THERMAL LOSSES IN SEALED, GAS-FILLED FLAT PLATE SOLAR COLLECTORS

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ABSTRACT

A sealed space between absorber and cover glass makes it possible reducing the influence of humidity condensate and dust at the same time as the enclosed space can be filled with a suitable gas for lowering the losses. This paper is about the size of the losses in these collectors. A calculating model of a gas-filled flat plate solar collector was built in Matlab with standard heat transfer formulas. It showed that the total loss can be reduced up to 20% when changing to an inert gas. It is also possible using a much shorter distance and still achieve low losses at the same time as the mechanical stresses in the material is reduced.

1. INTRODUCTION

The main aim is to study the possible improvements regarding thermal performance of flat plate collectors with different gas fillings between absorber and cover glass more in detail. Other gases than air requires a sealed space that has the advantage of reducing the influence of condensate and dust on the selective absorber surface and lowering the heat losses. The disadvantage is however that a more complicated design is required in order to manage variations of the pressure in, or the volume of, the gas filling. This work concentrates on the collector performance when the gas pressure is 100 kPa, i.e. close to the ambient air pressure..

2. BODY OF PAPER

2.1 Basic Theory

The top loss is a heat flow from absorber, via gas, glass, upside convection layer and out to the ambient air. The heat

flow through the gas from absorber to glass is made of conduction, convection and radiation. Equilibrium exists, so the heat transfer from absorber to glass is the same as the heat transfer from glass to the ambient air. Conduction and convection is usually expressed in the same formula, because they are dependant on each other. The total top loss q_{top} will be according to (1).

$$q_{top} = q_{cpg} + q_{rpg} = q_{cga} + q_{rga} \quad (1)$$

When calculating the backside losses the approach is the same, except that it is possible just calculating a pure conduction in the insulation between absorber and back side sheet. There are also losses to the edges. Here they are counted as a part of the top respectively the backside losses.

There is also a useful heat transfer, q_w , which goes from the absorber plate to the heat transfer media. The mean path to go is from the middle of one flange of a fin (i.e. $\frac{1}{4}$ of the tube spacing) + the boundary between absorber and tube + the mean cross section way in the tube + the convection layer from tube to the heat transfer media.

The heat transfer via convection and conduction from absorber to glass (q_{cpg}) is the one which is affected by changing gas. The q_{cpg} can be expressed as in (2).

$$q_{cpg} = Nu * k * S * (T_p - T_g) \quad (2)$$

As can be seen from formula 2, a heat flow is dependent of the conductivity k , shape factor S and temperature difference in the same way as a pure conduction driven heat transfer. The only difference between conduction and convection is the Nusselt number, Nu . The conductivity and the Nusselt number will be affected when changing gas.

Most of the formulas describing the different heat transfers in the different heat transfer medias are standard formulas.

Holman has done an almost complete compilation of suitable heat transfer formulas for setting up a model like the model used for calculations in this article (Holman 2002). The relation between Nusselt and Rayleigh numbers in inclined air layers is thoroughly investigated by [1] and [2]. The dependency of the Rayleigh number and the slope on the Nu number is shown in (3)

$$Nu = 1 + 1.44 \left[1 - \frac{1708(\sin 1.8\beta)}{Ra \cos \beta} \right]^* \left[1 - \frac{1708}{Ra \cos \beta} \right]^+ + \left[\left(\frac{Ra \cos \beta}{5830} \right)^{1/3} - 1 \right]^+ \quad (3)$$

Note: The + exponent means that only positive values of the terms in the square brackets are to be used. (3) is valid in a range of the slope between 15 to 60°.

2.2 Calculations

All calculations were made in Matlab, a suitable tool for practising numerical methods. At given conditions the unknown factors were possible finding by iterations.

When simulating unlinearities on the absorber, the calculation between absorber and glass was done several times with different l_{pg} . The resulting q_{pg} was a mean value of these calculations. That approach was also used when simulating a variation of temperature on the absorber but then the T_p varied. The T_p varies depending on distance out on the fin and the warming of the heat transfer media and the variation is not linear. A simple way to compensate for this was to weight the hotter parts a little bit different to the colder ones which were done in this model. The approach was a gamma correction used for interpolate the temperatures between the ends as in (4).

$$y = x^\gamma \quad (4)$$

$x=0$ corresponds to the coldest point and $x=1$ to the hottest. The temperature is calculated as in (5)

$$T(x) = T(0) * y + T(1) * (1 - y) \quad (5)$$

The y-values between $x=0$ and $x=1$ can be varied so a curve representing the temperatures in between can get a more representative temperature distribution. In reality the mid points are warmer than what a linear interpolation calculates. In this calculations a $\gamma=0.5$ was used.

Material data, i.e. gas properties was lineared in Matlab. Input data comes from measurement series[3-6]. Physical dimensions and other typical solar collector properties are taken from a model made by Schüco, called Premium. It was state of the art flat plate solar collectors among the ones sold by companies until 2006 in Sweden. The reason why this model was chosen is that if the gas filled collectors cannot achieve better performance than the already existing good ones there is no good reason to develop any with this

new technology unless it is possible to reduce the cost of manufacturing. Another reason was that the physical sizes were known and that it was possible to verify the model with a known collector.

A shape factor also had to be constructed, which accounted for how much more the edge affected the heat loss when l_{pg} varied in the calculations. The Schüco Premium has no side wall insulation and the distance between absorber and side wall was estimated to be 10 mm. The shape factor was calculated with a FEM-model assembled in Matlab. It all ended up with a shape factor which can be seen in (6)

$$S = \frac{A}{L_{pg}} (s1 + s2 * l_{pg} + s3 * l_{pg}^2) \quad (6)$$

The factors of a, b and c in the polynomial in (6) were found to be: $s1 = 0.995 \pm 0.005$, $s2 = 1.64 \pm 0.1$ and $s3 = 21 \pm 2$ in the range of $l_{pg} = 0.001 \dots 0.1$ m. This makes a maximum error of 2.5 % from the polynomial, but the heat transfer around the edge is not uniform, so the error is probably bigger.

2.4 Results

The model was validated with test results from the reference collector as well as a model set up in CodePro.exe [7], [8]. The performance were expressed as the c-factors for the efficiency formula which can be seen in (7).

$$\eta = c0 - c1 \frac{(T_w - T_a)}{G} - c2 \frac{(T_w - T_a)^2}{G} \quad (7)$$

The $c1$ and $c2$ factors are the most interesting factors here, since they are temperature dependant. The different $c1$ and $c2$ values can be seen in Table 1.

Table 1: Validation of model for reference collector

Value	Test	CodePro	Model
$c1$	3.518	3.845	3.97
$c2$	0.012	0.00352	0.0042
ΔU from test, $T_w=10^\circ\text{C}$		7%	10%
ΔU from test, $T_w=50^\circ\text{C}$		-3%	2%
ΔU from test, $T_w=100^\circ\text{C}$		-12%	-7%

The model is almost correct at $T_w=50^\circ\text{C}$ and underestimates the losses on high temperatures and overestimates them on low temperatures.

In the following the influence of gas filling on the overall heat loss is studied. It is assumed that the collector has the same physical dimensions and properties as a Schüco Premium, except l_{pg} , that the l_{pg} varies 2 mm, mean $T_w = 50^\circ\text{C}$, highflow system with $0.05\text{m}^3/(\text{h} \cdot \text{m}^2)$, slope = 45° , a

variation of l_{pg} of 2mm. The overall loss coefficient's dependency of the l_{pg} for different gas fillings can be seen in Fig 1.

Fig 1: Losses vs distance l_{pg} at $T_w=50^\circ\text{C}$ and $\beta = 45^\circ$

The result is that the overall loss coefficient can be lowered with about 20% changing to the best alternative, here Xenon (Xe). All studied gases have the same appearance. There is a minimum on every curve at a distance of 25..50 mm, depending of gas (and of course the edge insulation), which here will be called normal distance minimum. It is also a minimum on a very shorter distance, somewhere between 3 and 9 mm, depending of gas, which here will be called short distance minimum. Between the two minimums on each curve there is a hump near the short distance minimum, called local maximum.

Traditionally the normal distance minimum is used, or at least something near the minimum. The usage of the normal distance minimum in an enclosed collector will cause some problems. It will be a large amount of gas and the variations of temperature, which is normal for a solar collector in use, will cause either volume variations or pressure variations or a mixture of them in the enclosed gas, due to the common gas law. Therefore it is interesting analyze the short distance minimum.

The short distance minimum (from 3 – 8 mm depending on gas) is an interesting point of the curves in Fig 1. It appears on the longest distance where Nu still is 1 or very near 1. The short distance minimum has the wanted performance and a much less volume of gas is needed which causes smaller stresses in the material compared to the normal distance minimum. The fly in the ointment is that the distance is more critical.

A sensibility test was done which can be seen in [Table 2](#).

Table 2: Validation of model for reference collector

	Air	CO ₂	Ar	Kr	Xe	Ref.
U (W/(m ² *K))	4.56	4.62	4.04	3.76	3.56	4.68
Ideal l_{pg} (mm)	8.5	5.6	7.8	5.2	3.9	40
$\delta l_{pg} + 10\%$	-0.2%	-0.3%	-0.3%	-0.3%	-0.3%	0.0%
$l_{pg} + 10\%$	-1.1%	-0.7%	-1.1%	-0.6%	-0.4%	0.1%
$\delta l_{pg} + 10\%$, $l_{pg} + 10\%$	-1.1%	-0.8%	-1.2%	-0.7%	-0.5%	0.1%
$T_w + 5K$	1.9%	1.8%	1.9%	1.9%	1.9%	2.0%
Slope -10°	-1.3%	-1.4%	-1.2%	-1.1%	-1.1%	-0.9%
$T_w + 5K$, slope -10°	0.7%	0.4%	0.7%	0.8%	0.8%	1.1%
$l_{pz} + 10\%$	1.5%	1.5%	1.5%	1.6%	1.6%	1.4%
Wind $+10\%$	-0.3%	-0.3%	-0.3%	-0.3%	-0.3%	-0.3%

The changed parameters are in the left column and the changes of U from the starting point are in the columns to the right. The varied parameters are: + 10% change of unlinearity of the absorber plate (δl_{pg}), +10% deviation from the l_{pg} , a combination of δl_{pg} and l_{pg} where both were 10% larger, +5K change of heat transfer media temperature, , a change in the slope by -10° , a combination of T_w and slope (+5K, -10°), +10% change in wind speed, and a change in plate thickness by 10%.

The most spectacular observation is that a rise in temperature will lower the U-value. At the same time as the upward losses grows the losses due to the fin efficiency decreases. Totally the losses seem not increase as much as the temperature changes which leads to a lower U-value, but remember: this special effect is depending on the thickness of absorber and the irradiation and the model also underestimated the c2 factor. The thickness of the absorber will also give us a better U-value when only changing that.

Variations of l_{pg} and δl_{pg} got an expected result, and it is not any superposition relationship between them. If l_{pg} grows, the U-value will reach the local maximum as most, and then it will decrease again.

The slope and wind also have some influences, but not much more than on the reference.

Fig 2 shows the result in an ordinary collector efficiency diagram for a better understanding of the influence of changing gas in a collector with no expansion. The curve for the reference collector applies to 40 mm, while all others curves apply for the short distance minima, i.e. for a much thinner box. CO₂ and thin air have about the same performance as the reference collector. All the inert gas

filled collectors achieve better performance, and the succession order is Xe, Kr and Ar (with the best first).

Fig 2 : Efficiency curve

2.5 Discussion

Models are models and the truth is out there with experimenting tests. Nevertheless a model can compare different alternatives and find a succession order and it can also find out some promising aspects. The sources of errors are several in the model, as it usually is in convection theory. Nevertheless, all gases are treated in the same way and it is the differences between the gases this article wants to focus on.

As can be seen, short distances can be useful even if you don't hit the absolute minimum. Moderate deviations from the optimal l_{pg} only gives moderate more losses of performance. It is also interesting to see that a "thin" collector with inert gas still gives lower losses compared to a common "normal thick" air filled solar collector.

3. NOMENCLATURE

A	Area (m^2)
$c_{0..c2}$	factors describing losses in efficiency equation
G	Irradiance (W/m^2)
k	Heat conductivity ($W/(m*K)$)
l	Length (m)
Nu	Nusselt number (-)
q	Heat transfer per area (W/m^2)
Ra	Rayleigh number(-)
S	Conduction shape factor (m)

s	Factor in edge shape polynomial
T	Temperature (K)
U	Heat transfer per area and temp. diff ($W/(m^2*K)$)
x	Relative position between nodes (0..1)
y	Weight factor (0..1)
γ	Gamma interpolation correction factor(-)
Δ, δ	Difference
η	Efficiency (-)

Subscripts

a	Ambience
b	Backside
c	Convection
g	Cover Glass
p	Absorber plate
z	z-axis (height)
r	Radiation
w	Heat transfer media

5. REFERENCES

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