Journal of Energy and Power Engineering 19 (2025) 1-9
doi: 10.17265/1934-8975/2025.01.001

Thermodynamic Modeling and Energy Analysis of Organic Rankine Cycle with a Power Capacity of 60 kWe

Serigne Thiao[1], Awa Mar[2], Omar Sylla[1], Mamadou Sow[1], Diouma Kobor[1] and Issakha Youm[3]

1. Laboratory of Chemistry and Physics of Materials (LCPM), Department of Physics, Assane Seck Universty of Ziguinchor, BP 523 Ziguinchor, Senegal

2. Pole of Sciences and Technologies, Higher School of Engineering Sciences and Techniques, Amadou Moktar MBOW University of Dakar (UAM), BP 45927 Dakar, Senegal

3. Laboratory of Solar Energy, Materials and Systems, Departement of Physics, Cheikh Anta DIOP University, BP 5005 Dakar, Senegal

Abstract: The objective of this study is to establish a thermodynamic model of an ORC (organic Rankine cycle) for power electricity. A case study was proposed in an area where direct solar irradiation is abundant. The number of heliostats used in the system as a function of the DNI (direct normal irradiation) was studied. The efficiencies of ORC and receiver, the power of turbine, pump, evaporator and receiver as a function of thermodynamic parameters such as temperature, pressure at the level of different components of the system are studied. The results obtained show that the number of heliostats used decreases when the DNI increases. For a DNI of 700 W/m^2 to 500 W/m^2, the number of heliostats goes from 280 to 60. ORC efficiency and turbine power increase respectively from 11% to 22% and from 20 kW to 50 kW when the condenser temperature decreases. Also it is noted an increase of receiver efficiency when evaporator temperature increases.

Key words: ORC, DNI, heliostat, receiver, turbine.

Nomenclatures

$\dot{Q}$	Power [kW]
$\dot{W}$	Power [kW]
A	Area [m^2]
h	Enthalpy [kJ/kg]
$\dot{m}$	Mass flow[kg/s]
η	Efficiency [-]
DNI	Direct normal irradiation [W/m^2]
N	Number of heliostat [-]
ORC	Organic Rankine cycle

Indices

air	Air
$fied$	Fied
hel	Heliostat
rec	Receiver
$loss$	Loss
p	Pump
net	Net output
in	Inter
$Turb$	Turbine

Corresponding author: Serigne Thiao, PhD, associate professor, research fields: renewable energy and energy conversion.

1. Introduction

Since 1973 world energy consumption has been dramatically increased and global demand for energy has increased significantly due to the continued increase of world population and the fast development of industry [1, 2]. According to Li et al. [3], current world electricity production comes mainly from the combustion of fossil fuels which cause serious problems in terms of air pollution, excess CO_2 emissions and depleting energy resources. Most of the increase in average temperature of planet observed since the middle of 20th century is most likely attributable to the increase of concentrations of anthropogenic greenhouse gases. Between 1997 and

2004, an increase of 70% of greenhouse gas emissions was noted, mainly due to energy production and transport leases. In addition, the price of fossil fuels is doubled between 1996 and 2010. All of these conditions have led to the implementation of research programs for innovative solutions for new electricity power plants. It is therefore urgent to generate more energy from low enthalpy heat sources such as solar thermal, biomass, geothermal [4-7]. Thermal energy is the largest source of energy in the world [8]. The use of alternative clean energy sources such as solar, geothermal, biomass, and the development of efficient system of energy conversion are part of the possible techniques to tackle the problem of energy shortage and, at the same time, mitigate the negative environmental impacts caused by the excessive consumption of fossil fuels [2, 9, 10]. In Africa particularly in Senegal where one have abundant solar radiation, energy production methods using solar energy such as solar photovoltaics, are extremely popular, with many implementations already successful. There are other methods of using low temperature heat sources such as solar thermal energy for electricity production. There are already many studies in the literature on power plants using low pressure and low temperature heat [11]. The awareness of the use of low enthalpy heat sources has captivated energy researchers. The ORC (organic Rankine cycle) system is a solution for the production of electricity from low to medium enthalpy heat sources in the range of 80 to 350 ℃. It is a technology that exploits low enthalpy heat. The ORC is suitable for a wide range of applications, including low enthalpy geothermal sources, waste heat from industrial processes, biomass and concentrated solar thermal. Due to the use of low boiling temperature organic fluids as the heat transfer fluid instead of water, the ORC is characterized by its ability to efficiently convert low enthalpy heat into more economical useful work compared to conventional Rankine cycle of steam, particularly in applications ranging from a few kW to 1 MW [2, 12, 13]. In terms of working fluid, ORC uses organic refrigerants with low condensation and low boiling point; allowing it to use a low heat level with low pressure and low temperature [11, 14]. These organic fluids have interesting and advantageous characteristic properties compared to a water/steam system [15, 16]. According to Chen et al. [17], ORCs and Kalina cycles have demonstrated their ability to efficiently exploit such sources with an advantage due to their less complexity and the less maintenance required. The ORC is an ideal solution when the size of the application is too small as in the case of our study. A computer model is developed in the EES (Engineering Equation Solver) software to model the system.

2. Methodology

2.1 Thermal Solar Field Coupled to the ORC

The combined system is composed of two subsystems: solar field and ORC. The solar field is composed of parabolic solar collector. The solar radiation received by the solar field is transformed into heat. The primary working fluid recovers heat at the receiver and passes into evaporator ORC. A good part of this heat is transferred to the organic working fluid R245fa. The latter is brought to the enthalpy h_{7ORC} then driving the turbine. The mechanical power is then converted into electrical power by the alternator. The steam leaving the turbine is directed to the condenser where it is cooled. The organic fluid returns to the evaporator ORC at temperature T_3 through pump 2 and thus the ORC system starts again. Pump 1 return the primary fluid to the receiver at temperature T_9 and thus system begins again. A schematic diagram of a basic ORC system with a solar thermal field is given by Fig. 1.

Technologies such as linear Fresnel concentrators are particularly suitable for ORC because they require low investment costs and they work at lower temperatures [18, 19]. ORCs appear to be a promising technology for reducing investment costs on a small scale: they can work at lower temperatures and the total installed power can be reduced down to the kilowatt scale. The ORC is a well-known thermodynamic cycle that is practically used to convert heat into work [20].

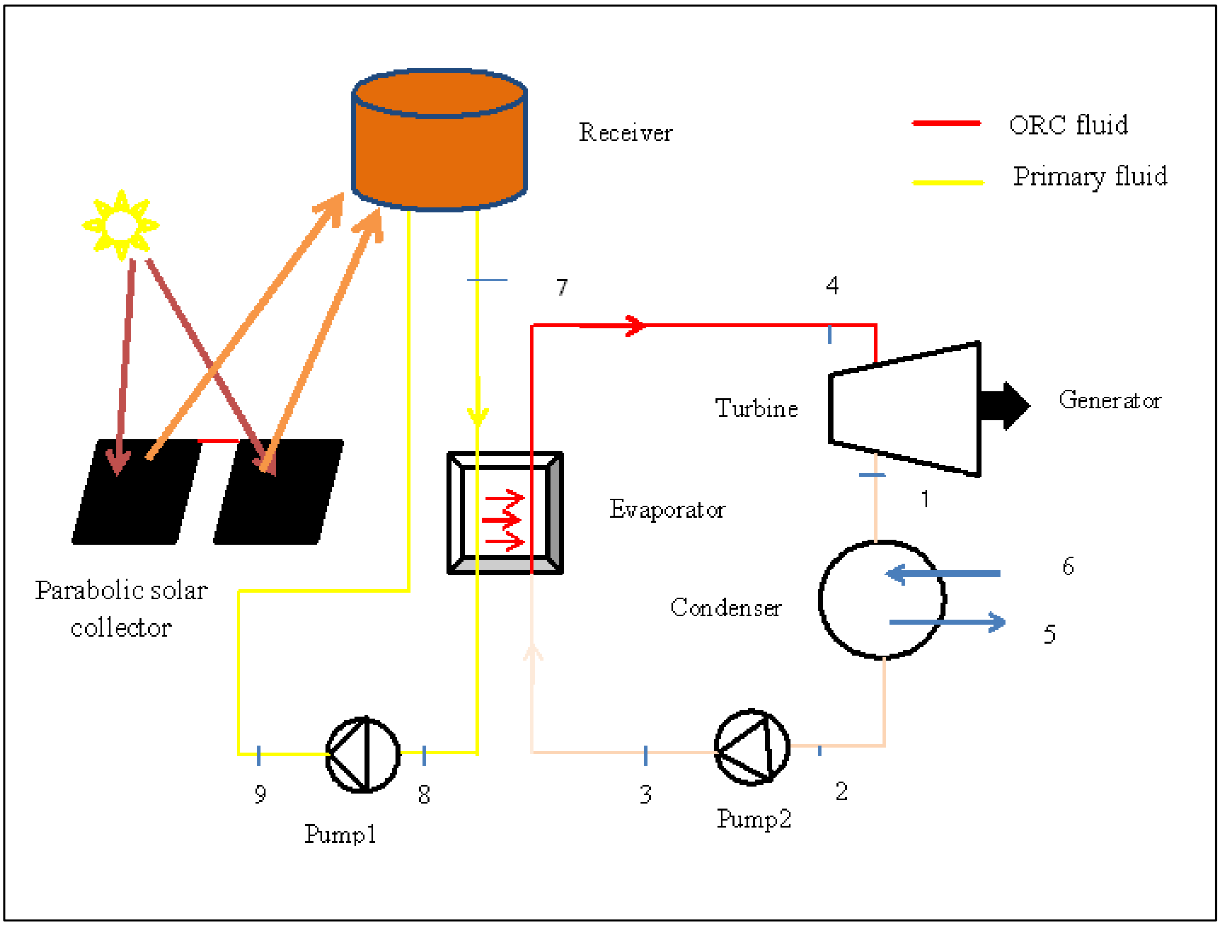

Fig. 1 Schematic diagram of basic ORC coupled with solar thermal field.

Kane et al. [21] studied the coupling of linear Fresnel solar collector with an ORC of 9 kWe in cascade. An overall efficiency (solar to electricity) of 7.74% was obtained, with a collector efficiency of 57%.

2.2 Thermodynamic Analysis

The efficiency of the heliostat field determines how much solar thermal energy reaches the receiver through the DNI (direct normal irradiation) of the Sun. A portion of the total solar irradiation arrived onto the heliostats was lost to the environment, leaving the remainder for the receiver in the form of usable heat. Thus the power received by the receiver is given by [16].

2.2.1 Energy Balance of the Solar Field Subsystem

2.2.1.1 Solar Field

Solar irradiation received by the solar field is converted into heat. The fraction absorbed by the receptor is given by Eq. (1).

$$\dot{Q}_{rec,in} = \eta_{fied}.\dot{Q}_{sun} = \eta_{fied}.DNI.A_{hel}.N_{hel} \quad (1)$$

However, there are always losses in the atmosphere due to conduction, convection and reflection. This power lost at the solar field level is noted $\dot{Q}_{rec,loss}$. The remaining power $\dot{Q}_{rec,net}$ is transferred to the working fluid or heat fluid transfer, i.e. the primary fluid. Eq. (1) then becomes:

$$\dot{Q}_{rec,in} = \dot{Q}_{rec,net} + \dot{Q}_{rec,loss} = \dot{m}_{air}(h_7 - h_8) + \dot{Q}_{rec,loss} \quad (2)$$

Receiver efficiency is given by:

$$\eta_{rec} = \frac{\dot{Q}_{rec,net}}{\dot{Q}_{rec,in}} \quad (3)$$

The ratio between the net power and the energy available on the heliostat field was used to define the overall energy efficiency of the solar power plant studied [16, 19-23].

$$\eta_{en,plant} = \frac{\dot{W}_{net}}{\dot{Q}_{Sun}} \quad (4)$$

2.2.1.2 Pump 1

The primary fluid loses its heat at the outlet of evaporator ORC. It is pumped towards the solar field by pump 1. The balance equation is given by:

$$\dot{W}_{P1} = \dot{m}_1(h_9 - h_8) \quad (5)$$

2.2.1.3 Receiver

The heat received by the receiver through the heliostats is considered to be the useful part of solar radiation. Thus, Eqs. (1) and (2) of the energy balance at the receiver level can be rewritten as:

$$\dot{Q}_{rec,in} = \dot{m}_{air}(h_7 - h_8) + \dot{Q}_{rec,loss} \quad (6)$$

2.2.2 The Energy Balance at the Level of the ORC Subsystem

2.2.2.1 Evaporator

In the evaporator, the heat of primary fluid coming from the receiver is transferred to the organic fluid. The energy balance is given by Eq. (7).

$$\dot{Q}_{evap} = \dot{m}_{air}(h_7 - h_8) + m_{ORC,fluid}(h_4 - h_3) \quad (7)$$

2.2.2.2 Turbine

The power of the turbine and its efficiency are given by Eqs. (8) and (9).

$$\dot{W}_{Turb} = \dot{m}_{ORC,fluid}(h_4 - h_1) \quad (8)$$

$$\eta_{Turb} = \frac{h_4 - h_1}{h_4 - h_{1S}} \quad (9)$$

2.2.2.3 Condenser

In the condenser, the organic fluid leaving the turbine is condensed. The heat contained in the organic fluid is dissipated into the ambient environment. This heat dissipation is given by Eq. (10).

$$h_2 - h_5 = h_1 - h_6 \quad (10)$$

2.2.2.4 Pump 2

Liquid organic fluid is sent into the evaporator ORC by pump 2. Energy balance and efficiency are given by Eqs. (11) and (12).

$$\dot{W}_{P2} = \dot{m}_2(h_3 - h_2) \quad (11)$$

$$\eta_{p2} = \frac{h_{3S} - h_2}{h_3 - h_2} \quad (12)$$

2.3 Case Study

A case study was carried out in a village in eastern of Senegal. In this rural area the electricity network is even not available. Likewise the price of the electricity is unaffordable. Using solar energy as energy source could be an alternative solution for Senegal because the country is located in a high solar density area. In this case of study, the power of the power plant depends on the maximum consumption power which is determined from the energy needs of the village. This maximum power consumption is visualized through the load profile and is equal to 60 kWe.

3. Results and Discussions

After carrying out the energy analysis of the different components of the system, the results obtained using EES software will be detailed and discussed.

Fig. 2 gives ORC efficiency as a function of mass flow.

The air flow coming from the receiver is fixed constant and the flow rate of ORC fluid is varied. Fig. 2 shows that more the flow rate increases, the efficiency also increases. Indeed, when the flow rate is equal to 1 kg/s, the efficiency is around 27%. This is logical because for the efficiency EES takes into account the fluid flow speed and the Reynolds number. In addition, ORC uses low temperature heat sources.

Fig. 3 gives the number of solar thermal collectors as a function of the DNI.

Fig. 3 shows the variation in the number of heliostats as a function of the DNI. It is noted that the number of heliostats varies inversely as a function of the DNI, that due to when the DNI increases, the number of heliostats necessary for better functioning of ORC decreases. However, if the DNI is equal to 547 W/m^2, the number of heliostats is around 65 heliostats. This is reassuring insofar as DNI is abundant in the study area.

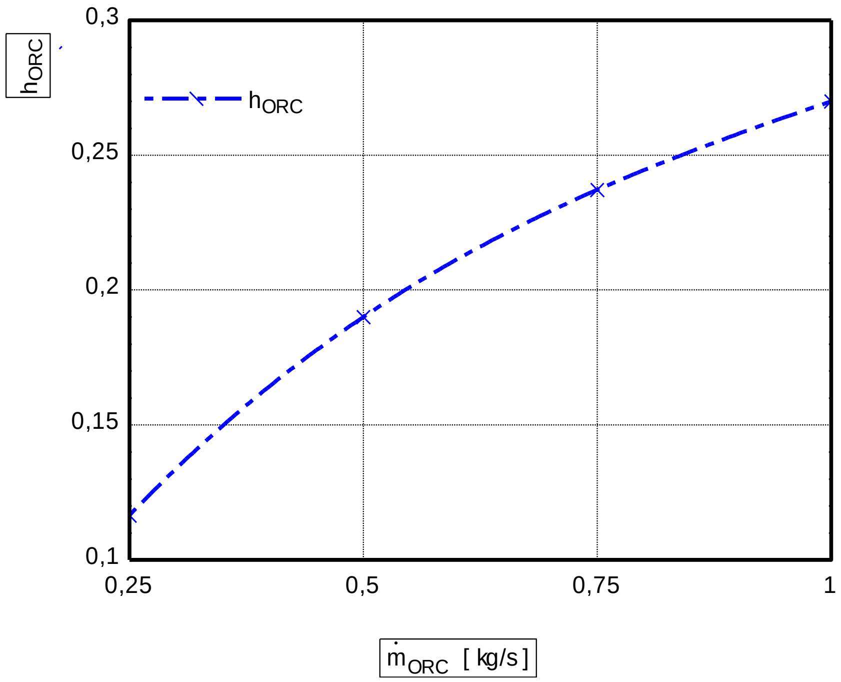

Fig. 2 ORC efficiency as a function of mass flow.

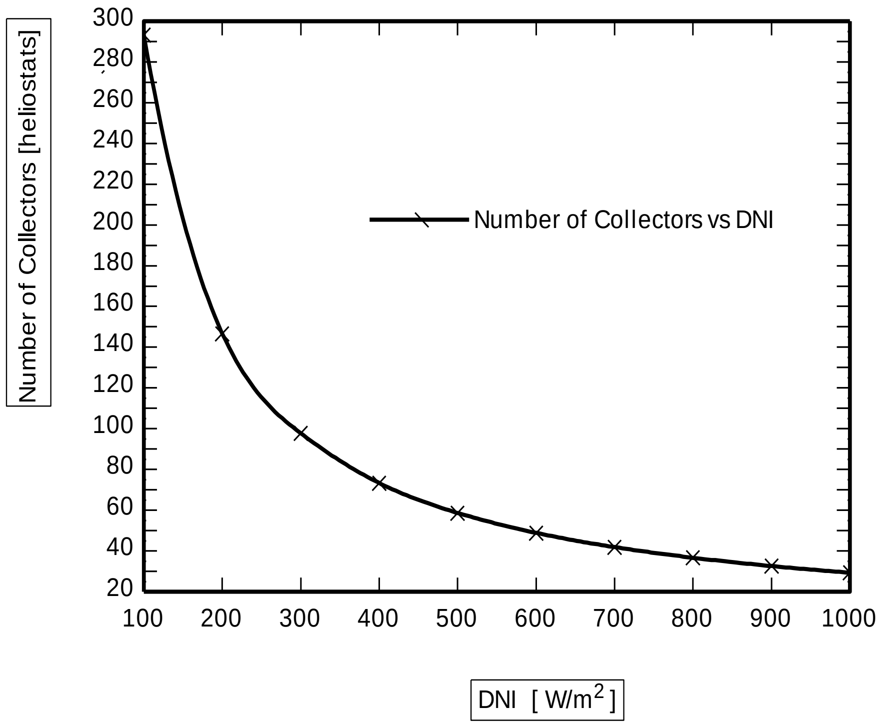

Fig. 3 Number of solar thermal collectors depending on the DNI.

Fig. 4 gives ORC efficiency and receiver efficiency as a function of air temperature at the evaporator inlet.

By fixing the air flow at 1 kg/s, Fig. 4 shows that ORC efficiency decreases when the air temperature at the evaporator inlet increases. Thus ORC efficiency can be optimized as a function of the evaporator temperature. Likewise, it is noted that receiver efficiency increases with evaporator temperature. When evaporator temperature inlet increases then $\dot{Q}_{rec,net}$ also increases. So Fig. 4 shows the evaporation temperature at which

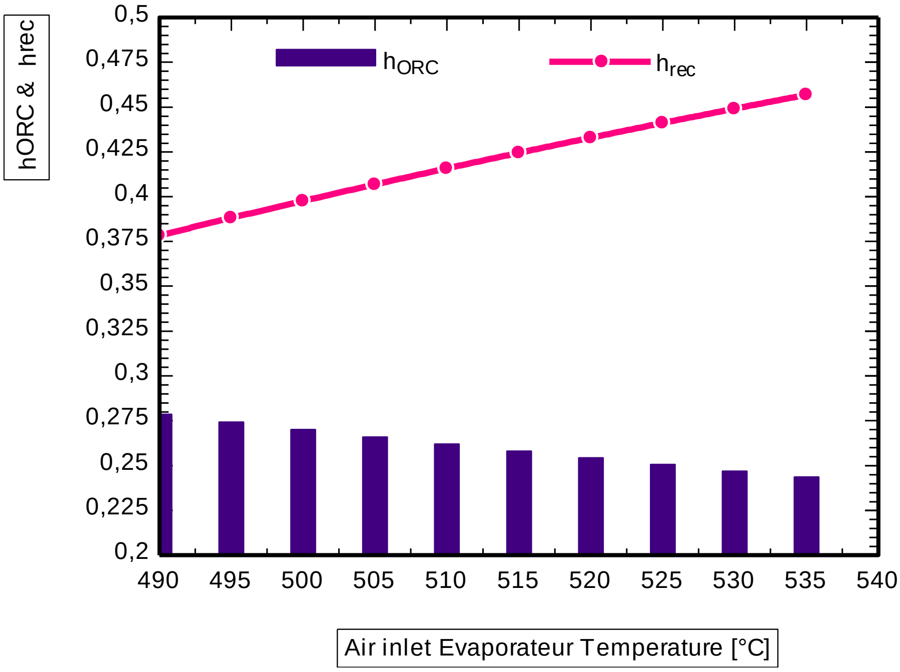

Fig. 4 ORC and receiver efficiency as a function of evaporator inlet air temperature.

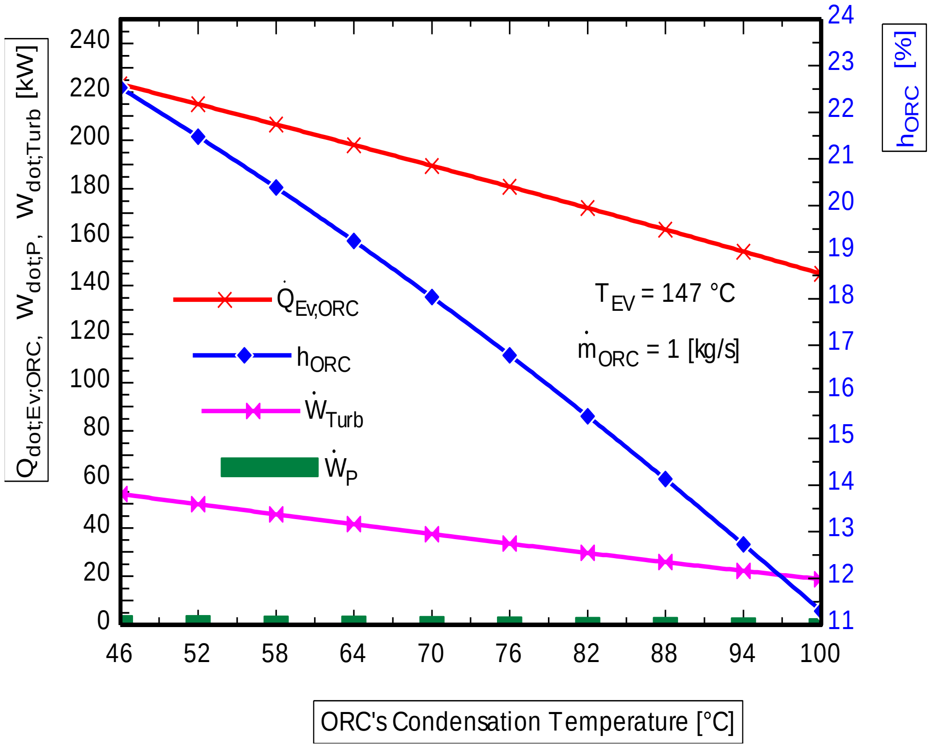

Fig. 5 Variation of pump power, turbine power, evaporator power and ORC efficiency as a function of ORC condensing temperature.

the organic Rankine cycle must operate to be optimized in power as a function of the temperature of the hot source. This result confirms the fact that ORCs allow low temperature heat recovery, i.e. below 300 ℃.

Fig. 5 gives the variation of the power of the pump, the turbine, the evaporator of the ORC and the efficiency of the ORC as a function of the condensation temperature of the ORC.

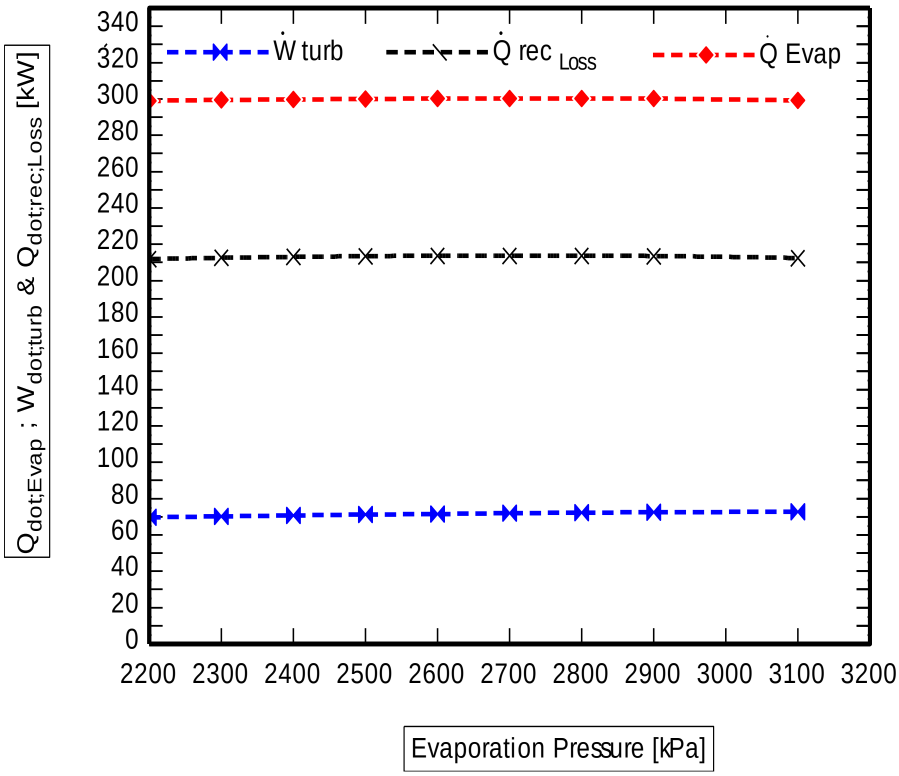

Fig. 6 Evolution of turbine power, receiver power lost and evaporator power as a function of evaporation pressure.

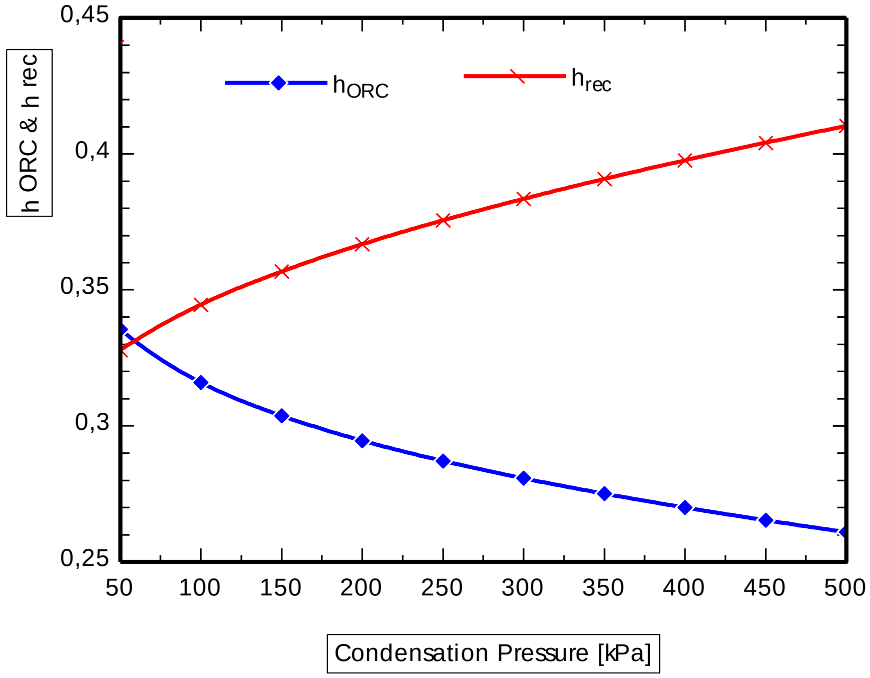

Fig. 7 Evolution of ORC efficiency and efficiency receiver as a function of the condensing pressure.

Fig. 5 shows that the pump power is almost constant for condenser temperatures below 70 ℃. ORC efficiency increases as the condenser temperature decreases. Likewise, evaporator power and turbine power decrease when the condenser temperature of ORC increases. For certain temperatures, the condenser behaves as a hot source and becomes incapable to condense the organic fluid. It is therefore necessary to cool the condenser.

Fig. 6 gives the evolution of the turbine power, receiver power and evaporator power as a function of evaporation pressure

The pressure ratio between evaporation pressure P_{evap} and the condensation pressure P_{cond} is kept constant and fixed at 8. Fig. 6 shows that if one increase P_{evap}, P_{cond} also increases but it is noted that turbine power, receiver loss power and evaporator power remain constant. So the pressure ratio has no effect on the power.

Fig. 7 gives the evolution of ORC efficiency and receiver efficiency as a function of condensing pressure.

Mass flow rates of URC fluid and air are set at 1.5 kg/s and 1 kg/s respectively. Likewise, evaporator pressure is equal to 90% of the critical pressure of fluid organic, i.e. 32 bar. Fig. 7 shows that ORC efficiency and that of receiver vary inversely. ORC efficiency is equal to 0.27 when the pressure of the steam expanded in the turbine increases to 4 bars before arriving in the condenser. On the other hand, receiver efficiency increases considerably up to 0.4, which would be due to an increase of power at the receiver level.

4. Conclusion

At the end of this study one managed to show the feasibility of the project in a country where DNI is abundant. The effects of thermodynamic parameters such as temperature and pressure were analyzed. The number of heliostats used in the system as a function of the DNI was studied. ORC efficiency and receiver efficiency, turbine power, pump power, evaporator ORC power and receiver power as a function of thermodynamic parameters at the level of the different components of the system are studied. The results obtained show that the number of heliostats used decreases when the DNI increases. For a DNI of 500 W/m^2, the number of heliostats decreases from 280 to 60. ORC efficiency and turbine power increase respectively from 11% to 22% and from 20 kW to 50 kW when the condenser temperature decreases.

References

[1] Saadon, S., and Mohd Nasir, N. A. 2019. "Modelling and Thermal Analysis of Organic Rankine Cycle with Superheater and Preheater." *Journal of Advanced Research in Fluid Mechanics and Thermal Sciences* 55 (2): 142-9.

[2] Hossin, K., Mahkamov, K., and Belgasim, B. 2020. "Thermodynamic Analysis and Sizing of a Small Scale Solar Thermal Power System Based on Organic Rankine Cycle." *J. Sustain. Dev. Energy water environ. Syst.* 8 (3): 493-506. https://doi.org/10.13044/j.sdewes.d7.0294.

[3] Li, L., Ge, Y. T., Luo, X., and Tassou, S. A. 2016. "Thermodynamic Analysis and Comparison between CO_2 Transcritical Power Cycles and R245fa Organic Rankine Cycles for Low Grade Heat to Power Energy Conversion." *Applied Thermal Engineering* 106: 1290-9.

[4] Manolakos, D., Papadakis, G., Kyritsis, S., and Bouzianas, K. 2007. "Experimental Evaluation of an Autonomous Low-Temperature Solar Rankine Cycle System for Reverse Osmosis Desalination." *Desalination* 203: 366-74.

[5] Uris, M., Linares, J. I., and Arenas, E. 2014. "Techno-Economic Feasibility Assessment of a Biomass Cogeneration Plant Based on an Organic Rankine Cycle." *Renewable Energy* 66: 707-13.

[6] Qiu, G. Q., Shao, Y. J., Li, J. X., Liu, H., and Riffat, S. B. 2012. "Experimental Investigation of a Biomass-Fired ORC-Based Micro-CHP for Domestic Applications." *Fuel* 96: 374-82.

[7] Heberle, F., and Bruggemann, D. 2010. "Exergy Based Fluid Selection for a Geothermal Organic Rankine Cycle for Combined Heat and Power Generation." *Appl. Therm. Eng*. 30: 1326-32.

[8] Ravindra Kumar, P., Ramachandra Raju, V., and Ravi Kumar, N. 2016. "Simulation and Parametric Optimisation of Thermal Power Plant Cycles." *Perspectives in Science* 8 304-36.

[9] Bao, J., and Zhao, L. 2013. "A Review of Working Fluid and Expander Selections for Organic Rankine Cycle." *Renewable and Sustainable Energy Reviews* 24: 325-42. https://doi.org/10.1016/j.rser.2013.03.040.

[10] Ozlu, S., and Dincer, I. 2015. "Development and Analysis of a Solar and Wind Energy Based Multigeneration System." *Solar Energy* 122: 1279-95. https://doi.org/10.1016/j.solener.2015.10.035.

[11] Mahlia, T. M. I., Syaheed, H., Abas, A. E. P., Kusumo, F., Shamsuddin, A. H., Ong, H. C., and Bilad, M. R. 2019. "Organic Rankine Cycle (ORC) System Applications for Solar Energy: Recent Technological Advances." *Energies* 12: 2930. doi: 10.3390/en12152930.

[12] Schuster, A., Karellas, S., Kakaras, E., and Spliethoff, H.

2009. "Energetic and Economic Investigation of Organic Rankine Cycle Applications." *Applied Thermal Engineering* 29 (8-9): 1809-17. https://doi.org/10.1016/j.applthermaleng.2008.08.016 6.

[13] Quoilin, S., Orosz, M., Hemond, H., and Lemort, V. 2011. "Performance and Design Optimization of a Low-Cost Solar Organic Rankine Cycle for Remote Power Generation." *Solar Energy* 85 (5): 955-66. https://doi.org/10.1016/j.solener.2011.02.010.

[14] Bellos, E., and Tzivanidis, C. 2018. "Investigation of a Hybrid ORC Driven by Waste Heat and Solar Energy." *Energy Convers. Manag.* 156: 427-39.

[15] Angelino, G., and Colonna di Paliano, P. 1998. "Multi-component Working Fluids for Organic Rankine Cycles (ORCs)." *Energy* 23: 449-63.

[16] Liu, B., Chien, K. C., and Wang, C. C. 2004. "Effect of Working Fluids on Organic Rankine Cycle for Waste Heat Recovery." *Energy* 29: 1207-17.

[17] Chen, H., Goswami, D. Y., and Stefanakos, E. K. 2010. "A Review of Thermodynamic Cycles and Working Fluids for the Conversion of Low-Grade Heat." *Renewable and Sustainable Energy Reviews* 14 (9): 3059-67. https://doi.org/10.1016/j.rser.2010.07.006.

[18] Quoilin, S., Van Den Broek, M., Declaye, S., Dewallef, P., And Lemort, V. 2013. "Techno-Economic Survey of Organic Rankine Cycle (ORC) Systems." *Renewable and Sustainable Energy Reviews* 22: 168-86.

[19] Ford, G. 2008. "CSP: Bright Future for Linear Fresnel Technology?" *Renewable Energy Focus* 9 (5): 48-51.

[20] Martinez, S., Michaux, G., Salagnac, P., and Bouvier, J.-L. 2017. "Micro-combined Heat and Power Systems (Micro-CHP) Based on Renewable Energy Sources." *Energy Conversion and Management* 154: 262-85.

[21] Kane, M., Larrain, D., Favrat, D., and Allani, Y. 2003. "Small Hybrid Solar Power System." *Energy* 28 (14): 1427-43.

[22] Thiao, S., Mar, A., Dioh, R. M. N. D., Mbow, C., and Youm, I. 2018. "Feasibility Study of Solar Thermal cooling Warehouse on a remote Area: Estimation of the Solar Thermal Collector Area and Efficiency." *International Journal on Energy Conversion* 6 (3): 90-6.

[23] Ferreira, C., and Kin, D. S. 2014. "Techno-Economic Review of Solar Cooling Technologies Based on Location Specific Data." *Int. J. Refrigeration* 39: 23-37.