Optimized coupling analysis of Internal Combustion Engine (ICE)-ORC-MCRS

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Abstract. This paper refers to recovering of waste heat from hot gases discharged by internal combustion engines (ICE) at partial or full load using a Rankine Organic (ORC) cycle which drives a compressor of a mechanical compression refrigeration system (MCRS) to convert heat recovered into cooling power. The subject is of particular interest to the automotive and refrigeration industry with the aim of finding a feasible solution for the recovery of residual heat and turning it into a real source of commercial or industrial refrigeration. The heat recovery system is based on an ORC driven by SES36, a working fluid with good thermodynamic properties for the recovery of low temperature heat, as well as refrigeration or air conditioning refrigeration cycles. A scheme was proposed that combines the two systems so as to use the same working fluid, thus using a small number of subassemblies. After the energetic and exergetic calculations, it was proved that the ICE-ORC-MCRS system is of real interest due to the recorded performances, while the optimization of the thermodynamic scheme can be upgraded by improving the heat transfer at the level of the exchangers, but also constructively trying to streamline the scheme with new heat recovery.

1. Introduction
Concerns about air pollution and fuel price escalation through the 1970’s till present have led to attempts to find viable alternatives to internal combustion engines (ICE). However, low power, high cost, and low efficiency of other power cycles prevented a substantial change in propulsion of the vehicle. Instead, the focus was on waste heat recovery systems to increase cogeneration efficiency and to add engine power by converting residual heat energy into a more useful, either mechanical or electrical form. Research done in this domain has led to increased ICE efficiency. However, lowering fuel prices in the 1980s and improving engine performance have marginalized the implementation of waste heat recovery systems [1,2].

Recent works published by major car manufacturers Honda, Toyota, BMW and Volvo show renewed interest in the recovery of residual heat in order to increase the thermal efficiency of the ICE. The reason for this shift is fuel price escalation as well as future rules on CO₂ emissions. In addition, these systems are increasingly viable due to recent technological advances that have increased the efficiency of individual components in waste heat recovery systems. Among these improvements we can list the removal of leakage losses and the improvement of the rotor profiles of the expander, as well as the improvement of the heat transfer at the level of the exchangers. At present, about 30% to 40% of the combustion energy in internal combustion engines is converted into useful mechanical
work, and the remainder is the residual heat expelled to the environment through combustion gases and engine cooling systems. Recovery and use of this waste heat saves fossil fuels and reduces the amount of greenhouse gases released into the environment. Also, residual heat can be used to generate electricity by increasing the efficiency of ICE-EG (electrical generator). Rankine Organic Cycle (ORC) systems are suitable for recovery of waste heat from low and medium temperature sources [1,2].

A particular interest has been given in the literature to various ORC configurations and working fluids for recovering heat from exhaust gases and engine coolant. Shu and al. compared the performance of three ORC double-circuit regeneration systems with the performance of a simple ORC system. The residual heat of the exhaust gases, engine coolant and heat of the high temperature cycle was recovered by these four systems [3].

More recently, interest in ORC applications of lower capacities also increases, partly because 90% of global waste heat is available in units of 10-250 kW. Also, smaller modular ORC systems can have easier access to the market, with the beneficiary being able to directly use waste heat and electricity generated by ORC systems [4].

ORC technology has not yet been used on a commercial scale in small-scale power plants although smaller ORC systems have significant potential to use near the areas where heat sources exist for various applications such as: power generation for isolated homes, cogeneration plants or heat-treated heat pumps [5]. For this reason, a number of theoretical and experimental researches have been carried out in recent years with the aim of developing compact units suitable for small capacities applications [6-12].

The objective of this paper is to identify and optimize the constructive solutions of the ORC-MCRS mixed cycle coupling through an exergetic and energetic analysis. The analysis aims to find the optimal operating and design structure for the ORC-MCRS system in order to obtain the maximum refrigeration load.

2. Construction drawing and working parameters

The proposed concept for analysis combines an ORC cycle with a mechanical sulphur compression refrigeration system (MCRS) to form a thermally activated cooling system as in figure 1. The crankshaft of the ORC and MCRS compressor are coupled directly to reduce energy conversion losses.

![Figure 1. Basic scheme and thermodynamic cycle of the ORC-MCRS system proposed for analysis.](image-url)
it can produce electricity when it is not a cold needed requirement. This applies to applications where heat from the heat source is available in all seasons. During hot summer periods, all available heat can be converted into mechanical energy and then cooling power, and in winter when no cold is needed, the system can produce electricity [13].

The configuration of the ORC-MCRS installation as well as the working fluid were established following a comprehensive study of the functional schemes encountered in the literature. Working fluids were selected through a bibliographic research based on ORC-specific criteria but also good results in MCRS systems. The study and results are presented in [14], [15]. Due to the constructive peculiarities of the mixed cycle under the research of this project, the SES36 substance was chosen as the working fluid. In table 1 it can be seen the thermo-physical and environmental characteristics of the SES36 work agent where \( t_{cr} \) is the critical temperature, \( p_{cr} \) is the critical pressure, \( \text{ODP} \) is the ozone depleting potential, and \( \text{GWP}_{100yr} \) is the global warming potential for 100 years.

| Fluid name | Fluid type | \( t_{cr}[^{0}\text{C}] \) | \( p_{cr}[^{\text{bar}}] \) | \( \text{ODP} \) | \( \text{GWP}_{100\text{yr}} \) | Safety measures |
|------------|------------|----------------|----------------|---------------|----------------|----------------|
| SES 36    | Dry        | 177.55         | 28.49          | 0             | 4121           | A1             |

The input data for the optimized coupling analysis found in table 2 were taken from the internal combustion engine to equip the ICE-ORC experimental equipment in the Department of Thermotechnics, Engines, Heat & Refrigeration Equipment. The stand was designed and built to experimentally investigate the possibilities of improving the performance coefficient of an internal combustion engine by coupling it with an ORC plant to recover the heat dissipated by the engine. The execution of this stand was funded under the research contract PN-II-PT-PCCA-2011-3.2-0059 “High-efficiency micro-cogeneration hybrid group equipped with ORC electronically assisted” acronym – GRUCOHYB. The realization of the micro-cogeneration hybrid group is the subject of a patent which, upon completion of the research, will be a prototype of a cogenerative system with a high performance coefficient that can supply electricity and heat that can be used in isolated areas not connected to the national electricity grid [2,3, 14,17,18].

| Engine type | Diesel |
|------------|--------|
| Engine model | 4TNV98TGGHEHR |
| Engine manufacturer | YANMAR |
| Maximum mechanical power | 40 kW |
| Exaust gas temperature 100% load | 480 °C |
| Exaust gas masic flow | 0,0534 kg/s |
| Engine water cooling temperature | 75 °C |
| Engine water masic flow | 0,3 kg/s |
| ORC condenser water inlet temperature | 15 °C |
| ORC condenser water masic flow | 0,5 kg/s |

The working conditions, determined from the parameters of the heat source and the cooling water of the condenser, for the ORC-MCRS are shown in table 2. The maximum vaporization temperature was also imposed taking into account the restriction as the value of the flue gas.
temperature at exit from the boiler evaporator is not less than 140°C to avoid drops of sulphuric acid (H₂SO₄) in the evaporator and implicitly corrosion.

3. System apour ng

The mathematical model is based on mass, energy and exergy balances for different constructive structures of the ORC-MCRS scheme.

The importance of exergy analysis is the ability to effectively enter each process in the system and to reveal the location and extent of a malfunction. Based on the exergy analysis, a strategy can be established to improve the performance of a system by making constructive and operational changes [4].

The basic scheme of the ORC-MCRS system from which this study was started is characterized by the following thermodynamic working fluid processes: apour overheating in the ORC boiler, apour expansion to condensation pressure in the ORC Expander, condensation to saturated ORC-MCRS condenser, compressing the liquid in the ORC pump, compressing the overheated apour in the MCRS Compressor to the condensing pressure, laminating the liquid agent into the MCRS Expansion Valve(VL), vaporizing and apour overheating in the MCRS Vaporizer resulting in the plant’s useful effect.

The required data and parameters are: ambient temperature \( t_0 = 25 \, ^\circ C \); mass flow rate of gas \( \dot{m}_{ga} = 0.0534 \, kg/s \); combustion gas temperature of ICE at boiler input \( t_{iga} = 480 \, ^\circ C \); combustion gas outlet temperature in boiler \( t_{ego} = 480 \, ^\circ C \); specific heat of combustion gases \( c_{ga} = 1.14 \, kJ/kg \); condenser water outlet temperature \( t_{ew} = 22 \, ^\circ C \); evaporation temperature in the evaporator of the MCRS \( t_v = 22 \, ^\circ C \).

The analysis was based on the following assumptions:

- all processes are considered to have constant flow;
- there are no heat losses in heat exchangers;
- pressure losses in all parts of the installation are neglected;
- vaporization and condensation processes are isobar processes;
- detenting, compressor compression and pump pressure increase are processes with entropy difference;
- the analysis of system is considered to be directly engaged with the \( W_{cp} = \dot{W}_{cp} \), the power transfer being made without mechanical loss, the crankshaft of the MCRS compressor and ORC expander are coupled directly;
- the working fluid in the ORC-MCRS system is SES36;
- a temperature difference between the combustion gas outlet temperature and the working fluid vaporization temperature of the Boiler \( \Delta t_{minB} \) is considered, thus resulting in the boiling temperature of the working fluid in the Boiler \( \Delta t_B \):

\[
\Delta t_B = \Delta t_{ego} - \Delta t_{minB}
\] (1)

- a temperature difference between the water outlet temperature and the condensing temperature of the working fluid in the Condenser is considered \( \Delta t_{minCd} \), resulting in condensing temperature of the working fluid in the Condenser \( \Delta t_c \):

\[
\Delta t_c = \Delta t_{ew} - \Delta t_{minCd}
\] (2)

- choose the degree of overheating in the boiler of the ORC, \( \Delta t_{siB} \). The temperature of the working fluid results at the entrance to the expander is \( t_3 \):
\[ t_j = t_B + \Delta t_{siB} \]  

- the isentropic efficiency of the Expander \( \eta_{iD} \), the \( \eta_{sCp} \) compressor and the pump \( \eta_{sP} \) will be taken into account.

### 3.1. Energetic analysis

Based on the thermodynamic parameters in the system key states determined by the Engineering Equation Solver (EES)[16] for the SES36 working fluid, the following calculates:

- the working fluid mass flow rate of the ORC cycle:
  \[ \dot{m}_{\text{fluid ORC}} = (\dot{m}_{\text{ga}} \cdot t_{\text{ga}} \cdot \Delta t_{\text{ga}}) / (h_3 - h_2) \]  

where enthalpy values in specific points of the cycle are marked with \( h \) and \( t_{\text{ga}} \) is the temperature difference between in and out of the combustion gases in the heat exchanger:

\[ \Delta t_{\text{ga}} = t_{\text{iga}} - t_{\text{ega}} \]

then the theoretical power produced by the expander can be calculated:

\[ \dot{W}_D = \dot{m}_{\text{fluid ORC}} \cdot (h_3 - h_4) \]  

the theoretical power consumed by the working fluid drive pump:

\[ \dot{W}_P = \dot{m}_{\text{fluid ORC}} \cdot (h_2 - h_1) \]  

the theoretical power of the Boiler evaporator is calculated:

\[ \dot{Q}_B = \dot{m}_{\text{fluid ORC}} \cdot (h_3 - h_2) \]  

the mechanical work required to drive the compressor is calculated:

\[ |t_{\text{CP}}| = h_8 - h_7 \]

it is possible to calculate the mass flow of working fluid driven by the compressor:

\[ \dot{m}_{\text{fluid MCRS}} = \dot{W}_{CP} / t_{\text{CP}} \]  

calculate the theoretical power of the Condenser:

\[ \dot{Q}_{Cd} = (\dot{m}_{\text{fluid ORC}} + \dot{m}_{\text{fluid MCRS}}) \cdot (h_5 - h_1) \]

the theoretical Vaporizer power of the MCRS cycle is calculated:

\[ \dot{Q}_{Vp} = \dot{m}_{\text{fluid MCRS}} \cdot (h_7 - h_6) \]

total flow of working fluid in the ORC-MCRS plant:

\[ \dot{m}_{\text{fluid ORC MCRS}} = \dot{m}_{\text{fluid ORC}} + \dot{m}_{\text{fluid MCRS}} \]

and the theoretical performance coefficient of the ORC-MCRS cycle:

\[ \text{COP}_{\text{ORCMCRS}} = \dot{Q}_{Vp} / (\dot{Q}_B + \dot{W}_P) \].
3.2. Exergy analysis

As we observe, the energy analysis stops at the outer boundary of the system, accounting for only the input and output energy without taking into account losses, so it does not provide information on how the heat transfer can be optimized to minimize the damage. In energy analysis it matters only the quantity and makes no difference between the qualities of the products obtained such as the power produced in the Expander, the electric power or the cooling power.

Exergy expresses the true measure of the quantity and quality of a certain energy in correlation with the intensive environmental parameters. Exergy is destroyed in a specific process of energy conversion.

Minimizing exergy destruction in the key components of an energy system provides the strategy to be followed for optimizing the structure and how the system works.

The destruction of exergy in each operating area of the ORC-MCRS system is calculated based on the Gouy-Stodola equation or exergetic equilibrium [4].

Heat transferred from combustion gases:

$$\dot{m}_{\text{fluidORC}} = m_{\text{ga}} \cdot c_{\text{ca}} \cdot \Delta t_{\text{ga}}.$$  

(15)

The average thermodynamic temperature of the combustion gases in the Boiler in Kelvin:

$$T_{\text{Kg}} = (t_{\text{ga}} - t_{\text{ega}}) / \ln(T_{\text{Kega}} / T_{\text{Kega}}),$$

where $T_{\text{Kega}}$ is the temperature of the combustion gases at the boiler inlet and $T_{\text{Kega}}$ is the temperature of the combustion gases at the boiler outlet in Kelvin.

Total exhaust of combustion gases:

$$\dot{E}_{\text{ExtQg}} = \dot{Q}_{\text{ga}} \cdot (1 - T_{\text{Ko}} / T_{\text{Kg}}),$$

(17)

where $T_{\text{Ko}}$ is the ambient temperature in Kelvin.

Then calculate the heat required to heat the working fluid in the liquid state in the boiler:

$$\dot{Q}_{\text{inc}} = \dot{m}_{\text{fluidORC}} \cdot (h_{\text{20}} - h_{\text{2}}),$$

(18)

exergy of the heat of heating the working fluid in the boiler:

$$\dot{E}_{\text{x, inc}} = \dot{m}_{\text{fluidORC}} \cdot (h_{\text{20}} - h_{\text{2}} - T_{\text{Ko}} \cdot (s_{\text{20}} - s_{\text{2}})),$$

(19)

where entropy values in specific points of the cycle are marked with $s$ and $t_{\text{ginc}}$ is the inlet temperature required to heat the working fluid in the liquid state in the boiler.

The heat given by the combustion gases for the heating process of the working fluid in the liquid state:

$$\dot{Q}_{\text{ginc}} = m_{\text{ga}} \cdot c_{\text{ca}} \cdot (t_{\text{ginc}} - t_{\text{ga}}).$$

(20)

This results in the inlet temperature $T_{\text{Kginc}}$ required to heat the working fluid in the liquid state in the boiler, in Kelvin:

$$T_{\text{Kginc}} = t_{\text{ginc}} + 273.15.$$  

(21)

The average thermodynamic temperature of the combustion gases in the heating process in boiler in Kelvin:

$$T_{\text{Kginc}} = (t_{\text{ginc}} - t_{\text{ega}}) / \ln(T_{\text{Kginc}} / T_{\text{Kega}}).$$

(22)
Results the total exergy exhaust of combustion gases required to heat the working fluid in the liquid state in the boiler:

\[ \text{Ext}_{\text{ginc}} = \dot{Q}_{\text{ginc}} \cdot (1 - T_{K0} / T_{Kginc}) . \]  

(23)

Destruction of exergy due to heat transfer to the finite temperature difference for heating the boiler working fluid:

\[ I\Delta_{\text{ginc}} = \text{Ext}_{\text{ginc}} - \text{Ext}_{\text{abinc}} . \]  

(24)

The heat required to vaporize the working fluid in the boiler:

\[ \dot{Q}_v = \dot{m}_{\text{fluidORC}} \cdot (h_{3i} - h_{20}) . \]  

(25)

Exergy heat of vaporization of the working fluid in the boiler:

\[ \text{Ext}_{\text{abv}} = \dot{m}_{\text{fluidORC}} \cdot (h_{3i} - h_{20} - T_{K0} \cdot (s_{3i} - s_{20})) . \]  

(26)

The heat given by the combustion gases for the evaporation process of the working fluid:

\[ \dot{Q}_{\text{g}} = \dot{m}_{\text{ga}} \cdot c_{\text{ca}} \cdot (t_{\text{igv}} - t_{\text{ginc}}) , \]  

(27)

where \( t_{\text{igv}} \) is the inlet temperature of the working fluid in vapour state in Celsius and \( T_{Kigv} \) is in Kelvin:

\[ T_{Kigv} = t_{\text{igv}} + 273.15 . \]  

(28)

The mean thermodynamic temperature of the combustion gases in the evaporator is:

\[ T_{Kg} = (t_{\text{igv}} - t_{\text{ginc}}) / \ln(T_{Kigv} / T_{Kginc}) . \]  

(29)

Results the total exergy exhaust of combustion gases required for the evaporation of the working fluid:

\[ \text{Ext}_{\text{g}} = \dot{Q}_v \cdot (1 - T_{K0} / T_{Kg}) . \]  

(30)

Destruction of exergy due to the transfer of heat at finite temperature difference in the evaporator:

\[ I\Delta_{\text{g}} = \text{Ext}_{\text{g}} - \text{Ext}_{\text{abv}} . \]  

(31)

The heat required to overheat the working fluid in the boiler:

\[ \dot{Q}_{\text{sinc}} = \dot{Q}_B - \dot{Q}_{\text{inc}} - \dot{Q}_v . \]  

(32)

Exergy heat of overheating of working fluid in boiler:

\[ \text{Ext}_{\text{ab-inc}} = \dot{m}_{\text{fluidORC}} \cdot (h_3 - h_{3i} - T_{K0} \cdot (s_3 - s_{3i})) . \]  

(33)

The average thermodynamic temperature of the combustion gases for the overheating process in boiler in Kelvin:

\[ T_{Kg_{sinc}} = (t_{\text{igo}} - t_{\text{igv}}) / \ln(T_{Kgiga} / T_{Kg_{sinc}}) . \]  

(34)

Results the total exergy exhaust of combustion gases required to overheat the working fluid in the superheated vapour state in the boiler:

\[ \text{Ext}_{\text{ginc}} = \dot{Q}_{\text{sinc}} \cdot (1 - T_{K0} / T_{Kg_{sinc}}) . \]  

(35)

Destruction of exergy due to heat transfer to the finite temperature difference for overheating of the working fluid:
\[ I \Delta_{\text{sinc}} = \text{Ext}_{\text{sinc}} - \text{Ext}_{\text{absinc}}. \]  

(36)

Total destruction of exergy due to heat transfer in boiler:

\[ I \Delta_{\text{TB}} = \Delta_{\text{rinc}} + \Delta_{\text{v}} + \Delta_{\text{finc}}. \]  

(37)

Destruction of exergy in the Expander:

\[ I_D = \dot{m}_{\text{fluidORC}} \cdot T_{K0} \cdot (s_4 - s_3). \]  

(38)

The loss of exergy in the Condenser:

\[ P_{\text{Cd}} = \dot{m}_{\text{fluidORCMCRS}} \cdot (h_3 - h_1 - T_{K0} \cdot (s_5 - s_1)). \]  

(39)

Destruction of exergy due to mixing of the working fluid:

\[ I_{\text{am}} = T_{K0} \cdot (\dot{m}_{\text{fluidORCMCRS}} \cdot s_5 - \dot{m}_{\text{fluidORC}} \cdot s_4 - \dot{m}_{\text{fluidMCRS}} \cdot s_8). \]  

(40)

Destruction of exergy in the Pump:

\[ I_p = \dot{m}_{\text{fluidORC}} \cdot T_{K0} \cdot (s_2 - s_1). \]  

(41)

Destruction of exergy in the expansion valve:

\[ I_{\text{VL}} = \dot{m}_{\text{fluidMCRS}} \cdot T_{K0} \cdot (s_6 - s_1). \]  

(42)

Destruction of exergy in the Compressor:

\[ I_{\text{CP}} = \dot{m}_{\text{fluidMCRS}} \cdot T_{K0} \cdot (s_8 - s_7). \]  

(43)

The exergy of the heat of vaporization of working fluid in the MCRS Vaporizer:

\[ \text{Ext}_{Q0} = \dot{m}_{\text{fluidMCRS}} \cdot (h_9 - h_8 - T_{K0} \cdot (s_7 - s_6)). \]  

(44)

Exergetic efficiency:

\[ \eta_{\text{ex}} = \frac{\text{Ext}_{Q0}}{\text{Ext}_{Q_{\text{ex}}}} \cdot 100 \]  

(45)

\[ A = \text{Ext}_{Q_{\text{ex}}} \]  

(46)

\[ B = \text{Ext}_{Q_{\text{ex}}} + P_{\text{Cd}} + I \Delta_{\text{TB}} + I \Delta_{D} + I \Delta_{\text{am}} + I \Delta_{p} + I \Delta_{\text{VL}} + I \Delta_{\text{CP}} \]  

(47)

Percentage of exergy destruction on each equipment reported for total gas exhaust:

\[ \psi_{\text{Cd}} = \frac{P_{\text{Cd}}}{A} \cdot 100 \]  

\[ \psi_{TB} = \frac{I \Delta_{TB}}{A} \cdot 100 \]  

\[ \psi_{D} = \frac{I \Delta_{D}}{A} \cdot 100 \]  

\[ \psi_{\text{am}} = \frac{I \Delta_{\text{am}}}{A} \cdot 100 \]  

\[ \psi_{p} = \frac{I \Delta_{p}}{A} \cdot 100 \]  

\[ \psi_{\text{VL}} = \frac{I \Delta_{\text{VL}}}{A} \cdot 100 \]
Compressor

heating the boiler fluid

\[
\psi_{Cp} = \frac{I\Delta_{Cp}}{A \cdot 100}
\]

vaporization of the fluid in the boiler

\[
\psi_{\Delta_{Tinc}} = \frac{I\Delta_{Tinc}}{A \cdot 100}
\]

overheating of the fluid in the boiler

\[
\psi_{T sinc} = \frac{I\Delta_{T sinc}}{A \cdot 100}.
\]

Exergy destruction percentages for the three fluid conditions in the boiler evaporator vs. total extinction destruction due to heat transfer in the boiler:

heating the boiler fluid

\[
\rho_{\Delta Tinc} = \frac{I\Delta_{Tinc}}{I\Delta_{TB} \cdot 100}
\]

vaporization of the fluid in the boiler

\[
\rho_{\Delta Tv} = \frac{I\Delta_{Tv}}{I\Delta_{TB} \cdot 100}
\]

overheating of the fluid in the boiler

\[
\rho_{T sinc} = \frac{I\Delta_{T sinc}}{I\Delta_{TB} \cdot 100}.
\]

4. Results and discussions

A temperature difference between the combustion gas outlet temperature and the working fluid evaporation temperature in the boiler, \( \Delta t_{\text{minB}} \), has been imposed. For the energetic and exergy analysis of the ORC-MCRS basic scheme, the following input data were used: ambient temperature \( t_0 = 25 \, ^\circ\text{C} \); mass flow rate of gas \( \dot{m}_g = 0.0534 \, \text{kg/s} \); combustion gas temperature of ICE at boiler input \( t_{ga} = 480 \, ^\circ\text{C} \); combustion gas outlet temperature in boiler \( t_{gga} = 480 \, ^\circ\text{C} \); specific heat of combustion gases \( c_g = 1.14 \, \text{kJ/kg} \); condenser water outlet temperature \( t_{ew} = 22 \, ^\circ\text{C} \); evaporation temperature in the evaporator of the MCRS \( T_{ev} = 22 \, ^\circ\text{C} \); the power transfer is considered made without mechanical loss, the crankshaft of the MCRS compressor and ORC expander are coupled directly; a temperature difference is considered between the combustion gas outlet temperature and the working fluid vaporization temperature of the boiler \( \Delta t_{\text{minB}} = 20 \, ^\circ\text{C} \); the degree of overheating in the boiler of the ORC \( \Delta t_{\text{orb}} = 10 \, ^\circ\text{C} \); a temperature difference between the water outlet temperature and the condensing temperature of the working fluid in the Condenser is considered \( \Delta t_{\text{minCd}} = 8 \, ^\circ\text{C} \).

| \(\text{COP}_{\text{MCRS}}\) [-] | \(\text{COP}_{\text{ORC}}\) [-] | \(\text{COP}_{\text{ORCMCRS}}\) [-] | \(\eta_{ex}\) [%] | \(\psi_{Cp}\) [%] | \(\psi_D\) [%] | \(\psi_{ATB}\) [%] | \(\psi_{ATinc}\) [%] |
|---|---|---|---|---|---|---|---|
| 8.393 | 0.1453 | 1.219 | 11.14 | 4.435 | 4.923 | 57.45 | 22.14 |

| \(\psi_{ATv}\) [%] | \(\psi_{T sinc}\) [%] | \(\psi_P\) [%] | \(\psi_{Pice}\) [%] | \(\psi_{VL}\) [%] | \(\psi_{am}\) [%] | \(Q_h\) [kW] | \(Q_{Cd}\) [kW] |
|---|---|---|---|---|---|---|---|
| 32.41 | 2.896 | 0.4345 | 18.25 | 1.191 | 0.2558 | 20.71 | 46.17 |

| \(Q_{\text{Cp}}\) [kW] | \(W_{\text{p}}\) [kW] | \(W_{\text{p}}\) [kW] | \(\dot{m}_{\text{fluidMCRS}}\) [kg/s] | \(\dot{m}_{\text{fluidORC}}\) [kg/s] | \(\dot{m}_{\text{fluidORCMCRS}}\) [kg/s] | \(Q_{\text{gs}}\) [kW] | \(E_{\text{gs}}\) [kW] |
|---|---|---|---|---|---|---|---|
| 25.37 | 3.023 | 0.1087 | 0.1706 | 0.09885 | 0.2694 | 20.71 | 9.799 |
The isentropic efficiency of the Expander $\eta_{sD} = 0.85$; the isentropic efficiency of the MRCS Compressor $\eta_{sCp} = 0.85$; the isentropic efficiency of the ORC Pump $\eta_{sP} = 0.6$.

Figure 2. The energy and exergetic results presented schematically using the interface projected in the EES software.

The results of the energetic and exergy analysis are presented centrally in table 3 and punctual in figure 2 illustrating the operation scheme of the ORC-MCRS system. In the same figure can be found values for the temperatures in the key system states as well as the mass flows in the ORC-MCRS cycle.

Figure 3. Variation of exergetic efficiency and coefficient of performance in relation to the temperature difference between the currents passing through the evaporator boiler.

In order to os sis and optimize the basic cycle of the ORC-MCRS installation, diagrams with exergy losses were made according to the variation of the decisonal parameters $\Delta t_{minB}$, $\Delta t_{minCd}$, $\Delta t_{silB}$, which directly determines the exergetic and energetic efficiency. Thus $\Delta t_{minB}$ requires the
vaporization temperature and $\Delta t_{\text{minC}}$ impose the condensation temperature and we notice that much of the energy is consumed for the heating of the working fluid.

Exergy analysis is the only analysis that can be used in each process to see which part is the most destructive.

It was taken for simulation a $\Delta t_{\text{minB}}$ of less than 20K and was varied from grade by grade, and we observe that the exergetic efficiency and the coefficient of performance of the ORC-MCRS system improves to small temperature differences between the two currents passing through the boiler vaporizer.

![Figure 4](image1.png)

**Figure 4.** Lost exergy values for heating, vaporisation, overheating depending on the temperature difference between the currents passing through the evaporator boiler.

Depending on the variation of the temperature difference between the currents passing through the boiler evaporator, we can see in figure 4 that the loss of exergy differs for the heating and vaporization processes and remains almost constant in the overheating process also because this process represents only 2.89% of the losses recorded. From the graph we can see that with the decrease of the temperature difference, the exergy losses due to the heating process decrease and the ones due to the evaporation process increase, this process having the highest percentage of destruction.

![Figure 5](image2.png)

**Figure 5.** Variation of exergetic efficiency and total exhaust heat exchanger MCRS depending on degree of overheating in ORC boiler $\Delta t_{\text{B}}$.
Next, a degree of overheating in boiler $\Delta t_{\text{siB}}$ below 10K was taken for simulation, and it was varied from grade to grade, so it is seen from figure 5 the increase of exergetic efficiency and total exergy as the boiler overheating increases.

Depending on the degree of overheating in the boiler, we can see in figure 6 the variation of the exergetic losses for heating, vaporization and overheating, as expected the most significant increase of the exergetic losses is recorded in the overheating process with the increase of the degree of overheating.

![Figure 6. Lost exergy values for heating, vaporization, overheating depending on the degree of overheating in the ORC boiler $\Delta t_{\text{siB}}$.](image)

However, given the high percentage of heating and vaporization processes, it is noted that the increase of the $\Delta t_{\text{siB}}$ overheating leads to the reduction of the total exergetic losses in the boiler as shown in the diagram in figure 5 and the flow of the working fluid in the circuits is also sensitive influenced by this overheating.

Another variable that directly influences the exergy losses in the ORC-MCRS system is also the temperature difference between the currents passing through the condenser heat exchanger. We can see from figure 7 that with the increase in the temperature difference, the exergetic yield decreases as well as the coefficient of performance of the ORC-MCRS system.

![Figure 7. Exergetic yield and coefficient of performance of the ORC-MCRS system depending on the temperature difference between the currents passing through the condenser heat exchanger.](image)
Figure 7. Variation of exergetic efficiency and the coefficient of performance in relation to the temperature difference between the currents running through the Condenser.

Regarding the loss of exergy due to heating, vaporization and overheating of the working fluid in the boiler evaporator we notice that it varies with the increase in the temperature difference $\Delta T_{\text{minCd}}$. Exergetic losses due to the heating process decrease, the exergetic losses due to the evaporation process increase and the exergetic losses due to the overheating process remain constant as shown in figure 8.

Figure 8. Lost exergy values for heating, vaporisation, overheating of the boiler depending on the temperature difference between the currents running through the Condenser.

Figure 9. Lost exergy values from the ORC-MCRS system components depending on the temperature difference between the currents passing through the boiler vaporizer.
In the diagrams of figures 9 and 11 the exergy losses in the main components of the ORC-MCRS system are presented simultaneously, depending on the temperature difference between the currents passing through the boiler evaporator and the condenser, but also the degree of overheating in the boiler evaporator in figure 10, resulting from simulation of theoretical operation.

**Figure 10.** Lost exergy values from the ORC-MCRS system components depending on the boiler overheating degree.

As could be expected, major fluctuations in losses in sensitive areas influenced by a decisional parameters are observed.

**Figure 11.** Lost exergy values in the ORC-MCRS system subassemblies depending on the temperature difference between the currents passing through the Condenser.
The diagram of figure 12 shows the enthalpy variation and the temperature of the combustion gases with red and the composite curve of enthalpy variation and temperature in the heating, vaporization and overheating processes of the SES36 working fluid in the boiler heat exchanger marked with blue.

![Figure 12. The T-ΔH diagram of the two currents running through the boiler vaporizer.](image)

The diagram in figure 13 represents the same compound curves, this time, depending on the exergetic factor ($\theta$). We note that much of the amount of exergy introduced through the combustion gases is consumed in the evaporator boiler as follows: 22.14% in the heater, in the evaporator 32.41% and only 2.89% in the superheater. The recorded values are due both to the working agent and to the large temperature difference in the boiler.

In figure 13, the surface between the combustion gas curve and the $H$-axis gives the measure of the total exergy of the heat transferred from the combustion gases, and the surface between the composite curve of the working fluid in the three heating-vaporization-overheating steps and the $H$-axis gives us the measure of exergy received by the working fluid.

![Figure 13. Diagram $\theta$-Δ$H$ of the two currents passing through the boiler.](image)
The areas between the two curves give us the measure of destruction of exergy in each distinctive part: heater, evaporator and superheater.

From figure 13, it becomes obvious that the largest area of the two curves (and the greatest destruction of the exergy) is associated with the heat transfer on a finished surface at a too high temperature difference in the heating and vaporization step of the work.

5. Conclusions
In order to reduce the surface corresponding to the destruction of exergy in the heater, the ORC fluid inlet temperature should be increased.

Exergy analysis of the core cycle highlights deficiencies in this scheme, reveals the operational possibilities and constructive changes to improve its performance.

On the MCRS side, we note that the destruction of exergy at the expansion valve is low (about 1%), which indicates the good thermodynamic properties of the SES36 working fluid.

We note that we can reduce the loss of exergy in heating the working fluid so that the boiler takes less heat from the flue gas to heat the working fluid.

We note that we can perform this preheating by installing a heat exchanger before the boiler through which we will pass the relatively high temperature working fluid from the outlet of the expander.

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Acknowledgments

The authors gratefully acknowledge the support provided by Executive Agency for Higher Education, Research, Development and Innovation Funding of Romania. Through the Program PN-II-PT PCCA-2011-3.2-0059, Grant No.: 75/2012.