Analysis of the energy performance of sunspaces: a new method

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Abstract

Environmental sustainability is an important issue concerning most of all building design and construction. An efficient system to decrease energy requirements during the heating season is the use of passive solar devices, among them integrated sunspaces.

The research presented in this paper deals with:

1. the comparison of different methods used to calculate energy behaviour of sunspaces, in particular their contribution to living space heating;

2. the development of these methods in order to achieve results closer to real situation.

This paper marks an intermediate stage in a work that is still in progress. The main goal is to develop a theoretical system and to find the parameters to be controlled in order to compare the energy performance of real sunspaces with the one of scale models. In this way, it is possible to build scale models (less expensive and easily to manage) and to verify the design assumption with in field preliminary tests. In order to get scale models into proportion and to manage correctly the results of the monitoring campaign, the changing of the physical quantities concerning the sunspace thermal behaviour has been theoretically studied while the model linear dimensions are reduced. Particular attention is given to radiative heat exchange among internal and external surfaces. Finally, the comparison of the results are presented and discussed.

1. Sunspace as a passive solar system

Increasing the exploitation of solar energy is an effective strategy to decrease heating requirements of buildings. According to Edward Mazria, the passive solar systems differ from the active solar ones because based not on mechanical devices but only on natural phenomena and presenting a stronger integration in the buildings' shape [1]. Sunspace is a particular typology of solar passive system. The solar radiation gets into the glazed room and heats its surfaces, which emit longwave radiation. Glass is much more transparent to solar radiation than to longwave radiation, so the larger portion of thermal radiation is trapped in the sunspace. At the same time heat is also transferred from inner surfaces of the sunspace to air, which is confined by the glazed surfaces. Therefore, during winter sunny days inner temperature can be much higher than outer one (greenhouse effect). Heat can be transferred from the sunspace to inner spaces by means of ventilation (natural or mechanical) or through the partition wall. When solar radiation is low or absent, the sunspace helps to decrease heat losses from the building.

Moreover, building a sunspace is not only an energetic issue. Sunspaces often increase the architectonic value of a building and realize a room where human comfort is guaranteed even in wintertime, not only by a thermal point of view but considering also the possibility to live an “outer” space with the same temperature of the inner ones.

2. Open questions

The availability of evaluation methods for the energy behaviour of sunspaces is of primary importance in order to make smart choices regarding their design and management. Unfortunately, methods used nowadays are too simplified or not enough developed to give suitable realistic indications to designers and to users, laying often to the construction of sunspaces where comfort conditions and energy requirements are not completely satisfied.

3. Analysis and development of tools concerning sunspace design

3.1 Mathematical models and computer software

A good prediction of sunspaces behaviour needs proper mathematical models. Wall (1996) [2] differentiates between two design phases: “At an early stage of design, the need is for design tools which are easy to use and yield results quickly. (..) Orders of magnitude are sufficient to assess the effect of different measures and the difference between configuration. (..) For the design of e.g. heating and cooling systems at a later stage, a dynamic calculation program may be needed” (p. 57). A goal of this research is to create user friendly tools (graphs, tables, indications) starting from results of advanced methods and from empirical data, useful to designers in the early stage of the design phase. Wall (1996) highlights that some software are suitable for dealing with buildings having windows of normal size but they are not suitable to deal with rooms having large glazed surfaces, in particular because in the second case the thermal radiative exchange is much more complicated. To very tall sunspaces it could be important modelling air temperature stratification as well.

3.1.1 Method 5000 and ANNEX E of the standard UNI EN ISO 13790:2008

Method 5000 [3] and the method presented in the ANNEX E of UNI EN ISO 13790:2008 [4] are based on quasi-stationary models (losses are obtained by a stationary calculation and solar and inner heat gains are reduced by an utilization factor that considers thermal inertia of building). Advantage of sunspace are calculated with a monthly procedure.

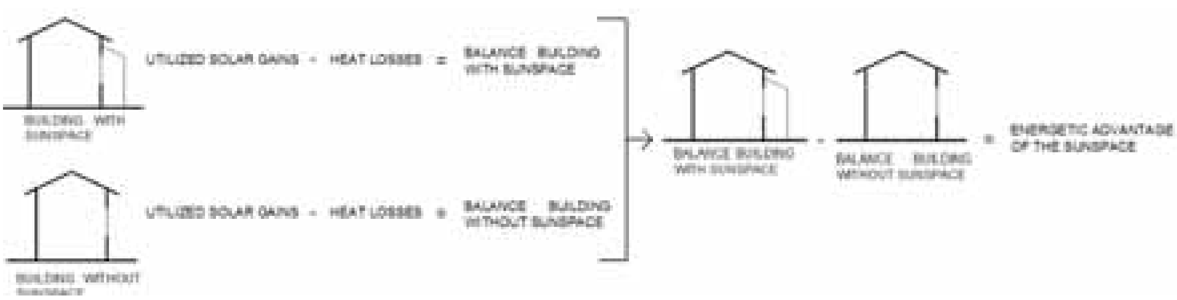


Fig. 1. Logic scheme of Method 5000 and ANNEX E of the standard UNI EN ISO 13790

Both methods consider “direct heat gains” (through windows and walls placed between inner environment and sunspace) and “indirect heat gains” (relating to warming of sunspace caused by solar radiation). The two methods mostly differ in the way they calculate “indirect heat gains”. Method 5000 calculates the "heat trapped in sunspace" considering solar heat coming through the glazed envelope of the sunspace, subtracting heat going directly to inner environment (and already considered within "direct heat gains") and using empirical coefficients related to the part of heat losses to external environment. Instead, standard UNI EN ISO 13790 calculates heat absorbed by every opaque surface inside the sunspace, subtracts heat going directly to inner environment and multiplies the remaining part by a factor depending on heat transfer coefficients in order to take into account different heat dispersions.

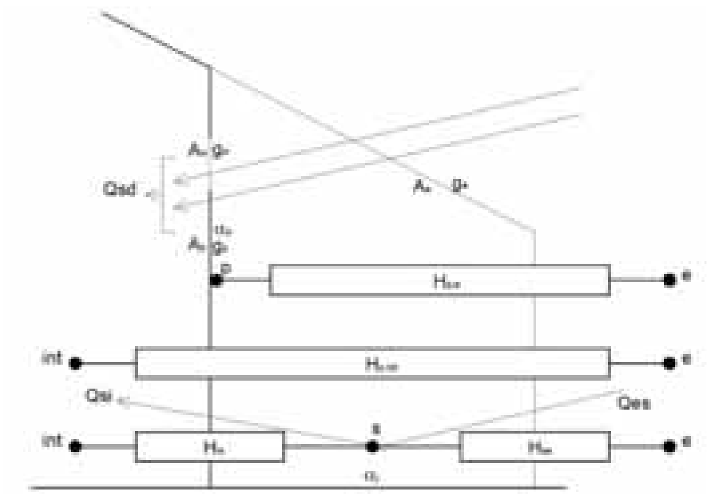


Fig. 2. Calculation scheme of heat gains in a sunspace
according to standard UNI EN ISO 13790:2008

Also heat gains due to inner opaque surfaces are calculated in different ways by the two methods: method 5000 multiplies heat absorbed by wall by 0.11U (where U is its thermal transmittance) while the UNI EN ISO 13790 method multiplies it by ratio between heat transfer coefficients $\frac{H_{p,tot}}{H_{p,e}}$, considering therefore the sunspace envelope properties as well.

3.1.2 A non-stationary model: A.K. Athienitis and M. Santamouris (2002)

A.K. Athienitis and M. Santamouris (2002) [5] proposed a non-stationary model in which thermal inertia of wall separating inner environment and sunspace is considered (Fig. 3). Basically, it is an equivalent network constituted by nodes, resistances and capacities. An approximation by finite differences method is proposed.

3.1.3 New analytical model and scale models

Compared to A.K. Athienitis and M. Santamouris (2002), the new model proposed considers heat capacity of sunspace air, heat striking the floor, heat transfer to the ground and ventilation between inner environment and sunspace. By considering heat capacity of air, the variations of its temperature value can be considered. Ground temperature at a certain depth is considered constant (it is a valid approximation depending on the period of time taken into account). Two different sunspaces are considered: the first one is a full scale prototype (subscript "pr"), the second one is a scale model (subscript "m"), i.e. it is different from the prototype only because of a proportional variation of geometrical dimensions.

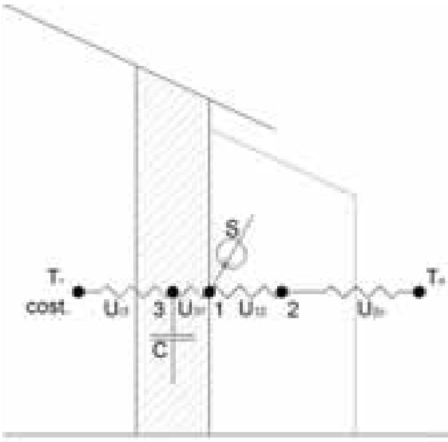
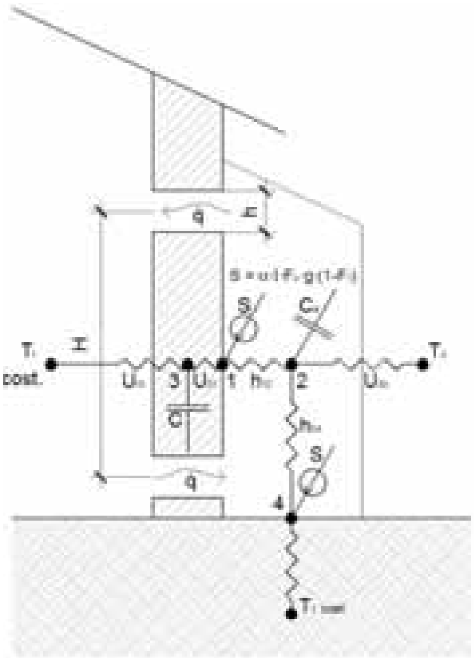


Fig. 3. Model by A.K.Athienitis and M.Santamouris (2002). Unknown quantities are temperatures T_1, T_2, T_3.



Known physical quantities

T_o climatic data [K]

I climatic data [W/m^2]

α absorption coefficient for solar radiation of the opaque part

F_{sh} shading reduction factor

g total solar energy transmittance of the glazed envelope

F_F frame area fraction

T_r constant depending on heating plant [K]

T_t constant (if time step is little enough) [K]

U thermal transmittances and surface coefficients of heat transfer [W/m^2]

C and C_a thermal capacities [J/K]

Unknown physical quantities

temperatures T_1, T_2, T_3, T_4

Fig. 4. New model proposed for heat transmission in a sunspace

According to p. 45 of CISBE (1994) [6] air flux between sunspace and internal environment is considered equal to

$$\Phi = \frac{A}{C_d} \cdot \sqrt{\frac{\Delta T \cdot gH}{T_2}} = \frac{hL}{C_d} \cdot \sqrt{\frac{|T_2 - T_r| \cdot gH}{T_2}} \qquad [m^3/s] \qquad (10)$$

C_d is the discharge coefficient and its typical value is 0,6.

Heat flux from node 2 to node r due to ventilation is therefore

$$\dot{Q}_{vent} = \dot{m}c_p \cdot (T_2 - T_r) = \rho\Phi c_p \cdot (T_2 - T_r) = \rho \cdot \frac{hL}{C_d} \cdot \sqrt{\frac{|T_2 - T_r| \cdot gH}{T_2}} \cdot c_p \cdot (T_2 - T_r) \quad [W] \qquad (11)$$

$$\dot{Q}_{vent} > 0 \iff T_2 > T_r$$

Heat entering internal environment is

$$\dot{Q}_{ent} = \dot{Q}_{vent} + \dot{Q}_{wall} \qquad (12)$$

The physical quantities regarding the time step j-th are denoted by subscript j and Δt is the interval between two successive time steps. Similarly to A.K. Athienitis and M. Santamouris (2002), energetic balances of nodes whose temperatures are unknown are inserted in a system, where an approximation by finite differences method is used.

$$\begin{cases} U_{12}A_w \cdot (T_{1\,j} - T_{2\,j}) + U_{2o}A_s \cdot (T_{o\,j} - T_{2\,j}) + U_{24}A_f \cdot (T_{4j} - T_{2\,j}) - \rho \cdot \frac{hL}{C_d} \cdot \sqrt{\frac{|T_{2\,j} - T_r| \cdot gH}{T_2}} \cdot c_p \cdot (T_2 - T_r) = C_a \cdot \frac{(T_{2\,j+1} - T_{2\,j})}{\Delta t} \\ U_{r3}A_w \cdot (T_r - T_{3\,j}) + U_{31}A_w \cdot (T_{1\,j} - T_{3\,j}) = C \cdot \frac{(T_{3\,j+1} - T_{3\,j})}{\Delta t} \\ U_{31}A_w \cdot (T_{3\,j} - T_{1\,j}) + U_{12}A_w \cdot (T_{2\,j} - T_{1\,j}) + SA_w = 0 \\ U_{24}A_f \cdot (T_{2\,j} - T_{4\,j}) + U_{4t}A_f \cdot (T_t - T_{4\,j}) + SA_f = 0 \end{cases}$$

therefore

$$\begin{cases} T_{2\,j+1} = \frac{\Delta t}{C_a} \cdot \left[U_{12}A_w \cdot (T_{1\,j} - T_{2\,j}) + U_{2o}A_s \cdot (T_{o\,j} - T_{2\,j}) + U_{24}A_f \cdot (T_{4j} - T_{2j}) - \rho \cdot \frac{hL}{C_d} \cdot \sqrt{\frac{|T_{2\,j} - T_r| \cdot gH}{T_2}} \cdot c_p \cdot (T_{2\,j} - T_r) \right] + T_{2\,j} \\ T_{3\,j+1} = \frac{\Delta t}{C/A_w} \cdot [U_{3r} \cdot (T_r - T_{3\,j}) + U_{31} \cdot (T_{1\,j} - T_{3\,j})] + T_{3\,j} \\ T_{1\,j+1} = \frac{U_{31}T_{3\,j+1} + U_{12}T_{2\,j+1} + S}{U_{31} + U_{12}} \\ T_{4\,j+1} = \frac{U_{24}T_{2\,j+1} + U_{4t}T_t + S}{U_{24} + U_{4t}} \end{cases}$$

$T_{3\,j+1}$ and $T_{2\,j+1}$ are calculated by the first two equations then $T_{1\,j+1}$ is calculated by the third equation and $T_{4\,j+1}$ by the fourth one.

Equations are valid both for prototype and for scale model. At the j-th instant, temperatures of prototype and of scale model are physically imposed to be equal. The goal is to design a scale model so that temperatures are equal on successive time step (the j+1-th one) as well to the real scale ones.

$$(T_{2\,j+1})_m = \frac{\Delta t}{(C_a)_m} \cdot [U_{12} \cdot (A_w)_m \cdot (T_{1\,j} - T_{2\,j}) + U_{2o} \cdot (A_s)_m \cdot (T_{o\,j} - T_{2\,j}) + U_{24} \cdot (A_f)_m \cdot (T_{4j} - T_{2j})]$$
$$- \frac{\Delta t}{(C_a)_m} \cdot \left[\rho \cdot \frac{(h)_m (L)_m}{C_d} \cdot \sqrt{\frac{|T_{2\,j} - T_r| \cdot g \cdot (H)_m}{T_2}} \cdot c_p \cdot (T_{2\,j} - T_r) \right] + T_{2\,j}$$

and

$$\left(T_{2\,j+1}\right)_{pr} = \frac{\Delta t}{(C_a)_{pr}} \cdot \left[U_{12} \cdot (A_w)_{pr} \cdot \left(T_{1\,j} - T_{2\,j}\right) + U_{2o} \cdot (A_s)_{pr} \cdot \left(T_{o\,j} - T_{2\,j}\right) + U_{24} \cdot (A_f)_{pr} \cdot (T_{4j} - T_{2j})\right]$$

$$- \frac{\Delta t}{(C_a)_{pr}} \cdot \left[\rho \cdot \frac{(h)_{pr}(L)_{pr}}{C_d} \cdot \sqrt{\frac{|T_{2\,j} - T_r| \cdot g \cdot (H)_{pr}}{T_2}} \cdot c_p \cdot (T_{2\,j} - T_r)\right] + T_{2\,j}$$

Since K_L is the scale factor of linear dimensions (but wall thickness and dimension h of openings are not scaled by it; the reason is explained afterwards):

$$(L)_m = \frac{(L)_{pr}}{K_L} \qquad (H)_m = \frac{(H)_{pr}}{K_L} \qquad (A)_m = \frac{(A)_{pr}}{K_L^2}$$

Replacing, in the expression of $\left(T_{2\,j+1}\right)_m$, $(L)_m$ with $(L)_{pr}/K_L$, $(H)_m$ with $(H)_{pr}/K_L$ and $(A)_m$ with $(A)_{pr}/K_L^2$ it is evident that equation $\left(T_{2\,j+1}\right)_m = \left(T_{2\,j+1}\right)_{pr}$ is verified if:

- $K_L^2 \cdot (C_a)_m = (C_a)_{pr}$
- $(h)_m = \frac{(h)_{pr}}{\sqrt{K_L}}$

To satisfy the first equality, additional masses must be added in scale model.

To satisfy equality $\left(T_{3\,j+1}\right)_m = \left(T_{3\,j+1}\right)_{pr}$ it is imposed $\frac{(C)_m}{(A_w)_m} = \frac{(C)_{pr}}{(A_w)_{pr}}$

This means that equality of areic thermal capacities is imposed; it is achievable by building the wall of the scale model by the same material and by the same thickness than the wall of the prototype.

The third and the fourth equations of the system depend on the first two equations and therefore conditions $\left(T_{1\,j+1}\right)_m = \left(T_{1\,j+1}\right)_{pr}$ and $\left(T_{4\,j+1}\right)_m = \left(T_{4\,j+1}\right)_{pr}$ are automatically satisfied if $\left(T_{2\,j+1}\right)_m = \left(T_{2\,j+1}\right)_{pr}$ and $\left(T_{3\,j+1}\right)_m = \left(T_{3\,j+1}\right)_{pr}$ are satisfied.

We can conclude that if scale model is built so that $K_L^2 \cdot (C_a)_m = (C_a)_{pr}$ and $(h)_m = \frac{(h)_{pr}}{\sqrt{K_L}}$ and same initial temperatures are physically imposed on prototype and on scale model, it is possible to measure T_1, T_2 and T_3 in scale model and the equality $(T)_p = (T)_m$ is valid for each successive instant and for each considered point.

Considering now heat fluxes entering inner environment, it is possible to write the following

$$\left(\dot{Q}_{vent}\right)_m = \rho \cdot \frac{(h)_m(L)_m}{C_d} \cdot \sqrt{\frac{|T_2 - T_r| \cdot g \cdot (H)_m}{T_2}} \cdot c_p \cdot (T_2 - T_r)$$

replacing dimensions relating to scale model with the ones relating to prototype

$$\left(\dot{Q}_{vent}\right)_m = \rho \cdot \frac{(h)_{pr}/\sqrt{K_L} \cdot (L)_{pr}/K_L}{C_d} \cdot \sqrt{\frac{|T_2 - T_r| \cdot g \cdot (H)_{pr}/K_L}{T_2}} \cdot c_p \cdot (T_2 - T_r) = \frac{\left(\dot{Q}_{vent}\right)_{pr}}{K_L^2}$$

$$\left(\dot{Q}_{wall}\right)_m = U_{3r} \cdot (A)_m \cdot (T_3 - T_r) = \frac{\left(\dot{Q}_{wall}\right)_{pr}}{K_L^2}$$

According to (12) $\left(\dot{Q}_{ent}\right)_{pr} = K_L^2 \cdot \left(\dot{Q}_{ent}\right)_m$

So, the total amount of heat entering the prototype is K_L^2 times the one entering the scale model.

3.1.4 Detailed calculation of the temperature of a sunspace: radiative heat transfer

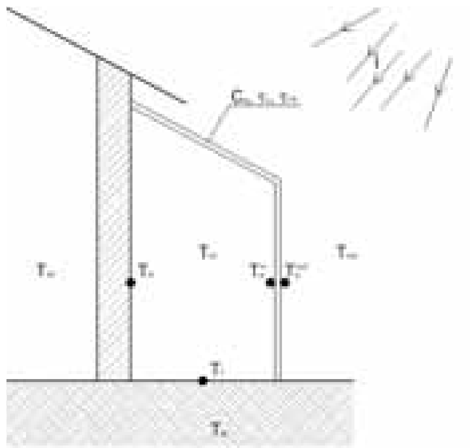


Fig. 5 Model for a detailed calculation of the temperature of a sunspace

In the new model proposed, infrared radiative heat transfer is modelled according to J.P. Holman (1992) [7] (pp. 449-450 and pp. 457-459), which provides indication to model radiative heat transfer through partially transparent surfaces. The model is an equivalent network of "surface resistances" (between correspondent blackbody emissive power $(E)_n$ and radiosity B) and "space resistances" (between radiosities of different surfaces) and is represented in Fig. 6. E means "emissive power relative to thermal radiation", "B" indicates the radiosity. The subscripts mean: w=wall, f=floor, g=glass (apex "in" for the internal surface, apex "out" for the external one), e=external environment. ρ and τ are reflectance relative to thermal radiation and transmittance relative to thermal radiation of the glass respectively.

The following hypotheses are made:

- the radiation is only diffuse
- the surfaces are part of grey bodies (i.e. properties do not depend on wavelength)
- the surfaces relative to external environment have much bigger extension than the other ones, therefore the resistance between B_e and $(E_e)_n$ is much littler and we can assume $B_e = (E_e)_n$
- every surface has uniform value of physical properties
- the air do not absorbs thermal radiation.

For what concerns solar radiation, the part reflected from opaque surfaces is divided into two parts depending on view factors: one reflected to another opaque element and one reflected to glazed surface. The most part of the latter one is disperse to outer environment depending on the transparent factor.

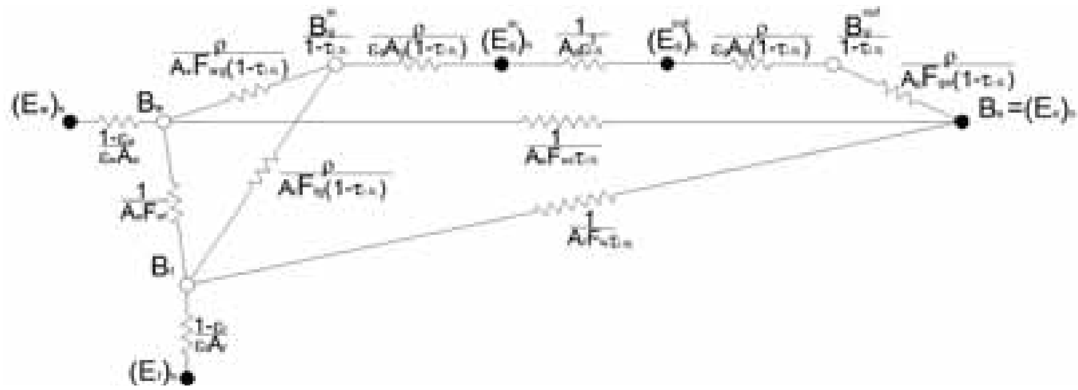


Fig. 6 Equivalent network of resistances for calculation of heat transfer by infrared radiation

3.2 Future developments: results by models and by physical realization of sunspaces

Models are going to be utilized in order to evaluate how some input data (e.g. sunspace geometry, management of openings and shadings, properties of constructive elements…) influence output data.

Future development is the integration of the model presented in 3.1.3 with the one of infrared radiation presented in 3.1.4. because existing models available today are too simplified for what concerns the influence of radiative exchange between sunspace and outer environment.

Then, analytical models are going to be validated through comparison with empirical data by means of the construction and monitoring of real models, both full-scale and a scale model (1:2), starting in winter 2010/2011.

4. Conclusions

A new method is proposed in order to properly design sunspaces considered as passive solar devices. In particular, a new approach is proposed and discussed with the aim of studying small scale model sunspaces positioned in open space conditions with true boundary conditions, so to better control all parameters involved reducing time, space and economic impact of the research activity itself.

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