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Thermal performance of gas-filled flat plate solar collectors

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Abstract

A sealed space between absorber and cover glass in a flat plate solar collector makes it possible to reduce 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 heat losses. This article describes the influence of different gases on the heat losses in a typical flat plate solar collector. A model of a gas-filled flat plate solar collector was built in Matlab with standard heat transfer formulas. The results show that the overall heat loss can be reduced by up to 20% when changing from air to an inert gas. It is further possible to reduce the distance between absorber and cover in order to reduce the mechanical stresses in the material with similar heat losses.

Keywords: Solar collectors; Modeling; Heat losses; Inert gases

1. Introduction

There is continuous endeavour to improve the performance and design of flat plate solar collectors. So far, only a small number of products with other gas fillings than air have been marketed, and these without great success. The main aim, here, is a more detailed study of the possible improvements regarding thermal performance of flat plate collectors with different gas fillings between absorber and cover glass. Other gases than air require a sealed space and this has the advantage of reducing the influence of condensate and dust on the selective absorber surface and of lowering the heat losses. The disadvantage is that a more complicated design is required in order to manage variations in the pressure and the volume of the gas filling. The thermal performance can be further improved by lowering the pressure of the gas. In a pressure range between ~10 Pa–10 kPa called the continuous area, the heat transfer properties of a gas are almost constant and it would be possible to achieve performance near that of vacuum tube collectors (Beikircher et al., 1996; Benz and Beikircher, 1999). Such a low pressure collector requires an even more complex design that can handle the pressure differences between inside and outside. This work concentrates, however, on the collector performance when the gas pressure is near 100 kPa, i.e., close to the ambient air pressure, since there are very few studies on this topic.

2. Basic theory

Heat transfer from an absorber is of two types. The desired heat transfer is to the heat transfer medium. There is also unwanted heat transfer in the form of heat losses which can be through the front cover, the back and the edges of the collector. The edge losses are normally small compared with the top and back losses for flat plate solar collectors typically around 10% of the total losses (Tabor, 1958). Hence the edge losses will be included in the front and back losses, respectively, and the losses can be described as:

$$q_l = q_{top} + q_{back} \tag{1}$$

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Nomenclature

A	area (m^2)	β	slope (°)
$a1$	loss coefficient when mean temperature of heat transfer medium is the same as the ambient temperature (W/m^2, K)	γ	gamma interpolation correction factor (–)
$a2$	temperature dependant loss coefficient (W/m^2, K^2)	δd	non-linearities of distance (m)
d	distance (m)	η	efficiency (–)
F'	collector efficiency factor (–)	ΔT	temperature difference (K)
$G(\tau\alpha)$	absorbed solar energy in the absorber plate (W/m^2)	*Subscripts*	
k	heat conductivity (W/(m K))	a	ambient
Nu	Nusselt number (–)	b	back, bond
q	heat transfer (W)	c	convection
Ra	Rayleigh number(–)	fin	fin
S	conduction shape factor; if a wall $S = A/d$ (m)	g	cover glass
s	factor in edge shape polynomial (–)	l	loss
T	temperature (K)	p	absorber plate
U	heat transfer per area and temp. diff (W/m^2, K)	r	radiation
x	relative position between nodes (0…1)	t	tube in absorber
y	weight factor (0…1)	top	top
		u	useful
		w	heat transfer medium

The top loss (q_{top}) is a heat flow from the absorber, via the gas and the glass, out to the ambient air. The heat flow through the gas from the absorber to the glass comprises conduction, convection and radiation. Conduction and convection are usually expressed in the same formula, because they are dependant on each other. In this case the flow will be called q_{cpg}, where q stands for heat transfer, c for convection (and conduction), p for absorber plate and g for cover glass. The radiation is called q_{rpg}, where r stands for radiation. The total top loss q_{top} will be:

$$q_{top} = q_{cpg} + q_{rpg} \tag{2}$$

There is the same heat flow from the glass to the ambient air. The nomenclature is q_{cga} and q_{rga}, i.e., the convection and the radiation respectively from the cover glass to ambient air. q_{cga} is, in reality, a convection film just over the glass where the heat is transferred out to the ambient air. As it is assumed to be uniform here, as in one dimensional heat transfer, q_{top} also will be:

$$q_{top} = q_{cga} + q_{rga} \tag{3}$$

Since there is still the same flow from the glass to ambient air as the flow from absorber to glass, (2) and (3) lead to:

$$q_{cpg} + q_{rpg} = q_{cga} + q_{rga} \tag{4}$$

All four components in (4) are dependant on temperatures. q_{cpg} and q_{rpg} are dependant on T_p and T_g while q_{cga} and q_{rga} are dependant on T_g and T_a. With a high T_g the sum of q_{cpg} and q_{rpg} will be low, but the sum of q_{cga} and q_{cga} will be high, With a low T_g it will be vice versa. Consequently, there must be a value for T_g where equilibrium exists to satisfy formula (4). T_g is assumed to be constant in the heat flow direction through the glass.

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

As already mentioned, the useful heat transfer is from the absorber plate to the heat transfer medium, q_u. Fig. 1 shows a cross section of a solar collector and Fig. 2 shows a blow-up of the section around the absorber plate connection to the tube. The mean path for the useful heat transfer is from the middle of one flange of a fin (i.e., ¼ of the tube spacing), through the bond and the cross section in/wall of the tube, and then through the convection layer inside the tube into the heat transfer medium. All heat transfer is by conduction except for the convection layer inside the tube. The influence of the change of resistance in the convection layer due to changes in temperature on the total heat loss is not of interest in this case since it is assumed to be small and therefore the layer is set as a constant heat resistance. q_u will be dependant on T_p, T_w and the heat resistance en

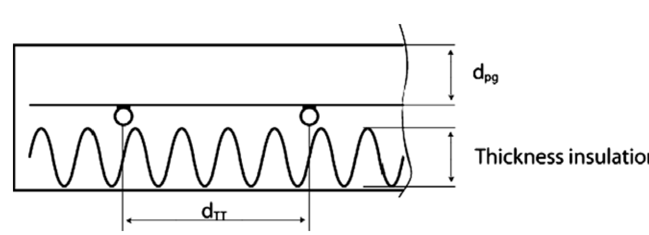

Fig. 1. Cross section showing tube spacing, d_{pg} and insulation.