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Theoritical Analysis on Thermal Energy Storage using Phase Change Materials Capsules for Solar Organic Rankine Cycle Power Generation System

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Abstract:

This paper presents a theoretical analysis on packed bed thermal energy storage using phase change material (PCM) to improve the performance of solar organic Rankine cycle power generation system by extending its operational time demand to the night. The transient behavior of packed bed thermal energy storage using spherical capsules filled with paraffin wax as a storage unit has been studied numerically by finite difference technique for charging and discharging process .The effects of different parameters such as inlet temperature, mass flow rate, bed porosity , diameter of capsule and turbine inlet temperature have shown to be dominant parameters in determining the effective thermal efficiency of the solar ORC. It is seen that, turbine inlet temperature is plays an important role in the geometrical parameters of the packed bed .Comparison between theoretical results and experimental results of previous study for solar water heating system has been carried out where good agreement is noticed.

Keywords: Solar Energy Storage, Organic Rankine Cycle (ORC), Phase Change Material (PCM).

Introduction

An alternative solution for the luck of electricity in Iraq is being highly needed to solve this situation. One of these ways is to use the solar systems, as Iraq has a very good geographical location concerning the utilization of the solar energy. This kind of energy is used these days to shortfall in electricity energy, especially in remote areas where there is no source for electric power. One of these systems is the solar organic Rankin cycle thermal. This system works with variety of temperature ranges, especially the low ranges. Although it is limited to solar energy, this style of system is very active with the storage of that wasted energy and it is fully compatible with periods of use. Organic Rankine Cycles (ORC) have received increasing attention for power generation purposes due to their potential for utilizing heat from low temperature sources and their favorable characteristics for integration into future distributed energy systems, Heat recovery using organic Rankine cycles (ORCs) is considered to be important method among renewable energy processes because of its capability to generate power from the available waste heat and natural heat sources such as solar radiation ,ocean thermal sources ,geothermal sources ,biomass and waste heat from fuel combustion or industrial processes [1-5]

One of the options to improve energy efficiency is to develop energy storage devices and systems in order to reduce the mismatch between supply and demand. Such devices and systems also improve the performance and reliability by reducing peak loads and allowing systems to work within an optimal range. Thus, they play a preponderant role in conserving energy [6].

Solar energy is a promising renewable energy resource due to its characteristics of abundance, cost-free and cleanness to the environment, but the extensive utilization is impeded by its intermittent characteristic. An efficient and reliable thermal energy storage system is one of the promising solutions for this time dependent limitation. Thermal energy can be stored as a change in internal energy of material as sensible heat, latent heat and thermochemical or combination of these, compared with other methods of heat storage, latent thermal energy storage (LTES) has shown some outstanding advantages of large heat storage capacity in the form of latent heat and small temperature variation during storage/retrieval which made its hot research topic in recent years [7]. Latent heat storage in a Phase Change Material (PCM) is very attractive because of its high storage density with small temperature swing. It has been demonstrated that, for the development of a latent heat storage system, the choice of the PCM plays an important role in addition to heat transfer mechanism in the PCM [8].

Phase Change Materials have been used in different applications. Some examples of the use of PCM are: conditioning of building, ice storage, cooling food and milk products and green houses, medical applications, such as operating tables, waste heat recovery, heating water and cooling, solar power plant, heat pump system and computer cooling [9-13].

The finite difference technique in Matlab is used to study the effects of various geometrical and operational parameters on packed bed thermal storage system in ORC. The spherical PCM capsules in two cases of charge and discharge process for numerical modeling conducted in one dimension transient heat transfer model are being solved for the mathematical model and the simulation.

Solar Organic Rankine Cycle System:

Solar thermal power cycles can be classified as low, medium and high temperature cycles. Low-temperature cycles work at maximum temperatures of about 100ºC [14]. A Schematic of the low-temperature system using flat-plate collectors works on the Rankine cycle as shown in Fig.(1).

An organic Rankine cycle (ORC) engine is a standard steam engine that utilizes heated vapour to drive turbine. However, the vapour is heated organic chemical instead of superheated steam [15].There are four processes in the organic Rankine cycle similar to the Rankine cycle working for steam and liquid, each changing the state of the working fluid . These states are defined by a number in the diagram, as illustrated in Fig .(2).Firstly the working fluid is pumped from low to high pressure by a pump. Pumping requires a power input, (for example mechanical or electrical) in this stage the pump requires little input energy. Secondly, the high pressure enters storage tank where it is heated at constant pressure and becomes a dry saturated vapour or superheated vapour,as in Fig.(2) .Thirdly, the dry saturated vapour expands through a turbine to produce work transform to electrical generator to produce power .Ideally, this expansion is isentropic .This decreases the temperature and pressure of the vapour, and some condensation may occur. Fourthly, the wet vapour then enters a condenser where it is cooled to become saturated liquid at constant pressure and this liquid then re-enters the pump and the cycle repeats. The pressure and temperature of the condenser are fixed by the temperature of the cooling coils as the fluid is undergoing a phase –change.

The selection of the working fluid is critical to achieve high-thermal efficiencies as well as optimum utilization of the available heat source. Also, the organic working fluid must be carefully selected based on safety and technical feasibility. The first step when designing an ORC cycle application is the choice of the appropriate working fluid. The choice of working fluid is important since the fluid must not only possess thermo physical properties that match the application, but has chemical stability at the desired working temperature [16]. The working fluid used in this study is an organic substance which has low boiling point and low latent heat for using solar heat source. Here, R152a, R123 and R227 are chosen as the components of the mixture because they have zero ODP (Ozone Depression Potential) and lower GWP (Global Warming Potential), which has less environmental impact.

Thermal Energy Storage:

The discrepancy between energy supply and demand can be overcame by the implementation of a proper energy storage system. The latent thermal energy storage employing a PCM is the most effective way of the thermal energy storage due to its advantages of high energy storage density and its isothermal operating characteristics during solidification and melting processes. In a latent heat storage system, energy is stored during melting while it is recovered during solidification of a PCM [17]. The thermal energy storage system under investigation is latent heat storage system using Phase Change Material (PCM) spherical capsules assembled as packed bed.

A latent heat thermal storage system has the following three main components [18]:

1- PCM suitable for the desired temperature range.

2- Container for the PCM (encapsulation of PCMs).

3- Heat exchange surface area required for transferring the heat from heat source to PCM.

The major factors to be considered in thedesign of a storage unit containing a PCM include: (a) temperature limits within the unit is to operate; (b) the melting-freezing temperature of the PCM; (c) the latent heat of the PCM; (d) the thermal load; and (e) configuration of the storage bed. Other design factors which are important, but relatively less critical, are storage unit pressure drop and pumping power. Heat is transferred to or from a heat transfer fluid (HTF) as the heat transfer fluid flows through the voids in the bed. During charging mode when the solar radiation is high, the hot HTF carrying energy from the solar collector and the pump is turned on and the system continuously circulates through the packed bed in the axial direction from the top of the bed to the bottom then it is used to meet the load through a heat exchanger (to evaporate organic working fluid in ORC) and the PCM inside the capsules absorbs latent heat and melts. During discharging mode after sunset time, the pump is set off, while HTF is heated from the PCM by freezing the encapsulated PCM. The HTF is then used to meet the load through a heat exchanger. In both operating modes (charging and discharging), the difference between the mean temperature of HTF and the phase change temperature must be sufficient to obtain a satisfactory rate of heat transfer to evaporate the organic working fluid in ORC used for power generation by expand through the turbine, and from PCM to the heat sink.

Mathematical Models

A mathematical model of packed bed thermal energy storage , concerned with the low –temperature solar organic Rankine cycle is developed to determine the Operational characteristics of the ORC engine for a given set of operational inputs and equipment characteristics. The following assumptions are used in the mathematical model are developed to simulate the ORC:

• Each component is considered steady-state steady-flow.

• Kinetic and potential energy are negligible.

• Evaporating and condensing processes are isobaric.

• Heat losses (other than in condenser) are negligible.

And the following assumptions are used in the mathematical model are developed to simulate the HTF and PCM:

•One-dimension heat transfer (temperature distribution only in flow direction).

•Unsteady state.

•The heat exchange between PCM and HTF is by convection.

•Constant thermophysical properties for PCM and HTF with respect to temperature.

•Conduction heat transfer in flow direction is not considered.

•Melting and solidification processes occur at the same and constant temperature.

Based on the above assumptions, the thermal energy conservation equations and boundary conditions for the ORC-PCM are given below:

Solar collector

Solar collectors are the key component of active solar-heating systems. They gather the sun's energy, convert its radiation into heat, and then transfer that heat to a fluid. In a solar organic Rankine cycle system, the flat plate solar collector is the main part of the system for obtaining fluid temperature less than 100 ºC. Hence, the optimal performance of the solar collector is highly important. The important parts of a typical flat plate solar collector, as shown in Fig.(3),are the black solar energy-absorbing surfaces, with means for transferring the absorbed energy to a fluid; a glass cover which reduces convection and radiation losses to the atmosphere; and back and lateral insulation to reduce the conduction losses as the geometry of system permits. In the present study, the back and lateral conduction losses are assumed to be negligible.

The solar thermal flat-plate collector model uses data from an environment Baghdad solar radiation data analysis and based on internal and external energy balance of the absorber solved in iteration loops to determine the temperature distribution in main parts of solar collector. In the steady state solar ORC thermal energy, input process is carried out with collectors. The useful energy delivered by the flat-plate collector (FPC) depends on the absorbed solar heat flux (H), the ambient temp T_a and the collector input/output water temperatures. The useful heat gain (Q_u) by the water is given, as [19]:

$$Qu = \overset{\circ}{m}_w C_{Pw}\left(T_{wout} - T_{win}\right) \quad \ldots\ldots\ldots\ldots (1)$$

where T_{win} , T_{wout} , C_{pw} and $\overset{\circ}{m}_w$ are the water inlet, outlet temperatures in collector, heat capacity and mass flow rate of the water, respectively. Under steady state conditions, the useful energy output is defined as the difference between the solar radiation reaching the absorber plate and the thermal loss (Q_L) [20].

$$Qu = A_a\left[(\tau\alpha)H - U_L\left(T_{pm} - T_a\right)\right] \quad \ldots (2)$$

The mean absorber plate temperature T_{pm} is difficult to calculate or measure, since it is a function of the collector design, the incident solar radiation and the entering fluid conditions [21]. So, Equation (2) is reformulated in terms of the inlet fluid temperature and the collector heat removal factor under steady state conditions according to principle Hottel-Whillier equation for useful heat gain, as shown below[22,23]:

$$Qu = F_R A_a \left[(\tau\alpha) H - U_L (T_{wi} - T_a) \right] \quad \ldots (3)$$

where A_a, F_R , τ,α and U_L are the aperture area, collector heat removal factor which is the quantity that relates the actual useful energy gained of collector to the useful gained by the air , transmission coefficient of glazing, absorption coefficient of plate and overall heat transfer coefficient of the absorber respectively.

The temperature of the circulated water at the solar collector exit is calculated from the equation:

$$T_{wo} = T_{wi} + \frac{F_R A_a \left[(\tau\alpha) H - U_L (T_{wi} - T_a) \right]}{m_w^\circ C_{Pw}} \quad \ldots (4)$$

Thermal Storage Tank

The schematic diagram of the one-dimensional physical model in the thermal storage of ORC system using spherical capsules phase change materials surrounded by heat transfer fluid HTF is shown in Fig.(4). The dimensions of simulation case are 0.8 m in length (L), 0.45 m in outer diameter of tank (D2),0.40 m in inner diameter (D1). The PCM (Phase change material) is based on the properties of paraffin wax, as listed in Table (1). The dimension of the PCM capsules is 80mm in diameter (d) with wall thickness of 0.5mm which is packed inside the tank above a grid plate placed at the bottom of tank .This grid plate acts as a support to the packed sphere, and is also useful in the distribution of HTF. The tank is divided into N layers each of height equal to the spherical capsules diameter. Thermal performance of a packed bed is concerned with heat transfer from flowing water to PCM packed in a container and vice versa. Hot water flows from top to bottom of the bed and heat transfer takes place from water to storage material during charging phase and evaporator of organic fluid in ORC.

The rate of energy transfer to or from the storage material elements in a packed bed is:

a-for charging process (temperature of HTF higher than the PCM melting temperature):

$$\dot{Q}_{Storage} = \dot{Q}_{Collectot} - \dot{Q}_{Evaporater} - \dot{Q}_{Losses} - \dot{Q}_{PCM} \quad \ldots\ldots (5)$$

b- for charging process (temperature of HTF lower than the PCM melting temperature):

$$\dot{Q}_{Storage} = \dot{Q}_{PCM} - \dot{Q}_{Evaporater} - \dot{Q}_{Losses} \quad \ldots (6)$$

The internal energy stored in water (HTF)

$$\dot{Q}_{Storage} = \varepsilon \rho_w C_{Pw} V \frac{dT_s}{dt} = \varepsilon \rho_w C_{Pw} V \left(\frac{T_m^{i+1} - T_m^i}{\Delta t}\right) \quad \ldots\ldots (7)$$

Where sub index m indicates the tank where the temperature is being calculated, while i and $i+1$ represent two successive time intervals

Rate of energy transfer from water exit from collector to bed element of thickness (Δx) is given by:

$$\dot{Q}_{Collector} = -m_w^\circ C_{Pw} \left[T_{m+1}^i - T_{m-1}^i \right] \quad \ldots\ldots (8)$$

The energy removal to the tank and supply to evaporate organic fluid can be expressed in coil and tube heat exchanger as:

$$Q^{\bullet}_{Evaporator} = m^{\circ}_{f} C_{Pf} \left(T_{co} - T_{ci}\right) = m^{\circ}_{f} C_{Pf} \left(T_{3} - T_{2}\right) \quad \dots\dots\dots (9)$$

Energy losses from the storage wall to the environment:

$$Q^{\bullet}_{Losses} = U_T A_T \left(T^{i}_{m} - T_a\right) \quad \dots\dots\dots (10)$$

The heat transfer to or from spherical PCM

$$Q^{\bullet}_{PCM} = h_{eff} A_p \left(T_p - T_s\right) = h_{eff} A_p \left(T_p - T^{i}_{m}\right) \quad \dots. (11)$$

for solid phase:

$$\left(1-\varepsilon\right) L \frac{d\Phi}{dt} = h_{eff} \left(T_s - T_m\right) \quad \dots. (12)$$

Where L is the solidification latent heat of PCM.Φ is the solid fraction of PCM and T_s is solidification temperature of PCM.

For liquid phase:

$$\left(1-\varepsilon\right) \rho_l C_{pl} \frac{dT_P}{dt} = h_{eff} \left(T_p - T_m\right) \quad \dots. (13)$$

In order to calculate the heat transfer between the HTF and PCM, one must take the thermal resistance of solidified PCM and the spherical capsule into consideration. So, the effective coefficient is presented [25]:

$$h_{eff} = \frac{h}{1 + M_c + M_s} \quad \dots\dots\dots\dots\dots\dots\dots (14)$$

The convective heat transfer coefficient between HTF and PCM capsules for charging process is determined using the correlation given in Equation (13),[24]

$$Nu = \frac{hd}{k_w} = 2 + 1.1\left[6\left(1-\varepsilon\right)\right]^{0.6} \mathrm{Re}^{0.6} \mathrm{Pr}^{1/3} \quad \dots\dots. (15)$$

And for discharging process it is determined using the correlation given in Equation (16), [26]:

$$Nu = \frac{h_{\infty} d}{k_w} = 2 + 0.65\left[\frac{\mathrm{Pr}}{0.846 + \mathrm{Pr}} Ra\right]^{1/4} \quad \dots.. (16)$$

N_u, R_e, R_a, p_r are Nusselt number, Reynolds number, Rayleigh number and Prandtl number of HTF, respectively .

$$M_c = \frac{R_c}{R_h} = \frac{hr_o\left(r_o - r_i\right)}{k_c r_i} \quad \dots\dots\dots (17)$$

$$M_s = \frac{R_s}{R_h} = \frac{hr_o^2\left(r_i - r_p\right)}{k_s r_i r_p} \quad \dots\dots\dots\dots.. (18)$$

Where M_s and M_c are the ratios of the thermal resistance of solidified PCM and cover of capsules to the thermal resistance due to convection on the external surface of the capsules, respectively.

Substitution of Equations (7), (8), (9), (10) and (11) into Equation (5) yields:

$$\varepsilon\rho_w C_{Pw} V(\frac{T_m^{i+1}-T_m^i}{\Delta t}) = -m_w^\circ C_{Pw}\left[T_{m+1}^i - T_{m-1}^i\right] - m_f^\circ C_{Pf}\left(T_3 - T_2\right) - U_T A_T\left(T_m^i - T_a\right) - h_{efff} A_p\left(T_p - T_m^i\right)$$
…… (19)

$$\varepsilon\rho_w C_{Pw} V(\frac{T_m^{i+1}-T_m^i}{\Delta t}) = h_{efff} A_p\left(T_p - T_m^i\right) - m_f^\circ C_{Pf}\left(T_3 - T_2\right) - U_T A_T\left(T_m^i - T_a\right)$$ ….. (20)

Simplifying of Equations (17) and (18) to solve temperature distribution by explicit finite difference technique depending on the initial and boundary condition is carried out according to Fig. (4):

$$T_m^{i+1} = C_1 T_m^i - C_2 T_{m+1}^i + C_3$$ ………. (21)

$$T_m^{i+1} = C_4 T_m^i + C_5$$ ………… (22)

Where C_1, C_2 , C_3, C_4 and C_5 are constants from Equations (19) and (20).

Initial conditions

$T_s(x,o) = T_a$ For HTF

$T_p(x,o) = T_a$ For PCM

Boundary conditions

$T_s(0,t) = T_{in}$ For HTFat inlet of storage tank

$\frac{\partial T_s}{\partial x} = 0$ For HTFat exit of storage tank

$\frac{\partial T_p}{\partial x} = 0$ For PCM at inlet of storage tank

$\frac{\partial T_p}{\partial x} = 0$ For PCM at exit of storage tank

Organic Rankine Cycle

In order to analyze thermodynamic behavior of ORC, the mathematical models for the components in an ORC are simulated under study- state condition. This simulation requires such conditions as the turbine inlet pressure P_2 and turbine outlet pressure P_1.The cycle efficiency are kept constant under the condition that each pressure ratio (P_2 / P_1) and the turbine inlet vapor is set to b superheated or saturated phase. The working fluid passing through the condenser is assumed to be saturated liquid (T_1 , P_1).In this simulation, each phase of the working fluid is expressed by the equations given below [27]:

Pump Phase: process $(1 \rightarrow 2)$ is a simple pumping of the saturated liquid from low to high pressure, as the fluid is liquid at this stage, the pump requires little input energy.

$$W_p = \frac{m_f^\circ (P_2 - P_1)}{\rho * \eta_p}$$ ………… (23)

Where ρ and η_p are the density of working fluid and isentropic effeminacy of pump respectively.

The specific enthalpy of the working fluid at the pump outlet h_2 is:

$$h_2 = h_1 + \frac{W_p}{m_f^\circ}$$ …………… (24)

Where h_1 is the specific enthalpy of the working fluid at the pump inlet.

Evaporator Phase: process $(2 \rightarrow 3)$ the high pressure liquid enters evaporator at constant pressure by heat exchange with storage which becomes dry saturated vapor at turbine inlet. The evaporator outlet specific enthalpy of working fluid is given by the following equation:

$$h_3 = h_2 + \frac{Q_{in}}{m_f^\circ} \quad \text{………………. (25)}$$

where h_2 and h_3 are the specific enthalpy at the evaporator inlet and outlet respectively.

and Q_{in} which is equal to the gained power by solar collector transfer from HTF to the working organic fluid in the coiled and tube heat exchanger is calculated as:

$$Qi = m^\circ C_{pf}(T_3 - T_2) \quad \text{………….. (26)}$$

where T_2, T_3, C_p and m_f° are the fluid inlet, outlet temperatures in evaporator , heat capacity and mass flow rate of the agent fluid, respectively.

Turbine Phase: process $(3 \rightarrow 4)$ the dry saturated vapor expands adiabatically through turbine generation power .This decreases the temperature and pressure of the vapor, and some condensation occurs.

$$W_t = m_f^\circ \eta_t (h_3 - h_4) \quad \text{………….. (27)}$$

Condenser Phase: process $(4 \rightarrow 1)$ the vapor enters the condenser and cools at constant pressure and temperature to be a saturated liquid. The pressure and temperature of the condenser are fixed by the temperature of the cooling coils as the fluid undergoes phase-change.

$$Q_c = m_f^\circ (h_4 - h_3) \quad \text{………….. (28)}$$

Rankine cycle efficiency:

$$\eta_R = \frac{W_t - W_p}{Q_{in}} \quad \text{…..……… (29)}$$

Numerical solution

The numerical resolutions of the set of equations are solved using MATLAB program. The program calculates the solar gain for the specified system, based on the insolation, the ambient temperature, the latitude, the parameters specifying the solar collector system, the volume of storage tank, the demand turbine power generation in ORC system Explicit finite difference method used to solve Equations (21) and (22) inside thermal energy storage .

Results and discussion

In this part, numerical results will be presented and discussed for one dimension mathematical model thermal energy storage for Solar ORC power generation system using PCM (Paraffin wax) capsules. The equations obtained for the model have been solved numerically by using finite difference technique in MTLAB program to investigate the effect of the various parameters on transient thermal performance of packed bed thermal energy storage.

The temperature distribution for HTF and PCM along the storage axial depth for different time intervals for charging and discharging process is shown in Fig.(5). During the charging process, the temperature quadratic increases for both HTF and PCM Fig.(5.a) and the temperature of HTF is higher than the temperature of PCM as a result of increasing the heat from solar collector. During the discharging process, a temperature quadratic decreases for both HTF and PCM Fig.(5.b) and the temperature of PCM is higher than the temperature of HTF when the pump of solar collector starts the heating loop.

Figure (6) shows a typical HTF temperature distribution with respect to axial position for four time interval during charging process. It can be seen that, the temperature of HTF changes drastically at the upper part of the storage tank are decreases slowly at the lower part and temperature increases with time along the axial depth of tank. The effect of water mass flow rate on the temperature of HTF is quite apparent in Fig.(7) it can be seen that the temperature of HTF rises with the increase in water mass flow rate and the maximum temperature occurs for water mass flow rate of 6 kg/min which is attributed to the increase in the heat transfer between water and PCM capsules.

In Figs.(8) and(9)The effect of geometrical parameters of system on the temperature distribution of HTF along the axial position of storage tank can be seen. The temperature of HTF rises with the increase in the number of capsules (which decreases the bed porosity) as a result of increasing the exchange surface of heat transfer between HTF and PCM capsules .The effect of capsule diameter is shown in Fig.(9) where the results ensure the increase of the temperature of HTF for all position of storage tank with the increase in the capsules diameter from 80 mm to 160 mm at the same porosity .

Instantaneous variation of heat stored and released in the storage tank as a function of the initial storage temperature shown in Fig.(10) and (11),Fig. (10)shows the instantaneous variation of heat stored as a function of the inlet HTF temperature. It can be observed clearly that, the instantaneous heat stored increases with the increase in inlet HTF temperature, and the heat stored curves begin to collapse onto one after 20 minutes during the process of thermal energy storage and continue to be stored for a long time ,Fig. (11) shows the instantaneous variation of heat release as a function of the inlet HTF temperature for discharging process, It can also be seen that, the instantaneous heat release increases significantly with the increase in inlet HTF temperature and quadratic decreasing with time until complete discharging process finishes at time equal to 260 min after charging process time.

Cumulative variation of heat stored and released in the storage tank as a function of the initial storage temperature for complete phase change during the charging and discharging process from starting working of pumps for HTF and organic fluid shown in Figs.(12) and (13).It can be seen from these figures that, heat storage and release increase significantly with the increase in inlet HTF temperature and increase rapidly with time ,for charging process. The stored energy increases continuingly due to the high temperature differences between HTF and PCM as a result of continuous solar collector water heating at sun shine .Even after the sun set and during the discharging process, the energy release decreases with duration

and the ORC stays working and slowly off.

The effect of turbine inlet temperature (TIT) of ORC on temperature of HTF along the axial position of storage tank is shown in Fig.(14) and it can be seen from this figure that the temperature of HTF is inversely proportional with TIT of ORC due to the increase in heat transfer received from HTF to the coil and tube heat exchanger to evaporate organic fluid which increases the efficiency of ORC with the highest value of TIT.

Figure (15).shows the variation of the ORC thermal efficiency with the turbine inlet temperature (TIT), while the turbine inlet pressure (TIP) is kept constant for output turbine power (TOP)5kW for different organic fluid. From the figure, it is shown that the ORC efficiency for all working fluids is a nearly linear function of temperature. It is observed that, R152a is of the highest ORC efficiency at TIT 90 ºC and R227 is not suitable for low temperature because of its lowest efficiency other organic fluids. The efficiency of ORC is much lower as a result of the lower temperature range for all working fluids used.

Figure (16). illustrates the effect of incident solar radiation on inlet temperature of HTF to storage tank which is exit temperature from solar collector with different values of HTF mass flow rates(from 3kg/min to 6kg/min),where temperature increases with the incident radiation for all values of mass flow rates and the maximum increase occurs at mass flow rate of 3 kg/min .It is noted that, the increase in mass flow rate will decrease the inlet temperature of HTF which is of opposite influence on the temperature distribution inside the packed bed storage (temperature of HTF rises with the decrease in water mass flow rate, as shown in Fig.(7)).

To examine the validity of model of the present work a comparison between the present model result and available experimental solar water heating system published data shown in Fig.(17) is carried out concerning the temperature distribution of HTF during the charge process at the same conditions for storage tank. The model results are in good agreement with the experimental results from reference [28].

Conclusion:

A theoretical study has been conducted to investigate the processes of charge and discharge of PCM spherical capsules in thermal energy storage used for domestic solar organic Rankine cycle power generation system. Based on the present results, the following conclusions can be drawn:

1-The highly inlet HTF temperature reduces the total charging time of PCM by increasing the stored heat. However, the reduction of HTF temperature reduces the performance of the solar ORC system.

2-Solar radiation increases the inlet HTF temperature with lower mass flow rate (increasing solar radiation 100 W/m²causes an increase in HTF temperature about 10 °C) and the total charging time of PCM decreases with the increase in inlet HTF temperature or increase flow rate.

3-Lower porosity indicates longer working time of power generation by solar organic Rankin cycle due to longer time required for complete discharging process (decreasing porosity 0.1causes an increase in HTF temperature about 1.5 °C) which increasing discharging time for PCM.

4-The turbine inlet temperature is an important design parameter for the thermal storage in a packed bed used in solarORC system, increasing in TIT causes a decrease in HTF temperature by factor 1/10 which increasing in charging time of PCM.

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Nomenclature

A_a = Aperture area of collector (m^2)

A_T = Surface area of cylindrical tank (m^2)

A_P = Surface area of spherical capsule (m^2)

C_{pw} = Heat capacity of water (J/kg °C)

C_{pf} = Heat capacity of organic fluid (J/kg °C)

C_{ps} = Heat capacity of solid phase PCM (J/kg °C)

C_{pl} = Heat capacity of liquid phase PCM (J/kg °C)

GWP= Global Warming Potential

H = Incident solar radiation(W/m^2)

k_c=Thermal conductivity of capsule material (W/m °C)

k_s= Thermal conductivity of solid phase PCM (W/m °C)

k_w= Thermal conductivity of water (W/m °C)

heff= Effective coefficient of convective heat transfer (W/m^2 °C)

h_∞= Convection heat transfer coefficient (W/m^2 °C)

h= Specific enthalpy (kJ/kg)

L = Solidification latent heat of PCM (kJ/kg)

N_u = Nusselt number

ODP= Ozone Depletion Potential

ORC= organic Rankine Cycle

Pr= Prandtl number

PCM= Phase Change Material

Re=Reynolds number

Ra=Rayleigh number

r_i = Inner radius of capsules (m)

r_o = Outer radius of capsules (m)

r_p = Solid – liquid interface (m)

T_a = Ambient temperature (˚C)

TIP= Turbine inlet pressure (pa)

TIT= Turbine inlet temperature (˚C)

TOP= Turbine output power (kW)

ε = Porosity of packed bed

Φ = Solid fraction of PCM

ρ = Density(kg/m³)

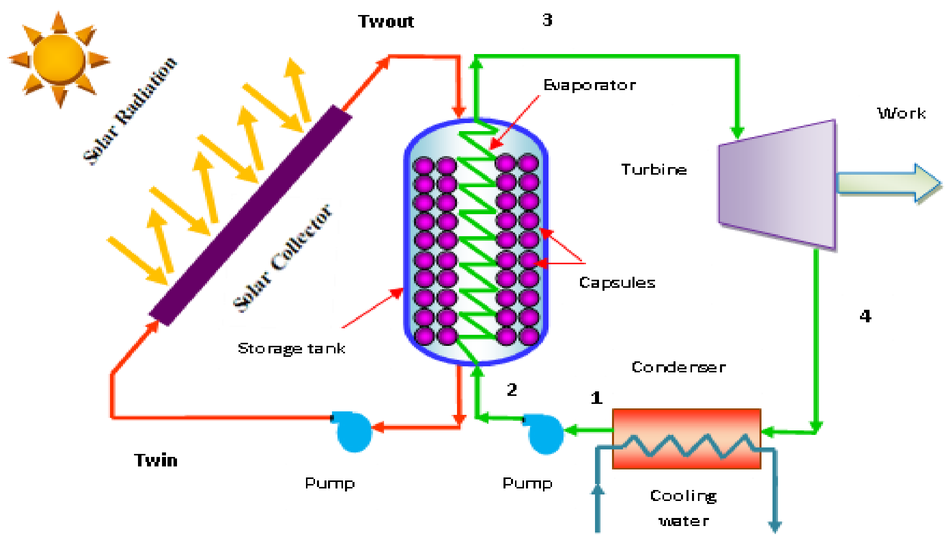


Fig. (1): Schematics of the low temperature solar organic Rankine cycle system containing latent heat thermal energy storage with spherical capsules.

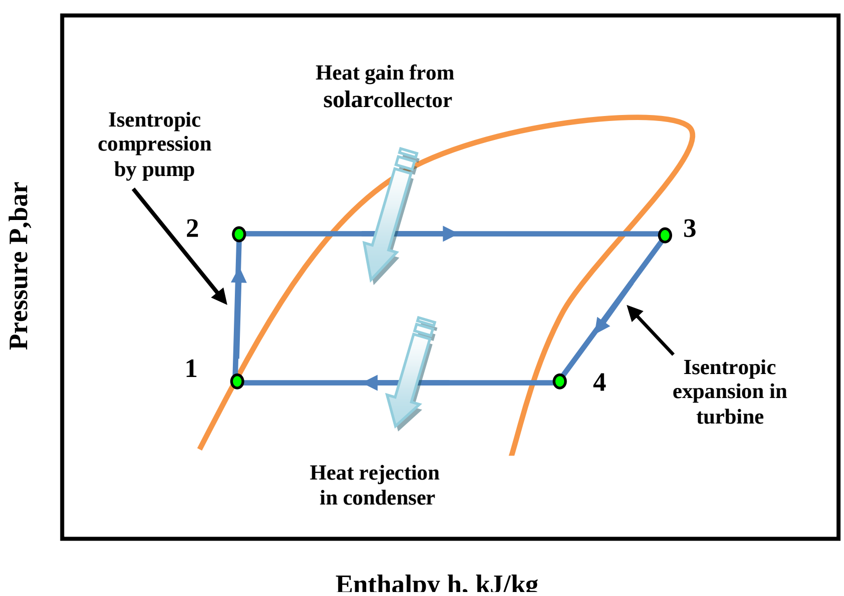


Fig.(2): p-h diagram of a typical Rankine cycle

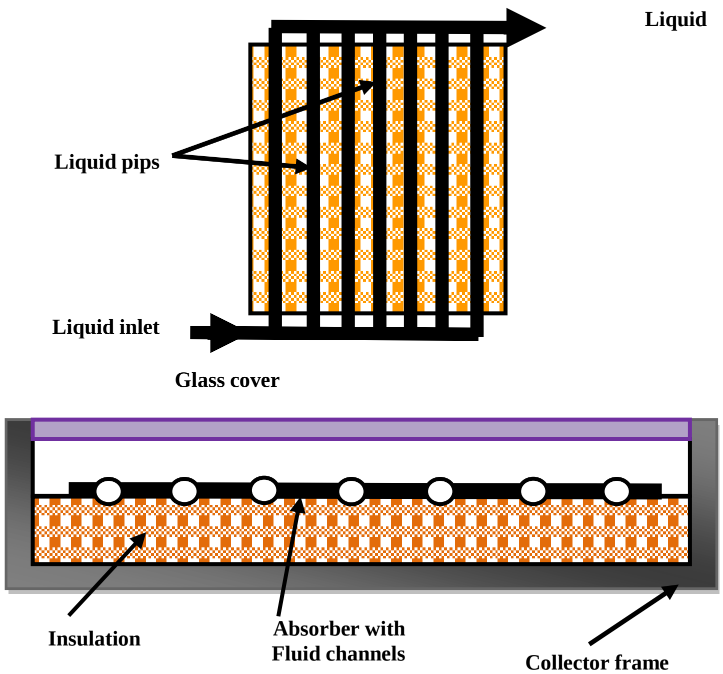


Fig.(3): Flat plate solar collector model

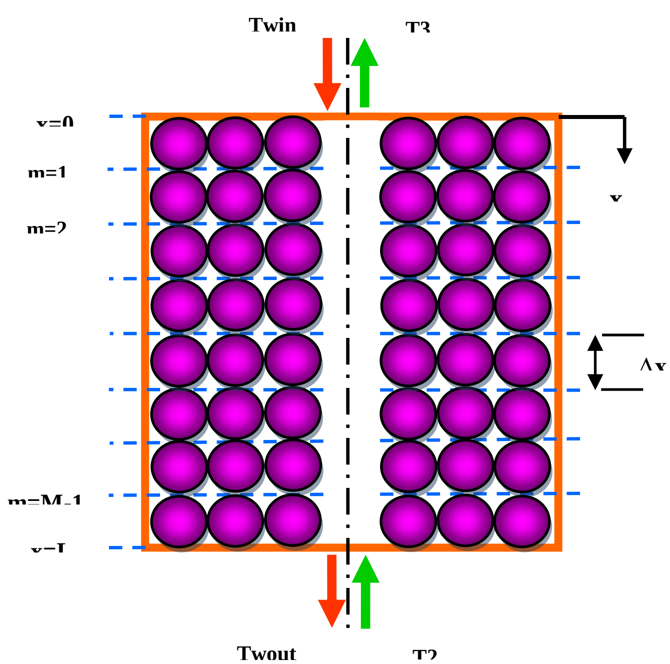
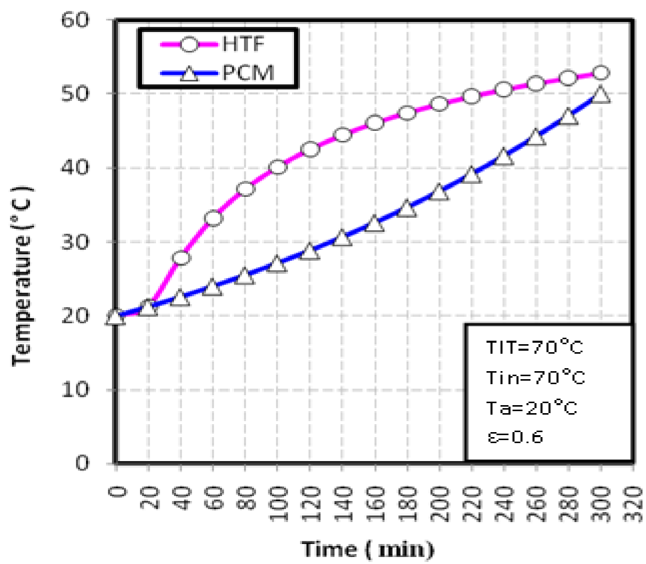
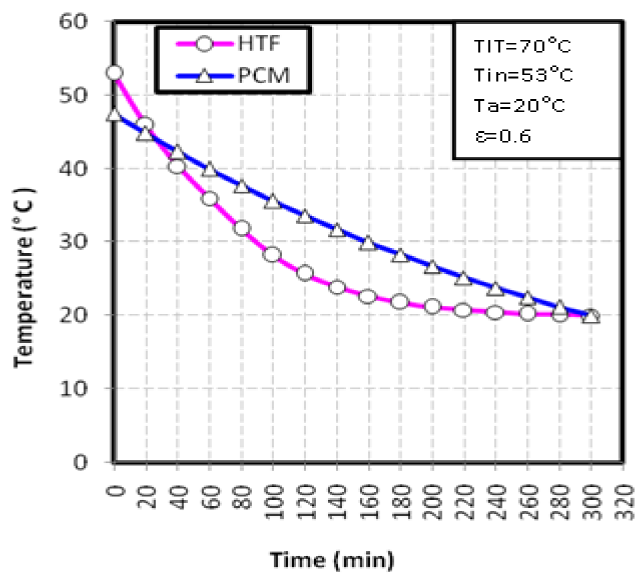


Fig.(4): thermal storage model for configuration numerical analysis



a-charging process



b-discharging process

Fig.(5):Temperature variations of HTF and PCM capsules at the top layer

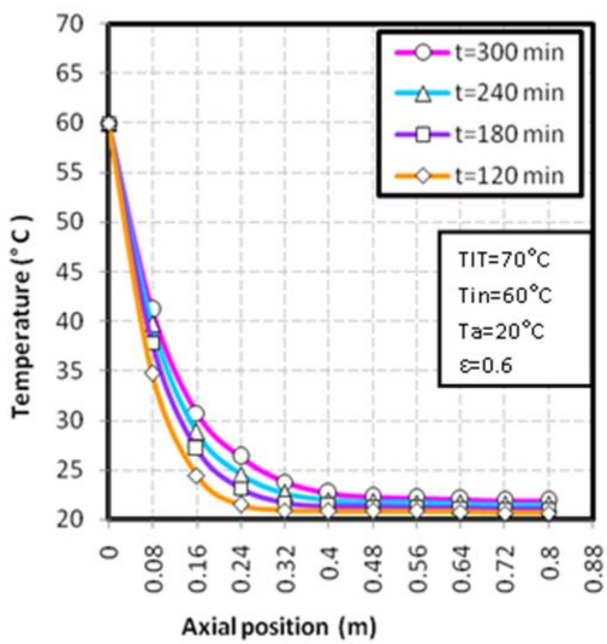


Fig.(6): Effect of time on Temperature c

HTF

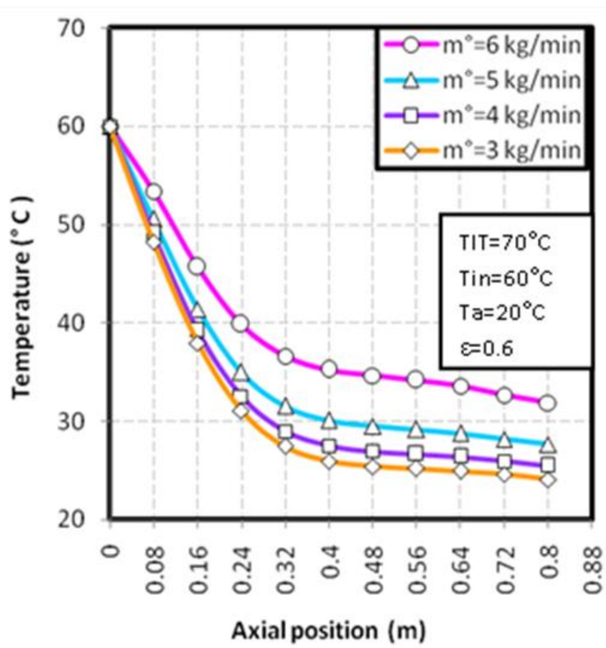


Fig.(7): Effect of flow rate on Temperature of HTF

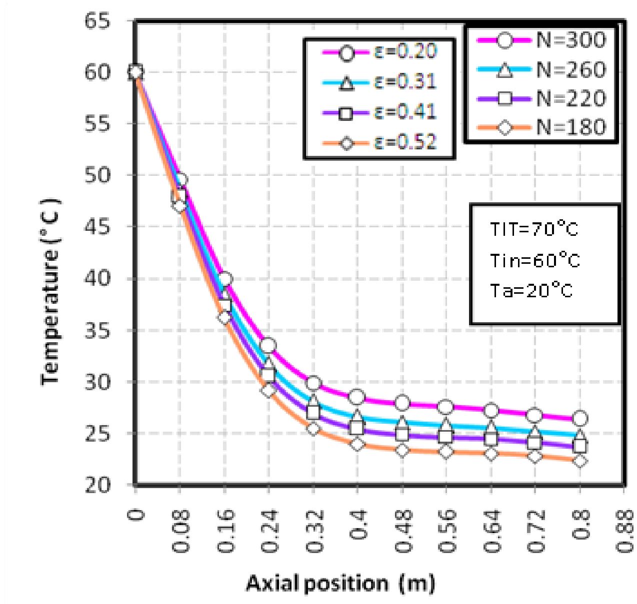


Fig.(8): Effect of porosity of bed on Temperature of HTF

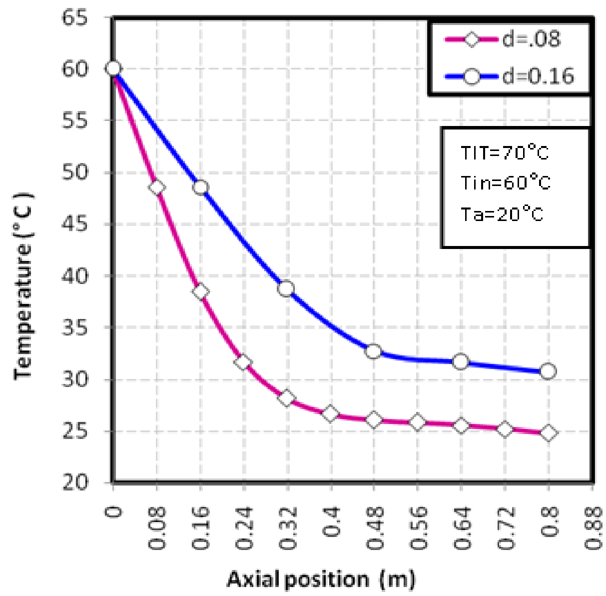


Fig.(9): Effect of diameter of capsules on Temperature of HTF

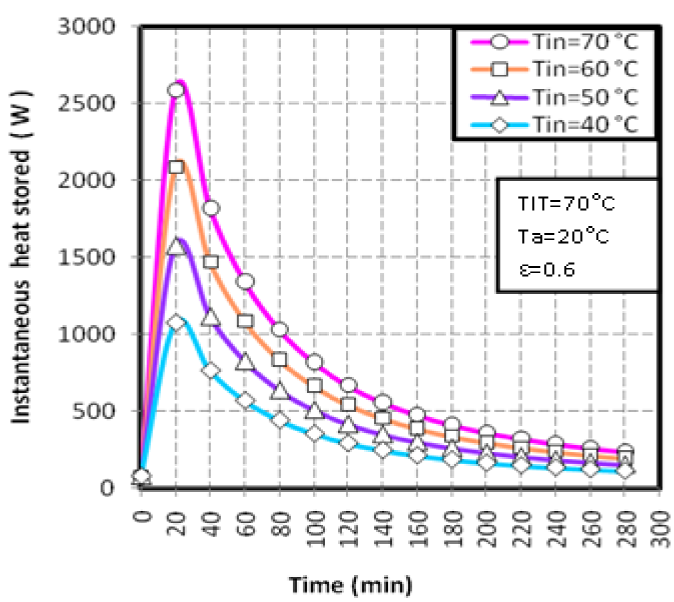


Fig.(10): Instantaneous heat stored for charging process

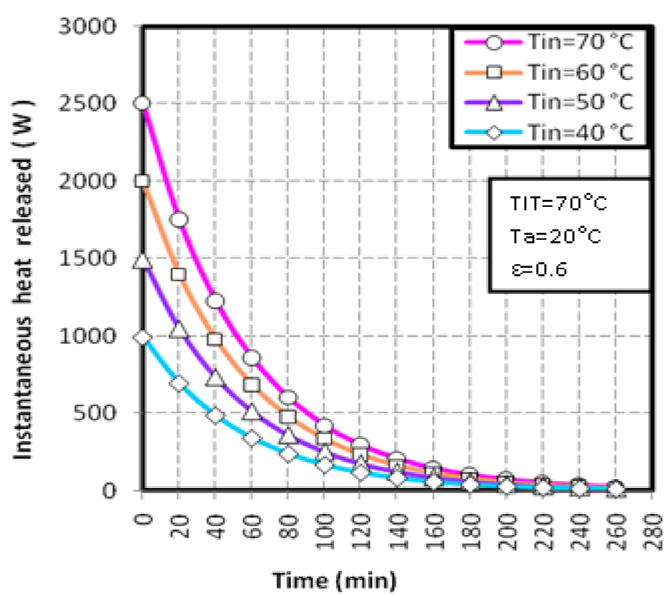


Fig.(11): Instantaneous heat released for discharging process

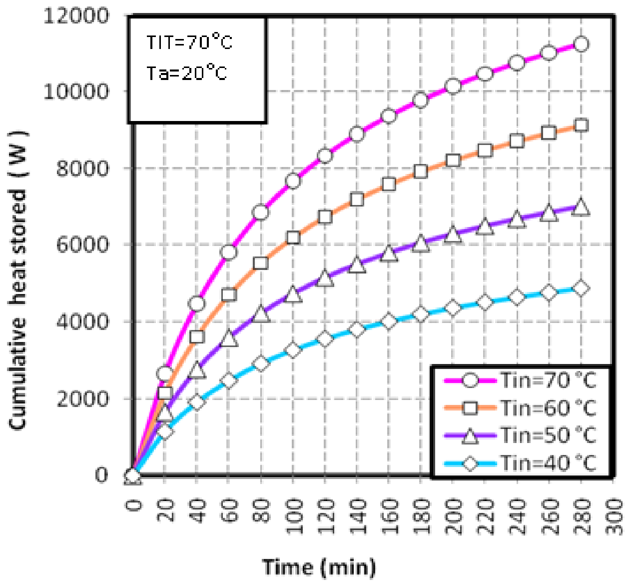


Fig.(12): Cumulative heat stored for charging process

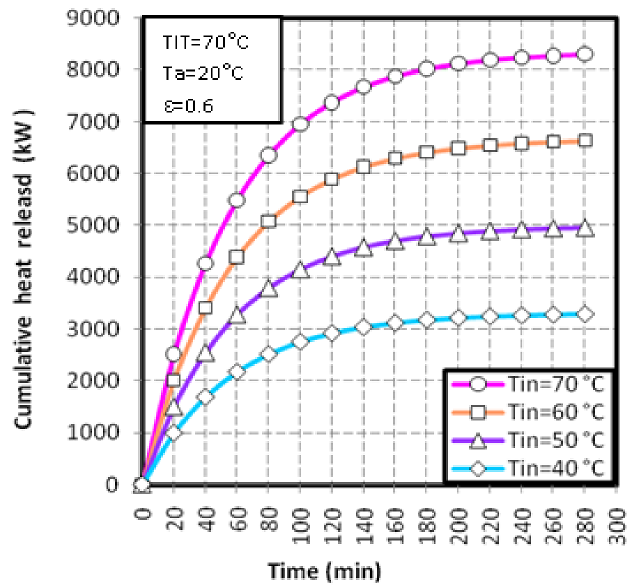


Fig.(13): Cumulative heat released for discharging process

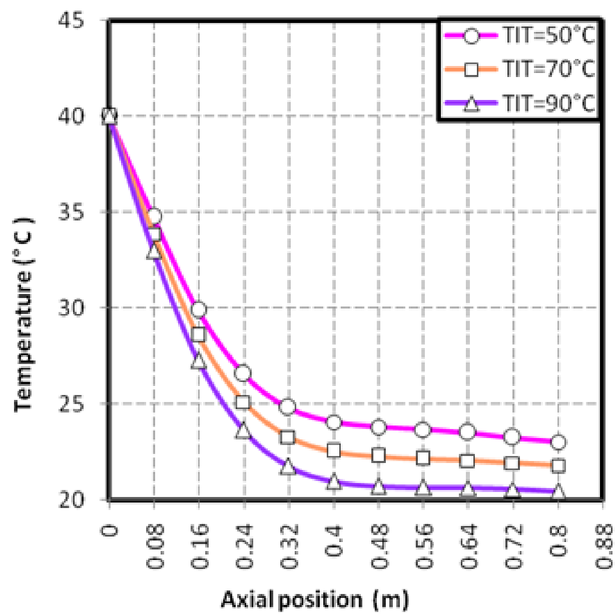


Fig.(14): Effect of turbine inlet temperature on Temperature of HTF

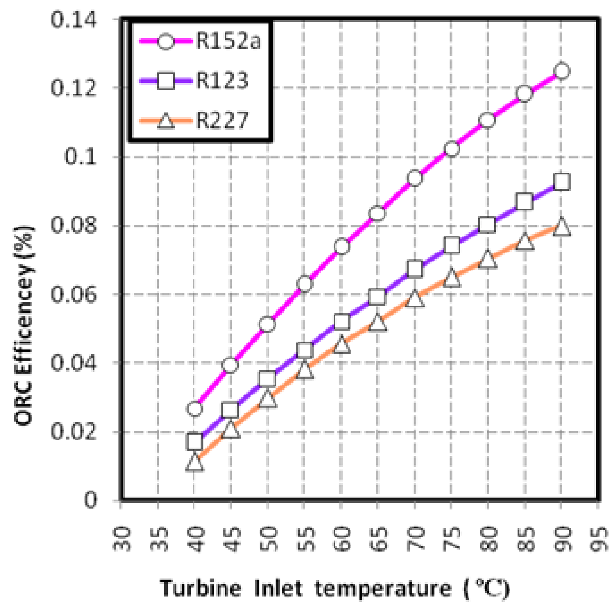


Fig.(15): Variation of the ORC efficiency with turbine inlet temperature

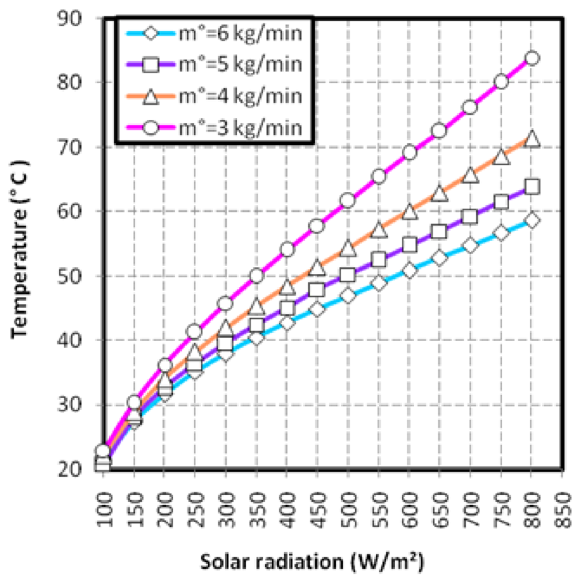


Fig.(16): Effect of Solar radiation on HTF inlet temperature to storage tank

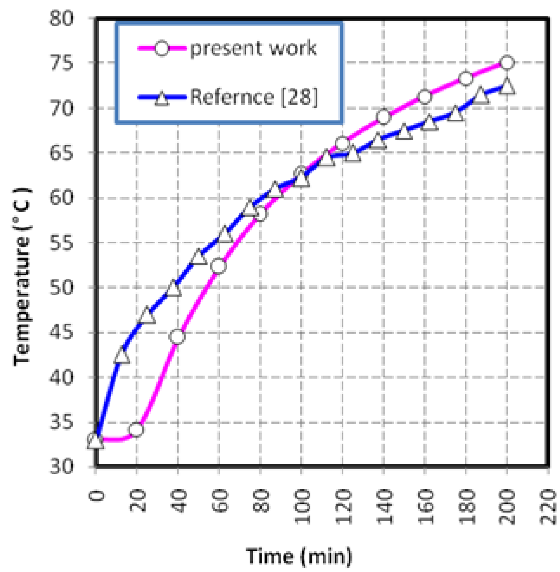


Fig.(17): Comparison between predicted temperature distribution and experimental from reference [28]

Table (1): Thermo-physical properties of paraffin wax [24].

Melting temperature (°C)	Specific heat (kJ/kg°C)		Density (kg/m³)		Thermal conductivity (W/m°C)		Latent heat of fusion (kJ/kg)
	solid	liquid	solid	liquid	solid	liquid	
50	2.9	2.1	930	830	0.24	0.15	190

Table (2): parameters used in numerical simulation.

Length of the collector	2 (m)	Length of the absorber plate	1.95 (m)
Width of the collector	1.5 (m)	Width of the absorber plate	1.45 (m)
Diameter of the tube	16 (mm)	Overall heat transfer coefficient (U_L)	8.9 $(W/m^2 °C)$
Number of tubes used	7	Collector heat removal factor(F_R)	0.87
Slope of the collector	40°	Absorptance of glass cover (α)	0.92
Turbine efficiency (η_t)	0.85 (%)	Transmittance of glass cover (τ)	0.85
Pump efficiency (η_p)	0.85 (%)	Ambient temperature (T_a)	20 (°C)
Condenser temperature (T_1)	300 (K)	Storage tank losses coefficient (U_T)	5.0 $(W/m^2 °C)$
Condenser pressure (P_1)	1 (bar)	Number of collectors used	3

Table (3): Environmental impacts and properties of organic working fluids [5,6].

Working fluid	ODP	GWP 100	T_c (ºC)	P_c(bar)	T_{boi} (ºC) at 1 bar
R125a	0	140	123.26	45.17	-24
R123	0	120	101.78	36.6	-16.4
R227	0.02	93	183.79	30.00	27.8

التحليل النظري لخزن الطاقة الحرارية باستخدام كبسولات المواد متغيرة الطور في نظام توليد الطاقة بدورة رانكن العضوية الشمسية

حيدر محمد جفال
الجامعة المستنصرية – هندسة البيئة

الخلاصـــــة:

يعرض هذا البحث تحليلاً نظرياً لخزن الطاقة الحرارية في مهد محشو بمادة متغيرة الطور لتحسين أداء دورة رانكن العضوية الشمسية لنظام توليد الطاقة الكهربائية وذلكبتمديد وقت التشغيل المطلوب إلى الليل .السلوك الغير مستقر لخزن الطاقة الحرارية في مهد محشو بكبسولات مملوءة بشمع البرافين كوحدة خزن تم دراسته عددياً باستخدام طريقة الفروق المحددة لعملتي الشحن والتفريغ.تأثيرات محددات مختلفة مثل درجة حرارة الدخول ,معدل التدفق ,مسامية المهد ,قطر الكبسولة ودرجة حرارة الدخول للتوربين وضحت لتكون محددا مهيمناً في إيجاد الكفاءة الحرارية الفعالة لدورة رانكن العضوية الشمسية .لوحظ إن درجة حرارة الدخولالى التوربين تلعب دورا قويا في المعايير الهندسية للمهد المحشو.تمت مقارنة النتائج النظرية مع نتائج عملية لدراسة سابقة لنظام تسخين المياه وظهر توافق جيد بين الدراستين.