OPERATION OPTIMIZATION AND PERFORMANCE STUDY OF SOLAR POWERED ORGANIC RANKINE POWER PLANT WITH REGENERATOR

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ABSTRACT

This paper presents operation optimization and performance study of solar powered organic Rankine power plant with regenerative feed using toluene as the cycle working fluid. In the operation optimization, the optimised parameters are : extraction fraction , intermidiate pressure and mass flowrate of the cycle working fluid and they are optimised with respect to efficiency and the net power output. In the performance study, the effects of hot fluid inlet temperature, high pressure turbine inlet temperature, condenser pressure and boiler pressure on efficiency and system net power generation have been investigated. Considered performance parameters are efficiency and the net power output. The optimization result reveals that increasing the extraction fraction, intermediate pressure and working fluid mass flow rate increases the efficiency and the net power output at the beginning until they attain their corresponding maximum values and continuing to increase these values causes the reduction of both efficiency and the net power output. The performance study also indicates that increasing the hot fluid inlet temperature and boiler pressure causes an increase in net power output and thermal efficiency. But increasing high pressure turbine inlet temperature and the condenser pressure results in reduction of both efficiency and the net power output.

Keyword : *Organic rankine cycle, solar power generation, regenerative feed, efficiency, power output.*

1.0   INTRODUCTION

Concentrating Solar Power (CSP) systems have been implemented with a variety of collector systems such as parabolic trough, solar dish, solar tower or the Fresnel linear collector. However, most of the currently installed concentrated solar power plants use steam Rankine cycle in the power block. This technology requires a minimum power of a few MW in order to be competitive and involves high collector temperature. Particularly in the case of small-scale systems, organic Rankine cycles (i.e. a Rankine cycle using an organic fluid instead of water) show a number of advantages over the steam cycle. These include a lower working temperature, the absence of droplets during expansion, low maintenance requirements and the simplicity (fewer components). According to McMahan, those advantages make the organic Rankine cycle (ORC) technology more economically attractive

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when used at small and medium power scales [1]. Several research studies have been done on ORC, particularly on solar organic Rankine cycles both theoretically by Davidson [2] and Probert [3], and experimentally by Monahan [4] as early as in the 1970s with reported overall efficiencies varying between 2.52% and 7%.

However, only some studies involved in performance study and operation optimization of the organic Rankine cycle power plant. In fact, with any given flow and temperature conditions of heat source, power may be generated with different set of operating variables. But only one set of operating variables results in maximum power output or thermal efficiency. The optimal operation results from tradeoffs between energy generation and consumption of different components. For instance, increasing working fluid mass flow rate increases expander power output but also consumes more pump power. Reducing the condensing temperature requires more power consumption of the air cooled condenser fan, but allows increasing on expander power output.

The purpose of this study is to accomplish operation optimization and performance study of solar powered ORC power plant and search the optimal operating parameters in order to achieve the best system thermal efficiency and the maximum system net power generation.

2.0   SYSTEM DESCRIPTION

As shown in Figure 1, the organic Rankine cycle solar powered plant used in this study is composed of turbines, boiler, condenser, pumps, regenerative feed, solar collector field and other auxiliary equipments. Heat from the heat source (solar receiver) is pumped into boiler where toluene is evaporated and changed into vapor. Toluene in the form of vapor is then delivered to the high pressure turbine at the boiler pressure (state 2) and expands isentropically to an intermediate pressure (state 3) as shown in figure 2. Some toluene in the vapor form is extracted at this state and routed to the regenerative feed heater, while the remaining part of toluene continues to expand isentropically through the low pressure turbine and then to the condenser pressure (state 4). The toluene leaves the condenser as a saturated liquid at the condenser pressure (state 5). The condensed toluene then enters an isentropic feed pump, where it is compressed to the intermediate pressure at state 6, and is routed to the regenerator heater, where it mixes with the toluene vapor extracted from the turbine.

The fraction of the toluene extracted is such that the mixture leaves the heater as a saturated liquid at the heater pressure (state 7). A second pump raises the pressure of the Toluene to the boiler pressure (state 1). The cycle is completed by heating the toluene in the boiler to the turbine inlet state (state 2).
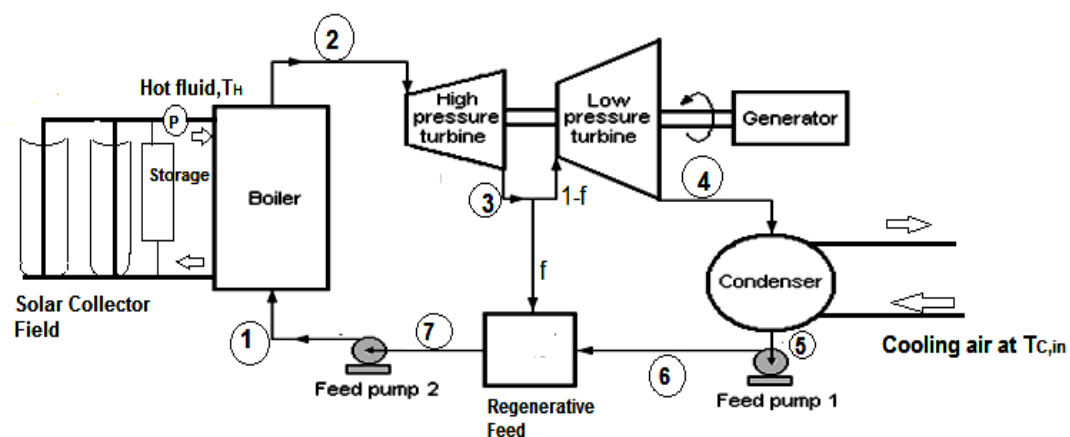

Figure 1: Schematic of Solar Powered ORC Power plant

The device where working fluid is heated by regeneration is called a regenerator (regenerative feed). Regeneration not only improves cycle efficiency, but also provides a convenient means of de aerating the regenerator (i.e removing the air that leaks in at the condenser) to prevent corrosion in the boiler. It also helps control the large volume flow rate of the working fluid at the final stages of the turbine (due to the large specific volumes at low pressures).

The percentage of the total cycle mass flow rate used for the regenerator heater is termed as the extraction fraction f, and must be carefully optimized for maximum power plant thermal efficiency since increasing this fraction causes a decrease in turbine power output [5].

The heat transfer fluid is heated up in the collector field and driven to the boiler by the heat transfer fluid pump. In the boiler, heat will be exchanged between the high temperature heat transfer fluid and the cycle working fluid, toluene. A thermal storage is installed in order to attenuate the fast fluctuations of solar radiation during the day and to maintain stable operation of ORC plant.
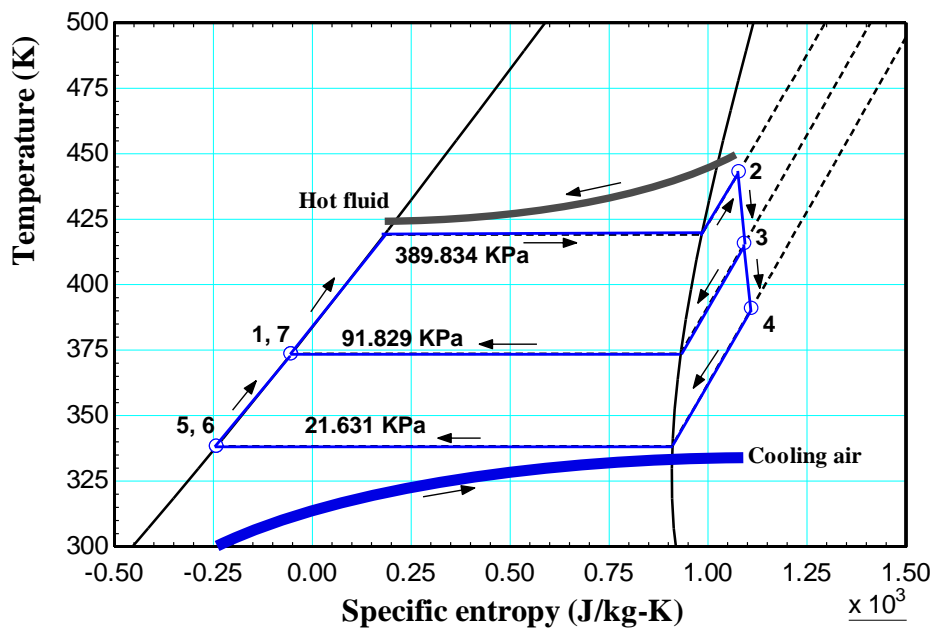

Figure 2: Temperature versus Specific entropy diagram of ORC

3.0   SYSTEM MODELING

The system is modeled by treating each of the major components i.e., boiler, condenser, turbine, pump and regenerative feed as single control volume. The following assumptions were made during system modeling.

  i.   Steady operating conditions exist.
 ii.   Kinetic and potential energy changes are negligible.
iii.   Friction and pressure losses in pipes and components are neglected.

The heat addition and rejection is obtained by energy balance across the boiler and condenser. The work interactions are also obtained by the same method.

The boiler heat addition (heat transfer rate from hot fluid) can be modeled by

$$\dot{Q}_H = \dot{m} \cdot (h_2 - h_1) \tag{1}$$

Evaporating temperature can be obtained as in the following.

$$\dot{Q}_H = \varepsilon_b \cdot \dot{m}_H \cdot c_H \cdot (T_{H,in} - T_b) \tag{2}$$

where ε_b is effectiveness of the boiler given by the following equation.

$$\dot{e}_b = 1 - \exp\left[\frac{-UA_b}{\dot{m}_H \cdot c_H}\right] \tag{3}$$

The energy balance over the condenser gives condenser heat transfer rate of

$$\dot{Q}_C = \dot{m} \cdot (1 - f) \cdot (h_4 - h_5) \tag{4}$$

where f is the extraction fraction.

And the condensing temperature, T_{cond} is obtained from the following equation.

$$\dot{Q}_C = \dot{e}_{cond} \cdot \dot{m}_C \cdot c_C \cdot (T_{cond} - T_{C,in}) \tag{5}$$

where condenser effectiveness, ε_{cond} is given by:

$$\dot{e}_{cond} = 1 - \exp\left[\frac{-UA_{cond}}{\dot{m}_C \cdot c_C}\right] \tag{6}$$

The regenerator heater is generally well insulated ($Q = 0$) and does not involve any work interactions ($W = 0$). By neglecting kinetic and potential energy changes of the streams, energy balance on regenerator results in the following equation.

$$fh_3+(1-f)h_6=h_7 \tag{7}$$

Energy balance on the turbines and pumps gives the following equations. The power output of high pressure turbine and low pressure turbine are given by the following equations.

$$W_{Hpt} = m(h_2-h_3) \tag{8}$$
$$W_{Lpt} = m(1-f)(h_3-h_4) \tag{9}$$

Similarly, the power input to the high pressure pump and low pressure pump are given by the following equations respectively.

$$W_{HPp} = m(h_1-h_7) \tag{10}$$
$$W_{LPp} = m(1-f)(h_6-h_5) \tag{11}$$

The equations of power output and thermal(cycle) efficiency of the ORC power plant are given by the following equations.

$$W_{net} = W_{Hpt} + W_{Lpt} - W_{HPp} - W_{LPP} \tag{12}$$
$$\eta_{cycle} = W_{net} / Q_H \tag{13}$$

The cycle efficiency in equation 13 is also called thermal efficiency .

The intermediate pressure which is equal to (P_3, P_6 or P_7) is obtained by the following equation

$$P_{int} = P_{cond} \cdot \sqrt{\frac{P_b}{P_{cond}}}$$

(14)

The thermodynamic properties(tempreture,pressure and enthalpies)of working fluid (toluene) at various states of the ORC are obtained from the built-in functions inside EES software [6].

4.0 SOLAR ENERGY MODELING

In this study, Parabolic Trough Concentrator (PTC) will be integerated as a solar field to the organic rankine power plant . Hence ,in what follows,the thermal performance equations will be developed for PTC.

The useful solar heat collected is transferred to hot liquid storage tank from which the boiler is supplied with input thermal energy.Necessary equations for modelling solar energy are as follows .

The useful heat gain rate from each PTC solar collector is given by , Duffie [7] as

$$\dot{Q}_u = F_R \cdot A_a \cdot \left[I_c \cdot t \cdot a_p \cdot r - U_L \cdot \left(\frac{T_{f,in} - T_a}{CR} \right) \right]$$

(15)

The instantaneous collector efficiency is the ratio of useful heat gain from collector to the solar energy .It is given by the following equation.

$$\dot{h}_c = F_R \cdot \left[t \cdot a_p \cdot r - U_L \cdot \left(\frac{T_{f,in} - T_a}{CR \cdot I_c} \right) \right]$$

(16)

Heat removal factor, F_R is given by the following equation:

$$F_R = \frac{\dot{m}_H \cdot C_{p,dt}}{p \cdot D_o \cdot L_t \cdot U_L} \cdot \left[1 - \exp\left(\frac{- F' \cdot p \cdot D_o \cdot U_L \cdot L_t}{\dot{m}_H \cdot C_{p,dt}} \right) \right]$$

(17)

Where the collector efficiency factor F' and overall heat loss coefficient U_L are given by the following equations [9].

$$\dot{F}' = \frac{1}{U_L \cdot \left[\frac{1}{U_L} + \frac{D_o}{D_i \cdot h_f} \right]}$$

(18)

$$\frac{\dot{1}}{U_L} = \frac{1}{C_3 \cdot (T_{pm} - T_c)^{0.25} + \frac{S \cdot (T_{pm}^2 + T_c^2) \cdot (T_{pm} + T_c)}{\frac{1}{e_p} + \frac{D_o}{D_{ci}} \cdot \left[\frac{1}{e_c} - 1 \right]} + \frac{D_o}{D_{co}} \cdot \left[\frac{1}{h_w + S \cdot e_c \cdot (T_c^2 + T_a^2) \cdot (T_c + T_a)} \right]}$$

(19)

The empirical Equation (19) was developed by (Mullic and Nanda)[8]. The constant C_3 has been obtained from correlation of Raithby and Hollands [8] as follows.

$$C_3 = \frac{17.74}{(T_{pm} + T_c)^{0.4} \cdot D_o \cdot (D_o^{-0.75} + D_{ci}^{-0.75})}$$
(20)

The cover temperature T_c is given by the following equation :

$$\frac{T_c - T_a}{T_{pm} - T_a} = 0.04075 \cdot \left[\frac{D_o}{D_{co}}\right]^{0.4} \cdot h_w^{-0.67} \cdot \left[2 - 3 \cdot e_p + \frac{(6 + 9 \cdot e_p) \cdot T_{pm}}{100}\right]$$
(21)

Where mean plate temperature(T_{pm}) should be in the range : $333 < T_{pm} < 513K$.
The total rate of solar energy input to the solar field containing a total number of n_c collectors is obtained by :

$$Q_{solar} = A_a \cdot n_c \cdot I_c$$
(22)

The total rate of useful heat transfer gained from solar field is given by

$$Q_{field} = Q_u \cdot n_c$$
(23)

The field efficiency which is defined as the ratio of total rate of useful heat transfer gained from solar field to the total rate of solar energy input to the solar field is given by the following equation.

$$\eta_{field} = Q_{field} / Q_{solar}$$
(24)

Eventually the overall efficiency which is the product of cycle efficiency and field efficiency is given by the following equation.

$$\eta_{overall} = \eta_{cycle} \cdot \eta_{field}$$
(25)

5.0   METHODOLOGY OF THE STUDY

By choosing the values of the following design parameters , all the above modelling equations has been simulated by Engineering Equation Solver(EES) [6]. Because the number of design and input parameters which may be varied in the ORC is large, it is not feasible to develop performance data for the entire range of these quantities. For this reason, the technique of varying one parameter at a time while holding all others fixed was applied. The result is the response of the system performance for each of the variables being analyzed.

5.1   Design input parametrs of Organic Rankine cycle power plant
In order to optimise the ORC power plant, the following parameters were taken.

i.   For the tubines and pumps the following efficiency values were taken : $h_{Hpt} = 0.87$ (High pressure turbine efficiency), $h_{Hpt} = 0.84$ (Low Pressure turbine efficiency) and $h_{HPp} = 0.58$ (High pressure Pump efficiency) , $h_{LPp} = 0.52$ (Low pressure pump efficiency)
ii. The overall fluid to fluid conductance of the boiler, $(UA)_b$ and condenser, $(UA)_c$ are respectively taken to be 33KW/°Cand 35KW/°C.
iii. The rate of heat input to the boiler by the solar collectors(Solar field) has been taken to be 500KW($Q_H = 500KW$).In all the work that follow, this is the system heat input value which is considered to be constantly supplied by the solar parabolic trough collector and hence, optimisation and parametric study will be done with respect to this heat input value.
iv. The cooling air mass flowrate has been taken to be $m_c = 11.09kg/s$ and high temperature fluid mass flow rate, $m_H = 8.6$ kg/s .

v. The specific heat capacity of heating fluid (DOWTHERM A) and cooling air are $C_H=2000J/Kg.K$ and $C_c=1000J/Kg.K$ respectively.

vi. The cooling air inlet temperature, $T_{c,in}$ has been taken to be equal to ambient temperature(that is $T_{c,in} = 25^oC$). The hot fluid inlet temperature $T_{H,in}$, which is the temperature of the hot heat transfer fluid coming from the solar parabolic trough collector to the boiler.Its value is 200^oC and the high pressure turbine inlet temperature is 170^oC.

vii. Toluene has been taken as a working fluid for the cycle of the considered ORC power plant.

viii. The heating fluid and the cooling medium used are synthetic organic heat transfer fluid (DOWTHERM A) and air respectively.

5.2.1 Parameters of the Solar Parabolic Trough Collectors

In order to integrate the ORC with solar collector and to optimise the overall efficiency , the Parabolic Trough Collector(PTC) whose dimmension shown in Figure 3 was considered.The parabolic Trough Collector with $30m^2$ aperture area has been chosen.The main parameters are as shown in Table 1.The datas are from the manufacturer's product information. For the purpose of optimisation and to see the variation of some parameters over a day, a particular date on January 21st, 2012 in Riyadh has been considered. Hence,solar angle data and solar radiation calculation has been done based on the location of Riyadh on the date mentioned.In order to calculate the slope of the aperture plane and angle of incidence, Tracking Mode II was adopted [8].

Table 1: Parameter of the Parabolic Trough Collectors (PTCs) [9].

Item	Value
Collector aperture area(m^2)	30
Collector aperture (m)	2.5
Length (m)	12
Rim angle (^{o})	31
Focal Distance (m)	2.25
Concentration Ratio	12.2
Inner diameter of the reciever (m)	0.0605
Outer diameter of the reciever(m)	0.0635
Inner diameter of glass envelop(m)	0.099
Outer diameter of glass envelop(m)	0.102
Mirror reflection coefficient	0.90
Glass transmission coefficient	0.9[a]
Absorber absorption coefficient	>0.9[a]
Absorber emissive coefficient	0.08@100^oC,0.20@400^oC
Absorber tube conductivity coefficient(W/(mK)	1.2
Glass conductivity coefficient(W/(mK)	40

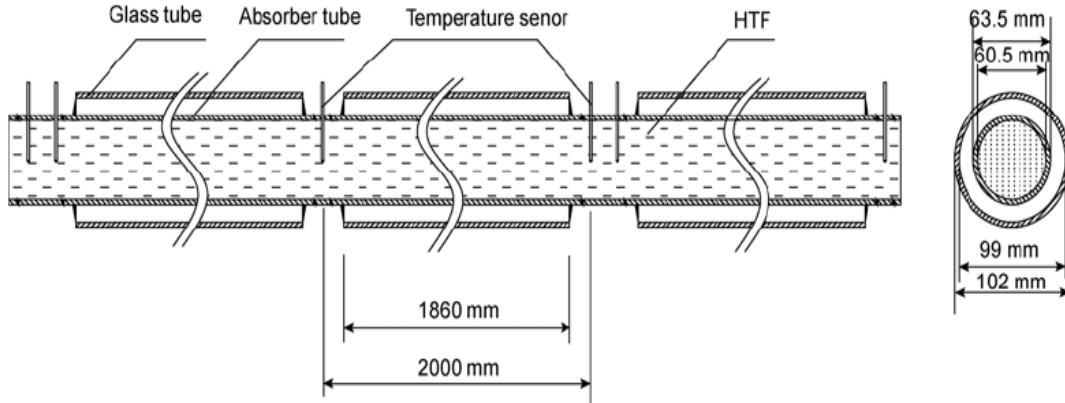[a] Data from the manufacturer's product information [9].

Figure 3 : Schematic of the Cross section of Parabolic Trough Collector [9].

6.0   RESULTS AND DISCUSSION

6.1   Operation optimisation

In the operation optimisation,the objective is to achieve the best cycle efficiency (i.e system thermal efficiency),overall efficiency or to achieve the highest net power generation. The parameters to be optimised are : Extraction fraction f, Intermidiate pressure (P_{int}) and mass flowrate of the working fluid (m)and they will be optimised with respect to efficiency and the net power output.The results are discussed as in the following.

Figures 4 and 5 show the effect of extraction fraction, f, of the total cycle mass flow rate extracted for the regenerative feed heater on efficiency and the net power output respectively. It can be seen that both efficiency and the net power output increase with the fraction f reaching their maximum value at f of 0.106 and then decrease.

Figures 6 and 7 show the effect of intermediate pressure on power output and efficiency respectively. In both figures we can clearly see that the cycle efficiency, overall efficiency and power output first increase reaching a maximum value and then decrease.This indicates that there is an optimum value of intermediate pressure (P_{int}) where the cycle efficiency ,overall efficiency and the net power output reach their maximum value.
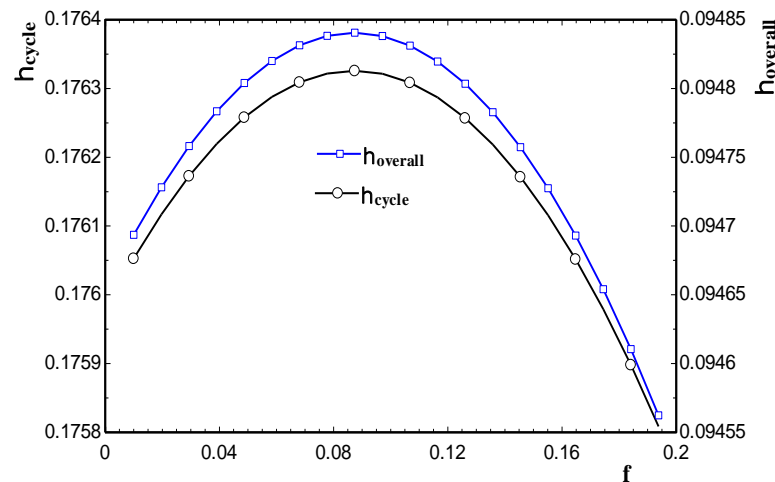

Figure 4 : Effect of extraction fraction on cycle efficiency and overall efficiency.
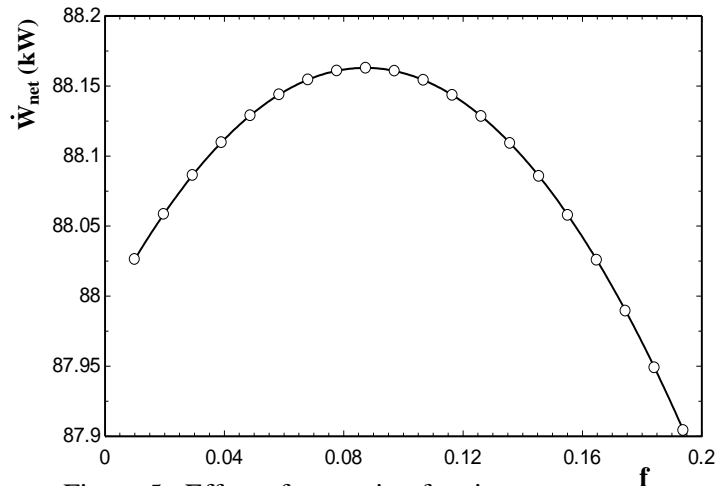
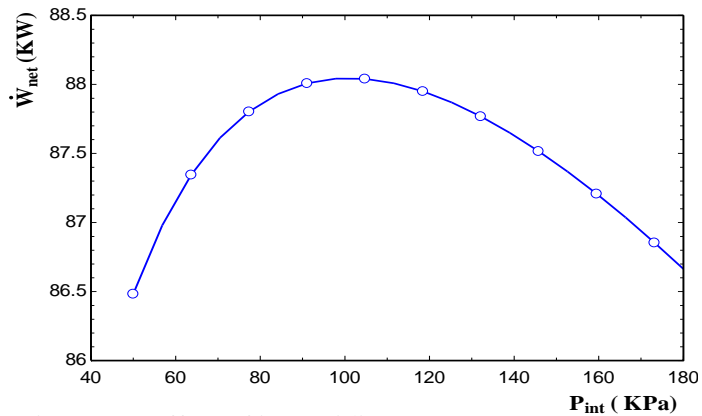Figure 5 : Effect of extraction fraction on net power output

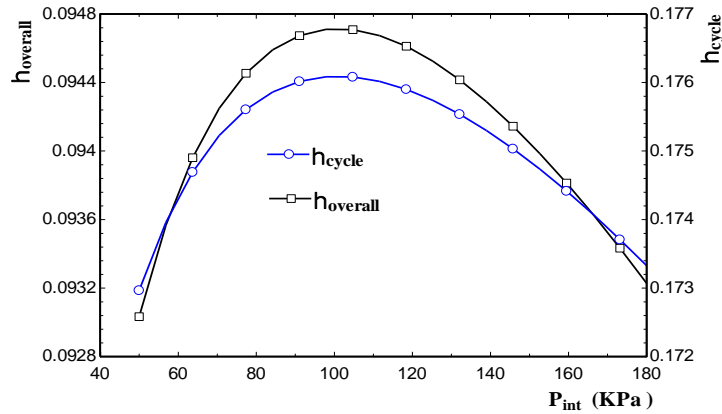Figure 6 : Effect of intermidiate pressure on power output

Figure 7 : Effect of intermidiate pressure on overall and cycle efficiency.

Figures 8 and 9 show the effect of toluene(cycle working fluid) mass flowrate on power output and efficiencies respectively. It can be clearly seen that increasing mass flowrate of toluene causes both power output and efficiency to increase until they attain their maximum values. However,continuing to increase the mass flowrate causes the power output and the efficiency to decrease. It is true that increasing working fluid mass flow rate results in more power input to the pump. On the other hand, increasing mass flow rate increases turbine power output. This is the reason why the working fluid mass flow rate should be optimized.
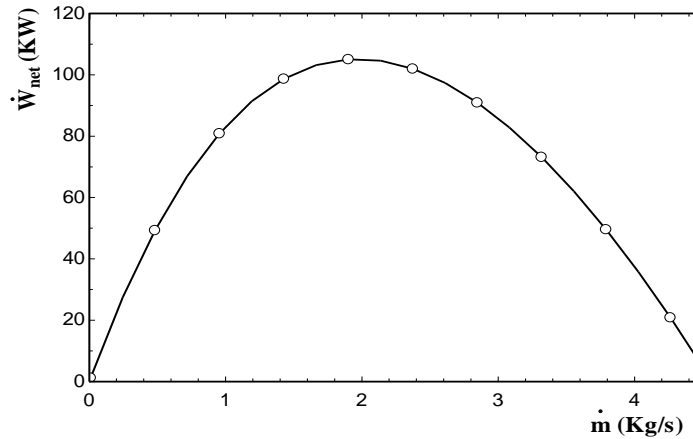
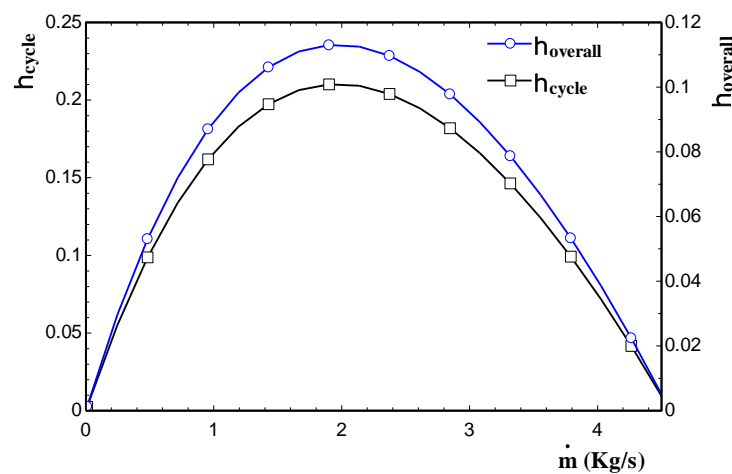Figure 8 : Effect of toluene mass flowrate on power on output

Figure 9 : Effect toluene mass flowrate on efficiency

6.2   Performance study of solar powered ORC plant

The solar powered ORC power plant shown in Figure 1 is simulated by using Engineering Equation Solver and by using input parameters shown in section 5.1. The plant simulation has been done based on mathematical modeling described in Sections 3 and 4.

The objective of performance study of solar powered ORC plant is to illustrate the effect of parameters like : hot fluid inlet temperature ($T_{H,in}$), high pressure turbine inlet temperature ($T_{t,in}$) ,condenser pressure (P_{cond}) and boiler pressure (P_b) on cycle efficiency (thermal efficiency), overall efficiency and net power generation. In addition to the above parameters, the variation of total solar radiation, useful heat gain from the parabolic trough collectors and instantaneous collector efficiency over a day in Riyadh have also been studied. The design input parameters are : heat input to the boiler, pump and turbine efficiency, overall fluid to fluid conductance, hot fluid inlet temperature from collector, cooling air inlet temperature and mass flow rate are fixed at their design values shown in Section 5.1. Thermodynamic properties are calculated by Engineering Equation Solver which already has a built-in function for thermodynamic properties.

Figure 10 indicates the effect of inlet temperature of hot fluid coming from solar collector on net power output and the efficiencies. The figure clearly shows that increasing the hot fluid inlet temperature to the boiler increases the net power output, overall efficiency and the cycle efficiency. But increasing this temperature causes the field efficiency of the solar collector to decrease. The total effect is that the overall efficiency increases as this temperature increases.
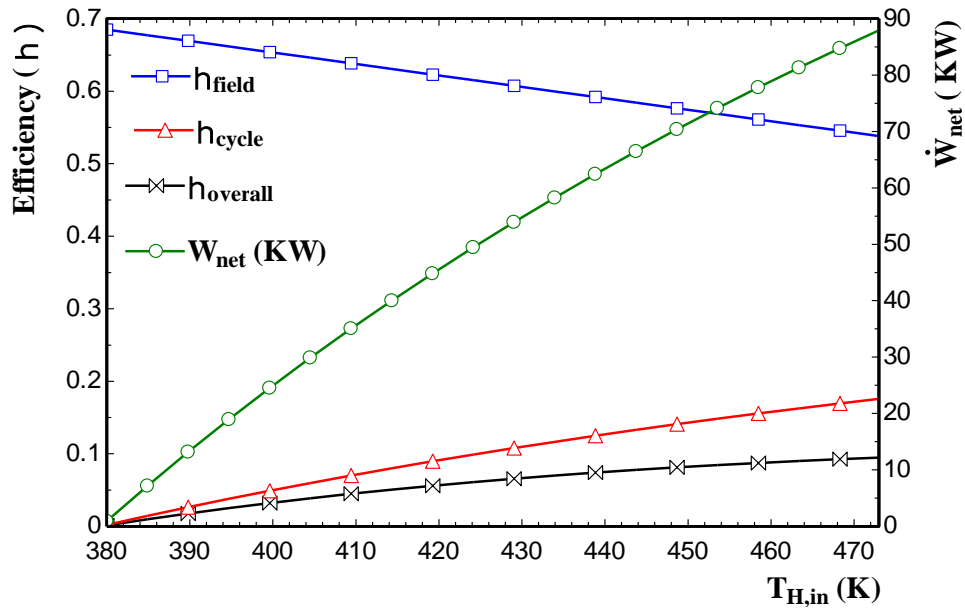
Figure 10 : Effect of hot fluid inlet temperature on net power output and efficiency.

Figure 11 shows the variation of efficiency and net power output with high pressure turbine inlet temperature. Increasing the turbine inlet temperature decreases the system thermal efficiency (cycle efficiency), overall efficiency and the net power generation. It is seen that it is better to operate the power plant at temperature of 443K (170°C) which is the design temperature. The reason is that, for ORC power plant, the turbine inlet temperature is limited by the material from which the turbine blade is made of.
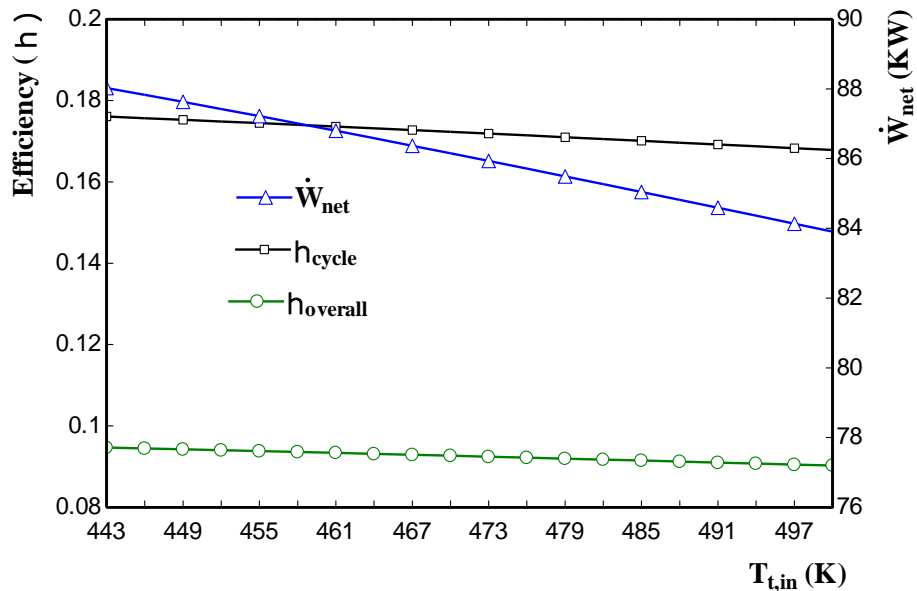

Figure 11 : Effect of high pressure turbine inlet temperature on net power output and efficiency.

Figure 12 shows the variation of efficiency and net power output with condenser pressure. It can be clearly seen that increasing the condenser pressur reduces both efficiency and the net power output. Hence, from the stand point of maximising the system thermal

efficiency and net power power output, it can be clealy seen that it is better to operate the power plant at lowest condenser pressure of 21kPa.

Figure 13 shows the variation of efficiency and net power output with boiler pressure. We can see that increasing the boiler pressure causes both efficiency and the net power out put to increase. If we want to maximise the system thermal efficiency, overall efficiency and net power power output, increasing the boiler pressure seems to be better. But increasing the boiler pressure is limited by energy input to the boiler and the boiler heat transfer area which are directly related to economics. Hence it is better to operate the power plant at the boiler pressure optimised for both economy and efficiency.
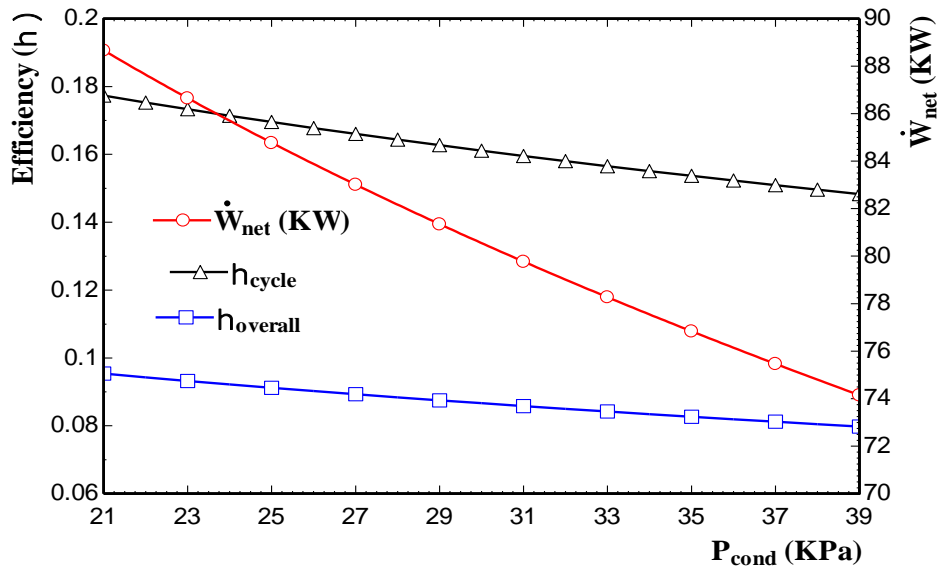

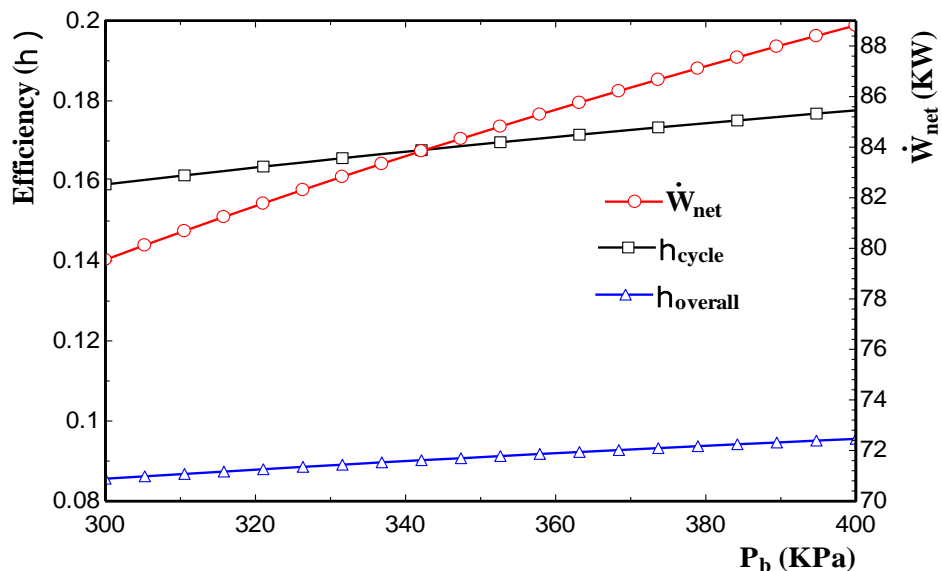Figure 12 : The effect of condeser pressure on net power output and efficiecy.

Figure 13 : The effect of boiler pressure on net power output and efficiecy.

Figure 14 shows the variation of useful heat gained from the solar collector and the instantaneous collector efficiency over a day in January 21st, 2012 in Riyadh. It can be seen that both efficiency and the useful heat gain increases from 8:00 AM until they attain their maximum value at 12:00AM. However after 12:00 AM, they reduce. From Figure 15,

similar behaviour for both field efficiency and total solar radiation incident on solar collector were observed.

Figure 16 indicates the variation of total useful heat collected from the solar field (PTC collectors) and the total solar radiation incident on solar collector. Here again the same behaviour was observed as that of Figures 14 and 15. The total solar radiation incident on PTC collectors in Riyadh is obtained by multiplying the ASHRAE clear sky model with adjustment factor in January as suggested by Sami A.Al-sanea [10] and it is shown in Table 2.

In all the three Figures, there is a high fluctuation of performances. Because of this, thermal storage is installed in order to attenuate the fast fluctuations of solar irradiation during the day and to maintain stable operation of ORC plant.
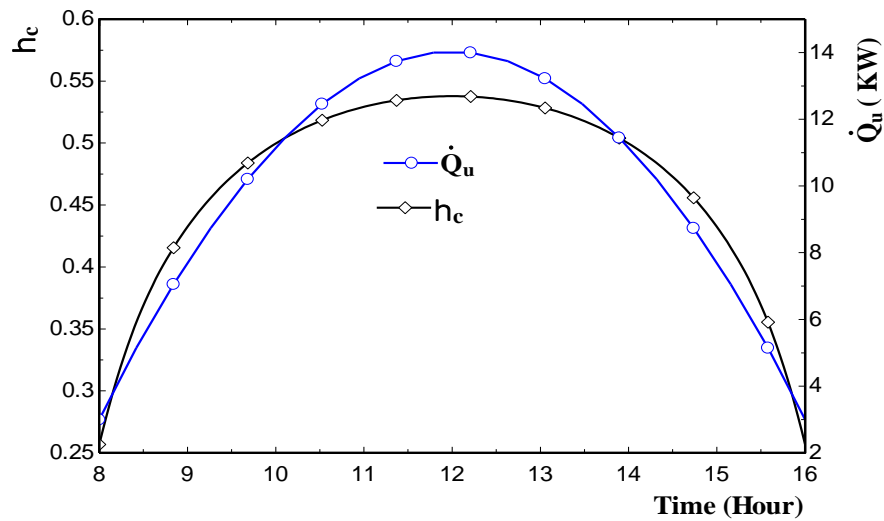

Figure 14 : Variation of useful heat gain from the PTC collector and the instantaneous collector efficiency over a day of January 21st, 2012 in Riyadh.
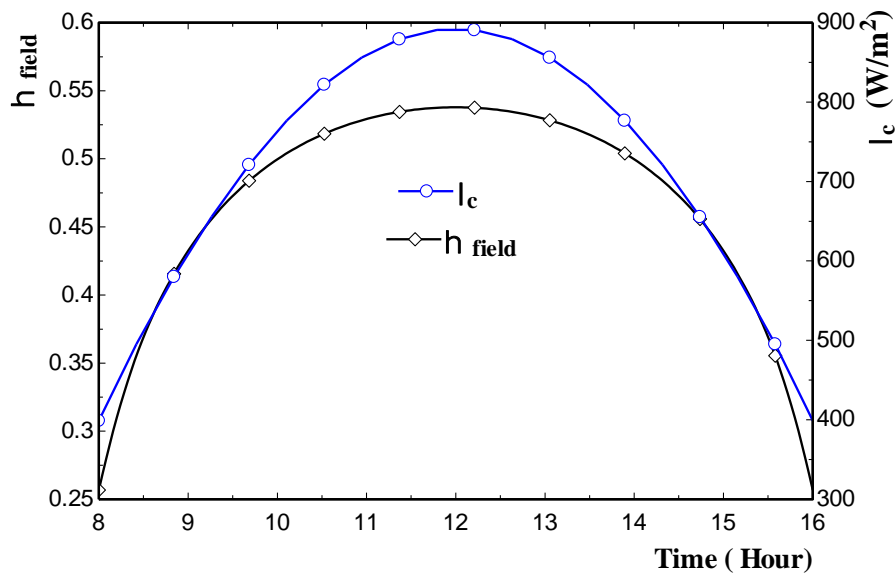

Figure 15 : Variation of total solar radiation incident on PTC collector and field efficiency over the day of January 21st , 2012 in Riyadh.
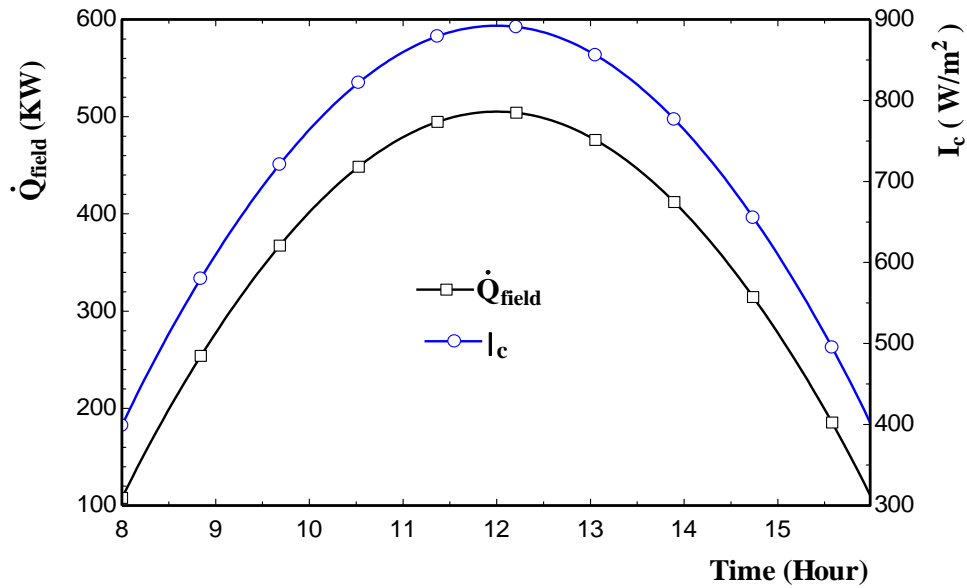
Figure 16 : Variation of total useful heat gain from the field and the total solar radiation over the day of January 21st, 2012 in Riyadh.

Table 2 : Solar Insolation incident on Parabolic Trough Collector, I_c(W/m²) for Riyadh, Saudi Arabia in 2012.

Hour	January	April	June	July	August
7:00	83.85	236.2	368.2	170.1	250.9
8:00	399.1	445.1	554.4	422.1	475.4
9:00	609.7	635.3	725.9	640.9	678.5
10:00	763.9	784.7	863.7	809	837.9
11:00	859.7	879.7	952.4	914.6	939
12:00	892.3	912.2	983	950.6	973.7
13:00	859.7	879.7	952.4	914.6	939
14:00	763.9	784.7	863.7	809	837.9
15:00	609.7	635.3	725.9	640.9	678.5
16:00	399.1	445.1	554.4	422.1	475.4
17:00	83.85	236.2	368.2	170.1	250.9

7. 0   CONCLUSIONS

This paper presents the operation optimization and performance study of solar powered organic Rankine power plant with regenerative feed using toluene as the working fluid. Mathematical model for the boiler, condenser, turbine, pump, parabolic trough solar collector and other accessory components have been developed to evaluate the plant performance. The operation optimization and performance study indicate that

i.   The extraction fraction, toluene mass flow rate and intermediate pressure are optimized in order to maximize the system efficiency and the net power generated.
ii. Increasing the extraction fraction of the total mass flow rate for the regenerative feed increases the efficiency and the net power output at the beginning until they attain their corresponding maximum values and continuing to increase the extraction fraction causes the reduction of both efficiency and the net power output.
iii. The effect of both intermediate pressure and the toluene mass flow rate on efficiency and the net power output is such that both efficiency and the net power output increase at the beginning until they attain their maximum value and then decrease.
iv. Increasing the hot fluid inlet temperature causes an increase in net power output, thermal efficiency and the overall efficiency of the system.
v.   The high pressure turbine inlet temperature has the effect of decreasing both efficiency and the net power generated.
vi. Increasing the boiler pressure causes an increase in both efficiency and the net power output, but increasing the condenser pressure results in reduction of both efficiency and the net power output.
vii. The useful heat gain from the parabolic trough collector, total solar radiation and the collector efficiency vary over a day in such a way that they increase starting from 8:00AM reaching their corresponding maximum value at 12:00AM and they keep on declining after 12:00 AM.

Nomenclatures

A_a effective collector aperture area
C_H hot fluid heat capacity
C_c cooling air heat capacity
CR   concentration ratio
D_o Absorber inner diameter
D_i Absorber outer diameter
D_{ci} glass cover inner diameter
D_{co} outer cover outer diameter
ηc instantaneous collector efficiency
η_{cycle} thermal efficiency of the plant
η_{field} efficiency of solar field composed of collectors
$\eta_{overall}$ overall efficiency of the solar powered ORC system.
η_{Hpt} high pressure turbine efficiency
η_{Lpt} low pressure turbine efficiency
η_{Hpp} high pressure pump efficiency
η_{Lpp} low pressure turbine efficiency
ρ Reflectivity of concentrator surface
τ transmissivity
m_H Hot fluid mass flow rate
m_C Cooling air mass flow rate
m mass flow rate of toluene
ε_b effectiveness of boiler
ε_{cond} effectiveness of condenser
f extraction fraction
F' Collector Efficiency Factor
F_R collector heat removal factor
h enthalpy
I_c total solar radiation incident on collector
Q_H Rate of heat transfer from hot fluid in boiler (Rate of heat input to the boiler)
Q_u useful heat rate gained by the collector

Q_{solar} total rate of solar energy input to the solar field
Q_{field} total rate of useful heat transfer gained from solar field
P_b boiler pressure
P_{cond} condenser pressure
P_{int} Intermediate pressure
T_a ambient temperature
T_b boiler temperature
T_{cond} condenser temperature
$T_{f,in}$ collector fluid inlet temperature
$T_{t,in}$ high pressure turbine inlet temperature
$T_{H,in}$ hot fluid inlet temperature
T_{pm} mean plate temperature
X quality
UA_b conductance of boiler
UA_{cond} conductance of condenser
U_L collector overall loss coefficient
W_{net} net power produced
W_{Hpt} high pressure turbine power
W_{Lpt} low pressure turbine power
W_{Hpp} high pressure pump power
W_{Lpp} low pressure pump power

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