COMPARATIVE ANALYSIS BETWEEN A PARABOLIC TROUGH COLLECTOR AND A COMPACT LINEAR FRESNEL REFLECTOR TO BE USED AS DIRECT GENERATOR OF AN ADVANCED ABSORPTION COOLING SYSTEM

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Abstract

A comparative analysis between a parabolic trough collector (PTC) and a compact linear Fresnel reflector (CLFR) is presented. A mathematical model for each concentrator is developed, a computational algorithm in order to solve the rigorous mathematical model is proposed, and a numerical simulation exploring the critical variables that define the operational performance of both concentrators is done. According to the obtained results, it can be concluded that the parabolic trough solar collector has important advantages that makes it the best option to be used as a direct generator of an advanced solar-GAX cooling system. Its thermal efficiency is about 10.5% higher than that of the Fresnel concentrator, therefore, the energy demand of the cooling cycle could be satisfied with a smaller equipment, if this concentrator is used.

1. Introduction

In recent years, an increasing interest in research and development of sustainable cooling technologies able to compete or even replace conventional vapour compression systems have been observed, because of the drawbacks of the latter (very high power required for their operation and serious environmental problems associated with the refrigerants used in this technology).

Absorption cooling systems thermally activated by solar collectors are considered as a very promising alternative since their main energy source is obtained from renewable resources, and at the same time, they use natural refrigerants that do not contribute to the global warming and do not damage the ozone layer. Two technologies converge in these systems: Absorption cooling technology and solar concentrating technology. Throughout the years, absorption cooling systems have evolved, and at the moment, the cycles with internal heat exchange between the absorber and the generator (GAX) are the most attractive due to their higher levels of efficiency and lower number of components compared to other advanced cycles. Regarding the concentrating solar technology, originally, PTC and CLFR were designed to heat a fluid, but now they are used to generate vapour directly in their receptors [18]. This technology is known as *direct steam generation* (DSG) and is mainly used in electric power generation [5]. Kalogirou et al. [9] proposed a direct steam generation system by using flash evaporation, where steam is generated after warming up water at high pressure, and later the pressure is reduced in an expansion tank. By doing this, only one phase is present in the receiver of the PTC, therefore, the problem related to the thermal stress caused by the temperature gradient, when two phases are present is avoided. Flores and Almanza [6] studied a PTC with a bimetallic receiver for direct steam generation, looking to solve the problem of deformation caused by the non-uniform heating appearing due to the existence of a stratified flow in two phases in the receiver. The Fresnel solar concentrator was invented by Francia [7] and in the last years, several

studies have been conducted in order to evaluate its technical and economic feasibility to be used in systems of direct steam generation for electricity production [11, 12]. Velázquez et al. [17] proposed a CLFR to generate steam directly in its receiver and to activate an advanced absorption cooling system. They reported that a Fresnel concentrator is a good option to be used as direct generator of an advanced absorption cooling system in view of the fact that, the process can be done at high temperatures beneficial to the cooling cycle. In addition, the efficiency of the concentrator is not affected in any significant way.

Based on the advantages that represent the direct steam generation in solar concentrators and the attractiveness of an advanced absorption cooling system activated directly by a solar concentrator, in this work, a mathematical model of a PTC and a CLFR is developed. A numerical simulation for each one of them exploring the most important variables that define its operational behavior and a comparative analysis to determine the best option to be used as direct generator of an advanced absorption cooling system is carried out.

2. Physical Description

Figure 1(a) shows a schematic diagram of a PTC. It consists of a reflective surface with a parabolic shape that reflects all the normally incident energy over its aperture area towards a receptor tube placed on the focus line. The receptor tube is surrounded by a glass cover with air in the annulus in order to reduce the convection loses between the receptor and the surroundings. Figure 1(b) shows the CLFR scheme. It is composed of flat mirrors set placed at the correct distance of the center to avoid shading between them, and at specific angle to reflect the beam incident normally over its aperture area towards a common focus, where a receptor tube surrounded by a glass cover with air in annulus is placed.
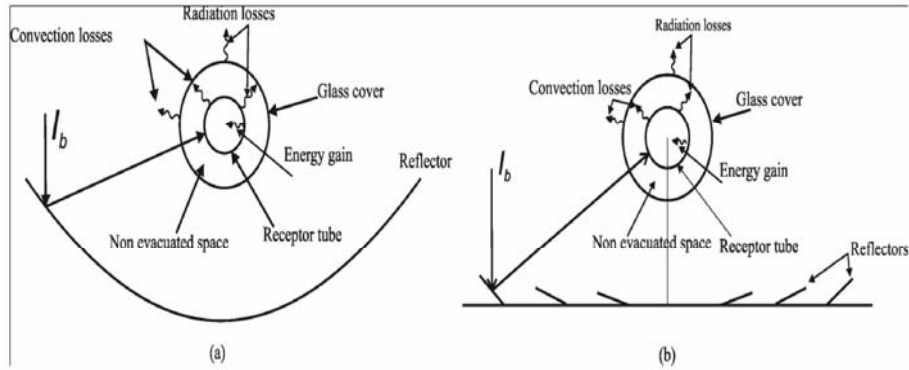
Figure 1. Schematic diagram of the systems evaluated.

3. Operative Description

The solar concentrators studied are used as direct generators of an advanced solar cooling system; they receive a saturated ammonia water mixture at 186°C, 18 bar, 6.73% ammonia, and 0.0197kg/s. When the mixture receives latent heat along the receptor tube, a phase change takes place and at the end of the receptor, the mixture is in two phases for its posterior separation and reintegration to the cooling cycle.

4. Methodology

Figure 3 shows the algorithm developed for the resolution of the mathematical model and the study of the different parameters that define the operational performance of the solar concentrators. Using this tool, each one of the concentrators was dimensioned in order to satisfy the thermal requirements of an advanced absorption cooling system of 10.5kW capacity. The following assumptions have been made in the simulation study.

   ● Both concentrators are 3m width.

   ● Both concentrators have a perfect solar tracking system in two axes.

   ● The aperture plane of both concentrators is normal to the incident solar beam radiation.

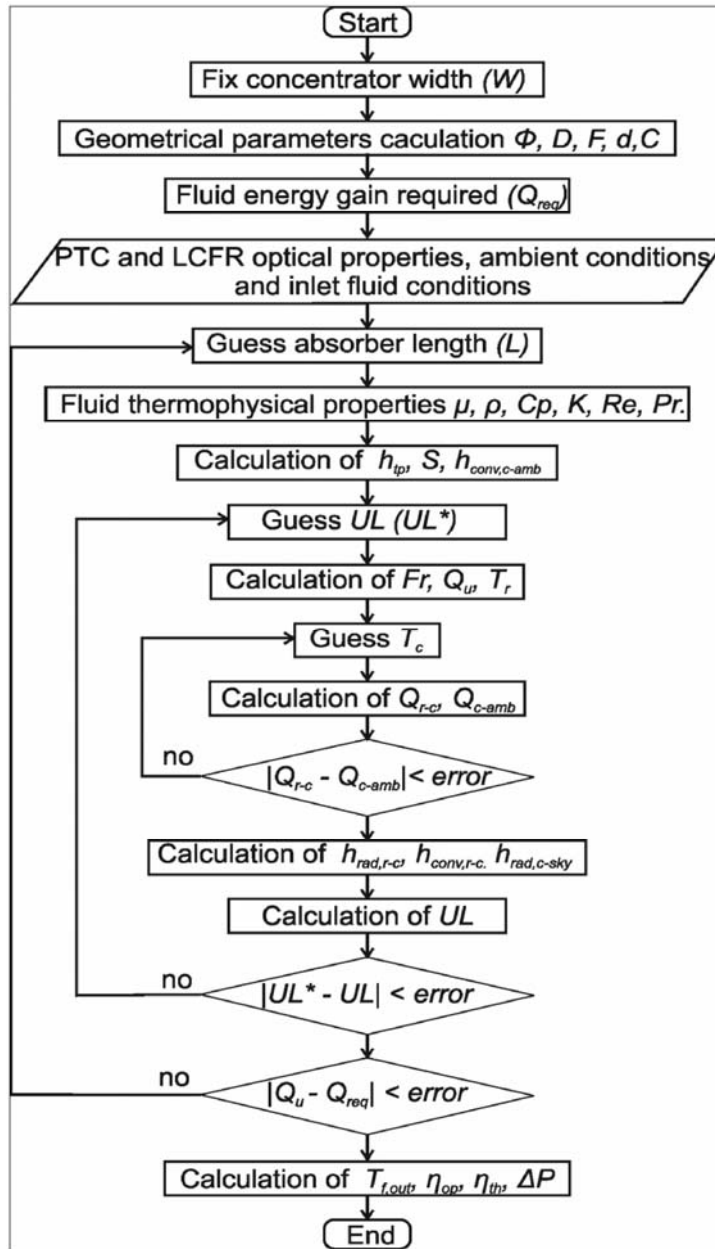● Width of receptor tube diameter of the CLFR is equal to the mirrors width.

Figure 2. Computational algorithm used in the design of the systems.

4.1. Optical and geometrical analysis

Position and angle of the first mirror in the CLFR are calculated by the following equations [10]:

$$d_1 = \frac{W}{2} + f \tan \xi, \tag{1}$$

$$\theta_1 = \frac{1}{2} \tan^{-1} \left(\frac{d_1 + \frac{W}{2} \cos \theta_1}{f - \frac{W}{2} \sin \theta_1} \right). \tag{2}$$

Position and angle of the second to "n" mirrors are calculated by an iterative method by using the following equations:

$$G_n = W \sin \theta_{n-1} \tan(2\theta_n + \xi), \tag{3}$$

$$d_n = d_{n-1} + W \cos \theta_{n-1} + G_n, \tag{4}$$

$$\theta_n = \frac{1}{2} \tan^{-1} \left(\frac{d_n + \frac{W}{2} \cos \theta_n}{f - \frac{W}{2} \sin \theta_n} \right). \tag{5}$$

For a PTC, the minimal diameter required in the receptor tube in order to receive all the reflected rays from the reflective surface, is calculated by [4]:

$$D = \frac{W_c \sin 0.267}{\sin \phi}, \tag{6}$$

where the aperture angle is defined by:

$$\phi = \tan^{-1} \left[\frac{8 \left[\frac{f}{W_c} \right]}{16 \left[\frac{f}{W_c} \right]^2 - 1} \right], \tag{7}$$

geometrical concentrating ratio is calculated by:

PTC:

$$C = \frac{W_c - D_{r,ext}}{\pi D_{r,ext}}. \tag{8}$$

CLFR:

$$C = \frac{\sum_{i=1}^{N} \cos \theta_N}{\pi D_{r,ext}}.$$

(9)

Solar radiation absorbed by the area of the receptor tube in both concentrators, is calculated by the following expression [1]:

$$S = I_b \rho_o \tau \alpha \gamma K_a.$$

(10)

Optical efficiency for both concentrators is defined in the following form:

$$\eta_{op} = \frac{S}{I_b}.$$

(11)

4.2. Thermal analysis

Considering the similitude in the receptor tube system and that both concentrators use the same working fluid, the mathematical model of the heat transfer process is the same in both concentrators, therefore, useful heat gain is calculated by [4]:

$$\dot{Q}_u = Fr(W - D_{r,ext})L\left[S - \frac{U_L}{C}(T_{f,ent} - T_{amb})\right].$$

(12)

Overall heat transfer losses coefficient depends on the individual heat transfer coefficients and is defined by:

$$U_L = \left[\frac{A_r}{(h_{conv,c-amb} + h_{rad,c-\text{cie lo}})A_c} + \frac{1}{(h_{rad,r-c} + h_{conv,r-c})}\right]^{-1}.$$

(13)

Radiation heat transfer coefficient between receptor tube and glass cover, is estimated by the following correlation:

$$h_{rad,r-c} = \frac{\sigma(T_r^2 + T_c^2)(T_r + T_c)}{\dfrac{1 - \varepsilon_r}{\varepsilon_r} + \dfrac{1}{F_{r-c}} + \dfrac{(1 - \varepsilon_c)D_{r,ext}}{\varepsilon_c D_{c,int}}}.$$

(14)

Convection heat transfer coefficient between receptor tube and glass cover, is calculated by:

$$h_{conv,r-c} = \frac{2k_{eff}}{D_{r,ext} \ln(D_{c,int} / D_{r,ext})},$$ (15)

where k_{eff} is obtained from [15] correlation:

$$\frac{k_{eff}}{k} = 0.317(Ra^*)^{1/4},$$ (16)

Ra^* is the modified Rayleigh number and is calculated by:

$$(Ra^*)^{1/4} = \frac{\ln(D_{c,int} / D_{r,ext})}{b^{3/4}\left[\dfrac{1}{D_{r,ext}^{3/5}} + \dfrac{1}{D_{c,int}^{3/5}}\right]^{5/4}} Ra^{1/4},$$ (17)

$$b = \frac{D_{c,int} - D_{r,ext}}{2}.$$

Radiation heat transfer coefficient between glass cover and surroundings, is calculated by [14]:

$$h_{rad,c-sky} = \sigma\varepsilon_c(T_{sky}^2 + T_c^2)(T_{sky} + T_c).$$ (18)

Convection heat transfer coefficient between glass cover and surroundings, is estimated by the correlation [8].

$$h_{conv,c-amb} = CRe^n \frac{k_{air}}{D_{c,ext}},$$ (19)

where C and n are constants defined as follows:

for $40 < Re \le 4000$ $C = 0.615, n = 0.466,$

for $4000 < Re \le 40000$ $C = 0.174, n = 0.618,$

for $40000 < Re \le 400000$ $C = 0.0239, n = 0.805.$

Convection heat transfer coefficient from internal wall of the receptor tube toward two phase ammonia water mixture, is calculated by using Mishra et al. [13] correlation as follows:

$$h_{tp} = 65h_l\left(\frac{1}{x_{tt}}\right)^n Bo^m,$$ (20)

where $n = 0.5$ and $m = 0.15$ [16], x_{tt} is   the   Lockhart-Martinelly parameter defined by:

$$x_{tt} = \left[\frac{1 - X_g}{X_g}\right]^{0.9}\left[\frac{\rho_g}{\rho_l}\right]^{0.5}\left[\frac{\mu_l}{\mu_g}\right]^{0.1},$$ (21)

h_l is the convection heat transfer coefficient for the liquid phase, and it is calculated by the [3] correlation:

$$h_l = 0.023Re^{0.8}Pr^{0.4}\frac{K}{D_{r,\,int}},$$ (22)

Bo is the boiling number defined as follows:

$$Bo = \frac{q}{G\lambda}.$$ (23)

Global efficiency for both collectors is calculated by:

$$\eta_{th} = \frac{\dot{Q}_u}{A_{ref}I_b}.$$ (24)

Fluid temperature at the end of the receptor tube, is calculated by using an iterative method guessing temperature values until the next balance is reached:

$$\dot{Q}_u = \dot{m}(\Delta H),$$ (25)

ΔH is the difference between the inlet and outlet enthalpy.

Transport properties are evaluated by using Conde correlations [2]; thermodynamics properties are estimated by Ziegler and Trepp correlations [19].

5. Results and Discussion

A numerical simulation study in order to size, predict the operational performance and to select the best option to be used as direct generator of an advanced absorption solar cooling system between a PTC and a CLFR has been carried out. Materials utilized and dimensions obtained were selected in order to obtain maximum optical and thermal efficiency considering the following conditions:

● Thermal energy demanded by an advanced absorption cooling cycle (12.23kW).

● 3m width for both concentrators.

● Meteorological conditions of Mexicali B. C. Mexico.

● Receptor tube diameter and glass cover diameter commercially available.

Table 1 shows the most important design variables and dimensions for each one of the concentrators. In the study of the solar concentrating technology, one of the most important aspects is the optical and geometrical study. Figure 3 demonstrates how some of the geometrical parameters such as focus distance, receptor tube diameter, and geometrical concentrating ratio were chosen for both concentrators. For the PTC (3a), the maximum concentration ratio is 32.9 with a focus distance of 0.75m, while for the CLFR (3b), the concentration ratio is 14.2 with a focus distance of 2m. The difference in the concentration ratio is due to the fact that PTC takes advantage of all the aperture area, while in the CLFR, the mirrors are separated and as a consequence, the effective area is lesser than in the PTC. By means of similar parametric studies, the number of mirrors in the CLFR, the receptor tube diameter, the glass cover diameter, and the rest of dimensions were specified as shown in Table 1.

Table 1. Design and working conditions of the systems studied

| Parameter	CLFR	PTC
Mass flow rate	0.0197kg/s	
Inlet temperature	184.98°C	
Inlet pressure	18 bar	
Inlet ammonia concentration	7.18%	
Ambient temperature	39°C	
Receptor internal diameter	0.05250m	0.0199m
Receptor external diameter	0.06032m	0.0267m
Cover internal diameter	0.08m	0.045m
Cover external diameter	0.085m	0.05m
Width	3m	
Length	8.71m	7.18m
Receptor absorbance	0.9 (dimensionless)	
Shape factor due to inexact concentrator orientation	1.0	1.0
Number of mirrors	46	-
Focal distance	2m	0.75m
Mirror reflectivity	0.92 (dimensionless)	
Receptor emittance	0.35 (dimensionless)	
Cover emittance	0.88 (dimensionless)	
Receptor conductivity (carbon steel)	55W/m K	
Air velocity	2.3m/s	
Geometrical concentration ratio	14.22	32.91
Solar beam radiation	900W/m^2	
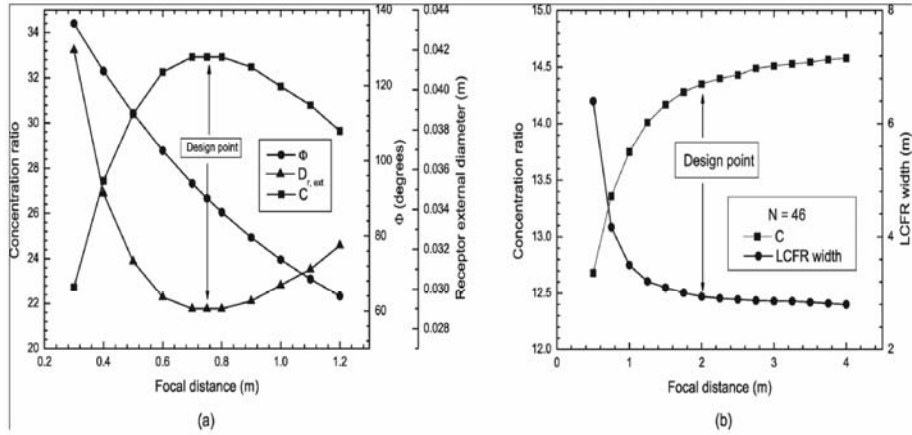
Figure 3. Geometrical parameters selection for the solar collectors.

Figure 4 illustrates how the useful heat demanded by the cooling system is satisfied in different length for each concentrator. In the case of the PTC, 7.18m are needed, while the CLFR requires 8.71m. This is due mainly to the difference in the receptor tube diameter in each concentrator, the PTC has a receptor diameter of 0.0267m, which is about half than that of the CLFR. When the diameter is bigger, the heat transfer area increases causing higher thermal losses to the surroundings. This effect is also observed in the individual heat transfer coefficients, which depend of the transversal area and heat transfer area. The higher geometrical concentration ratio in the PTC is also reflected in the higher useful heat gain compared with the CLFR. As shown in Figure 4, the outlet quality of the mixture is higher too in the PTC, because if more heat is transferred, more ammonia vapour is released.
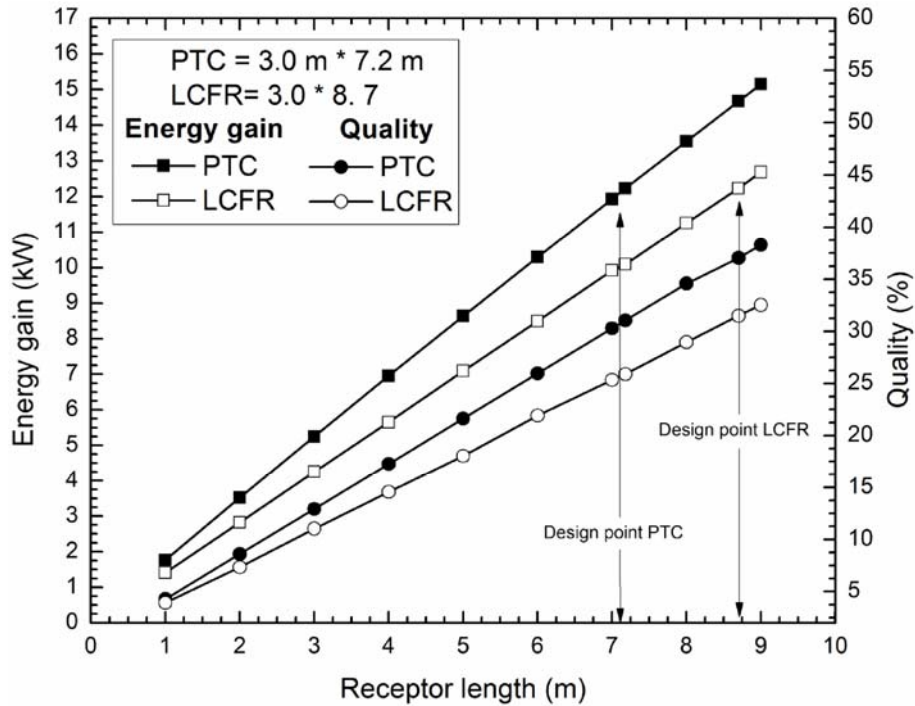
Figure 4. Energy gain and output quality against receptor length comparison.

Figure 5 shows useful heat transfer and thermal efficiency against different levels of solar beam radiation for both concentrators, as shown. There is a difference between the efficiency of the concentrators, that of the PTC being 10.5% higher than that of CLFR. As mentioned above, this is due to the difference of heat transfer area, in the individual heat transfer coefficients and in the geometrical concentration ratio in both concentrators.

The performance of the solar-GAX cycle directly activated by the solar concentrators and the efficiency of both concentrators varying the generator temperature are shown in Figure 6. It can be seen that unlike conventional solar thermal cooling systems activated by flat collectors, in this configuration, with generator temperature increase, the efficiency of the cooling cycle is incremented without affecting in a significant way the collector efficiency.
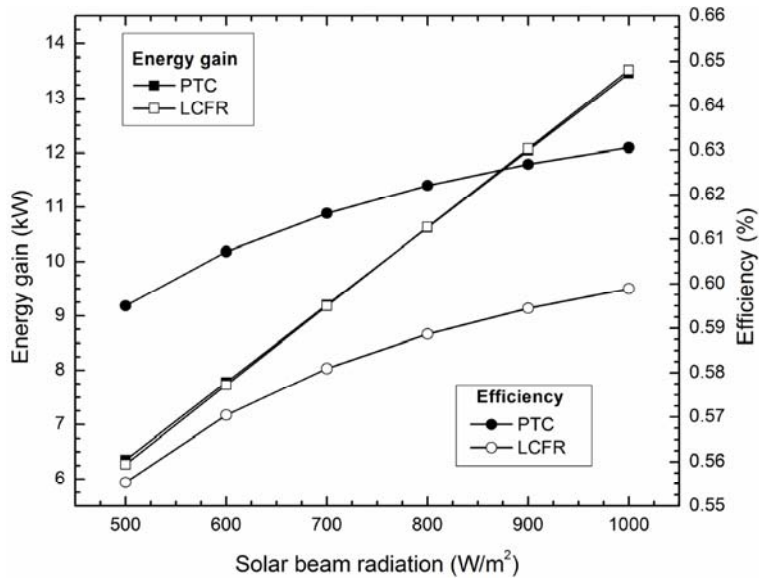
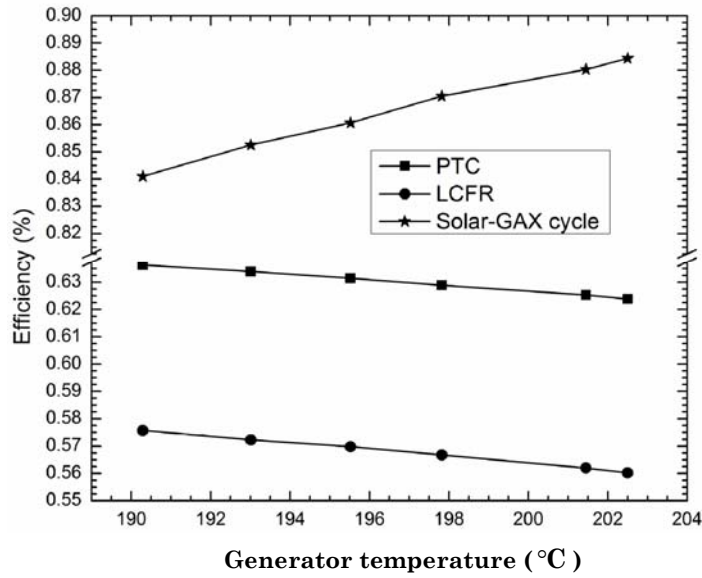**Figure 5.** Energy gain and efficiency against solar beam radiation comparison.

Figure 6. Solar-GAX and collector efficiencies against generator temperature.

6. Conclusion

A comparative analysis between a PTC and a CLFR in order to determine which one of these would be the best choice to be used as direct generator of an advanced absorption cooling cycle has been carried out. According to the obtained results, it can be concluded that both concentrators can provide the quantity and quality of thermal energy demanded by the cooling cycle, although, the PTC has several advantages that makes it the best option. Its thermal efficiency and geometric concentration ratio are 10.5% and 131%, respectively, higher than that of the CLFR, therefore, a smaller system is required to satisfy the energy demanded by the cooling cycle (PTC = $3.0m^* 7.18m$; CLFR = $3.0m^*$ $8.71m$). On the other hand, by using a solar concentrating collector as a direct generator in an advanced cooling cycle, allows working at higher temperature favouring the COP of the cooling cycle without affecting in any significant way the efficiency of the collector.

Acknowledgement

Nomenclature

A_c Internal heat transfer cover area $[\text{m}^2]$.

A_r External heat transfer receptor area $[\text{m}^2]$.

Bo Boiling number [dimensionless].

Cp Specific heat at constant pressure $[\text{Jkg}^{-1}\text{K}^{-1}]$.

C Geometrical concentration ratio [dimensionless].

D Diameter [m].

F Focal distance [m].

G Flow mass velocity $[\text{kg}\,\text{m}^{-2}\text{s}^{-1}]$.

H Enthalpy $[\text{Jkg}^{-1}]$.

h Heat transfer coefficient $[\text{W}\,\text{m}^{-2}\text{K}^{-1}]$.

I_b Solar beam radiation $[\text{W}\,\text{m}^{-2}]$.

K Thermal conductivity $[\text{W}\,\text{m}^{-1}\text{K}^{-1}]$.

K_a Incident angle modifier [dimensionless].

L Length [m].

$\dot{m}$ Mass flow rate $[\text{kg}\,\text{s}^{-1}]$.

Pr Prandtl number [dimensionless].

Q Heat flow per unit area $[\text{W}\,\text{m}^{-2}]$.

$\dot{Q}_u$ Energy gain [W].

Ra Rayleigh number [dimensionless].

Re Reynolds number [dimensionless].

S Solar absorbed energy per unit area [dimensionless].

T Temperature [°C].

U_L Overall heat transfer losses $[\text{W}\,\text{m}^{-2}\text{K}^{-1}]$.

W Mirror width [m].

W_c PTC width [m].

Subscript

r Receptor

amb ambient

c　　　Cover

conv　　convective

ent　　Inlet

ext　　external

f　　　Fluid

int　　internal

l　　　Liquid

op　　Optical

rad　　radiative

tp　　Two phases

g　　　Vapour

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