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NEW TECHNIQUE TO EVALUATE THE OVERALL HEAT LOSS COEFFICIENT FOR A FLAT PLATE SOLAR COLLECTOR

NOVA TEHNIKA ZA OCENO CELOTNEGA KOEFICIENTA TOPLOTNE IZGUBE ZA PLOŠČATI SONČNI KOLEKTOR

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Keywords: flat plate solar collector, overall heat loss coefficient, heat removal factor

Abstract

Low-temperature solar systems mostly use flat plate solar collectors. Good design and correct dimensioning of a solar heat generator are based on precise knowledge of the characteristics of the flat plate solar collector on site. The present work considers a flat plate solar air collector, a flat plate solar water collector, and a flat plate solar water collector with air absorber cooling. The investigation intends to shed light on a procedure to determine the overall heat loss coefficient and the heat removal factor using recorded system temperatures, operating parameters, and environmental data. The Hottel-Whillier-Bliss equation gives the collector useful energy. This expression is used to generate a correlation for the collector efficiency through a linear fitting. We calculate the overall heat loss coefficient of the collector from the slope of the collector efficiency curve. However, we need to know the heat removal factor of the collector. In this study, we present a new technique to calculate the heat removal factor. Then we deduce the collector overall heat loss coefficient.
Results show that, very often, the overall heat loss coefficient for the flat plate solar air collector and the flat plate solar water collector determined with this new method is higher than the one calculated with the empirical formula proposed by Klein.

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However, the experimental overall heat loss coefficient for the flat plate solar water collector with air absorber cooling is smaller than the one calculated with the empirical formula proposed by Klein. The analysis shows that the overall heat loss coefficient determined with the new technique seems more realistic since all phenomena occurring during the heat transfer from solar irradiance incident on the absorber plate and transmitted to the transport fluid are considered.

Povzetek

Nizkotemperaturni sončni sistemi večinoma uporabljajo sončne kolektorje z ravno ploščo. Dobra zasnova in pravilne dimenzije solarnega generatorja toplote temeljijo na natančnem poznavanju značilnosti ploščatega sončnega kolektorja na lokaciji. Pričujoče delo obravnava ravno zračni ploščati sončni kolektor, vodni ploščati sončni kolektor in ploščasti sončni kolektor vode z zračnim absorberjem. Analiza v članku namerava osvetliti postopek določanja celotnega koeficienta toplotnih izgub in faktorja odvajanja toplote z uporabo zabeleženih temperatur sistema, obratovalnih parametrov in okoljskih podatkov. Enačba Hottel-Whillier-Bliss daje raziskovalcem koristne informacije. Ta izraz se uporablja za ustvarjanje korelacije za učinkovitost kolektorja z linearno vgradnjo. Skupni koeficient toplotnih izgub kolektorja izračunamo iz naklona krivulje izkoristka kolektorja. Poznati pa moramo faktor odvajanja toplote kolektorja. V tej študiji predstavljamo novo tehniko za izračun faktorja odvajanja toplote. Nato izračunamo koeficient celotne toplotne izgube kolektorja.

Rezultati kažejo, da je skupni koeficient toplotne izgube za ravno ploščasti zračni sončni kolektor in vodni ploščati sončni kolektor, določen s to novo metodo, večji od tistega, izračunanega z empirično formulo, ki jo je predlagal Klein.

Vendar je eksperimentalni skupni koeficient toplotne izgube za vodni ploščni sončni kolektor z zračnim absorberjem manjši od tistega, izračunanega z empirično formulo, ki jo je predlagal Klein. Analiza kaže, da se zdi skupni koeficient toplotne izgube, določen z novo enačbo, bolj realen, saj so upoštevani vsi pojavi, ki se pojavijo med prenosom toplote zaradi sončnega obsevanja, ki vpade na ploščo absorberja in se prenese v transportno tekočino.

1 INTRODUCTION

With the trend of decarbonization, many countries are moving toward using alternative energy sources such as solar, wind, biomass, geothermal, and others. For the supply of heat, solar water heating systems (SWHS) are widely used, especially in countries with good solar radiation. These systems are sufficiently mature to replace gas and other fossil fuel to supply heat in domestic or industrial applications. Solar air collectors are also integrated into many heating systems such as air space heating, drying applications, such as agricultural products, timber, biomass cultivation, waste biomass, building materials and desalination, and regulating microclimate in agricultural products storage facilities. Improvement of system efficiency and managerial flexibility of the energy obtained can expand the use of solar energy. In a scenario still characterized by strong growth in the installed solar collector capacity, *[1]*, even relatively small improvements may lead to a large increase in the overall energy production in absolute terms. Rigorous studies may also lead to adequate previsions.

For this reason, research and development into the optimization of the collector characteristics and solar field design play key roles. Flat plate solar collectors (FPSC) are the main and most used solar water heating systems component. It is made of an absorber plate covered above by a transparent sheet and surrounded by an isolating material. Incident solar radiations onto the absorber are absorbed by the latter and transferred as heat to a fluid flowing within the absorber. Therefore, a flat plate solar water collector is a special type of heat exchanger.

In contrast, a flat plate solar air collector (FPSAC) is a simple solar heating system with a low convective heat transfer coefficient between the absorber surface and the flowing air, resulting in a low heat transfer rate. Low thermal conductivity of air and high heat loss to the environment are the drawbacks of solar air collectors, *[2, 3].* Solar thermal flat plate collector performance depends very much on the absorbed solar radiation and the energy lost to the environment.

Many authors have examined various configurations to assess the influence of geometrical characteristics and operating conditions on the thermal efficiency of flat plate solar water collectors. Important parameters such as absorber thickness, riser position, the shape of the tube's cross-section, plate material, coating effect on absorptivity, cover transmissivity, fluid properties, and mass flow rate were investigated *[4]*. Other studies investigate cover absorber distance, heat transfer between absorber plate and cover sheet, the roughness of absorber plate surface, or use of transparent insulation material, *[5].*

Lowering heat losses from the absorber to the surrounding environment is an important issue for FPSC. The use of selective coating techniques was an innovation to reduce thermal radiation heat transfer from the absorber while maintaining high plate absorptivity, *[6].* Others investigated heat convection between the glass cover and absorber plate and the influence of the distance between these two components, *[7].*

Several other studies have been made on flat plate solar water heaters, *[8-11]*, mainly on the determination of the global heat loss coefficient and the heat removal factor of the collector. Klein, *[12]*, proposed a mathematical model, based on the theory of Bliss-Whillier, *[13, 14]*, which estimates the collector efficiency factor and the heat removal factor of the collector considering many simplifying assumptions. The performance of solar air collectors is usually affected by the low heat transfer coefficient between the air and the absorber plate, *[10].* The choice of the optimal collector depends on the temperature level required by the specific application and on the climatic conditions at the installation site. Therefore, in terms of efficiency, each collector displays features, making it most suitable to a given application. Several new applications of solar energy have appeared and intensified their use, as shown by works developed by Esen, *[15, 16].*

Efforts have been made to combine a number of the most important factors into a single formulation to have a mathematical model, which will describe the collector's thermal performance in a computationally efficient manner. Evaluating the thermal loss coefficients is the fundamental task to assess the flat plate solar collector performance. The FPSC global heat loss coefficient U_C (W/m^2.K) is the sum of the top loss (U_t), the bottom loss (U_b), and the edge loss (U_e) coefficients. This loss value is established between the collector and its surrounding by conduction, infrared solar radiation, and convection heat transfer. A good approximation of the FPSC global heat loss coefficient will lead to an effective solar water system design.

The present work aims to determine the overall loss heat coefficient of the flat plate solar collector using a new technique to evaluate the heat removal factor of the collector, taking into consideration experimental data to minimize the effect of assumptions made in other studies. This more realistic deduced overall heat loss coefficient will be compared to the one obtained by the Klein empirical formula, *[12].*

2 SYSTEM THEORY

Under steady-state conditions, the collector delivers a useful energy rate (Qu) equal to the rate of radiation energy absorbed by the collector minus the energy rate transmitted to its surroundings.

$$Q_u = A_c\left[I_c(\tau\alpha) - U_c\left(T_p - T_a\right)\right] = \dot{m}C_p[T_o - T_i] \quad (2.1)$$

After some reshuffling, it becomes, *[17]*:

$$Q_u = A_c FR[I_c(\tau\alpha) - U_c(T_i - T_a)] \quad (2.2)$$

with F_R, the heat removal factor that is determined by:

$$F_R = \frac{\dot{m}Cp(T_o - T_i)}{A_c[I_c(\tau\alpha) - U_c(T_i - T_a)]} \quad (2.3)$$

The collector efficiency is given by:

$$\eta = \frac{Q_u}{A_c I_c} = F_R\left[(\tau\alpha) - \frac{U_c(T_i - T_a)}{I_c}\right] \quad (2.4)$$

Which is equivalent to:

$$\eta = \frac{\dot{m}C_p(T_o - T_i)}{I_c A_c} \quad (2.5)$$

Equations (2.4) and (2.5) will let us draw the efficiency curve versus $\frac{(T_i - T_a)}{I_c}$ from experimental data.

Transforming equation (2.2), we obtain:

$$Q_u = F_R Q_u + A_c F_R U_c(T_p - T_i) \quad (2.6)$$

From equation (2.6), we obtain the temperature difference between the absorber temperature and the fluid inlet temperature:

$$T_p - T_i = \frac{Q_u}{A_c F_R U_C}(1 - F_R) \quad (2.7)$$

Taking into consideration equation (2.4), we obtain:

$$F_R = \frac{\eta I_c}{I_c \tau\alpha - U_C(T_i - T_a)} \quad (2.8)$$

From equations (2.4) and (2.7), we obtain:

$$A_c F_R U_C = \eta A_c I_c \frac{(1 - F_R)}{T_p - T_i} \quad (2.9)$$

Combining equation (2.9) and equation (2.4), we obtain:

$$\eta = F_R \tau\alpha - \eta(1 - F_R)\theta \quad (2.10)$$

in which

$$\theta = \frac{T_i - T_a}{T_p - T_i} \quad (2.11)$$

From equation (2.10), we derive F_R

$$F_R = \frac{\eta(1+\theta)}{\tau\alpha+\eta\theta} \tag{2.12}$$

We observe that F_R can be calculated based on physical parameters and operating parameters, including the absorber temperature, environment temperature and inlet fluid temperature. The collector efficiency coefficient is given by equation (2.4) and equation (2.5).

Considering the experimental data, we plot the efficiency coefficient versus $\frac{(T_i-T_a)}{I_c}$. From the slope $(F_R U_C)$ of the experimental curve η versus $\frac{(T_i-T_a)}{I_c}$ we determine the overall experimental loss coefficient knowing that F_R can be determined by equation (2.12).

For comparison, we calculate the heat loss coefficient using the widely used empirical equation given by Klein, *[12]*:

$$U_t = \frac{1}{\frac{N_g}{\frac{C}{T_p}\left[\frac{T_p-T_a}{N_g+f}\right]^{0.33}}+\frac{1}{h_w}} + \frac{\sigma(T_p^2+T_a^2)(T_p+T_a)}{\frac{1}{\varepsilon_p+0.05N_g(1-\varepsilon_p)}+\frac{2N_g+f-1}{\varepsilon_g}-N_g} \tag{2.13}$$

with

$$f = (1-0.04h_w+0.0005h_w^2)(1+0.091N_g) \tag{2.14}$$

$$C = 365.9(1-0.00883\beta+0.000129\beta^2) \tag{2.15}$$

$$h_w = \frac{8.6V^{0.6}}{L^{0.4}} \tag{2.16}$$

Due to a low bottom casing temperature, the radiative heat exchange with the surrounding can be neglected. Only the convection term is considered, *[17]*; thus, the heat transfer coefficient for the heat transmitted from the back to the surrounding is given by:

$$U_b = \frac{1}{\frac{t_b}{k_b}+\frac{1}{h_{c,b-a}}} \tag{2.17}$$

Similarly, the heat transfer coefficient for the heat transferred from the collector edges to the environment, still assuming that radiation heat transfer is negligible, is given by:

$$U_e = \frac{1}{\frac{t_e}{k_e}+\frac{1}{h_{c,e-a}}} \tag{2.18}$$

The overall heat loss coefficient for the collector is the sum of the heat loss coefficient for the top, the heat loss coefficient for the bottom, and the heat loss coefficient for the edges.

$$U_c' = U_t + U_b + U_e \tag{2.19}$$

Regarding the flat plate solar water collector with air absorber cooling, we replace the previous expression by the sum of heat rate for both fluids, and Ti takes the smallest inlet temperature from both fluids.

3 EXPERIMENTAL SET-UP

To determine the collector characteristics experimentally, primarily the overall heat loss coefficient, an experimental bench was set up (Fig.1). In the first configuration, a cross-section of the solar air collector shows a glass cover, an air gap between the glass cover and the absorber plate, the absorber plate fins, the air channel under the absorber, back and edge insulation and the casing.

In the second configuration, a cross-section of the solar water collector shows a glass cover, an air gap between the glass cover and the absorber plate, a grooved plate to which riser tubes are embedded, back and edge insulation and the casing. The third configuration is identical to the solar water collector to which an air channel is added between a finned absorber and the back insulation.

The used cover glass has low iron content in the three cases and is sealed to the casing with windshield silicone sealer. The absorber plate of 1 mm thickness is made of aluminium in the three cases. The absorber coating is matt black painting. In the second and third configurations, the embedded tube risers are made of copper. The back and the edges are insulated using rigid polyurethane panels.

The benches are instrumented with thermocouples type K to measure temperatures (cover, absorbing plate, inlet and outlet fluids, collector back, ambient, etc.). Air speed was measured using a hot wire CFM anemometer with 0.01m/s resolution. A Kipp and Zonen pyranometer mounted at the collector orientation is used to measure the global incident solar radiation. During the experiment, recorded parameters come under the required environment conditions (Table 1).

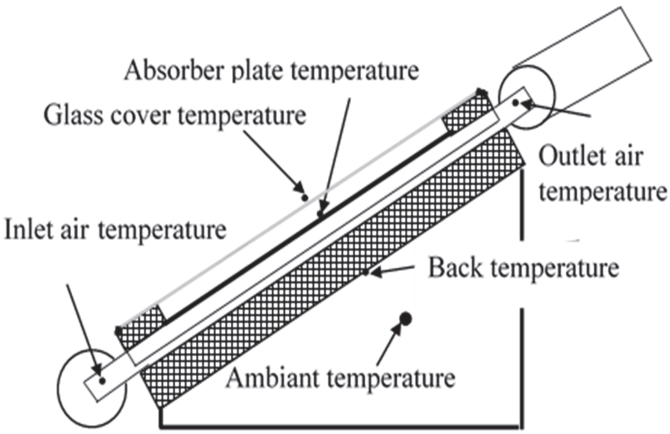
Physical properties of the materials were obtained from their documentation. All geometrical dimensions are given in the figures and Table 2. Airflow is generated with a centrifugal exhaust fan. In the case of the water collector, a water pump is used to perform forced circulation in the solar collector circuit. A flow meter is inserted in the water circuit in order to record the flow rate. Figure 2 shows the solar collectors constructed and used for the experiment.

Table 1: *Required environmental conditions (ASHRAE 93)*

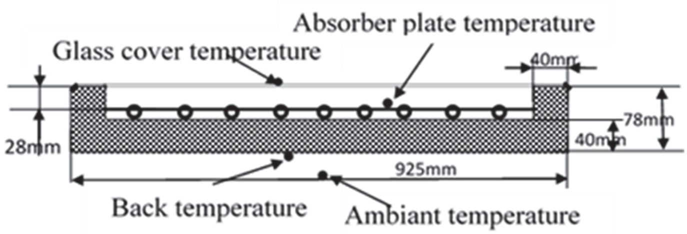
Variable	Absolute limits
• Total solar irradiance normal to sun (W/m^2)	• 790 (minimum)
• Diffuse fraction (%)	• 20 (maximum)
• Wind speed (m/s)	• 2.2<u<4.5
• Incidence angle modifier	• 98%<normal incidence value <102%

Table 2: *Physical characteristics*

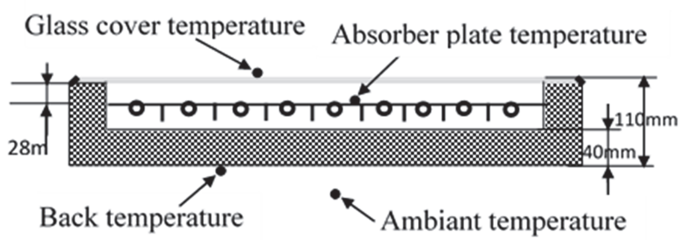
Element collector designation	Dimension (mm)
Riser diameter	12
Header diameter	18
Flat plate collector length	1920
Number of riser tubes	09
Fin height	28



(a) Solar air flat plate solar collector



(b) Solar water flat plate solar collector



(c) Solar water flat plate collector with air absorber cooling

Figure 1: *Cross-sections of the different configurations*

Solar air flat plate solar collector

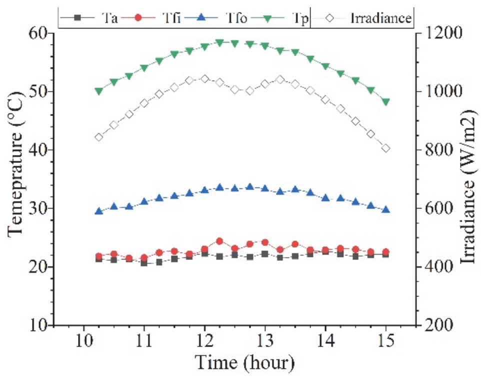
Solar water flat plate solar collector

Solar water flat plate collector with air absorber cooling

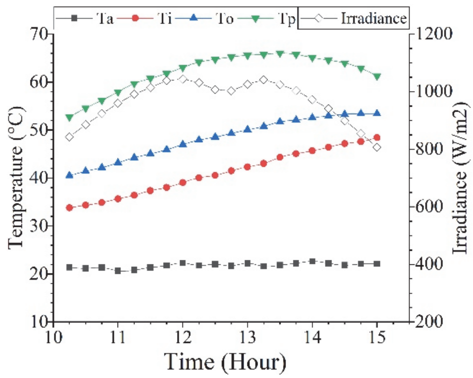
Figure 2: *Pictures of the solar collectors set up*

4 RESULTS AND DISCUSSION

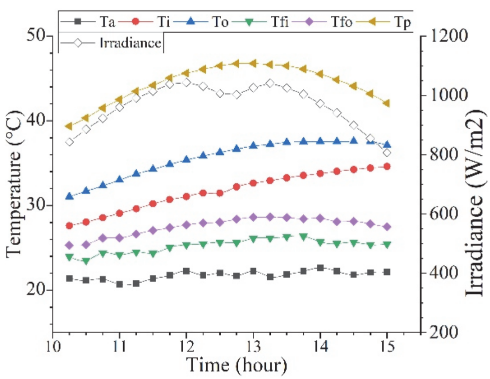
Figure 3 shows various parameters used to calculate the heat removal factor according to equation (2.12) and the drawing of the efficiency curves for the different collectors.
As it might be noticed, all inlet parameters are practically stable during this measurement period (10h- 15h), which lets us consider the quasi-stationary state. Incident solar radiation is over 800 W/m^2.



(a) Air collector



(b) Water collector



(c) Water collector with air absorber cooling

***Figure 3:** Temperatures and solar irradiance for the different configurations*

Figure 4 shows the heat removal factors versus the absorber plate temperatures. We observe a small fluctuating value of F_R around a horizontal line, which might be considered a reference value for this parameter. This indicates that the absorber temperature has a slight influence on F_R. In contrast, the heat transfer coefficient between the absorber and the fluid flow significantly impacts F_R. This confirms that F_R is an important design parameter as it is a measure of thermal resistance encountered by the absorbed solar radiation in reaching the collector fluid.

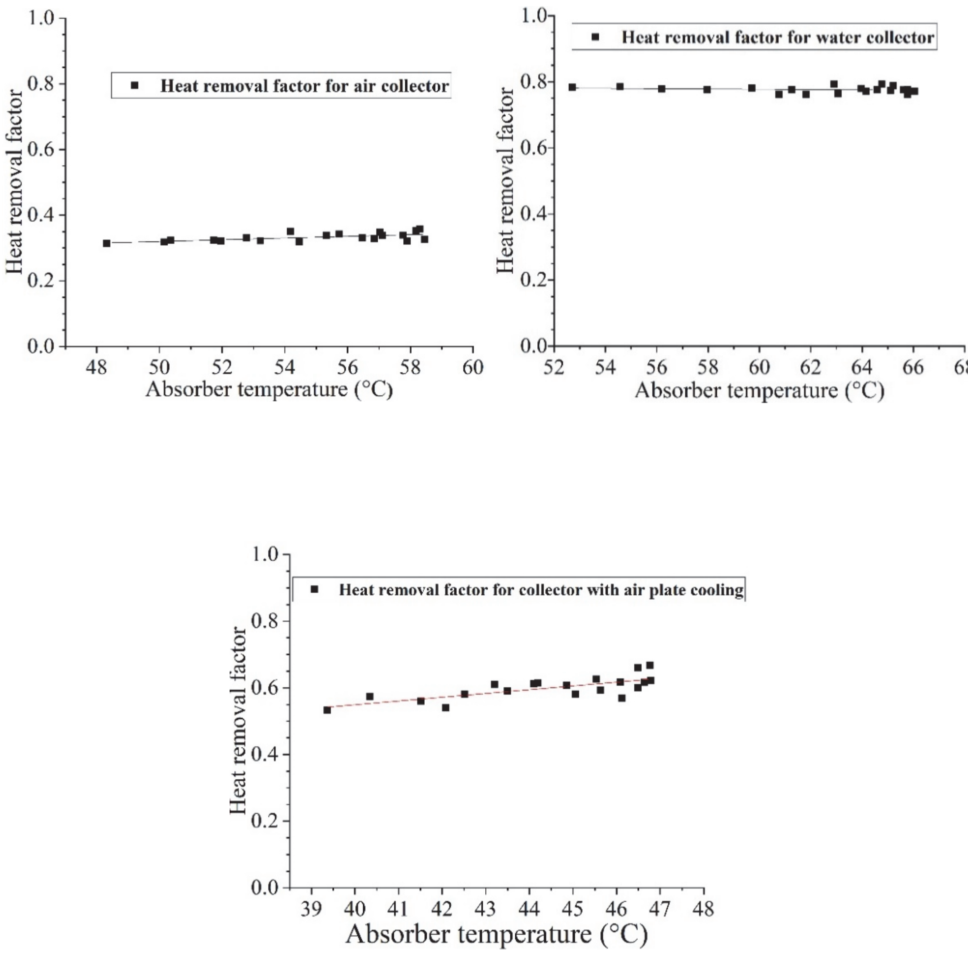


***Figure 4:** Heat removal factor for the different configurations*

Figure 5 shows collector efficiencies (η) versus $\frac{(T_i - T_a)}{I_g}$, using experimental data for each configuration. The efficiency coefficient is determined for a mass flow rate of 0.05 kg/s for the air in the solar air collector, 0.11 kg/s for the air in the solar water collector with air absorber cooling, and 0.027 kg/s for water in the solar water collector. The reported values are calculated using experimental data and equation (2.5). It is to recall that, in the case of the configuration of water solar collector with air absorber cooling, Q_u is the sum of the energy obtained by the flowing water and the energy taken away by the cooling air.

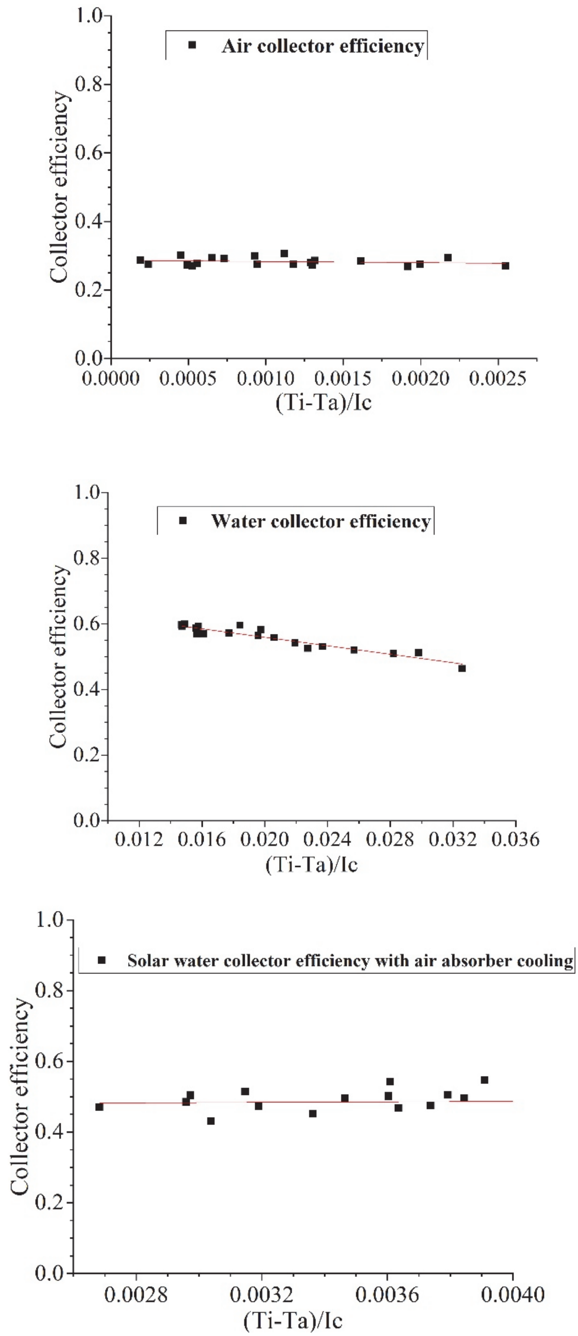


Figure 5: *Efficiency coefficients for the different configurations*

The fitting curve lets us determine the slope. Using the slope and the heat recovery factor calculated with equation (2.12), we can evaluate each collector's average overall heat loss coefficient. Considering the instantaneous data (efficiency and heat recovery factor), we can also determine the collector's instantaneous overall heat loss coefficient.
Figures 6, 7, and 8 show the instantaneous overall heat loss coefficients calculated from Klein's empirical formula and the one deduced experimentally for each configuration versus the absorber temperature. We observe that the overall heat loss deduced experimentally is larger for air collector and water collector. The result is more realistic since it is deduced experimentally.
It considers the most influential parameter, which is absorber temperature and all phenomena occurring in the process of heat transmission in any direction. The efficiency coefficient of the solar air collector remains almost constant, regardless of the incident solar flux. This is accompanied by a decrease in the heat loss coefficient. The efficiency coefficient of solar water collector decreases with high irradiance, although there is a decrease in the heat loss coefficient.

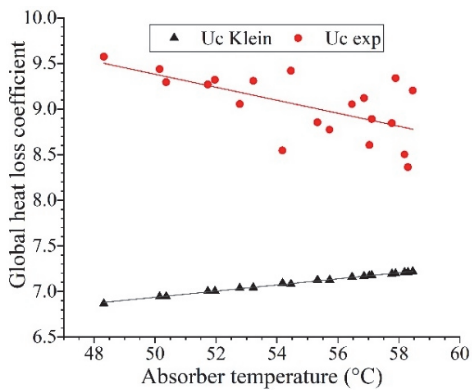


***Figure 6:** Global heat loss coefficient for solar air collector*

In the case of a solar water collector with air absorber cooling, the efficiency coefficient is stable at 0.5. The air absorber cooling significantly influences the heat loss coefficient since it decreases with an increase in absorber temperature. In the three cases, the heat loss coefficient calculated with Klein's empirical formulae increases with absorber temperature increase.
In contrast, the experimental heat loss coefficient decreases for solar air collector and water solar collector with air absorber cooling and uniform for solar water collector with increasing absorber temperature. The average efficiency coefficient of the solar air collector is smaller than the one of the solar water collector with air absorber cooling (Table 3).

Table 3: Experimental characteristics for the three solar collector configurations (Uc in W/m^2.K)

Collector	η	Slope	F_R	Uc_exp	Uc_klein
Air collector	0.283	3.341	0.332	10.063	7.099
Water collector	0.578	6.459	0.776	8.323	7.328
Water collector with air absorber cooling	0.485	3.561	0.599	5.945	6.716

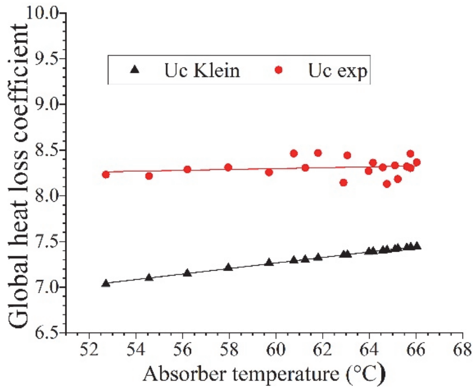


Figure 7: *Global heat loss coefficient for solar water collector*

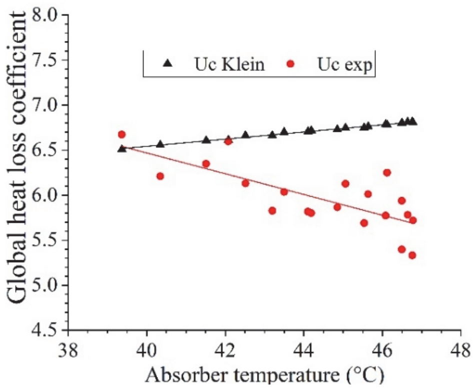


Figure 8: *Global heat loss coefficient for water collector with absorber back cooling*

5 CONCLUSION

The evaluation of the thermal loss coefficients is a fundamental task to assess the flat plate solar collector performance, whether in a single solar domestic hot water or a solar collector heat generator or to heat a building or drying system. A new technique for evaluating a global heat loss coefficient of flat plate solar collector was implemented based on experimental data. This technique considers the operating parameters such as temperatures, operating conditions, and geometric characteristics of the collector. The heat removal factor is first deduced using a new formulation based on the most influential parameter: the absorber temperature. From the slope of the efficiency coefficient fitting line, the global heat loss coefficient is determined. The result analysis and the comparison with thermal loss coefficients obtained from Klein's empirical formula indicate that this technique generates a more realistic value for the global heat loss coefficient. The coefficient deduced with this procedure will let the designer and the solar system developer be more confident in determining the dimensions and the expected power. The rapid determination of $F_{R,}$ which is an important design parameter, lets the developer appreciate thermal resistance encountered by the absorbed solar radiation in reaching the collector fluid. A precise knowledge of the characteristic of the collector lets the designer correctly establish the dimensions of any solar heating system.

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Nomenclature

A_c	Total collector aperture area (m^2)
$h_{c,b-a}$	Convection heat loss coefficient from back to ambient (W/m^2 .K)
$h_{c,e-a}$	Convection heat loss coefficient from edge to ambient (W/m^2.K)
h_w	Wind heat transfer coefficient (W/m^2.K)
I_c	Irradiance on the collector (W/m^2)
k_b	Conductivity coefficient of back insulation (W/m.K)
k_e	Conductivity coefficient of edge insulation (W/m.K)
L	Collector length (m)
$\dot{m}$	Mass flow rate of fluid (kg/s)
N_g	Number of glass covers
Q_u	Rate of useful energy delivered by the collector (W)
T_a	Average ambient temperature (° C)
t_b	Thickness of back insulation (m)
t_e	Thickness of edge insulation (m)
T_{fi}	Air collector inlet temperature (°C)
T_{fo}	Air collector outlet temperature (°C)
T_i	Water collector inlet temperature (°C)
T_o	Water collector outlet temperature (°C)
T_p	Average temperature of the absorbing surface (°C)
U_b	Bottom heat loss coefficient (W/m^2.K)
U_e	Edges heat loss coefficient (W/m^2.K)
U_c	Solar collector overall heat loss coefficient (W/m^2.K)
U_t	Top heat loss coefficient (W/m^2.K)
V	Wind velocity (m/s)