International Journal of Engineering Research & Technology (IJERT)
ISSN: 2278-0181
Vol. 4 Issue 02, February-2015

Mathematical Modelling of a Solar Parabolic Trough Collector

Coupling with an absorption machine

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***Abstract*— The depletion of fossil fuels and the irreversible damage caused by their combustion is one of the biggest current problems of our planet. Buildings are among the most energy-intensive in the world with more than 30% of the energy used for heating, cooling and domestic hot water production. The use of solar energy for air conditioning is a promising concept, since the cooling needs coincide mostly with the availability of solar radiation. In addition, solar air conditioning thermal systems permit reducing electricity consumption, using harmless refrigerants, reducing noise and simplifying problematic in terms of electricity infrastructure in rural areas.**

The contribution presents a mathematical modeling and a numerical simulation of a Solar Parabolic Trough Collector, *SPTC*. Validated according to the final test results of the Schott HCE (Sandia laboratories), the model is implemented as a new type (type240) in TRNSYS 17 library, thus it can be used in projects involving solar concentration (Solar Concentrating Power, Process Heat...). A solar air conditioning example is studied under TRNSYS environment.

Keywords—Parabolic trough collector; solar cooling; absorption machine; TRNSYS; mathematical modelling

LIST OF SYMBOLS

Symbol	Definition	Unit
A	Area	m^2
a	Aperture	m
C	Concentration ratio	-
C	specific heat	J/(kg K)
D	Diameter	m
F	View factor	-
f	Focal length	m
G	Irradiation	W/m^2
h	Heat transfer coefficient	$W/(m^2K)$
h	concentrator height	m
h_d	Annulus gas conduction heat transfer coefficient	$W/(m^2K)$
k_a	Thermal conductivity of the annulus gas at standard temperature and pressure	$W/(m^2K)$
l	Collector length	m
M	Mass	kg
Nu	Nusselt number	-
Pr	Prandtl number	-
p_v	Annulus gas pressure	N/m^2
Q	Energy per unit time	W
R	Reflectance	-
Re	Reynolds number	-
T	Temperature	K
V	Wind velocity	m/s
x, y	Coordinates	m
Greek		
α	Absorptance	-
δ	Molecular diameter	m
ε	Emittance	-
γ	Ratio of specific heats	-
φ	Insolation rate	-
σ	Stefan Boltzmann constant	$W/(m^2K^4)$
τ	Transmittance	-
ϑ	Free mean path	m
Subscripts		
a	Ambient, aperture, atmosphere	-
b	Absorber	-
e	External	-
f	Fluid	-
g	Glass	-
i	Internal	-
r	Reflective	-
s	Sky	-

I. INTRODUCTION

Estimates of global energy consumption from 2003 to 2030 indicate an increase of about 71% [1]. According to the International Energy Agency, most of this energy is produced from fossil fuels [2]. At present, fossil fuels are still abundant, but it is now clearer than ever that fossil fuel resources are being depleted at an incredible speed. In addition, their combustion contributes significantly to greenhouse effect and indirectly to depletion of the ozone layer, where high altitude emissions might be a specific problem [3]. Emissions from fuel combustion increased significantly to 29 Gt of CO_2 in 2007 since the industrial revolution which causes global surface temperature rising about 0.66 °C since 1900, according to the National Oceanic and Atmospheric Agency (NOAA) [4].

The use of renewable energy sources such as solar, wind, geothermal are very interesting alternatives to meet world energy needs. The possibility of cold production from solar energy has been initiated by technological developments in solar energy.

In recent years, these environmental consequences have forced many nations in the world to harvest the abundant solar energy for their domestic and industrial applications.

Solar thermal systems, in addition to the well-known advantages of renewable resources are very suitable for air-conditioning and refrigeration demands, because solar radiation availability and cooling requirements usually coincide [5,6]. Solar air-conditioning and refrigeration facilities can also be easily combined with space heating and hot water applications, increasing the yearly solar fraction of buildings.

Conventional cold producing machines that are based on vapor compression principle are primary electricity consumers and their working fluids are being banned by international legislation. Solar powered cooling systems as a green cold production technology are the best alternative. Absorption refrigeration is a mature technology that has proved its applicability with the possibility to be driven by low grade solar and waste heat [3].

In the Solar Heating and Cooling Technology Roadmap published by International Solar Energy in 2012 it's estimated to 750 the number of installed solar cooling systems worldwide in 2011, including small capacity (< 20 kW) plants. The IEA roadmap also mentions recent developments of big plants, as that of the United World College in Singapore, completed in 2011, with a cooling capacity of 1470 kW and a collector field of 3900 m^2 executed on an energy services company (ESCO) model [7] .

Parabolic trough collectors (PTC) are the most proven, widespread and commercially tested technology available for solar harnessing. The majority of the parabolic trough plants deployed operate at temperatures up to 400 °C using synthetic oil as heat transfer fluid (HTF) [8].

The PTC system uses mirrored surfaces of a linear parabolic concentrator/reflector to focus direct solar radiation to a tubular absorber tube located along the focal line of the parabola. Then, the concentrated solar radiation is absorbed and converted into thermal energy by the heat transfer fluid (HTF) flowing through the tube [8,9,10].

In this paper, a solar PTC mathematical model is developed. It constitutes the basis of a numerical code performed in Matlab. It is validated with the final test results for the Schott HCE on a LS-2 Collector (Sandia laboratories) [11], and then can be used in solar concentrating projects. An example of its integration to an absorption air conditioning project is presented under TRNSYS environment.

II. METHODOLOGY OF THE WORK

The paper presents a mathematical modeling and a numerical simulation of a solar parabolic trough collector. The model is validated according to the test results for the Schott HCE on a LS-2 Collector at Sandia laboratories. Results of the simulation are presented. After implementing the model as a new type (type240) in TRNSYS 17 library, it is integrated into an absorption air conditioning project. The project concerns the air conditioning of a block of offices for teachers in the region of Tlemcen, (Algeria). TRNbuild is used to create the description file of the block. This file will be used as an external file for the type 56. Results of the simulation are presented.

III. SOLAR PARABOLIC TROUGH COLLECTOR MODELLING

In solar refrigerators, concentration collectors act as the heat source of the machine. They comprise a focal absorbent tube, a reflecting surface and one or more glass protecting the collector from heat losses. Solar concentration can reach a temperature which can activate dissociation in the solution contained in the generator and hence releasing the refrigerant.

Many significant simulation studies on the complex process of the photo-thermal conversion have been carried out. They generally aimed to describe the mechanism working, to study the performance-reliability improvement and the cost reduction. Most models were assumed that the solar flux and the fluid flow were uniform or constant, or that the correlations in the models were based on an assumption of a uniform or constant temperature [12].

Kalogirou [13] presented a detailed thermal model of a parabolic trough collector written in the Engineering Equation Solver (EES) and validated with known performance of existing collectors to be at last used to perform analysis of a collector going to be installed at Archimedes Solar Energy Laboratory at the Cyprus University of Technology.

Xu et al [12] presented a study on the comparison of three outdoor test methods for determining the thermal performance of parabolic trough solar collectors. The methods are respectively the steady-state method in the ASHRAE 93 standard, the quasi-dynamic method in the EN 12975-2 standard and a new dynamic method developed by the authors. The comparison shows the advantages and disadvantages of these models according to both the practical operation and weather conditions.

A detailed numerical heat transfer model based on the finite volume method is presented by Hachicha et al [14] the different elements of the receiver are discretized into several segments in both axial and azimuthal directions. The set of algebraic equations are solved simultaneously using direct solvers. Results obtained shown a good agreement with experimental data.

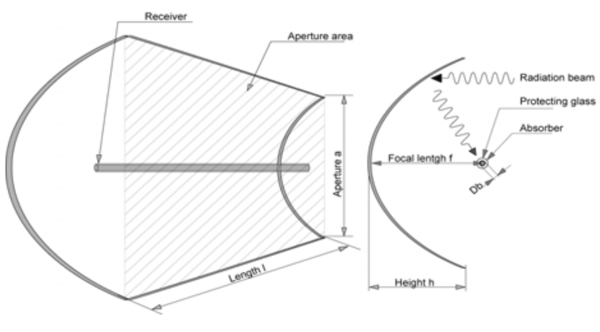

Fig.1. Geometry of the solar parabolic trough collector

As shown in Figure 1, the linear parabolic concentrator form is given by [15]:

$$y^2 = 4fx \tag{1}$$

where f is the focal length of the collector.

Thermal energy received at the focal line is absorbed by a metallic pipe inside an evacuated glass tube. The geometric concentration ratio is a measure of the average concentration. Its value depends directly on the collector geometry. It can be calculated by the ratio of the collector aperture area (A_a) to the receiver area (A_r) [15]:

$$C = \frac{A_a}{A_r} \tag{2}$$

The actual concentration coefficient is calculated using the following equation [6]:

$$A_a = l\left[\left(\frac{a}{2}\sqrt{\left(\frac{4h}{a}\right)^2+1}\right)+2f\ln\left(\frac{4h}{a}\sqrt{\left(\frac{4h}{a}\right)^2+1}\right)\right] \tag{3}$$

with

$$h = \frac{a^2}{16f} \tag{4}$$
$$A_r = \pi l D_b$$

a and *l* are respectively the collector aperture diameter and the length of the concentrator, D_b is the mean diameter of the absorber

In dynamic modeling, energy stored in certain levels is considered. These levels are considered as nodes at a uniform temperature. This method consists in subdividing a system into several subsystems. But before establishing the energy balance of the collector, modeling assumptions must be considered.

A. Assumptions

The following simplifying assumptions have been made:

- Due to vacuum, heat transfer between the glass and the absorber is only due to radiation and conduction and therefore convection is negligible [16].
- The working fluid is incompressible and without phase change.
- The absorbent tube coincides with the focal line.
- The temperature is uniform at each node.
- The glass is considered opaque to infrared radiation.
- Temporal variations along the absorber and the glass thickness are negligible.
- The solar flux at the absorber is uniformly distributed.

B. Mathematical modeling

Figure 2 shows the different types of heat transfer involved. According to this representation, a homogeneous temperature prevails at each node and an axisymmetric heat transfer occurs. An electrical analogy of the problem is shown in Figure 3.

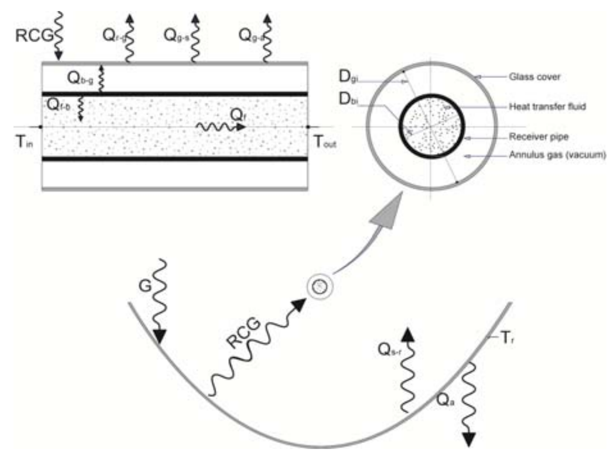

Fig. 2. Energy balance

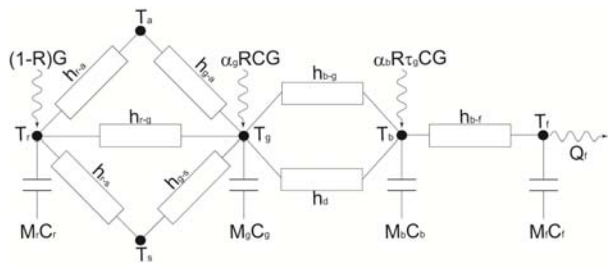

Fig.3. Electrical analogy of the problem

The amount of heat exchanged between the nodes can be written as:

$$Q_{ij} = A_{ij}h_{ij}\left(T_i - T_j\right) \tag{5}$$

i and *j* are the two neighboring nodes, A_{ij} and h_{ij} are respectively the surface and heat transfer coefficient between the two *i* and *j* nodes.

Each node stores a quantity of heat which depends on the capacity of the used material. Temperature change over time is given by:

$$Q_i = M_i C_i \frac{dT_i}{dt} \tag{6}$$

Heat transfers between reflector and air is given by:

$$M_r C_r \frac{dT_r}{dt} = \begin{pmatrix} (1-R)G - h_{r-a}(T_r - T_a) \\ -h_{s-r}(T_r - T_s) - h_{r-g}(T_r - T_g) \end{pmatrix} \quad (7)$$

Between glass and reflector, it's expressed by:

$$\begin{aligned} M_g C_g \frac{dT_g}{dt} &= \alpha_g A_g R G + A_g h_{r-g}(T_r - T_g) \\ &- A_{ge}\left(h_{g-a}(T_g - T_a) + h_{s-g}(T_g - T_s)\right) \\ &+ A_{gi}\left(h_{b-g} + h_d\right)(T_b - T_g) \end{aligned} \quad (8)$$

Similarly by absorber and glass:

$$\begin{aligned} M_b C_b \frac{dT_b}{dt} &= \tau_g \alpha_b A_r R G - A_{bi} h_{b-f}(T_b - T_f) \\ &- A_{gi}\left(h_{b-g} + h_d\right)(T_b - T_g) \end{aligned} \quad (9)$$

And finally between HTF and absorber:

$$M_f C_f \frac{dT_f}{dt} = A_{pi} h_{p-f}(T_p - T_f) - \dot{m} C_f \frac{dT_f}{dx} \quad (10)$$

With M_i and C_i is the mass and specific heat of the i node material, $\dot{m}_f$ is the working fluid mass flow, T_s the sky temperature calculated by [8]:

$$T_s = 0.0552 T_a^{1.5} \quad (11)$$

Many empirical correlations are used to determine directly the heat transfer coefficients between the different nodes.

<u>Reflector and sky</u> [16]

$$h_{s-r} = \sigma \varepsilon_r (T_s + T_r)(T_s^2 + T_r^2) \quad (12)$$

<u>Reflector and glass</u> [15,16]

$$h_{r-g} = \frac{\sigma (T_g^2 + T_r^2)(T_g + T_r)}{\frac{1-\varepsilon_g}{\varepsilon_g} + \frac{1}{F_{rg}} + \frac{(1+\varepsilon_r) A_r}{\varepsilon_r A_{ge}}} \quad (13)$$

with

$$F_{rg} = \frac{\left[(W_1 + W_2)^2 + 4\right]^{\frac{1}{2}} - \left[(W_2 - W_1)^2 + 4\right]^{\frac{1}{2}}}{2W_1}$$

$$W_1 = \frac{D_{ge}}{f}$$

$$W_2 = \frac{a}{f}$$

<u>Between reflector and air</u> [17]

$$h_{r-a} = \frac{8.6 V^{0.6}}{l^{0.4}} \quad (14)$$

<u>Between glass and sky</u> [16]

$$h_{s-g} = \sigma \varepsilon_g (T_s + T_g)(T_s^2 + T_g^2) \quad (15)$$

<u>Between glass and air</u> [12]

$$h_{g-a} = \frac{4V^{0.58}}{D^{0.42}} \quad (16)$$

<u>Between glass and absorber</u> [15, 16, 18, 19]

$$h_{b-g} = \frac{\sigma (T_g^2 + T_b^2)(T_g + T_b)}{\frac{1-\varepsilon_g}{\varepsilon_g} + \frac{1}{F_{gb}} + \frac{(1+\varepsilon_b) A_{gi}}{\varepsilon_b A_{be}}} \quad (17)$$

with

$$F_{gb} = \frac{1}{X}\left[\frac{1}{\pi}\left(\cos^{-1}\left(\frac{B}{A}\right)\frac{1}{2Y}\left(C\cos^{-1}\left(\frac{B}{XA}\right) + B\cos^{-1}\left(\frac{1}{X}\right) - \frac{\pi}{2}A\right)\right)\right]$$

$$X = \frac{D_{gi}}{D_{be}}, \quad Y = \frac{2l}{D_{be}},$$

$$A = X^2 + Y^2 - 1, \quad B = X^2 Y^2 - 1, \quad C = \sqrt{(A+2)^2 - (2X)^2}$$

<u>Between absorber and fluid</u> [20]

For turbulent flow, Re > 10^4:

$$Nu = 0.125\left(0.790\log(\mathrm{Re}) - 1.64\right)^{-2} \mathrm{Re}\,\mathrm{Pr}^{0.34} \quad (18)$$

For laminar flow, Re<10^4:

$$Nu = 3.66 + \frac{0.0668\left(\frac{D_{bi}}{l}\right)\mathrm{Re}\,\mathrm{Pr}}{1 + 0.04\left[\left(\frac{D_{bi}}{l}\right)\mathrm{Re}\,\mathrm{Pr}\right]^{0.67}} \quad (19)$$

IV. SOLAR PARABOLIC TROUGH COLLECTOR BEHAVIOR

Programming equations of the parabolic is performed in Matlab. Iterative Gauss method is used to solve these equations after

finite difference discretization. The Parabolic trough solar collector unsteady equations are discretized and solved by the finite volume method. A Crank-Nicolson scheme with Δt=1s was adopted for the time step.
Calculations are performed using a uniform mesh containing 100 nodes, iterative Gauss method was used and the program was written in Matlab. The collector components' geometrical and thermophysical properties and HTF properties are summarized in Table I and Table II.

Table I. and II illustrate respectively the collector components geometrical and thermal properties and HTF properties.

TABLE I. COLLECTOR PROPERTIES

	Symbol	Parameter	Value	Unit
Collector	l	Collector length	7.800	m
	a	Aperture	5.000	m
	f	Focal length	1.840	m
Reflector	Cp_r	Specific heat	0.581	$kJ/(kg.K)$
	ρ_r	Density	2,400	kg/m^3
	e_r	Thickness	0.005	m
	ε_r	Emittance	0.140	–
	R_r	Reflectance	0.930	–
Glass	Cp_g	Specific heat	1.090	$kJ/(kg.K)$
	ρ_g	Density	2,230	kg/m^3
	D_{gi}	Internal diameter	0.109	m
	D_{ge}	External diameter	0.115	m
	τ_g	Transmittance	0.935	–
	ε_g	Emittance	0.860	–
	α_g	Absorptance	0.020	–
Absorber	Cp_b	Specific heat	0.500	$kJ/(kg.K)$
	ρ_b	Density	8,020	kg/m^3
	D_{bi}	Internal diameter	0.066	m
	D_{be}	External diameter	0.070	m
	ε_b	Emissivity	0.140	–
	α_b	Absorptance	0.906	–
	p_v	Vacuum pressure	0.010	bar

TABLE II. FLUID PROPERTIES

Parameter	Value	Unit
Specific heat	1,617	$kJ/(kg.K)$
Density	936	kg/m^3
Conductivity	0.134	$W/(m.K)$
Viscosity	9.1	$Pa.s$
Critical temperature	367	$°C$
Critical pressure	10.9	bar
Critical density	3.22	l/kg

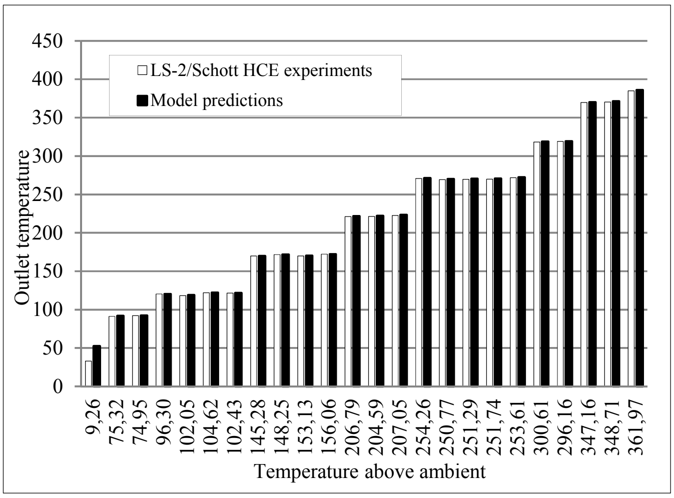

Fig. 4. Outlet HTF temperature vs. temperature above ambient

Figure 4 shows a comparison of the evolution the outlet HTF temperature with respect to the temperature above ambient between model predictions and test data of the LS-2/Schott HCE. Except the first value, a good agreement is found that the difference between predicted and experimental values of temperature does not exceed one degree (1K).

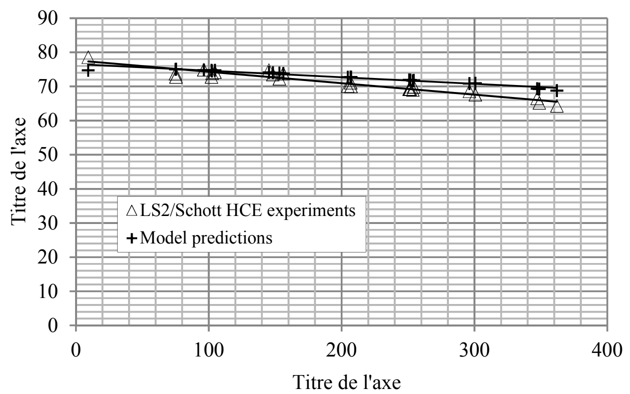

Fig.5. Peak efficiencies vs. average HTF temperature through the HCE above ambient

Similarly, figure 5 shows a comparison between predicted and measured peak efficiencies versus the HTF temperature above ambient. Here again, a good agreement is found. Nevertheless, a maximum prediction error of 6.56% is observed between predicted and measured values changes according to the temperature to obtain

These results allow for the use of the model for predicting the approximate temperature of the fluid outlet and the energy required to activate desorption of the binary solution in the absorption machine.

International Journal of Engineering Research & Technology (IJERT)
ISSN: 2278-0181
Vol. 4 Issue 02, February-2015

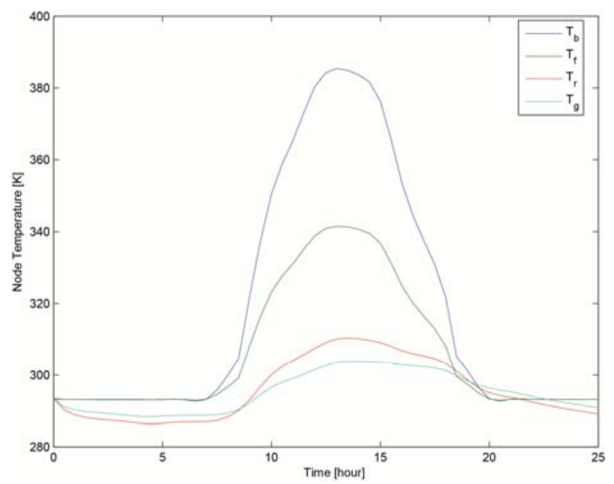

Fig. 6. Daily Temperature evolution at the different nodes on September 1st

Simulations allow studying the temperature evolution and hence the effectiveness of the collector at any time of the day throughout the year. Temperature evolution at each node is in a good agreement with the studied literature. Absorber temperature is greater compared to other temperatures due to its position in the focal line and its absorptivity, Figure 6.

Simulations allow also studying the efficiency of some collectors and to detect the most suitable energy solution. Figure 7 illustrates a comparison of collectors' solar fractions relative to the surface and allows choosing the most appropriate collector for the solar cooling installation, thus optimizing its surface.

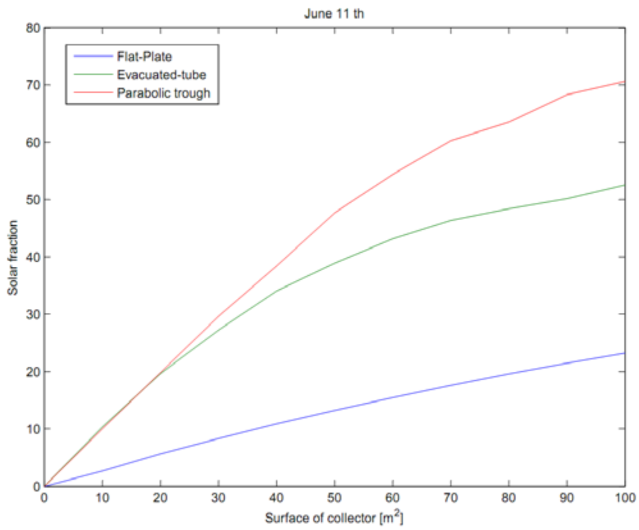

Fig. 7. Solar fraction for different collector types

It shows that the flat plate collector has the lowest solar fraction compared to the two other collectors. For the two others, the preference depends on climatic conditions and surface. For small areas, the evacuated tube collector is more convenient. But when using larger areas, the concentration collector becomes the most reliable.

V. Implementation in TRNSYS environment

TRNSYS is an open modular structure with open source code which simulates energy system sectors. To create a model, the user is able to create custom components or choose from the TRNSYS standard library of components. TRNSYS has been used extensively to simulate solar energy applications and conventional buildings.

A. Absorption machine

According to the IEA project, absorption technology is largely predominant with 72%, followed by adsorption technology and solid desiccant cooling technology with respectively 17% and 10% of installed power. Liquid desiccant cooling is almost nonexistent with less than 1% installed capacity [21].

Absorption chillers most commonly traded use water lithium bromide (H_2O LiBr) systems and are dedicated to air conditioning. Refrigeration industry uses ammonia water (NH_3 H_2O) systems. In this paper, TRNSYS Type107 is used to simulate the absorption machine.

B. Building

The building considered is a block of 5 offices for teachers and a hall, Figure 8. The hall has is 24 m^2, each office is 12 m^2 and the height is 3m. Figure V shows the layout and orientation of this structure. This building is located in Tlemcen in North West of Algeria (3° 38 west longitude and 34 ° 53 north latitude at 600 m altitude).

This block consists of 4 types of walls:

<u>Interior wall:</u> consisting of a layer of 15 cm thick clay bricks centered between two layers of 1 cm thick mortar.
<u>Exterior wall:</u> consisting of 5 layers, two 1.5 cm thick layers of mortar either side of the wall. Between these layers two 10 cm clay brick walls separated with a 7 cm thick air gap.
<u>Ceiling:</u> Composed of three layers, outside is 12 cm thick heavy concrete, a layer of 12 cm thick slab block and finally, the interior consists of a 1.5 cm thick mortar layer.
<u>Floor:</u> Composed of three layers: tile, mortar and heavy concrete respectively, 2 cm, 3 cm and 12 cm.
Each office is occupied by two people (light work) with a 230W personal computer each and a 5 W/m^2 lamp. Office windows have an area of one square meter each and all doors have two square meters each.
TRNSYS 17 is the final simulation environment. This allows incorporating components or software programmed in other languages. The simulation of the system is structured in three main parts: (1) heat production, (2) cold production loop and (3) coupling with the building. TRNbuild is used to create the description file of the block. This file will be used as an external file for the type 56.

C. Air conditioning system

The objective of the selected system, figure 9, is the study of heating as well as air conditioning. In heat production loop, the circulation of the coolant is ensured by a pump (P1) between the collector (CS) which acts as a converter of solar energy into thermal energy and the storage tank (R), which constitutes the primary circuit or heat production loop.

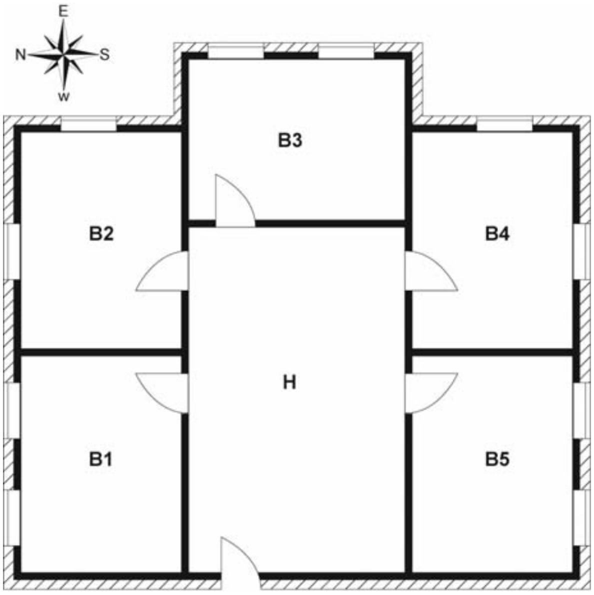

Fig. 8. Studied building layout

A second loop in which heat is transported from the reservoir (R) to the hot side of the absorption machine (TTM) in the case of cooling or to the heat exchanger (E1) in the case of heating. The change in fluid flow between the two circuits is provided respectively by an overflow valve (V1) and a mixer (V2). In this loop, there are two main components, the pump (P2), which provides circulation of the coolant (secondary circuit) and the auxiliary heater (H) when the collector cannot reach the desired temperature.

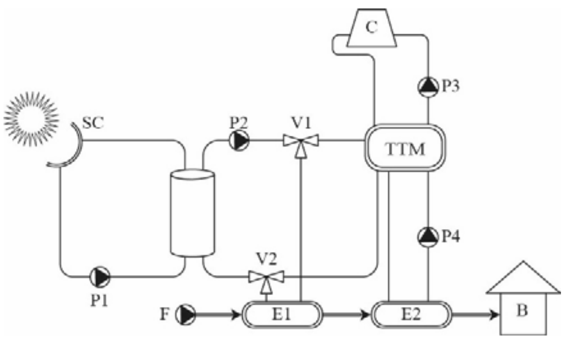

Fig. 9. Solar air conditioning system

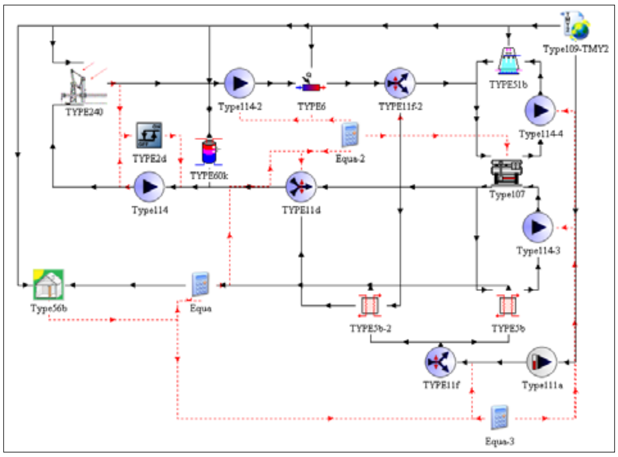

Fig. 10. The system under TRNSYS environment

The chosen system also includes a loop comprising a condensation pump (P3) and a cooling tower (CT). Another loop, which allows the coupling of the plant with the building, it consists of a heat exchanger (E2) combined to cold side of the absorption chiller to cool the ventilation air.

Before presenting simulation results, it is necessary to recall the main models and default values of the different components. The system under TRNSYS environment is illustrated in figure 10.

A controller actuates the pump (P1) when the solar energy is enough to maintain a temperature level higher than that of the hot water storage tank. In the following simulations, the fixed values of the temperatures of zones are 20 °C for air conditioning and 18 °C for heating.

One of the most interesting features of the solar cooling and one of its main advantages is the suitability of a temporal point of view between the need for cold and solar gain. This is true both annually and daily during summer. As can be seen in Figure 11, daily adequacy in Tlemcen is almost perfect during the five chosen days (from 08/15 to 08/19). The figure shows the evolution of the cooling demand for all the office parts. This variable demand during the day can be satisfied through a strategy of distributing cold depending on the time and place (location of the room). The phasing between the need for cold and solar gain is generally excellent, with sometimes a slight delay of a few hours which can be compensated by storage.

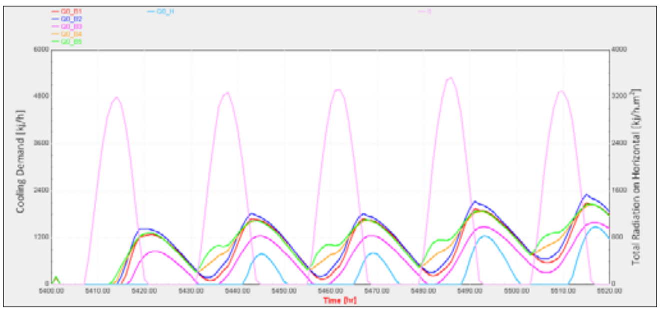

Fig. 11. Evolution of cooling demand

VI. CONCLUSION

In this paper, a thermal model of a solar parabolic trough collector is presented. Numerical simulations of the model are performed in Matlab. Temperature evolution at different nodes is presented and a comparison between of different solar collectors is made.

Then, the numerical model is implemented in TRNSYS 17 environment as a new type (type240). This type is integrated in a solar absorption air conditioning system. The project is dedicated to ensure a comfort temperature of 20 °C in teachers' offices under climatic conditions of the region of Tlemcen (Algeria).

Conclusions made are:

i. Temperature evolution at each node is in a good agreement with the studied literature. Absorber temperature is greater compared to other temperatures due to its position in the focal line and its absorptivity.
ii. In addition to the influence of solar radiation, wind velocity and angle of incidence also affect the temperature of the collector.
iii. The new type 240 can be implemented in solar energy projects studied under TRNSYS projects.
iv. Phasing between the need for cold and solar gain is generally excellent, with sometimes a slight delay which can be compensated by storage.
v. The use of solar energy for air conditioning is a promising alternative.

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