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Optimal Operating Conditions of an Organic Rankine Cycle under Steady Heat Input

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Abstract: In the frame of a small scale concentrating solar power plant design and construction in Ouagadougou, an ORC (organic rankine cycle) machine is used to operate as a power block. In order to characterize the system and further optimize its operating conditions, a model is developed in this paper to describe the operating process of the thermodynamic cycle. The so-called model is obtained by interconnecting the mathematical models of its components. Then, simulations are conducted to determine the parameters that impact mostly the efficiency of the system. Furthermore, authors conducted an exergy analysis of the system to evaluate the exergy destruction of every component for a chosen operating condition.

Key words: CSP (concentrating solar power), ORC, exergy analysis, energy analysis, modelling.

Nomenclature

General

CD	Condenser	[-]
C_p	Specific heat capacity	[kJ·kg^{-1}·K^{-1}]
EV	Evaporator	[-]
H	Specific enthalpy	[kJ·kg^{-1}]
m	Mass flow rate	[kg·s^{-1}]
p	Pressure	[bar]
Q	Thermal power	[kW]
RC	Recuperator	[-]
S	Specific entropy	[kJ·kg^{-1}·K^{-1}]
T	Temperature	[K]
W	Thermodynamic work	[kW]
X	Overall exergy	[kW]

Subscripts/Superscripts

0	Dead state
1	Outlet of the condenser
2	Outlet of the pump
3	Inlet of the evaporator
4	Outlet of the evaporator
5	Outlet of the turbine
6	Outlet of the recuperator
d	Destroyed
ex	Exergy
gen	Generated
in	Inlet
irr	Irreversible
out	Outlet
PP	Pump
TB	Turbine
rev	Reversible
u	Useful

Greek symbol

ε	Effectiveness	[-]
η	Efficiency	[-]
ψ	Specific exergy	[kJ·kg^{-1}]

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1. Introduction

Currently, ORCs (organic rankine cycles) receive a growing interest because they allow the valorization of low grade temperature heat sources [1], which is a way to increase the energy efficiency of various industrial processes or maximize the exploitation of thermal energy from renewable sources. This is especially the case for µ-CSP (micro-concentrating solar power) plants, whose heat sources are usually below 250 °C [2]. In the context of the development of a µ-CSP named "CSP4Africa" [2], it has been decided to use a commercial ORC machine. For the operating of this ORC, parameters such as mass flow rates of the HTF (heat transfer fluid) and the working fluid, evaporator pressure, etc., need to be selected. Then, questions were raised regarding the best operating conditions, disregarding any internal regulation of the machine. In this paper, we investigate the effect of the operating conditions on the performance of the power cycle. The aim is to get the best combination of these parameters in order to extract the maximum work from a power supply of 100 kW, available with an inlet temperature of 210 °C.

For this purpose, we first review the various models of ORCs available in the literature. Basing on this literature work, we present a simplified model of ORC. This model is then used to investigate the performance of the selected ORC machine under various operating conditions.

1.1 Description of the Power Block of the CSP4Africa Micro-CSP Plant

The overall design of the CSP4Africa demonstration power plant (Fig. 1) has been presented by N'Tsoukpoe et al. [2].

The power block is a commercial ORC coupled with a dry-cooler. The characteristics of the power block are presented in Table 1. The characteristics of these heat exchangers, which are plate-type heat exchangers made of stainless steel, are provided in Table 2. The ORC uses Novec 649 as working fluid with a nominal electric efficiency of 8.6%.

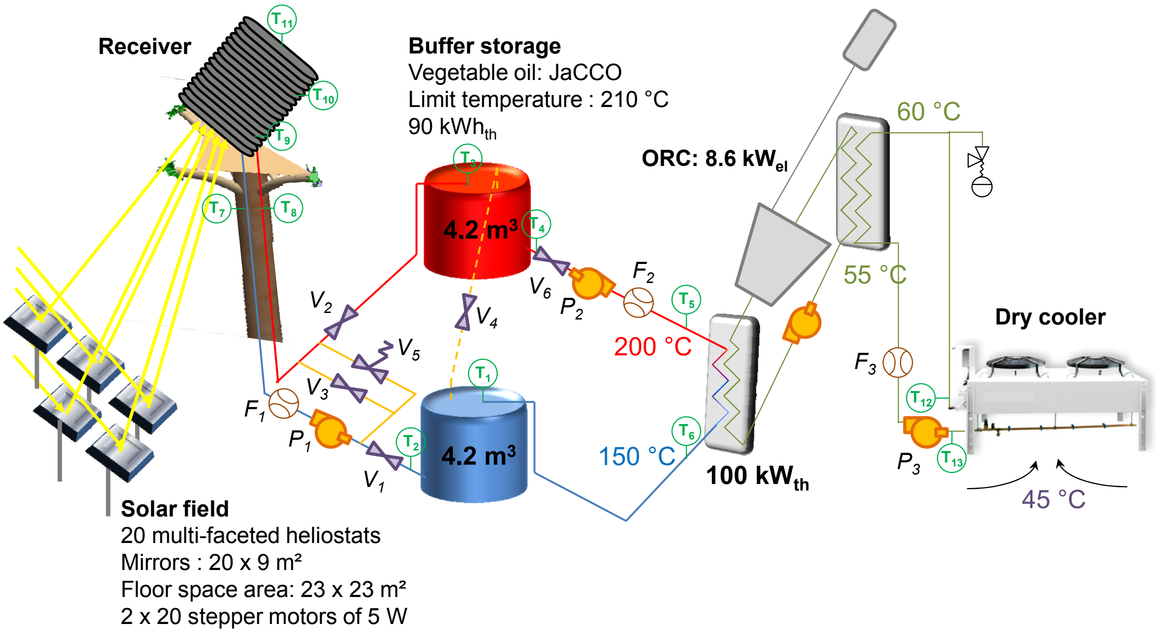

Fig. 1 Layout of the analyzed micro-CSP plant [2].

Table 1 Main features of the power block in nominal conditions.

Plant nominal size	8.6 kW$_e$
Thermal power input at evaporator	100 kW$_{th}$
Evaporator inlet temperature	200 °C
Evaporator outlet temperature	150 °C
Cooling system	Dry cooling
Design point ambient temperature	45 °C
Condenser inlet temperature	45 °C
Condenser outlet temperature	55 °C
ORC design point efficiency	8.6%

Table 2 Characteristics of the heat exchangers of the ORC.

Component	EV	CD	RC
Total heat exchange area (m²)	10.1	12.6	7.42
Width of a plate (m)	0.243	0.243	0.364
Height of a plate (m)	0.525	0.525	0.374
Thickness of a plate (mm)	1	1	1
Number of plates	80	100	60
Spacing of plates (mm)	0.5	0.5	0.5

For the demonstration plant, the nominal solar thermal power supplied to the conversion cycle is set to 100 kW_{th}.

The system includes two thermal storage tanks; the storage plays a role of buffer storage and is mainly devoted to the management of the intermittent energy supply such as during the passage of clouds.

However, heat from the storage supplements the direct generated heat from the sun when this direct heat does not reach 100 kW_{th}, especially at the start-up and next to the shutdown of the plant. One of the ideas to make the power generation from the sun more sustainable, considering local context, is the use of JaCCO (*Jatropha curcas* crude oil), a locally produced non-edible vegetable oil, as high temperature heat transfer fluid.

Fig. 2 shows a view of the ORC system used in this work. State 1 refers to the outlet of the condenser and the inlet of the pump. State 2 refers to the outlet of the pump and the inlet of the recuperator. State 3 refers to the outlet of the recuperator and the inlet of the evaporator while State 4 refers to the outlet of the evaporator and the inlet of the turbine. State 5 refers to the outlet of the turbine and the inlet of the recuperator while State 6 refers to the outlet of the recuperator and the inlet of the condenser.

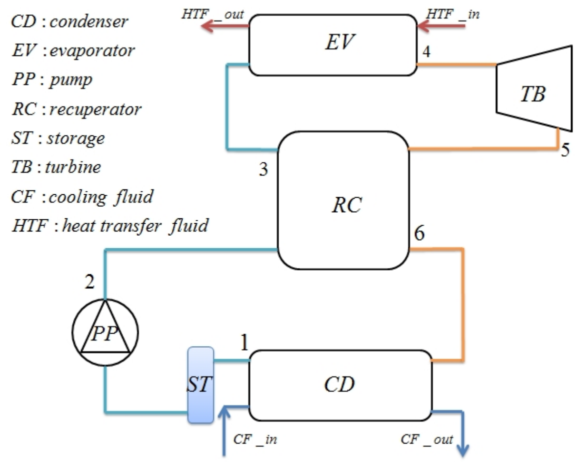

Fig. 2 Schematic view of the ORC system.

2. Modeling Methods for Organic Rankine Cycles

In order to determine the optimal operating conditions of an ORC machine or to maximize its efficiency, a thermodynamic optimization is required and is based on the prediction of the performances of the ORC. However, in the case of projects using ORC machines, two cases are generally observed:

- The first case, which uses new ORC systems that must be totally sized taking into account the requirements of the application;
- The second case, which deals with existing systems.

As a result, the prediction of the performances of these ORC systems is usually based on two approaches:

- The first approach, more adapted to the first case, uses mathematical models to describe the operating process of every component of the system such as the evaporator, the recuperator, the condenser, the pump, the turbine and sometimes the storage tank; the model of the whole system is then obtained by connecting models to their components.
- The second approach focuses more on the overall modeling of the whole system by considering it as a single component where some parameters can be tuned or modified in order to optimize its impact and performances.

In the first approach where every single component in the system is described by a mathematical model, the operating regimes of the system need to be considered. In fact, according to the application for which the ORC system is set up, the characteristics such as temperature and mass flow rate of the hot reservoir can vary or can be stationary constraining then the ORC system to operate in dynamic or in stationary regimes. In this first approach, the most critical components of the ORC system in term of modeling are heat exchangers since the changes in the characteristics of the hot reservoir affect their performances contrary to electromechanical components (pumps, turbines, expanders) which are always modeled in stationary regimes.

Literature on the ORC gives some methods to model heat exchangers (evaporator, condenser and recuperator): the ε-NTU method, the LMTD method, the Finite Volume discretization and the Moving Boundary model. It is important to notice that according to the operating regime of the ORC, some of the models are more suited than others. Thus, ε-NTU and the LMTD methods are regularly used to describe exchangers in stationary regimes while Finite Volume Discretization and Moving Boundary models are more suited to dynamic modeling.

The ε-NTU method is generally used when the mass flow rates of the two fluids, their inlet temperatures in the exchanger, the type and the dimensions of the latter are known [3]. The work therefore consists in calculating the thermal power exchanged by the two fluids and their temperatures at the outlet of the exchanger. However, a major problem occurs: the phase change of the working fluid makes the ε-NTU method unsuitable since the specific heat of the aforementioned fluid can no longer be considered constant throughout the exchanger. Vargas et al. [4] divide the heat exchanger into three virtual zones corresponding to the state of the working fluid, namely the liquid zone, the boiling zone and the superheating zone. He thus applies the ε-NTU method to each of these three zones, starting with the superheating zone where the temperature of the hot reservoir is known. He thus gradually determines the temperatures of the two fluids at the boundaries of each zone, including the temperatures at the outlet of the exchanger.

This method has been adopted by Bamgbopa and Uzgoren [5] for the study of a steady state ORC. The authors also proposed an algorithm based on this method for the calculation of the surfaces of each of the zones concerned, knowing the total surface of the exchanger. They applied this approach to the heat exchangers of their ORC. This algorithm, however, seems more suitable for modeling an exchanger whose surface is known.

A similar but not very detailed approach has been used in Ref. [6] for tubular bundle heat exchangers. The Distributed Modeling method is first used to subdivide the evaporator into three consecutive segments, delineating the different states of the working fluid. At each segment, the efficiency and the heat power exchanged are calculated. The thermal powers of the three segments are then summed to find the thermal power exchanged by the two fluids in the evaporator or in the condenser.

The LMTD method also subdivides the heat exchanger in three parts according to the state of the fluid. Lecompte et al. [7] use LMTD method to divide their evaporator in three zones. They obtained a system of 8 equations with 8 unknowns that they solve using "Trust Dogleg Region" algorithm. They use similar approach to model their condenser for which the complete calculation procedure is inspired by Bell et al. [8] and Stewart et al. [9].

Quoilin et al. [10] use a totally different approach. Based on the LMTD method, they proposed a designing algorithm for the evaporator and the condenser that are plate exchangers. The algorithm is based on the choice of a pinch, a pressure drop and an arbitrary plate width. An iterative process, based on the variation of the width of the plate, ends the algorithm when the pressure drop calculated from the formulas is equivalent to the one chosen arbitrarily.

Manente et al. [11] expressed the thermal power received at the evaporator and the heat rejected at the condenser as a function of the heat transfer coefficient, the surface of the heat exchangers and the LMTD. In view of the variation of the hot reservoir temperature, an off-design method is used to describe the transient behavior of the tube heat exchangers and evaluate these various mentioned power sources; in case of variation of the hot reservoir temperature, the new heat transfer coefficient of the heat exchanger is obtained as a function of the new and old mass flow rates and the heat transfer coefficient of the nominal point. That is to say the thermal power received or rejected by the heat exchangers is obtained by the new overall heat transfer coefficient.

In the case of transient regime operation, heat exchangers are modeled by Finite Volume method or Moving Boundary model. Regarding the Finite Volume method in the case of ORC modeling, Bamgbopa and Uzgoren [5] used this approach to resolve the equation related to the energy of the fluid across the heat exchanger. Since only the thermodynamic properties of the two fluids at the inlet of the countercurrent exchanger are known, the differential equations of each node are solved from the inlet of the refrigerant to its outlet. The equations of the finite volumes of the working fluid are described using their enthalpy; those of the heat transfer fluid and the wall separating the two fluids are described based on their temperatures.

The Moving Boundary Model is a transient model that helps in describing the process of evaporation, condensation and steam generation. As a model with a reduced number of variables, it represents an important tool for the determination of the control parameters. However, it presents a drawback: the number of equations increases when the operating conditions of the exchangers increase [12].

The Moving Boundary Model divides the heat exchanger into three zones delimited by the state of the fluid as it is done with the ε-NTU and LMTD methods. The idea of a model is to follow dynamically the lengths of the different zones [12] according to the variation of the hot stream parameters.

The model is governed by three main equations that are the mass balance, the energy balance and the differential energy balance at the wall of the heat exchanger. These equations are reduced to one dimension following the assumptions made during the modeling of heat exchangers. Zhang et al. [13], Wei et al. [14] and Zhang et al. [15] use Moving Boundary Model to describe the operation of heat exchangers of the ORC machines used for the recovery of fatal heat.

Wei et al. [14] make a comparison of the moving boundary model with a new discretization technique; this discretization approach differs from the others in this way that it introduces an overall average amount of momentum balance through the exchanger channel instead of a local momentum balance through the exchanger channel or the heat exchanger channel or local momentum balances for each cell. It emerged that the moving boundary model is less complex than the discretization technique used, because it has fewer variables and a higher computation speed.

In the case of concentrated solar power project in Burkina Faso, an ORC machine is used for electricity generation. In order to describe its behavior and search for its optimal operating conditions, a mathematical model is used to define its operating process. Assuming that plant integrates a two-tank storage system to provide the heat transfer fluid with constant temperature and mass flow, the evaporator, which is the most critical component for the heat transfer is modeled in stationary regime using the approach presented by Vargas et al. [2] while the condenser is described by a pinch point model. As we do not know the characteristics of the turbine and the pump, they were modeled by their isentropic efficiency.

3. Steady State Modeling of the System

According to its description, the power block of CSP4Africa is an ORC system provided by the manufacturer. It includes an evaporator, a recuperator, a condenser, a turbine, a pump and a storage tank as shown in Fig. 2.

The three heat exchangers are of plate type from the manufacturer SWEP. The pump is a G25 Series pump provided by Hydra-Cell. The characteristics of the turbine are kept confidential by the manufacturer.

In this section, a mathematical model is developed to describe the performance of the ORC system in various operating conditions. Since CSP4Africa integrates a storage system, the inlet temperature of the heat transfer fluid is assumed to be constant;

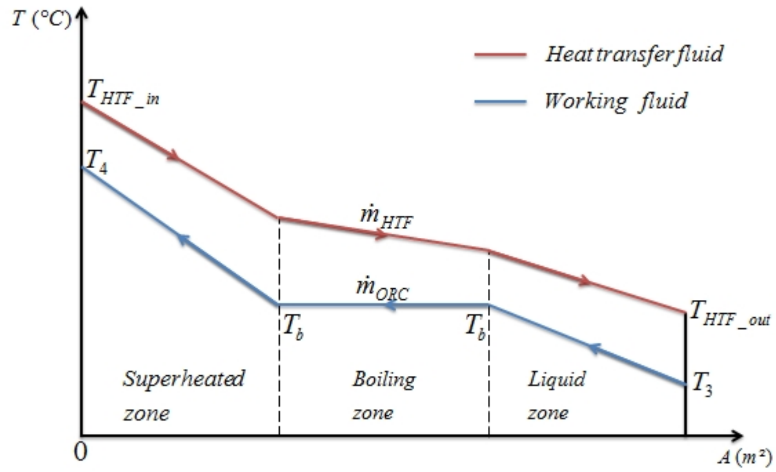

Fig. 3 Temperature distribution of the two fluids across the evaporator.

therefore, ORC system will operate in steady state regime. The model is then intended to be a stationary model.

Each component of the system is described by a mathematical model, which accounts for its characteristics. Finally, the models of all components are interconnected to obtain the model of the whole system.

3.1 Model of the Evaporator

It is considered as the most critical component of the system since it gives to the system the heat received from the hot reservoir. In the frame of this work, and considering that the working fluid (Novec 649) undergoes phase change in the component, it is important to consider all the states of the refrigerant across the evaporator.

A model based on ε-NTU approach described by Vargas et al. [4] is then used. The approach subdivides the heat exchanger in three zones according to the state of the refrigerant: subcooled region, boiling region and superheated region. Fig. 3 shows the different zones of the evaporator.

The inlet parameters of the evaporator model are the inlet temperature of the heat transfer fluid, the inlet temperature of the refrigerant, the evaporating pressure of the refrigerant and the characteristics of the heat exchanger.

As the mass flow rates of the heat transfer fluid and the refrigerant are unknown parameters, a mass flow rate ratio is introduced in the model to assess its impact on the performance of the system.

An iterative procedure is integrated in the model and ends when the sum of the areas of the states of the refrigerant equals to the total area of the heat exchanger.

3.2 Model of the Recuperator

The function of the recuperator in such a system is to increase the temperature of the refrigerant at the outlet of the pump before entering the evaporator and thus decrease its superheated vapor temperature at the outlet of the turbine. This process helps in increasing the energy efficiency of the thermodynamic cycle.

In the case of this study, the recuperator exchanges heat between the refrigerant at superheated state and its subcooled liquid state as shown by Fig. 4. It is assumed that no phase change happens in the component.

The application of the first to the recuperator yields Eq. (1).

$$h_3 - h_2 = h_5 - h_6 \quad (1)$$

Knowing the temperatures of the fluid at the outlet of the pump and the turbine, the effectiveness of the

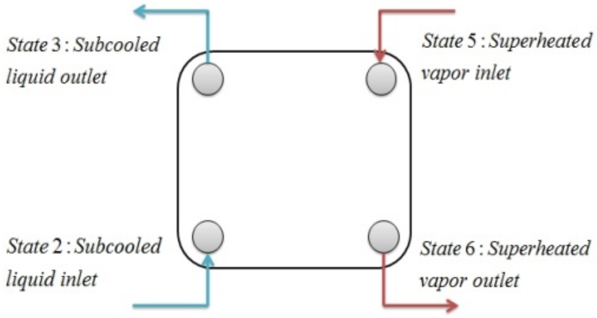

Fig. 4 Schematic view of the recuperator.

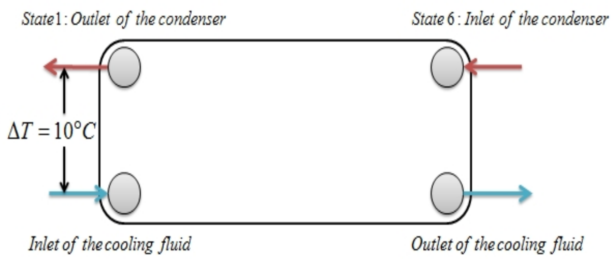

Fig. 5 Schematic view of the condenser.

recuperator may be expressed by Eq. (2).

$$\varepsilon_{RC} = \frac{T_3 - T_2}{T_5 - T_2} \quad \text{if } C_{p23} \le C_{p56}$$
$$\varepsilon_{RC} = \frac{T_5 - T_6}{T_5 - T_2} \quad \text{if } C_{p56} \le C_{p23} \tag{2}$$

where C_{p23} is the average specific heat capacity of the working fluid between 2 and 3, and C_{p56} is the average specific heat capacity of the working fluid between 5 and 6.

From the manufacturer website [16], a software for calculating performance of heat exchangers is used to simulate the operating process of the recuperator and determine its effectiveness, depending on the operating conditions.

3.3 Model of the Condenser

In order to simplify the modeling of this component, a pinch point approach is applied. A temperature difference is assumed between the outlet of the refrigerant and the inlet of the cooling fluid as described in Fig. 5.

Assuming an effectiveness of 0.9, the outlet temperature of the cooling fluid is then calculated by Eq. (3).

$$T_{CF\_out} = T_{CF\_in} + \varepsilon_{CD} \cdot (T_{CF\_in} - T_6) \tag{3}$$

Since their characteristics are kept confidential by the manufacturer, the electromechanical components such as pump and turbine are modeled by their isentropic efficiencies. The expressions of these isentropic efficiencies are respectively given by Eqs. (4) and (5).

$$\eta_{is\_TB} = \frac{\Delta h_{irr}}{\Delta h_{rev}} = \frac{h_5 - h_4}{h_{5'} - h_4} \tag{4}$$

$$\eta_{is\_PP} = \frac{\Delta h_{rev}}{\Delta h_{irr}} = \frac{h_{2'} - h_1}{h_2 - h_1} \tag{5}$$

3.4 Efficiency of the ORC System

The efficiency of the overall system is calculated as a ratio of the net mechanical work of the system by the heat input at the evaporator. It is given by Eq. (6).

$$\eta_{cycle} = \frac{W_{TB} - W_{PP}}{Q_{HTF}} \tag{6}$$

CSP expected efficiencies are low in thermal plants for electricity generation. The effort to be made, both for this new plant and for others around the world, will consist in improving the performance of the overall cycle. However, it should be known that efficiencies of thermal engines are always limited. This limitation is primarily theoretical due to the 2nd law of thermodynamics, but it is also practical due to the intrinsic efficiencies of the devices that make up the cycle. This is why energy and exergy analysis are needed for the improvement of the cycle overall efficiency.

Energy analysis deals only with quantities of energy, meaning 1st law of thermodynamics. Only exergy analysis deals with quality of energy. Energy analysis alone cannot show how each individual device of the cycle can be improved. Exergy analysis tells us instead what part of the useful energy is involved and how far we can be from the efficient use of this useful fraction of energy. Exergy also allows for a more accurate comparison of energy systems. A solar power plant like this one with its low thermal levels with a maximum efficiency as low as 20% can be more efficient in term of efficient use of exergy than coal plant whose efficiency is close to 40%. These are some of the questions that the last part of this paper attempts to answer.

In this paper, the reference state of exergy or dead state is:

$P_0 = 101.3$ kPa; $T_0 = 298$ K

In a Rankine cycle, each of the four main devices of the cycle operates as a control volume (open system). We have therefore considered the following calculation rules for the specific exergy of the fluid:

$$\psi = (h - h_0) - T_0 \cdot (s - s_0) \tag{7}$$

When the specific heat is constant, the exergy of a liquid without phase change is reduced to:

$$\psi = c_p \cdot \left[(T - T_0) - T_0 \cdot \ln(T/T_0)\right] \tag{8}$$

The exergy of a flow system with *m* flow rate is:

$$X = \dot{m} \cdot \psi \tag{9}$$

The variation of exergy after a transformation is then:

$$\Delta X = \dot{m} \cdot \Delta\psi \tag{10}$$

$$\Delta X = \dot{m} \cdot \left[(h_f - h_i) - T_0 \cdot (s_f - s_i)\right] \tag{11}$$

The exergy balance of any organ in the cycle obeys the equation:

$$\Delta X = X_e - X_i - X_d \tag{12}$$

In a steady state this equation becomes

$$X_e = X_i + X_d \tag{13}$$

The exergy efficiency of the turbine in these conditions is defined by

$$\eta_{ex,TB} = \frac{W_u}{W_{rev}} \tag{14}$$

$$\eta_{ex,TB} = \frac{h_{out} - h_{in}}{\psi_{out} - \psi_{in}} \tag{15}$$

As for the pump:

$$\eta_{ex,PP} = \frac{W_{rev}}{W_u} \tag{16}$$

$$\eta_{ex,TB} = \frac{\psi_{out} - \psi_{in}}{h_{out} - h_{in}} \tag{17}$$

For a heat exchanger such as: a condenser, a steam generator, or a recuperator this efficiency will be:

$$\eta_{ex} = 1 - T_0 \cdot \left(\frac{S_{gen}}{\dot{m}_{HTF} \cdot (\psi_{in} - \psi_{out})}\right) \tag{18}$$

$$\eta_{ex} = \frac{\dot{m}_{CF} \cdot (\psi_{CF,out} - \psi_{CF,in})}{\dot{m}_{HTF} \cdot (\psi_{HTF,out} - \psi_{HTF,in})} \tag{19}$$

which means: exergy recovered from cold fluid over exergy available on hot fluid.

The energy and exergy balances of the four devices of the cycle can be calculated as follows.

3.5 Isentropic Efficiency of Pump: η_{PP}

Energy losses

$$W_{PP,irr} - W_{PP,rev} = \dot{m} \cdot (h_{out} - h_{in}) \cdot \left(\frac{1}{\eta_{PP}} - 1\right) \tag{20}$$

$$W_{PP,irr} - W_{PP,rev} = \dot{m} \cdot (h_{2'} - h_2) \tag{21}$$

Exergy destruction

$$\Delta X = \dot{m} \cdot (\psi_{2'} - \psi_2) \tag{22}$$

3.6 Isentropic Efficiency of Turbine: η_{TB}

Energy losses

$$W_{TB,rev} - W_{Tb,irr} = \dot{m} \cdot (h_{out} - h_{in}) \cdot (1 - \eta_{TB}) \tag{23}$$

$$W_{TB,rev} - W_{TB,irr} = \dot{m} \cdot (h_{5'} - h_5) \tag{24}$$

Exergy destruction

$$\Delta X = \dot{m} \cdot (\psi_{5'} - \psi_5) \tag{25}$$

3.7 Heat Exchangers: Steam Generator, Condenser, and Recuperator

Energy losses

$$\Delta E = \dot{m}_{HTF} \cdot (h_{HTF,in} - h_{HTF,out}) - \dot{m}_{CF} \cdot (h_{CF,in} - h_{CF,out}) \tag{26}$$

Exergy destruction

$$\Delta X = \dot{m}_{HTF} \cdot (\psi_{HTF,in} - \psi_{HTF,out}) - \dot{m}_{CF} \cdot (\psi_{CF,in} - \psi_{CF,out}) \tag{27}$$

4. Simulation and Results

The model describing the behavior of the ORC system is implemented in Matlab. The inlet parameters of the model are the evaporating pressure of the refrigerant, the inlet temperature of the heat transfer fluid, the inlet temperature of the cooling fluid, the isentropic efficiencies of the turbine and pump and the characteristics of the heat exchanger. The efficiency of the cycle is then simulated. To evaluate the importance of these simulations, parameters that impact the most the system efficiency are chosen among the inlet parameters.

The results presented in this section show the trends of the ORC system efficiency as a function of the evaporating pressure, the ambient temperature and the isentropic efficiency of the turbine.

Fig. 6 shows the overall trend of the efficiency of the whole system. The curves were obtained by considering the isentropic efficiency of the turbine as a parameter. Results show that the efficiency of the system increases with the evaporating pressure and also with the isentropic efficiency.

Considering that the temperature of the heat transfer fluid at the inlet of the evaporator is constant for all the simulations, it can be concluded that operating at higher pressure ensures a better thermal match between the hot reservoir and the refrigerant in the evaporator.

Fig. 7 highlights the overall efficiency of the system as a function of the ambient temperature.

The isentropic efficiency of the turbine is varied from 0.6 to 0.9. Results show that the isentropic efficiency of the turbine has a huge impact on the efficiency of the system, even though latter decreases with the increase of the ambient temperature. The trend of these efficiencies is realistic since the inlet temperature of heat transfer fluid is maintained constant during the simulation.

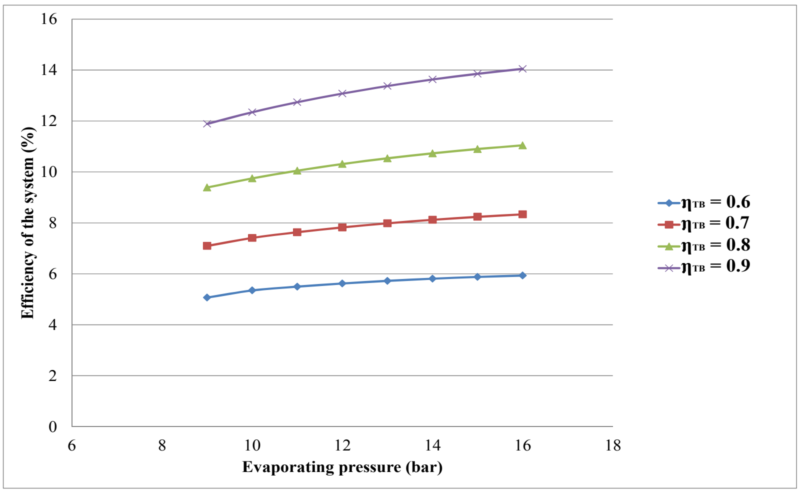

Fig. 6 Trend of the system efficiency as a function of evaporating pressure.

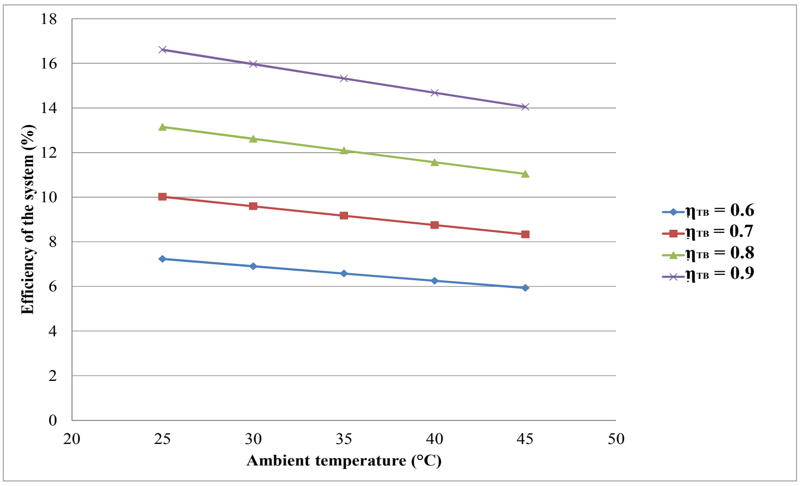

Fig. 7 Trend of the system efficiency as a function of ambient temperature.

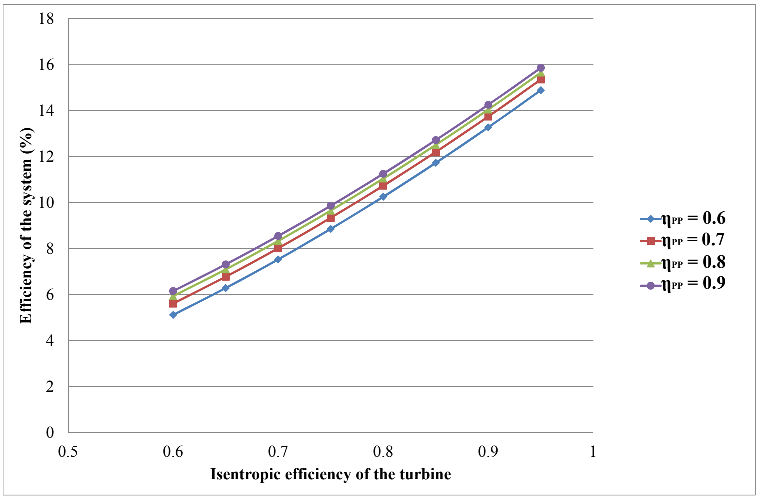

Fig. 8 Trend of the system efficiency as a function of the isentropic efficiency of the turbine.

In Fig. 8, the efficiency of the system is computed against the isentropic efficiency of the turbine. The isentropic efficiency of the pump is used as a parameter and varied from 0.6 to 0.9.

The effect of the pump isentropic efficiency on the system efficiency is very weak; a difference of 1% is observed between the system efficiencies when considering isentropic efficiency of the pump at 0.6 and 0.9.

For an isentropic efficiency of the turbine of 0.85, a pump efficiency of 0.80, an evaporative pressure of 1.6 MPa and ambient temperature of 45 °C, the efficiency of the cycle is 12.51%. For a 100 kW thermal power it leads to 12.51 kW power generation.

4.1 Exergy Analysis and Discussions

Table 3 gives the physical properties and functions needed for the analysis.

Under these conditions, the overall efficiency of the cycle was found to be 12.51%. Figs. 9 and 10 below summarize the energy and exergy balances achieved. Fig. 9 gives this balance in absolute value and Fig. 10 in relative value. Fig. 9 shows that the largest energy losses are in the order: recuperator, condenser, and steam generator. These losses are lower when considering exergy. The recuperator can be incriminated with the greatest losses in quality as well as quantity of energy. The losses of the turbine and the pump are negligible. The exergy losses of these mechanical parts are also negligible.

These two figures take stock of the waste of energy and exergy destruction and show especially in which sector an improvement effort is reasonably possible. According to the two figures, the recuperator is the first organ to be incriminated because of its high energetic loss and exergy destruction.

The analysis of this 100 kW thermal plant, with 12.51% overall efficiency shows that the energy losses are down to 2.8 kW for the turbine and only 160 W for the pump. For heat exchangers these losses are higher and up to 34 kW for the recuperator, 26 kW for the condenser and only 2.8 kW for the steam generator. It must be noted that the high energy losses of the recuperator include the part of unrecovered energy of de-superheating and condensation of the working fluid. For the condenser these losses are due to the convection and radiation losses from the body of this device.

The overall exergy destruction of the pump and turbine together is close to 1 kW only. It is the recuperator that destroys the maximum useful energy (12 kW) followed by the condenser (7 kW) and the steam generator (5 kW). These figures are to be compared to the 12.51 kW power generation of the plant.

The contribution of pumps and turbines to improve the overall performance of the cycle remains marginal.

Table 3 Novec 649 properties at various stages of the cycle.

State	*P* (bar)	*T* (K)	*H* ($kJ \cdot kg^{-1}$)	*S* ($kJ \cdot kg^{-1} \cdot K^{-1}$)	Ψ ($kJ \cdot kg^{-1}$)
1	1.24	328.15	260.84	1.2026	1.6074
2	16	328.77	261.82	1.2026	2.5874
2'	16	328.99	262.06	1.2034	2.5889
3	16	396.96	341.67	1.423	16.7251
4	16	480.74	483.44	1.7483	61.5069
5	1.24	441.85	455.49	1.7483	33.5569
5'	1.24	446	459.69	1.7577	34.9543
6	1.24	364.09	380.08	1.5608	14.0501
Dead State	1.0133	298.15	227.42	1.0959	0

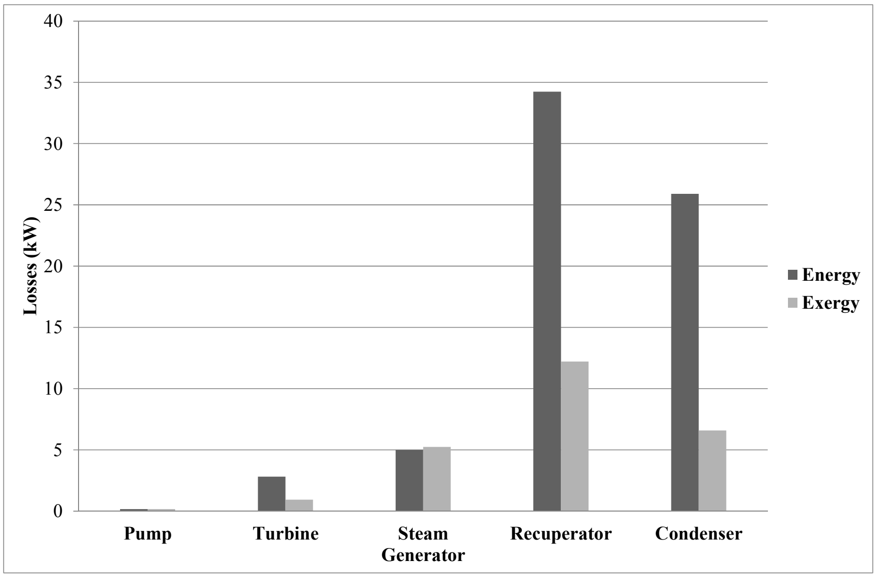

Fig. 9 Energy losses and exergy destruction in kW.

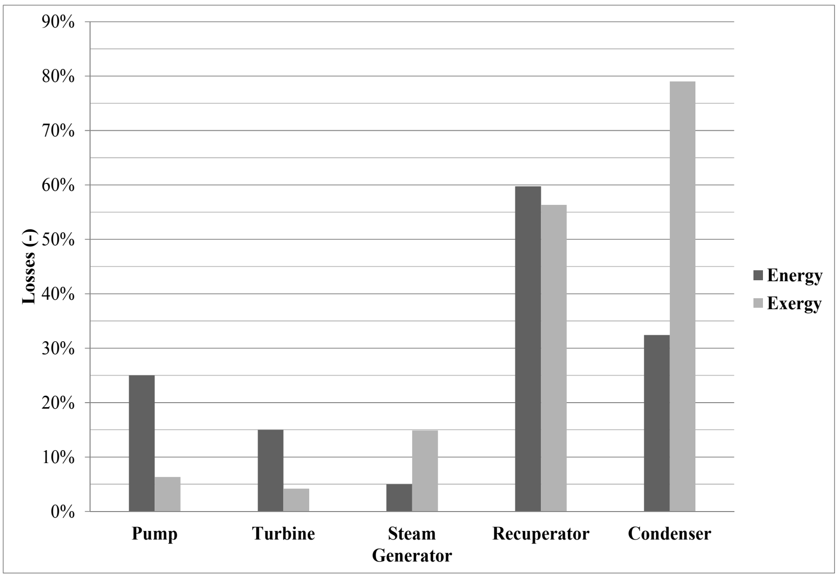

Fig. 10 Energy losses and exergy destruction in %.

In this study all purely thermal organs have high energy losses and therefore require special attention like insulation in order to reduce their losses of useful energy. The magnitude of the losses as compared to the power generation shows that any improvement of the components of the cycle will tremendously improve the overall efficiency of the ORC cycle.

5. Conclusions

This paper is a contribution to the thermodynamic analysis of organic Rankine cycles, which are receiving a growing interest because they allow the valorization of low grade temperature heat sources. First, a review on the modeling of ORC has been made. It appears that there are very few works on the modeling of ORC and the mostly used modeling methods are usually based on relatively simplified models. Based on this review, a model of ORC has been developed and applied to the 100 kW_{th} ORC machine of the CSP4Africa micro-CSP plant. It comes out that it is the recuperator that destroys the maximum useful energy (12 kW) followed by the condenser (5 kW) and the condenser (7 kW) for a net mechanical output of 12.51 kW. The operating conditions and the performance of the cycle will be optimized after model validation by experimental data from the machine that is presently undergoing tests at the Fondation 2iE.

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