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# Energy performance characterisation of vented opaque envelope through simplified methodologies

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## Abstract

Opaque vented façades are innovative and widely-used technological systems adopted both in new constructions and in building renovations. According to European Directive 2010/31/EU (EPBD recast) on the energy performance of buildings, each Member State should give priority to passive cooling techniques in order to enhance building performance during the summer period. For this purpose opaque vented envelope could be an appropriate technological solution to reduce the summer peak loads and the energy consumption. Although the EPBD recast has submitted the development of calculation methods for the energy performance evaluation to the European Committee of Standardisation (CEN), there is a lack in European Standards on the calculation of non-conventional building envelope performance, including vented façades.

The object of the present work is the thermal performance characterization of vented vertical opaque enclosures in real conditions of use, through simplified parameters.

Starting from EN ISO 6946 and EN ISO 13786, new equivalent thermal parameters are defined, such as the equivalent steady state thermal transmittance, the equivalent periodical thermal transmittance and the time shift.

Equivalent parameters are obtained by evaluating surface inside face conduction in the opaque components, under stabilized periodic external conditions, for the summer design day. An equivalent outside temperature is used, which considers both the convective and the radiative thermal exchanges (solar and infrared waves), for different boundary conditions (orientation). The tool used for calculations is based on the conduction transfer function – CTF – method, as implemented in the thermal dynamic simulation program *Energy Plus*.

Through a sensitivity analysis, different opaque enclosures are analyzed, varying the design parameters such as the thickness, the height and the length of the vented cavity.

## 1. Introduction

Opaque vented façades are largely used both for existing buildings' renovations and for new buildings to improve the thermal performance of the envelope and the architectural design quality of the external skin.

An opaque vented façade is a double skin façade made up of two opaque building elements separated by an air gap. The outer component (baffle) is generally a thin layer attached to the load bearing wall by specific mechanical systems. The inner component is the wall itself, traditionally composed of a massive layer (brick, concrete etc.) coated by a thermal insulation layer. Through the gap a natural air flow is created through specific openings by means of the combined effect of the wind forces and the stack effect.

In summer period the advantages of an opaque vented façade are related to the reduction of the thermal load due to direct solar radiation by means of the shading effect of the baffle and of the natural convection inside the air gap.

In order to calculate the opaque vented wall performance, several works focus on CFD analysis (Sanjuan et al. 2011, Patania et al. 2010) while others apply a zonal approach (Marinosci et al. 2011, Chan et al. 2009) which is simpler than the CFD approach but quite precise. Both the numerical models are validated through experimental data (Peci López et al. 2012, Giacola et al. 2012, Sanjuan et al. 2011).

The detailed evaluation of the vented façade thermal performance is quite complex and requires a complete thermofluid-dynamic analysis of the vented air gap, an accurate knowledge of heat transfer coefficients and the knowledge of each input parameter affecting the results. On the other hand simple calculation methods can be applied.

Despite the growing interest in this technological solution and the correlated scientific research based on detailed calculation methods, only a few studies (Balocco 2002, Ciampi et al. 2003) refer to simplified methods, which enable the estimation of the vented façade performances in an easy but rigorous way.

In this paper a simplified calculation method is presented in order to provide equivalent dynamic thermal parameters (periodic thermal transmittance, time shift) for different vented façade configurations.

The use of these thermal parameters can be a useful simple tool for designers and industries to evaluate the performance of this technology.

## 2. Case study

In order to evaluate the thermal performance of the opaque vented solution, a test-room has been considered (Figure 1).

The test room is surrounded by opaque adiabatic components except for the analyzed vented façade. No window has been considered.

The layers constituting the adiabatic components have been chosen according to EN ISO 13791 while their thermophysical properties have been adopted according to UNI 10351.

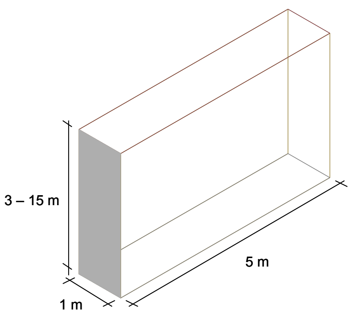


Fig. 1 – 3D model of the test-room analyzed. In grey colour the vented façade

| | Layers | | $s$ | $\rho$ | $c$ | $\lambda$ |
|---|---|---|---|---|---|---|
| | (ext-int) | | cm | kg/m³ | J/(kg K) | W/(m K) |
| INTERNAL WALL | I | Gypsum plasterboard | 1,2 | 900 | 880 | 0,21 |
| | II | Thermal insulation | 10 | 30 | 840 | 0,04 |
| | III | Gypsum plasterboard | 1,2 | 900 | 880 | 0,21 |
| FLOOR | I | Waterproofing | 0,4 | 1500 | 1500 | 0,23 |
| | II | Concrete | 6 | 2000 | 880 | 1,40 |
| | III | Thermal insulation | 4 | 50 | 840 | 0,04 |
| | IV | Concrete | 18 | 2400 | 880 | 2,10 |
| | V | Thermal insulation | 10 | 50 | 840 | 0,04 |
| | VI | Acoustic underlay | 2 | 400 | 880 | 0,06 |
| ROOF | I | Waterproofing | 0,4 | 1500 | 1300 | 0,23 |
| | II | Thermal insulation | 8 | 50 | 840 | 0,04 |
| | III | Concrete | 20 | 2400 | 880 | 2,10 |

Table 1 – Thermo physical characteristics of adiabatic components

For each simulation, the thermophysical properties (thermal conductivity, thickness, density and specific heat) of the external vented massive layer as well as the case study height (3 and 15 m respectively) have been changed, while the adiabatic components have been set as constant. See Table 1.

Moreover the test-cell has been considered South, North, East and West oriented in order to evaluate the influence of the orientation on the energy performance of the vented façade, while the indoor air temperature is maintained constant at 26 °C.

### 2.1 Naturally vented wall

The analysed wall is a naturally vented wall. It is composed of a massive layer (dotted in Figure 2), a thermal insulation layer, a naturally vented cavity and a baffle.

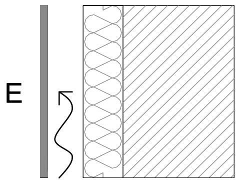


Fig. 2 – Vented façade layers.

In order to evaluate the influence of the main thermophysical properties of the massive layer on the dynamic thermal parameters of the vented wall, a sensitivity analysis has been carried out varying the thermal conductivity ($\lambda$), the thickness ($s$), the density ($\rho$) and the specific heat ($c$).

The four parameters have been varied simultaneously within specific range values according to a random analysis as implemented in *SimLab 2.2*. A hundred solutions have been chosen. Table 2 shows the maximum and minimum values of the thermal parameters range.

| | | Min | Max |
|---|---|---|---|
| $s$ | [m] | 0,10 | 0,50 |
| $\lambda$ | [W/(m K)] | 0,15 | 2,00 |
| $\rho$ | [kg/m³] | 400 | 2400 |
| $c$ | [J/(kg K)] | 840 | 2700 |

Table 2 – Minimum and maximum values of thermo-physical characteristic of the massive layer

Surfaces properties influencing convective and radiative heat transfer have been chosen as constant values.

Concerning the thermal insulation layer, fixed thermophysical properties have been chosen: $\lambda$ = 0,04 W/(m K); $\rho$ = 30 kg/m$^3$; $c$ = 840 J/(kg K) with a constant thickness of 0,08 m.

The thermal transmittance of the wall is calculated according to EN ISO 6946 for each configuration and varies from 0,22 to 0,43 W/(m$^2$K).

It is important to point out that the thermophysical properties of the massive layer have been chosen to consider most of the existing building material (wood, concrete, brick etc.). Moreover, the range of variation of vented façade thermal transmittance as well as the range of its dynamic thermal properties have been chosen with respect to the national current limit values.

In order to evaluate the influence of the geometrical characteristic of the air cavity on the vented façade performance, three thicknesses of the air gap have been considered: 5 – 10 – 15 cm. The ventilation openings at the top and at the bottom of the wall are considered to be of the same length and depth of the baffle and of the air cavity respectively.

## 3. Calculation methods

The *EnergyPlus* dynamic simulation tool has been used to calculate the conductive heat flux through the inner surface of the vented wall by means of the conduction transfer function calculation method.

### 3.1 Exterior naturally vented cavity

The opaque vented envelope is a traditional opaque wall whose outer layer consists of a thin and only resistive coat (baffle) separated from the load bearing wall by a vented air cavity.

As the baffle is sufficiently thin and highly conductive, it is possible to consider a single temperature for both sides and along its area. Moreover, the baffle is opaque to shortwave and longwave radiation and it completely covers the underlying layers avoiding solar energy to reach the underlying layers. The baffle is a continuous surface: the natural ventilation of the gap only depends on the openings at the top and at the bottom of the cavity.

The baffle temperature is calculated through the heat balance equation in the baffle surface's control volume (see Figure 3) as in equation

$$\theta_{s,baff} = \frac{\left(I\alpha + h_{\mathrm{cv}}\theta_{\mathrm{ae}} + h_{\mathrm{r,air}}\theta_{\mathrm{ae}} + h_{\mathrm{r,sk}}\theta_{\mathrm{sk}} + h_{\mathrm{r,gr}}\theta_{\mathrm{gr}} + h_{\mathrm{r,cav}}\theta_{\mathrm{se}} + h_{\mathrm{cv,cav}}\theta_{\mathrm{cav}}\right)}{\left(h_{\mathrm{cv}} + h_{\mathrm{r,air}} + h_{\mathrm{r,sk}} + h_{\mathrm{r,gr}} + h_{\mathrm{r,cav}} + h_{\mathrm{cv,cav}}\right)} \quad (1)$$

where $I$ is the solar irradiance reaching the outer side of the baffle; $h_{\mathrm{cv}}\theta_{\mathrm{ae}}$ and $h_{\mathrm{cv,cav}}\theta_{\mathrm{cav}}$ are the convective heat exchanges of the baffle with the external environment and the air cavity respectively; $h_{\mathrm{r,air}}\theta_{\mathrm{air}}$, $h_{\mathrm{r,sk}}\theta_{\mathrm{sk}}$, $h_{\mathrm{r,gr}}\theta_{\mathrm{gr}}$, $h_{\mathrm{r,cav}}\theta_{\mathrm{se}}$ are the radiative heat exchanges between the baffle and the external air, the sky, the ground and the underlying component surface respectively.

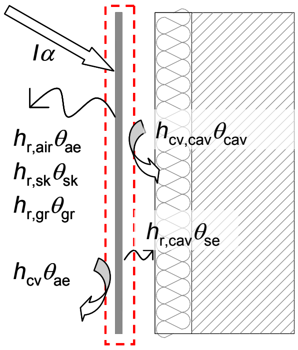


Fig. 3 – Baffle surface heat balance scheme

The volume of air located between the baffle and the underlying wall components is the cavity. It is possible to calculate a uniform air temperature of the air cavity through the heat balance equation (2) (see Figure 4).

$$\theta_{cav} = \frac{Ah_{cv,cav}\theta_{se} + Ah_{cv,cav}\theta_{s,baff} + \dot{m}c\theta_{ae}}{\left(Ah_{cv,cav} + Ah_{cv,cav} + \dot{m}c\right)} \qquad (2)$$

where $A$ is the surface surrounding the cavity involved in convective heat exchange. The heat balance equation takes into account the baffle ($Ah_{cv,cav}\theta_{s,baffle}$) and the outer surface of the massive wall ($Ah_{cv,cav}\theta_{se}$) convective heat exchange within the cavity, as well as the heat exchange due to the air mass flow from natural forces ($\dot{m}\,c\theta_{ae}$).

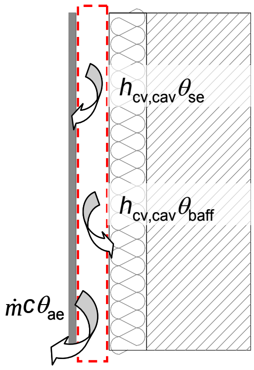


Fig. 4 – Cavity air heat balance scheme

In order to take into account natural ventilation air exchanges, the ASHRAE HOF (2009) model has been considered.

According to this model, the air mass flow from natural forces is calculated through equation

$$\dot{m} = \rho\dot{V}_{tot} \qquad (3)$$

where $\rho$ is the density of the air and $\dot{V}_{tot}$ is the total volumetric flow rate of air ventilating in and out of the cavity due to natural buoyancy and wind forces:

$$\dot{V}_{tot} = \dot{V}_{wind} + \dot{V}_{thermal} \qquad (4)$$

Wind forces are calculated as:

$$\dot{V}_{wind} = C_v A_{in} w \qquad (5)$$

where $C_V$ is the effectiveness of the openings that depends on opening geometry and opening orientation respect to the wind direction; $A_{in}$ is the half of the total area of the openings; $w$ is the local wind speed.

A typical range of $C_V$ values is 0,25 – 0,35 for diagonal wind and 0,5 – 0,6 for perpendicular wind.

Natural buoyancy phenomena are taken into account according to equation (6) or (7).

$$\dot{V}_{thermal} = C_D A_{in}\sqrt{2g\Delta H_{NPL}(\theta_{cav} - \theta_{ae})/\theta_{cav}} \qquad (6)$$

if $\theta_{cav} > \theta_{ae}$

$$\dot{V}_{thermal} = C_D A_{in}\sqrt{2g\Delta H_{NPL}(\theta_{ae} - \theta_{cav})/\theta_{ae}} \qquad (7)$$

if $\theta_{ae} > \theta_{cav}$ and baffle is vertical.

where $C_D$ is the discharge coefficient for the opening and it depends on opening geometry; $g$ is the gravitational constant; $\Delta H_{NPL}$ is the height from the midpoint of the lower opening to the Neutral Pressure Level and is equal to ¼ of the height of the component (in case of vertical component) or ¼ sen$\beta$ where $\beta$ is the component tilt.

ASHRAE HOF provides a typical range of $C_D$ values varying from 0 to 1,5 and a fixed value (0,65) for unidirectional air flow rate.

In order to investigate the influence of the vented opaque component design on its dynamic thermal performance, different geometrical characteristics of the air gap have been considered: three thicknesses of vented cavity (0,05 – 0,10 – 0,15 m) and two heights of the wall (3 – 15 m) as described in previous section.

In Table 3 the input parameters considered for the simulations are shown.

| Input data | Values | | |
|---|---|---|---|
| Height of the wall (m) | 3 | | 15 |
| Thermal emissivity of exterior baffle material $\varepsilon$ [-] | 0,9 | | |
| Solar absorptivity of exterior baffle $\alpha$ [-] | 0,6 | | |
| Height scale for buoyancy – driven ventilation $\Delta H_{NPL}$ [-] | 0,75 | | 3,75 |
| Effective thickness of cavity behind exterior baffle [m] | 0,05 | 0,10 | 0,15 |
| Roughness of exterior surface | Smooth | | |
| $C_V$ [-] | 0,25 | | |
| $C_D$ [-] | 0,65 | | |

Table 3 – Input parameters considered for the simulations

## 3.2 Equivalent dynamic thermal parameters

EN ISO 13786 has been considered. This technical standard is based on the admittance method introduced by N.O. Milbank and J. Harrington-Lynn (1974), and supplies a simplified calculation model that considers 24 h sinusoidal boundary conditions.

The main simplification of the model is due to the use of a sinusoidal trend of external temperature varying cyclically around a mean value (Baratieri et al. 2009).

In order to represent in a more realistic way the boundary conditions influencing the heat flow through a wall, an equivalent external temperature has been considered ($\theta_{e,eq}$) as in equation.

$$\theta_{e,eq} = \theta_{ae} + \frac{I\alpha + h_{r,gr}(\theta_{gr} - \theta_{ae}) + h_{r,sk}(\theta_{sk} - \theta_{ae})}{h_e} \quad (8)$$

The use of an equivalent external temperature allows us to take into account as driven forces not only the external temperature $\theta_{ae}$, but also the effects of the solar radiation $I\alpha$, the radiative heat exchange between the component and the ground $h_{r,gr}(\theta_{gr}-\theta_{ae})$ and between the component and the sky $h_{r,sk}(\theta_{sk}-\theta_{ae})$.

$h_e$ is the outdoor surface heat transfer coefficient, that includes the convection coefficient $h_{cv}$ and the radiative ones, between the component and the air $h_{r,air}$, the ground $h_{r,gr}$ and the sky $h_{r,sk}$ respectively

$$h_e = h_{cv} + h_{r,air} + h_{r,gr} + h_{r,sk} \quad (9)$$

In order to calculate the equivalent dynamic thermal properties of a naturally vented wall, a summer design day has been considered for the city of Turin.

In Table 4 the geographical data of the location and the climatic data of the summer design day are shown.

| | | | |
|---|---|---|---|
| PLACE | Location | Turin | |
| | Longitude | 45,08 | [°] |
| | Latitude | 7,68 | [°] |
| | Altitude | 239 | [m] |
| SUMMER DESIGN DAY | $\theta_{db,max}$ | 30,7 | [°C] |
| | $\Delta\theta_{ae}$ | 11 | [°C] |
| | $I_{m,North}$ | 79,8 | [W/m$^2$] |
| | $I_{m,South}$ | 150,0 | [W/m$^2$] |
| | $I_{m,East}$ | 177,3 | [W/m$^2$] |
| | $I_{m,West}$ | 211,5 | [W/m$^2$] |
| | Wind speed | 0,8 | [m/s] |

Table 4 – Geographical data and climatic data of summer design day for Turin

In the summer design day of Turin, the equivalent external temperature varies according to exposure as shown in Figure 5.

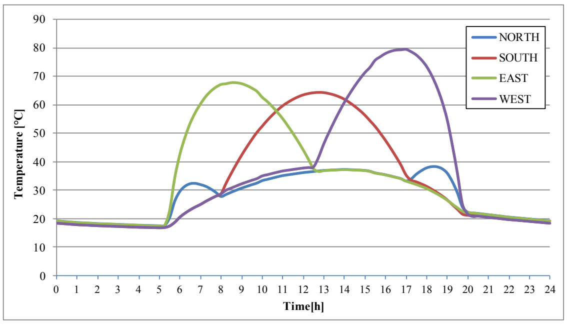


Fig. 5 – Profiles of external equivalent temperature on summer design day for different exposures.

The equivalent dynamic thermal properties taken into account are periodic thermal transmittance $Y_{ie}$ and the time shift $\varphi$.

According to the definition in EN ISO 13786, the equivalent periodic thermal transmittance has been defined as the ratio between the daily maximum $\Phi^{dyn}_{cd,si,max}$ and minimum $\Phi^{dyn}_{cd,si,min}$ opaque inner surface heat flux difference, and the outdoor detailed equivalent temperature maximum $\theta_{e,eq,max}$ and minimum $\theta_{e,eq,min}$ difference (Corrado and Paduos, 2009).

$$Y_{\mathrm{ie}} = \frac{\left(\Phi_{\mathrm{cd,max}} - \Phi_{\mathrm{cd,min}}\right)_{si}^{dyn,CTF}}{\left(\theta_{\mathrm{e,eq,max}} - \theta_{\mathrm{e,eq,min}}\right)} \qquad (10)$$

The conductive heat flux has been calculated through the *EnergyPlus* dynamic model using the conduction transfer function calculation heat balance algorithm.

The equivalent time shift is defined as the delay between the daily maximum conductive heat flux value on the inner surface $\Phi_{cd,si,\max}^{dyn}$ and its correspondent heat flux maximum value not considering components thermal inertia $\Phi_{cd,si,\max}^{st}$, corresponding to the maximum of the external equivalent temperature.

Time shift has been calculate as in equation (11)

$$\begin{cases} t_{\phi cd,si,\max}^{dyn} > t_{\phi cd,si,\max}^{st} \Rightarrow t_{\phi cd,si,\max}^{dyn} - t_{\phi cd,si,\max}^{st} \\ t_{\phi cd,si,\max}^{dyn} \le t_{\phi cd,si,\max}^{st} \Rightarrow t_{\phi cd,si,\max}^{dyn} - t_{\phi cd,si,\max}^{st} + 24 \end{cases} \qquad (11)$$

where $t_{\phi cd,si,\max}^{dyn}$ is the hour of the design day in which the maximum conductive heat flux occurs; $t_{\phi cd,si,\max}^{st}$ is the time of the design day at which the maximum conductive heat flux occurs through the same wall neglecting its thermal inertia, that is the time of the maximum external equivalent temperature.

The equivalent time shift would be the same as the EN ISO 13786 time shift only if the external equivalent temperature profile were a sine curve with a period of 24 hours.

In order to obtain the equivalent time shift, the conductive heat flux has been calculated through the *EnergyPlus* dynamic simulation tool twice: the first time the thermal inertia of opaque components has been considered while the second time it has been neglected.

In Figure 6 the difference between the conductive heat flux profile is shown by considering (_MASS) or not (_NO MASS) the thermal inertia of the wall, for a light solution corresponding to the lowest $\zeta$ value.

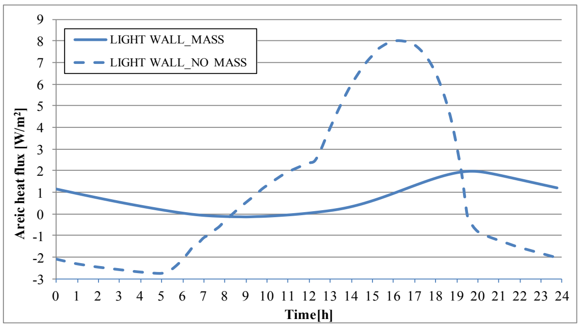


Fig. 6 – Conductive heat flux trend: comparison between light and heavy walls.

## 4. Results

### 4.1 Dynamic thermal parameters

The dynamic equivalent thermal parameters are represented versus $\zeta$. EN ISO 13786 introduces $\zeta$ as the parameter representing the ratio between the thickness of the considered layer and its penetration depth $\delta$. The penetration depth is a function of the thermal diffusivity $a$ related to the considered time period $T$ :

$$\zeta = \frac{d}{\delta} = \frac{d}{\sqrt{a\frac{T}{\pi}}} = \frac{d}{\sqrt{\frac{\lambda \cdot T}{\rho \cdot c \cdot \pi}}} \qquad (12)$$

Applying equation (12) to the massive layer of the opaque envelope technical solutions derived from the random analysis, the corresponding $\zeta$ values lies within the range 0,64 to 10,49. Higher $\zeta$ values correspond to higher thermal inertia of the technical solutions.

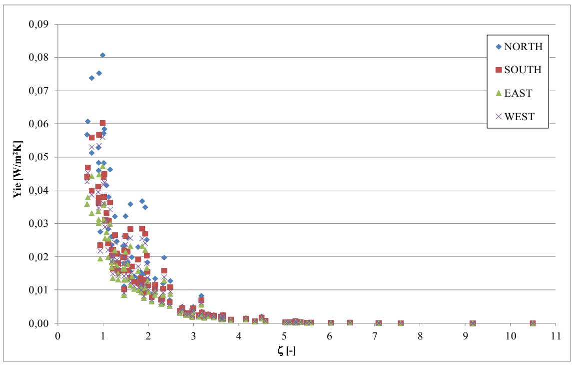


Fig. 7 – Equivalent periodic thermal transmittance versus $\zeta$ for different exposures (one storey wall with 0.05m vented cavity)

The results show the periodic thermal transmittance exponentially decreases for increasing values of $\zeta$. The light solutions give the highest $Y_{ie}$ values, for heavy solutions $Y_{ie}$ tends to zero; the dynamic thermal performance could be generally considered very good independently from $\zeta$ (values lower than 0,08 W/(m²K)).

By increasing the thickness of the vented cavity from 0.05 m to 0.15 m, both the one storey and the five storey wall do not get significant deviations from the results shown in Figure 7.

The influence of the exposure on the periodic thermal transmittance is noticeable for low values of $\zeta$. By considering the same solution ($\zeta$ = 0,9) for different exposures, the north side gives the highest $Y_{ie}$ value (around 0,08 W/(m²K)) while the east side the lowest one (around 0,04 W/(m²K)).

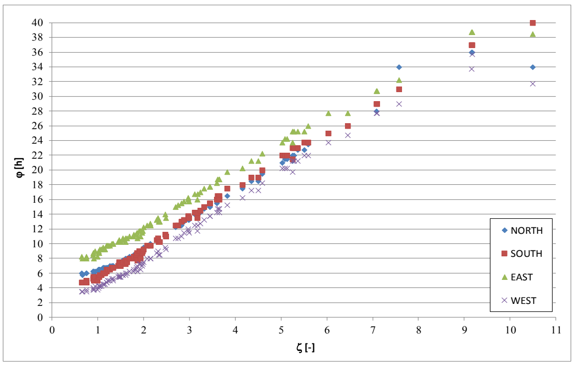


Fig. 8 – Time lag versus $\zeta$ for different exposures (one storey wall with 0.05m vented cavity).

The analysis shows that time shift is an increasing function of $\zeta$ and the trend is linear: heavy solutions obtain highest $\varphi$ values, for light solutions $\varphi$ tends to zero; for values of $\zeta$ higher than 5,5, the time shift exceeds 24 hours.

As for the periodic thermal transmittance, by increasing the thickness of the vented cavity from 0.05 m to 0.15 m, both the one storey and the five storey wall do not show significant deviations from the results shown in Figure 8.

Despite the periodic thermal transmittance, the exposure influence on the time shift is noticeable both for heavy and light solutions: for increasing $\zeta$ the time shift deviation among exposures is maintained constant. North and south exposures obtains similar results; east side obtains the highest values of time shift because of the external equivalent temperature trend: despite from its high value, the peak is relevant during the early hours of the morning, when the wall is discharged because of the night thermal exchange; exactly the opposite reasoning could be argued for the west exposure.

From what has been observed, it is possible to conclude that ventilation reduces the conductive heat flux entering the opaque component.

## *4.2* Equivalent thermal transmittance

Rather than in terms of periodic thermal transmittance, the variation of the conductive heat flux can be better performed introducing a parameter defined as the "equivalent thermal transmittance" $U_{eq}$ representing the ratio between the summer design day conductive heat flux mean value and the average temperatures difference between the internal and external environments in the same design day:

$$U_{eq} = \frac{\overline{\Phi}_{cd,si}}{\overline{\theta}_{e,eq} - \theta_{ai}} \tag{13}$$

Therefore, the equivalent thermal transmittance considers the effect of the vented cavity on the stationary conductive heat flux : as being evaluated in daily average conditions, $U_{eq}$ is not influenced by the thermal inertia of the component.

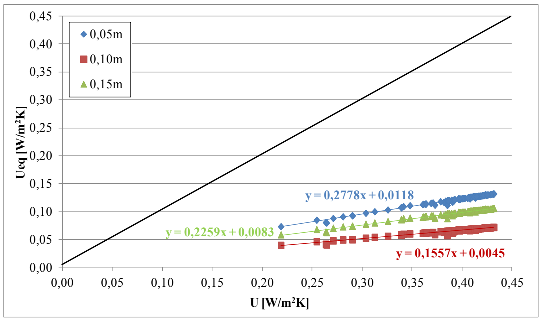


Fig. 9 – Equivalent versus theoretical thermal transmittance for West oriented 5 storey opaque vented wall, and different cavity thickness

Figure 9 shows the relation between the thermal transmittance $U$ evaluated in steady state conditions – according to EN ISO 6946 - and the equivalent thermal transmittance $U_{eq}$ defined in dynamic conditions. The introduction of a vented cavity deeply reduce the conductive heat flux and consequently $U_{eq}$.

Increasing the thermal conductance, the ratio between the steady state and the equivalent thermal transmittance shows a rising linear function and the deviation depends on the thickness of the vented cavity: by considering medium resistive solutions, for each exposition the 0.05 m cavity reduces the thermal transmittance of about 3.5 times, the 0.10 m cavity 6.5 times and the 0.15 m cavity around 4.5 times.

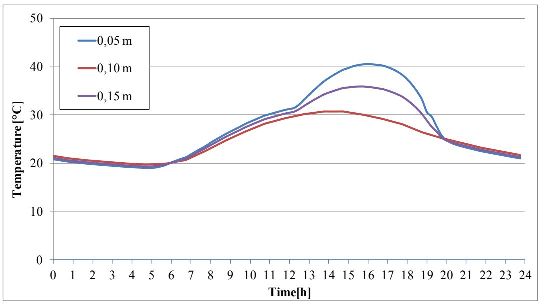


Fig. 10 – Summer design day trends of the cavity temperature for a West oriented 5 storey wall, for 0.05 m, 0.10 m and 0.15m thickness of the cavity

Solutions characterized by 0,15 m thickness of the vented cavity present $U_{eq}$ values higher than the 0, 10 m ones; Figure 10 shows results depending on the temperatures within the vented cavity: increasing the cavity thickness from 0.10 m to 0.15 m the conductive heat flux referred to solutions with the same resistive and massive thermal characteristics increases too, because of the cavity overheating. That means the vented cavity of 0.10 m thickness shows better thermal performances.

The same considerations could be made for each orientation, cavity thickness and number of storey considered.

## 5. Conclusion

The present work introduces a methodology to evaluate the thermal performances of opaque vented solutions. A case study has been evaluated for different heights of the component, vented cavity thickness and exposures. The thermal performance has been evaluated in dynamic conditions, referring to the Turin summer design day. Equivalent thermal parameters have been then calculated for the case study, like periodic thermal transmittance, time shift and thermal transmittance.

The results show the equivalent periodic thermal transmittance is a decreasing exponential function of $\zeta$, that represents the ratio of the thickness of the massive layer to the penetration depth. For really massive solutions $Y_{ie}$ tends to zero. The vented cavity thickness and the boundary conditions only influence light solutions, but deviations are not significant.

The results also show the time shift is an increasing linear function of $\zeta$; for really massive solutions $\varphi$ exceed the 24 hours. The time shift does not seem to be significantly influenced by the thickness of the cavity or the height of the panel. Indeed, the same technical solutions with different exposures show deviations that remain constant with increasing $\zeta$ values and depend on the joint between the thermal inertia of the component and the daily trend of the equivalent external temperature.

The introduction of a vented cavity deeply influences the conductive heat flux; the parameter that better represents the phenomenon is the equivalent thermal transmittance.

The proposed methodology could be a valid tool for industries and designers to easily perform opaque vented innovative technologies for different boundary conditions.

## 6. Nomenclature

### Symbols

| | |
|---|---|
| $a$ | thermal diffusivity (m²/s) |
| $A$ | surface (m²) |
| $c$ | specific heat (J/(kg K)) |
| $g$ | gravitational constant (= 9,81 m/s²) |
| $h$ | surface heat transfer coefficient (W/(m²K)) |
| $I$ | solar irradiance (W/m²) |
| $\dot{m}$ | air mass flow (kg/s) |
| $R$ | thermal resistance ((m²K)/W) |
| $s$ | thickness (m) |
| $U$ | thermal transmittance (W/(m²K)) |
| $\dot{V}$ | volumetric flow rate (m³/s) |
| $Y_{ie}$ | periodic thermal transmittance (W/(m²K)) |
| $\alpha$ | solar absorptivity (-) |
| $\delta$ | periodic penetration depth of a heat wave in a material (m) |
| $\Phi$ | heat flux (W) |
| $\varphi$ | time shift (h) |
| $\lambda$ | thermal conductivity (W/(m K)) |

| | |
|---|---|
| θ | temperature (°C) |
| ρ | density (kg/m$^3$) |
| ζ | ratio of the thickness of the layer to the penetration depth |

## Subscripts/Superscripts

| | |
|---|---|
| ae | external air |
| ai | internal air |
| baff | baffle |
| cav | cavity |
| cd | conduction |
| cv | convection |
| dyn | dynamic |
| e | external |
| eq | equivalent |
| gr | ground |
| i | internal |
| in | inlet |
| max | maximum |
| min | minimum |
| r | radiative |
| s | surface |
| sk | sky |

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