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MODELING OF AN ORGANIC RANKINE CYCLE FOR LOW TEMPERATURE HEAT SOURCES

Arthur Batista Martins Lott
Pontifícia Universidade Católica de Minas Gerais, Department of Mechanical Engineering, Belo Horizonte, Brazil
arthurlott@hotmail. com

Arthur Pacheco luz
Pontifícia Universidade Católica de Minas Gerais, Department of Mechanical Engineering, Belo Horizonte, Brazil
arthur_pacheco.luz@hotmail.com

João Arthur Daconti e Silva
Pontifícia Universidade Católica de Minas Gerais, Department of Mechanical Engineering, Belo Horizonte, Brazil
dacontisilva@hotmail.com

Cristiana Brasil Maia
Pontifícia Universidade Católica de Minas Gerais, Department of Mechanical Engineering, Belo Horizonte, Brazil
cristiana@pucminas.br

Sérgio de Morais Hanriot
Pontifícia Universidade Católica de Minas Gerais, Department of Mechanical Engineering, Belo Horizonte, Brazil
hanriot@pucminas.br

Abstract. *Rankine cycles represents today a strong and competitive energy cycle to produce energy from waste heat recovery or renewable sources. This article brings a simple mathematical modeling for Organic Rankine cycles, a variant of Rankine cycles that working with organic fluids. It's showing their standard components and how they affect the cycle, the most common working fluids for the cycle and the power generation results for different fluids in a simple and ideal configuration.*

Keywords: *Organic Rankine cycle, mathematical modelling, low temperature thermal sources, organic fluid.*

1. INTRODUCTION

Organic Rankine cycles represent an alternative way to generate or recover energy from different sources of heat, so, a study of this kind of Rankine cycle becomes advisable. Factors such as the oil crisis of 1970, geopolitical issues and environmental aspects are increasingly on the subject in important discussions around the world. From this point on, we seek to understand the mathematical modeling that describes its behavior, bringing with important details that need to be considered, such as the transition from a single-phase to two-phase flow in the evaporator presented by Helvaci (2017).

In the same way that Wang (2013) demonstrates in his model of the organic Rankine cycle directed to solar energy, modeling the cycle for small-scale energy generation, as well as Moreira (2017) in he's model for reuse of heat in a cement plant, the intention of this article is to present the mathematical modeling that describes the organic Rankine cycle for low temperature alternative heat sources and also present results obtained by the presented models.

2. SOLAR ORGANIC RANKINE CYCLE

Organic Rankine cycles are a variation of the conventional Rankine cycle, which uses water vapor as the working fluid. The main difference between these two cycle models is the amount of energy that the cycle needs to produce work as presented by Muñoz (2013).

Organic rankine cycles works with basically hydrocarbons, siloxanes and refrigerants as working fluids. A great advantage of these fluids is the possibility of working with lower temperatures. The organic fluids utilized by the cycle contemplates more than one class of fluids, but, according to (Landelle, et al., 2017) the working fluids can be classified from their use as shown in Figure 1. This classification also gives us the information that most organic Rankine cycles work with refrigerants as working fluid. 52% of the working fluids are HFC, 20% are HCFC, 7% are hydrocarbons, 6% HFE, 4% mixtures, 2,5% PFC, 2,5% CFC, 2,5% HFO and 5% of other fluids, and this data provides us with information that the biggest part of organic rankine cycles work with refrigerants as working fluid, according to (Landelle, et al., 2017) the biggest portion of utilized working fluids. Still according to (Landelle, et al., 2017) the fluids are composed basically by R245fa (52%), R123(18%) and R134a(7%). On solar organic rankine cycles more specifically, the R245fa and R134a are largely used for applications that work below 250 ºC according to (Abelowafa, et al., 2018).

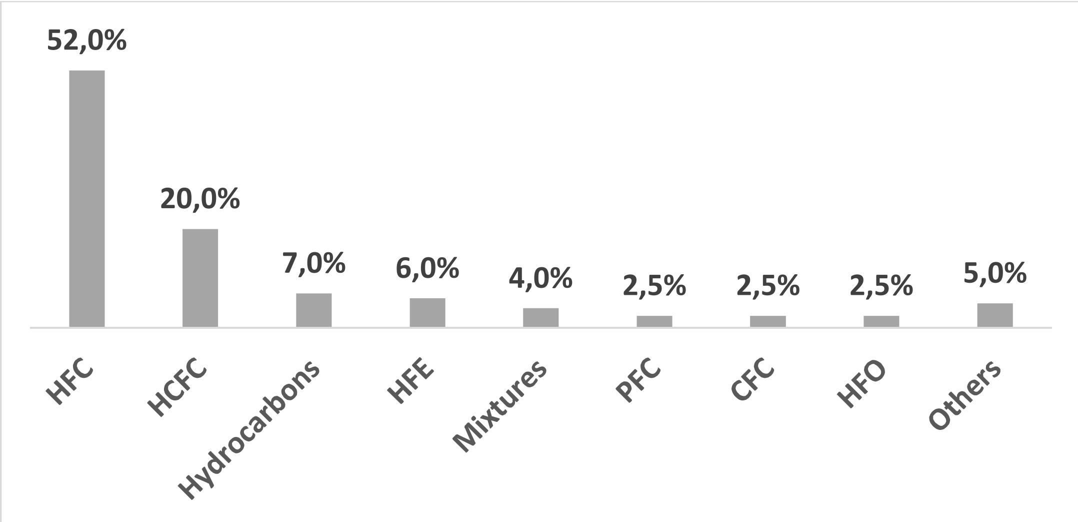

Figure 1. Percentage of the use of organic fluids in ORCs. Adapted from Landelle et al. (2017).

According to Macchi and Astolfi (2016), the ORC is applied in basically 4 types of thermal sources, they are: reuse of the residual heat of processes and equipment, Biomass, Geothermal energy and Solar energy. For example, Muñoz (2013) developed a study that evaluated the thermodynamic viability of the ORC using three different thermal sources, geothermal, solar and biomass. It was also done a modeling of the cycle with different fluids to obtain the generated power and the yield. According to the author, among those three sources studied, the one that generated the most power was solar energy, but the one that obtained the greatest efficiency was the energy coming from the biomass.

2.1. Components

Rankine cycles are composed by four standard equipment's as shown in figure 2, they are the expander, the condenser, the pump and the evaporator. As indicated by (Landelle, et al., 2017), the expander is a fundamental component to study when analyzing the power capacity of the cycle since the expander efficiency is correlated with its power scale. Also according to (Li Zhao, 2013), along with the power generation capacity of the turbine are related parameters as: pressure ratios, lubrication requirements, complexity, reliability, cost, among other factors that impact on the choice of turbine for the system.

Pumps are used to generate a mass flow of the working fluid on the cycle, however the most important parameter of the pump is its power consumption, because it interferes directly on the power output of the system once some portion of the power produced by the expander is used to drive the pump. The relation between the work generated by the turbine and the work spent by the pump on the conventional Rankine cycle is generally low, but for the organic Rankine cycle this ratio is a considerable parameter, stating that pump efficiency are important for cycles with low temperatures, such as organic Rankine cycles that work with some fluids as described earlier in this article.

Evaporators and condensers are components that must adapt to the thermal source of the cycle, so these components are sized according to applications for renewable energy sources, so, despite classifying the cycle, the major interest of the study is the scale of energy generated by the turbine, but (Helvaci and Khan, 2017) showed that, on the mathematical model of the cycle, both components need attention once the fluid flowing through them shows significant changes, such as single-phase flow to two-phase flow on the evaporator, meaning significant changes on the result of a mathematical model of the cycle.

2.2. Energetic Analysis and Mathematical Modeling

Considering a cycle as a control volume and, through the mass balance Equation (1) and the law of Conservation of Energy Equation (2), it is possible to determine the power of the turbine and the pump, the heats exchanged in the evaporator and the condenser and, the efficiency of the cycle (Moreira, 2017).

The law of conservation of mass can be applied by the equation (1):

$$\frac{dm_{cv}}{dt} = \sum \dot{m}_i - \sum \dot{m}_o \tag{1}$$

Where:
$\frac{dm_{cv}}{dt}$ – mass flow rate [kg/s];
$\dot{\boldsymbol{m}}_{\boldsymbol{\iota}}$ and $\dot{\boldsymbol{m}}_{\boldsymbol{o}}$ – mass flow rate in and out, respectively [kg/s];

First law of thermodynamics can be applied by de equation (2):

$$\frac{de_{cv}}{dt} = \dot{Q_{cv}} - \dot{W_{cv}} + \sum \dot{\boldsymbol{m}}_{\boldsymbol{\iota}} \left(h_i + \frac{1}{2}V_i^2 + gz_i\right) - \sum \dot{\boldsymbol{m}}_{\boldsymbol{o}} \left(h_o + \frac{1}{2}V_o^2 + gz_o\right) \quad (2)$$

Where:
$\frac{de_{cv}}{dt}$ – energy rate of change [kW];
$\dot{Q_{cv}}$ – Heat transfer to the cycle;
$\dot{W_{cv}}$ – Power transfer to the environment;
h_i and h_o – Input and output enthalpy of the system, respectively [kJ/kg];
$\frac{1}{2}V_i^2$ and $\frac{1}{2}V_o^2$ – Input and output Kinetic energy, respectively [m²/s²];
gz_i and gz_o – Input and output Potential energy, respectively [m²/s²].

Equation (2) shows the overall energy balance of the cycle, but only with the overview of the cycle can not obtain the respective energy values and the working fluid status for each component of the cycle. For those reason, a particular evaluation is made in each component, as equations (3), (4), (5) and (6) present.

For the evaporator, $\dot{Q}_{evap} = \dot{m}(h_o - h_i)$ (3)
For the expander, $\dot{W}_{exp} = \dot{m}(h_i - h_o)$ (4)
For the condenser, $\dot{Q}_{cond} = \dot{m}(h_o - h_i)$ (5)
For the pump, $\dot{W}_{pump} = \dot{m}(h_i - h_o)$ (6)

Where:
$\dot{Q}_{evap}$ – Energy transferred in the evaporator [kW];
$\dot{W}_{exp}$ – Power generated in the expander [kW];
$\dot{Q}_{cond}$ – Energy transferred in the condenser [kW];
$\dot{W}_{pump}$ – Power spent at the pump [kW].

By the reason that the expander and the pump are the generator and the energy consumer respectively, they need an evaluation of their efficiencies, once those efficiencies interact directly with the overall efficiency of the cycle and for this there are equations (7) and (8).

For the expander, $\eta_t = \frac{h_1 - h_2}{h_1 - h_{2s}}$ (7)
For the pump, $\eta_t = \frac{h_{2s} - h_1}{h_2 - h_1}$ (8)

Where:
η_t – Isentropic component efficiency;
h_1 e h_2 – Input and output enthalpy of the component, respectively [kJ/kg];
h_{2s} – Enthalpy of the component isentropic state [kJ/kg].

The power produced in the cycle can be obtained by the difference between the power generated by the expander and the power consumed in the pump, according to equation (9).

$$W_{liq} = \dot{W}_{exp} - \dot{W}_{pump} \quad (9)$$

Finally, the cycle thermal efficiency is calculated by (10):

$$\eta_{th} = \frac{\dot{W_{liq}}}{\dot{Q_{\iota}}} \quad (10)$$

Where:
η_{th} – Thermal cycle efficiency [%];

W_{liq} – Power produced [kW];
Q_i – Heat transfer to inside cycle [kW].

3. RESULT AND DISCUSSION

Figure 2 presents a conventional Rankine cycle with its 4 basic components and a T-S diagram with the ideal state of the fluid in each steady state process. Process 1>2 consists of an isentropic expansion of the fluid in the turbine, process 2>3 represents a isobaric rejection of heat in the condenser until the fluid becomes saturated liquid state, process 3>4 consists of a isentropic increasing pressure using the pump, 4>1 shows a isobaric heat transfer in the evaporator until the fluid is in saturated vapor state.

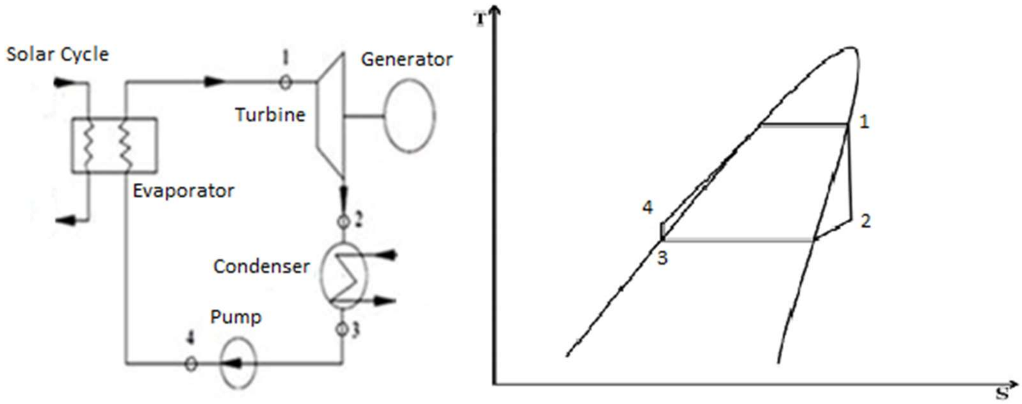

Figure 2. Simple cycle ORC, configuration and e diagram T-S. Adapted from (Muñoz, 2013).

From this configuration represented by Figure 2, and assuming the equations already presented in the previous section, figure 3 shows the results of power produced by the organic Rankine cycle and the thermal efficiency of the same by changing the input temperature of the fluid in the expander for different working fluids.

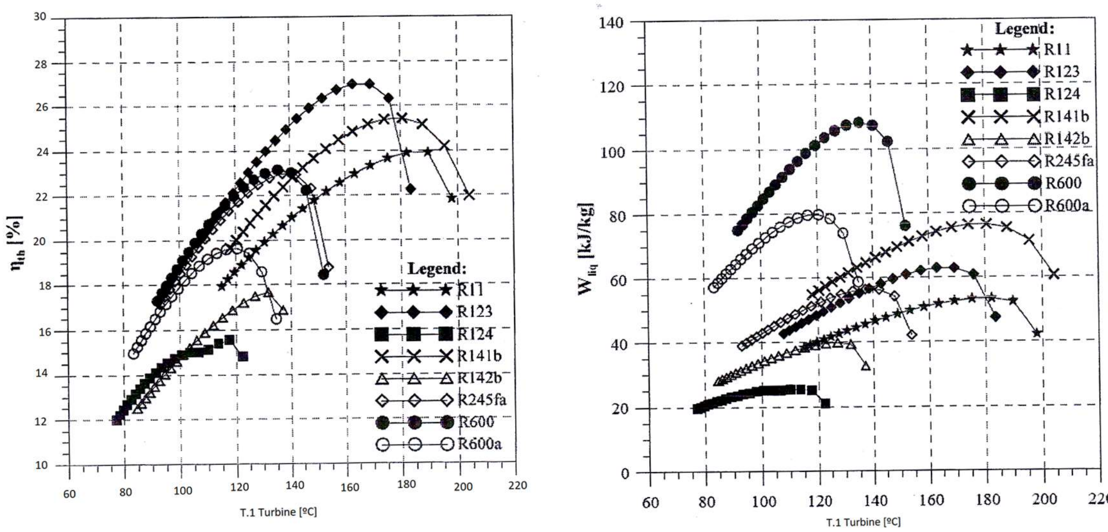

Figure 3. Generated power and efficiency for an ideal cycle with temperature variation at the turbine inlet using different working fluids. Adapted from (Moreira, 2017).

From the results of figure 3, it is possible to note that as well as the efficiency, the power generated in the cycle increase with the increase of the temperature at the turbine inlet, but the cycle loses this characteristic when the temperature approaches to the critical temperature of the fluid.

Other results also for the configuration presented in Figure 2 were obtained, this time with boundary conditions and pre-defined fluids as Tables 1 and 2 present. The fluids chosen were R-134a and R-245fa.

Variable	Value
Power generated in the expander (kW)	2
Temperature of the working fluid at the evaporator outlet (ºC)	95
Cooling water temperature at the condenser inlet (ºC)	25
Cooling water temperature at the condenser outlet (ºC)	30
ORC pump inlet pressure (bar)	1
Ambient pressure (bar)	1

Table 1 – Input data of the cycle, prepared by the author.

The other contour conditions presented in Table 2 were extracted from the literature cited in this article.

Variable	Value
Pressure at the turbine inlet (bar)	16
Isentropic efficiency of pumps (%)	70
Isentropic turbine efficiency (%)	85

Table 2 – Other input data of the cycle, made by the author.

From the data presented above it was possible to obtain the thermodynamic properties of each cycle in each state as presented in Tables 3 and 4 for each state of the cycle for the fluids R-134a and R-245fa, respectively.

Status	*T* [ºC]	*P* [bar]	*v* [m³/kg]	*h* [kJ/kg]	*s* [kJ/kg.K]	*X* [-]	Fluid status
1	57,92	16	1,208E-02	275,33	0,8982	1	Saturated steam
2S	-26,43	1	1,846E-01	223,39	0,8982	0,963	Liquid + steam
2	-26,43	1	1,915E-01	231,18	0,9386	0,999	Liquid + steam
3	-26,43	1	7,250E-04	16,29	0,0678	0	Saturated liquid
4S	-25,54	16	7,272E-04	17,38	0,0678	-	Compressed liquid
4	-25,17	16	7,278E-04	17,84	0,074	-	Compressed liquid

Table 3 – Thermodynamic properties of the cycle for the R-134a fluid, made by the author.

Status	*T* [ºC]	*P* [bar]	*v* [m³/kg]	*h* [kJ/kg]	*s* [kJ/kg.K]	*X* [-]	Fluid status
1	110,91	16	1,050E-02	480,20	1,7962	1	Saturated steam
2S	29,88	1	1,808E-01	429,15	1,7962	-	Superheated steam
2	38,17	1	1,867E-01	436,81	1,8211	-	Superheated steam
3	14,81	1	7,322E-02	219,08	1,0679	0	Saturated liquid
4S	15,28	16	7,301E-04	220,18	1,0679	-	Compressed liquid
4	15,64	16	7,311E-04	220,65	1,0695	-	Compressed liquid

Table 4 – Thermodynamic properties of the cycle for the R-245fa fluid, made by the author.

After the calculation of the properties, it was possible to obtain the results of the mass flow through the cycle, the pump power, the rejected heat in the condenser and the heat supplied to the ORC. Those data are shown in Tables 5 and 6. It was also possible to compare the thermal efficiencies for each fluid according to Figure 4.

Thermal efficiency [%]	Heat in Evaporator [kW]	Heat in the Condenser [kW]	Power at the pump [kW]	Mass flow [kg/s]
16,50	11,66	9,73	0,070	0,043

Table 5 – Results for the R-134a fluid, made by the author.

Thermal efficiency [%]	Heat in Evaporator [kW]	Heat in the Condenser [kW]	Power at the pump [kW]	Mass flow [kg/s]
21,78	8,90	6,96	0,062	0,039

Table 6 – Results for the R-245fa fluid, made by the author.

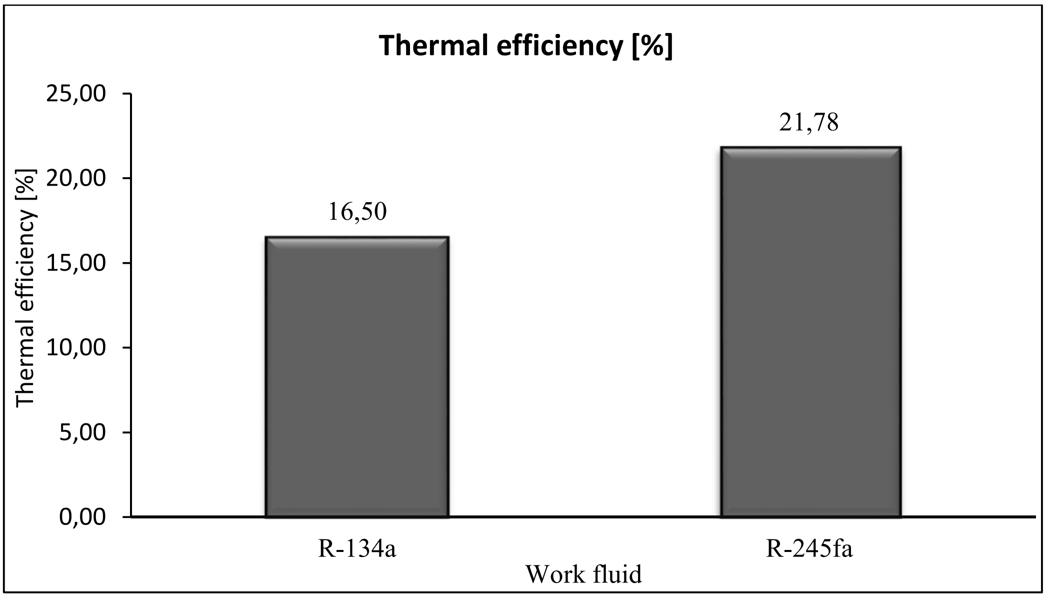

Figure 4 – Thermal efficiency of the selected work fluids, made by the authors.

It is observed from Figure 4 that the R-245fa fluid presented the same thermal efficiency range as Figure 3 (b) presented for several temperature ranges contributing to the validation of the calculations. In addition, it is worth mentioning that the small amount of fluid required by the cycle is justified by the low power generated.

4. CONCLUSION

The results show that when applying the mathematical model presented to the Organic Rankine cycle with low temperature heat source for different working fluids, higher power was obtained with those who has greatest difference between the critical temperature and the input temperature of the expander Moreira (2017). In addition, it was demonstrated that the organic Rankine cycle follows the thermodynamic laws and that the cycle modeling is similar to the conventional Rankine.

It is still possible to conclude that for the low temperature organic Rankine cycle with simple configuration the fluid who presented the highest efficiency was the dry fluid R123 but from the point of view of energy production the fluid with the highest generation capacity was the R600, isentropic fluid Moreira (2017).

The working fluid is a determining factor in the efficiency of the cycle, because it defines important parameters of the cycle, such as evaporator outlet temperature and turbine inlet pressures and condenser outlet.

5. ACKNOWLEDGEMENTS

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