Journal of Applied Energies

Performance of reheat organic Rankine cycle using wet fluids

Youcef Abdellah Ayoub Laouid[1*], Cheikh Kezrane[1], Yahia Lasbet[2]

[1]Laboratoire de Développement en Mécanique et Matériaux (LDMM), Université de Djelfa Route de Moudjebara, 17000 Djelfa, Algeria.

[*]Correspondance. Email address: youcef.ayoub.la@gmail.com

Abstract: An efficient extraction of power can be achieved through an organic Rankine cycle, utilizing low-grade heat obtained from waste energy sources. The current study focuses on exploring a reheated Organic Rankine Cycle using various wet organic fluids. Within the reheat Organic Rankine Cycle model, there are several components, including an economizer, evaporator, superheater, reheater, two turbines, a pump, and condensers. The net power output and thermal efficiency are estimated with variable turbine inlet temperatures and reheat pressures. The optimum turbine inlet temperatures that maximized the net power output were 358K and 363K for Propylene, Difluoroethane, and cyclopropane, respectively. However, the higher turbine inlet temperature will lead to significant thermal efficiency. The net power recorded an increase exceeding 13% for all the fluids studied.

Keywords: Organic Rankine Cycle; Net power output; Thermal efficiency; Reheat pressure.

1. INTRODUCTION

Over the past few decades, the increase in fossil fuels consumption has contributed to many serious environmental problems, such as global warming, ozone layer depletion, acid rain,

and atmospheric pollution. Therefore, the utilization of waste heat sources becomes more attractive. Researchers proposed many thermodynamic cycles such as the Organic Rankine cycle, supercritical $Co_{2,}$ and triangle cycle. Among the previous process, ORC is more straightforward and more widely used. Many studies have been carried over the past decades to determine the performance of ORC. Dai et al. [1] analyzed and compared ten working fluids such as ammonia, R236EA, and R113 on different ORC configurations. They used exergy efficiency as an objective function. They found that the R236EA had the most outstanding exergy efficiency. The addition of an internal heat exchanger could not improve the performance of ORC under the same waste heat source temperature. Li [2] investigated different ORC configurations based on the heat source temperature. He selected only isentropic and dry fluids, and he found that ORC with internal heat exchanger and regenerative ORC had the best thermal efficiency compared to the other configurations. Imran et al. [3] made a thermo-economic optimization of basic and regenerative ORC. They selected the thermal efficiency and specific investment cost as the objective functions. Their results showed that the R245fa is the best working fluid and the basic ORC had the lower typical investment cost and thermal efficiency than the regenerative ORC. Other related works may be found in Refs. [4-9].

The main aim of this study is focused on the reheat ORC with wet fluids under the same waste heat source. The effect of expander inlet temperature and the reheat pressure on thermal efficiency and net power output is explored.

2. THERMODYNAMIC ANALYSIS OF ORGANIC RANKINE CYCLE

2.1. Design and model description

A typical reheat ORC consists of a pump, an evaporator, two expanders, and a condenser, as shown in Figure 1. The working fluid is first pumped into the evaporator, gaining heat from the heat source and turning into superheated vapor. Then the vapor expands in the first turbine, and thus power is generated, then the vapor enters the reheater, which absorbs heat from the heat source before passing through a second, lower-pressure turbine. The reheat temperatures are equal to the inlet temperatures, whereas the reheat pressure is lower than the original evaporator pressure. The exiting fluid from the lower-pressure turbine flows into the condenser and is cooled to a saturated liquid. The present study assumes that (a) the system is in a steady-state, (b) there are no pressure drops in the economizer, evaporator, superheater, reheater, condenser, and pipes, (c) no heat loss in all components, (d) the reheat pressure is

the average pressure of condensing pressure and evaporating pressure [2]. The efficiency of the low pressure and high pressure turbines is taken as 0.8, the efficiency of the pump is taken as 0.7 and the pinch temperature difference in the evaporator is considered to be 8 K°. Other operating conditions were assumed as listed in Table 1. The performance of reheat ORC was studied by using the first law of thermodynamics.

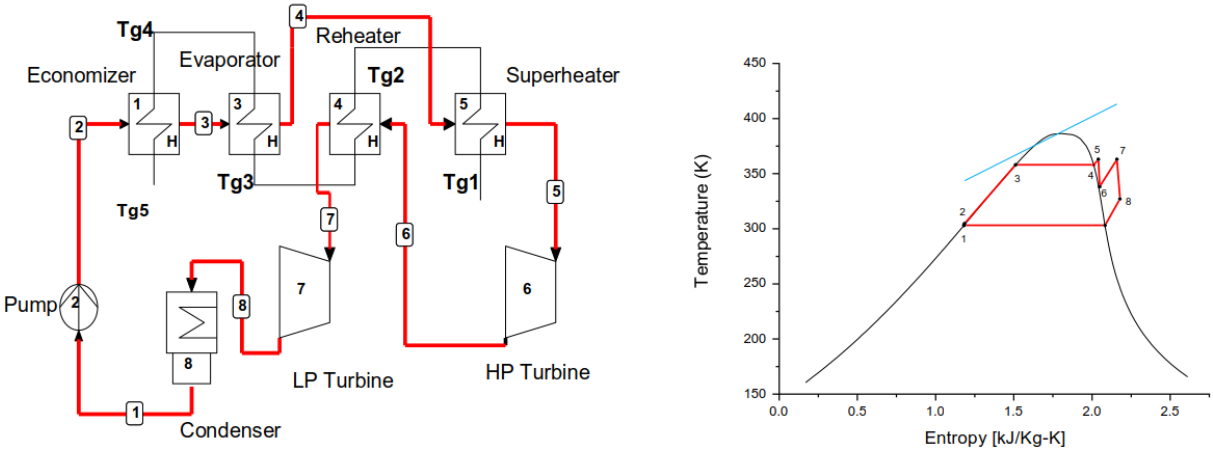


Figure 1: System design and cycle T-s diagram of the reheat ORC system.

The energy balance in economizer, evaporator, super heater, and reheater are:

$$\dot{m}_{wf}\,(h_3 - h_2) = \dot{m}_g Cp_m (Tg_4 - Tg_5)$$

$$\dot{m}_{wf}\,(h_4 - h_3) = \dot{m}_g Cp_m (Tg_3 - Tg_4)$$

$$\dot{m}_{wf}\,(h_5 - h_4) = \dot{m}_g Cp_m (Tg_1 - Tg_2) \quad (1)$$

$$\dot{m}_{wf}\,(h_7 - h_6) = \dot{m}_g Cp_m (Tg_2 - Tg_3)$$

where $\dot{m}_{wf}$ and $\dot{m}_g$ are the mass flow rates of the working fluid and the gas, Cp_m is The mean specific heat capacity of the gas at constant pressure, h is the specific enthalpies of the working fluid at each state and Tg is the temperature of heat source at each state.

The turbine power for both HP and LP turbine can be calculated as:

$$W_T = \dot{m}_{wf}\,(h_5 - h_6) + \dot{m}_{wf}\,(h_7 - h_8) \quad (2)$$

The heat rejection in the condenser is:

$$\dot{Q}_{con} = \dot{m}_{wf}\,(h_8 - h_1) \quad (3)$$

The work input by the pump is:

$$W_{Pump} = \dot{m}_{wf}\,(h_8 - h_1) \quad (4)$$

The heat added to the cycle is the sum of $\dot{Q}_{eco}$,$\dot{Q}_{eva}$,$\dot{Q}_{Sup}$ and $\dot{Q}_{\mathrm{Re}h}$ which can be expressed as:

$$\dot{Q}_{in} = \dot{m}_{wf}\,(h_5 - h_2) + \dot{m}_{wf}\,(h_7 - h_6) \tag{5}$$

The net power output, which is the difference between the power generated by the two turbines and the power consumed by the pump, can be calculated by:

$$W_{net} = W_T - W_{Pump} \tag{6}$$

The thermal efficiency of ORC systems is defined as the ratio of the net power output to the heat addition:

$$\eta = \frac{W_{net}}{\dot{Q}_{in}} \tag{7}$$

The pinch point is:

$$Tg_4 = T_3 + \Delta T_{pp} \tag{8}$$

The turbine inlet temperature is:

$$T_5 = T_4 + 5 \tag{9}$$

The above equations can calculate the net output power, the thermal efficiency and the mass flow rate of the working fluid, and the hot gas temperatures.

Table 1: Assumed operating conditions [10].

Temperatures of heat source (K)	413
Mass flow rate of hot gas	14
Specific heat at constant pressure for hot gases (kJ/kg·K)	1.1
Condensing temperature (K)	303

Table 2: Properties of the selected working fluids.

	Working fluid		
Propertie	Propylene	Difluoroethane	Cyclopropane
M (kg/kmol)	42.08	66.05	42.08
T_{cr} [K]	364	386	398
P_{cr} [MPa]	4.55	4.5	5.6

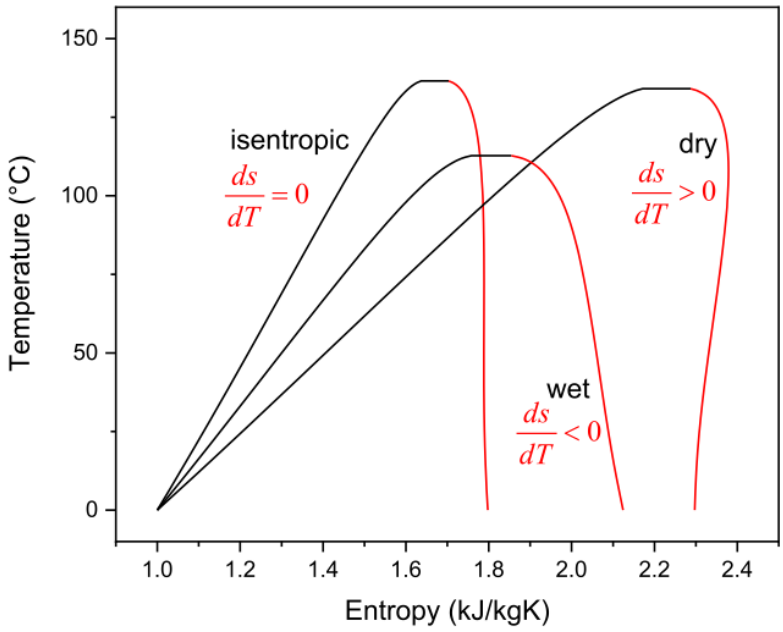


Figure 2: Types of working fluids: dry, isentropic and wet [11].

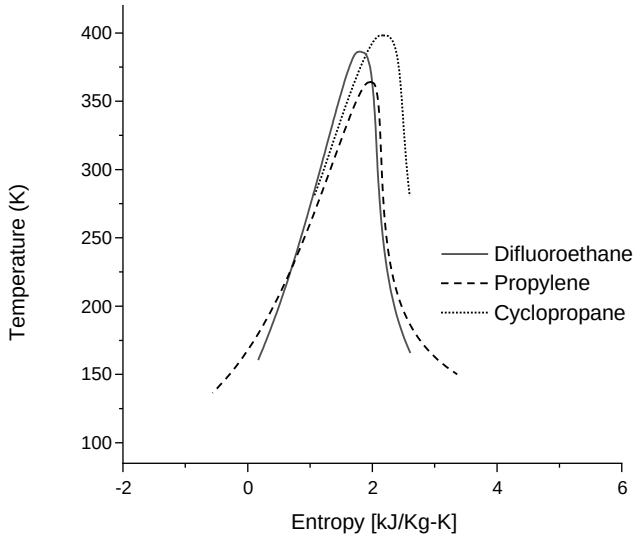


Figure 3: T-s diagrams of the working fluid.

2.2. Working fluid

According to [12], there are three types of working fluids: dry, isentropic, and wet fluids depending on the slope of saturation vapor curve on T-s diagram dT/ds. Figure 2 shows the three types of fluids in a T–s diagram. For environmental reasons, some working fluids have been phased out, such as R-11, R-12, R-113, R-114, and R-115, while some others are being phased out in 2020 or 2030 (such as R- 21, R-22, R-123, R-124, R-141b, and R-142b) [12]. In this work, Propylene, Difluoroethane and Cyclopropane will be considered. Table 2 shows

their critical properties, and Figure 3 shows the working fluid candidate in T-s diagram. All the data in Table 2 comes from Coolprop [10] and Ref [13].

3. RESULTS AND DISCUSSION

3.1. Effect of turbine inlet temperature on net power output and thermal efficiency

In this section, we set the evaporation temperature variation from 323K to T_{cr} −10 limit the subcritical cycle and keep an appropriate temperature difference between the hot gas and the working fluid. The turbine inlet temperature is the evaporation temperature plus 5 K to ensure that the exit of working fluid from the HP turbine lies in the superheated vapor region.

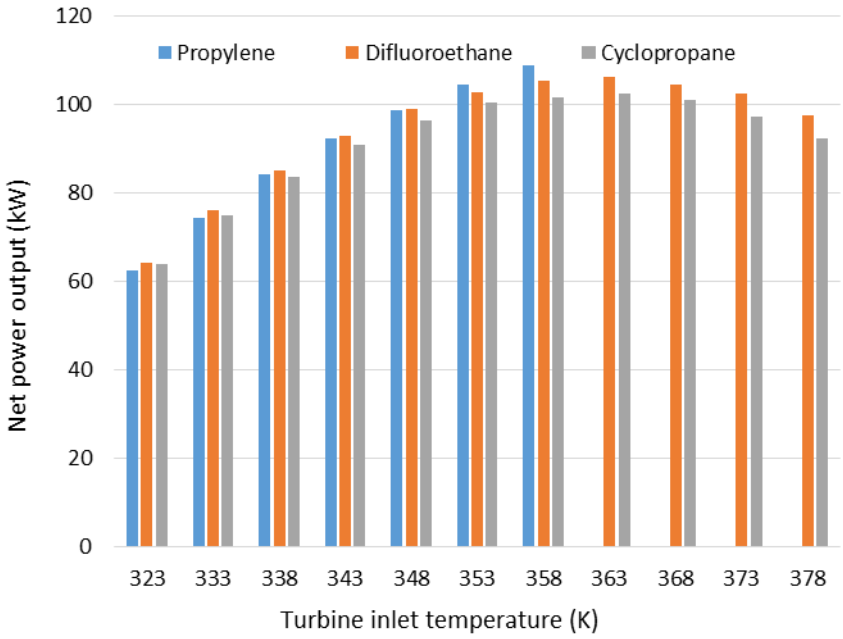


Figure 4: Net power output with turbine inlet temperature.

Figure 4 illustrates the net power output with the turbine inlet temperature. The net power output increases first and then decreases after reaching its maximum at the turbine inlet temperature of 358K with a net power output of 108.63 kW for Propylene. Differently, the net power output of Cyclopropane increases and reaches its maximum at a turbine inlet temperature of 363 K. The turbine inlet temperature of Propylene is lower because of its lower critical temperature. Not always, the higher turbine inlet temperature causes a significant net power output. Figure 5 shows the variation of thermal efficiency with the turbine inlet temperature. For all working fluids, the thermal efficiency increases with the increase of turbine inlet temperature, while the cyclopropane presents the maximum thermal efficiency compared to Difluoroethane and Propylene.

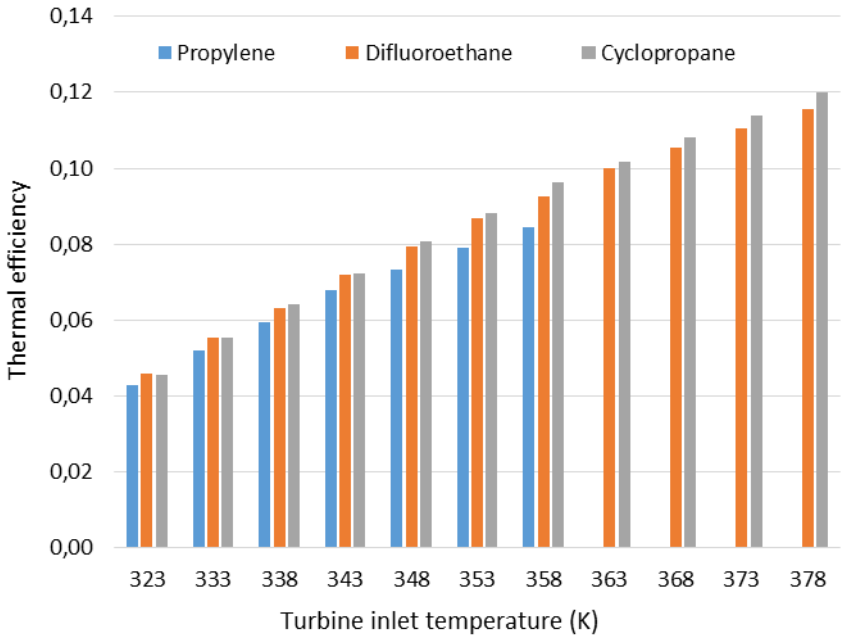


Figure 5: Thermal efficiency with turbine inlet temperature.

3.2. Effect of reheat pressure on net power output and thermal efficiency

In this section, we set the evaporation temperature as 358K, and the maximum reheat pressure is $P_{ev} * 0.9$.

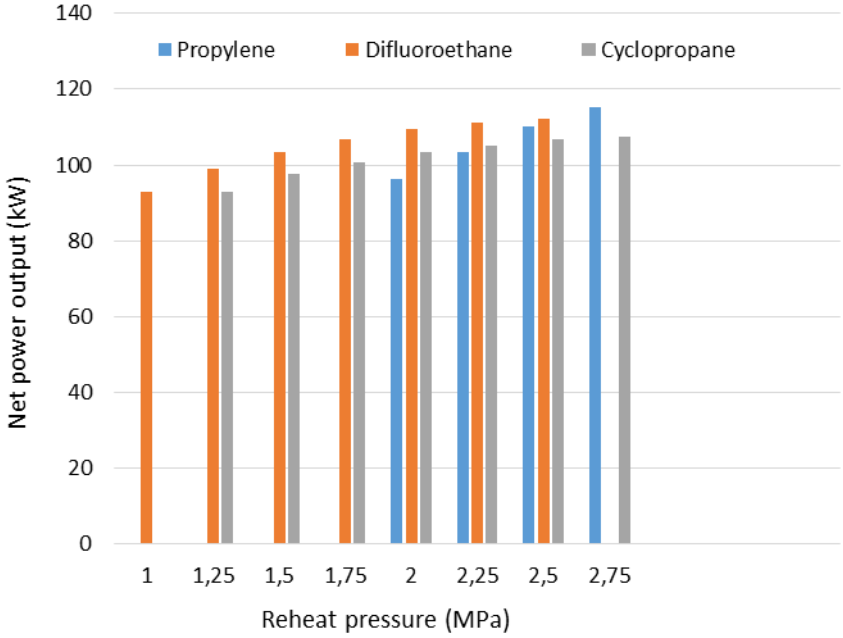


Figure 6: Net power output with reheat pressure.

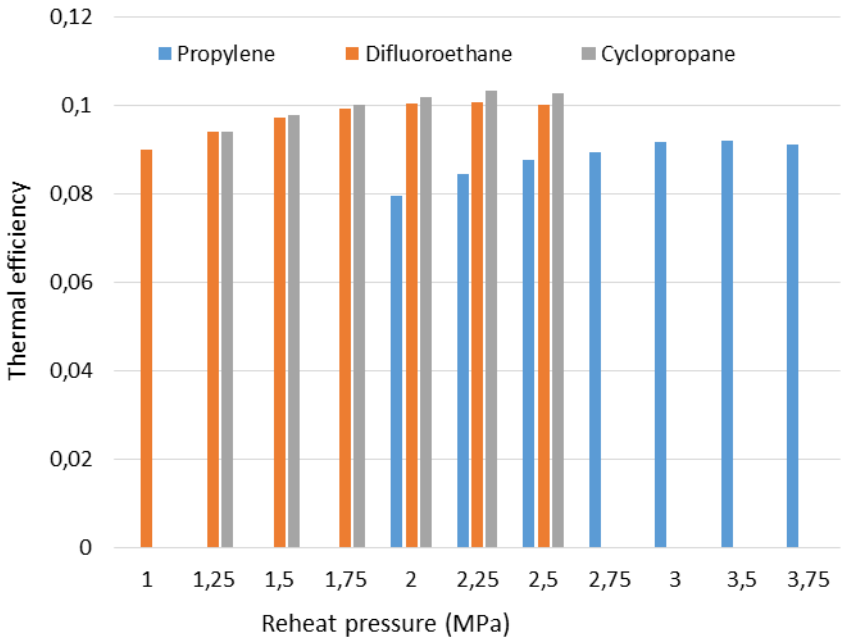


Figure 7: Thermal efficiency with reheat pressure.

Figure 6 presents the effects of the variation of the reheat pressure on the net power output. The net power output increases with the increase of the reheat pressure, and Propylene has the highest net power output with a value of 115 kW. In comparison, Difluoroethane and cyclopropane have maximum values of 112 kW and 107 kW, respectively. Figure 7 presents the effects of the variation of the reheat pressure on thermal efficiency. There is a marginal increase in thermal efficiency for all working fluids, which is about 1.05%, 1.15%, and 0.9% for Difluoroethane, Propylene, and cyclopropane, respectively. From Figures 6 and 7, the increase of reheat pressure gives only marginal growth in the efficiency of the cycle. Still, it raises the net power output by making higher pressures possible and keeping the quality of working fluid at turbine exhaust remains in the vapor region.

3. CONCLUSION

In this work, the effects of turbine inlet temperature and reheat pressure on the net power output and the thermal efficiency were studied. Three wet working fluids were analyzed for the reheat ORC, given the waste heat source temperature 413. The main results are as follows: The optimum turbine inlet temperatures that maximized the net power output were 358K and 363K for Propylene, Difluoroethane, and cyclopropane, respectively. However, the higher turbine inlet temperature will lead to significant thermal efficiency.

The increase of reheat pressure gave only marginal growth in the efficiency of the cycle. Still, it increased the net power output by 18.7 kW, 18.9 kW, and 14.15 kW for Propylene, Difluoroethane, and cyclopropane, respectively.

REFERENCES

[1] Chen, H., Goswami, D. Y., 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), 3059-3067.

[2] Dai, Y., Wang, J., Gao, L. (2009). Parametric optimization and comparative study of organic Rankine cycle (ORC) for low grade waste heat recovery. *Energy Conversion and Management*, *50*(3), 576-582.

[3] Xi, H., Li, M. J., He, Y. L., Tao, W. Q. (2015). A graphical criterion for working fluid selection and thermodynamic system comparison in waste heat recovery. *Applied Thermal Engineering*, *89*, 772-782.

[4] Li, G. (2016). Organic Rankine cycle performance evaluation and thermoeconomic assessment with various applications part II: Economic assessment aspect. *Renewable and Sustainable Energy Reviews*, *64*, 490-505.

[5] Zheng, X., Luo, X., Luo, J., Chen, J., Liang, Y., Yang, Z., ... & Wang, H. (2020). Experimental investigation of operation behavior of plate heat exchangers and their influences on organic Rankine cycle performance. *Energy Conversion and Management*, *207*, 112528.

[6] Bademlioglu, A. H., Canbolat, A. S., Kaynakli, O. (2020). Multi-objective optimization of parameters affecting Organic Rankine Cycle performance characteristics with Taguchi-Grey Relational Analysis. *Renewable and Sustainable Energy Reviews*, *117*, 109483.

[7] Köse, Ö., Koç, Y., Yağlı, H. (2020). Performance improvement of the bottoming steam Rankine cycle (SRC) and organic Rankine cycle (ORC) systems for a triple combined system using gas turbine (GT) as topping cycle. *Energy Conversion and Management*, *211*, 112745.

[8] Fan, G., Du, Y., Li, H., Dai, Y. (2021). Off-design behavior investigation of the combined supercritical CO2 and organic Rankine cycle. *Energy*, *237*, 121529.

[9] Wang, S., Liu, C., Li, Q., Liu, L., Huo, E., Zhang, C. (2020). Selection principle of working fluid for organic Rankine cycle based on environmental benefits and economic performance. *Applied Thermal Engineering*, *178*, 115598.

[10] http://www.coolprop.org/

[11] Laouid, Y. A. A., Kezrane, C., Lasbet, Y., & Pesyridis, A. (2022). Performance analysis and optimisation of a reheat organic Rankine cycle. *International Journal of Sustainable Energy, 41*(7), 909-931. doi: 10.1080/14786451.2021.1990294.

[12] Imran, M., Park, B. S., Kim, H. J., Lee, D. H., Usman, M., Heo, M. (2014). Thermo-economic optimization of Regenerative Organic Rankine Cycle for waste heat recovery applications. *Energy Conversion and Management, 87*, 107-118.

[13] Li, G. (2016). Organic Rankine cycle performance evaluation and thermoeconomic assessment with various applications part I: Energy and exergy performance evaluation. *Renewable and Sustainable Energy Reviews, 53*, 477-499.