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Comparison of Working Fluids of an Organic Rankine Cycle with Recuperator for a Small solar power plant CSP

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Abstract This work deals with the comparison and classification of working fluids of an organic Rankine cycle with recuperator (ORC) of a Small solar power plant CSP of 3 kW. Several organic working fluids (R500, R152a, R134a, R717 and R290) were evaluated and compared for better optimization of the system.The energy efficiency, exergy efficiency, cycle heat input, total irreversibility, volume flow, mass flow rate, pressure ratio, toxicity, flammability, ODP (Potential for depletion of Ozone) and GWP (global warming potential) were used for the comparison of the various fluids.The results of such a comparison show that the fluids R152a and R134a appear as the most suitable fluids for ORC applications with a temperature between 80 °C. and 130 °C. followed by R290 and R717 due to their flammability. The exergy analysis of the cycle made it possible to evaluate the recovery potential of the noble energies and the level of the irreversibilities of the various components. A low irreversibility rate maximizes the exergy efficiency. The evaporator and recuperator contribute 74% and 10% to the total irreversibility of the cycle, respectively, followed by the condenser and the turbine. The heat input necessary to produce 3 kW ranges from 55.01 kW (R7171) to 48.95 kW (R134a).

Keywords Working fluid, Organic Rankine cycle, Energy and Exergetic Analysis, Equation Engineering Solver (EES)

1. Introduction

Access to modern energy services must be seen as a universal right to lift the underdeveloped countries out of poverty. According to the International Energy Agency [1], about 22% of the world's population still lacks access to electricity. In 2008, this represented 1.5 billion people, most of whom lived in remote areas and were difficult to connect to national or regional networks because of the high connection costs for some areas. The IEA estimates that 85% of these people live in rural areas of developing countries and the majority are in sub-Saharan Africa and South Asia. According to [2] the minimal need for electricity for people to read at night, pumping a minimum amount of drinking water and listening to radio broadcasts is only 50 kWh per person per year. Faced with this situation, it is urgent to solve the energy poverty problem of these rural populations. Currently, there are a variety of viable and competitively priced renewable energy solutions such as solar, geothermal, biomass and industrial thermal rejection that can be used to meet the priority needs of these populations. These sources can not, however, be converted economically into electricity by the conventional Rankine cycle because of their low temperature heat [3]. However, there are other thermodynamic conversion cycles compatible with these sources such as the Organic Rankine (ORC) cycle [4], the Kalina cycle [5], the

Supercritical cycle [6], the Goswami cycle [7] And the trilateral cycle [8]. Among them, the organic Rankine cycle is however less complex and requires less maintenance with a competitive investment cost compared to other cycles [3]. Several studies have been carried out on low temperature ORC systems [9], [10], [11], [12] and [13] studied the different configurations of a low power ORC cycle designed for the production of electricity. [14] and [15] evaluated the performance of cooling systems coupled to Rankine engines. [17], [18] and [19] proposed the use of ORC technology for desalination of seawater. [20], [21] and [22] studied and analyzed the Performance of ORC systems for the recovery of waste heat. [23] and [24] also studied several configurations of modified ORC cycles to increase yield.
The performance and economic profitability of an organic Rankine cycle are related to the thermo-physical properties of the working fluid [19]. [25] analyzed the relationship between the properties of working fluids and the economic and thermodynamic profitability of an ORC cycle from a theoretical and analytical point of view. Their results showed that poor choice could lead to a less efficient and costly cycle. Several authors have studied the performance of different working fluids in order to select the optimum working fluid for the organic Rankine cycle. [26] analyzed the performance of different working fluids as a function of the operating conditions, in particular the evaporation pressure and the condensation temperature. [3] summarized the selection criteria for working fluids and studied the influence of these properties on the performance of the ORC cycle. [27] analyzed the thermodynamic characteristics and performance of 20 working fluids of a low temperature ORC cycle. Fluids with a critical temperature above 75 ° C were studied. [28] studied 13 working fluids for an ORC cycle in order to optimize the total exchange surface of the exchangers. [29] studied the performance of an ORC cycle for electricity production using working fluids with low global warming potential (GWP).
This work was interested in standard ORC cycles, ie without recuperator. In recent years ORC cycles with recuperator have been developed in order to increase the performance of ORC systems. Several authors have studied the performance of different working fluids for this type of system. [30] compared the energy and exergy performance of several working fluids for an ORC cycle with recuperator. [31] evaluated experimentally and compared the performance of different working fluids in an ORC cycle with recuperator. Fluids with a critical temperature greater than 150 °C were studied. To our knowledge, the performances of working fluids whose critical temperature is less than 150 °C of an ORC cycle with recuperator have never been studied. The objective of this study is to compare and classify five working fluids of critical temperature between 80 °C and 130 °C of a 3 kW ORC cycle with recuperator according to energy, exergy, environmental and safety criteria.

2. Material and Methods

Materials

The ORC system proposed consists of a diaphragm type pump, three evaporator heat exchangers, recuperator and condenser and a micro turbine / Scroll compressors, as shown in Figure 1.

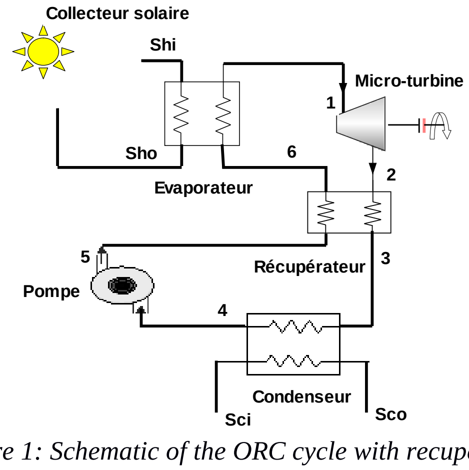

Figure 1: Schematic of the ORC cycle with recuperator

The micro-turbine considered here is similar to that used in the work of [32] and [33]. The pump supplies working fluid to the evaporator where it is heated and vaporized by the heat transfer fluid coming from the solar collector. The enthalpy of the high-pressure steam produced at the inlet of the micro-turbine is then converted into work. The low pressure steam at the outlet of the micro-turbine is directed towards the recuperator where it is cooled and then towards the condenser where it is liquefied. The available liquid is then reinjected into the evaporator by the pump to start a new cycle. The whole process described above is shown in the diagram T-s in figure 2.

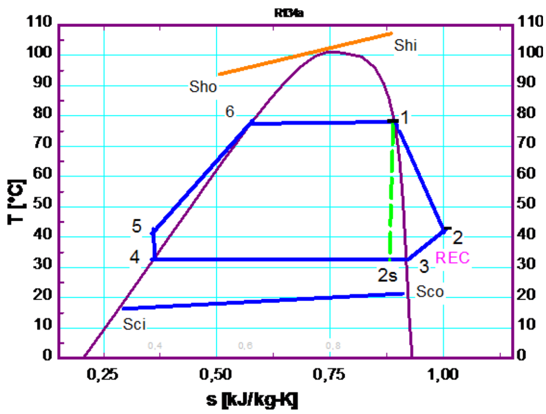

Figure 2: ORC cycle with recuperator

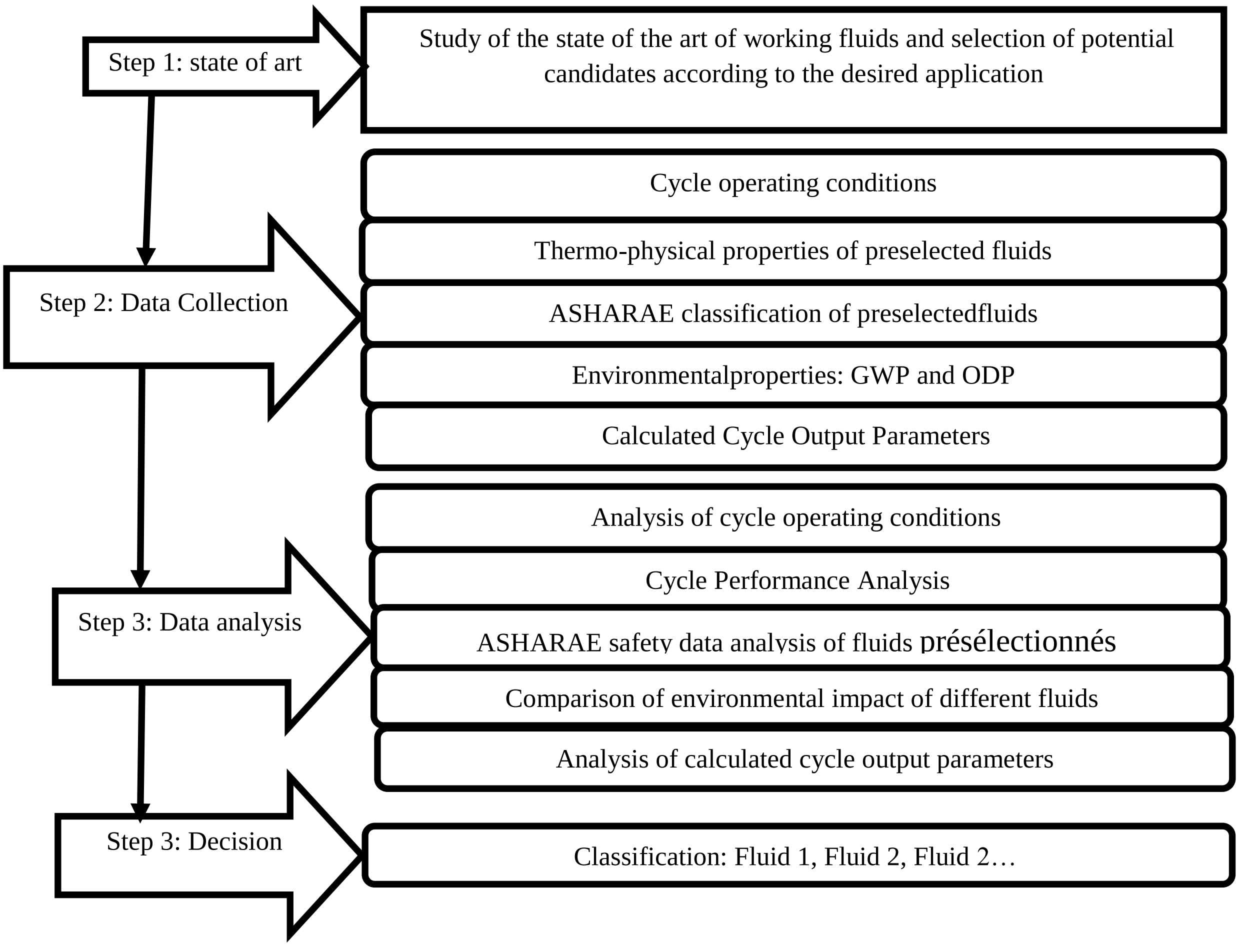

Figure 3: Methodology for selecting a working fluid

Methodology for selecting a working fluid

In practice, there is no working fluid that meets all environmental, safety and performance criteria. A compromise must therefore be found for each application. In this work the selection of potential working fluids is carried out in several steps. A preliminary selection was made on the basis of critical temperature, operating pressure, environmental and safety criteria. Eleven working fluids were then pre-selected. These fluids were also subjected to a comparison of other criteria. According to [34] the temperature difference between the critical working fluid temperature and the maximum operating temperature of the cycle should not exceed 10-15 ° C. According to [24] the desired maximum operating temperature must be less than 0.96 times the critical working fluid temperature. On the basis of these two criteria we made a second selection. The fluids retained are given in Table 2. In the following, we will compare the various pre-selected working fluids on the basis of energy efficiency, exergy efficiency, cycle heat input, total irreversibility, volume flow, mass flow rate, Pressure ratio, toxicity, flammability, ODP (Potential for Ozone Depletion) and GWP (Global Warming Potential). The general procedure for the selection of potential working fluids is illustrated in the diagram in fig. 3.

Table 1: Physical properties, environmental and safety data of pre-selected work fluids

Fluid	T_{cri} [°C]	P_{cri} [MPa]	P_{max} [MPa]	P_{min} [MPa]	PR	Groupe de securité*	ODP	GWP (100 ans)
R290	96.68	4.247	2.85	1.218	2.34	A3	0	~20
R500	102.1	4.17	2.74	1.00	2.74	A1	0.74	8100
R152a	113.3	4.520	2.108	0.794	2.65	A2	0	124
R717	132.3	11.333	3.709	1.351	2.74	B2	0	<1
R134a	101	4.059	2.366	0.887	2.66	A1	0	1430

Table 2: ASHARAE Classification of Working Fluids *[35]

Increase Of flammability	A3	B3
	A2	B2
	A1	B1
	Increased toxicity	

Modeling of the ORC cycle

The various components of the ORC cycle are open systems that exchange matter constantly. For each component of the cycle it is associated with a corresponding control volume. The application of the principles of conservation of mass and energy to a control volume between the instant t to t + Δt can be reduced, under steady conditions, to the following equations:

$$\begin{cases} \sum_e \dot{m}_e + \sum_s \dot{m}_s = 0 \\ \dot{Q} + \dot{W} + \sum_{e,s} \dot{m}_{e,s}\left(h + \frac{1}{2}V^2 + gz\right) = 0 \end{cases} \qquad (1)$$

For most thermal machines in steady state with unidirectional flow, variations of kinetic and potential energy are often negligible, equation (1) is further simplified:

$$\dot{Q} + \dot{W} + \dot{m}(h_e - h_s) = 0 \qquad (2)$$

Equation (2) will be particularly used in the study of the ORC cycle . Indeed, in components that only exchange heat with the external environment, $\dot{W} = 0$ and (2) becomes:

$$\dot{Q} = \dot{m}(h_s - h_e) si \dot{W} = 0 \qquad (3)$$

On the other hand, for adiabatic transformations where there will be mechanical energy production , $\dot{Q} = 0$ and (3) becomes:

$$\dot{W} = \dot{m}(h_s - h_e) si \dot{Q} = 0 \qquad (4)$$

The equation of exergy destruction rate (or irreversibility flow) for a steady-state ORC cycle can be expressed as:

$$\dot{I} = T_0\left(\sum \dot{m}_s \dot{s}_s - \sum \dot{m}_e \dot{s}_e - \sum \frac{\dot{Q}_k}{T_k}\right) \qquad (5)$$

For the supposed adiabatic components this equation can be reduced to:

$$\dot{I} = \dot{T}_0\left(\sum \dot{m}_s \dot{s}_s - \sum \dot{m}_e \dot{s}_e\right) \qquad (6)$$

Equations energy and exergy of the ORC cycle

For the two components considered adiabatic (the pump and the turbine), the power exchanged with the outside is given by an adaptation of equation (4):

Detente in the micro-turbine:

$$\dot{W}_t = \dot{m}(h_2 - h_1)\eta_{mt} \qquad (7)$$

Flow rate of irreversibility (exergy destroyed in the micro-turbine):

$$\dot{I}_t = T_0\dot{m}(s_2 - s_1) \quad (8)$$

Mechanical power of the pump:

$$\dot{W}_p = \dot{m}(h_5 - h_4)/\eta_p = \dot{m}v_4(P_5 - P_4)/\eta_p \quad (9)$$

Flow rate of irreversibility (exergy destroyed in the pump):

$$\dot{I}_p = T_0\dot{m}(s_5 - s_4) \quad (10)$$

For the two heat transfer components, there is no exchange of work with the outside so that equation (3) is used to obtain the calorific power exchanged with the outside:

Thermal power received by the working fluid (evaporator):

$$\dot{Q}_e = \dot{m}(h_1 - h_5) \quad (11)$$

Flow rate of irreversibility (exergy transported between inlet and outlet of the evaporator):

$$\dot{I}_e = T_0[\dot{m}_h(sh_0 - sh_i) - \dot{m}(s_5 - s_1)] \quad (12)$$

Thermal power discharged to the cold well (condenser):

$$\dot{Q}_c = \dot{m}(h_3 - h_4) \quad (13)$$

Flow rate of the irreversibility (exergy transported between the inlet and the outlet of the condenser):

$$\dot{I}_e = T_0[\dot{m}_c(sc_0 - sc_i) - \dot{m}(s_4 - s_3)] \quad (14)$$

Thermal power regained at recuperator:

$$\dot{Q}_r = \dot{m}(h_2 - h_3) \quad (15)$$

Flow rate of the irreversibility (exergy transported between the inlet and the outlet of the recuperator):

$$\dot{I}_r = T_0\dot{m}[(s_6 + s_3) - (s_2 + s_5)] \quad (16)$$

ORC cycle performance:

The performance of the ORC system is determined by the following equations:

Mechanical power:

$$\dot{W}_{net} = \dot{W}_t - \dot{W}_p \quad (17)$$

Thermal efficiency (efficiency in the sense of the first thermodynamic principle):

$$\eta_{\mathrm{I}} = \frac{\dot{W}_{net}}{\dot{Q}_e} \quad (18)$$

The total irreversibility of the cycle:

$$\dot{I}_{tot} = \sum_i \dot{I}_i = \dot{I}_t + \dot{I}_e + \dot{I}_c + \dot{I}_p + \dot{I}_r \quad (19)$$

Yield in the sense of the second principle of thermodynamics:

$$\eta_{\mathrm{II}} = \frac{\eta_{\mathrm{I}}}{(1 - {T_c}/{T_h})} \quad (20)$$

$T_0, T_c\, and\, T_h$ are the reference, cold source and hot source temperatures respectively.

The output volume flow of the turbine Vt2 determines the size of the turbine and influences the cost of the system. Therefore, working fluids with low volume flow are preferred for economic reasons.

$$V_{t2} = \frac{\dot{m}}{\rho_2} \quad (21)$$

Where $\dot{m}$ and ρ2 are the mass flow and the density at state point 2.

Due to simplify the complexity of the thermodynamic model. The operating conditions of the ORC cycle are given in Table 1 with the characteristics of the micro-turbine and of the pump. The heat transfer fluid at 80 °C is supplied by the solar collectors. The condenser is cooled by ambient air. It is assumed that the system is located in a rural area in Mauritania where the average monthly ambient temperature is about 30 °C. The temperature of the hot source can vary from 60 °C to 100 °C.

Table 3: The input data for the ORC model analysis

Evaporation temperature	Te	80°C
Condensation temperature	Tc	30°C
Mechanical efficiency of the micro-turbine	η_{mt}	0.63
Isentropic efficiency of the micro-turbine	η_{st}	0.7
Pump output	η_p	0.8
Reference temperature	T_0	25°C

3. Results and Discussions

Table 4 shows the results of comparison of the performances of the various working fluids of the ORC system for a power of 3 kW.

Table 4: Comparison of the performances of the different working fluids for a power of 3 kW

Fluide	P_{max} [MPa]	$\dot{m}$ (kg/s)	η_I [%]	η_{II} [%]	I_{tot} [kW]	Q_e [kW]	V_{t2} [m^3/h]	x_2 [%]
R290	2.85	0.147	5.87	10.44	12.88	51.09	21.82	92.1
R500	3.735	0.290	5.92	10.54	12.49	50.62	24.34	92.9
R152a	2.108	0.174	5.9	10.52	12.4	50.69	28.34	92.1
R717	3.709	0.042	6.20	11.03	11.82	48.07	15.1	84.5
R134a	2.366	0.280	5.75	10.23	12.98	55.02	26.26	93.4

Moderate pressures in the cycle

High pressure at the inlet of the turbine leads to problems of mechanical stresses. According to [36] moderate steam pressures in the range of 0.1-2.5 MPa and a pressure ratio (PR) of about 3.5 is reasonable. From Table 4, the fluids R152a, R134a have low pressure values in the condenser. Fluids R500, R290 and R717 have pressures greater than 2.5 MPa in the evaporator. All fluids are characterized by a pressure ratio of less than 3.5 MPa. The fluids R152a and R134a meet the criteria cited above and are therefore the best candidates from the point of view of the moderate vapor pressures in the cycle.

Volume flow at the output of the micro turbine

The results in Table 4 show that the R717 has the lowest volume flow rate. A fluid with a low volume flow is preferable for two reasons. On the one hand it allows to choose a micro-turbine of reduced size, and on the other hand it minimizes the losses of loads of connection pipes. Figure 6 shows the evolution of the volume flow as a function of the inlet temperature of the micro turbine. It can be seen that as the inlet temperature of the micro-turbine increases, the volume flow decreases. Figure 4 shows that the fluid R717 has the lowest volume flow rate whatever the temperature of the inlet of the turbine.

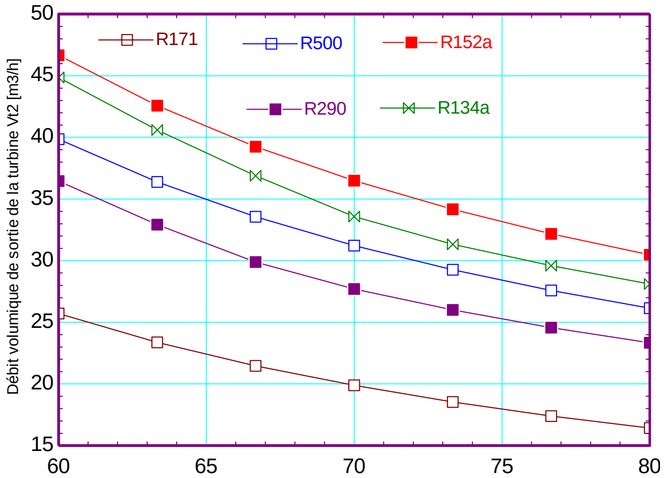

Figure 4: Volume flow vs input temperature for various working fluids at Tc = 30 °C

Energy efficiency of the system

The results in Table 4 show that the energy efficiency of the ORC cycle varies from 5.75% to 6.2%. Figure 5 shows the evolution of the energy efficiency as a function of the inlet temperature of the micro turbine. It shows that an increase in the inlet pressure of the micro turbine results in an increase in the energy efficiency of the system. Figure 5 shows that the R152a has the highest efficiency. For high inlet pressures of the micro-turbine, the R717 becomes more effective beyond these pressures. R717 and R152a are more efficient in view of the effectiveness. It is also observed that the boiling temperature is not a sufficient criterion to judge the effectiveness of the fluid contrary to the results found by [24].

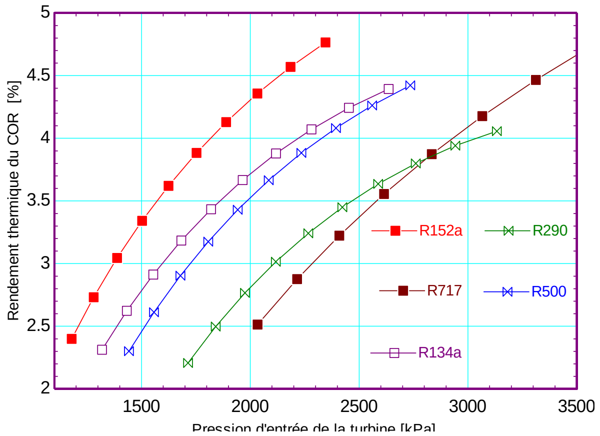

Figure 5: Thermal efficiency vs input pressure for various working fluids at Tc = 30 °C

Exergetic performance of the system

From Table 4 the exergy yield varies from 11.03% (R717) to 7.23% (R134a). The R134a fluid has the lowest exergy efficiency due to its high irreversibility, which is 12.98 kW. Figure 6 illustrates the evolution of the exergy efficiency as a function of the input pressure of the micro turbine. It shows that the R152a fluids have the highest exergy efficiency. For inlet pressures of the micro-turbine higher than the maximum pressure of R152a, the R717 becomes more efficient. This justifies that the boiling temperature is not the only criterion to judge the effectiveness of the fluid contrary to the results found by [24].

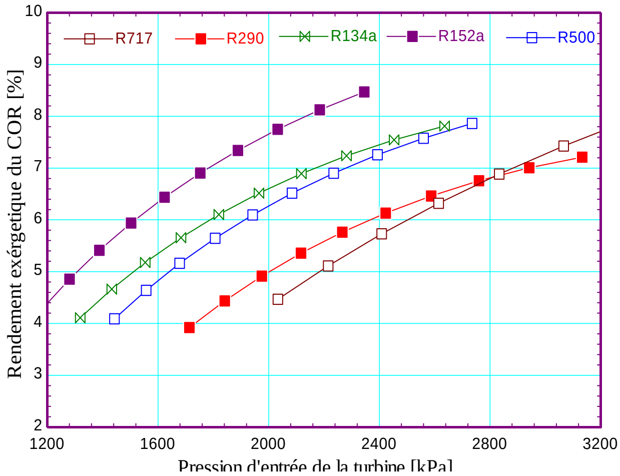

Figure 6: Exergy efficiency vs input pressure for various working fluids at Tc = 30 °C

Flow of irreversibility

The results in Table 4 show that the total irreversibility rate of the ORC cycle varies in the range 11.82 to 12.98 kW. A low irreversibility rate maximizes the exergy efficiency of the ORC cycle. Fluids R717 and R134a give the highest and lowest rates of irreversibility, respectively. Figure 7 shows the distribution of the irreversibility of the various components and for different fluids. The evaporator and recuperator contribute 74% and 10% of the total irreversibility of the cycle, respectively, followed by the turbine (Figure 8). Figures 9 and 10 show the evolution of the temperature and input pressure of the micro-turbine on the total irreversibility of the cycle. The total irreversibility of the system decreases when the pressure at which the temperature at the inlet of the micro turbine increases. The lowest irreversibility flow rate is obtained for R152a above the maximum pressure of R152a (Figure 9). Figure 10 shows that the R7171 has the lowest irreversibility rate irrespective of the variation of the temperature at the input of the micro turbine.

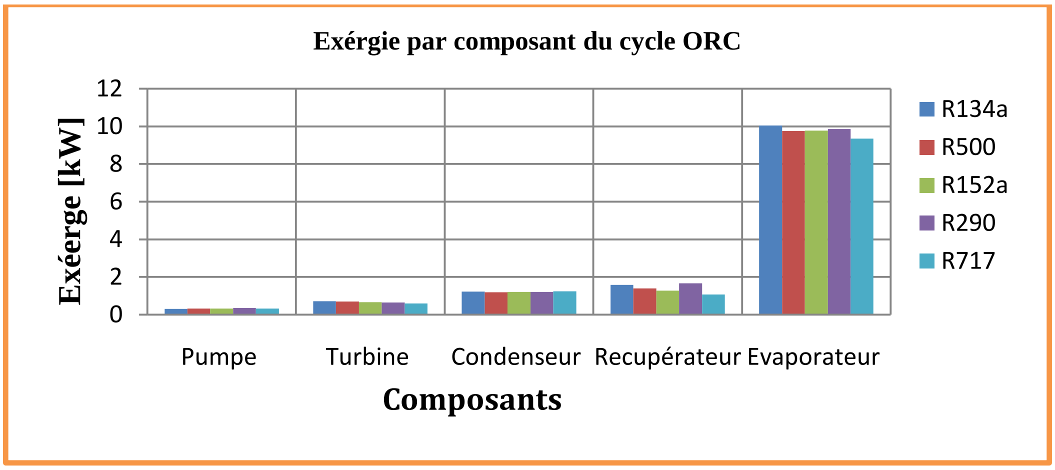

Figure 7: Exergy of each component for various fluids

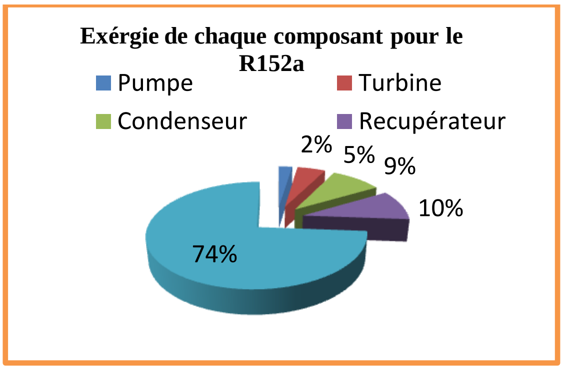

Figure 8: Exergy of each component for working fluids R152a

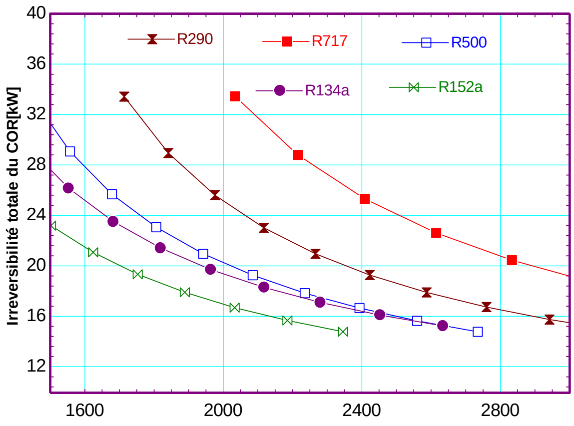

Figure 9: Total irreversibility vs input pressure for various working fluids at Tc = 30 °C

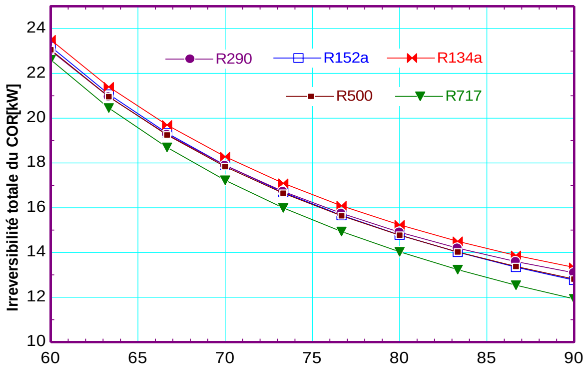

Figure 10: Total irreversibility vs input temperature for various working fluids at Tc = 30 °C

Mass flow rate of the cycle

The results in Table 4 show that the R717 has the lowest mass flow rate. A low mass flow rate is advantageous and leads to a low heat input. Figure 11 shows the evolution of the mass flow rate as a function of the inlet temperature of the micro turbine. The mass flow rate decreases as the temperature at the inlet of the micro turbine increases. Figure 11 shows that the R717 has the lowest mass flow regardless of the variation in temperature at the inlet of the micro turbine. For economical reasons, fluids with low mass flow rates are particularly advantageous, especially for high capacity.

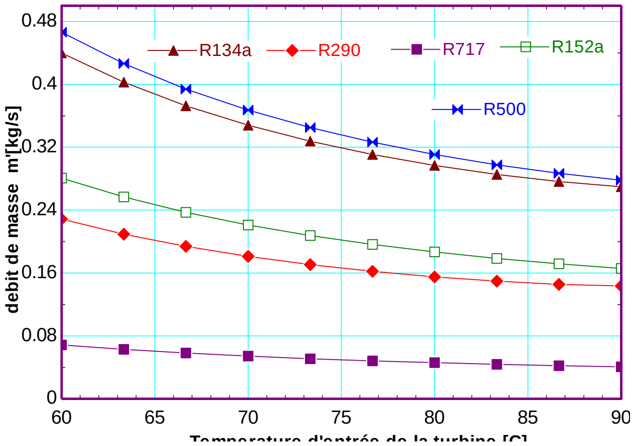

Figure 11: Variation of the mass flow rate as a function of the inlet temperature of the micro turbine

Supply of heat from the hot source

According to Table 4, the heat input needed to produce 3 kW varies from 51.01 kW (R717) to 54.95 kW (R134a). Figure 12 shows the evolution of the thermal power of the hot source as a function of the inlet temperature of the micro turbine. It can be seen that the power decreases as the inlet temperature of the turbine increases. Figure 12 shows that the R717 has the smallest supply of heat regardless of the variation in temperature. Low input power from the hot source minimizes the surface area of the solar collector and is an important part of the cost of the overall system.

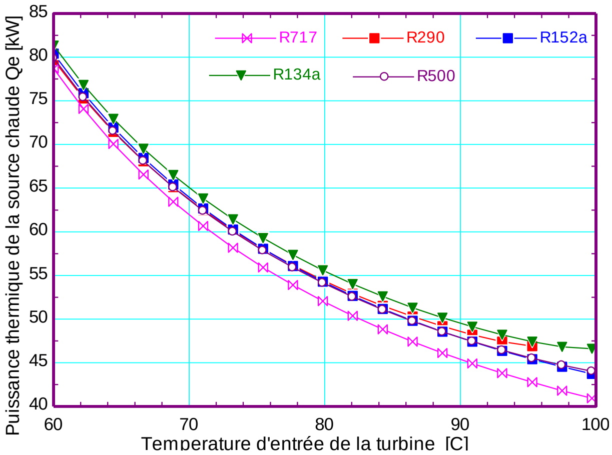

Figure 12: Variation of heat input vs the inlet temperature of the turbine

Title of Steam

The results in Table 4 show that the vapor pressure of the R717 fluid is very low, which is a disadvantage and leads to the presence of droplets during the expansion process.

Safety and environmental impact

Some substances, mainly refrigerants, deplete the ozone layer and / or contribute to global warming. Because of their negative effects, there is a need to choose those that have less harmful effects on the environment. Table 4 shows the classification of the various pre-selected working fluids on the basis of environmental and secretarial criteria. It shows that the R500 fluid is very harmful because of its non-zero ODP and excluded from the selection. Fluids R717 and R290 have a low GWP followed by R152a and R134a with a somewhat high GWP. According to the ASHAR classification, R134a is the best non-flammable or toxic candidate (A1) followed by low-flammability and non-toxic R152a (A2). R290 (A3) more flammable than R152a and non-toxic. R717 (B2) slightly flammable and toxic.

Finally, Table 5 gives the optimal classification of different working fluids suitable for ORC applications from 80 ° C to 130 ° C

Table 5: Synthesis of Optimal Working Fluid Selection

Fluid	Pmax	$\dot{m}$	η_I	η_{II}	φ	Itot	Qe	Vt2	x2	Toxicity	Inflammability	ODP	GWP	Decision	ranking
R290														**Accepted**	**3**
R500														**Rejeted**	**5**
R152a														**Accepted**	**1**
R7117														**Accepted**	**4**
R134a														**Accepted**	**2**
Légende	bad	Good	Very good		Excellent										

3. Conclusion

This work presents the classification and performance analysis of five working fluids in a low-temperature organic solar Rankine cycle. Theoretical performances of the thermodynamic and environmental properties of these fluids were evaluated and compared. Several criteria were used for the comparison: moderate cycle vapor pressures, energy and exergy yields, mass and volume flow rates, cycle heat input, total irreversibility, steam titer at the outlet of the micro turbine, Safety and environmental data. The fluids favored by moderate vapor pressures in the cycle are: R152a, R134a and R500. This is very advantageous from the point of view of security and the cost of the system. The lowest volume flow rates were observed for R717, R290 and R152a, which is preferable for economic reasons as well. From the point of view of effectiveness, high-boiling fluids such as R152a and R717 are very effective, but the presence of droplets during the relaxation process is a disadvantage. Next International regulations (Kyoto and Montreal protocols) R500 is harmful to the environment. In conclusion, R125a appears to be the best candidate for ORC applications with temperatures ranging from 80 followed by R134a. The two fluids R290 and R717 offer excellent performance but require safety precautions because of their flammability and toxicity respectively.

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