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Research Article

ENERGY, ANERGY AND EXERGY ANALYSIS OF ORGANIC RANKINE CYCLE FOR A RESIDENTIAL APARTMENT USING SOLAR ENERGY BASED ON ASPEN PLUS

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A B S T R A C T

The utilization of solar energy is on a rise due to serious global energy crisis.This paper presents the simulation generated data in energy and exergy analysis to determine the performances for an organic Rankine cycle (ORC). The system was simulated to supply both electrical and thermal energy to an existing residential apartment. The simulation of the operating ORC was performed using Therminol 55 as the heat transmitting fluid within the oil tank, pump and evaporator and using R-245fa as the working fluid within the cycle. The highest power output was obtained when the expander inlet pressure was 2MPa and the solar source temperature was 163.5°C, delivering an output power of 33.34kW with an achievement of overall process efficiency of 14.55%. The proposed system is intended to be potentially attractive for real estate investors in Thika town with good solar irradiation and without (or with very high cost) access to the general public electricity supply. Detailed analysis of energy and exergy in the organic Rankine cycle has been done in order to determine the best rational use of energy and suggestions on exergy saving are provided based on simulated results.

INTRODUCTION

Global electricity requirement is on the rise and future electricity demand is expected to increase up to 33% by 2020 and 84% by 2035[1]. Following the trend which has been observed in the last 10 years, the cost of electricity is expected to increase gradually. In addition to the environmental challenges, such as global warming, ozone layer depletion and increase in pollution levels, there is a great concern and a heavy influence on worldwide energy policy. Therefore, in order to reduce the environmental impacts and fossils fuels usage, it is necessary to emphasize the use of emerging technology and well-known renewable energy approaches. The organic Rankine cycle (ORC) is a Clausius Rankine cycle which utilizes organic fluid as a working fluid. ORC has the great potential of converting low temperature heat (<300°C) to electricity [1-2].

The ORCs having heat input and power output with the potential of reversing low grade heat into high grade electricity possess the following merits compared to steam based water Rankine cycle: simple to operate on low temperature heat source; maintenance and operation costs are low; compact; simple start and stop procedure; some models do not require water consumption; quiet during operation; requires smaller expanders with higher rotational speeds; the turbine in organic Rankine cycle only requires single stage expanders and the components are commonly available in the market today. The first and second laws of thermodynamics are applied in organic Rankine cycle [2-3]. The thermoeconomics of a Rankine system is strictly linked to the thermodynamic properties of the working fluid used in the system. Many studies of solar ORC power plant exist in the literature, the working fluid selection is the main point in most of the studies and many factors have influenced on it, in medium temperature levels of 150 ^{0}C to 200 ^{0}C. R254ca, R254fa, isobutane and isopentane are usually investigated. Poor selection of a working fluid could result in low efficiency and increase in cost of the ORC plant. Most refrigerants which have lower boiling point than water such as pentane, butane, and hexane can be used in ORC as the working fluids. The working fluids selected are heated with low temperature heat from the collector or recovered waste heat and have properties that differ from those of water in different aspects [3]. Examples of countries where organic Rankine cycles have been installed include: Canada, USA, Germany and Italy though the applications have also been reported in Morocco, Romania, India, Swaziland, Austria, Russia, Finland and Belgium [3].

The following technologies are the main applications used towards harvesting solar energy:

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- Solar photovoltaic (PV) power: A series of photovoltaic panels are used to convert sunlight into electricity through the photo- voltaic effect.
- Solar thermal energy: To collect heat from the sun, solar collector is used. The captured heat will be used directly or stored for providing warm water, space heating and drying purposes depending on the application.
- Solar photovoltaic thermal energy: This is a mix of solar thermal and photovoltaic (PV) components integrated into one system and able to generate electricity and heat simultaneously.

Low efficiency is the main crucial factor that affects the popularity of PV module. Until today, PV modules available commercially in the market from various manufacturers have efficiency ranging from 6% to 16%, however the claimed efficiency is at a temperature of 25°C [4]. In reality, especially for areas with hot weather, their ambient temperature during a sunny day would be more than 35°C, but the increase in PV temperature leads to the decline in module efficiency [4]. In the near future, the use of solar energy is expected to be crucial and to increase significantly hence there is dire need to improve the performance of thermal power plants integrated with solar thermal energy [5]. Modern steam power plants operating at a steam pressure over 200 bar and at a temperature above 600°C. As a rule, they produce an electric output power of several units of MWe. Consequently, the suitable sources of heat for such particular power plants are central receiver system and parabolic trough concentrators but dish concentrators can also be used in driving more compact thermal engine with less power output. Derscha *et al.* [6] carried out a comprehensive research to compare the performance of integrated solar combined cycle systems (ISCCs) with a solar electric generating system (SEGS) and found that integrated solar combined cycle systems are more superior to solar electric generating systems. Analyzing exergy in ORCs system is an approach that explains and calculates the different parts of exergy destruction rate (endogenous/exogenous and unavoidable/avoidable) in each component of an energy system [7, 8, 9]. In other words, in the energy system, energy is always conserved, and exergy shows the quality of energy. Analyzing exergy can be done for both simple and complex organic Rankine cycles by considering each and every component separately, which is a crucial factor in efficiency, improvements and performance evaluation in the organic Rankine cycle with the main aim of overall performance optimization and identification of exergy of destruction sites [10-11]. Christos Trivanidis *et al.* conducted a research using an organic Rankine cycle coupled with PTC, analyzing it parametrically in order to optimize it financially and energetically using various working fluids [12].

Studies were performed on a hybrid solar gas-turbine plant and identification of the sources of energy losses in their systems [13, 14, 15]. Karellas and Braimakis [16] conducted thermodynamic and economic analyses of a solar biomass system driven tri-generation using vapor compression cycles and ORCs. They recorded thermal efficiency of 5.5% and the exergy efficiency of 7.6% in the ORC system. In developing countries, the use of solar energy for electricity generation for small and medium scale using ORC system with liquids as the working fluids is of great importance and is expected to increase in most countries [17, 18]. Currently, commercially there exists in USA one ORC plant of a size 1MW which was supplied by ORMAT Company; it uses n-pentane as the working fluid with a conversion ratio of solar to electric equal to 0.084. In 2009 a prototype solar ORC was constructed in Armenia and Spain within the frame of POWERSOL project, and SES36 was used as the working fluid with an output of 5 kWe and attained an ORC efficiency of 14% [18-20]. Simulation using a simulation software Aspen plus v8.0 is helpful in designing a solar PVT with optimum and economic conditions to achieve high process efficiency. A comprehensive analysis done on the components conducted taking into account anergy, exergy, or energy analysis of the solar PV thermal plant using ORC in a residential apartment. The aim of this paper is to present a detailed exergy, energy and entropy analysis of ORC for a solar PV thermal plant in a residential apartment with 10 floors and 30 units per floor located in Thika, Nairobi, Kenya, using the geographical conditions of the area and simulated using Aspen plus v8.0. Aspen plus is a computer aided software which uses the under lying physical relationships such as material and energy balances, thermodynamic equilibrium; rate equations etc. to predict process performances e.g., stream properties, operating conditions and equipment sizes etc.

System Description

Modelling and simulation of the process

In this study the system is modelled using the process simulator Aspen plus. Solar ORC system is a promising technology to reduce the investments cost at small scale: the system can work at low temperatures and the total installed power can be scaled down to kW levels. There are three main subsystems in the system simulated ORC, solar plants, and cold-hot water section. The solar collector has the optical part with the energy conversion path in a solar thermal energy system, from solar radiation to thermal. Solar collectors perform the function of adding thermal energy to a liquid medium flowing though the collectors for usage or storage. Solar irradiation is incident to a photovoltaic type solar collector. The ORC is integrated with the solar collector field through the oil tank which contains the heat transmitting fluid. The heat transmitting fluid is heated and pumped from pump P-2 to the oil tank and to the evaporator, and then goes back to the pump. In the evaporator there is heat exchange between the heat transmitting fluid and the working fluid from the pump P-1, the high pressure stream produced in the evaporator is used to run the turbine/expander which is then connected to generator producing electricity. The working principal of the system is presented in Figure 1.

The outlet stream of turbine is cooled in the condenser and converts it to saturated liquid [21, 22]. The solar heated fluid (Therminol 55) from the tank is passed into the plate type heat exchanger (evaporator) and after being cooled down is pumped back to the tank through pump (P-2). The organic working fluid 1,1,1,3,3-pentafluroropropane (R-245fa), vaporizes when passed into the evaporator, which drives the magnetically coupled scroll expander to produce mechanical power.The expander/turbine is connected to generator. After expansion in the process, R-245fa vapor is condensed back into the liquid state, that is, the superheated R-245fa at the outlet of the turbine is condensed to saturated liquid in the condenser. The liquid R-245fa is again pressurized by the pump, completing the ORC. Lastly, R-245fa in its liquid form runs into the refrigerant tank to complete its cycle. Mechanical power is produced by the expander in the ORC unit and then coupled to an electrical generator for electricity generation. In some instances, in order to reuse the heat after the expender/turbine, a recuperator may be included in the ORC system to preheat the working fluid (not in this paper scope). A recuperator is essential in improving thermal efficiency of system but at an extra cost, hence an increased power output can be achieved for a decreased heat input in the ORC system. The relatively low efficiency realized with solar ORC is mainly due to low temperature range obtained from the photovoltaic collectors [4].

Solar Resource

The present study was carried out using dataset for an average monthly daily information obtained from Kenya meteorological department, Dagoretti station, the meteorological dataset includes Diffuse Horizontal Irradiance (DHI), Global Horizontal Irradiance (GHI), Direct Normal Irradiation (DNI), solar azimuth and elevation, temperature, relative humidity and wind velocity. The simulation was conducted with the aim of creating a platform for real estate investors in residential apartments in Thika located at (0^0, 37^0, 41'E) with an average monthly daily solar insolation ranges from 4.4 kWh/m^2/day to 6.37 kWh/m^2/day and the average monthly temperature ranging from 19.43 ^oC to 21.76 ^0C as shown in the Figure 2 and 3 respectively. The average sunlight hours in Thika per year is 2600 (of possible 4383 hours) with an average of 8 hours of sunlight per day. The data provided by the meteorological department was used as input to the model, the analysis was done to provide a baseline data for the investors in the area using Aspen plus v8.0. The analysis shows that Thika has reliable energy resources with little variability and the transparency of the sky is also encouraging as it shows that there is little obstruction to the solar radiation.

PV system in ORC

PV technology with a secondary thermal system that operates on a waste heat is used in this simulation, commonly referred to as PVT system. Thermodynamics analysis using the software generated data has been carried out on the system between the temperature range 80 ^0C to 180 ^0C proving the applicability of the system in Thika and its effectiveness, using therminol oil as the heat transmitting fluid (HTF) in the oil tank (o-tank) direct energy storage TES system and an ORC power plant. HTF storage tank (o-tank) is installed in order to attenuate the fluctuation of solar irradiation during the day and maintain stable operation of ORC engine (see Figure 1)

The HTF temperature required by ORC system in consideration, makes therminol oil the most suitable option for temperature levels 80 ^0C - 180 ^0C. The PV section includes the photovoltaic arrays comprising of several sub arrays, the total modules area is better calculated in the following section.

The correlations expressing PV cell temperature (T_c) as a function of weather variables in Thika such as ambient temperature (T_{amb}), area wind speed (V_w), solar radiation, material and system dependent properties such as glazing cover, transmittance etc. The effect of temperature on the electrical efficiency of a PV cell was calculated using the equation

$$P_{max} = I_{max}V_{max} = (FF)I_{sc}V_{OC} \quad (1)$$

Both open circuit voltage (V_{OC}) and fill factor (FF) decreases substantially with temperature because the thermally excited electrons start to dominate the electrical properties of the semiconductor, while the short circuit current increases slightly, leading to a linear equation in the form:

$$\eta_c = \eta_{T\,ref}[1 - \beta_{ref}(T_C - T_{ref}) + \gamma log_{10} I \quad (2)$$

$$\eta_c = \eta_{T\,ref}[1 - \beta_{ref}(T_C - 25) + \gamma log_{10} 1000 \quad (3)$$

Given that $\beta_{ref} = 0.0045K^{-1}, \gamma = 0.12$ for crystalline silicon module in the PVT system and $I = 1000W/m^2$.

β_{ref} depends on the PV material and reference temperature, it's given by;

$$\beta_{ref} = \frac{1}{T_x - T_{ref}} \quad (4)$$

Where T_x is the high temperature at which PV module electrical efficiency drops to Zero, for crystalline silicon solar cell $T_x = 270^0C$

Based on the average monthly daily information for Thika town, the monthly average efficiency can be determined from the following equation;

$$\bar{\eta} = \eta_{T\,ref}[1 - \beta_{ref}(\overline{T_c} - \overline{T_{ref}}) - \beta_{ref}\frac{\overline{(\tau\alpha)}}{\eta U_L}\overline{VH_T}] \quad (5)$$

The over bar denotes the monthly average quantities, n is the number of hours per day, U_L-the overall thermal loss coefficient, H_T is the monthly average insolation on the plane of the array, V is the dimensionless function of such quantities, as the monthly average clearness index and the ratio of monthly total radiation on the array to the horizontal surface (see Figure 2 and 3).

Solar thermal cycle

Diffuse radiation or scattered radiation and direct radiation or beam radiation which is the radiation that arrives on the ground without being scattered by the clouds are the two main components of solar radiation. Global radiation is the total radiation received which is the sum of scattered radiation and direct radiation. There are two main types of (water/air) PV/T solar collectors available in the market today namely: non-concentrated solar collectors and concentrated solar collectors. The solar energy is collected using photovoltaic solar collector operating at a temperature ranging from 80°C - 180°C. To calculate the absorber surface temperature for the collector, T_{ab}, the following formula was applied [27].

$$T_{ab}^4 = (1 - \eta)\frac{A_{ap}s}{A_{ab}\sigma}, \quad (6)$$

Where A_{ap}, s, σ, A_{ab}, η are aperture area, solar flux, Stefan Boltzmann constant, absorber area and collector efficiency respectively.

The photovoltaic collector efficiency can be calculated with the following relation:

$$\eta_{col} = \frac{ixv}{I_{t,a}A_{ap}} = \left(\frac{\eta_G}{\eta_{std}}\right)\left(\frac{\eta_{G,T}}{\eta_G}\right)\left(\frac{\eta_{G,E(\gamma)}}{\eta_{std}}\right), \quad (7)$$

where i is the electrical current through the cell; v represents the voltage across the cell; $I_{t,a}$stands for the global irradiance of the solar aperture; and $\frac{\eta_G}{\eta_{std}}$, $\frac{\eta_{G,T}}{\eta_G}$, $\frac{\eta_{G,E(\gamma)}}{\eta_{std}}$ are the ratios of efficiency of the panel at specified irradiance level to efficiency at standard irradiance, efficiency of the panel at specified cell temperature to its efficiency at specified irradiance level, and efficiency of panel at specified air mass to its efficiency at specified irradiance level respectively.

The solar flux was taken to be 1000 W/m^2 [26]. Diffuse radiation is calculated using correlations from Orgill and Hollands [28];

$$\frac{I_d}{I} = \begin{Bmatrix} 1.0 + 0.24k_T \\ 1.557 - 1.84k_T \\ 0.177k_T \end{Bmatrix} \begin{matrix} k_T < 0.35 \\ 0.35 < k_T < 0.75, \\ k_T > 0.75 \end{matrix} \tag{8}$$

where clear sky index $k_T = \frac{I}{I_0}$ where I and I_0 are measured global radiation and the calculated global radiation which depends on latitude, solar time for the Thika area and day of the year. Francis Oloo *et al.* [31] reported annual mean radiation of 6.5kWh/m^2 in Thika town (data available at http://swera.unep.net) and a daily defuse radiation in central Kenya is 5.0MJ/m^2/day with a clearness index of the range 0.4 and 0.7. To obtain the direct radiation, the diffuse radiation was subtracted from value for the global radiation and the required power output and hot water supply was achieved from total panel area of 950m^2, the reflectivity of the aperture surface was taken as 0.96.

Working Fluid Selection

Substances which are most commonly considered as the working fluids of the low-temperature solar ORC are shown in Table 1.

Table 1 Thermophysical properties of the selected pure working fluids [1]

Fluid	M (g/mol)	T_{crit} (°C)	P_{crit} (bar)	$T_{boiling}$ (°C)
Cyclohexane	84.2	280.5	40.8	80.3
Hexane	86.2	234.7	30.3	68.3
Isobutane	58.1	134.7	36.3	-12.1
Isohexane	86.2	224.6	30.4	59.8
Isopentane	72.1	187.2	33.8	27.4
Pentane	72.1	196.6	33.7	35.7
R245fa	134	154	36.5	14.8
R365mfc	148.1	186.9	32.7	39.8
Ammonia	17.0	405.8	112.8	-33.3

The following condition is considered in selecting R -245fa as the suitable working fluid, the critical temperature of the working fluid and the critical pressure was examined. Table 1, shows various properties for the various working fluids as compared to R-245fa. The thermodynamic properties like enthalpy, entropy, flow rates at various nodes have been generated by the model which are used in the calculation of performance parameters like components effectiveness/ efficiency.

Apart from the organic substances, NH_3 has also been considered mainly due to its good characteristics like fluid flow properties. These working fluids have been analyzed in some low-temperature solar ORC investigations in various research papers.

However, compared to other organic fluids, ammonia has been found to be having saturation pressure which is considerably higher than many other organic fluids and its high toxicity. Most fluids shown in Table 1 are ozone friendly substances with few of them having low global warming potential.Taking ambient temperature and pressure values of 25°C and 101.325 kPa respectively [2, 5, 29, 30].

In this simulation, Therminol 55 was used as HTF and 1,1,1,3,3-pentafluroropropane (R254fa) was used as the working fluid in the ORC system. The 1,1,1,3,3-pentafluoropropane (R-245fa) was offering the best overall efficiency as the working fluid despite the solar plant exhibiting lowest efficiency. An isomer of 1,1,1,3,3-pentafluroropropane (R-245fa), is used by UTC technologies in their commercialized ORC system [33] taking into consideration the global environmental impact, toxicity and flammability of the working fluid.

The choice of 1,1,1,3,3-pentafluroropropane (R-245fa) as the working fluid was based on the thermodynamic investigation, considering all the assumptions in Table 2, the operational condition of R245fa was determined as follows:

Table 2 Operation assumptions

Parameters	value	units
T_{cond}	60	°C
ΔT_{SUPH}	8	°C
ΔT_{oil}	103.5	°C
T_0	25	°C
P.P	5	°C
P_{max}/P_{crit}	0.9	-
$\prod_{T,max}$	60	°C
$T_{oil,\ max}$	290	°C

$$T_3 = 130^0C,\ (P_{min} = P_{sat}) \tag{9}$$

$$P_{max}1 = \prod_{T,max}.P_{min} = 60P_{min} \tag{10}$$

$$P_{max}2 = 0.9P_{crit} = 0.9x36.5 = 32.85bar \tag{11}$$

$$T_3 = T_{sat} + 8 \tag{12}$$

At isentropic point;

$$h_4 = h(p = p_{min}, s = s_3) \tag{13}$$

$$h_4 = h_3 - \eta_{is,T}(h_3 - h_4) \tag{14}$$

$$T_1 = T_C = 30^0\text{C, when } h_1 = h, T = T_1, s_1 = s \tag{15}$$

Therefore $h_{2,isen} = h(p - p_{max})$

$$h_2 = h_1 + w_{pump} = h_1 + \frac{h_{2,isen} - h_1}{\eta_{isen,pump}} \tag{16}$$

$$P_e = m_{ORC}.\eta_{Gen}.(h_3 - h_4) - \frac{m_{ORC}.w_{pump}}{\eta_{motor}} \tag{17}$$

Given that$P_e = 33.34kW$

$$Q_{net} = m_{ORC}(h_3 - h_2) = m_{oil}.\text{cp}_{oil}.\Delta T_{oil} \tag{18}$$

$$\eta_{ORC} = \frac{P_e}{m_{ORC}(h_3 - h_{2A})} = \frac{33.34kW}{Q_{net}} \tag{19}$$

Energy balance in the evaporator

$$\text{m}_{oil} = \frac{Q_{net}}{\text{cp}_{oil}.\Delta T_{oil}} = \frac{33.34kW}{79\ cp_{oil}.\eta_{ORC}} \tag{20}$$

$$T_{oil,in} = T_{sat} + PP + \frac{m_{ORC}.(h_3 - h(p = p_{max}))}{m_{oil}.\text{cp}_{oil}} = T_{sat} + 5 + 79\frac{\eta_{ORC}.(h_3 - h(p = p_{max}))}{(h_3 - h_4) - \frac{w_{pump}}{\eta_{motor}}} \tag{21}$$

The following screening properties were also taken into consideration: low ozone depletion potential (ODP), availability in the market, economical, higher efficiencies when used as a working fluid, low specific volumes, moderate pressures in the heat exchangers within the ORC system, low global warming potential (GWP), long service life, and super pump ability at low temperature among other properties.

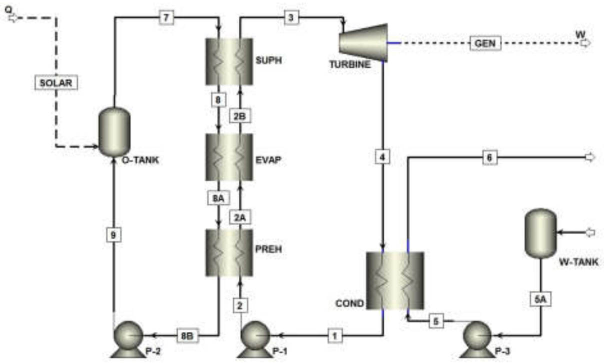

Figure 1 Schematic diagram of the organic Rankine cycle driven by solar energy

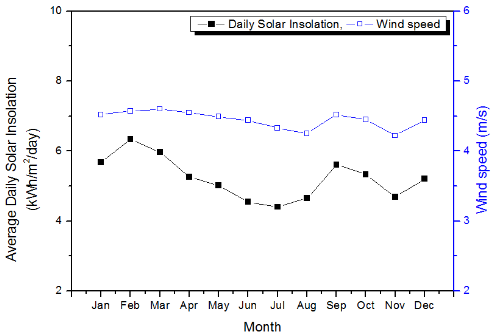

Figure 2 The average Monthly daily solar insolation and wind speed for Thika

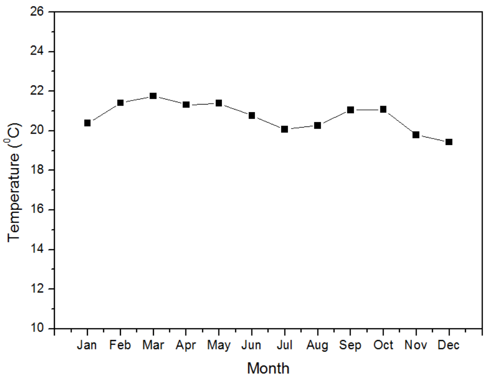

Figure 3 The average Monthly daily temperature for Thika

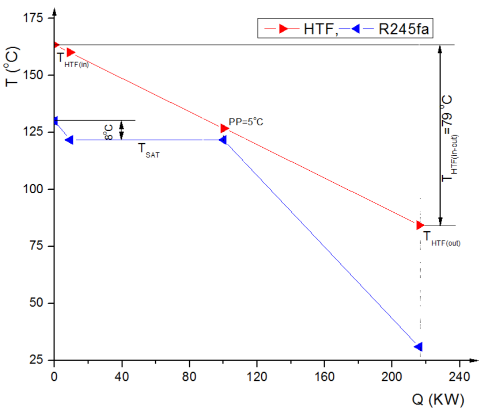

Figure 4 T-Q heat exchange between Heat Transmitting fluid (HTF) and working fluid

Therminol 55 used in this simulation as HTF is a synthetic hydrogen-carbon mixture with a noncorrosive nature to the metals used in heat transfer. It has kinetic viscosity of 19mm^2/s at 40°C, average molecular weight of 320g/mol and a critical pressure of 13.2 bar.

Various usual and realistic assumptions have been put into consideration. In order to simulate this system using Aspen plus v8.0 in a dynamic basis. First of all, the system nominal electricity power is selected to be 33.34kW el for all the examined cases. The maximum temperature of the heat transfer fluid in the inlet of the evaporator was set to be 31^{0}C, because this study aims to this temperature region. The HTF transfers its heat to the ORC system and its temperature is reduced by 79 ^{0}C . The pinch point in the evaporator was selected to be at 50 ^{0}C, according to T-Q diagram in Figure 4.

Load Estimation

In the present study a building apartment located in Thika is considered, and it is a 10 floor building with a total covered area of 33000 m^2 (3300 m^2 per floor) comprising of total 300 units (30 units each floor). The apartment includes corridor with the total covered area of 3000 m^2 (i.e. 300 m^2 per floor); dimension of corridor is 150 m (L) × 2 m (W), and that of one unit is 10 m (L) × 10 m (W) which is calculated to be a living area of 100 m^2 per unit. The roof to ceiling height is at 4 m. The average occupant per unit in the apartment is 5 people, total number of occupants is 1500 (i.e. 30 units × 10 floors × 5 persons). The required heating and power loads of this house are supposed to be provided by the proposed ORC cycle. To determine the heating and cooling loads requirements, the following assumptions were considered: wall insulation thickness is taken to be 1 inch; every window has internal shading; glasses are single pane; the building orientation is towards the west; and the roof is flat air vented with 1-inch insulation. In estimation of the electrical power required in the above mentioned residential apartment, the following appliances with the average power required for each one is assumed.

Table 3 Input values to the system

Parameters	Value	Units
Solar collector temperature t_7	163.5	°C
Isentropic efficiency for the turbine	80	%
pump isentropic efficiency	75	%
Electrical generator efficiency	95	%
Ambient temperature T_0	25	°C
Pinch point temperature of the evaporator & condenser	5	°C

Table 4 Result analysis simulated at different evaporator exit temperatures

Stream Parameters	129 °C		130 °C		131 °C	
	T (°C)	P (MPa)	T (°C)	P (MPa)	T (°C)	P (MPa)
ORC Working Fluid at Point 1	30	0.17	30	0.18	30	0.18
ORC Working Fluid at Point 2	31.06	2	31.06	2	31.06	2
ORC Working Fluid at Point 3	129	2	130	2	131	2
ORC Working Fluid at Point 4	60	0.12	60	0.12	60.02	0.1
Water (Inlet)	15	0.5	15	0.5	15	0.5
Water (Outlet)	54.87	0.5	54.89	0.5	53.33	0.5
Heat Transmitting Oil (Inlet)	163	0.35	163.5	0.35	164.5	0.35
Heat Transmitting Oil (Outlet)	84.5	0.35	84.5	0.35	84.5	0.35

Light (300 W), TV-LCD (158 W), washing machine (345 W), refrigerator (140 W), fan coil (140 W), computer (150 W), the total power consumption is taken to be 1.3 kW per house per day. To estimate the required load for domestic hot water supply, the maximum consumption is considered for each unit. Based on the calculations, the system is supposed to provide warm water at a rate of 3600 kg/h, and a temperature of 55 ℃ (∓0.5) to cater for the residents water consumption requirements. Given that each person consumes approximately 55 - 60 litres/day (shower/ bathing/ washing etc.), the total water consumption is 3600 litres per hour.

Energy, Anergy and Exergy analysis

Energy is neither destroyed nor created according to the first law of thermodynamics, hence energy is conserved. The first law of thermodynamics highlights on quantity of energy and fails to account for the quality of energy in the system.The following mass and energy balance equations are applied to investigate the heat transfer and work for a steady state condition with an assumption that $\boldsymbol{\Delta KE = 0\ and}\ \Delta\mathbf{PE} = \mathbf{0}$ [21, 23, 24]. Two major assumptions that are taken into consideration: 1) a steady state conditions are reached by the system, and 2) heat losses to the environment and the drop in pressure in the pipeline are neglected.

Table 5 Results for T = 130 °C

Parameters	Units	Value
W Turbine	kJ/hr	1.33×10^5
Q Condenser	kJ.hr	6.49×10^5
W pump	kJ/hr	5.49×10^3
Q Evaporator	kJ/hr	7.77×10^5
$mORC$	kg/hr	3.00×10^3
η_{ORC}	%	14.55
T_4	°C	60
T_7	°C	163.5
Enthalpy at point 1	kJ/kg	4569
Enthalpy at point 3	kJ/kg	4310
Enthalpy at point 4	kJ/kg	4351
Enthalpy at point 5	kJ/kg	16000
Enthalpy at point 7	kJ/kg	9280

Table 6 Exergy & anergy result analysis

Parameter ($\times 10^3$)	Evaporator	Turbine	Condenser	Pump
Ex^F [kJ/hr]	777.27	154.30	189.45	5.49
Ex^P [kJ/hr]	151.21	133.30	9.62	4.20
I [kJ/hr]	626.07	21.00	179.83	1.29

Equations used to perform the analysis for a basic ORC in Figure 1 are:

$$\sum \dot{m}_i - \sum \dot{m}_o = 0, \tag{22}$$

$$\dot{Q} + \sum \dot{m}_i h_i = \dot{W} + \sum \dot{m}_0 h_0, \tag{23}$$

where $\dot{W}$, $\dot{m}$, h, $\dot{Q}$, ‘i’, ‘o’ are work rate, rate of mass flow, specific enthalpy, heat transfer rate, subscript for inlets, and subscript for outlets, respectively.

Energy Analysis

Turbine

$$\dot{W}_T = \dot{m}_{ORC}(h_{3,ORC} - h_{4,ORC}) \tag{24}$$

Condenser

$$\dot{Q}_{COND} = \dot{m}_{ORC}(h_{4,ORC} - h_{1,ORC}) \tag{25}$$

Pump

$$\dot{W}_p = \dot{m}_{ORC}(h_{2,ORC} - h_{1,ORC}) \tag{26}$$

Preheater, Evaporator & Superheater

$$\dot{Q}_{Total} = \dot{m}_{ORC}(h_{3,ORC} - h_{2,ORC}) \tag{27}$$

$$\dot{m}_{ORC}(h_{3,ORC} - h_{2A,ORC}) = \dot{m}_{oil}(h_7 - h_{8A}) \tag{28}$$

$$\dot{m}_{ORC} = \frac{\dot{m}_{oil}(h_7 - h_{8A})}{(h_{3,ORC} - h_{2A,ORC})} \tag{29}$$

$$T_{2A} = T_{8A} - T_{PP} \tag{30}$$

T_{PP} is the pinch point which is fixed at 5 ^{0}C,

Net power

$$\dot{W}_{ORC} = \dot{W}_T - \dot{W}_P = \dot{W}_{3,4} - \dot{W}_{1,2} = \dot{Q}_{EVAP} - \dot{Q}_{COND} \tag{31}$$

$$\dot{W}_{ORC} = \dot{Q}_{2,2A} + \dot{Q}_{2A,2B} + \dot{Q}_{2B,3} - \dot{Q}_{COND} \tag{32}$$

Where $\dot{Q}_{2,2A}$,$\dot{Q}_{2A,2B}$,$\dot{Q}_{2B,3}$ represent the preheating, evaporating and superheating sections, and

$$T_{2A} = T_{2B} = T_{EVAP} \tag{33}$$

Thermal Efficiency

$$\eta_{ORC} = \frac{W_T - W_P}{Q_{EVAP}} \tag{34}$$

From the equation below, Exergy is a thermodynamic property which is dependent on the system surrounding environment (reference environment) in an extent that no changes in its intensive properties such as pressure (P_0), and temperature (T_0) of the system under analysis are observed. Hence the maximum theoretical work attainable from the interaction of a system with its surrounding environment up to when equilibrium state between them is reached and can also be seen as the departure state of a system from the reference environment. Its analysis makes use of both the first and the second law of thermodynamics which assists in overcoming of the various shortcomings of energy analysis.It is important in identifying the process inefficiencies magnitude, causes and locations of inefficiencies [32]. The specific exergy, energy and anergy for a flow stream disregarding potential and kinetic energies are given as:

$$Exergy = (h - h_0) - T_0(s - s_0) \tag{35}$$

$$Energy = (h - h_0) \tag{36}$$

$$Anergy = T_0(s - s_0) \tag{37}$$

The anergy $T_0(s - s_0)$, which is also known as the exergy destruction, and exergy efficiency for all components in the ORC system will be now defined below based on Figure 1. The anergy can be calculated as indicated in the equation below;

$$\dot{E} = T_0 S_{gen} \tag{38}$$

where $T_0$is the dead state temperature, and S_{gen} is the rate of entropy generation.

The system components exergy analysis is as follows [26].

Condenser

$$\dot{E}x^P_{Cond} = Q\dot{}_{Cond}\left(1 - \frac{T_0}{T_{mW}}\right) \tag{39}$$

$$T_{mW} = \frac{h_{4,ORC} - h_{1,ORC}}{S_{4,ORC} - S_{1,ORC}} \tag{40}$$

$$\dot{E}x^F_{Cond} = \dot{m}_{ORC}\left(h_{1,ORC} - h_{4,ORC} - T_0(S_{1,ORC} - S_{4,ORC})\right) \tag{41}$$

$$\dot{I}_{Cond} = \dot{m}_{ORC}(Ex_{1,ORC} - Ex_{4,ORC}) - Q\dot{}_{Cond}\left(1 - \frac{T_0}{T_{mW}}\right) \tag{42}$$

Turbine

$$\dot{E}x_T^P = \dot{W}_T \tag{43}$$

$$\dot{E}x_T^F = \dot{m}_{ORC}\left(h_{4,ORC} - h_{3,ORC} - T_0\left(S_{4,ORC} - S_{3,ORC}\right)\right) \tag{44}$$

$$\dot{I}_T = \dot{m}_{ORC}.T_0\left(S_{4,ORC} - S_{3,ORC}\right) \tag{45}$$

Pump

$$\dot{E}x_{Pump}^p = \dot{m}_{ORC}\left(h_{2,ORC} - h_{1,ORC} - T_0\left(S_{2,ORC} - S_{1,ORC}\right)\right) \tag{46}$$

$$\dot{E}x_{Pump}^F = \dot{W}_P \tag{47}$$

$$\dot{I}_{Pump} = \dot{m}_{ORC}.T_0\left(S_{2,ORC} - S_{1,ORC}\right) \tag{48}$$

Preheater, Evaporator & Superheater

$$\dot{E}x^p = \dot{E}x_{PREH} + \dot{E}x_{EVAP} + \dot{E}x_{SUPH} \tag{49}$$

$$\dot{E}x^F = \left(\dot{E}x_{PREH} + \dot{E}x_{EVAP} + \dot{E}x_{SUPH}\right)\left(1 - \frac{T_0}{T_{mH}}\right) \tag{50}$$

$$Where\ T_{mH} = \frac{T_{7,oil} - T_{8B,oil}}{In\left(\frac{T_{7,oil}}{T_{8B,oil}}\right)}$$

$$\dot{E}x_{PREH} = \dot{m}_{ORC}\left(h_{2,ORC} - h_{2A,ORC} - T_0\left(S_{2,ORC} - S_{2A,ORC}\right)\right) \tag{51}$$

$$\dot{E}x_{EVAP} = \dot{m}_{ORC}\left(h_{2B,ORC} - h_{2A,ORC} - T_0\left(S_{2B,ORC} - S_{2A,ORC}\right)\right) \tag{52}$$

$$\dot{E}x_{SUPH} = \dot{m}_{ORC}\left(h_{3,ORC} - h_{2B,ORC} - T_0\left(S_{3,ORC} - S_{2B,ORC}\right)\right) \tag{53}$$

$$\dot{E}x^p = \dot{m}_{ORC}\left(h_{3,ORC} - h_{2,ORC} - T_0\left(S_{3,ORC} - S_{2,ORC}\right)\right) \tag{54}$$

$$\dot{I} = \dot{Q}\left(1 - \frac{T_0}{T_{mH}}\right) - \dot{m}_{ORC}\left(\dot{E}x_{3,ORC} - \dot{E}x_{2,ORC}\right) \tag{55}$$

where $\dot{I}_{Cond}$, $\dot{I}_T$ and $\dot{I}_{Pump}$ are the irreversibilities in the condenser, turbine and pump, respectively.

RESULTS AND DISCUSSION

The analysis was performed based on exergy, mass and energy balances and the following assumptions were taken into consideration for simulated ORC system:

- The organic Rankine cycle system runs as a steady process.
- The chemical exergy of the materials is not taken into consideration.
- The expander/turbine and the pumps in the modelled system are adiabatic devices.
- The working fluid which is 1, 1, 1, 3, 3-pentafluroropropane (R-245fa) and at the pump inlet/condenser outlet is a saturated liquid.
- In any of the organic Rankine cycle devices, the pressure drop in the condenser and evaporator are neglected.

The model described in Figure 1 was used to simulate the behavior of the ORC system by varying the outlet temperature of the solar collector from 80 ^{0}C to 180 ^{0}C with a step size of 1 ^{0}C and hence variation in evaporator exit temperature in Table 4 shows the analysis of the results simulated at three consecutive evaporator exit temperatures using Aspen plus v8.0 simulator i.e. 129 °C, 130 °C and 131 °C. The desired results were obtained at 130°C and the analysis was done based on collector temperature 163.5 °C and evaporator exit temperature of 130 °C.

Table 5 and Table 6 indicate the energy and exergy rates respectively for the ORC system, optimum temperature at the evaporator exit (T = 130 °C)

Figure 5 and Table 6 show the amount of exergy destruction in various components and based on the above presentation, it can be easily used in identifying the locations where there is minimum and maximum exergy destruction percentage. The analysis assists in highlighting the improvements in the system components and improving the overall energy efficiency of the ORC system. From the above analysis, the major source of exergy destruction is on the evaporator, which accounts for over 70% of the total exergy destroyed in the system followed by condenser, turbine and pump in the respective order. The largest exergy was destroyed in the evaporator due to the difference in temperature change between the incoming and leaving hot water in the component. It can be realized based on the calculations and simulated results that further improvement can be achieved through careful design of the evaporator and operation at optimum temperature. This improvement opportunity for evaporator can significantly improve the efficiency of the ORC system. The pump has negligible amount of exergy destruction from the simulated results in the ORC components and may not need to be improved. Based on the results, it can be highlighted that there is still room for performance improvement opportunities of the ORC system by improving each and every component's performance in the ORC.

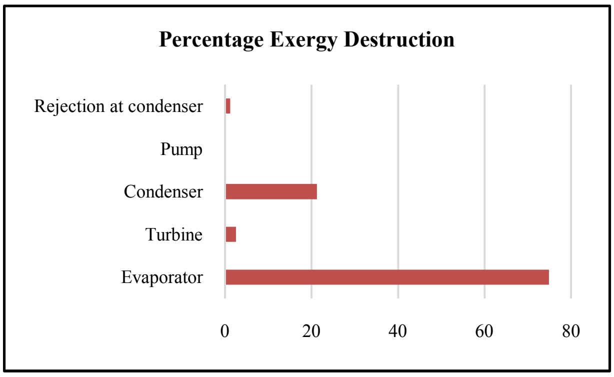

Figure 5 Percentage exergy of destruction in the ORC system components

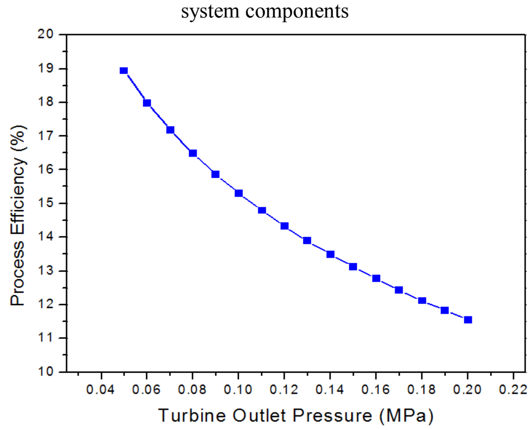

Figure 6 Variation of process efficiency with Turbine outlet pressure

In Figure 6, lowering the process efficiency tends to enhance turbine outlet pressure and vice versa. Thus the energy is decreased with increasing condenser pressure because there is a rise in the exergy destruction in the condenser.

Figure 7 shows the variation of the process efficiency when the condenser outlet temperature was varied.The process efficiency decreased with the increase of condenser outlet temperature, and this can be attributed to increase of outlet temperature in the turbine hence the heat exchange in the condenser increases. The condensing pressure in the ORC system has a distinct importance on the ORC system's performance since it assists in determining the overall heat rejection temperature, which is a crucial parameter for improving the efficiency in ORC apart from the heat input temperature. The condensing pressure varies inversely as the overall energy and exergy efficiency.

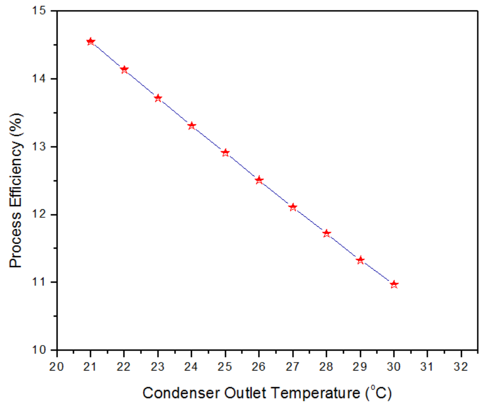

Figure 7 Variation of process efficiency with condenser outlet temperature

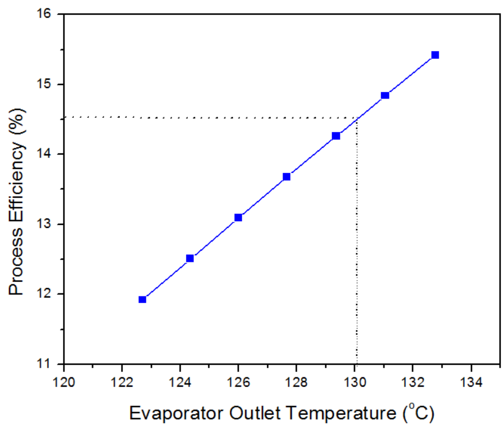

Figure 8 Variation of process efficiency with evaporator outlet temperature

The calculations of the working fluid mass flow rate, enthalpy, entropy and mole flow rate of the refrigerant R245fa, the corresponding, heat transmitting fluid (Therminol 55), hot water fluid and cooling water were calculated using Peng Robinson equation of state property set.

Figure 8 shows the variation of process efficiency with the outlet temperature in the evaporator. The process efficiency directly varies with evaporator outlet temperature. An overall efficiency is achieved at 14.55 % at evaporator outlet temperature of 130 ^{0}C.

Validation of Results

To validate the simulated model, various literatures and research work were carefully compared with the results achieved in this paper. The simulated data was validated by comparison with Quoilin *et al.* [35] work, under the same working fluid. The numerical results obtained show a very good agreement with the simulated data results in Table 7.

The ORC efficiency is 11.2 %, with R245fa as the working fluid. The absolute difference is 13.5 ^{0}C, with a relative difference of 9%. The absolute difference in ORC efficiency is 3.35 %, whereas the relative difference in pinch point is 37.5 %. Moreover, the difference in collector's efficiency and the solar hours are almost the same, about 59% and 8 hours respectively. The possible difference in power output mainly arise from the pressure drop of the working fluid variation and differences in various component's specifications.

Table 7 Results validation

Parameters	Ref. value[35]	Present value	Units
T_{evap}	150	163.5	°C
T_{amb}	15	25	°C
η_{ORC}	11.2	14.55	%
$\eta_{Col} T_0$	61.6	59	°C
P.P	8	5	°C
n	8	8	hrs.
$\prod_{T,max}$	60	60	
HTF	Monoethylene glycol	Therminol 55	

BouLawz Ksayer [36] using 1,1,1,3,3-pentafluroropropane (R-245fa) as the working fluid evaluated a solar ORC for domestic hot water production and electricity generation. The process efficiency predicted was 14.5% during the peak solar hours, and with the same working fluid used in this paper, the results tally with the simulated data obtained using Aspen plus v8.0 from this paper with best performance efficiency at 14.55% achieved. Suresh *et al.* [18] using the small scale ORC system for rural areas electrical power generation application determined experimentally that the highest exergy of destruction occurred in the evaporator and in the pump there was negligible exergy of destruction occurrence.This is in line with the results achieved in this paper based on the simulated results and the calculations done using Peng Robinson equation of state.

CONCLUSIONS

In this paper, the exergetical performance of ORC solar system has been analyzed using Aspen plus v8.0 simulation software based on the second law of thermodynamics.The exergy of destruction is greatest in the evaporator component in the system. The following can be concluded from this study, increasing the turbine inlet temperature results in increasing the system net electric efficiency, net electric power, and exergy efficiency leads to improved exergetic and energetic system performance, which leads to decrease in exergy reduction rate. The exergy destruction reduction methods within the components in the ORC system are proposed to model the energy transfer process of the collector in the solar power plant, the proposed model allows the performance of the system for a wide range of working and ambient conditions. This analysis shows a beneficial energy guide for the simulations under the optimal working condition (130°C) using solar as the source of energy simulations and subsequent calculations were carried out to research the heat transfer and exergy transfer performances along the ORC. The energy use efficiencies of the photovoltaic collector under different working conditions are revealed. The ORC system is an environmental-friendly technology, cost effective, simple to operate and it is suitable for Thika area given that the area experience a good solar conditions.

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