International Journal of Engineering and Innovative Research

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THERMODYNAMIC MODELLING OF A SOLAR POWERED ORGANIC RANKINE CYCLE

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(**Received:** 13.03.2019; **Revised:** 17.04.2019; **Accepted:** 21.04.2019)

ABSTRACT: A considerable amount of thermal energy is available in the form of renewable energy source and this can reduce the consumption of fossil fuels. The solar organic rankine cycle is a promising technology which uses energy from the sun as a source of power and this does not affect the environment. However, due to the recent global warming, environmental pollution and energy crises coupled with the instability of oil prices, interest in renewable energy for mitigating these issues is growing once again. The aim of this study is to develop a model for evaluating and predicting the net power output and performance of a solar powered organic rankine cycle and to validate the model using experimental data.

A thermodynamic analysis was carried out to see how feasible the power plant will operate on the chosen site, simulation were done in a Matlab environment , parametric and sensitivity analysis were also carried out to know the parameters that effect the system the most. A model was developed to predict the net power output and thereby performing a performance analysis. The model was validated using an experimental setup by Braden Lee Twomey, 2015 at University of Queensland Australia.

Measured/calculated and predicted net power output of the solar organic rankine cycle using R134a are 0.905kW, 0.913kW, 0.919kW and 0.908kW, 0.929kW, 0.920kW respectively. Measured/calculated and predicted net power output of the solar organic rankine cycle using R245fa are 0.973kW, 0.976kW, 0.979kW and 1.041kW, 0.940kW, 0.953kW respectively. The organic rankine cycle efficiencies and the overall solar organic rankine cycle efficiencies using R134a are 0.093, 0.086, 0.077 and 0.000028, 0.000030, 0.000032 respectively.

The organic rankine cycle efficiencies and the overall solar organic rankine cycle efficiencies using R245fa are 0.200, 0.185, 0.167 and 0.000060, 0.000065, 0.000069 respectively.

From the above result it can be deduced that the measured and predicted net power output are close with very little percentage error and as such the model is able to perform a performance analysis of the system.

Keywords: Efficiency, Net Power Output, Solar Organic Rankine, Thermal Energy.

1. INTRODUCTION

Energy is one of the primary causes of a nation's development and sustenance. It has been reported that the global demand for primary energy is on the steady increase and if the demand is maintained at a conservative average rate of 2%, the total global energy demand will increase by 100% in 30 years. The accelerated consumption of fossil fuels, if not abated would lead to a major health and energy crisis. However, the utilization of low-grade energy has attracted appreciable attention due to its potential in relaxing environmental pollution and fossil fuel consumption [1].

Worldwide energy demand has been rapidly increasing but the fossil fuel to meet the demand is being drained, for the past 20 years efficient uses of low-temperature energy source such as geothermal energy, exhaust gas from gas turbine system, biomass combustion, waste heat from various industrial processes, and solar energy have attracted much attention and researches about them become more and more important. The organic rankine cycle (ORC) and the power generating system using binary mixture as a working fluid have been focused on as they are proven to be the most feasible methods to achieve high efficiency in converting the low-grade thermal energy to more useful forms of energy. ORC is a rankine cycle where an organic fluid is used instead of water as working fluid. Particularly in low temperature applications many benefits may be obtained by using ORC instead of steam rankine process [2].

By powerful nuclear fusion reaction, the Sun produces staggering amounts of energy and much of that energy is dispersed in space and practically all of it is lost. The energy intercepted by Earth over a period of one year is equal to the energy emitted in just 14minutes by the Sun. The Sun releases an enormous amount of radiation energy to its surroundings and when the energy arrives at the surface of the Earth, it has been attenuated twice by both the atmosphere (6% by reflection and 16% by absorption) and the clouds (20% by reflection and 3% by absorption), as shown in Fig. 1 [4].

Solar collectors and thermal energy storage components are the two core subsystems in solar thermal applications. A solar collector which is the special energy exchanger converts solar irradiation energy to the thermal energy of the working fluid in solar thermal applications. Solar collectors need to have good optical performance in order to absorb heat as much as possible. For solar thermal applications, solar irradiation is absorbed by a solar collector as heat and then is transferred to the working fluid. The heat carried by the working fluid can be used to either provide domestic hot water or to charge a thermal energy storage tank from which the heat can be drawn for use later [3].

The aim of this research is to develop a model for predicting the net power output of a solar organic rankine cycle as shown in figure 1.

2. THE SOLAR ORC SYSTEM CONFIGURATION

The solar evacuated tube collector is heated using the energy from the sun, the heat from the sun causes the fluid in the collector (heat transfer fluid) to gain an increase in temperature and it then flows to the evaporator/storage unit where there is heat exchange between the heat transfer fluid (water) and the working fluid (refrigerant).

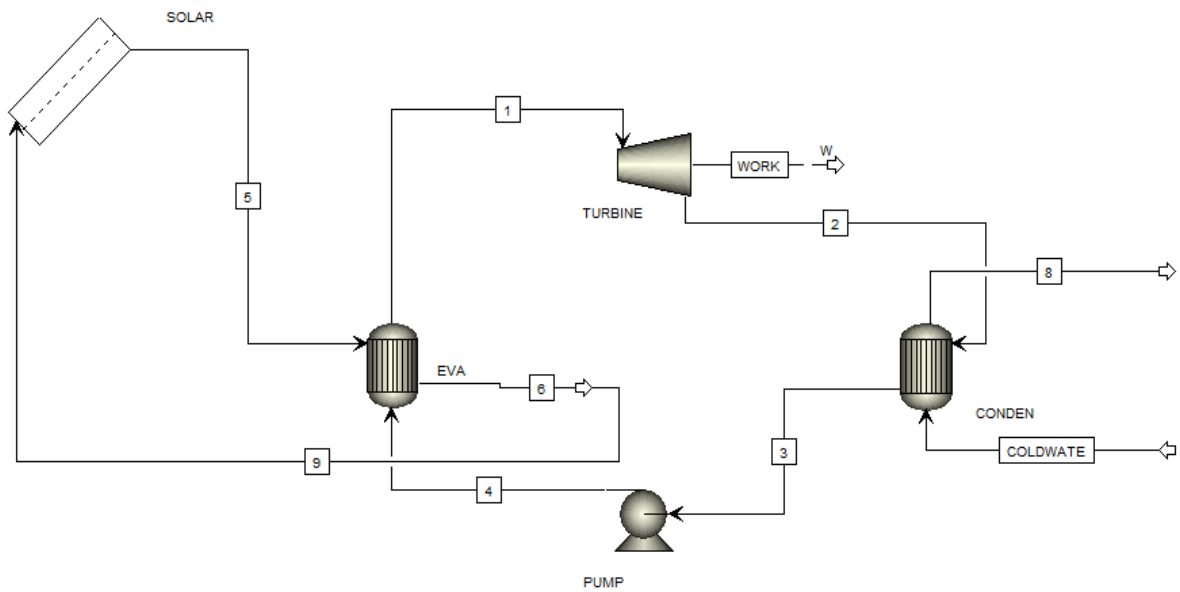

Figure 1. Schematic diagram of solar ORC.

As the working fluid gains more energy and increase in temperature, it moves to the expander/turbine where it causes expansion within the system and it connects to the generator for producing electricity. The remaining part of the refrigerant goes to the condenser where it loses energy in form of a drop in temperature as it mixes with cold water. The working fluid is then pumped back to the evaporator and the cycle continues.

2.1. Working Fluid Selection

R134a and R245fa are choosing for low-temperature and medium-temperature solar ORCS as working fluid respectively, considering the following factors like flammability, toxicity, and environmental conditions. Water is used as the heat transfer fluid and this solar ORC can be mostly suitable for remote places of developing countries that lack electricity. Therefore, this technology is expected to be used for small distributed power generations systems.

3. THERMODYNAMIC ANALYSIS

Data were collected putting into mind the site and location which is been used as case study. 1kW was fixed at the turbine/expander output, thermodynamic assumptions were used to conduct the thermodynamic analysis of this research putting into consideration the boundary conditions and the operational parameters are shown in Table 1.

The system was then simulated using MATLAB environment under steady state condition. A model was developed using dimensional analysis for performance prediction. The following equations were used for the analysis:

i. **Evaporator model**

$$\boldsymbol{Q_e = m_f(h_1 - h_4) = m_f C_{p,wf}(T_{C.I} - T_{C.O})} \quad \textbf{(1)}$$

The total heat rate in the evaporator from the heat transfer fluid (HTF) into the working fluid is given by using equation 1.

$Q_e = Heat\ rate\ of\ the\ evaporator, m_f = massflow\ rate\ of\ the\ heat\ trasfer\ fluid,\ h_1 = Specific\ enthalpy\ at\ inlet,\ h_4 = Specific\ enthalpy\ at\ outlet,\ C_{p,wf} = Specific\ capacity\ of\ water, T_{C.I} = Temperature\ at\ inlet\ of\ the\ evaporator$

$T_{C.O} = Temperature\ at\ outlet\ of\ the\ evaporator.$

ii. **Expander/turbine model**

$$\boldsymbol{W_t = m_f(h_1 - h_{2S})\eta_t\eta_g = m_f(h_1 - h_2)\eta_g} \quad \textbf{(2)}$$

The organic fluid which is the working fluid vapour passes through the expander/turbine to generate mechanical power. The turbine/expander can be analyzed using equation 2.

Where $\eta_t\ and\ \eta_g$ are the turbine/expander isentropic efficiencies and generator efficiency respectively. h_1, h_{2S}, h_2 are the specific enthalpies of the working fluid at the expander inlet, expander outlet under ideal and actual conditions respectively.

iii. **Condenser model**

$$\boldsymbol{Q_C = m_f(h_2 - h_3) = m_f C_{p,w}(T_{C.O} - T_{C.I})} \quad \textbf{(3)}$$

The exhaust vapour at the expander exit is directed to the condenser where it is converted to the liquid state by rejecting its heat to the cooling water. This condenser heat rate can be estimated using equation 3.

$Where$ $\boldsymbol{Q_C, m_f, h_2, h_3}$ are the heat generated, mass flow rate of the working fluid, specific enthalpy at condenser inlet and outlet respectively $\boldsymbol{m_w}, \boldsymbol{C_{p,w}}, \boldsymbol{T_{C.O}}\ and\ \boldsymbol{T_{C.I}}$ are the cooling water mass flow rate, specific capacity of water, cooling water temperature of outlet and inlet of the condenser respectively.

iv. **Pump model**

$$\boldsymbol{W_p = \frac{m_f v_3 (P_4 - P_3)}{\eta_p} = \frac{m_{f(h_{4s} - h_3)}}{\eta_p}} \quad \textbf{(4)}$$

The power consumed by the working fluid pump is estimated using equation 4. $Where\ \boldsymbol{W_p}, \boldsymbol{m_f}, \boldsymbol{v_3}, \boldsymbol{P_4}, \boldsymbol{P_3}, \boldsymbol{h_4}, \boldsymbol{h_3}\ and\ \boldsymbol{\eta_p}$ are pump work, working fluid mass flow rate, specific volume, pressure outlet, pressure inlet, specific enthalpy outlet, specific enthalpy inlet and pump efficiency of the pump respectively.

v. **Net power output and system efficiency**

The net power output generated by the SORC system is given as:

$$\boldsymbol{W_{net} = W_t - W_p} \quad \textbf{(5)}$$

The thermal efficiency of the ORC is the ratio of the net power output to the heat input in the evaporator. It can be expressed as:

$$\boldsymbol{\eta_{ORC} = \frac{W_{net}}{Q_e}} \quad \textbf{(6)}$$

The overall efficiency of the solar ORC system can be defined as follows:

$$\boldsymbol{\eta_{OVR} = \frac{W_{net}}{G_I \times A_{COL}}} \quad \textbf{(7)}$$

Table 1. Input parameters

S/N	COMPONENT	LEVEL 1	LEVEL 2	LEVEL 3
1	Low/medium SORC area	$32m^2/16m^2$	$30m^2/15m^2$	$28m^2/14m^2$
2	Low/medium solar ORC temperature	95 ^OC/130 ^OC	90 ^OC/120 ^OC	85 ^OC/110 ^OC
3	Pinch temperature difference at evaporator	10 ^OC	10 ^OC	10 ^OC
4	Low/medium ORC evaporator temperature	85 ^OC/ 120 ^OC	80 ^OC/ 110 ^0C	75 ^OC/ 100 ^OC
5	Expander power output	1kW	1kW	1Kw
6	Condenser outlet temperature	42 ^0C	42 ^0C	42 ^0C
7	Expander efficiency	70%	70%	70%
8	Pump efficiency	70%	70%	70%
9	Solar collector efficiency	70%	70%	70%
10	Generator efficiency	95%	95%	95%

3.1. Model Formulation and Performance Prediction

Dimensional analysis was carried out in order to establish an empirical correlation relative to each parameter. It is a mathematical technique which utilizes the knowledge of the dimensions by considering units of measurement.

$\boldsymbol{W_{net}}$ is a variable which depends on other variables as shown in the functional equation 3.8.

$$\boldsymbol{W_{net} = f(A, m_f, G_I, t, h)}\ \textbf{ or }\ \boldsymbol{f(A, m_f, G_I, t, h, W_{net}) = 0} \quad \textbf{(8)}$$

The net power output is dependent on mass flow rate, enthalpy, density, viscosity, temperature, pressure and area. Where;

$$W_{net} = ML^2T^{-3} \tag{9}$$
$$A = L^2 \tag{10}$$
$$m_f = MT^{-1} \tag{11}$$
$$G_l = MT^{-3} \tag{12}$$
$$t = \theta \tag{13}$$
$$h = Ml^2T^{-2} \tag{14}$$

Based on the above observation using equation (9 – 14), the pi groups are modelled coupled with the repeating variables ($M = A, m_f, G_L, t$) as follows;

$$\pi_1 = A^{a_1}, m_f^{b_1}, G_L^{C_1}, t^{d_1}, W_{net} \tag{15}$$

$$\pi_2 = A^{a_2}, m_f^{b_2}, G_L^{C_2}, t^{d_2}, h \tag{16}$$

$$f_1(\pi_1, \pi_2) \tag{17}$$

Using equation (3.20 – 3.22) the pi groups are modelled to obtain the following result

$$\pi_1 = \frac{W_{net}}{A\, G_l} \tag{18}$$

$$\pi_2 = \frac{h}{A\, G_l^{\frac{1}{2}}\; m_f^{\frac{1}{2}}} \tag{19}$$

$$f\left[\left(\frac{W_{net}}{A\, G_l}\right), \left(\frac{h}{A\, G_l^{\frac{1}{2}}\; m_f^{\frac{1}{2}}}\right)\right] \tag{20}$$

$$W_{net} = A\, G_l\, \emptyset\left(\frac{h}{A\, G_l^{\frac{1}{2}}\; m_f^{\frac{1}{2}}}\right) \tag{21}$$

From various literatures reviewed it was observed that low and medium SORC system works between 60^0C – 200^0C [Orosz and Dickes, 2017] using an evacuated tube. Refrigerant R134a was used for low temperature SORC considering its critical temperature of 101.06 ^{0}C and R245fa was used for medium SORC considering its critical temperature of 154.01^0C. Using simulated results and a correlation factor $R^2 = 0.987$ and $R^2 = 0.347$ regression analysis was done as shown in figure 2 and 3 respectively. Hence from the regression analysis Eq. (22 and 23) were obtained as follows:

$$\boldsymbol{Y = -6.5 \times 10^{-5} + 6.84 \times 10^{-5} X_1} \quad \textbf{(22)}$$

$$\boldsymbol{Y = -5.61715 \times 10^{-4} + 1.32115 \times 10^{-4} X_1} \quad \textbf{(23)}$$

By comparing Eq. (21) with Eq. (22 and 23), the empirical model or correlation for R134a and R245fa can be obtained as follows:

$$\boldsymbol{W_{net} = A\, G_l\, [-6.5 \times 10^{-5} + 6.84 \times 10^{-5} \left(\frac{h}{A\, G_l^{\frac{1}{2}}\, m_f^{\frac{1}{2}}} \right)]} \quad \textbf{(24)}$$

$$\boldsymbol{W_{net} = A\, G_l [-5.61715 \times 10^{-4} + 1.32115 \times 10^{-4} \left(\frac{h}{A\, G_l^{\frac{1}{2}}\, m_f^{\frac{1}{2}}} \right)]} \quad \textbf{(25)}$$

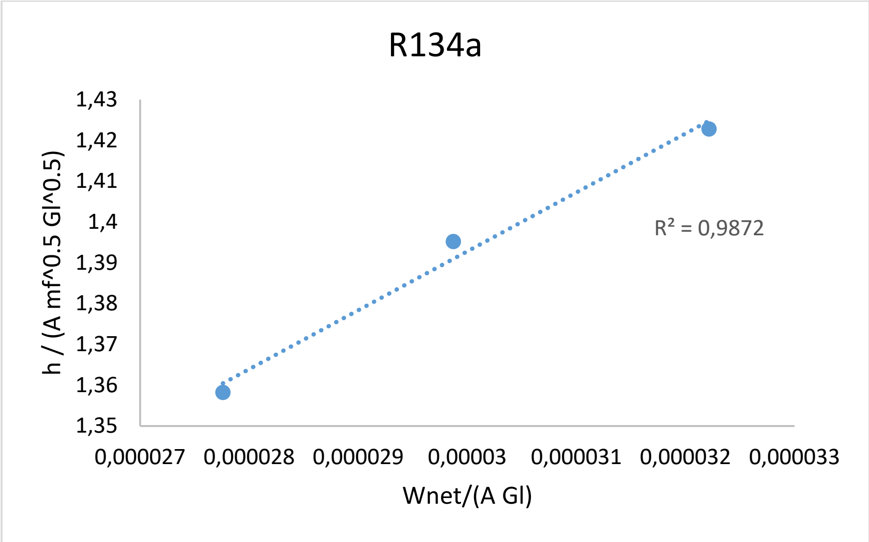

Figure 2. Regression plot of $\left(\frac{W_{net}}{A\, G_l}\right)$ as a function $\left(\frac{h}{A\, G_l^{\frac{1}{2}}\, m_f^{\frac{1}{2}}}\right)$ for R134a

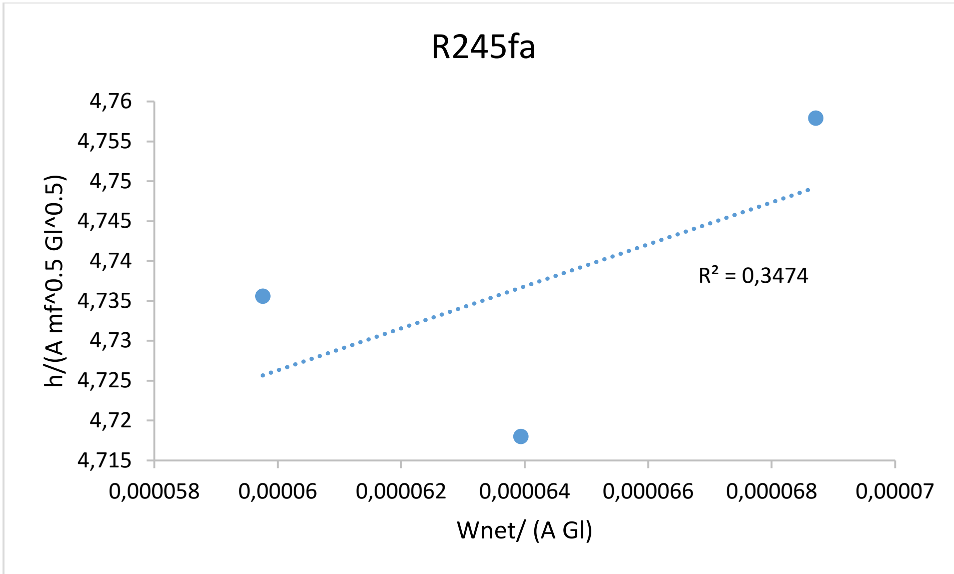

Figure 3. Regression plot of $\left(\frac{W_{net}}{A\,G_l}\right)$ as a function $\left(\frac{h}{A\,G_l^{\frac{1}{2}}\,m_f^{\frac{1}{2}}}\right)$ for R245fa

4. RESULTS

Table 2, 3 and 4 shows the result gotten form the thermodynamic analysis of using the equations from section 3.0.

Table 2. Result for both R134a and R245fa for level 1

Parameters	R134a	R245fa
h_1	427.760 kJ/kg	484.390 kJ/kg
h_2	409.830 kJ/kg	435.160 kJ/kg
h_{2S}	402.130 kJ/kg	414.060 kJ/kg
h_3	259.410 kJ/kg	255.290 kJ/kg
h_4	261.040 kJ/kg	256.580 kJ/kg
h_{4S}	260.550 kJ/kg	256.190 kJ/kg
m_f	0.059 kg/s	0.021 kg/s
Q_E	9.782 kW	4.871 kW
Q_C	8.825 kW	3.846 kW
W_P	0.096 kW	0.028 kW
W_{net}	0.905 kW	0.973 kW
η_{ORC}	0.093	0.200
η_{SORC}	0.000028	0.000060

Table 3. Result for both R134a and R245fa for level 2

Parameters	R134a	R245fa
h_1	428.810 kJ/kg	479.740 kJ/kg
h_2	412.240 kJ/kg	435.160 kJ/kg
h_{2S}	405.140 kJ/kg	416.050 kJ/kg
h_3	259.410 kJ/kg	255.290 kJ/kg
h_4	260.780 kJ/kg	256.300 kJ/kg
h_{4S}	260.370 kJ/kg	255.990 kJ/kg
m_f	0.064 kg/s	0.024 kg/s
Q_E	10.670 kW	5.275 kW
Q_C	9.710 kW	4.247 kW
W_P	0.087 kW	0.024 kW
W_{net}	0.913 kW	0.976 kW
η_{ORC}	0.086	0.185
η_{SORC}	0.000030	0.000065

Table 4. Result for both R134a and R245fa for level 3

Parameters	R134a	R245fa
h_1	429.020 kJ/kg	474.260 kJ/kg
h_2	414.170 kJ/kg	435.160 kJ/kg
h_{2S}	407.810 kJ/kg	418.410 kJ/kg
h_3	259.410 kJ/kg	255.290 kJ/kg
h_4	260.550 kJ/kg	256.060 kJ/kg
h_{4S}	260.210 kJ/kg	255.830 kJ/kg
m_f	0.071 kg/s	0.027 kg/s
Q_E	11.940 kW	5.874 kW
Q_C	10.970 kW	4.842 kW
W_P	0.081 kW	0.021 kW
W_{net}	0.919 kW	0.979 kW
η_{ORC}	0.077	0.167
η_{SORC}	0.000032	0.000069

5. CONCLUSION

A new model was developed to estimate net power output, the net power output gotten from R134a are 0.908kW, 0.929kW and 0.920 kW. The model estimated using R245fa generated the following net power 1.041kW, 0.940kW and 0.953kW. The model was also used to validate an experimental solar organic rankine cycle by Braden Lee Twomey, 2015 at University of Queensland Australia. The measured and calculate output was 1.0365kW and the predicted net output is 1.1937kW using R134a. The measured/calculated net power gotten was 3.853kW and the predicted net power was 3.833kW using R245fa.

From the above result it can be deduced that the measured and predicted results are close with very small percentage error and as such the model is able to perform a performance analysis of the system.

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