ENERGY AND EXERGY ANALYSIS OF AN ORGANIC RANKINE CYCLE

Andreea MARIN[1], Alexandru DOBROVICESCU[2], Lavinia GROSU[3], Adina GHEORGHIAN[4]

In this paper, energy and exergy analysis is used to show the exergetic performances for an Organic Rankine Cycle (ORC). This system supplies with electrical energy an existing building with a known configuration. The ORC system is supplied with solar energy from an evacuated solar tube collector.

The use of solar energy and the ORC system instead of conventional energy sources reduces pollution and offers energy savings.

The simulations of the operating for the ORC system have been performed using several working fluids. The optimization procedure based on the exergetic analysis enabled the choice of the best working fluid. The optimization looked for reducing the exergy destruction to the key parts of the system.

The careful selection of the working fluid is essential to reduce the exergy destructed and increase the exergy efficiency for the entire installation. Throughout this work, we were able to prove that this proposed configuration of a solar powered Organic Rankine Cycle is a viable solution capable of satisfying the consumer's needs. The use of an ORC for producing mechanical work from low grade heat source is a valuable solution.

Keywords: Organic Rankine Cycle, solar eneregy, energy and exergy analysis

1. Introduction

In the current economical and energetic context, the implementation of technologies using renewable energies as heating sources is offering a double advantage: the reduction in pollution and in the fuel cost. Considering these two goals, in this paper it was studied the production of electricity through an ORC system. One of the main concerns of the modern human is to provide comfort in buildings. The main utilities are: electricity, domestic hot water, heating/cooling according to external ambiance.

The solar energy represents the heat source that is offered at a maximum temperature of 140ºC by a Solar System Collector.

[1]PhD., Dept. of Engineering Thermodynamics, Internal Combustion Engines, Thermal and Refrigerating Equipments, University POLITEHNICA of Bucharest, Romania, e-mail: marin_andreea86@yahoo.ro

[2] Prof., Dept. of Engineering Thermodynamics, Internal Combustion Engines, Thermal and Refrigerating Equipments, University POLITEHNICA of Bucharest, Romania

[3] Prof., Laboratoire Energétique, Mécanique et Electromagnétisme, University PARIS X, France, e-mail:lgrosufr@yahoo.fr

[4] Junior Lecturer, Dept. of Engineering Thermodynamics, Internal Combustion Engines, Thermal and Refrigerating Equipments, University POLITEHNICA of Bucharest, Romania

The Organic Rankine Cycle is used for producing electricity in many plants, such as the plants in Altheim, Austria with a power production of 1MW_{el} and another plant is in Germany, Neustadt-Glewe with a power production of 0.2 MW_{el}, both are using initially n-perfluoropentane as working fluid, this fluid is not environmentally very friendly because it has a big GWP [1]. The ORC system is a promising process for conversion of low and medium temperature heat to electricity. This cycle is very similar to a steam Rankine cycle but it uses an organic fluid instead of water. The organic fluid can operate at a much lower evaporation temperature and pressure than a conventional turbine and still perform at a high efficiency [2]. These fluids are: HFC's Freon, ammonia, butane, iso-pentane, toluene and they have in general a high molecular mass.

There are several interesting characteristics and advantages in using an ORC to recover low-grade waste heat, including economic utilization of energy resources, smaller systems and reduced emissions, availability of its components and these systems need less maintenance.

Another advantage is that the ORC fluid can operate at low temperatures 70°- 90°C. At this temperature, a conventional steam turbine is inefficient. In the Organic Rankine Cycle is not necessary to superheat the fluid like in the case of the conventional steam power plant [3].

Many investigations about ORC were carried out and can be classified into three main categories according to their application domains. The first is geothermal utilization. Hettiarachchi et al. [4] evaluated optimum cycle performance when the evaporation and condensation temperatures and geothermal and cooling water velocities were varied. He compared the results obtained with working fluids that included ammonia, R 123, n-pentane, and PF5050. Saleh et al. [5] investigated 31 components working fluids for sub-critical and supercritical ORCs for geothermal power-plants. The second category reviewed is solar energy harnessing. Yamamoto et al. [6] estimated the optimum operating conditions of the ORC comparing R123 and water as working fluids. Tchanche et al. [7] comparatively analysed 20 working fluids for use in low-temperature solar organic Rankine cycle systems. The final category considered is low-grade waste heat recovery. Hung et al. [8] analysed the ORC efficiency using refrigerants benzene, ammonia, R11, R12, R134a and R113 as working fluids. Maizza and Maizza [9] investigated the thermodynamic and physical properties of 21 conventional fluids used in Organic Rankine Cycles heated by waste energy sources. Liu et al. [10] reviewed the effects of 10 values working fluids on the thermal and total heat recovery efficiencies of the ORC. Wei et al. [11],[12] analysed system performance and optimised the working conditions of an ORC using R245fa when a simulation of a dynamic model. Drescher and Bruggemann [13] developed software to find thermodynamically suitable fluids for ORC in biomass power and heat plants.

The efficiency of an ORC has estimated to be between 10 and 20 % depending on temperature levels of evaporator and condenser. Nowadays, the ORC systems are commercially available with a power between some kW until 3 MW. ORC uses the same components as a conventional steam turbine.

2. Description of the system

The analysed system has the following components: preheater, evaporator, superheater, turbine, condenser and a pump. In the heat exchanger the pumped ORC working fluid (R365mfc, Solkatherm SES 36 and R123) is preheated, evaporated and superheated before to enter in the ORC turbine. The exhausted vapour from the turbine is directed to the condenser where is cooled by cooling water, figure 1.

The solar energy is collected in an evacuated solar tube collector. This energy is utilized for heating an intermediary fluid, in our case water. The maximum reached by the heat carrier is 140°C. The electrical power need was estimated from an energy balance of the building by summarizing the different types of heat gains/losses: thermal losses through the walls, through thermal bridges and glass surfaces, inputs from occupants, lighting, fresh air and different types of equipments.

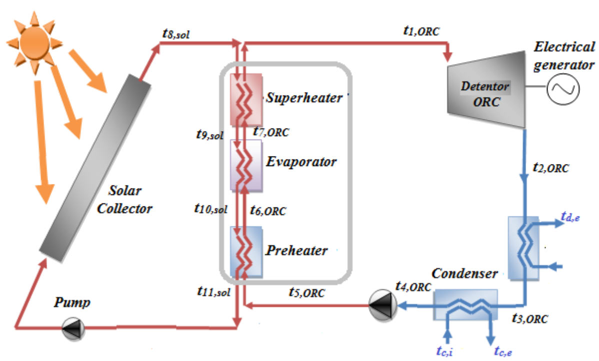

Fig. 1. Simplified scheme of the ORC system

3. Mathematical method

The solar energy is provided in an evacuated solar tube collector that operates in a range of temperatures 80° - 180°C.

The absorber surface temperature for the solar collector [14]:

$$T_{Ab}{}^{4} = (1 - \eta_{sol}) \frac{A_{Ap} S}{A_{Ab} \sigma} \quad (1)$$

The collector efficiency can be calculated with the following relation:

$$\eta_{sol} = \eta_0 - \frac{a_1 (T_m - T_0)}{G} - \frac{a_2 (T_m - T_0)^2}{G} \quad (2)$$

The optical efficiency (η_o) ,thermal loss coefficients, a_1 and a_2 have been set equal 0.71, 0.95, 0.005 following constructor recommendations. The solar flux is 800 W/m^2.

In this paper the optimization of the collectors and simulation for several fluids (table1) were done with the Thermoptim software and SOLKANE 8.0 [15][16] .

The choice of a working fluid that fulfils all the criteria required by the ORC system is a very difficult task. Jorge Façao et al. [3] examined the effects of using different agents on the performance of the ORC system from a thermal and an economical point of view taking also into account the fluid toxicity. The selection of the working fluids has a significant role in the design of the ORC systems. The strategy for choosing the suitable ORC fluid is given in Figure 2.

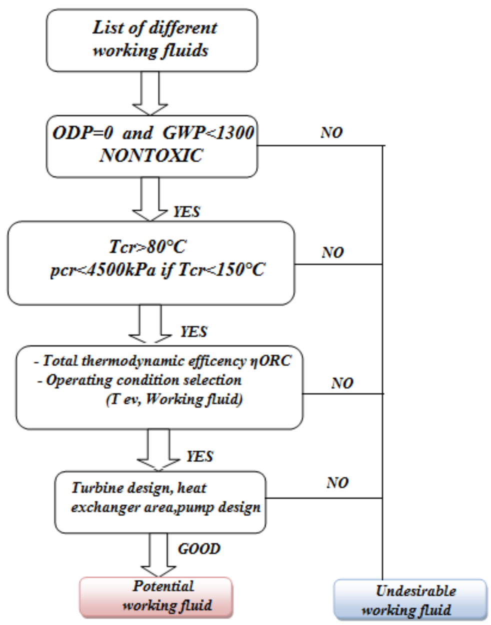

Fig 2. Screening method for fluid selection

Table 1.

Thermo-physical proprieties of different working fluids

Working fluids	M [kg/kmol]	p_c [bar]	T_c [K]	v_c [dm^3/kg]	ODP	GWP 100	Chemical formula
Solkane R 365mfc	*148.09*	*32.66*	*460*	*2.110*	*0*	*825*	$CF_3CH_2CF2CH3$
Solkatherm SES 36	*184.85*	*28.49*	*450.7*	*1.859*	*0*	*-*	*CF3CH2CF2CH3+PFPE*
R 123	*152.931*	*36.74*	*456.94*	*1.819*	*0.02*	*93*	*CHCI2CF3*

Characteristics of the fluids such as molar mass, critical pressure and temperature are shown in the Table 1.

It can be concluded that for a fixed heat source, the working fluid with the lower critical temperature possesses a superior performance. The critical temperature must lie well above the highest operating temperature of the cycle to minimize the irreversibility generated by heat transfer across a finite temperature difference within the evaporator. The working fluid should be non-corrosive, thermally and chemically stable at all operating temperatures and pressures.

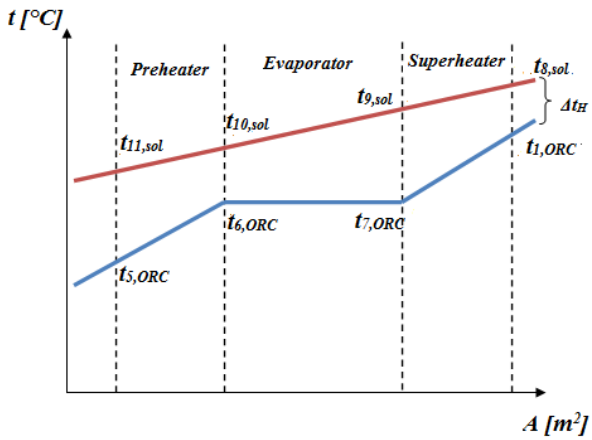

Fig 3. Heat exchanger diagram

Table 2.

Input parameters for the ORC system

Parameters	$t_{8,sol}$	Δt_H	Δt_{solHE}	$\dot{m}_{sol}$	$\Delta\ p_{sol}$	Δt_{sh}	$t_{4,ORC}$	$\dot{Q}_{sol}$	η_T	η_p
Unit	*°C*	*°C*	*°C*	*kg/s*	bar	*°C*	*°C*	kW	-	-
Value	140	20	40	0.14	0.5	10	30	23.4	0.85	0.6

Taking into account of the simulation, the parameters from Table 2, the simulation is made for 3 fluids. The results are compared in Table 3.

Table 3.

Mechanical power and efficiency of the ORC system for different working fluids versus high pressure

Fluids	p_{1ORC} ***[bar]***	$\dot{m}_{ORC}$ ***[kg/s]***	$\dot{W}_{ORC}$ ***[kW]***	η_{ORC} ***[%]***
R 365mfc	11.43	0.0787	3.612	15.43
SES 36	12.05	0.1048	3.557	15.2
R123	14.57	0.1007	3.785	16

Tables 2 and 3 show that the highest performance is obtained for the ORC with R123 but is not achieved the first step from the screening method because the ODP is 0,02. Taking all constraints into account, efficiency, toxicity, price, critical temperature and turbine inlet pressure, Solkatherm SES 36 recommends itself as a non-toxic and no-risk fluid. SES 36 is the best solution for a solar installation with ORC.

SES36, is recommended for its high η_{ORC}, mechanical work of turbine and its friendly behaviour in contact with environment.

3.1.1 Energy analysis

Two principal assumptions are considered: 1) the system reaches a steady state, and 2) pipe pressure drop and heat losses to the environment are neglected.

Equations used to perform the energy analysis are [17]:

$$\textit{Preheater:} \quad \dot{Q}_{\mathrm{Pr}e} = \dot{m}_{ORC}(h_{6,ORC} - h_{5,ORC}) \tag{3}$$

$$\textit{Evaporator:} \quad \dot{Q}_{Ev} = \dot{m}_{ORC}(h_{7,ORC} - h_{6,ORC}) \tag{4}$$

$$\textit{Superheater:} \quad \dot{Q}_{Sh} = \dot{m}_{ORC}(h_{1,ORC} - h_{7,ORC}) \tag{5}$$

$$\textit{Turbine:} \quad \dot{W}_{T} = \dot{m}_{ORC}(h_{2,ORC} - h_{1,ORC}) \tag{6}$$

$$\textit{Condenser:} \quad \dot{Q}_{Cd} = \dot{m}_{ORC}(h_{4,ORC} - h_{2,ORC}) \tag{7}$$

$$\textit{Pomp:} \quad \dot{W}_{P} = \dot{m}_{ORC}(h_{5,ORC} - h_{4,ORC}) \tag{8}$$

$$\dot{Q}_{HE} = \dot{Q}_{\mathrm{Pr}e} + \dot{Q}_{Ev} + \dot{Q}_{Sh} \tag{9}$$

$$\textit{Net power:} \quad \dot{W}_{ORC} = \dot{W}_{D} - \dot{W}_{P} \tag{10}$$

Thermal efficiency is:

$$\eta_{ORC} = \frac{\dot{W}_{D} - \dot{W}_{P}}{\dot{Q}_{HE}} \tag{11}$$

3.1.2 Exergy analysis

The exergy analysis is employed to evaluate the performance of the system based on the irreversibility that is occurring in every component of the ORC system (as non-isentropic expansion and compression) [18]

Heat exchanger

$$\textit{Product:} \quad \dot{E}x_{HE}^{P} = \dot{E}x_{\mathrm{Pr}e} + \dot{E}x_{Ev} + \dot{E}x_{Supra} \tag{12}$$

$$\textit{Fuel:} \quad \dot{E}x_{HE}^{F} = \dot{E}x_{Q_{HE}}^{T_{mH}} = \dot{Q}_{HE}\left(1 - \frac{T_0}{T_{mH}}\right) \tag{13}$$

$$\dot{E}x_{\mathrm{Pr}e} = \dot{m}_{ORC} \cdot (h_{6,ORC} - h_{5,ORC} - T_0(s_{6,ORC} - s_{5,ORC})) \tag{14}$$

$$\dot{E}x_{ev} = \dot{m}_{ORC}(h_{7,ORC} - h_{6,ORC} - T_0(s_{7,ORC} - s_{6,ORC})) \quad (15)$$

$$\dot{E}x_{Supra} = \dot{m}_{ORC}(h_{1,ORC} - h_{7,ORC} - T_0(s_{1,ORC} - s_{7,ORC})) \quad (16)$$

$$\dot{E}x_{HE}^{P} = \dot{m}_{ORC}(h_{1,ORC} - h_{5,ORC} - T_0(s_{1,ORC} - s_{5,ORC})) \quad (17)$$

$$\dot{I}_{HE} = \dot{E}x_{HE}^{F} - \dot{E}x_{HE}^{P} \quad (18)$$

$$\dot{I}_{HE} = \dot{Q}_{HE}(1 - \frac{T_0}{T_{mH}}) - \dot{m}_{ORC}(ex_{1,ORC} - ex_{5,ORC}) \quad (19)$$

Where $T_{mH} = \dfrac{t_{8,sol} - t_{11,sol}}{\ln(\dfrac{T_{8,sol}}{T_{11,sol}})}$ (20)

$$Ir_{HE} = \frac{\dot{I}_{HE}}{\dot{E}x_{HE}^{F}} \cdot 100 \quad (21)$$

Condenser

Product: $\dot{E}x_{Cd}^{P} = \dot{E}x_{Q_{Cd}}^{T_{mW}} = \dot{Q}_{Cd}(1 - \dfrac{T_0}{T_{mW}})$ (22)

Fuel:

$$\dot{E}x_{Cd}^{F} = \dot{m}_{ORC}(h_{4,ORC} - h_{3,ORC} - T_0(s_{4,ORC} - s_{3,ORC}) + h_{3,ORC} - h_{2,ORC} - T_0(s_{3,ORC} - s_{2,ORC})) \quad (23)$$

$$\dot{I}_{Cd} = \dot{m}_{ORC}(ex_{4,ORC} - ex_{2,ORC}) - \dot{Q}_{Cd}(1 - \frac{T_0}{T_{mW}}) \quad (24)$$

Where: $T_{mW} = \dfrac{h_{2,ORC} - h_{4,ORC}}{s_{2,ORC} - s_{4,ORC}}$ (25)

$$Ir_{Cd} = \frac{\dot{I}_{Cd}}{\dot{E}x_{HE}^{F}} \cdot 100 \quad (26)$$

Turbine:

Product: $\dot{E}x_{D}^{P} = \dot{W}_{D}$ (27)

Fuel : $\dot{E}x_{D}^{F} = \dot{m}_{ORC}(h_{2,ORC} - h_{1,ORC} - T_0(s_{2,ORC} - s_{1,ORC}))$ (28)

$$\dot{I}_{D} = \dot{m}_{ORC} \cdot T_0(s_{2,ORC} - s_{1,ORC}) \quad (29)$$

$$Ir_{D} = \frac{\dot{I}_{D}}{\dot{E}x_{HE}^{F}} \cdot 100 \quad (30)$$

Pump:

Product: $\dot{E}x_P^P = \dot{m}_{ORC}(h_{5,ORC} - h_{4,ORC} - T_0(s_{5,ORC} - s_{4,ORC}))$ (31)

Fuel: $\dot{E}x_P^F = \dot{W}_P$ (32)

$$\dot{I}_P = \dot{m}_{ORC} \cdot T_0(s_{5,ORC} - s_{4,ORC}) \quad (33)$$

$$Ir_P = \frac{\dot{I}_P}{\dot{E}x_{HE}^F} \cdot 100 \quad (34)$$

The exergy efficiency of ORC system evaluates the performance for waste heat recovery and is calculated below:

$$\eta_{exORC} = \frac{\dot{W}_D - \dot{W}_P}{\dot{E}x_{Q_{HE}}^{T_{mH}}} \quad (35)$$

4. Results and discussions:

The analysis was performed under the following assumptions for every working fluid:

- The temperature from the solar collector ($t_{8,sol}$) is 140°C;
- The ambient temperature is considered to be 20°C;
- The isentropic efficiency for the turbine was assumed to be 85%.
- The isentropic efficiency for the pump was assumed to be 60%.

The model described above was used to simulate the behaviour of the ORC system upon variation of the outlet temperature of the solar collector.

Table 4.

Results *for* $t_{8,sol}$=140

Parameters	*Units*	*R123*	*SES 36*	*R 365 mfc*
Qpreheater	*[kW]*	*10.96*	*12.50*	*11.83*
Qevaporator	*[kW]*	*11.39*	*9.49*	*10.39*
Qsuperheater	*[kW]*	*1.054*	*1.404*	*1.188*
Qcondenser	*[kW]*	*16.94*	*13.69*	*15.04*
Wturbine	*[kW]*	*3.942*	*3.701*	*3.725*
Wpump	*[kW]*	*0.155*	*0.144*	*0.105*
mORC	*[kg/s]*	*0.1007*	*0.1048*	*0.0787*
WORC	*[kW]*	*3.787*	*3.555*	*3.620*

Table 5.

Results for $t_{8,sol}$=140

	Heat exchanger			*Condenser*			*Turbine*			*Pump*		
	R123	*SES36*	*R236*	*R123*	*SES36*	*R236*	*R123*	*SES36*	*R236*	*R123*	*SES36*	*R236*
Ex^F *[kW]*	*5.94*	*5.94*	*5.94*	*0.56*	*0.46*	*0.48*	*4.56*	*4.23*	*4.23*	*0.155*	*0.145*	*0.105*
Ex^P *[kW]*	*5.35*	*5.42*	*5.25*	*0.42*	*0.34*	*0.37*	*3.94*	*3.70*	*3.72*	*0.005*	*0.006*	*0.003*
I [kW]	*0.59*	*0.52*	*0.69*	*0.14*	*0.12*	*0.10*	*0.62*	*0.53*	*0.50*	*0.151*	*0.138*	*0.102*

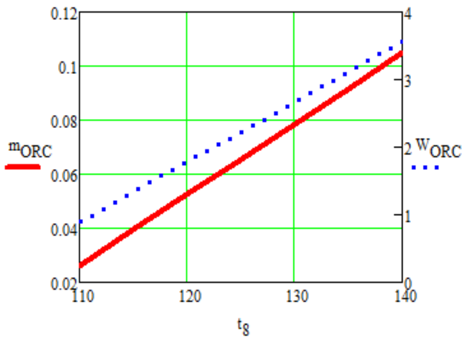

Fig 4. Variation of mass flow rate versus the high temperature from the system

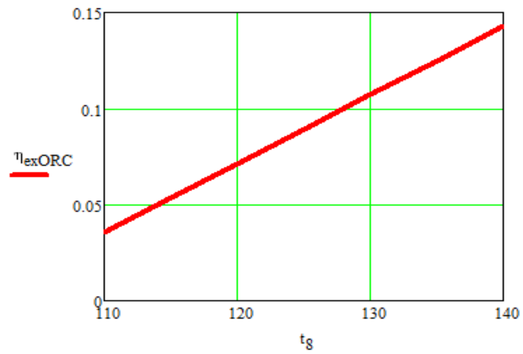

Fig 5.Exergy efficiency versus high temperature

Table 4 shows the results from the energy balance for maximum exit temperature from solar collector. Parameters from table 4 were calculated with software SOLKANE 8.0. In table 5 are illustrated the results of exergy balance and the most important exergy destructions are in the heat exchanger and expansion process. In figure 4 and 5, the simulations were made for the working fluid SES36, because it was considered more suitable from ecological point of view. Figure 4 shows that for the same heat flow in the solar collector but with an increasing exit temperature, the mechanical power and the flow of organic fluid are increasing. Heat input increases for the ORC system and also the power will increase. Exergy efficiency is decreasing with increasing of the high temperature from the solar collector, figure 5, because produced irreversibility in the heat transfer in the heat exchanger decrease.

5. Conclusions

In this paper, an Organic Rankine Cycle was studied using the first and the second law of thermodynamics.

The exergy efficiency increases, due to the decrease of the irreversibility in the evaporator process at higher temperatures from the solar collector. The increases of the irreversibility during the expansion process, that quantitatively represent, a large share of total exergy loss, have a major influence for the exergy efficiency of the system.

The experiment investigations of Organic Rankine Cycle show that using SES36 as working fluid is feasible and the performances are acceptable. This fluid is an ecological alternative for these systems.

Evolution of exergy efficiency of the ORC was followed for several input temperature. Exergy efficiency is decreasing with increasing of the high temperature, because the irreversibility from heat exchanger is decreasing.

The power generation efficiency is low at present; it could be enhanced by improving the Rankine cycle if the condensation heat could be recovered the thermal performance of the low temperature solar powered Rankine cycle would be improved greatly.

Thus in this work, exergy analysis is one more highlighted like a powerful tool for finding the optimum design and operating regime for an energetic system.

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