ELECTRICITY PRODUCTION SYSTEM FROM SOLAR-HEATED RANKINE CYCLE: MODELING AND SIMULATION

[1,2]Faouzi Nasri, [1,2*] Chaouki Ali & [3]Habib Ben Bacha
[1]LASEM (LAboratoire des Systèmes Electro-Mécaniques), National Engineering School of Sfax, B.P. W3038 Sfax, Sfax University – Tunisia
[2]Faculty of Sciences Gafsa, Campus Universitaire Sidi Ahmed Zarrouk - 2112 Gafsa – Tunisia
Gafsa University – Tunisia
[3]Al Kharj University , College of Engineering at Alkharj, BP 655 - 11942 – Kingdom of Saudi Arabia
E-mail: chaouki_ali@yahoo.fr, hbacha@Ksu.edu.sa, nasrifaouzi@yahoo.fr

ABSTRACT
This paper present a design of an electricity production system from a mechanical power generation based on solar heated Rankine cycle operating at low temperature range, developed at the Laboratory of Electromechanical Systems of the National Engineering School of Sfax – Tunisia. The proposed system permits to produce electricity from solar thermal energy and auxiliary heater upon a demand. The dynamic modelling of heat and mass transfer behaviours of the different sections (solar collector, storage tank, evaporator etc.) are presented. The mathematical balance equations are numerically resolved using the finite volume method. The numerical simulation permit to study the behaviour of the system and it would enhance the performance of such installation.

Keywords: *Electricity production, solar thermal energy, Organic Rankine Cycle, modeling, numerical simulation.*

1. INTRODUCTION

The Rankine cycle converts heat into work frequently found in electricity generation plants. Organic Rankine Cycle commonly uses industrial waste heat renewable energy for electricity production such as solar energy, geothermal energy.

Previous works were interested in producing of electricity by solar thermal energy. Lourdes et al [1], present the project entitled POWERSOL (Mechanical Power Generation Based on Solar Heat Engines) partially supported by the European Commission under the Specific program for research, The main POWERSOL objective is the development shaft power generation technology, based on solar thermal energy, The project focuses on the technological development of a solar thermal-driven mechanical power generation based on a solar-heated thermodynamic cycle. Zouhair et al [2], propose a hybrid thermal energy storage system for managing simultaneously the storage of heat from solar and electric energy. A system heat transfer model is developed and validated. Saad et al [3], present a unified model of a solar electric generation system in order to study different collector-field power-house arrangements, the model evaluates thermal properties, steam flow rate and pressure drop in a direct steam generation or an oil based collector field. D. Manolakos et al [4], present a paper concerning the design outline, of a low temperature solar organic Rankine cycle system for reverse osmosis desalination. E.J. Adam et al [5], present a Dynamic simulation of large boilers with natural recirculation nonlinear dynamic model for natural circulation drum-boilers that describes the complicated dynamics of water-in-tube boilers with natural recirculation the drum, downcomer, and riser components. Sylvain Quoilin et al [6] conducted an Experimental study and modeling of an Organic Rankine Cycle working with refrigerant HCFC-123 using scroll expander. Xudong Ding et al [7] proposed a hybrid modeling approach to model two-phase flow evaporators. Donghong Wei et al [8] analyzed thermodynamic performances and optimization of an ORC system under disturbances.

Theoretical performances as well as thermodynamic and environmental properties of few fluids have been comparatively assessed for use in low-temperature solar organic Rankine cycle systems by G. Kosmadakis al [9]. In this paper we present essentially the plate collector and the Evaporator/condenser model, the storage tank, Auxiliary tank, expander and the pump model are presented in previous work [10].

The work objective is the dynamic modeling and simulation of the unit and its implementation in the region of Gafsa- Tunisia. System description and modeling based on thermodynamic laws was made in a previous work [10].

NOMENCLATURE

A	Area,	m^2
C	Specific heat	$J/(kgK)$
$\dot{m}$	Mass flow rate,	kg/s
m	Mass,	kg
$\dot{Q}$	Energy,	W
I	Solar flux,	W/m^2
t	Time,	Second
T	Ambient temperature,	K
U	Overall heat loss coefficient of the CPC,	$W/(m^2K)$
δ_c	Control function,	
ρ	Mass density,	kg/m^3

2. TECHNICAL DESCRIPTION OF THE SYSTEM

The operation of the system of electricity production utilizing solar-heated Rankine Cycles is illustrated in the figure 1. The proposed system is consisted of several interconnected components with a three loops.

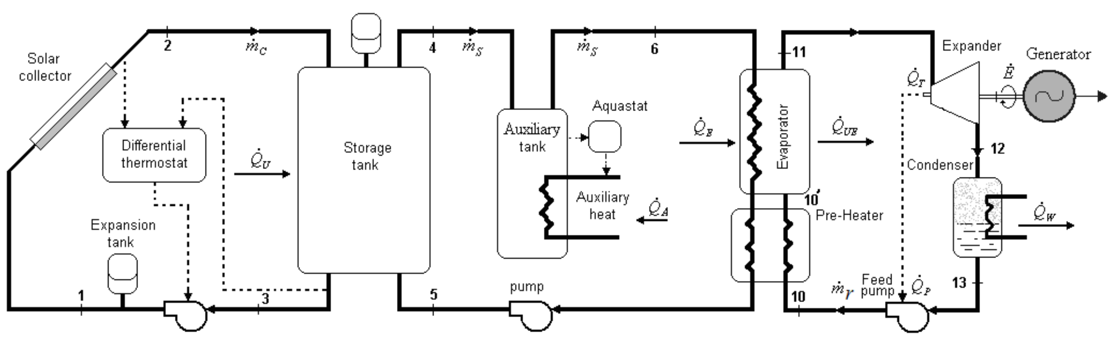

Figure 1: *Design of the system of electricity production from a mechanical power generation based on solar heated Rankine Cycles*

The first flow loop, whose primary function is to convert the incident solar radiation into thermal energy, this flow loop containing a flat plate solar collector, the storage tank, the collector pump, and The differential thermostat controller. Protection of the collector loop from overpressure by thermal expansion is provided by an expansion tank.

At the second flow loop, the working fluid from the storage tank is delivered to the auxiliary tank; the working fluid is heated by an auxiliary heater upon demand. The energy delivered from the auxiliary tank is used to drive the third flow loop. the second flow loop pump is controlled by a differential thermostat- potentiometers none presented in the figure, in order to maintain a constant temperature 90 °C at state 7.

The third flow loop is constituted of an evaporator, an expander, a condenser and a feed pump. The working fluid, HFC-134a, leaving the condenser at state 13, and pumped to the evaporator at state 10, at this point it enters the evaporator after to be pre-heated, where it is converted into a superheated vapour at state 11. The super-heated vapour is then driven to the expander where mechanical work is then generated. The saturated vapour at the expander outlet, at state 12, is directed to the condenser and condensed. The expander shaft is coupled to a generator which converts the rotary mechanical motion into the electrical energy.

Figure 2 illustrates the thermodynamic cycle (pressure – enthalpy) of the HFC 134a working fluid, the net work output. Temperature, pressure, enthalpy and entropy at different states are listed in table 1.

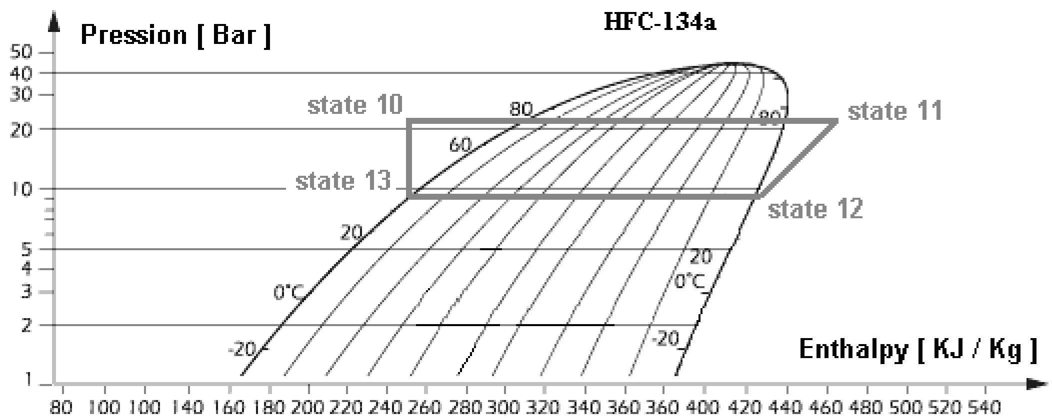

Figure 2: *Pressure-enthalpy Rankine cycle working fluid states (HFC-134a).*

State	T (°C)	P (Bar)	H (kJ / kg)	S (kJ / kg. K)
10- Pumped liquid,	35	21.16	248.7	1.166
10’- Saturated liquid,	70	21.16	303.9	1.332
11- Super-heated vapor	75	21.16	437.8	1.718
12- Saturated vapor	35	8.67	415.9	1.709
13- Saturated liquid	35	8.67	248.7	1.166

Table 1: State of Rankine Cycle

3. MATHEMATICAL MODEL

- **Flat plate solar collector modeling**

The model of the flat plate solar collector is given by

$$\frac{\partial T_f}{\partial t} = \frac{\left[-\left(\dot{m}_f . \xi\right) . \frac{\partial T_f}{\partial x} - T_f + f(t) \right]}{b} \qquad (1)$$

With:

$$\xi = \frac{C_f}{U_g . l} \qquad (2)$$

$$b = \frac{m_f C_f + m_{ab} . C_{ab}}{U_g . l . L} \qquad (3)$$

$$f(t) = \frac{B . I(t)}{U_g} + T_a(t) \qquad (4)$$

The rate of the total useful energy, $\dot{Q}_U$, delivered by the flat plate solar collector is given by:

$$\dot{Q}_U = A_c F_R \left[S - U_L \left(T_1 - T_a \right) \right] \delta_c = \dot{m}_f C_f (T_2 - T_1) \delta_c \qquad (5)$$

Where δ_c is the control function.

$\delta_c = 1$	when	$\left[S - U_L \left(T_1 - T_a \right) \right]$	is positive
$\delta_c = 0$	when	$\left[S - U_L \left(T_1 - T_a \right) \right]$	is negative

The control function δ_c implies that $\dot{Q}_U$ is positive only when the incident energy is greater than the losses from the collector.

- **Storage tank modeling**

At any instant, heat may be added to the storage tank by the collector ($\dot{Q}_U$), removed by the second loop ($\dot{Q}_D$), and lost to the storage surrounded by heat transfer through the storage tank walls ($\dot{Q}_{Lst}$). The instantaneous energy balance on the storage unit is then:

$$\left(MC_p\right)_{st} \frac{dT_S}{dt} = \dot{Q}_U - \dot{Q}_D - \dot{Q}_{Lst} \quad (6)$$

- **Auxiliary tank modeling**

Same thing as the storage tank, heat may be added/ removed to the auxiliary storage tank by the storage tank/heat exchanger, ($\dot{Q}_{SH}$), and added by the auxiliary heater ($\dot{Q}_{AH}$) and lost to the auxiliary tank surrounded by heat transfer through the auxiliary tank walls ($\dot{Q}_{Las}$).The instantaneous energy balance on the storage unit is then:

$$\left(MC_p\right)_{as} \frac{dT_{as}}{dt} = \dot{Q}_{AH} - Q_{SH} - \dot{Q}_{Las} \quad (7)$$

- **Evaporator / condenser modeling**

The figure 3 shows the evaporator diagram and the differential control volume used to develop the model.

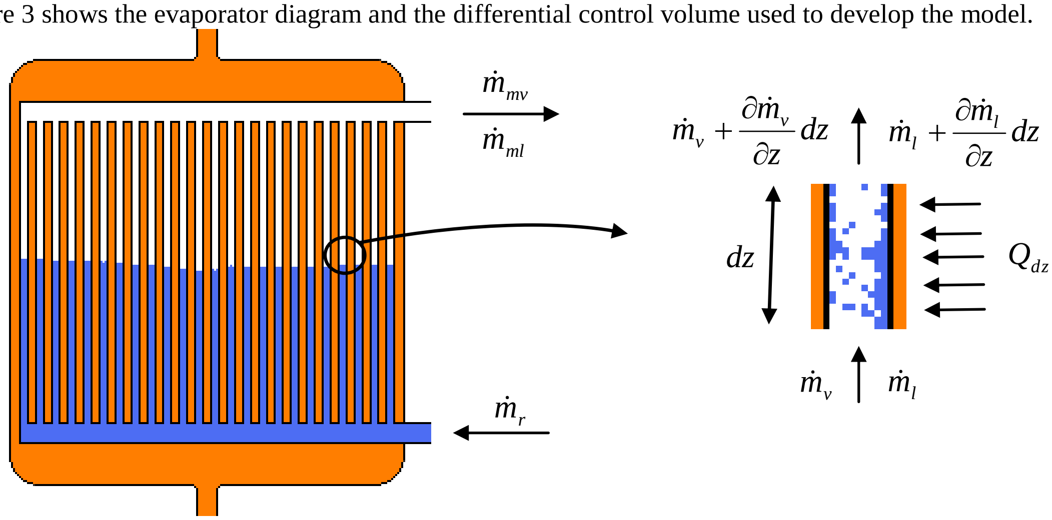

Figure 3: *Evaporator diagram.*

Mass conservation equation

The overall mass conservation can bi delivered by:

$$\frac{d}{dt}(V_r \rho_l + \bar{\alpha}_e \rho_v V_m + \frac{\partial}{\partial z}((1-\bar{\alpha}_e)\rho_v V_m) = \dot{m}_r - \dot{m}_{mv} - \dot{m}_{ml} \quad (8)$$

Vapor mass conservation of the control volume

$$\frac{\partial}{\partial t}(\alpha_e \rho_v) + \frac{\partial}{\partial z}(\alpha_e \rho_v w_v) = \Gamma_e \quad (9)$$

Liquid mass conservation of the control volume

$$\frac{\partial}{\partial t}((1-\alpha_e)\rho_l) + \frac{\partial}{\partial z}((1-\alpha_e)\rho_l w_l) = -\Gamma_e \quad (10)$$

Where w is the velocity (ms^{-1}), α is the volume vapor fraction

Momentum balance equation

$$\frac{\partial}{\partial t}((1-\alpha_e)\rho_l w_l + \alpha_e \rho_v w_v) + \frac{\partial}{\partial z}((1-\alpha_e)\rho_l w_l^2 + \alpha_e \rho_v w_v^2) = -((1-\alpha_e)\rho_l + \alpha_e \rho_v) g - \frac{\partial}{\partial z}(P) - \frac{4\tau_w}{D_i} \quad (11)$$

Energy conservation equations

$$\frac{\partial}{\partial t}[(1-\alpha_e)\rho_l (h_l + \frac{1}{2} w_l^2 - \frac{P}{\rho_l}) + \alpha_e \rho_v (h_v + \frac{1}{2} w_v^2 - \frac{P}{\rho_v})] + \frac{\partial}{\partial z}[(1-\alpha_e)\rho_l (h_l + \frac{1}{2} w_l^2) w_l + \alpha_e \rho_v (h_v + \frac{1}{2} w_v^2) w_v] = -[(1-\alpha_e)\rho_l w_l + \alpha_e \rho_v w_v] g + \frac{4Q_w}{D} \quad (12)$$

The shear stress , τ_w, is given by:

$$\tau_w = -\frac{D_i}{4}\left(\frac{dP}{dz}\right)_F \quad (13)$$

The total heat flux from the tube wall, Q_w, is estimated by:

$$Q_w = h_l.(T_w - T_l) \quad (14)$$

The heat transfer is given by:

$$h_l = 0.023 \frac{k_l}{D} R_{em}^{0.8} P_{rl}^{o.4} \quad (15)$$

The evaporation rate, Γ, is calculated from

$$\Gamma_e = \frac{4}{D} \frac{h_l.(T_w - T_l) - \sqrt{\alpha}.h_v.(T_l - T_v)}{h_{fv}} \quad (16)$$

The void fraction α can be determined by [11, 12]:

$$\alpha = \left(1 + \left(\frac{1-x_f}{x_f}\right)\left(\frac{\rho_v}{\rho_l}\right)^{2/3}\right)^{-1} \quad (17)$$

4. SIMULATION AND RESULTS

The developed equations in section 3 are used to simulate the different component of the unit. The electricity production system consists of 6 solar collector subfield located at Gafsa, each subfield connected to 1.1 m^3 cylindrical storage tanks with $U_{Lst} = 0.3\ Wm^{-2}K^{-1}$. The subfield solar collector has total area of 15 m^2 and uses water for the working fluid. Each solar collector has one cover glasses faces south at 45 degrees with collector heat removal factor $F_r = 0.8$. The energy delivered to the ORC from the third loop is 78,3 KW. Other system specifications and input data were presented in previous work [10]. The used condenser consists of 100 tubes of identical shape of 1m length and 10mm diameter.

Figure 4-7 present respectively the Subfield storage tank temperature and heat loss variation, collector heat loss and the useful energy in function of the absorbed solar radiation S with $\dot{m}_4 = \dot{m}_5 = 0$.

Figures 8-10 present respectively the pressure void fraction and vapor Relative velocity in the evaporator.

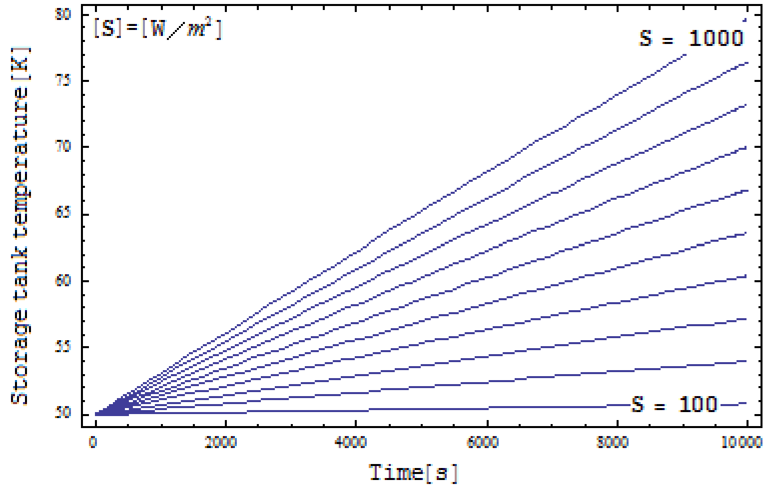

Figure 4: *Subfield storage tank temperature variation,* T_s *, with the absorbed solar radiation* S

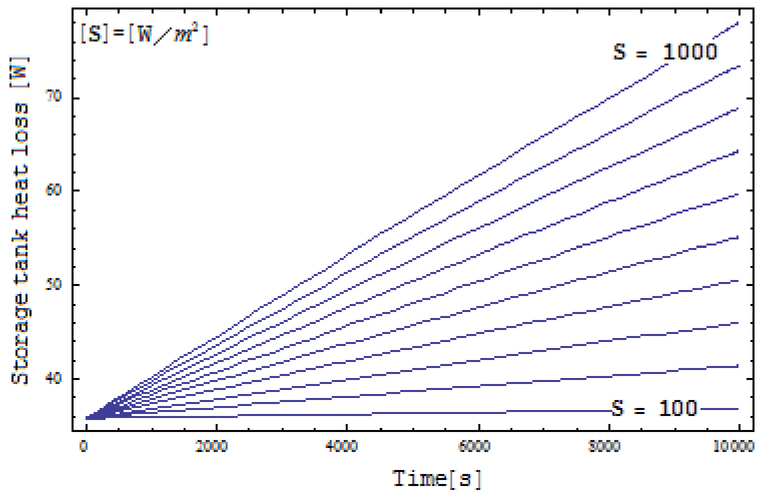

Figure 5: *Subfield storage tank heat loss,* $\dot{Q}_{Lst}$ *, with the absorbed solar radiation* S

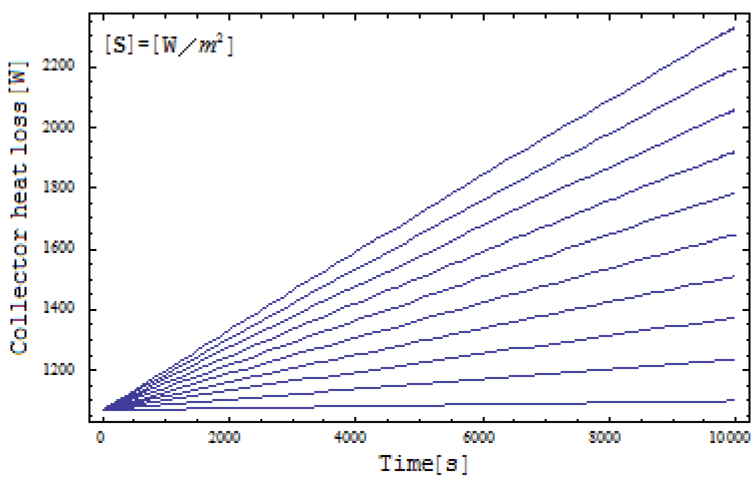

Figure 6: *Subfield collector heat loss with the absorbed solar radiation* S

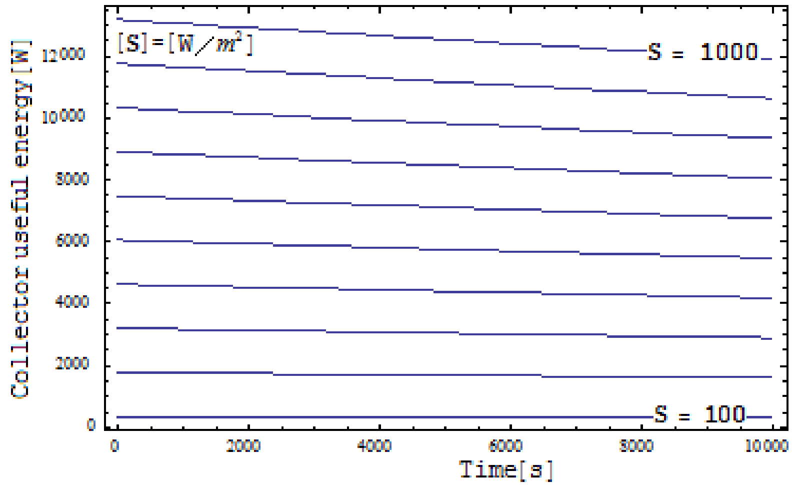

Figure 7: *Subfield collector useful energy,* $\dot{Q}_U$ *, with the absorbed solar radiation* S

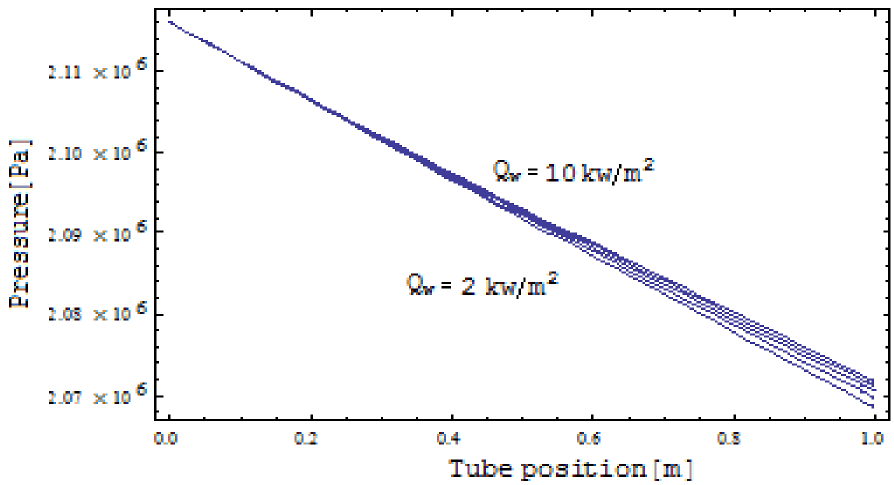

Figure 8: *pressure variation with tube position in the evaporator*.

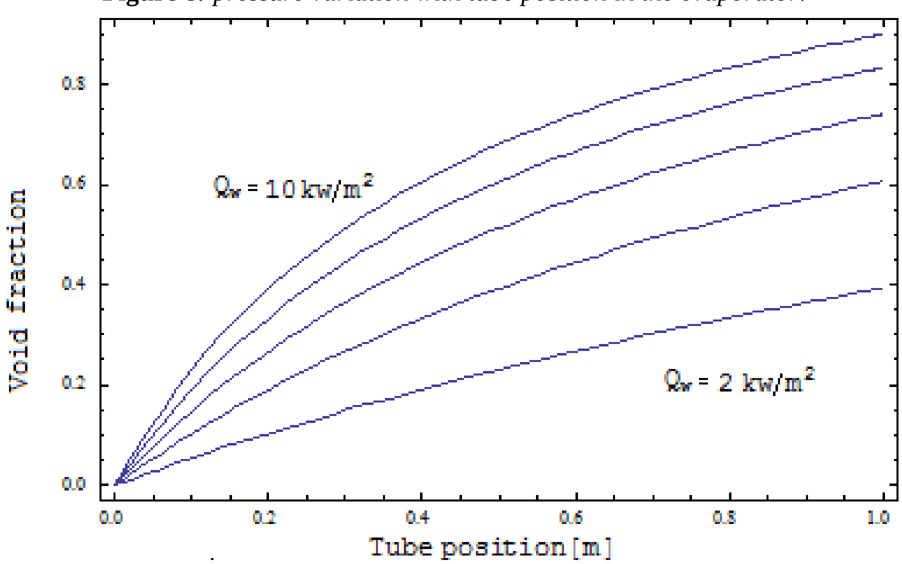

Figure 9: *void fraction variation with tube position in the evaporator*

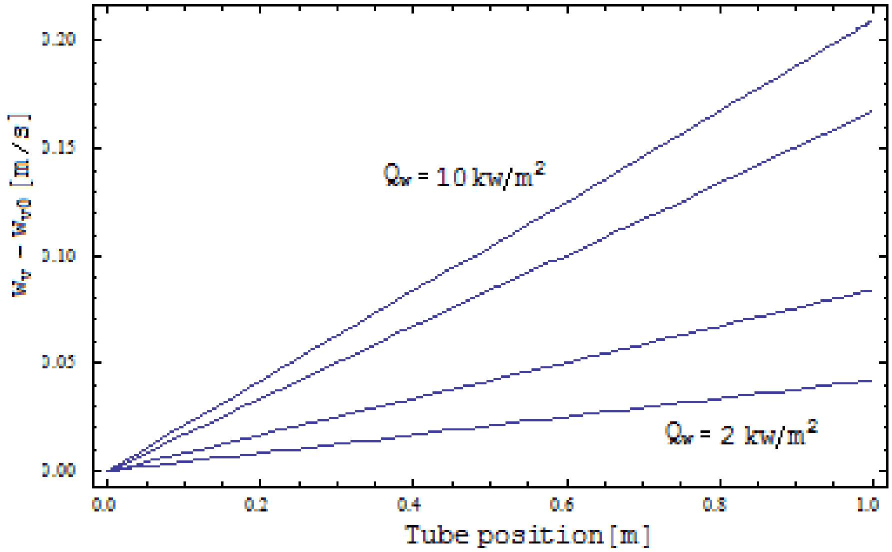

Figure 10: *vapor Relative velocity in tube position*

5. CONCLUSION

The mathematical model of the different sections of the electricity production system from a mechanical power generation based on solar heated Rankine cycle operating at low temperature range, permit to predict the system thermal behavior. The developed system, permit to supply the basic needs of electricity to small communities, rural locations, Saharan and isolated areas. Actually research and development of these types of installation are encouraged due to intensive use of energy and the historical records of petrol prices.

6. REFRENCES:

[1]. Lourdes García-Rodríguez, Julián Blanco-Gálvez, Solar-heated Rankine cycles for water and electricity production: POWERSOL project, *Desalination* 212 (2007) 311–318.

[2]. Zouhair Ait Hammou, Marcel Lacroix ,A hybrid thermal energy storage system for managing simultaneously solar and electric energy, *Energy Conversion and Management*, 47 (2006) 273-288.

[3]. Saad D. Odeh, Unified model of solar thermal electric generation systems, Renewable Energy 28 (2003) 755–767.

[4]. D. Manolakos, G. Papadakis, Essam Sh. Mohamed, S. Kyritsis, K. Bouzianas, Design of an autonomous low-temperature solar Rankine cycle system for reverse osmosis desalination, *Desalination* 183 (2005) 73–80.

[5]. E.J. Adam, J.L. Marchetti, Dynamic simulation of large boilers with natural recirculation, *Computers and Chemical Engineering 23 (1999) 1031–1040.*

[6]. Sylvain Quoilin, Vincent Lemort, Jean Lebrun, Experimental study and modeling of an Organic Rankine Cycle using scroll expander, *Applied Energy 87 (2010) 1260–1268* .

[7]. Xudong Ding, Wenjian Cai, Lei Jia, Changyun Wen, Evaporator modeling – A hybrid approach, *Applied Energy 86 (2009) 81–88.*

[8]. Donghong Wei, Xuesheng Lu, Zhen Lu, Jianming Gu, *Performance analysis and optimization of organic Rankine cycle (ORC) for waste heat recovery*, Energy Conversion and Management 48 *(2007) 1113–1119*

[9]. G. Kosmadakis, D. Manolakos, S. Kyritsis, G. Papadakis, Comparativethermodynamic study of refrigerants to select the best for use in the high-temperature stage of a two-stage organic Rankine cycle for RO desalination, *Desalination 243 (2009) 74–94* .

[10]. Nasri Faouzi, ChaoukiAli, Habib Ben Bacha, *Autonomous system of electricity production from a mechanical power generation based on solar-heated HFC-134a Rankine cycle*, international Journal of Research and Reviews in Applied Sciences (April 2010) 18-29.

[11]. Wisis Sripattrapan, Somchai Wongwises, *Two-phase flow of refrigerants during evaporation under constant heat flux in a horizontal tube*, International Communications in Heat and Mass Transfer, Volume 32, Issues 3-4, February 2005, Pages 386-402 .

[12]. John R. Thome, *engineering data book III, Wolverine Tube*, engineering innovation.