COMPUTATIONAL TOOL TO MODEL AND SIMULATE SOLAR ASSISTED ORGANIC RANKINE CYCLE WITH A THERMAL ENERGY STORAGE

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DECLARATION

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S.V.R. Gamage Date

The above candidate has carried out research for the Masters Dissertation under my supervision.

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Prof. K.K.C.K. Perera Date

To My Beloved Parents

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On the successful completion of my thesis, I would like to express my regards and thoughts, to acknowledge the contributions of all those, who stood beside me and supported me towards the completion of my MSc.

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ABSTRACT

The Organic Rankine Cycle (ORC) is considered as one of the most promising methods to convert low grade heat into the power. The ORC energy conversion process is much similar to the typical Rankine cycle except for the working fluid. The ORC applicability with low critical point organic fluids enables the operation of the system with low temperature heat sources. This makes low grade solar thermal, waste heat and geothermal suitable heat sources for power generation. Moreover, this applicability of small scale power generation makes it popular for standalone and low quantity heat source applications.

This thesis presents a novel design of solar collector field along with a thermal energy storage to generate electrical power using an ORC. Concentric and non-concentric solar collectors were used to design the cascade collector array considering two collector operating temperatures. Several different collector arrangements of flat plate, evacuated tube, compound parabolic trough and parabolic trough solar collectors were considered. To overcome the intermittent nature of solar irradiation and to extend the number of operational hours, a thermal energy storage system was integrated to the system. Encapsulated phase change materials submerged in a thermal oil bath was considered for this thermal energy storage.

For this investigation, the ORC system was designed according to the maximum load required. However, for the performance evaluation, part load system parameters variation was considered. Two systems were proposed for the evaluation process named system-1 and system-2. The system-1 consists with flat plate and evacuated tube solar collectors with low temperature thermal energy storage and system-2 contains evacuated and parabolic trough solar collectors with medium temperature thermal energy storage. The mathematical model is developed in this research to evaluate the energy flow through system components on an hourly basis. Hourly and seasonal variation of solar energy potential and energy demand were taken and used to simulate the mathematical model using a novel computational tool developed in this study. The system performances were evaluated based on collector area, the capacity of thermal energy storage and ORC thermal efficiency.

Results from the investigation depict the performance of the proposed cascaded solar collector field with different ORC working conditions in a Sri Lankan context. The system performance evaluation was done for five different organic fluids identify optimal working fluids for different system parameters. The evaluated results show the variation of power output, plant factor and system efficiency with different system configurations. The identification of best system performance should be based on both power output and plant factor. However, identification of optimal system depends on both thermodynamic and economic factors. Therefore based on an economic analysis, normalized energy costs can be calculated to identify the best operating conditions along with economic considerations.

Keywords: *Organic Rankine Cycle, PCM storage, Renewable energy, Solar Thermal*

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1 INTRODUCTION

1.1 Problem Statement

Rural electrification and development in energy infrastructure with renewable energy sources support for country energy security and economic growth. It is a challenging endeavor to extend the existing main grid in order to come up with this when considering present Sri Lankan context. Therefore, it is essential to innovative solutions in order to achieve this task [1], [2]. This work evaluates the capability of using solar thermal Organic Rankine Cycle (ORC) with a thermal storage for such standalone applications. The targeted problem is considering rural area distant from main grid to supply electrical power. The load requirement was targeted considering the average electricity usage of fifteen rural families in Sri Lanka. However, the proposing system can be integrated with other power sources and even with main grid.

1.2 Background

Rankine cycle (RC) can be taken as the thermodynamic cycle used to generate majority of the present electricity demand. In most of the instances, water is used as the working fluid of RC with high steam temperatures. However, thermal efficiency of RCs drops significantly when the temperature of steam drops below 370^{o}C [3]. Therefore, solar thermal electricity generation by using steam RC needs higher initial capital investment with number of techno economical challenges. In such circumstances, Organic fluids can be used to replace water in RCs in order to harness low grade thermal energy.

Mathematical modeling and experimental validation, simulation and optimization solar thermal ORC has recently become focused area of study. Design of solar collector is important when considering the overall efficiency of the ORC since it becomes the thermal source. Flat plate, evacuated type and compound parabolic collectors are taken into consideration in most of the instances when it comes to low temperatures ORC [4]–[8]. Usage of low temperature non-concentric solar collector may able to extract both diffuse and beam radiation and reduce the cost of the system. Heat losses of the solar collector field can be significantly reduced by subdividing solar field in to different temperature stages that consists of both concentric and non-concentric collectors [9]. Optimum balance between concentric and non-concentric collectors is about to taking into consideration. Therefore, optimum design of collector array and collector temperature need to be optimized when developing Solar assisted ORC. However, non-concentrated collectors match with present Sri Lankan context compared to concentrated collectors due to low clearness index.

It is important to come up with thermal storage when it comes to continues power generation. It also helps to smooth the heat supply with interrupted solar irradiation. Thermal energy can be stored either as latent heat or sensible heat. Latent heat thermal energy storage with Phase Change Materials (PCM) may reduce the size of the storage system and maintain constant temperature of storage. There are number of phase change materials in literature with various temperature ranges for different applications. When designing thermal energy storage with subdivided solar collectors, it is important to use two temperature thermal storages for reduce the heat losses and maintenance cost. To improve the heat transfer efficiency, encapsulated PCM with thermal oil is considered for the thermal storage [10]–[12].

Conducting a techno economic analysis considering continuous power supply is important to identify the optimum design parameters. Subdivided solar collector areas, thermal energy storage capacities, type of working fluid and ORC working conditions in Sri Lankan context should be identified though a techno economic analysis to implement this system in commercial scale.

1.3 Study Objectives

- Analyse the Organic Rankine Cycle performance variation with solar collector arrangement
- Evaluate the optimum design parameters, working conditions, working fluid, PCM that would match with Sri Lankan context

1.4 Research Methodology

- Carry out a detail literature survey about existing mathematical model of organic rankine cycle, latent heat thermal storage, solar thermal energy extracting, etc…
- Develop a detailed mathematical model about solar thermal energy extracting for ORC power generation with latent heat thermal energy storage
- Develop a computational tool to simulate the mathematical model
- Determine optimum design parameters, working conditions, working fluids, PCM that would match with Sri Lankan context

2 LITERATURE REVIEW

2.1 Theory

The typical Rankine cycle consists of an evaporator, expander, condenser and feed pump. The most common working fluid is water at high temperature. However, the ORC runs with organic fluids instead of water. The basic Rankine cycle involve with four thermodynamic processes with each components as follows.

1-2 Expander process : Isentropic expansion process

2-3 Condenser process : Isobaric heat rejection process

3-4 Pump process : Isentropic compression process

4-1 Evaporator process : Isobaric heat addition process

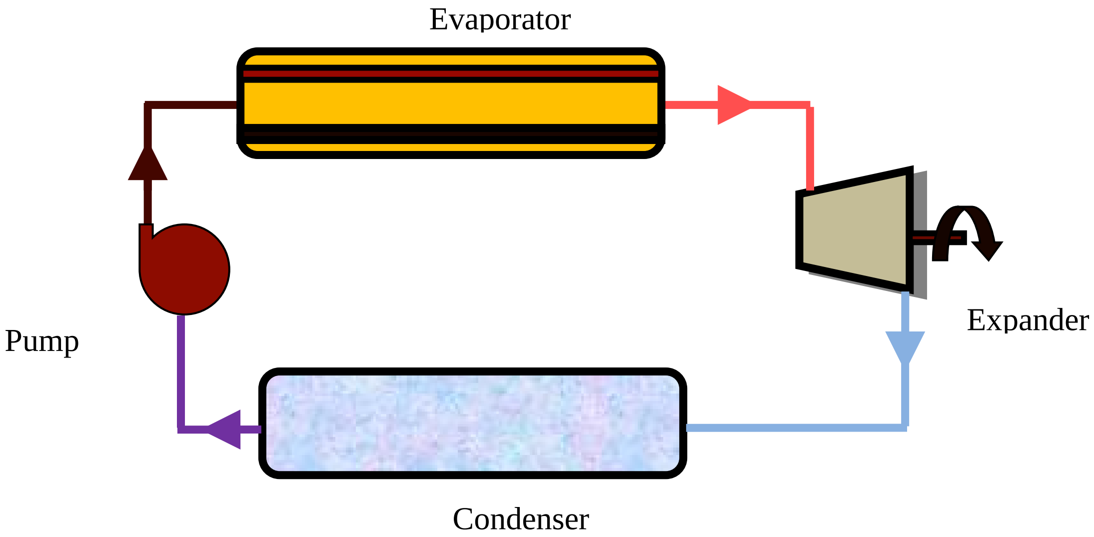


Figure 2.1: Basic Rankine cycle with major components

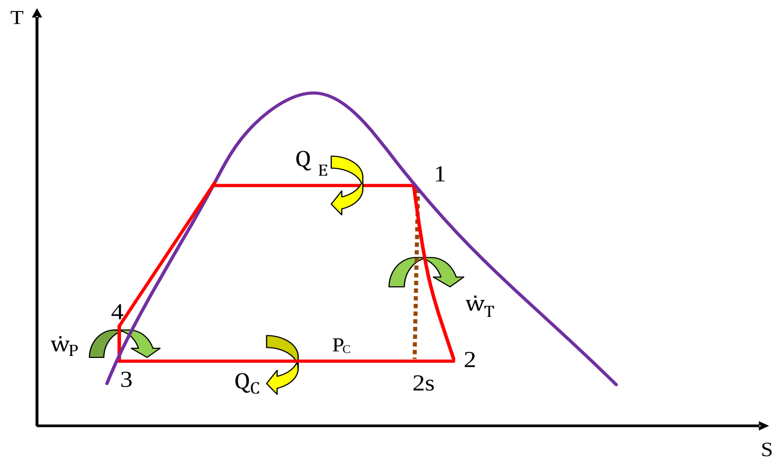


Figure 2.2 : Temperature vs entropy diagram for working fluid water

Rankine cycle (RC) can be taken as the thermodynamic cycle used to generate majority of the present electricity demand in the world. In most of the instances, water is used as the working of RC with high steam temperatures. However, thermal efficiency of RCs drops significantly when the temperature of steam drops below 370^{o}C [3]. Therefore, for low grade heat sources low critical point organic fluids are become good candidate.

Considering thermodynamic aspects working cycle ORC systems can be divided into three classes, based on the shape of the saturated vapor line in a temperature versus entropy diagram. Fluids having a positive slope for saturated vapor line can be identified as dry fluids, fluids with a nearly infinitely large slope are considered as isentropic fluids, and fluids having a negative slope are named as wet fluids. The illustration of those curves are shown in below figure (2.3)

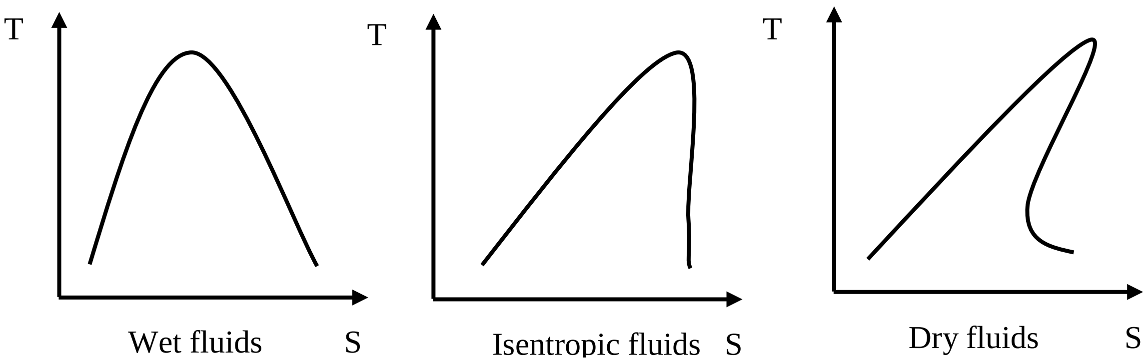


Figure 2.3 : Temperature vs entropy diagrams for different fluids

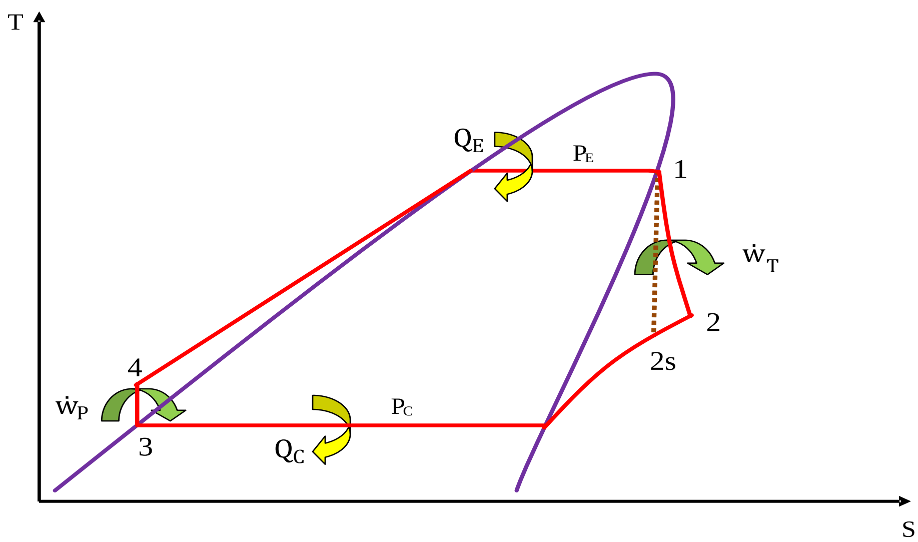
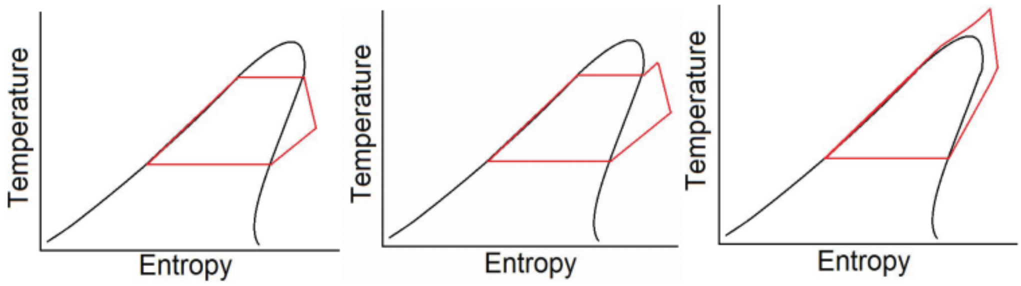


Figure 2.4 : Temperature vs entropy diagram for a dry organic fluid

There are another three types of ORCs according expander inlet conditions. The subcritical ORC system having a saturated vapor at expander inlet (figure 2.5), superheated ORC system having a slightly superheated vapor at the expander inlet and supercritical ORC system having a superheated vapor at expander inlet. In the supercritical ORC system, the evaporator process does not pass through a distinct two phase region like a conventional Rankine or organic Rankine cycle.



Subcritical Superheated Supercritical

Figure 2.5 : Temperature vs entropy diagrams for different expander inlet conditions

2.2 Working fluid selection

The selection of working fluid for an ORC is the most important part when it comes to designing stage. In the literature, still there is no any optimum working fluid because the performance of the working fluid is highly depends on the system parameters. Therefore according to ORC operating parameters working fluid should be selected [13]. Basically there are two types of working fluids as pure fluids and zeotropic mixtures.

Key things to consider to select ORC Working fluid [14]–[16]

- Positive or isentropic saturation vapor curve
- High vapor density
- Low viscosity
- Higher thermal conductivity
- Acceptable evaporating pressure
- Positive condensing gauge pressure
- High temperature stability
- The melting point should be lower than the ambient
- Safety level, toxicity and flammability
- Low ODP and GWP
- Good availability and low cost

Table 2.1 : Comparison between ORC and Steam power cycle

Advantages of the ORC	**Advantages of the steam cycle**
▪ No superheating requirement ▪ Good for small scale applications ▪ Lower turbine inlet temperature ▪ Lower evaporating pressure ▪ Higher condensing pressure ▪ Low temperature heat recovery	▪ Higher efficiency ▪ Low-cost working fluid ▪ Environmental-friendly working fluid ▪ Non-flammable, non-toxic working fluid ▪ Low pump consumption ▪ High chemical-stability working fluid

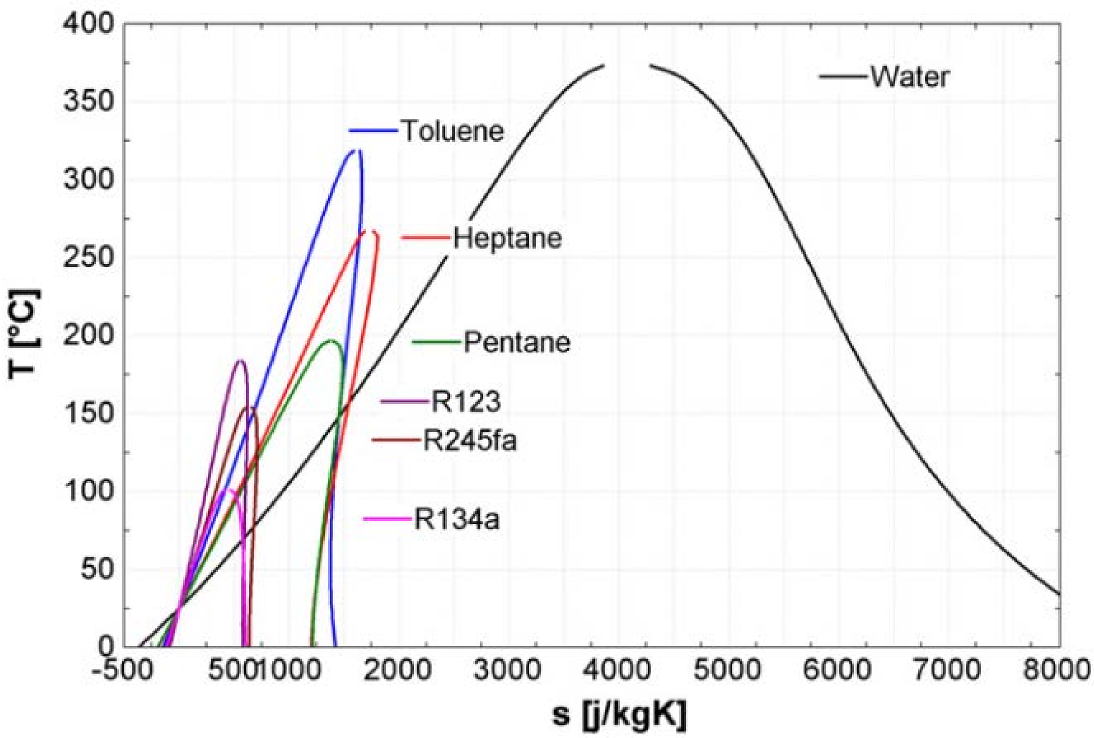


Figure 2.6 : T-S diagram for water and some organic fluids [17]

2.3 Organic Rankine Cycle design

2.3.1 Expander

The selection of expansion machine is one of the very important parts of ORC design. The system thermodynamic performance highly depends on expansion device. According to the range of power generating and application, type of expander should be selected.

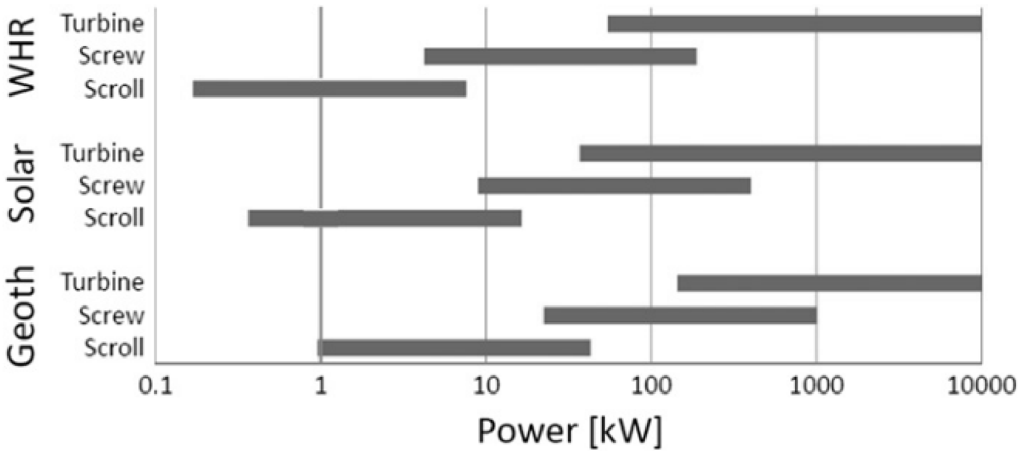


Figure 2.7 : Selection of expansion machine according to applications [17]

Basically there are two types of expansion machines for ORC as positive displacement machines and turbo expansion machines. However turbo expansion machines are mostly selected for high flow rate, high rotational speed applications. The ORC systems are mostly involved with lower flow rates, low speed and relatively low pressure ratios. Therefore positive displacement machines are mostly suitable for ORC systems.

2.3.2 Positive displacement expanders

The applications of positive displacement expanders in industry are less in popular. Therefore most of the expanders are used for ORC systems after modifying existing positive displacement compressors [17]. The mostly used positive displacement expanders are piston, screw, scroll and vane expanders. In literature commonly, used expander in small scale ORC systems (less than 50 kW) is scroll type following screw type (10-1000 kW).

Quoilin [18] has studied analytically and experimentally performance of scroll expanders for small scale ORC applications. What is more, he has investigated system behavior with different working fluids, working temperature, and power output necessary to utilize a small turbine. However, expander isentropic efficiency is highly depends on volume flow rate, rotational speed, volume ratio and pressure ratio. It is important to consider expander efficiency variation specially with variable load ORC systems. M. Astolfi [19] has investigated the expander efficiency variation with volume flow rate and volume ratio for screw expanders. From that investigation, empirical equation about expander efficiency has been proposed for screw expanders [20]–[23].

3 System description

For this investigation 37.5kW solar thermal plant was considered, with hourly variation of both electricity demand and solar irradiation level. The considered load curve variation between 28.0 kW and 37.5 kW. For the system thermodynamic performance analysis five organic working fluids were considered. The system performances are evaluated based on different system arrangements such as changing collector area, area ratio and thermal storage capacity. Two systems were proposed for the evaluation process. System-1 consists of flat plate and evacuated tube solar collectors with thermal energy storage at 149 ^oC. System-2 contains evacuated and parabolic trough solar collectors with thermal energy storage at 176 ^oC.

For the proposed system electricity load and solar irradiation are considered as external inputs. According to those two parameters and with selected design parameters system performances are expected to calculate. After evaluating system performance optimal collector area, working fluids, storage size and collector ratios are expected to decide.

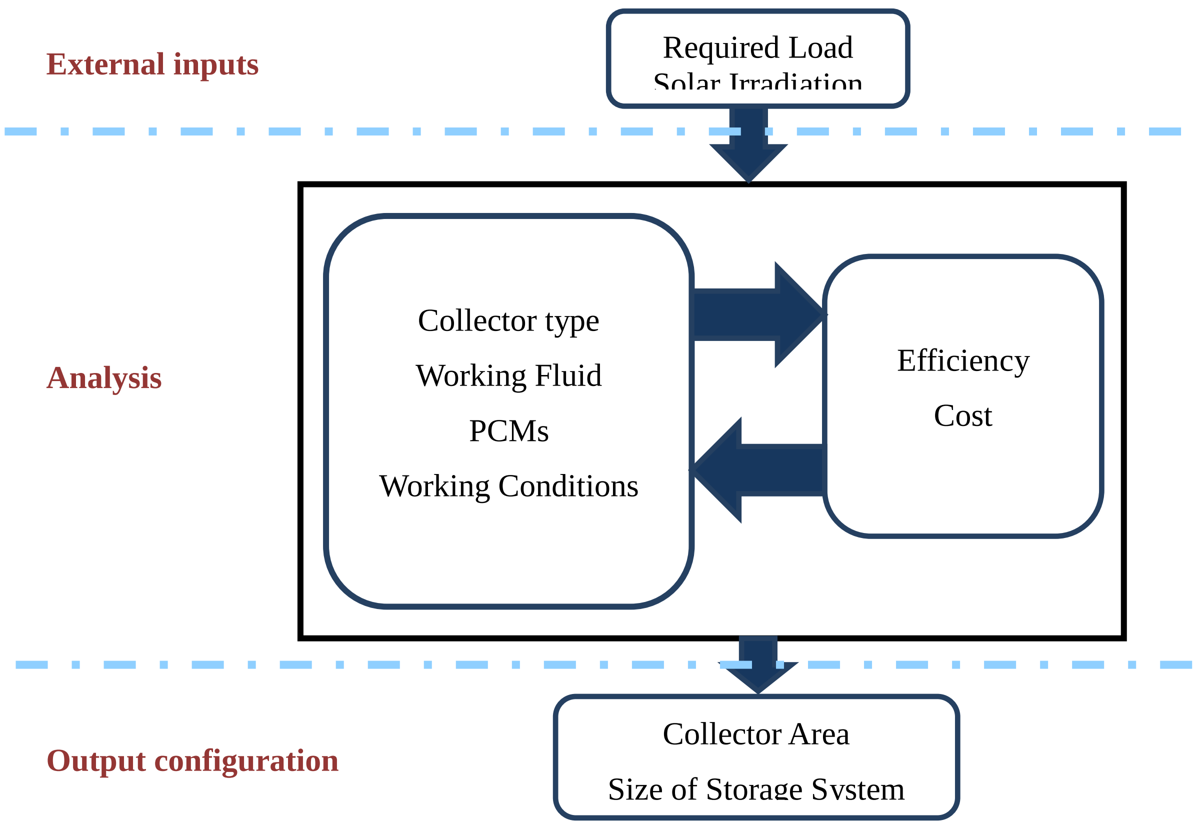


Figure 3.1 : Input and output parameters of the proposed system

3.1 Solar collector area

For this proposed system four types of solar collector systems were selected to investigate the system performance. However, for each evaluation process only two types of collectors were considered. Two stage cascade type arrangement was considered extract heat from solar irradiation. Basically four types of solar collector arrangements were identified as follows.

Configuration – 1 : Flat plate and evacuated type solar collector system

Configuration –2 : Flat plate and compound parabolic trough type solar collector system

Configuration – 3 : Compound parabolic trough and parabolic trough solar collector system

Configuration – 4 : Evacuated type and parabolic trough solar collector system

According to collector working configuration four system evaporator temperatures were selected. However, configurations 1 and 2 are operating at same temperature level while system 3 and 4 are operating at another temperature level. Therefore, the system was designed for two thermal storages, one for configuration 1,2 and second for configurations 2,3. According to thermal storage temperature suitable phase change materials were selected. In summary, two working systems are as follows.

System -1 : Non concentric collectors to extract solar heat, low temperature phase change material (near to 150 ^{o}C) for storage system.

System -2 : Both non concentric and concentric collectors to extract solar heat, medium temperature phase change material (near to 175 ^{o}C) for storage system.

3.2 Thermal energy storage

Thermal energy extract from the solar collectors are transferred to the ORC and thermal storages using conductive heat transfer thermal oil. Therminol Vp1 heat transfer oil considered as that heat transfer oil. The same thermal oil is used as thermal oil bath for encapsulated phase change material thermal energy storage.

The thermal energy storage system was designed considering phase change materials. The importance of phase change material storage system is to maintain constant storage temperature and reduce the storage volume. There are two phase change materials were selected for this analysis. The selected phase change materials were located inside the storage as encapsulated spherical shape capsules. All those encapsulated phase change material spheres were designed to place inside a thermal oil bath. Then for the analysis process, the thermal oil bath temperature was assumed as a constant. The charging and discharging rates variation was considered for this study. The details of those phase change materials are listed in below table

Table 3.1 : The details of two systems considered for this study as follows [24]

Design parameter	System -1	System -2
Collector type	Flat plate-evacuated type collectors / Flat plate - compound parabolic	Evacuated-parabolic trough collectors / Compound parabolic-parabolic trough collectors
Evaporator temperature range	130-142 ^oC	155-168 ^oC
Condenser temperature range	38-42 ^oC	38-42 ^oC
Pressure ratio range	7-15	7-20
Phase change material	KNO_2 $NaNO_3$	HCOONa – HCOOK
Thermal storage temperature	149 ^oC	176 ^oC
Heat transfer oil	Therminol Vp1	Therminol Vp1

Table 3.2 : Properties of PCM

Property	KNO_2 $NaNO_3$	HCOONa – HCOOK
Melting temperature (^oC)	149	176
Latent heat of fusion (kJ/kg)	124	175
Density (kg/m^3)	2080	1913
Energy density (kWh/m^3)	70	92
Cost by volume (£/m^3)	994	421
Cost by energy capacity (£/kWh)	14	4.6

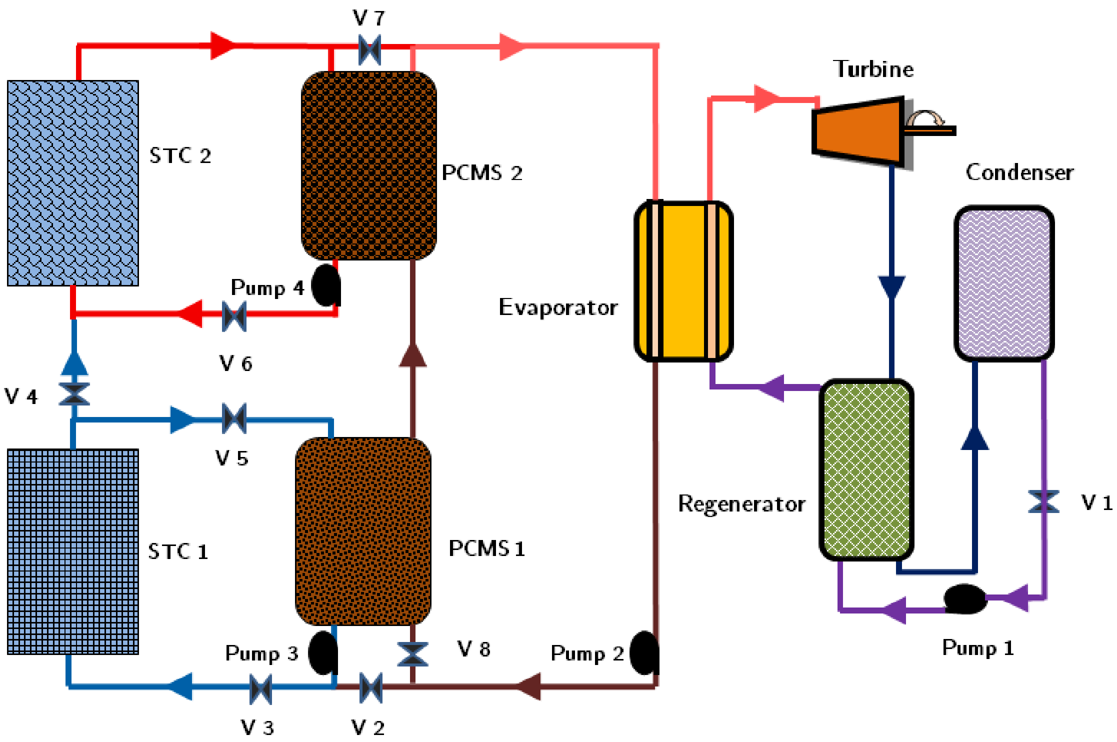


Figure 3.2 : Schematic diagram of the proposed system

3.3 Dispatch strategy – works in three modes

1. Charging Cycle
    - When thermal energy production is higher than required evaporator heat load.
2. Combined Discharge Cycle
    - When thermal energy production is not enough to supply required evaporator heat load.
3. Discharge Cycle
    - When solar irradiation is not available

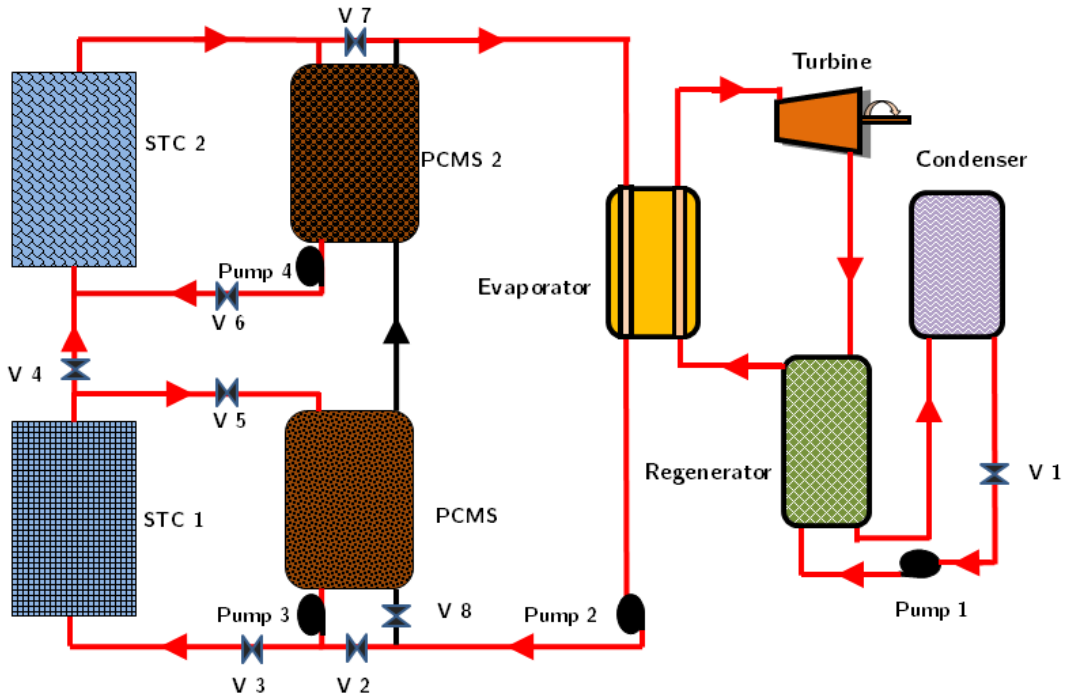


Figure 3.3 : Charging Cycle, valve 8 is closed, all pumps are working

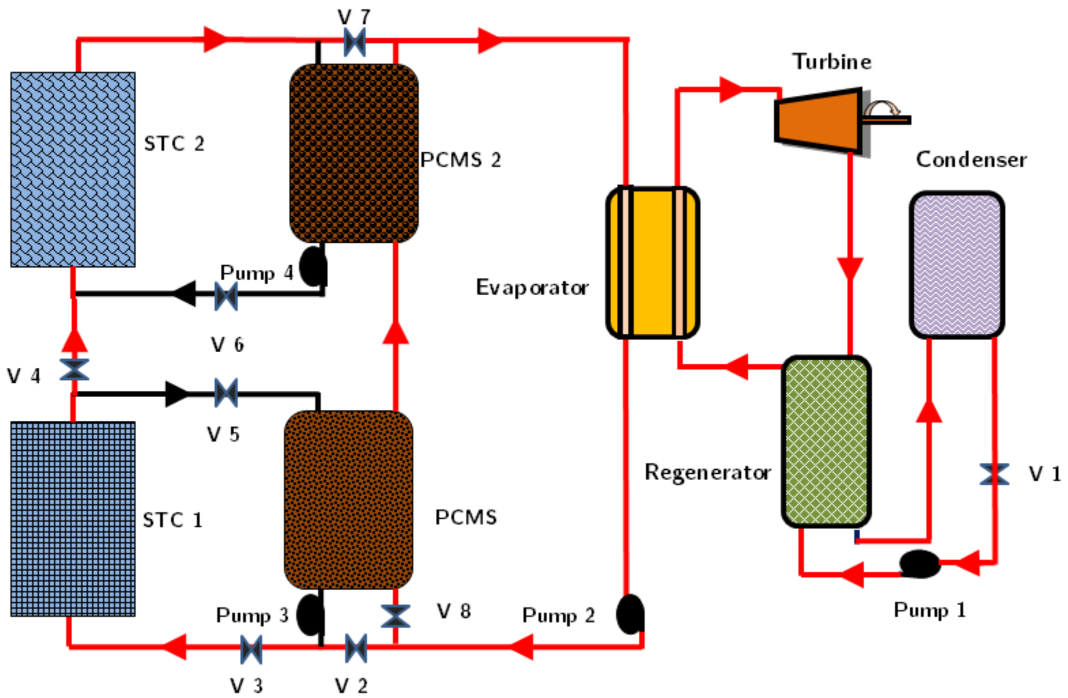


Figure 3.4 : Combined discharge cycle, valves 5,6 are closed, pumps 1,2 are working

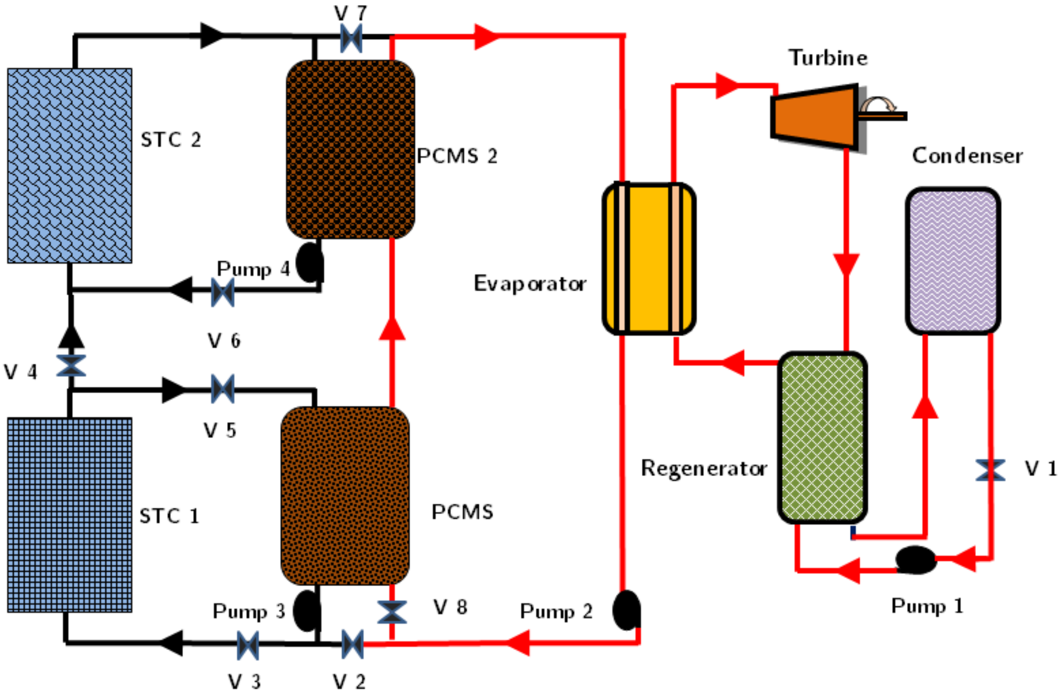


Figure 3.5 : Discharge cycle valves 1,8 are opened, pumps 1,2 are working

4 Mathematical model

4.1 Solar thermal collector system

In this study both non-concentric and concentric solar thermal collectors were considered. All major components of the irradiation counted for the study. Addition of beam, diffuse and reflected irradiation on the collector surface was used to calculate the extracting solar thermal heat. Both hourly solar irradiation on the tilted STC and its performance should have to be modeled in order to find the energy output from STCs. In order to achieve this, hourly global irradiation on a horizontal plane G is taken (Fig. 4.1) [25] and it is used to calculate hourly diffuse fraction f using Climed-2 model, which was later used to calculate diffuse solar radiation G_d on horizontal plane according to Eq (1).

$$G_d = f \cdot G \qquad (1)$$

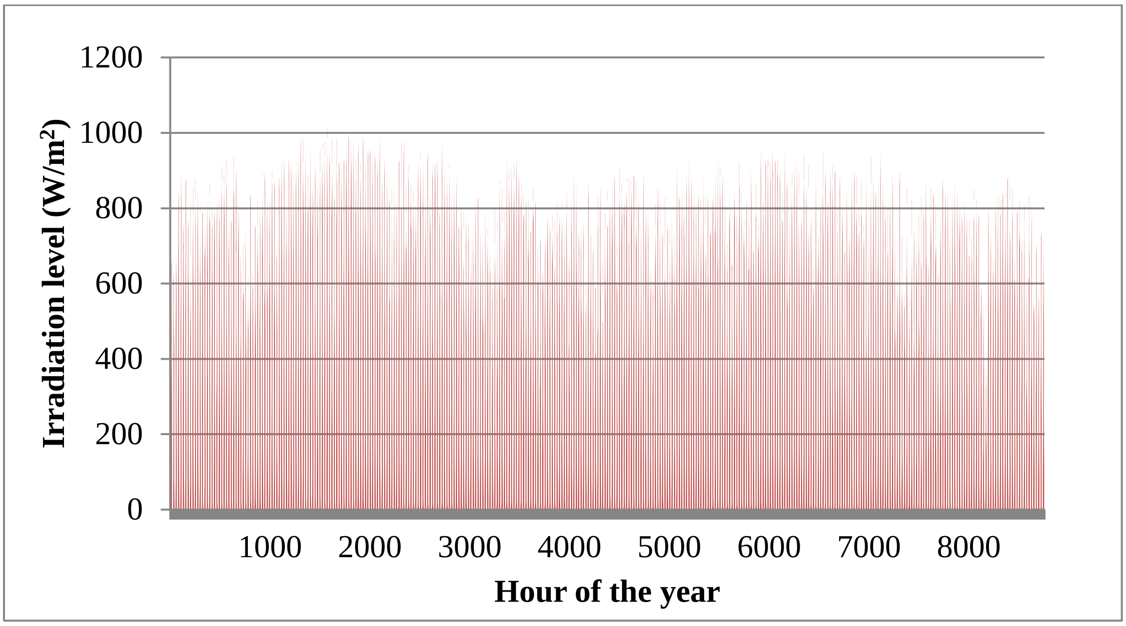


Figure 4.1 : Annual hourly solar irradiation variation [25]

Klucher model [26] was used to calculate diffuse solar radiation on tilted surface $G_{d\beta}$. Finally, beam radiation $G_{b,\beta}$ and reflected solar radiation $G_{r,\beta}$ were calculated using Eq. (2) and (3). These values were used to find total solar radiation on tilted surface G_β using Eq. (4).

$$G_{b,\beta} = (G - G_d) \cdot \frac{\cos(\theta)}{\cos(\theta_Z)} \quad (2)$$

$$G_{r,\beta} = \left(\frac{1}{2}\right) \rho \cdot G \cdot \left(1 - \cos(\beta)\right) \quad (3)$$

$$G_{\beta} = G_{d,\beta} + G_{b,\beta} + G_{r,\beta} \quad (4)$$

Where θ, θ_z and β denote angle of incidence for an arbitrarily inclined surface oriented toward the equator, zenith angle and tilt angle of STC.

Flat plate, evacuated, compound parabolic and parabolic trough type solar thermal collectors were taken into consideration in this work. The thermal efficiency of solar thermal collectors (η_c) can be calculated as a function of medium operating temperature of the STC (T_m), of the STC (T_m), ambient temperature (T_a) and solar irradiation on tilted STC (G_β) according to Eq. 5.

$$\eta_c(t) = C_o - C_1 \,.\, \frac{(T_m - T_a)}{G_\beta(t)} - C_2 \,.\, G_\beta(t). \left[\frac{(T_m - T_a)}{G_\beta(t)}\right]^2 \quad (5)$$

Table 4.1 :Efficiency parameters for different collector types

Solar thermal collector type	Curve coefficient			Ref
	C_o	C_1	C_2	
Flat plate	0.768	2.90	0.0108	[2]
Evacuated	0.612	0.54	0.0017	[12]
Compound parabolic trough	0.665	0.59	0.0019	[2]
Parabolic trough	0.700	0.20	0.0015	[12]

Finally, hourly net thermal energy outputs from solar thermal collectors are calculated using Eq. (6) where A_C denotes the solar thermal collector area.

$$Q_{col}(t) = 3600 \cdot A_C \,.\, G_\beta(t) \,.\, \eta_c(t) \quad (6)$$

The extracted thermal energy from the solar thermal collectors is continuously supplied to evaporator and excess energy stored in a latent heat thermal storage tank by passing conduction oil through the solar collectors. Conduction oil hourly mass flow through solar collectors is calculated using Eq. (7).

$$m_{col}(t) = \frac{Q_{col}(t)}{C_s \cdot (T_H - T_L)} \qquad (7)$$

Where C_s denotes specific heat capacity of conduction oil and T_H denotes the temperature of the solar collector outlet.

4.2 Thermal energy storage model

Main purpose of thermal storage is to store excess thermal energy from solar thermal collector system in order to support continuous power generation under timely varying energy load demand and solar irradiation. Phase change material is used as the thermal storage and energy is stored by melting the phase change material and releases by solidifying. It is essential to come up with dispatch strategy for both storing and releasing of energy. Dispatch strategy is based on the difference between solar energy potential and energy load demand and consists of three different states as charging cycle, combined discharge cycle and discharge cycle.

4.2.1 State 1 (Charging Cycle)

When thermal energy production is more than the requirement of energy production system starts to follow charging cycle. In this mode only valve 8 is closed and all the pumps are kept open in order to store excess thermal energy and conduction oil hourly mass flow through the evaporator in order to supply the energy load demand is calculated by using Eq. (8).

$$m_1(t) = \frac{Q_E(t)}{C_s \cdot (T_H - T_L)} \qquad (8)$$

Where $Q_E(t)$ denotes the evaporator heat requirement, and T_L denotes the conduction oil temperature at evaporator outlet. Excess conduction oil mass flow rate from the solar collectors $m_2(t)$ is evaluated using Eq. (9).

$$m_2(t) = m_{col}(t) - m_1(t) \qquad (9)$$

The amount of thermal energy stored, $Q_{supply}(t)$ in thermal storage can be determined using Eq. (9) and Eq. (10).

$$Q_{supply}(t) = m_2(t) \cdot C_s \cdot (T_H - T_L) \quad (10)$$

Subsequently available thermal energy capacity in the thermal storage available is calculated using Eq. (11) where $Q_{Losses}(t)$ denotes thermal losses.

$$Q_s(t) = Q_s(t-1) + Q_{supply}(t) - Q_{Losses}(t) \quad (11)$$

Finally melted phase change material percentage of the thermal storage M_p is computed using Eq. (12) where C_{LS} denotes latent heat of fusion and M_{PCM} denotes total mass of phase change material.

$$M_p = \frac{Q_s(t) \cdot 100}{C_{LS} \cdot M_{PCM}} \qquad (12)$$

4.2.2 State 2 (Combined Discharge Cycle)

When solar thermal energy collected from the collector is not enough to drive the cycle to provide the energy load required alone, system moves to combined discharge cycle (State 2). In this mode valves 5 and 6 are closed, pumps 1 and 2 are kept working and energy from both solar thermal collectors and thermal energy storage are used to supply the evaporator heat required. Thermal energy required from the thermal storage $Q_{require}(t)$ in this state is calculated using Eq. (13).

$$Q_{require}(t) = Q_E(t) - Q_{col}(t) \qquad (13)$$

Finally available thermal energy capacity is calculated using Eq. (14).

$$Q_s(t) = Q_s(t-1) - Q_{require}(t) - Q_{Losses}(t) \quad (14)$$

4.2.3 State 3 (Discharge Cycle)

Discharge cycle is used to when solar irradiation is not available. Stored thermal energy in phase change material is used in this state to provide the evaporator heat required. Only valves 1 and 8 are kept open and pumps 1 and 2 are kept working in this mode allowing conduction oil to flow through the thermal storage and evaporator. Thermal energy required from the thermal storage is calculated using Eq. 15 and available thermal energy capacity in thermal storage is calculated using Eq. 14.

$$Q_{require}(t) = Q_E(t) \qquad (15)$$

4.3 Mathematical Model for Organic Rankine Cycle

This section provides a detailed explanation about the dynamic mathematical model formulated in order to simulate the ORC. Four basic components of the cycle i.e. pump, condenser, turbine and evaporator is modeled in order to come up with work and heat transfer from these devices.

When considering the work transfer of the cycle, work input at pump and work output from the turbine has being coupled together. Therefore, work input at the pump is initially calculated using Eq. 16. In this equation $\dot{m}_{cycle}(t)$ denotes the mass flow rate of working fluid, P_E and P_C denote evaporator and condenser pressures, ρ_{wF} denotes density of working fluid and $\eta_p(t)$ denotes pump working efficiency.

$$\dot{w}_P(t) = \frac{\dot{m}_{cycle}(t)\cdot(p_E - P_C)}{\rho_{wF}\cdot\eta_p(t)} \quad (16)$$

The efficiency of pump is a function of cycle mass flow rate and calculated by using Eq. 17 [27]. Where $\dot{m}_{cycle,ref}$ denoted reference mass flow rate at peak demand and $\eta_{p,ref}$ denoted reference efficiency of the pump at peak demand.

$$\eta_p(t) = 2\cdot\ \eta_{p,ref}\cdot\left[\frac{\dot{m}_{cycle}}{\dot{m}_{cycle,ref}}\right] - \ \eta_{p,ref}\cdot\left[\frac{\dot{m}_{cycle}}{\dot{m}_{cycle,ref}}\right]^2 \quad (17)$$

Required turbine work is the addition of mechanical work required for electricity generation, pump work and condenser cooling system power. This is calculated according to Eq. 18 assuming constant generator efficiency (η_{gen}) and very small condenser cooling system power.

$$\dot{w}_T(t) = \frac{\dot{w}_{load}(t)}{\eta_{gen}} + \dot{w}_P(t) \quad (18)$$

For the proposed design screw type expander has been selected. The isentropic efficiency of the screw expander is a function of outlet volume flow rate and volume ratio. For this investigation hourly variation of load has been considered. Therefore expander efficiency varies with hourly changing volume flow rate. Referring below set of equations expander efficiency can be calculated [19].

$$\eta_p(t) = \mathrm{C}\,[0.9403305\ +\ 0.0293295\ln(V_{out}) - 0.0266298\,V_r\,] \quad (19)$$

Where C is a constant added to volume ratio greater than 7 and Vr is the expander fluid flow volume ratio between exhaust and inlet.

$C = 1 - 0.264 \ln (V_r/7)$ for $V_r > 7$

$V_r = V_{out}/V_{in}$

Turbine outlet working fluid enthalpy (h_2) is calculated using Eq. 20 where h_1 denotes turbine inlet working fluid enthalpy and h_{2S} denotes working fluid isentropic enthalpy at turbine outlet

$$h_2 = h_1 - \eta_t \cdot (h_1 - h_{2s}) \tag{20}$$

Finally evaporator hourly energy requirement $Q_E(t)$ is calculated by using Eq. 21 where h_4 denotes the working fluid enthalpy at pump outlet

$$Q_E(t) = \dot{m}_{cycle}(t) \cdot (h_1 - h_4) \tag{21}$$

5 Simulation method

ORC simulation process is developed by using a simulation program written in Matlab R2016a language Microsoft Windows® 07 environment. In this investigation, the system conditions are evaluated on hourly basis considering timely varying solar irradiation and required load demand. Load required is highly sensitive to the application. In this work, it was assumed that the load varies throughout the year according to summer - weekly load IEEE reliability test system, which is scaled to 37.5 kW (Fig. 5.1). Steady state conditions of system are assumed during each hour.

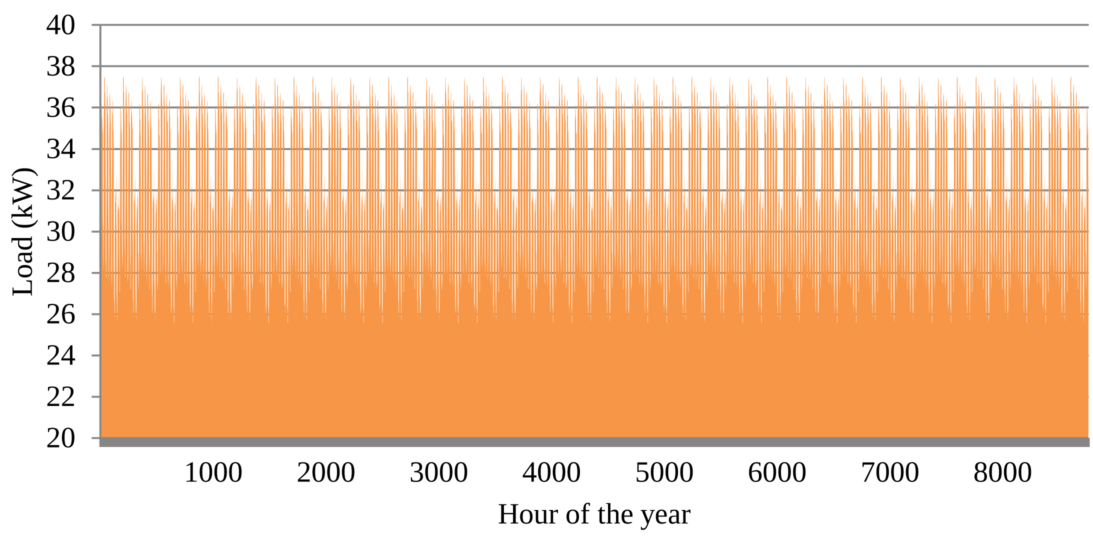


Figure 5.1 : Hourly variation of load though out the year

The simulation is conducted by evaluating thermal energy supply from solar collectors and thermal energy required for evaporator every hour. Then heat capacity of latent heat storage is calculated according to working mode of system discussed in Section 2.2. The latent heat storage capacity is considered as the key condition for continuous running for the system. The condition of the latent heat thermal storage is availability of the usable thermal energy inside the storage (availability of the molten phase change material) when irradiation level weak or not available. The performance evaluation process was done by in different ways. The proposed two systems were evaluated by changing parameters of collector area, thermal energy storage and ORC.

For this study four types of solar collectors were considered. However, for each system performance evaluation only two cascaded solar collectors were considered. Altogether there are four collector combinations proposed in table 3.1 were taken to evaluate. The combination of different concentric and non-concentric collectors may enable the extract heat at optimum collector temperature while reducing overall collector area. Actually, the low temperature collectors act as a preheater for high temperature collector. The simulation process was done by changing total collector area and collector ratio for each configuration. However, identification optimum collector configuration is based on both energy output and cost.

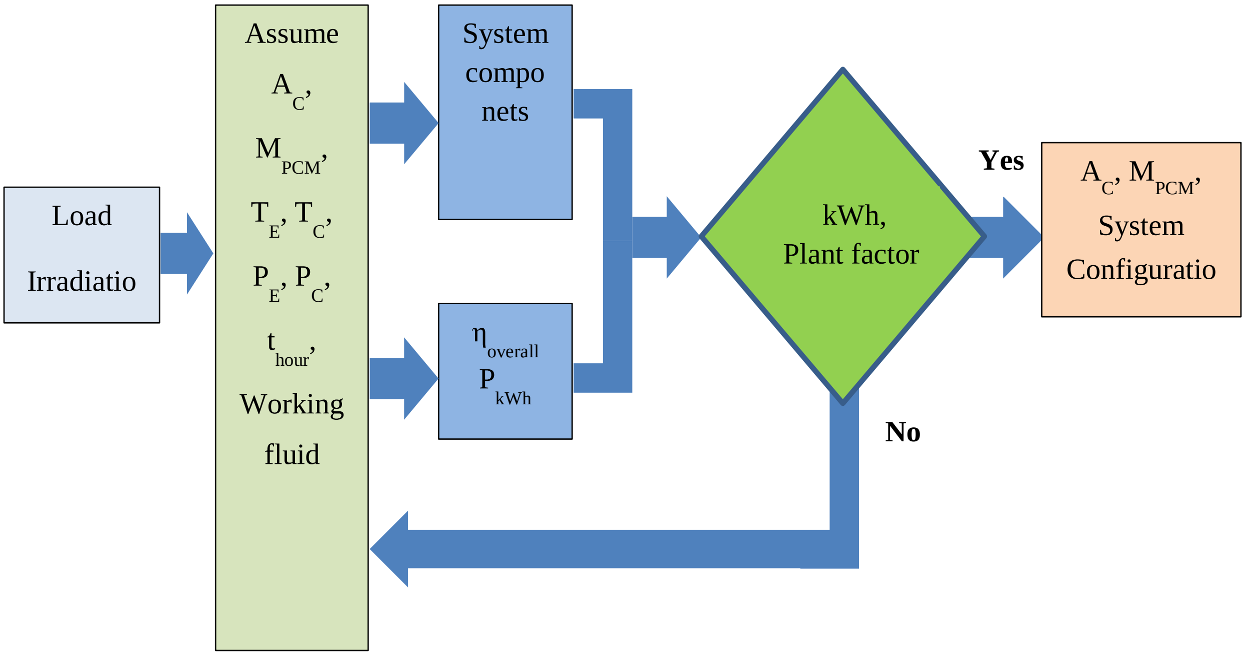


Figure 5.2 : Simulation method

The thermal energy storage system was taken to simulation process considering energy availability of the storage. During the charging mode storage energy level is increased and stores the thermal energy. Then during combined discharge and discharge modes storage energy level decreased according to required heat load. The charging and discharging rates of the storage system are taken in to this study. Both charging and discharging rates depend on storage energy level. For this study, all charging and discharging processes are kept within those heat transfer rates.

The ORC system simulation was conducted by considering energy flow of each component. Specially, the screw expander efficiency variation with system load is important to consider in this timely varying dynamic model. Therefore both expander and pump efficiency variations were calculated for each working conditions. The selected plate type heat exchangers effectiveness was taken to heat transfer analysis process. The pinch point temperature difference kept between 8-15 ^{o}C. The performance analysis process was done for five selected working fluids. The simulation parameters for each working fluid are shown in below tables 5.1

Table 5.1 :System -1 expander inlet outlet conditions

Working fluids	Expander inlet		Expander outlet	
	Pressure bar	Temperature ^{o}C	Pressure bar	Temperature ^{o}C
Neo-pentane	22	138	2.00	37
Pentane	13	139	1.00	36
R11	17	138	1.75	41
R365mfc	15	140	1.00	42
R141b	15	140	1.40	42

Table 5.2 :System -2 expander inlet outlet conditions

Working fluid	Expander inlet		Expander outlet	
	Pressure (bar)	Temperature ^{o}C	Pressure (bar)	Temperature ^{o}C
Pentane	21	167	1.00	36
R11	27	166	1.75	41
R365mfc	23	165	1.00	42
R141b	23	165	1.40	42

6 RESULTS AND DISCUSSION

6.1 System performance with area variation

The system performance analysis with solar thermal collector area size is the first key element of this investigation. The figure 6.1 and 6.2 illustrate the variation of power per unit area for different working fluids. The system-1 shows maximum power output values in area range between 1000 -1300 m^2 for many investigated working fluids. However, for the working fluid pentane, it takes 1600 m^2 to achieve maximum power output. The system - 2 shows maximum power output values in area range between 200 - 350 m^2 for the considered working fluids in this investigation. However, working fluids R11 and R365mfc have a slight variation of power per unit collector area from the beginning to maximum, and then gradually decrease with larger collector areas.

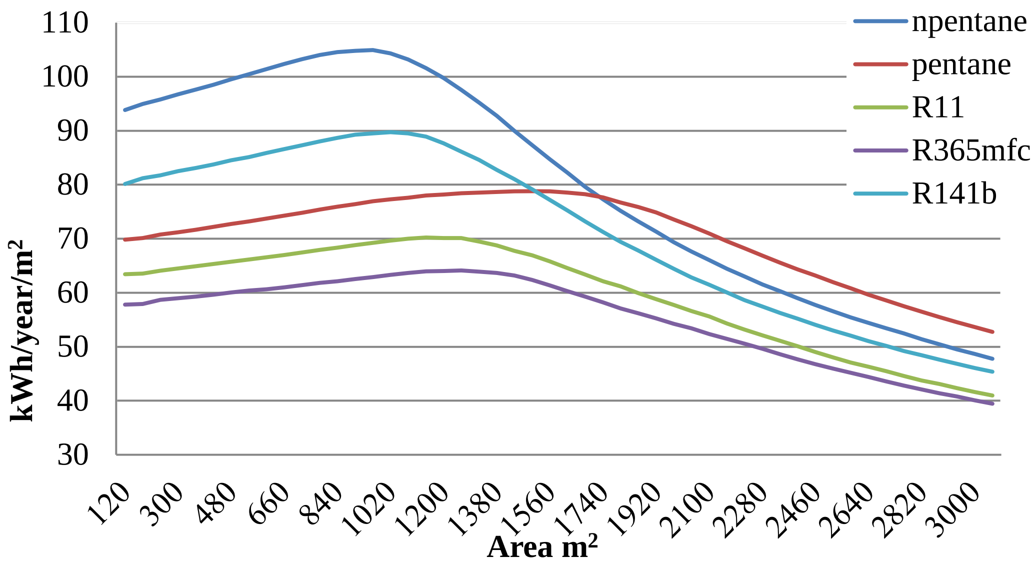


Figure 6.1 : System -1 annual power per unit collector area

The power output per area between system -1 and system -2 show nearly threefold difference for each fluid. On the other hand, system -1 takes nearly four times area to obtain maximum value compare with system -1. However, these optimal values are not the best operating points with consideration of plant factor and cumulative power. Therefore, the optimum operating points should be identified considering all those factors including cost elements. In summary from these two graphs, only considering power output per unit area n-pentane and pentane can be concluded as optimal working fluids.

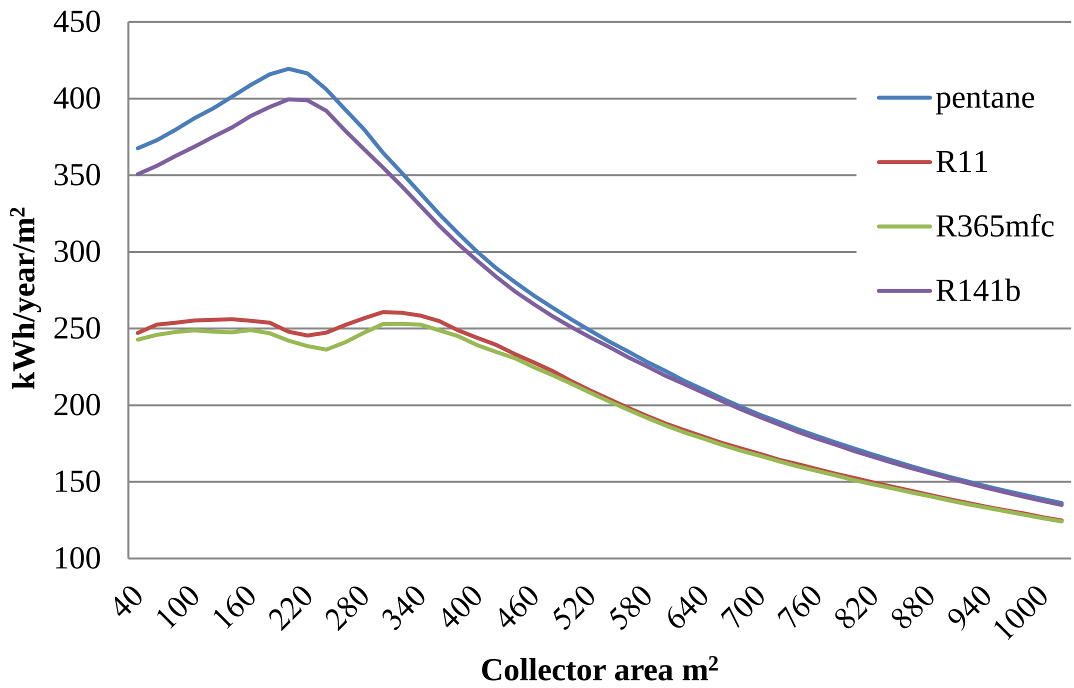


Figure 6.2 : System -2 annual power per unit collector area

6.2 System cumulative power variation

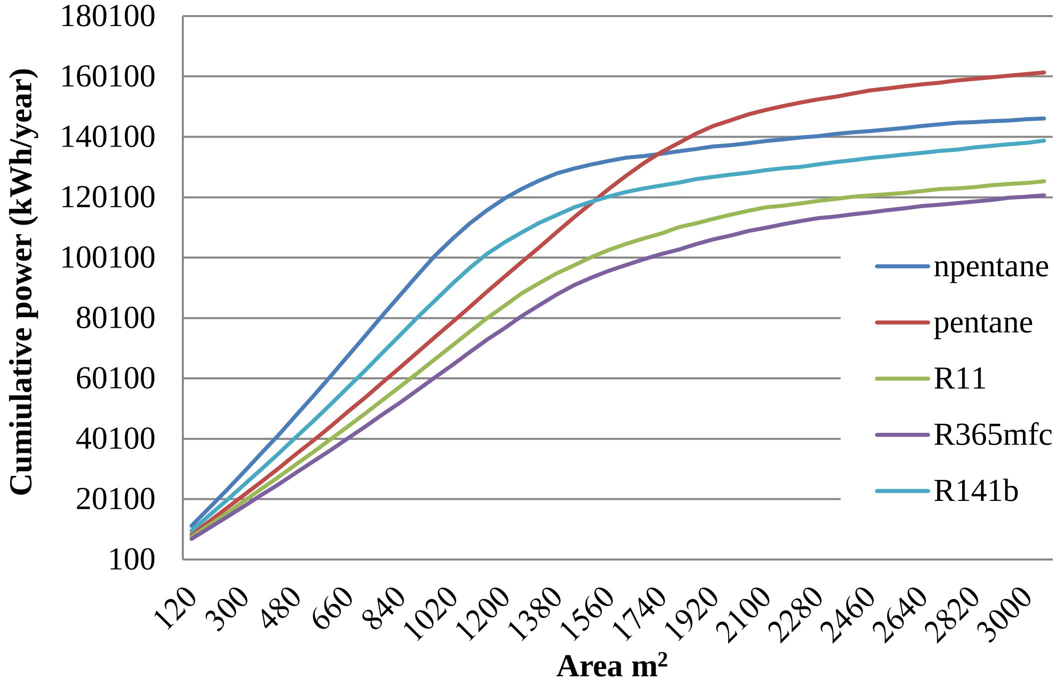


Figure 6.3 : System -1 annual cumulative power variation

The cumulative power variation with collector area shows that all fluids obtain nearly saturated values with larger collector area. For system -1 it takes 2000 m^2 and system -2 takes 600 m^2 to obtain saturated values.

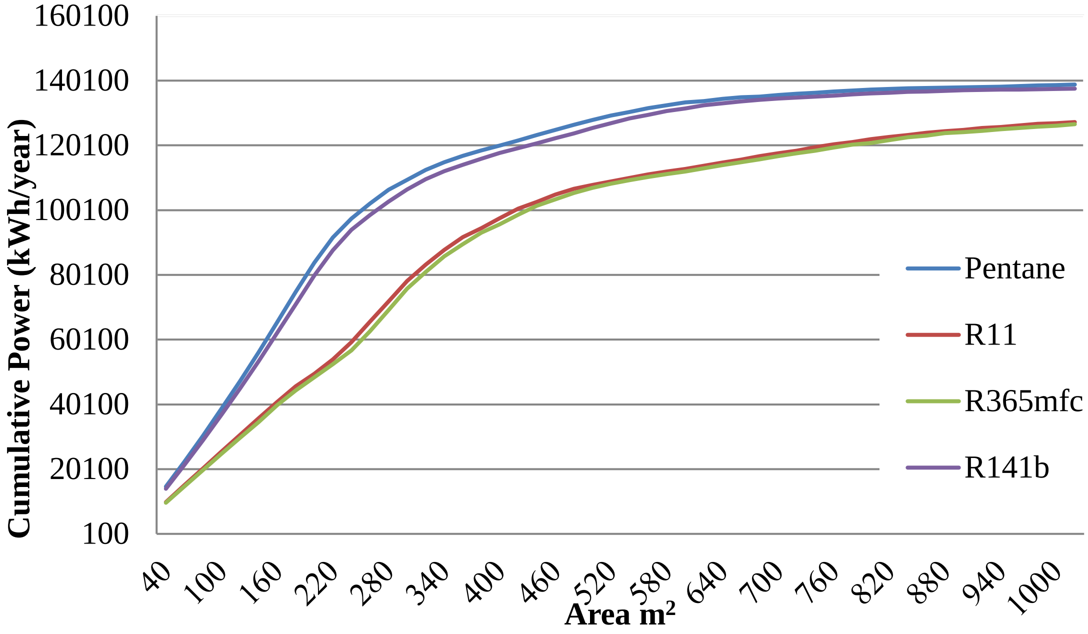


Figure 6.4 : System -2 annual cumulative power variation

6.3 Plant factor variation with area

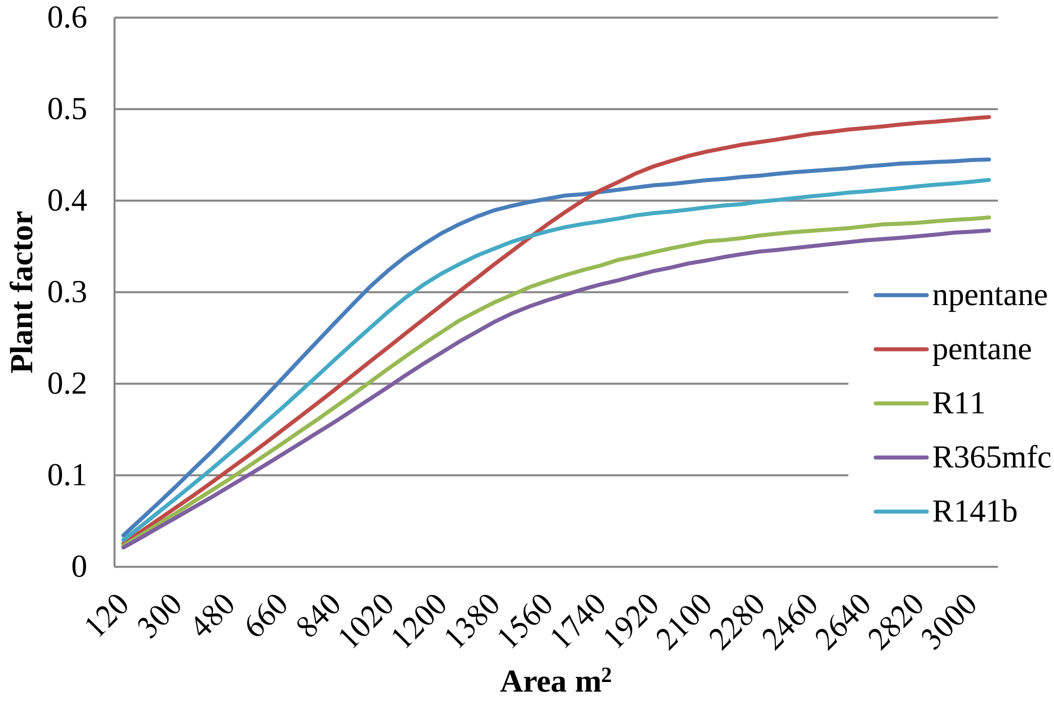


Figure 6.5 : System -1 plant factor variation with collector area

The plant factor variation with collector area shows that all investigated fluids having nearly constant plant factor values at higher collector areas. For the system -1 it takes nearly 2000 m^2 to obtain nearly constant plant factor value and system – 2 takes 750 m^2.

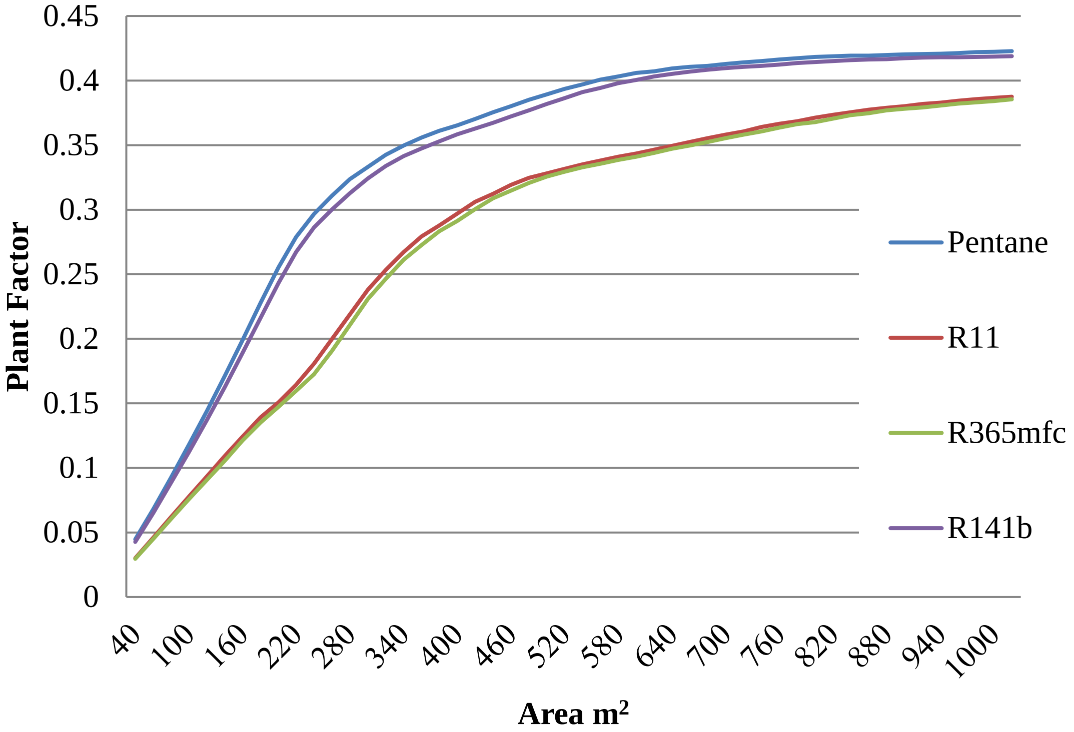


Figure 6.6 : System -2 plant factor variation with collector area

6.4 Plant factor variation with storage capacity

The plant factor variations with storage capacity shown in below graphs illustrate that plant factor values become saturate with larger capacity storages. From the graphs it shows that the best investigated working fluids in system -1 gives maximum plant factor near to 0.4 and in system -2 it remains near 0.55. However, the system -1 has slight variations of plant factor for all considered storage capacities. But system -2 shows slightly higher plant factor variation for evaluated storage capacities. The optimal working fluids for these two systems are pentane and R141b. However, the selection of best working storage capacity should be identified considering power output, plant factor and economical factors.

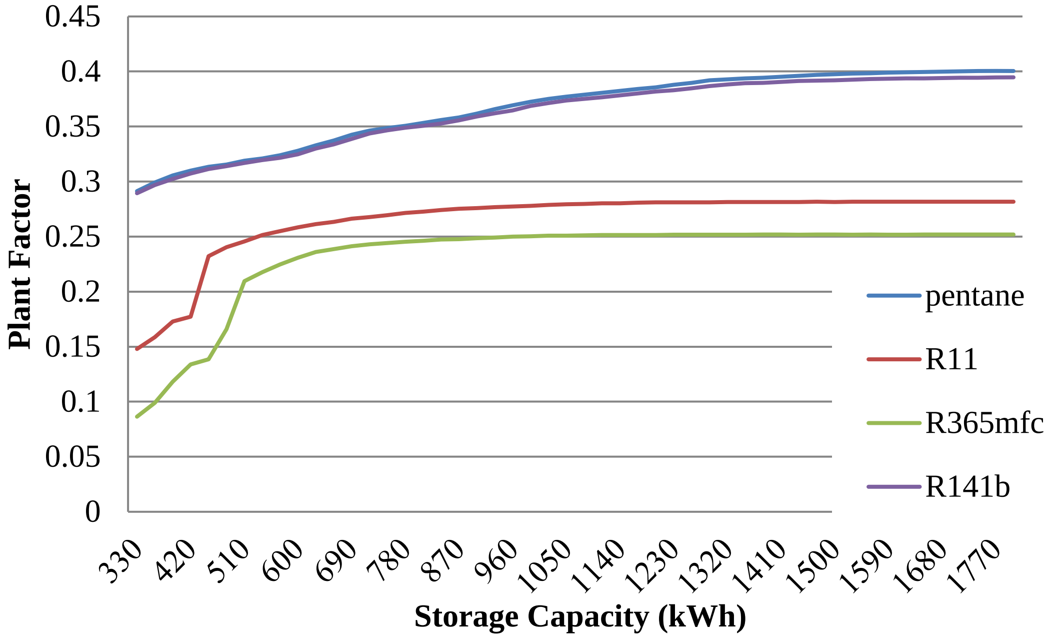


Figure 6.7 : Plant factor variation with storage capacity System-1

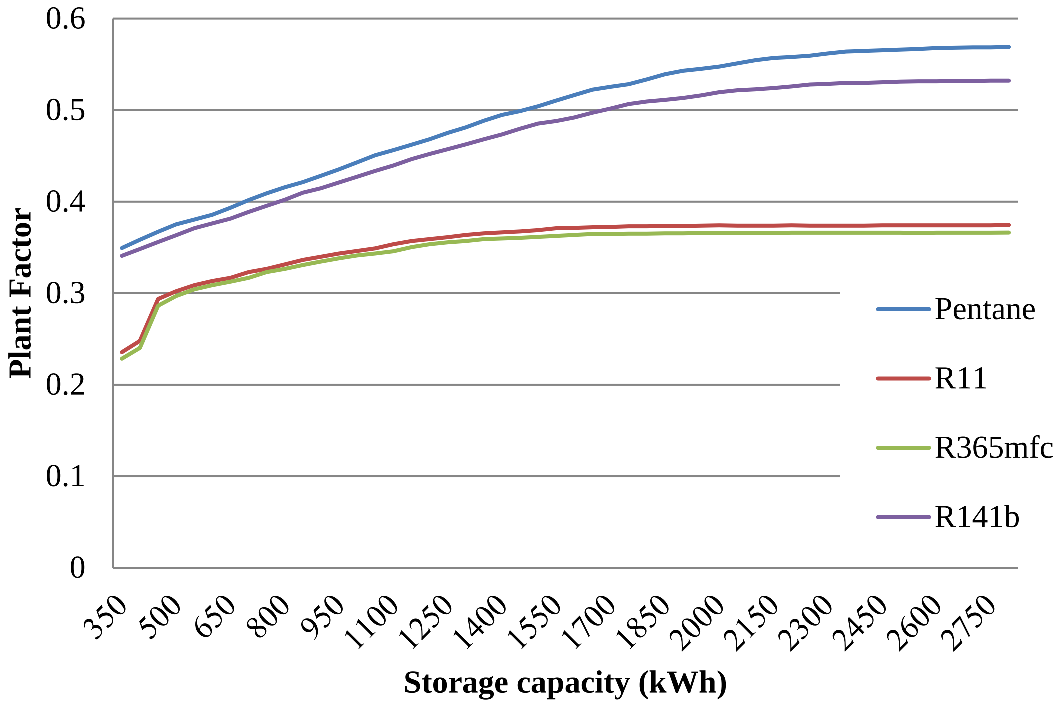


Figure 6.8 : Plant factor variation with storage capacity System-2

7 CONCLUSION

7.1 Conclusions

In this investigation performance of solar assisted organic Rankine cycle with a latent heat thermal energy storage has been studied. The mathematical model developed to evaluate system performances was simulated to obtain results considering both timely varying load demand and solar irradiation level. The main purpose of this study is to identify optimum system configuration, working conditions and working fluids. The results shown in chapter 6 describe the variation of power output with collector area and storage size.

The results from this study clearly show that, there is a significant performance difference between system -1 and system-2. For the performances with collector area and storage size system -2 shows nearly threefold high performance. However, system -2 involve with concentric solar collector system to obtain its high working temperature. Therefore it is difficult to predict how optimum the system without considering both thermodynamic and economic factors.

From this investigation, it has identified the working fluid performances for each configuration. From the results it shows that, pentane and R141b are having better working performances. However, there are many other organic fluids are yet to investigate with this model to identify the optimum working fluids for the proposed system.

In summary, from this investigation thermodynamic performances of proposed system have been investigated. The optimal working regions for different configurations have been identified. However, to identify the best optimum working parameters, it is required to study this system considering both thermodynamic and economic performances. The next stage of this study is develop and integrate a cost model for currently existing energy model to identify optimum working parameters.

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