EVALUATION OF A SIMPLIFIED MODEL FOR FAÇADE COLLECTORS

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ABSTRACT

This paper presents a simplified model for Transparent Solar Thermal Collectors (TSTC). An existing, validated detailed model was used to identify the formula's coefficients by comparing the simulation results of both models. The accuracy of the simplified model as well as the influence of the set of simulations used to establish the model has been investigated. The simplified model has been compared to other formulas. The model has proven to be able to predict accurately the collector efficiency.

INTRODUCTION

Transparent Solar Thermal Collectors

In order to meet the new energy regulations[1], the building facade must take a larger role in energy production. This is especially true for buildings with a low roof-to-facade ratio such as high-rise buildings and skyscrapers.

Transparent Building-Integrated Solar Thermal Systems (BIST) using a liquid heat transfer medium offer new and interesting options. They provide renewable energy, visual transparency, solar control and low primary energy demands. One transparent solar thermal facade collector is already being marketed[2].

In [3], carried out within the European project "Cost-Effective", a new type of Transparent Solar Thermal Collector (TSTC) was developed and optimized. At the same time a detailed model for TSTC was developed. Figure 1 shows a schematic illustrating a possible TSTC construction. The collector consists of a triple glazing that includes a transparent absorber between the cover glass pane and the central glass pane. The absorber is fitted with tilted slats, which block all direct radiation above the cut-off angle. The goal is to control the heat flow to the room and prevent glare in summer. In winter when the sun's position is lower in the sky, higher solar transmission results. . The collector allows a good view to the outside, and the glazing serves as insulation against the outdoor temperatures and the hot absorber. The goal is to decrease the energy demand for heating, cooling and lighting by using solar energy.

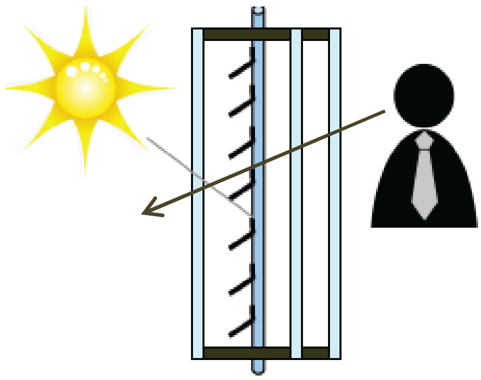

Figure 1 Schematic drawing of a TSTC

Figure 2 - TSTC, view from the inside.

Existing detailed model

The TSTC detailed model developed in [3] has many inputs and outputs. Inputs are, for example, the outdoor and indoor temperatures, the direct normal irradiation, the fluid inlet temperature, etc.

The two most significant outputs are:

- the heat flow from the component towards the interior Q_{int} (W*m^{-2} aperture), and

- the heat flow removed by the fluid Q_{use} (W*m^{-2} aperture).

The detailed model in [2] is a thermal multi-node model, using an energy balance for each node/layer. This energy balance is based on angle-dependent absorbance and transmittance (for each layer), temperature-dependent formulas for the convection and the infrared heat-exchange, and constant thermal resistances for conduction around the edges. The complex energy balance is then resolved by a solver to calculate the outputs.

The parameters of the model (spacing-properties, IR-values…) have been calibrated for the first prototypes by calorimeter measurements. This parameterization must be carried out anew for each type of collector.

A TRNSYS type was made of this detailed model. Many whole-year simulations using TRNSYS were then carried out with the developed TRNSYS type coupled to a building and its HVAC system. The simulations showed the energy saving potential of such TSTCs, as presented in [4].

The detailed model is validated against validated software, a flat plate collector measurement and measurements of a glazing unit with integrated blinds. Finally, the model is validated by calorimeter measurements of a transparent solar thermal collector prototype. The validation methodology is explained in [3].

Need of a simplified model

A simplified model of TSTC was already presented in [5]. However, this simplified model was not capable of accurately modeling Q_{int} and Q_{use}, since it neglects the effect of the operating mode on Q_{int} and underestimates Q_{use} with increasing fraction of diffuse irradiance. As a result, the calculated heating and cooling loads were sometimes over or underestimated, when the errors in the simplified model reinforced each other.

The detailed model is accurate but complex, with over 300 parameters to fit on the basis of measurements.

Apart of the simplicity of use, a simplified model is needed for several reasons:

-A simplified model with only 5 parameters to fit requires only a limited number of expensive calorimetric measurements.

- Computational time can be reduceded by using a simplified model instead of a detailed model.

- A simplified model may allow to compare façade collectors by including the simplified model in a norm, as it is done for standard roof collectors[6].

This papers aims to present a new simplified model, in order to allow a good modeling of Q_{use} and Q_{int}.

SIMPLIFIED MODEL

Existing model for opaque collectors

The efficiency $\eta = Q_{use}/G$ of collectors, as given, for example, in [6], is usually calculated by

$$\eta = \eta_0 - a_1 * X - a_2 * X^2 * G \quad (1)$$

With

- $X = \frac{T_m - T_{ext}}{G}$ [m^2*K *W^{-1}]
- a_1: linear heat loss coefficient in [W*K^{-1}*m^{-2}].
- a_2: second order heat loss coefficient in [W*m^{-2}*K^{-2}].
- G the global irradiance on the collector surface in W*m^{-2}.
- $T_m = \frac{T_{fluid,in} + T_{fluid,out}}{2}$
- T_{ext} the ambient temperature.
- η_0 the efficiency at zero temperature difference between the mean fluid and the ambient temperature.

New Model

Contrary to opaque collectors, the efficiency of a façade collector also depends on the temperature of the building interior. The following formula was developed:

$$\begin{aligned} \eta = \eta_0 \\ -a_{1,ext} * X - a_{2,ext} * X^2 * G \\ -a_{1,int} * Y - a_{2,int} * Y^2 * G \end{aligned} \quad (2)$$

With

- $X = \frac{T_m - T_{ext}}{G}$ [m^2*K *W^{-1}]
- $Y = \frac{T_m - T_{int}}{G}$ [m^2*K* W^{-1}]
- $a_{1,int}$: internal linear heat loss coefficient [W*K^{-1}*m^{-2}],
- $a_{2,int}$: internal second order heat loss coefficient [W*m^{-2}*K^{-2}],
- $a_{1,ext}$: external linear heat loss coefficient [W*K^{-1}*m^{-2}],
- $a_{2,ext}$: external second order heat loss coefficient [W*m^{-2}*K^{-2}],
- G: total irradiance on the collector surface W*m^{-2},
- $T_m = \frac{T_{fluid,in} + T_{fluid,out}}{2}$: the mean fluid temperature in the collector,

- T_{int}: the temperature of the building interior,
- T_{ext}: the ambient temperature,
- η_0: the efficiency at zero temperature difference between the fluid, the front and the back of the collector.[3]

The hypothesis is that the heat flux to the interior Q_{int} equals:

$$Q_{int} = a_{1,int} * Y * G + a_{2,int} * Y^2 * G^2 \qquad (3)$$

Parameterization of the new model

In order to test this simplified formula, the collector prototype developed in [3] was chosen. A validated, detailed model already exists for this prototype. This allows us to use accurate stationary simulations in order to parameterize the simplified formula.

For the parameterization, four input parameters were varied: the exterior temperature T_{ext}, the building interior temperature T_{int}, the direct normal irradiance on the collector G and the fluid inlet temperature $T_{fluid,in}$. The outputs were Q_{use}, the heat flux transmitted to the fluid, and so the efficiency η for each simulation.

First, we calculated η_0 by setting T_{int}=T_{ext} and varying $T_{fluid,in}$ until X=Y=0. η_0 was found to be equal to 0.6989.

Simulations were carried out using many combinations of the four input parameters, varying within following ranges:
- T_{ext} was varied within [-20;40] with 5°C steps.
- T_{int} was varied within [0;40] with 5°C steps.
- Irradiance was varied within [50;1100] with 50 W*m^{-2} steps.
- Tfluid,$_{in}$ was varied within [20;80] with 5°C steps.

All values were simulated or calculated on the basis of the aperture area of the TSTCs.

These 33,462 combinations of the inputs cover almost all possible situations. There are also unrealistic situations such as a T_{ext}=-20°C, T_{int}=40°C, G=1100 W*m^{-2}, $T_{fluid,in}$=30°C. This situation is unrealistic because such a high indoor temperature would not happen in reality with such a low outdoor temperature.

After all the combinations were calculated with the detailed model, the coefficients $a_{1,int}$, $a_{2,int}$, $a_{1,ext}$ and $a_{2,ext}$ of the simplified model were parameterized. The parameterization was done by using a solver to minimize the Root Mean Squared Error (RMSE) between detailed and simplified model. The parameterization leads to following formula:

$$\begin{aligned}\eta = {} & 0.6989 \\ & -4.506 * X - 0.00095 * X^2 * G \\ & -1.010 * Y - 0.003294 * Y^2 * G\end{aligned} \qquad (4)$$

For these values, the RMSE for the efficiency η was equal to 0.0300.

So the proposed formula is able to predict the efficiency η – and the resulting Q_{use} -with a good accuracy.

The coefficients determined show greater losses towards the exterior. This corresponds to the collector geometry, with a single glazing towards the exterior and a double glazing towards the interior.

Influence of the set of simulations chosen

To assess the impact of the set of simulations used to parameterize the formula, another set of simulations was used. This time, the simulation inputs were inspired by the standard EN 12 975-2. This standard gives indoor and outdoor measurement rules for determining efficiency curves of opaque collectors:
- The standard requires that at least four different values of $T_{fluid,in}$ are used. We used 13 values, with $T_{fluid,in}$ varying within [20;80] with 5°C steps.
- T_{ext} and the irradiance were taken from weather data (Stuttgart, Germany). Situations were chosen where the irradiance on the collector surface is above 700 W*m^{-2}, as required by the standard. Ten pairs of T_{ext} and irradiance taken from a summer week were simulated.
- T_{int} was varied within [15;35] with 5°C steps. This corresponds to the range of indoor temperatures you can encounter during a summer week in Germany.

All 650 combinations were simulated. The coefficients of the simplified formula were optimized to fit the simplified model to the detailed one in all 650 simulations.

The parameterization leads to the following formula:

$$\begin{aligned}\eta = {} & 0.6989 \\ & -4.792 * X - 0.004805 * X^2 * G \\ & -0.9566 * Y - 0.002373 * Y^2 * G\end{aligned} \qquad (5)$$

For these values, the RMSE for η was equal to 0.0023..

The same process was done for 10 winter situations:
- $T_{fluid,in}$ was varied within [20;80] with 5°C steps.

- T_{ext} and the irradiance were taken from the same weather data (Stuttgart). Only situations where the irradiance on the collector surface is above 700 $W*m^{-2}$ were chosen, as required by the standard. Ten pairs of T_{ext} and irradiance taken from a winter week were simulated.
- T_{int} was varied within [5;25] with 5°C steps, which corresponds to possible indoor temperatures during a winter week.

All 650 combinations were simulated. The coefficients of the simplified formula were optimized to fit the simplified model to the detailed one in all 650 simulations.

The parameterization leads to following formula:

$$\eta = 0.6989 \\ -4.396 * X - 0.005681 * X^2 * G \\ -0.9814 * Y - 0.002610 * Y^2 * G \quad (6)$$

For these values, the RMSE on the collector efficiency was equal to 0.0038.

From one set of simulations to the other, the parameters of the formula are close. The choice of the set of simulations used to parameterize the equation seems to have a limited impact of the coefficients, as can be seen on following figure. Especially for low X and Y value, which represent small differences between the fluid and respectively outside and inside temperature, the difference is low.

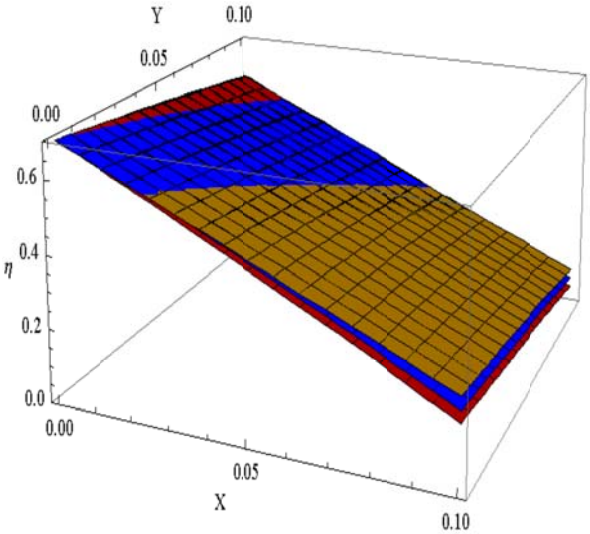

Figure 3 Efficiency surfaces with an irradiance on the collector area of 1000 $W*m^{-2}$, parametrized with the first voluminous simulation set (orange) , the summer cases (red) and the winter cases (blue).

However, the exact influence of choice of the 4 input values to parametrize the equation has not been investigated in this work.

Comparison with other models

For the realistic summer and winter cases, the model proposed in formula (2) was compared with two other models:

- The standard model for opaque collectors:

$$\eta = \eta_0 - a_1 * X - a_2 * X^2 * G \quad (1)$$

with

$$x = \frac{T_m - T_{ext}}{G}$$

- And the formula using the equivalent ambient temperature:

$$\eta = \eta_0 - a_1 * \left(\frac{T_m - T_{amb,eq}}{G}\right) - a_2 * \left(\frac{T_m - T_{amb,eq}}{G}\right)^2 * G \quad (7)$$

with

$$T_{amb,eq} = \frac{T_{ext} + T_{int}}{2}$$

Both models were parameterized for the winter and summer cases by minimizing the RMSE. The results can be seen in following table:

Table 1 RMSE between the efficiency calculated with several models and the efficiency calculated with the detailed model, for the summer and winter cases.

RMSE (on the collector efficiency η)	Standard Model (1)	Average temperature model (7)	Simplified Model (2)
Summer cases	0.0112	0.0203	0.0023
Winter cases	0.0125	0.0429	0.0038

One interesting result is that Formula (1) (standard model) works better than Formula (7) (average temperature model), without taking into account the indoor temperature. One explanation is that the exterior temperature has a larger influence for the thermal losses and so on the collector's efficiency, due to the collector's geometry.

The best results are given by the proposed simplified model (2).

Calculation of other outputs

The heat flux going from the collector towards the interior of the building, Q_{int}, is very important for the thermal simulation of building. We proposed to extrapolate Q_{int} from formula (2), with:

$$Q_{int} = a_{1,int} * Y * G + a_{2,int} * Y^2 * G^2 \quad (3)$$

However, for the three sets of simulations used (all combinations including unrealistic cases, summer cases, and winter cases), formula (3) was not able to model Q_{int}. For example, for the summer cases the RMSE value for Q_{int} was 19.7 W*m^{-2}. The mean value of Q_{int} over all summer cases for the detailed model was 47.9 W*m^{-2}. Even when fitting the coefficients to decrease the RMSE on Q_{int} (without regards to η), Formula (3) was not able to model Q_{int}. The RMSE was, for example, equal to 11.0 W*m^{-2} for the summer cases. On possible explanation is that the simplified model doesn't take into account the edge effects and the energy flowing from the exterior directly to the interior through the edge.

CONCLUSION

A new simplified model to calculate the efficiency of Transparent Solar Thermal Collectors (TSTCs) has been presented and compared with other models.

A detailed, validated model of a TSTC was used to parameterize the simplified model using different simulation data sets.

First investigations showed that the formula was not able to predict the heat flux Q_{int} from the collector to the interior. However, the formula has been proven to accurately model the collector's efficiency η for several sets of simulations. The influence of the choice of data set used to parameterize the simulations appears to be negligible.

This new simplified model is capable of modelling the collector gain. Further tests using other types of solar thermal façade collectors, such as [2] still need to be performed. The parameters can be fitted on the basis of measurement or on the basis of a physical model. It is still necessary to find a simplified model which is capable of predicting the heat flux Q_{int} to the interior.

One long term goal is to have a comparison tool for façade collectors, such as formula (1) for opaque collectors. Such a formula could be integrated in a future standard for façade thermal collectors.

NOMENCLATURE

Q_{int} = heat flow from the component towards the interior in W*m^{-2} aperture.
Q_{use} = heat flow removed by the fluid in W*m^{-2} aperture.
a_1 = linear heat loss coefficient in W*K^{-1}*m^{-2}.
a_2 = second order heat loss coefficient in W*m^{-2}*K^{-2}.
G = irradiance on the collector area in W*m^{-2}.
$T_m = \frac{T_{fluid,in}+T_{fluid,out}}{2}$, mean fluid temperature.
T_{ext} = ambient temperature.
T_{int} = temperature of the building interior.
$T_{amb,eq} = \frac{T_m - \frac{T_{ext}+T_{int}}{2}}{G}$ = equivalent ambient temperature.
η_0 = efficiency with no temperature difference between the fluid and the ambient temperature.
$a_{1,int}$ = internal linear heat loss coefficient in W*K^{-1}*m^{-2}.
$a_{2,int}$ = internal second order heat loss coefficient in W*m^{-2}*K^{-2}.
$a_{1,ext}$ = external linear heat loss coefficient in W*K^{-1}*m^{-2}.
$a_{2,ext}$= external second order heat loss coefficient in W*m^{-2}*K^{-2}.
G the total irradiance on the collector surface in W*m^{-2}.

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REFERENCES

1. *Directive 2010/31/EU of the European parliament and of the council of 19 May 2010 on the energy performance of buildings*. 2010, Official Journal of the European Union.
2. RobinSun, *TECHNICAL DOCUMENTATION ROBIN SUN SOLAR THERMAL GLASS*. 2011.
3. Maurer, C., *Theoretical and experimental analysis and optimization of semi-transparent solar thermal façade collectors*, in *Fakultät für Architektur*. 2012, Karlsruher Institut für Technologie (KIT): Freiburg. p. 118.
4. Maurer, C. and T.E. Kuhn, *Variable g value of transparent façade collectors*. Energy and Buildings, 2012. **51**: p. 177-184.
5. Maurer, C., et al. *Heating and cooling in high-rise buildings using façade-integrated transparent solar thermal collector systems*. in *Building Simulation 2011*. 2011. Sydney: IBPSA.
6. *Norm EN 12975-2:2006 - Thermal solar systems and components: Solar collectors*. 2006.