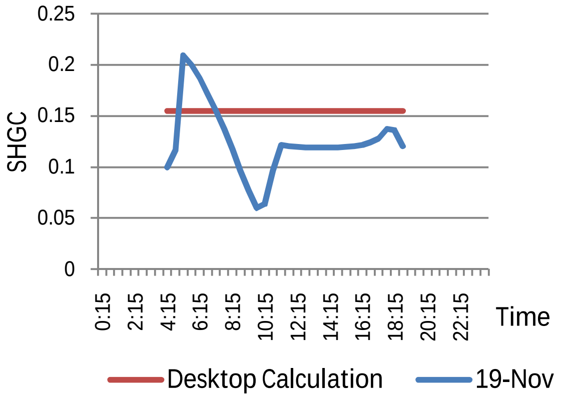

Figure 3 – Graph of SHGC against time for Peak SE incident Solar Radiation – 19 November

The results show higher SHGC figures than that calculated earlier in the day. This is believed to be a result of high levels of direct solar radiation on the glazed ends or returns of the construction which is directed into the conditioned space. Later in the day the solar angle is such that the incident solar radiation is approaching 900 or normal to the glazed returns and has much less of an effect. A general reduction in the SHGC is also observed as the incident solar angle increases.

At peak solar radiation on the external facade, between 7am – 8am, there is good correlation between the model calculated SHGC and that from the desktop analysis.

Reviewing the air movement within the cavity from the model yielded results at the peak solar incident radiation as shown in Figure 4. The image shows flow into the cavity as blue arrows with red arrows representing flow out.

The outflow of air through the top of the cavity at 0.1368m3/s compares to the desktop calculation estimate of 0.122m3/s. It should be noted however that these results are based on slightly different solar incident flux, with the desktop analysis using CIBSE figures and the model using the TRY weather file.

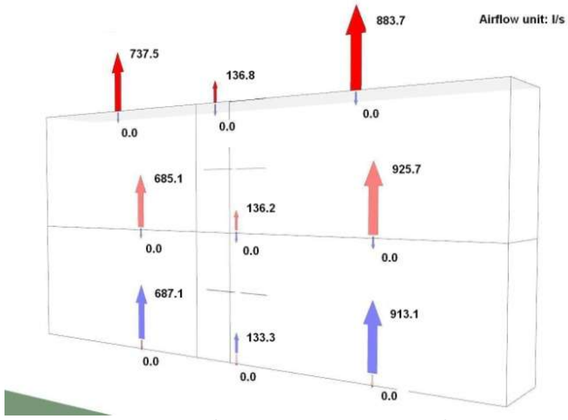

Figure 4 – Image of predicted cavity air flows at peak SE incident solar radiation (19 November)

The temperatures within the cavity spaces are shown in Figure 5. The biggest differential to ambient occurs in the early morning as expected. The maximum temperature differential predicted by the model is 3.1oC. This compares reasonably with the estimated temperature differences in the desktop validation calculations where a temperature difference between 3.0 °C and 3.4 °C was estimated.

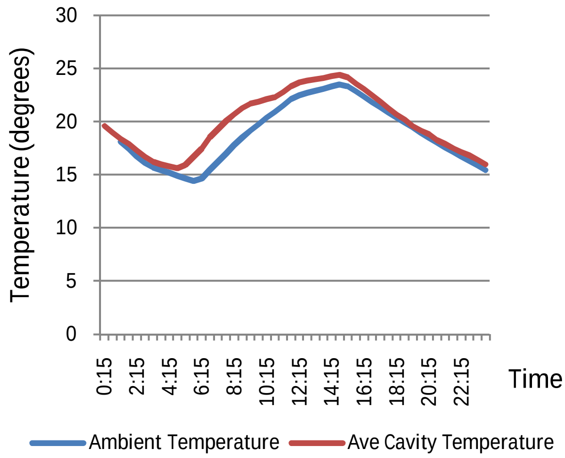

Figure 5 – Graph of average internal cavity temperature and ambient temperature against time (19 November)

Re-visiting Equation 5 the relationship established from the cavity convection analysis using ISO 15099:2003 section 7.4.2.3.

$$0.426 = \emptyset_{vi,I} \text{ x } (T_{gap,i} - T_{gap,o})$$

This relationship holds with the model outputs.

To further check the sensitivity of the calculated SHGC, analysis of the model results was investigated at other times across the year, other than under peak conditions and different orientations. Selections of these daily plots are shown in Figures 6 and 7.

These graphs show similarities with the previous plot for 19 November. The early morning peak previously described varies in magnitude across the year, peaking in summer due to the solar angle. During the peak incident solar gain on a SE surface the SHGC is reasonably consistent at around 0.15. Later in the day the SHGC trends at around 0.12 consistently for all days analysed.

As expected this graph is essentially the inverse of the SE exposure. Similarly the maximum SHGC values which now peak in the afternoon are at a maximum in the winter and minimum in the summer.

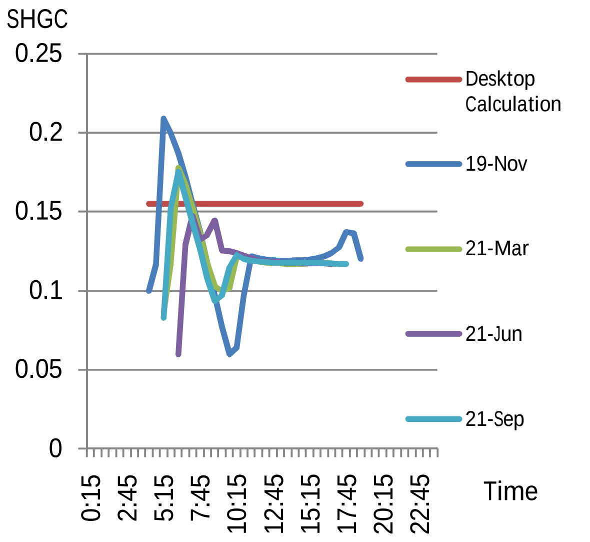

Figure 6 – Daily graphs SHGC SE Exposure – dates as shown

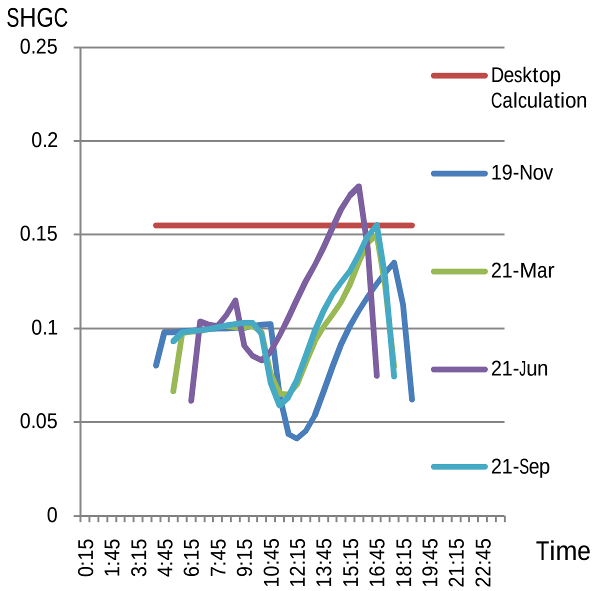

Figure 7 – Daily graphs SHGC NW Exposure – Peak Incident Solar Radiation

COMPUTATIONAL FLUID DYNAMICS (CFD) MODELLING

The analytical method for evaluating the SHGC value for a ventilated facade presented in this paper is fundamentally reliant on the absorbed energy of the outer pane of glazing being convected out of the cavity and not transferred into the internal space. To verify that this claim is valid a 2-dimensional CFD model was created to examine the air movement in the cavity. This used the CFD-ACE software package.

The 2D model created is a simplified representation of the geometry. Note that an idealised model was created, free of complex features such as structural framing and glass joints. This reflects the simplification that is by inherent in the analytical solution .

The glass walls of the cavity were set as adiabatic surfaces with a fixed temperature of 38°C and 28°C for the outer and inner panes, respectively. These values were obtained from the thermal model. The CFD model did not include any radiative heat gains as these are assumed to pass through the space with little additional heat gain to the air in the cavity.

The openings to the cavity at the top and bottom were set as passive boundaries without any form of forced convection and the same ambient pressure and temperature of 1atm and 16.5°C. This temperature is the average ambient temperature observed from the thermal model between the hours of 7am-8am on Nov 19, coincident with the glass surface temperatures described above.

The results of the simulation demonstrate an average velocity across the exhaust of 0.58m/s and a peak of 0.75m/s. This average velocity correlates to an average exhaust volumetric flow rates of 0.173m3/s. These values are illustrated in Figure 8.

The average temperature predicted in the cavity is 17.3°C with an exit temperature peaking at approximately 21°C. Notably in the temperature field shown in **Error! Reference source not found.** the air is stratified with the top one-third of the cavity having an average temperature of approximately 18.5°C. As the air temperature are still significantly lower than the inner pane glazing temperature of 28°C, there is no net conduction of heat into the space from the air cavity at the time of peak solar load.

The error margin of this result compared to that derived from the analytical solution presented in Method 1 is not a significant issue. The main outcome from the CFD analysis is the validation of the premise that air is exhausted from the cavity due to the natural convection driven by the stack effect.

This result validates the approach for excluding the quantity of absorbed solar energy in the outer pane of glazing in the determination of the SHGC for a double skin ventilated facade.

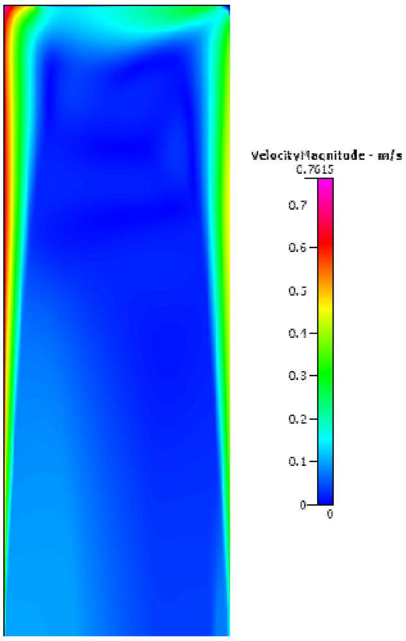

Figure 8 – Velocity profile in the ventilated cavity.

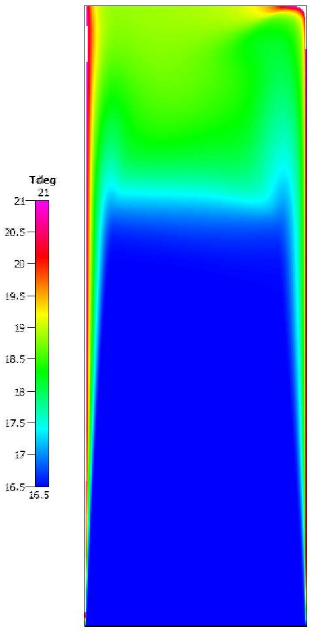

Figure 9 – Temperature profile in the ventilated cavity.

CONCLUSIONS

A desktop method for determining solar heat gain coefficient for a double skinned facade utilising first principles has been performed based upon the glazing properties of each glazing skin.

The assumption that the cavity is well ventilated and conduction gains are dissipated through natural ventilation and the stack effect in the cavity, have been tested through calculation. The results from these calculations suggest that the assumption is valid and that the stack effect within the cavity is not constrained.

Dynamic thermal modelling has been undertaken to verify the performance of the double skinned facade. This has a good level of correlation with the desktop analysis, both in terms of SHGC, predicted air flows and cavity temperatures. The effects of the glazed returns, not considered in the desktop analysis, do have an impact and are believed to be the cause of discrepancies observed. This is particularly evident in the early morning SHGC peaks observed for the actual south eastern exposure scenario.

Computational fluid dynamics modelling demonstrated that the air in the cavity is exhausted at sufficient volumetric flow rates to validate the approach for the exclusion of the absorbed solar energy in the outer pane of glazing.

Despite the increase in cavity temperatures this type of approach may be suitable for determining the properties required to support a Section J Deemed to Satisfy analysis subject to a check of U-value sensitivity.

The results show that a double cavity facade significantly improves SHGC performance when the cavity is well ventilated.

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