

Type of the Paper (Article, Review, Communication, etc.)

Exergoeconomic analysis of a mechanical compression refrigeration unit run by ORC

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Abstract: Increasing the efficiency of a Diesel internal combustion engine can be achieved by cogeneration of mechanical energy and cold. The cold is produced by a mechanical compression cycle (MRC) driven by an ORC system that recovers the heat emitted by the combustion gases. The work offers an exergoeconomic optimization methodology of the MRC_ORC group. The exergetic analysis highlights the changes that can be made to the system structure to reduce the exergy destruction associated with internal irreversibilities. Thus, the preheating with the help of an internal heat exchanger of the ORC fluid leads to a decrease in the share of exergy destruction in the ORC boiler by 4.19% and finally to an increase in the global exergetic yield by 2.03% and implicitly in the COP of the ORC-MRC installation. Exergoeconomic correlations are built for each individual piece of equipment. The mathematical model for calculating the monetary costs for each flow of substance and energy in the system is presented. Following the evolution of the exergoeconomic performance parameters, the optimization strategy is drawn. The conclusion is to reduce the exergy consumption in the system by choosing high-performance equipment. Following the optimization procedure, the cost of the produced cold unit drops by half. The cost of Diesel fuel has a major influence on the unit cost of cooling. A doubling of the cost of Diesel fuel leads to an 80% increase in the cost of the cold unit.

Keywords: exergy analysis; Diesel engine; heat exchanger; exergy destruction; exergoeconomic correlation; exergoeconomic cost assessing; exergoeconomic factor.

Citation: To be added by editorial staff during production.

Academic Editor: Firstname
Lastname

Received: date
Revised: date
Accepted: date
Published: date



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1. Introduction

The increase in the cost of fossil fuels, but also the demand for energy and the environmental concerns [1] have led to the analysis, design and development of thermal systems that can also exploit the energy potential of low-quality heat sources. One such thermal system, used on a large scale that uses a low grade heat source, is the Rankine cycle with organic fluids (ORC) [2].

Daniarta S. et al. [3] analyzed the benefits of recovering with an ORC the heat released by the baking ovens in an automotive paint shop. The use of the ORC succeeds in reducing the heat loss that accompanies the manufacturing process contributing to ameliorating the employees confort while producing also electrical energy.

Because the thermal efficiency of the ORC system is quite low [4], this one has been combined with other thermal systems with the main aim of improving the overall efficiency of the system by converting low quality energy into other useful forms of energy such as electricity, cold and heat. A presentation of the combined systems intended

for the simultaneous production (cogeneration) of energy and cold was made in [5, 6], and the works [7, 8] describe the systems intended for the simultaneous production of mechanical energy, heating and cooling (trigeneration).

The authors of the works [9–11] dedicated their research to the thermodynamic study of thermal systems that combine ORC and a refrigerating cycle with an ejector. The papers [12–14] describe a thermodynamic analysis of the combined power and cooling cycle, consisting of an ORC cycle and an absorption cycle that uses the ammonia–water mixture as the working fluid. Paper [15] presents a heat-fed refrigeration cycle concept that combines an ORC and a mechanical vapor compression (MRC) refrigeration plant using a unitary assembly consisting of a free piston compressor–expander. The two systems use the same working fluid and, in addition, share the same condenser. Also, in the paper [16] a thermally activated cooling cycle consisting of an ORC and an MRC is presented. The system can be powered by solar thermal energy, geothermal energy or various waste heat flows. The ORC expander shaft was directly coupled to the MRC compressor shaft to reduce power conversion losses.

Hu K. et al. [17] focus their interest on an analysis of a refrigerating system based on an ORC. The refrigeration is obtained in a two-stage transcritical CO₂ mechanical compression cycle. The analysis is based on economic, energetic and exergetic criteria. The influence of the intermediary and high pressure of CO₂ is shown. The analysis points out that the boiler of the ORC has the largest exergy destruction while the evaporator of the refrigerator has the highest investment cost.

Bett A.K. and Jalilinasrabady S. [18], combine exergy with Pinch analysis to optimize an ORC system that recovers geothermal energy. Different ORC fluids are comparatively analyzed. The use of a regenerative internal heat exchanger is considered. The variation in the Pinch temperature difference in the ORC evaporator imposes the inlet pressure of the ORC fluid in the turbine. An optimal boiling pressure for which the net power generated is found out for each ORC fluid.

Ochoa G. et al. [19] analyzes the use of solar energy as a heat source in ORC cycles.

At the same time, there are many recent studies that have focused on the subject of applying ORC technology to recover low-quality waste heat from Diesel engines [20–30], using ORC systems to produce additional mechanical energy or to drive an electric generator. The paper [31] presents an alternative of using the waste heat from the Diesel engine in an ORC which in turn produces the energy needed to drive the compressor of a MRC. MRC can produce a cooling effect at different temperatures. The advantage of a combined ORC–MRC system compared to absorption refrigeration systems [32] is that when the refrigerating effect is not required, all thermal energy can be converted into power and used for other applications [33–40].

In order to increase the overall performance of the combined cycle, numerous studies were carried out based on the principles of thermodynamics and economic analysis to determine the characteristics of the operating parameters such as: the right working fluid, the temperature of the heat source, the vaporization temperature of the working fluid, etc. which characterizes a higher thermal performance of the combined ORC–MRC system.

Camero A.B. et al. [41], used an advanced exergo–environmental analysis to discover the perspectives of improvement of an ORC that recovers the waste heat from a natural gas engine. The work based on the concepts of endogenous and exogenous exergy destructions points out that most of the key parts of the system are weakly connected with the other and can be optimized practically individually in order of their exergy destruction magnitude. The analysis reveals that the greatest endogenous exergy destruction happens in the heat transfer process between the engine exhaust gases and ORC fluid where about 70% of the available energy of the exhaust gases is destroyed. A reduction in this exergy destruction will increase the overall system performance.

Li T. et al. [42], analyze the use of different working fluids in a combined cooling, heating and power cogeneration system composed of an ORC supplied with geothermal heat and a two-stage mechanical compression refrigerator that offers two levels of cold

temperatures. The influence of the geothermal fluid temperature on the turbine net power output is discussed. The values of the mechanical power generated, cooling and heating when using different working fluids are shown.

In the paper [43] the authors proposed that the energy source supplying the secondary systems of the vehicle should be constituted by a combined ORC-MRC system. They used R134a, R245fa, n-propane and cyclopentane and found R134a to be the most suitable working fluid. The proposed combined system was determined to be 9.2% to 9.8% more thermally efficient than the conventional power system. In the paper [44], eight fluids such as R143a, R134a, R152a, ammonia, R22, R600, isobutane and propane were studied for the ORC-MRC system and they proposed isobutane as a suitable working fluid to obtain a coefficient of performance (COP) and an increased exergetic efficiency. In paper [45] a small-scale ORC-MRC system was investigated based on thermoeconomic criteria using R245fa as the working fluid. They analyzed the impact of condensing and vaporizing temperature, superheat and recovery heat exchanger on the thermo-economic performance of the ORC-MRC system. The authors concluded that adding the recuperator and superheat did not affect the thermoeconomic performance of the system compared to changing the condensing and vaporizing temperatures. The paper [46] aims to study 14 pure working fluids such as hexane, butane, R152a, R245fa, isobutane, RC318, R236ea, R236fa, R1234ze (E), C5F12, RE245cb2, R245ca, R6010 and proposed the working fluid R6012 to obtain an increased exergetic efficiency and COP of the ORC-MRC system. In the paper [47] the authors compared the ORC-MRC system with a commercial air conditioning system. They concluded that the COP of the combined system can be increased from 3.1 to 3.54 compared to the commercial air conditioning system by using the R600a-R123 fluid pair as the working fluid.

To further improve the overall performance of ORC-based systems, researchers have used several optimization techniques. The paper [48] presents the application of genetic algorithm (GA) for the thermo-economic optimization of the combined ORC-MRC system to determine the optimal component sizes and thermal performance of the system. They recommended R245fa as the working fluid for optimal thermo-economic performance of the system. In the paper [49] the authors used the TOPSIS technique based on the entropy method to determine the optimal mass fraction of mixtures to be used as working fluid in a simple ORC. It was concluded that the mass fraction 0.1/0.9 is optimal for mixtures such as pentane/butane, hexane/pentane and isohexane/pentane. Also, the paper [50] presents the multi-objective optimization of the simple ORC system using the Taguchi technique and gray relational analysis (GRA) to determine the order of importance of each input parameter in the system and their effect on the energetic and exergetic performance of the system. It was found that choosing optimal condensing and vaporizing temperatures, turbine isentropic efficiency and heat exchanger efficiencies were the most effective, than optimizing the superheat temperature, pinch temperature differences in condenser and evaporator and isentropic efficiency of the pump.

Based on the literature review, it can be deduced that previous studies have focused on the energetic and exergetic analysis of the ORC-MRC system, but very few studies are aimed at optimizing the performance of the ORC-MRC combined cycle.

The concept proposed for analysis in this paper combines an ORC cycle with a mechanical vapor compression refrigeration (MRC) cycle to form a thermally activated cooling system by recovering the heat from the combustion gases of a stationary internal combustion engine (ICE) through cogeneration of mechanical power and cold. The complex ORC-MRC system uses a direct gear ratio of the mechanical work generated by the ORC cycle expander. Thus, the shaft of the ORC expander directly engages the shaft of the MRC compressor and uses a single working fluid to drive both thermodynamic cycles chosen after extensive bibliographic research, namely R1224yd(Z). In this sense, a hydraulic route has been implemented that circulates the working fluid required for both cycles, with a single condenser heat exchanger.

The present work aims to identify and optimize the constructive solutions of the ORC-MRC coupled system by means of the exergoeconomic analysis. The exergoeconomic optimization method aims to identify the exergy destruction of each functional area and assign its monetary cost. Areas with high cost of exergy destruction will be the first targets of the optimization procedures, the effect of each local cost reduction will be verified at the global level. To quantify the value of a zonal destruction, the fuel, the product, the loss and the exergetic performance coefficient were evaluated for each operational area, with the aim of obtaining the maximum product from a limited resource, with minimum monetary cost.

The scheme of the ORC-MRC combined installation and the representation of the cycle in the p-h diagram are presented in figure 1.

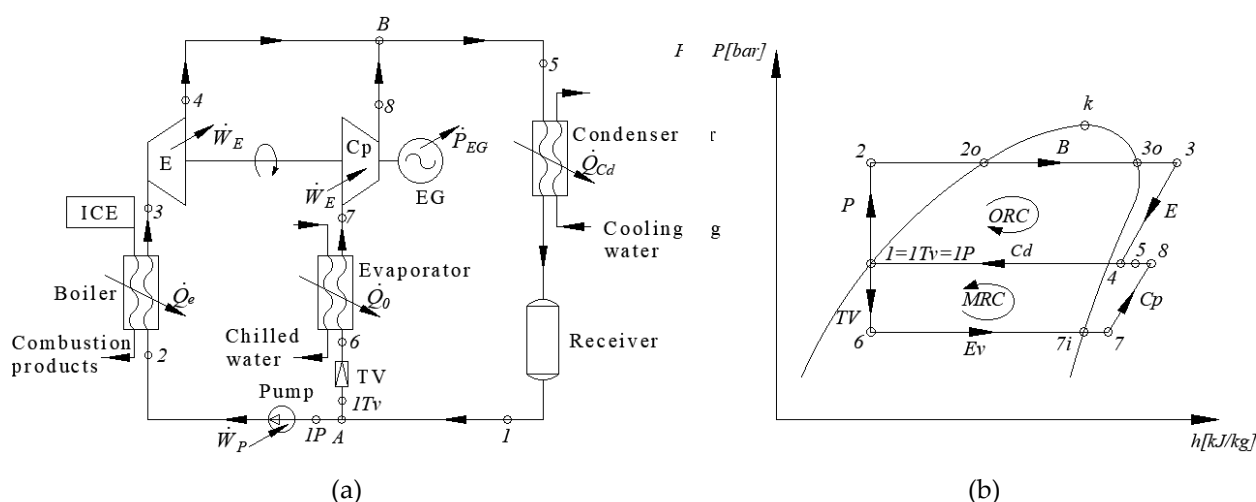


Figure 1. System ORC-MRC: (a) Flow chart; (b) ORC-MRC cycles in the p – h diagram.

The characteristic measures (Table 1) for the analysis of the optimized coupling were taken from the internal combustion engine (ICE) that equips the ICE-ORC experimental stand located within the Department of Thermotechnics, Engines, Thermal Equipment and Refrigeration of the National University of Science and Technology Politehnica Bucharest. The stand is equipped with a Diesel type engine model 4TNV98TGGEHR manufactured by Yanmar and 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 installation with the role of recovering the heat dissipated by the engine.

Table 1. The characteristics of the coupled system ICE-ORC-MRC

Main system parameters	Values
Maximum mechanical power	40 kW
The temperature of the combustion gases in the 100% regime, equal to the temperature of the hot fluid at the entrance to the ORC Boiler, $t_{\alpha} = t_{i,g}$	420°C
Outlet temperature of the combustion gases from the Boiler, $t_{\beta} = t_{o,g}$	140°C
Mass flow rate of combustion gases, $\dot{m}_g$	0.0534 kg/s
Specific heat of combustion gases, c_g	1.14 kJ/kgK
The temperature of the cooling water at the entrance to the condenser, $t_{\xi} = t_{iw,Cd}$	15°C
Condenser outlet water temperature, $t_{\tau} = t_{ow,Cd}$	22°C
Condenser cooling water mass flow rate, $\dot{m}_{w,c}$	0.5 kg/s

The temperature of the cooled water at the entrance to the evaporator, $t_{\gamma} = t_{iw,Ev}$	8°C
The temperature of the cooled water at the exit from the evaporator, $t_{\delta} = t_{ow,Ev}$	3°C
Ambient temperature, t_o	15°C
ORC-MRC working fluid	R1224yd(Z)

The operating conditions were established starting from the parameters of the heat source for the ORC (engine exhaust gases) and the condenser cooling water. The maximum vaporization temperature of the organic fluid was imposed taking into account the restriction that the temperature of the combustion gases at the exit from the Boiler vaporizer should not fall below 140 °C to avoid condensation and the formation of sulfuric acid (H₂SO₄) in the ORC Boiler and implicitly its corrosion.

Compared to refrigeration systems with absorption or adsorption the ORC – MRC combined system is flexible; during hot summer periods all available thermal energy can be converted into mechanical energy and then into a cooling effect, and during winter when there is no need for cold, the system can produce only electrical energy

2. Mathematical modeling of the system

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

The importance of the exergetic analysis consists in the possibility to identify and evaluate the magnitude of the dysfunctions in the system. Based on the exergetic analysis, a strategy can be established to improve the performance of a system by making constructive and operational changes.

The analysis of the functional scheme and of the installation cycle (Figure 1) highlights the sequence of work processes: (2-20) heating the ORC liquid up to boiling temperature, (20-30) vaporizing the ORC liquid in the ORC boiler, (30-3) superheating of the vapors in the ORC Boiler, (3-4) expansion of the vapors to the condensing pressure in the ORC expander, (7-8) compression of the superheated vapors in the MRC Compressor to the condensing pressure, (5) the result of the mixing of the streams with the states (4) and (8), (5-1) condensation to saturated liquid in the ORC-MRC condenser, (1TV) the part of the stream (1) that expands in the throttling valve (TV) of the refrigeration plant, (1P) the part of the stream (1) which is compressed in the ORC liquid pump (P), (1P-2) the liquid compression in the ORC pump, (1TV-6) the throttling of the liquid agent in the MRC throttling valve, (6-7) the vaporization and vapor overheating in the MRC evaporator, (A) nodal point for separation of the condensed liquid (1) in the streams (1TV) to the TV throttling valve of the MRC and (1P) to the ORC liquid pump, (B) nodal mixing point of currents (4) and (8).

The analysis was made based on the following assumptions:

- there is no heat loss in the heat exchangers;
- pressure losses in all components of the installation are neglected;
- expansion in the expander, compression in the compressor and pressure increase in the pump are irreversible adiabatic processes characterized by the isentropic efficiencies η_{sE} , η_{sCP} , η_{sP} ;
- it is considered that the expander is directly engaged with the compressor $\dot{W}_{CP} = \dot{W}_E$, the power transfer being done without mechanical losses;
- the working fluid in the ORC-MRC system is R1224yd(Z), a new type refrigerant used to replace R245fa.

The mathematical model of the operation of the ORC-MRC system is built based on the choice of some decision parameters.

2.1 Choice of decision parameters

2.1.1 The boiling temperature in the ORC boiler

The temperature difference Δt_{minB} is chosen between the outlet temperature of the combustion gases $t_{o,g}$ and the boiling temperature of the working fluid in the boiler t_B (Figure 2 (a)). The result is the boiling temperature of the working fluid in the boiler t_B .

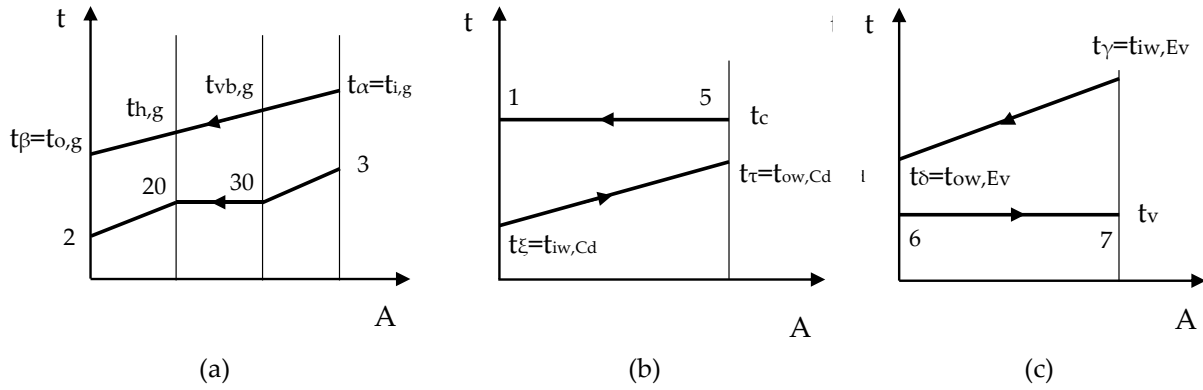


Figure 2. Temperature – Area diagrams: (a) ORC boiler; (b) ORC-MRC condenser; (c) MRC evaporator (for the ORC-MRC working fluid notations correspond to Figure 1)

2.1.2 Condensation temperature

The temperature difference Δt_{minCd} is chosen between the outlet temperature of the water from the condenser $t_{ow,Cd}$ and the condensation temperature of the working fluid t_c (Fig. 2(b)). It results:

$$t_c = t_{ow,Cd} + \Delta t_{minCd} \quad (1)$$

2.1.3 Vaporization temperature

The temperature difference Δt_{minEv} is chosen between the outlet temperature of the chilled water from the evaporator $t_{ow,Ev}$ and the evaporation temperature of the working fluid t_v (Fig. 2(c)). Result:

$$t_v = t_{ow,Ev} - \Delta t_{minEv} \quad (2)$$

2.1.4 The temperature of the ORC fluid at the entrance to the expander

The degree of superheating of the vapors in the boiler $\Delta t_{oh,B}$ is chosen (Figures 1(b), 2(a)), resulting temperature t_3 at the exit from the Boiler:

$$t_3 = t_B + \Delta t_{oh,B} \quad (3)$$

2.2 Energetic analysis

Based on the energy balances of the component devices, the flow rates of the working fluids, the thermal loads of the heat exchangers and the mechanical powers are calculated (Figure 1).

The organic fluid mass flow rate results from the energy balance of the ORC boiler:

$$\dot{Q}_B = |\dot{Q}_g| = \dot{m}_g \cdot c_g(t_{i,g} - t_{o,g}) \quad (4)$$

$$\dot{m}_{ORC} = \dot{Q}_B / (h_3 - h_2) \quad (5)$$

The power produced in the expander becomes:

$$\dot{W}_E = \dot{m}_{ORC}(h_3 - h_4) \quad (6)$$

Power input of the working fluid drive pump:

$$|\dot{W}_p| = \dot{m}_{\text{ORC}}(h_2 - h_1) \quad (7)$$

The specific mechanical work required to drive the compressor:

$$|W_{\text{cp}}| = h_8 - h_7 \quad (8)$$

Considering that the MRC compressor is directly driven by the expander without mechanical losses

$$|\dot{W}_{\text{cp}}| = \dot{W}_E \quad (9)$$

the mass flow rate of the fluid in the refrigeration cycle can be calculated.

$$\dot{m}_{\text{MRC}} = |\dot{W}_{\text{cp}}|/|W_{\text{cp}}| \quad (10)$$

The condenser is common to the two combined cycles so that its thermal power becomes:

$$|\dot{Q}_{\text{cd}}| = (\dot{m}_{\text{ORC}} + \dot{m}_{\text{MRC}})(h_5 - h_1) \quad (11)$$

The energy balance of the evaporator specifies the refrigeration power of the system

$$\dot{Q}_{\text{ev}} = \dot{m}_{\text{MRC}}(h_7 - h_6) \quad (12)$$

The energy performance coefficients, calculated for each individual subsystem and for the overall system, are:

$$\text{COP}_{\text{ORC}} = \frac{\dot{W}_E}{|\dot{Q}_B|} \quad (13)$$

$$\text{COP}_{\text{MRC}} = \frac{\dot{Q}_{\text{ev}}}{|\dot{W}_{\text{cp}}|} \quad (14)$$

$$\text{COP}_{\text{ORC-MRC}} = \frac{\dot{Q}_{\text{ev}}}{|\dot{Q}_B|} \quad (15)$$

2.3 Exergetic analysis

Exergy represents the maximum mechanical work that a system can release or the minimum mechanical work it must consume to reach total equilibrium with its environment. In energy transfer processes, part of the exergy is consumed (destroyed) due to irreversibility.

Minimizing exergy destruction in key components of an energy system provides the strategy to follow to optimize the structure and the way the system works.

The exergy destruction in each operating area of the ORC-MRC system is calculated based on the Gouy-Stodola equation or the exergy balance equations.

The analysis of heat transfer in the ORC boiler was carried out by dividing it into distinct areas for liquid heating, evaporation and vapor superheating:

- Liquid heating area (Figure 2 (a))

The heat required to heat the working fluid in the liquid state in the Boiler is:

$$\dot{Q}_h = \dot{m}_{\text{ORC}}(h_{20} - h_2) \quad (16)$$

From the energy balance of this area, the inlet temperature of the combustion gases in this area results.

$$\dot{Q}_h = \dot{Q}_{h,g} = \dot{m}_g c_g (t_{h,g} - t_{o,g}) \quad (17)$$

The exergy destruction associated with heat transfer at the finite temperature difference in the liquid heating zone becomes:

$$\dot{I}_{\Delta T, h} = \left| \dot{E}x_Q^{T_{g,h}} \right| - \dot{E}x_Q^{T_{ORC,h}} \quad (18)$$

$$\left| \dot{E}x_Q^{T_{g,h}} \right| = |\dot{Q}_{g,h}| \left(1 - \frac{T_0}{T_{g,h}} \right) \quad (19)$$

$$\dot{E}x_Q^{T_{ORC,h}} = \dot{m}_{ORC} (h_{20} - h_2 - T_0 (s_{20} - s_2)) \quad (20)$$

where:

$$T_{g,h} = \frac{t_{h,g} - t_{o,g}}{\ln \frac{T_{h,g}}{T_{o,g}}} \quad (21)$$

- ORC fluid evaporation zone (Figure 2 (a))

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

$$\dot{Q}_{vb} = \dot{m}_{ORC} (h_{30} - h_{20}) \quad (22)$$

The exergy of the boiling heat of the ORC fluid in the boiler becomes:

$$\dot{E}x_Q^{T_{ORC,vb}} = \dot{m}_{ORC} (h_{30} - h_{20} - T_0 (s_{30} - s_{20})) \quad (23)$$

On the side of the combustion gases in the evaporation area of the ORC fluid, the energy balance equation specifies:

$$|\dot{Q}|_{g,vb} = \dot{Q}_{ORC,vb} \quad (24)$$

$$|\dot{Q}|_{g,vb} = \dot{m}_g c_g (t_{vb,g} - t_{h,g}) \quad (25)$$

in which the variable is the temperature entering the area of the combustion gases, $t_{vb,g}$.

The average thermodynamic temperature of the combustion gases in the evaporation zone is calculated as follows:

$$T_{g,vb} = (t_{vb,g} - t_{h,g}) / \ln(T_{vb,g} / T_{h,g}) \quad (26)$$

The exergy of the combustion gases becomes:

$$\left| \dot{E}x_Q^{T_{g,vb}} \right| = |\dot{Q}|_{g,vb} \left(1 - T_0 / T_{g,vb} \right) \quad (27)$$

Exergy destruction due to heat transfer at finite temperature difference for the vaporization of the working fluid is calculated with the relation:

$$\dot{I}_{\Delta T, vb} = \left| \dot{E}x_Q^{T_{g,vb}} \right| - \dot{E}x_Q^{T_{ORC,vb}} \quad (28)$$

- Overheating zone of organic fluid vapors (Figure 2 (a))

Exergy destruction due to heat transfer at finite temperature difference for superheating the working fluid is:

$$\dot{I}_{\Delta T, oh} = \left| \dot{E}x_Q^{T_{g,oh}} \right| - \dot{E}x_Q^{T_{ORC,oh}} \quad (29)$$

$$\dot{E}x_Q^{T_{ORC,oh}} = \dot{m}_{ORC} (h_3 - h_{30} - T_0 (s_3 - s_{30})) \quad (30)$$

$$\left| \dot{E}x_Q^{T_{g,oh}} \right| = |\dot{Q}|_{g,oh} \left(1 - T_0 / T_{g,oh} \right) \quad (31)$$

$$T_{g,oh} = (t_{i,g} - t_{vb,g}) / \ln(T_{i,g} / T_{vb,g}) \quad (32)$$

Taking into account the exergy consumption in the boiler's functional areas, the total exergy destruction due to heat transfer at the finite temperature difference in the boiler becomes:

$$\dot{I}_{\Delta T,B} = \dot{I}_{\Delta T,h} + \dot{I}_{\Delta T,vb} + \dot{I}_{\Delta T,oh} \quad (33)$$

Taking into account Eqs (20, 28, 32), the total exergy consumption in the ORC boiler, which is also the exergetic fuel of the global system, is:

$$|\dot{E}x_{Q,g}| = |\dot{E}x_Q^{T_{g,h}}| + |\dot{E}x_Q^{T_{g,vb}}| + |\dot{E}x_Q^{T_{g,oh}}| \quad (34)$$

For the other component equipment of the ORC-MRC group (Figure 1), the following exergy consumptions or losses are identified:

- exergy destruction in the expander

$$\dot{I}_E = \dot{m}_{ORC} T_0 (s_4 - s_3) \quad (35)$$

- exergy loss in the condenser

$$\dot{I}_{Cd} = \dot{m}_{ORC-MRC} [h_5 - h_1 - T_0 (s_5 - s_1)] \quad (36)$$

- exergy destruction due to the mixing of the working fluid at the entrance to the common condenser (state 5 Figure 1)

$$\dot{I}_m = T_0 (\dot{m}_{ORC-MRC} \cdot s_5 - \dot{m}_{ORC} \cdot s_4 - \dot{m}_{MRC} \cdot s_8) \quad (37)$$

- exergy destruction in the pump

$$\dot{I}_P = T_0 \cdot \dot{m}_{ORC} (s_2 - s_1) \quad (38)$$

- exergy destruction in the throttling valve

$$\dot{I}_{TV} = T_0 \cdot \dot{m}_{MRC} (s_6 - s_1) \quad (39)$$

- exergy destruction in the compressor

$$\dot{I}_{Cp} = T_0 \cdot \dot{m}_{MRC} (s_8 - s_7) \quad (40)$$

The product of the combined ORC-MRC system is the exergy of the refrigerating effect achieved in the evaporator of the refrigerating cycle.

$$|\dot{E}x_Q^{T_{MRC,v}}| = |\dot{E}x_{Q_0}| = -\dot{m}_{MRC} (h_7 - h_6 - T_0 (s_7 - s_6)) \quad (41)$$

The exergetic performance of the combined ORC-MRC system is specified by the global exergetic efficiency and the share of exergy consumption and losses, associated with each equipment or process, in the exergy consumption of the system.

Noting that the product of the global system is defined by Eq. (41) and the fuel used upon entering the system by Eq. (34), the exergetic efficiency of the ORC-MRC system, becomes:

$$\eta_{ex} = \frac{|\dot{E}x_Q^{T_{MRC,v}}|}{|\dot{E}x_{Q,g}|} = \frac{|\dot{E}x_{Q_0}|}{|\dot{E}x_{Q,g}|} \quad (42)$$

The weight of a destruction or loss of exergy in the exergy consumption of the global system is defined as follows:

$$\psi_i = \frac{\dot{I}_i}{|\dot{E}x_{Q,g}|} \quad (43)$$

For the basic scheme of ORC-MRC (Figure 1), the energetic and exergetic study was carried out for the fluid R1224yd(Z), under the conditions: $t_0 = 15^\circ\text{C}$; $\dot{m}_g = 0.0534 \text{ kg/s}$; $t_{i,g} = 480^\circ\text{C}$; $t_{o,g} = 140^\circ\text{C}$; $c_g = 1.14 \text{ kJ/(kgK)}$; $t_{ow,Cd} = 22^\circ\text{C}$; $t_v = -5^\circ\text{C}$; $|\dot{W}|_{cp} = \dot{W}_E$; $\Delta T_{minB} = 10\text{K}$; $\Delta T_{oh,B} = 6\text{K}$; $\Delta T_{minCd} = 6\text{K}$; $\eta_{SE} = 0.85$; $\eta_{SCP} = 0.85$; $\eta_{SP} = 0.6$

The results of the energy and exergy analysis of the ORC-MRC coupled cycle are presented in table 2 and figure 3.

Table 2. The results of the energetic and exergetic analysis for the basic scheme ORC-MRC

COP_{MRC} [-]	COP_{ORC} [-]	$COP_{ORC-MRC}$ [-]	η_{ex} [%]	$\dot{m}_{MRC}$ [kg/s]	$\dot{m}_{ORC}$ [kg/s]	$\dot{m}_{ORC-MRC}$ [kg/s]	ψ_{Cp} [%]	ψ_E [%]	$\psi_{\Delta T,B}$ [%]	$\psi_{\Delta T,h}$ [%]
6.108	0.1657	0.9468	14.91	0.1089	0.07605	0.1849	4.605	5.464	51.45	28.07
$\psi_{\Delta T,oh}$ [%]	$\psi_{\Delta T,vb}$ [%]	ψ_P [%]	$\psi_{L,Cd}$ [%]	ψ_{TV} [%]	ψ_m [%]	$\dot{Q}_B$ [kW]	$\dot{Q}_{Cd}$ [kW]	$\dot{Q}_{Ev}$ [kW]	$\dot{W}_E$ [kW]	$ \dot{W} _P$ [kW]
2.425	20.96	0.8721	19.22	3.21	0.267	17.05	33.18	16.14	2.825	0.1823

The exergy analysis highlights the high exergy consumption due to the heat transfer at the finite temperature difference from the ORC boiler. The biggest temperature differences, which also determine the biggest exergy consumption, are found in the ORC liquid heating zone $\psi_{\Delta T,h} = 28.08\%$ and evaporation zone $\psi_{\Delta T,vb} = 20.96\%$; the reduction of these destructions is the way forward in the optimization procedure. A decrease in the temperature difference in the ORC boiler would lead to a decrease in the exergy destruction in this area leading to an increase in the cooling power of the combined system. The loss of exergy in the condenser is more related to the temperature difference between the ORC and MRC fluids and the cooling water, being influenced to some extent by the variation of the organic fluid mass flow rates to the variation of the other decision parameters.

In order to understand how the decision-making parameters functionally and constructively influence the evolution of the ORC-MRC system, a sensitivity study of the energetic and exergetic performances and of the destruction and exergy losses of the combined system to the variation of the decision-making parameters Δt_{minB} , Δt_{minCd} , $\Delta t_{oh,B}$ has been done.

2.4.1 The behavior of the ORC-MRC cycle at the variation of the temperature difference Δt_{minB} in the ORC boiler

The results of the sensitivity study to the change in temperature difference Δt_{minB} in the ORC boiler are presented in figure 3.

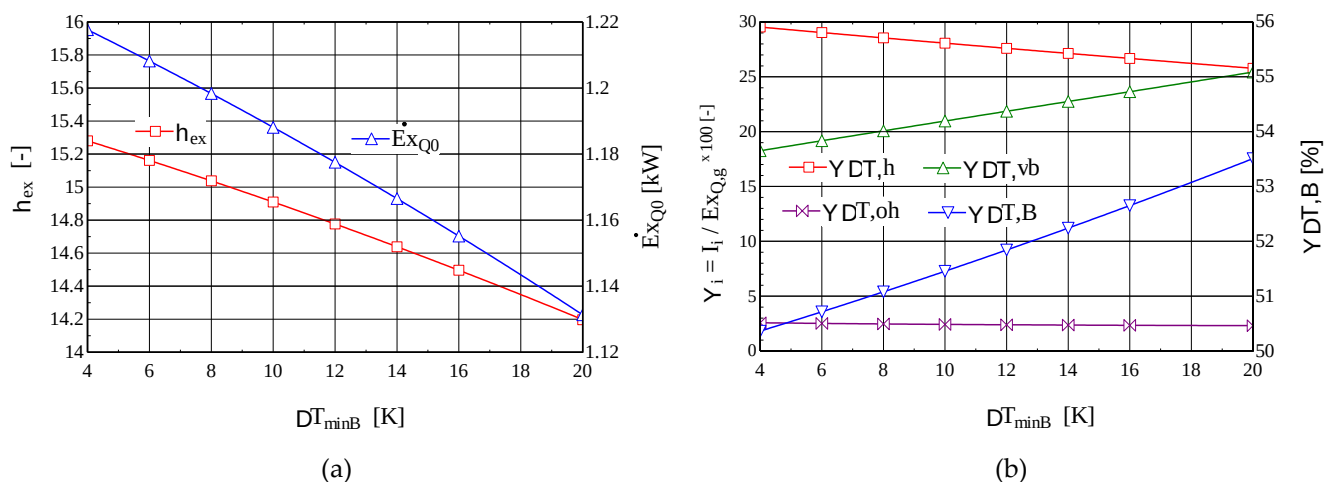


Figure 3. The ORC-MRC system: (a) the exergetic performance coefficient and the exergy of the heat current extracted in the MRC evaporator depending on the minimum temperature difference Δt_{minB} in the ORC boiler; (b) the share in the exergy consumption of the overall system, of the exergy destruction from the heating, boiling and overheating areas of the boiler, depending on the minimum temperature difference Δt_{minB}

When the temperature difference Δt_{minB} increases, the exergy destruction due to the heating process $\psi_{\Delta T,h}$ decreases, and that due to the vaporization process $\psi_{\Delta T,vb}$ increases (Figure 3 (b)). Overall, the exergy destruction in the boiler increases with the increase in the temperature difference between the fluids, which leads to a decrease in the exergetic efficiency of the cycle and a decrease in the exergy of the refrigerating power (Figure 3 (a)).

2.4.2 The behavior of the ORC-MRC cycle at the variation of the overheating degree $\Delta t_{oh,B}$ in the ORC boiler

Increasing the degree of overheating in the boiler has a positive effect, leading to an increase in the exergetic efficiency of the combined cycle and an increase in the exergy of the refrigerating power (Figure 4 (a)).

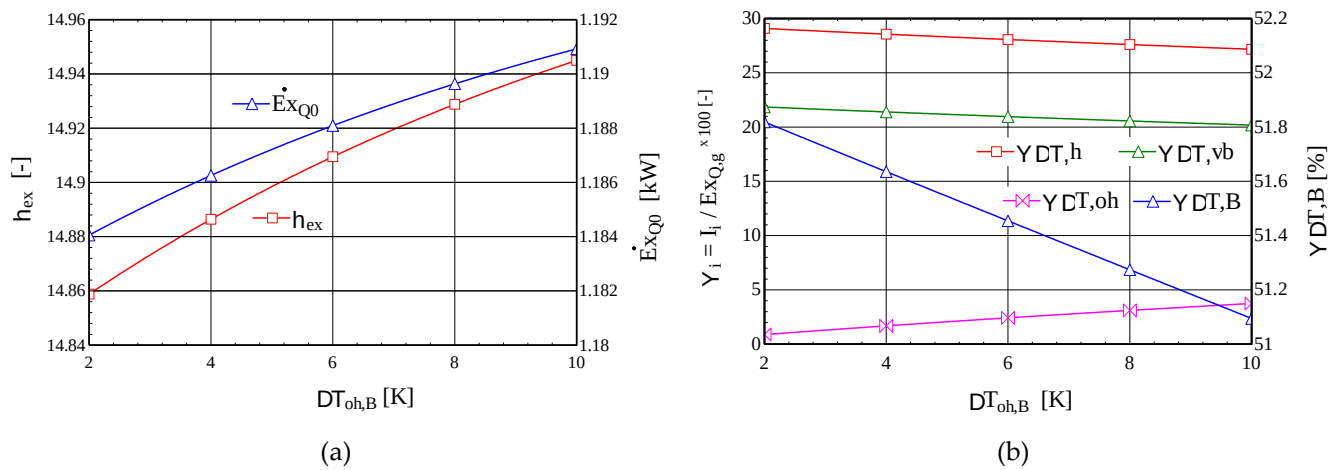


Figure 4. The ORC-MRC system: (a) the exergetic performance coefficient and the exergy of the heat stream extracted in the MRC evaporator depending on the degree of overheating in the ORC boiler, $\Delta t_{oh,B}$; (b) the share in the exergy consumption of the global system, of exergy destruction from the heating, boiling and overheating areas of the boiler, depending on the degree of overheating in the ORC boiler, $\Delta t_{oh,B}$

Under the conditions of constant thermal flow transferred to the ORC boiler from the combustion gases, the increase in the degree of overheating $\Delta t_{oh,B}$ of the ORC fluid leads to the redistribution of thermal loads from the boiler in the liquid heating, vaporization and superheating phases. The relative exergy destructions from the phases of liquid heating $\psi_{\Delta T,h}$ and vaporization $\psi_{\Delta T,vb}$, which have the highest weight, decrease faster than the increase in the weight of exergy destruction associated with overheating $\psi_{\Delta T,oh}$ (Figure 4 (b)); overall, the exergy destruction caused by the irreversibility of the heat transfer at the finite temperature difference in the boiler $\psi_{\Delta T,B}$ decreases, leading to the increase in the exergy efficiency of the global system and to the increase in the exergy of the refrigerating power (Figure 4 (a)).

2.4.3 The behavior of the ORC-MRC cycle at the variation of the minimum temperature difference Δt_{minCD} in the condenser

When the minimum temperature difference Δt_{minCD} in the condenser increases, the exergetic efficiency of the ORC-MRC system and the exergy of the refrigerating power decrease (Figure 5).

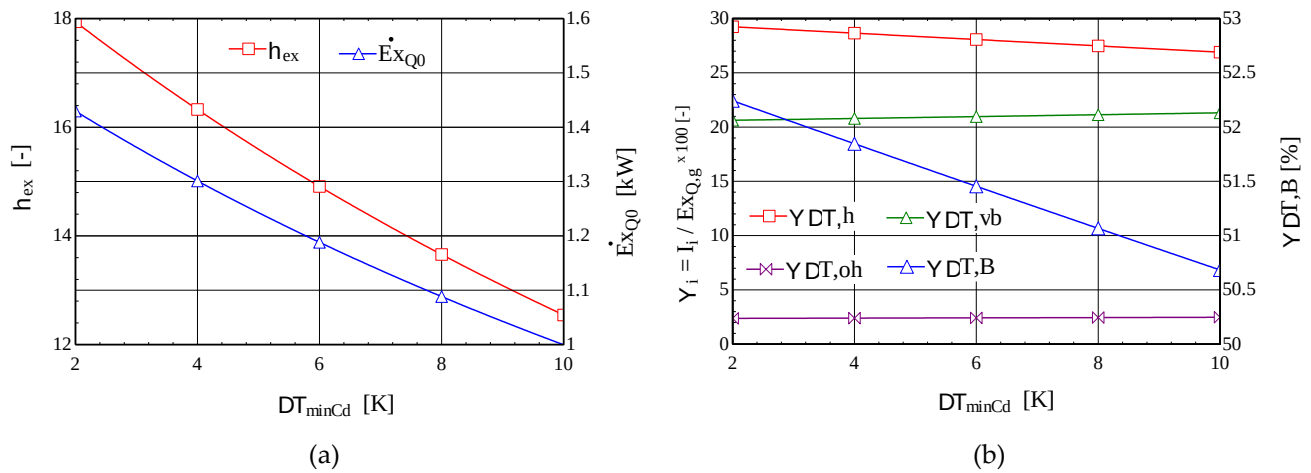


Figure 5. The ORC-MRC system: (a) the exergetic performance coefficient and the exergy of the heat current extracted in the MRC evaporator depending on the temperature difference Δt_{minCd} in the condenser; (b) the share in the exergy consumption of the overall system, of the exergy destruction in the heating, boiling and overheating areas of the boiler, depending on the temperature difference Δt_{minCd} in the condenser.

The increase in the minimum temperature difference Δt_{minCd} in the condenser leads to an increase in the condensation temperature, which leads to a decrease in the refrigerating power and its exergy $\dot{Ex}_{Q0}$ (Figure 5 (a)). The decrease in the exergy of the refrigeration power is the factor that determines the decrease in the exergy efficiency of the global system despite the reduction of the relative exergy destruction $\psi_{\Delta T,h}$ associated with the heating of the ORC fluid in the boiler and the consequent reduction of the total exergy destruction in the boiler $\psi_{\Delta T,B}$ (Figure 5 (b)).

In Figure 6 are represented in coordinates $(\theta = 1 - \frac{T_0}{T})$ (exergetic temperature factor) – H, the curve of the hot stream represented by the combustion gases, and the composite curve of the cold stream represented by the ORC fluid from the ORC boiler.

The area between the flue gas curve and the H axis gives the measure of the total exergy of the heat transferred from the flue gases, and the area between the composite curve of the working fluid (ORC), in the three stages, heating-vaporization-superheating, and the H axis gives the measure of the exergy received by the working fluid. The areas between the two curves give the measure of exergy destruction in each distinctive part: heater, vaporizer and superheater.

From figure 6 (a) it can be seen that for the simple ORC system (Figure 1) a large part of the amount of exergy introduced through the combustion gases is consumed in the boiler as follows: 28.07% in the heater, in the vaporizer 20.96% and only 2.425% in the superheater. The recorded values are due to the large temperature difference in the different areas of the boiler.

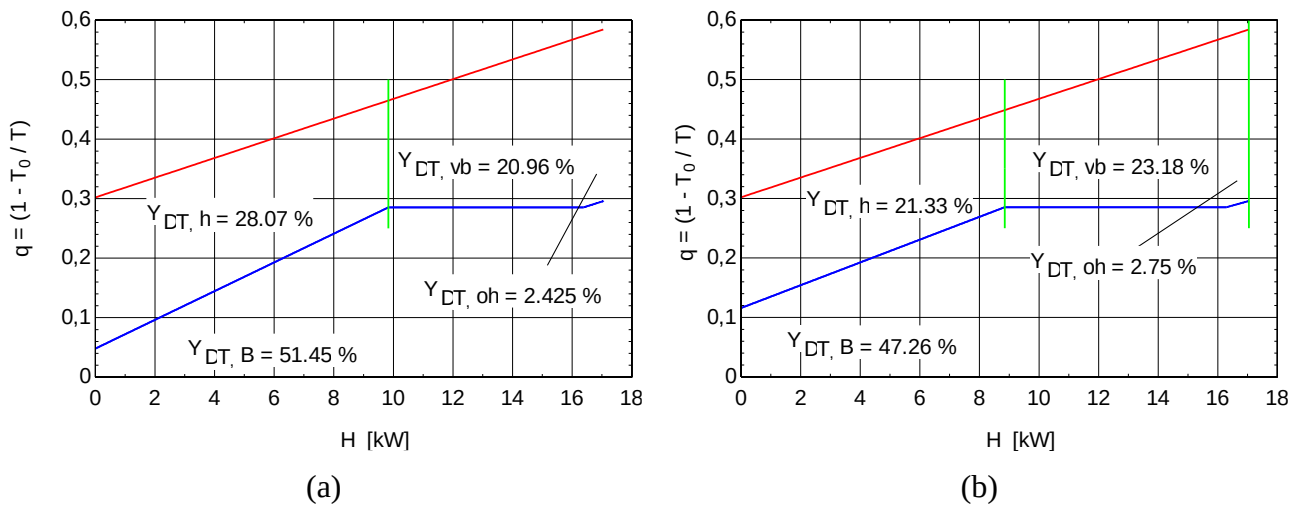


Figure 6. θ -H diagram of hot and cold currents from the ORC boiler: (a) simple ORC system (Figure 1); (b) ORC system with internal recuperative heat exchanger (Figure 7).

From figure 6 (a), it becomes evident that the largest area between the two curves (and the largest exergy destruction) is associated with heat transfer at a too large temperature difference in the heating and vaporization stage of the working fluid. The reduction of these surfaces and therefore of the exergy consumed, can be achieved, for the same vaporization temperature, by increasing the inlet temperature of the ORC fluid in the heating zone; this can be done, if the ORC fluid allows it (it is of a dry type) by changing the structure of the ORC cycle by introducing an internal recuperative exchanger (Figure 7). The biggest effect for reducing exergy destruction in the boiler is the increase of the vaporization temperature up to the limit allowed by the temperature difference at Pinch in this device.

2.5 ORC-MRC scheme with internal heat exchanger on the ORC circuit

Figure 7 shows the scheme of the ORC-MRC with internal heat exchanger on the ORC circuit.

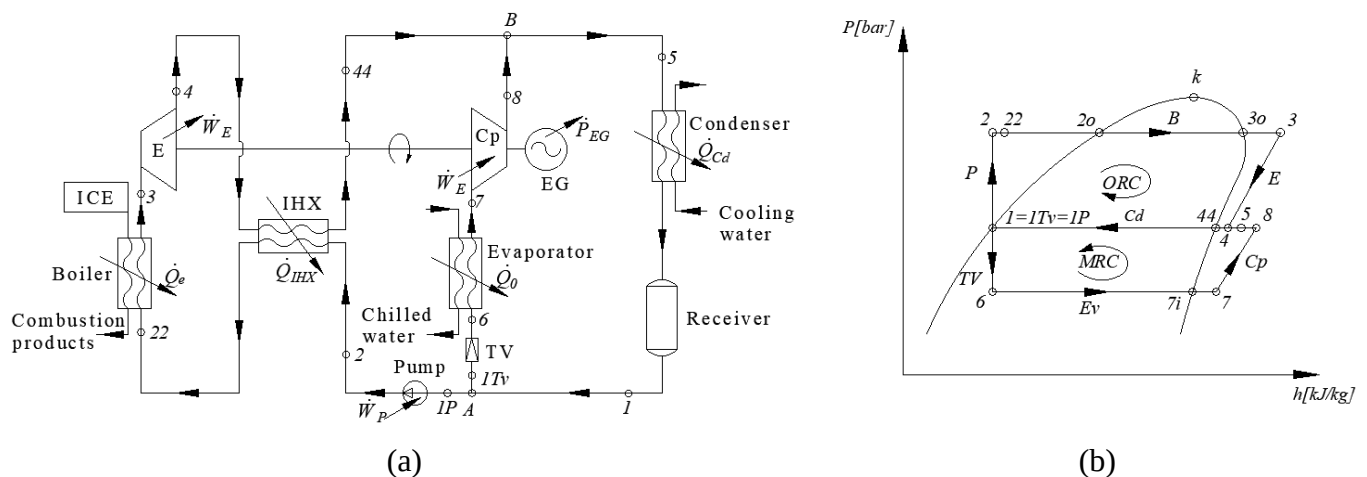


Figure 7. ORC - MRC system with ORC with internal regenerative heat exchanger; (a) Scheme of the ORC-MRC system; (b) the thermodynamic cycle of the ORC-MRC system in the p-h diagram.

The results of the mathematical modeling based on the exergetic analysis of the operation of the ORC-MRC cycle with an internal heat exchanger on the ORC circuit for a 100% engine load are presented in table 3.

Table 3. Energy and exergy results obtained for the ORC-MRC scheme with internal heat exchanger on the ORC circuit

COP_{MRC} [-]	COP_{ORC} [-]	$COP_{ORC-MRC}$ [-]	η_{ex} [%]	$\dot{m}_{MRC}$ [kg/s]	$\dot{m}_{ORC}$ [kg/s]	$\dot{m}_{ORC-MRC}$ [kg/s]	ψ_{Cp} [%]	ψ_E [%]	$\psi_{\Delta T,B}$ [%]	$\psi_{\Delta T,h}$ [%]
6.108	0.1883	1.076	16.94	0.1237	0.08642	0.2101	5.233	6.21	47.26	21.33

$\psi_{\Delta T,oh}$ [%]	$\psi_{\Delta T,vb}$ [%]	ψ_P [%]	$\psi_{L,Cd}$ [%]	ψ_{TV} [%]	ψ_m [%]	$\dot{Q}_B$ [kW]	$\dot{Q}_{Cd}$ [kW]	$\dot{Q}_{Ev}$ [kW]	$\dot{W}_E$ [kW]	$ \dot{W} _P$ [kW]
2.751	23.18	0.991	19.34	3.648	0.1194	17.05	33.38	18.34	3.21	0.2072

Analyzing Figure 6(a) compared to Figure 6(b), it is observed the decrease of the area between the two curves, when the internal heat exchanger is introduced in the ORC scheme, especially in the heating and vaporization stages of the working fluid. This decrease in the surface area leads to a decrease in the share of the exergy destruction in the ORC boiler by 4.19% and finally to an increase in the overall exergetic efficiency by 2.03% and implicitly in the COP of the ORC-MRC installation with internal heat exchanger.

3. Exergoeconomic optimization of the coupled system ORC-MRC-basic scheme

3.1. Exergoeconomic analysis - general principles

Exergoeconomic analysis is the only investigation method and optimization procedure that takes into account the fact that any thermodynamic system interacts with two environments:

- a physical environment determined by a system of intensive parameters such as pressure, temperature and chemical potential, and
- an economic environment characterized by prices of raw materials and equipment and sets of regulations to ensure sustainable development.

In defining the method of searching for optimal functional and constructive solutions, the exergoeconomic analysis is based on the union between the thermodynamics of irreversible processes and the economic analysis.

Given that energy is a conservative measure (Principle I of thermodynamics) and that, in principle, it is not consumed and therefore cannot appear, conceptually, in economic balance sheets, another, non-conservative concept must be found to define the consumption of usable energy, this is the exergy.

The exergoeconomic analysis lays the foundations for a methodology to search for the optimum, constructively and functionally in sight, offering the user clear rules for improving the studied system, whose effects can be followed step by step.

Unlike exergoeconomic analysis, any other optimization procedure based on statistical, evolutionary or mathematical algorithms represents black boxes for the user and the results must be accepted on trust. In addition, any mathematical optimization method performs the search within the limits of the specified scheme without providing any hints on its structural improvement.

The exergetic analysis method makes the connection between the system and its surrounding physical environment with which it interacts, using the concept of exergy that quantifies the value of use of each exergy stream highlighting the place and size of the exergy consumed (destroyed).

Written with an economic tone, the exergetic balance equation (Figure 8 (a))

$$\Sigma Ex_i = \Sigma Ex_o + \Sigma I \quad (44)$$

becomes:

$$\dot{E}x_F = \dot{E}x_P + \Sigma \dot{E}x_L + \Sigma \dot{I} \quad (45)$$

where:

$\dot{E}x_F$ – the exergy current of the resources used (generically called fuel)

$\dot{E}x_P$ – the exergy current of the product

$\dot{E}x_L$ – the exergy of the loss currents

$\dot{I}$ – consumption or destruction of exergy due to irreversibility (of work processes)

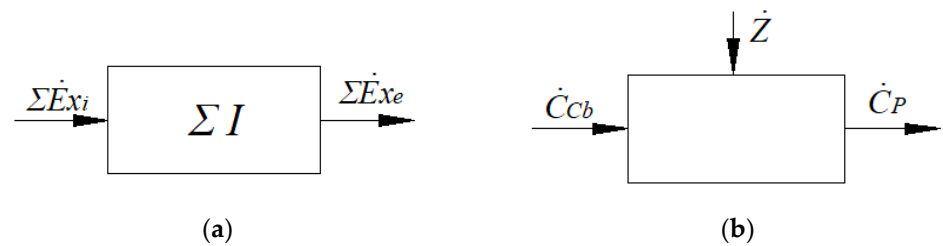


Figure 8. Balance sheets: **(a)** exergy balance for a control volume; **(b)** exergoeconomic balance

The exergoeconomic analysis makes the economic balance accounting both the monetary flows related to the operating process and those of investment (Figure 8 (b))

The monetary balance equation is of the form:

$$\dot{C}_P = \dot{C}_F + \dot{Z} \quad (46)$$

in which:

$\dot{C}_P \left[\frac{EUR}{s} \right]$ – the monetary cost of the product stream

$\dot{C}_F \left[\frac{EUR}{s} \right]$ – the monetary cost of the fuel flow

$\dot{Z} \left[\frac{EUR}{s} \right]$ – the amortization rate of the invested capital

For the calculation of the amortization rate of the invested capital, a capital recovery factor CRF with the following formula, is applied:

$$CRF = \left[1 + \frac{(1 + i_{ef})^n}{(1 + i_{ef})^{n-1}} \right] \quad (47)$$

where $i_{ef} = 0.05$ represents the interest (5% per year), and $n=10$ represents the economic life period.

The amortization rate of the invested capital for an equipment is calculated as follows:

$$\dot{Z} \left[\frac{EUR}{s} \right] = Z[EUR] \cdot \frac{CRF}{n_h \cdot 3600} \quad (48)$$

where $n_h = 7000$ hours is the number of operating hours per year, and the amortization rate of the invested capital per hour is:

$$\dot{Z}_h \left[\frac{EUR}{hour} \right] = Z \left[\frac{EUR}{s} \right] \cdot 3600[s] \quad (49)$$

If the exergy losses can be reduced by sealing, isolation or recovery, the reduction of exergy destructions, which is the essential source of increasing the performance

coefficient, is achieved by increasing the investment expenses. The antagonistic evolution of the operating cost (caused by exergy destruction) and the investment cost leads to the optimal solution given by the minimization of the total cost as the sum of the operating and investment costs.

The exergoeconomic optimization method aims to identify the exergy destruction of each functional area and assign its monetary cost.

Areas with high cost of exergy destruction will be the first targets of the optimization procedures, the effect of each local cost reduction will be verified at the global level.

In order to quantify the value of a zonal destruction, the fuel, the product, the loss and the exergetic performance coefficient will be evaluated for each operational area.

We will proceed with the search for the functional and constructive optimum for the ORC-MRC combined system in which the power Rankine cycle is a simple one without an internal recuperative heat exchanger.

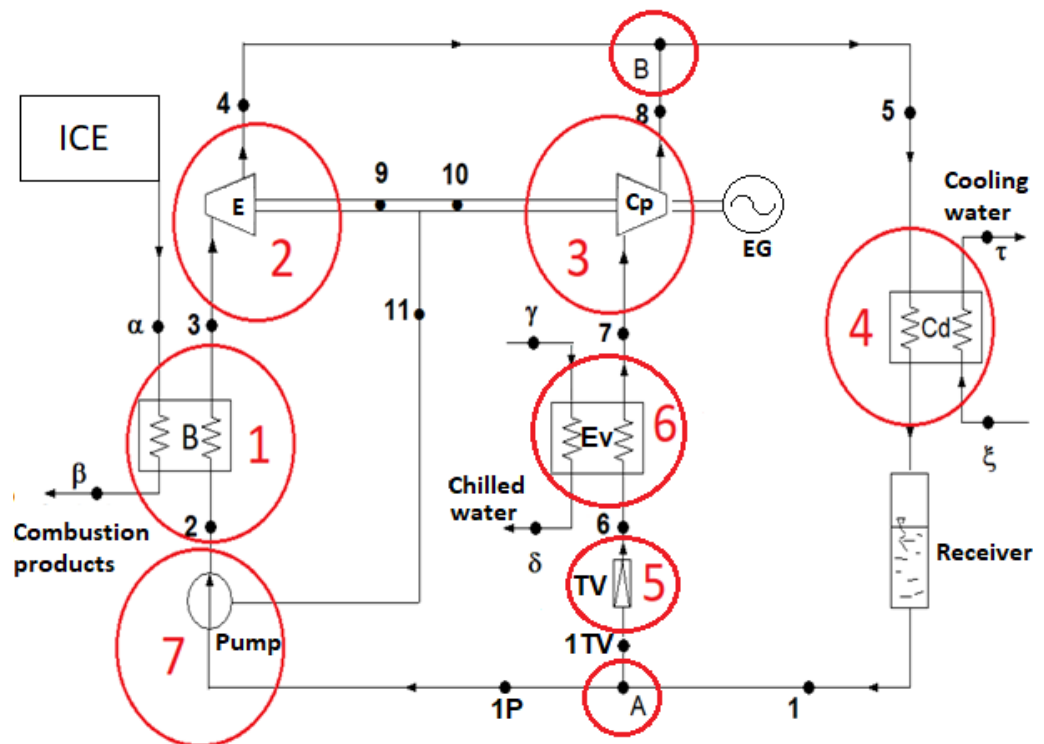


Figure 9. The ORC-MRC system divided in functional areas

The scheme of the MRC installation operated with the ORC system shown in figure 9 was divided into seven operating areas: 1) ORC Evaporator (Boiler); 2) Expander; 3) Compressor; 4) Condenser; 5) Throttling valve; 6) MRC Evaporator; 7) ORC liquid pump, to which node A for splitting and node B for mixing of substance currents are added.

The exergetic concept of Fuel, Product, Loss and Destruction for each functional area are presented in table 4 [51].

Table 4. The concepts of Fuel, Product, Exergy Loss and Destruction for functional areas

Equipment	Fuel	Product	Loss	Destruction
Boiler	$\dot{E}x_{\alpha}$	$\dot{E}x_3 - \dot{E}x_2$	$\dot{E}x_{\beta}$	$\dot{E}x_{\alpha} - \dot{E}x_3 + \dot{E}x_2 - \dot{E}x_{\beta}$
Expander	$\dot{E}x_3 - \dot{E}x_4$	$ \dot{W}_D $		$\dot{E}x_3 - \dot{E}x_4 - \dot{W}_D$
Compressor	$ \dot{W}_{Cp} $	$\dot{E}x_8 - \dot{E}x_7$		$ \dot{W}_{Cp} - \dot{E}x_8 + \dot{E}x_7$
Condenser	$\dot{E}x_5 - \dot{E}x_1$		$\dot{E}x_{\tau} - \dot{E}x_{\xi}$	$\dot{E}x_5 - \dot{E}x_1 - \dot{E}x_{\tau} + \dot{E}x_{\xi}$
Throttling valve	$\dot{E}x_{1TV}$	$\dot{E}x_6$		$\dot{E}x_{1TV} - \dot{E}x_6$
Evaporator	$\dot{E}x_6 - \dot{E}x_7$	$\dot{E}x_{\delta} - \dot{E}x_{\gamma}$		$\dot{E}x_6 - \dot{E}x_7 - \dot{E}x_{\delta} + \dot{E}x_{\gamma}$

ORC Pump	$ \dot{W}_p $	$\dot{E}x_2 - \dot{E}x_{1p}$	$ \dot{W}_p - \dot{E}x_2 + \dot{E}x_{1p}$
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Observing from relationship (46) that the destruction I of a zonal exergy is part of the exergy of the fuel, it is logical to assign it as the unitary monetary cost, the unitary cost of the fuel of that specific area.

3.2 Exergoeconomic monetary cost assessing

Each internal unitary monetary cost represents part of the overall system's fuel cost and capital investment. These costs are determined by the system's interaction with the market.

A methodology must be found to allocate these costs imposed by the market to each internal current.

There are 19 currents: 1-11, 1P, 1TV, α , β , γ , δ , τ , ξ for which 19 relationships between them are needed.

Nine monetary balance relationships of the equation type (3.3) can be written for the seven established functional areas and for the two nodes A and B, respectively.

The rest of the equations up to 19 will be auxiliary relationships between the monetary costs of the various exergy currents.

The auxiliary equations are built based on some rules that can be stated based on principles of economic common sense [51]:

Rule 1. For an unconsumed fuel stream exiting a subsystem, the monetary cost per unit of exergy is equal to the unit cost of the fuel stream feeding the subsystem.

Rule 2. If a subsystem has several products, the monetary cost per unit of exergy of each product is the same

Rule 3. If a component of a product has several output currents, each is assigned the same unit monetary cost.

Rule 4. In the absence of an external evaluation, the loss streams are assigned zero unitary monetary cost.

Rule 5. In the absence of an external assessment, the unitary exergy costs of the input currents in the global system are equal to their exergies.

A mathematical model based on the exergoeconomic analysis is written for each functional zone (Figure 9):

- Zone 1. The ORC evaporator (Boiler)

The exergoeconomic monetary balance cost (Eq. 46) of the ORC boiler is:

$$c_\alpha \cdot \dot{E}x_\alpha + c_2 \cdot \dot{E}x_2 - c_\beta \cdot \dot{E}x_\beta - c_3 \cdot \dot{E}x_3 + \dot{Z}_B = 0 \quad (50)$$

The exergy of combustion gases $\dot{E}x_\alpha$ is calculated as the sum of their thermomechanical and chemical exergy, at a certain operating mode of the engine, compared to the standard composition of the ambient environment.

Combustion gases being the unconsumed part of diesel fuel (internal combustion engine fuel) will be assigned the Diesel fuel unit exergetic monetary cost (Rule 1). It is considered that the mass exergy of diesel is $\dot{E}x_{CHF} = 41800$ [kJ/kg] and has a cost of 1.7 [EUR/kg]; it results:

$$c_F = c_\alpha = 4.07 \cdot 10^{-5} \left[\frac{\text{EUR}}{\text{kJ}} \right] \quad (51)$$

The exergy of current β (Figure 9) is a loss (Table 4) so that according to Rule 4

$$c_\beta = 0 \quad (52)$$

- Zone 2. Expander

The exergoeconomic monetary balance cost (Eq. 46) of the Expander is:

$$c_3 \cdot \dot{E}x_3 - c_9 \cdot \dot{W}_E - c_4 \cdot \dot{E}x_4 + \dot{Z}_E = 0 \quad (53)$$

Current 4 represents an unconsumed part of fuel 3 as per Rule 1 543

$$c_3 = c_4 \quad (54)$$

The unit monetary costs of currents 9, 10 and 11 are equal because they represent parts of the same product (Rule 3): 544 545

$$c_9 = c_{10} = c_{11} \quad (55,56)$$

- Zone 3. Compressor 546

The exergoeconomic monetary balance cost (Eq. 46) of the Compressor is: 547

$$c_7 \cdot \dot{E}x_7 + c_{10} \cdot |\dot{W}_{CP}| - c_8 \cdot \dot{E}x_8 + \dot{Z}_{CP} = 0 \quad (57)$$

- Zone B. Mixing point 548

In the mixing point B, the following monetary balance equation can be written 549

$$c_4 \cdot \dot{E}x_4 + c_8 \cdot \dot{E}x_8 = c_5 \cdot \dot{E}x_5 \quad (58)$$

- Zone 4. Condenser 550

The condenser of the ORC-MRC system is a dissipative zone and has no exergetic product (Table 4). 551 552

Its monetary balance equation based on exergoeconomic criteria is: 553

$$c_5 \cdot \dot{E}x_5 + c_\xi \cdot \dot{E}x_\xi - c_1 \cdot \dot{E}x_1 - c_\tau \cdot \dot{E}x_\tau + \dot{Z}_{Cd} = 0 \quad (59)$$

The exergy of the cooling water of the condenser not being used represents a loss. 554
According to Rule 4: 555

$$c_\xi = c_\tau = 0 \quad (60,61)$$

- Zone A. Splitting point 556

The following monetary balance equation can be written at the splitting point A 557

$$c_1 \cdot \dot{E}x_1 + c_{1TV} \cdot \dot{E}x_{1TV} - c_{1P} \cdot \dot{E}x_{1P} = 0 \quad (62)$$

where the specific monetary cost of currents 1TV and 1P is the same as representing a multiple product (Rule 2) 558 559

$$c_{1TV} = c_{1P} = c_1 \quad (63)$$

- Zone 5. Throttling valve 560

The exergoeconomic monetary balance cost (Eq. 46) of the Throttling valve is: 561

$$c_{1TV} \cdot \dot{E}x_{1TV} - c_6 \cdot \dot{E}x_6 + \dot{Z}_{TV} = 0 \quad (64)$$

- Zone 6. MRC Evaporator 562

The exergoeconomic monetary balance cost (Eq. 46) of the Evaporator is: 563

$$c_6 \cdot \dot{E}x_6 + c_\gamma \cdot \dot{E}x_\gamma - c_7 \cdot \dot{E}x_7 - c_8 \cdot \dot{E}x_8 + \dot{Z}_{Ev} = 0 \quad (65)$$

The exergy current 7 is the unconsumed part of the fuel 6 that feeds the evaporator. 564
According to Rule 1, the two currents have the same monetary unit cost: 565

$$c_7 = c_6 \quad (66)$$

According to Rule 5 a value is assigned to current γ 566

$$c_\gamma = 1 \quad (67)$$

It is observed that the value of the unitary monetary cost of the chilled water current γ is not important in the calculation because the state γ is taken as a reference for the calculation of the exergy of the chilled water ($\dot{E}x_\gamma = 0$)

- Zone 7. ORC liquid pump

For the ORC liquid pump the monetary cost balance equation becomes:

$$c_{1P} \cdot \dot{E}x_{1P} + c_{11} \cdot |\dot{W}_P| - c_2 \cdot \dot{E}x_2 + \dot{Z}_P = 0 \quad (68)$$

Equations (51 – 69) represent the 19 equations of the mathematical model for the calculation of the 19 monetary unit costs.

3.3 Exergoeconomic cost correlations of component equipment

Unlike thermoeconomic cost correlations in which the purchase cost of an equipment is estimated as a function of the amount of material resources used, in the case of exergoeconomic correlations, the cost is specified according to the exergetic performance coefficient (or defined based on the Second Law of Thermodynamics) or of the decision variables that define the exergetic performance (such as exergy destruction) and function of the size of the exergetic product of the equipment [4, 52-55].

- MRC compressor

The exergoeconomic cost correlation is of the form [52]

$$Z_{Cp} = 71 \frac{\dot{m}_{MRC}}{0,9 - \eta_{scp}} \cdot r_{cp} \cdot \ln r_{cp} \text{ [EUR]} \quad (69)$$

where the performance coefficient of the equipment defined on the basis of the Second Law of Thermodynamics is the isentropic efficiency η_{scp} of the compression process, and the exergetic product is

$$P_{ex)Cp} = \dot{m}_{MRC} \cdot r_{cp} \cdot \ln r_{cp} \left[\frac{kg}{s} \right] \quad (70)$$

which marks the antagonistic effect of the variation of the compressed gas flow rate $\dot{m}_{MRC}$ with the compression ratio $r_{cp} = \frac{p_c}{p_v} = \frac{p_8}{p_7}$ (Figure 9)

The investment cost Z_{Cp} [EUR] has been updated to the level of 2020.

- ORC expander

The exergoeconomic correlation for the acquisition cost of the expander (Zone 2 Fig 15) takes into account the isentropic efficiency of expansion η_{sd} as a measure of the perfection of the equipment through the prism of the Second law of Thermodynamics. The cost of the expander is proportional to its exergetic product as a measure of the mechanical work produced by expansion [52].

$$P_{ex)D} = \dot{m}_{ORC} \cdot \ln \frac{p_3}{p_4} \left[\frac{kg}{s} \right] \quad (71)$$

The notations correspond to Figure 9, which shows the technological scheme of the installation.

With these considerations and taking into account the cost update at the level of 2020, the exergoeconomic cost correlation for the expander becomes [106]

$$Z_E = 479 \frac{\dot{m}_{ORC}}{0,92 - \eta_{se}} \cdot \ln \frac{p_3}{p_4} \text{ [EUR]} \quad (72)$$

- ORC pump

For the pump, a form of exergoeconomic correlation similar to that used for the compressor is chosen [52]

$$Z_P = 10,38 \frac{\dot{m}_{ORC}}{0,9 - \eta_{sp}} r_P \cdot \ln r_P \text{ [EUR]} \quad (73)$$

in which the performance of the equipment in relation to the Second Law of Thermodynamics is described by the isentropic compression efficiency η_{sp} and the exergetic product correlates the pumped flow rate $\dot{m}_{ORC}$ with the pressure increase ratio $r_p = \frac{p_2}{p_1}$ (Figure 9)

d) Heat exchangers

In the case of heat exchangers, to specify the cost of the purchase, the starting point is the thermo-economic correlation

$$Z = c \cdot A \text{ [EUR]} \quad (74)$$

where $c[\text{EUR}/\text{m}^2]$ represents the cost of the heat exchange surface unit and $A[\text{m}^2]$, its surface.

In order to highlight the decision-making parameter that imposes the exergetic performance of the device, the heat exchange surface is explained according to the minimum temperature difference in the device. The minimum temperature difference $\Delta T_{\min}$ decides the amount of exergy destruction in the device. This decision-making parameter makes the balance between the operating expense of the device (exergy consumption) and its acquisition cost (heat exchange surface).

$$A = \frac{\dot{Q}}{U \cdot \Delta T_m} = \frac{\dot{m} \cdot \Delta h}{U \cdot \Delta T_m} \quad (75)$$

where $U[\text{W}/(\text{m}^2 \cdot \text{K})]$ represents the global heat exchange coefficient of the device, and ΔT_m is the average logarithmic temperature difference in the device.

d.1 The refrigerating unit evaporator

The temperature variation diagram as a function of the heat transfer surface is shown in Figure 2 (c), where:

$$\Delta T_{\min}^V = t_\delta - t_7 \quad (76)$$

$$\Delta T_m^V = \frac{t_\gamma - t_\delta}{\ln \frac{t_\gamma - t_V}{t_\delta - t_V}} = \frac{t_\gamma - t_\delta}{\ln \frac{t_\gamma - (t_\delta - \Delta T_{\min}^V)}{\Delta T_{\min}^V}} = \frac{t_\gamma - t_\delta}{\ln \frac{t_\gamma - t_\delta + \Delta T_{\min}^V}{\Delta T_{\min}^V}} \quad (77)$$

Considering the temperature drop on the cooled fluid side as an imposed design data

$$t_\gamma - t_\delta = a \quad (78)$$

relationship (78) becomes:

$$\Delta T_m^V = \frac{a}{\ln \frac{a + \Delta T_{\min}^V}{\Delta T_{\min}^V}} \quad (79)$$

Taking (80) into account, equation (75) becomes:

$$Z_V = c_V \frac{\dot{Q}_V \ln \frac{a + \Delta T_{\min}^V}{\Delta T_{\min}^V}}{a \cdot U_V} \quad (80)$$

Considering $c_V = 87 [\text{EUR}/\text{m}^2]$, $U_V = 500 [\text{W}/(\text{m}^2 \cdot \text{K})]$ and choosing $a = 16 \text{ K}$, Equation (81) becomes:

$$Z_V[\text{EUR}] = 10,87 \cdot \dot{Q}_V[\text{kW}] \ln(16 \cdot (\Delta T_{\min}^V)^{-1} + 1) \quad (81)$$

d.2 Condenser

Applying to the condenser the same scheme used in the case of the MRC evaporator, the exergoeconomic correlation is obtained:

$$Z_{cd} = c_{cd} \frac{\dot{Q}_{cd} \ln \frac{b + \Delta T_{min}^{cd}}{\Delta T_{min}^{cd}}}{b \cdot U_{cd}} \quad (82)$$

in which:

$$b = t_{\tau} - t_{\xi} \quad (\text{Fig 2 (b)}) \quad (83)$$

is imposed by the design, and

$$\Delta T_{min}^{cd} = t_c - t_{\tau} \quad (\text{Fig. 2 (c)}) \quad (84)$$

Considering that $c_{cd} = 254 [EUR/m^2]$, $U_{cd} = 1750 [W/(m^2 \cdot K)]$ and $b = 7 K$, Equation (83) becomes:

$$Z_{cd}[EUR] = 20,73 \cdot \dot{Q}_{cd}[kW] \ln \left(7 \cdot (\Delta T_{min}^{cd})^{-1} + 1 \right) \quad (85)$$

d.3 ORC boiler

In the case of the ORC boiler, the exergoeconomic correlation for the calculation of the purchase cost is of the form:

$$Z_B [EUR] = c_B \left[\frac{EUR}{m^2} \right] \cdot A_B [m^2] \quad (86)$$

where the heat transfer surface is the sum of the surfaces corresponding to the processes of heating the subcooled liquid (A_h), boiling (A_{vb}) and overheating (A_{oh}).

$$A_B = A_h + A_{vb} + A_{oh} \quad (87)$$

The heat exchange surfaces of each functional area of the boiler are calculated as follows:

- Liquid heating area

$$A_h = \frac{\dot{Q}_h}{U_h \cdot \Delta T_m^h} \quad (88)$$

with:

$$\dot{Q}_h = \dot{m}_{ORC}(h_{20} - h_2) \quad (\text{Fig. 2 (a)}) \quad (89)$$

$U_h = 70 [W/(m^2 \cdot K)]$ - the global heat transfer coefficient for the liquid heating process:

$$\Delta T_m^h = \frac{(t_{\beta} - t_2) - (t_{h,g} - t_{20})}{\ln \frac{(t_{\beta} - t_2)}{(t_{h,g} - t_{20})}} \quad (90)$$

Temperature $t_{h,g}$ on the heating gas side results from the energy balance of the liquid heating zone (Fig.2 (a)):

$$t_{h,g} = t_{\beta} + \frac{\dot{m}_{ORC}(h_{20} - h_2)}{\dot{m}_g \cdot c_g} \quad (91)$$

with $t_{\beta} = 140^\circ C$.

- Boiling area

$$A_{vb} = \frac{\dot{Q}_{vb}}{U_{vb} \cdot \Delta T_m^{vb}} = \frac{\dot{m}_{ORC}(h_{30} - h_{20})}{U_{vb} \cdot \Delta T_m^{vb}} \quad (92)$$

with $U_{vb} = 90 [W/(m^2 \cdot K)]$ - the global heat transfer coefficient for the boiling process:

$$\Delta T_{vb} = \frac{(t_{vb,g} - t_{h,g})}{\ln \frac{(t_{vb,g} - t_v)}{(t_{h,g} - t_v)}} \quad (\text{Fig. 2 (a)}) \quad (93)$$

Temperature $t_{vb,g}$ on the heating gas side results from the energy balance of the boiling zone:

$$t_{vb,g} = t_{h,g} + \frac{\dot{Q}_{vb}}{\dot{m}_g \cdot c_g} \quad (94)$$

• Overheating area:

$$A_h = \frac{\dot{Q}_{oh}}{U_{oh} \cdot \Delta T_m^{oh}} = \frac{\dot{m}_{ORC}(t_3 - t_v)}{U_{oh} \cdot \Delta T_m^{oh}} \quad (95)$$

with:

$U_{oh} = 50 [W/(m^2 \cdot K)]$ the global heat transfer coefficient for the overheating process

$$\Delta T_{oh} = \frac{(t_\alpha - t_3) - (t_{vb,g} - t_{30})}{\ln \frac{(t_\alpha - t_3)}{(t_{vb,g} - t_{30})}} \quad (96)$$

where $t_\alpha = 420^\circ\text{C}$

3.4 Exergoeconomic performance indicators

3.4.1. Monetary cost of zonal exergy destruction

To calculate the monetary cost of the exergy destruction in each functional area, the value of the zonal exergy destruction is multiplied by the unitary monetary cost of the local, zonal fuel (table 5)

Table 5. Unit cost of zonal fuel and monetary cost rate of zonal exergy destruction

Equipment	Zonal fuel unit cost $\left[\frac{EUR}{kJex}\right]$	Monetary cost rate of zonal exergy destruction $\dot{C}_I \left[\frac{EUR}{s}\right]$
Boiler	c_{gMAI}	$c_{gMAI}(\dot{E}x_\alpha - \dot{E}x_3 + \dot{E}x_2 - \dot{E}x_\beta)$
Expander	c_3	$c_3(\dot{E}x_3 - \dot{E}x_4 - \dot{W}_D)$
Compressor	c_{10}	$c_{10}(\dot{W}_{cp} - \dot{E}x_8 - \dot{E}x_7)$
Condenser	c_5	$c_5(\dot{E}x_5 - \dot{E}x_1 - \dot{E}x_\tau + \dot{E}x_\xi)$
Throttling valve	c_1	$c_1(\dot{E}x_{1TV} - \dot{E}x_6)$
Evaporator	c_6	$c_6(\dot{E}x_6 - \dot{E}x_7 - \dot{E}x_\delta + \dot{E}x_\gamma)$
ORC Pump	c_{11}	$c_{11}(\dot{W}_p + \dot{E}x_{1p} - \dot{E}x_2)$

3.4.2 Exergoeconomic factor

To appreciate how the investment expense saves the monetary cost of zonal exergy destruction, the exergoeconomic factor [56] is calculated for each piece of equipment, where the index k represents the piece of equipment.

$$f_k = \frac{\dot{Z}_k}{\dot{Z}_k + \dot{C}_{I,k}} < 1 \quad (97)$$

If f_k approaches 1, it means that the operational cost (of exergy destruction $\dot{C}_{I,k}$) is too low and the investment cost $\dot{Z}_k$ is too high. In this case the exergetic performance of the device must be relaxed and a higher exergy destruction should be accepted; the total cost consisting of operating and investment costs will decrease.

If f_k is too small it means that the monetary value of the exergy destruction is too high and it is recommended to purchase a better equipment which is obviously more expensive.

3.4.3 Zonal coefficients of performance

a) Exergetic efficiency

For each functional area, the Product and the Fuel were specified (Table 8.1). Their ratio gives the zonal exergetic efficiency [51].

$$\eta_{ex,k} = \left(\frac{P}{F} \right)_k 100 \quad (98)$$

For the global ORC-MCR system, the product is the heat exergy extracted from the chilled water in the MCR evaporator (evaporator product (Table 4)) and the fuel is the exergy of the combustion gases $\dot{E}x_\alpha$

$$\eta_{ex} = \frac{P_{Ev}}{\dot{E}x_\alpha} 100 \quad (99)$$

b) The relative weight of an exergy destruction in the fuel consumption of the global ORC-MCR system

The percentage weight of the zonal exergy destruction in the fuel consumption of the global system is calculated with the relationship:

$$\psi_k = \frac{I_k}{\dot{E}x_\alpha} 100 \quad (100)$$

The search procedure for the optimal functional and constructive ORC-MCR system will be guided by the values of the exergoeconomic performance indicators.

3.5 Results of exergoeconomic optimization

The following tables show the simulation results for the working fluid R1224yd(Z) depending on the variation of several decision parameters, including the minimum temperature difference in the Condenser ΔT_{minCd} , the minimum temperature difference in the Evaporator ΔT_{minV} , the minimum temperature difference in the Boiler ΔT_{minB} , as well as the variation of the efficiency of the Compressor η_{sCp} or the Expander η_{sE} . The unit cost of diesel was considered $c_{Diesel} = 1.7$ [EUR/l].

Table 6. The results obtained for $\Delta T_{minCd} = 8$ K; $\Delta T_{minV} = 8$ K; $\Delta T_{minB} = 30$ K; $\eta_{sCp} = 0.8$; $\eta_{sP} = 0.8$; $\eta_{sE} = 0.8$

Zone	η_{ex} [%]	$\dot{I}$ [kW]	Ψ [%]	$\dot{Z}$ [EUR/h]	CI [EUR/kJ]	$\dot{C}I = CI \cdot \dot{I}$ [EUR/h]	$\dot{C}I + \dot{Z}$ [EUR/h]	f
Boiler	31.43	4.211	37.17	0.01067	$4.07 \cdot 10^{-5}$	0.617	0.6277	0.01701
Expander	82.25	0.4939	4.359	0.01204	$2.021 \cdot 10^{-4}$	0.3593	0.3713	0.03242
Compressor	81.65	0.4035	3.561	0.005774	$2.471 \cdot 10^{-4}$	0.359	0.3648	0.01583
Condenser		1.2	13.63	0.00695	$2.535 \cdot 10^{-4}$	1.095	1.102	0.006306
MRC Evaporator	41.37	0.5094	4.496	0.002576	$5.432 \cdot 10^{-4}$	0.9962	0.9988	0.002579
ORC Pump	81.03	0.01719	0.1517	0.002539	$2.471 \cdot 10^{-4}$	0.0153	0.01783	0.1423

It results the cost of the product exergy unit c_R [EUR/kJex] (refrigerating power of the MRC Evaporator)

$$c_R = 13,15 \cdot 10^{-4} \text{ [EUR/kJex]}$$

The lowest exergetic efficiency η_{ex} and the highest relative destruction is in the Boiler. The small exergetic factor requires the choice of a larger heat exchange surface (a lower minimum temperature difference) to reduce exergy destruction. E.g:

$$\Delta T_{minB} = 20 \text{ [K]}$$

$$c_R = 12.35 \cdot 10^{-4} \text{ [EUR/kJex]}$$

and for $\Delta T_{minB} = 10 [K]$

$$c_R = 11.73 \cdot 10^{-4} [EUR/kJex]$$

The MRC evaporator has the exergoeconomic factor $f=0.002579$ being the lowest, as can be seen in table 6, which marks a high cost of exergy destruction compared to the cost of amortization of the invested capital. The suggestion is to reduce the temperature difference and thus increase the heat exchange surface of the evaporator.

Table 7. The results obtained for $\Delta T_{minCd}=8 K$; $\Delta T_{minB}=10 K$; $\eta_{sCp}=0.8$; $\eta_{sP}=0.8$; $\eta_{sD}=0.8$ when $\Delta T_{minV}=3 K$ is imposed.

Zone	η_{ex} [%]	I [kW]	Ψ [%]	$\dot{Z}$ [EUR/h]	CI [EUR/kJ]	$\dot{CI} = CI \cdot \dot{I}$ [EUR/h]	$\dot{CI} + \dot{Z}$ [EUR/h]	f
Boiler	34.59	3.852	34	0.01137	$4.07 \cdot 10^{-5}$	0.5645	0.5758	0.01975
Expander	82.43	0.5559	4.906	0.01372	$1.723 \cdot 10^{-4}$	0.3448	0.3585	0.03826
Compressor	81.61	0.4547	4.013	0.005171	$2.105 \cdot 10^{-4}$	0.3446	0.3498	0.01479
Condenser		1.376	15.64	0.007987	$2.228 \cdot 10^{-4}$	1.103	1.111	0.007186
MRC Evaporator	56.41	0.3789	3.344	0.005873	$5.443 \cdot 10^{-4}$	0.7425	0.7484	0.007848
ORC Pump	81.05	0.02545	0.2246	0.004271	$2.105 \cdot 10^{-4}$	0.01928	0.02356	0.1813

It results the cost of the product exergy unit $c_R = 9.681 \cdot 10^{-4} [EUR/kJex]$.

The condenser has the lowest exergoeconomic factor as shown in Table 7. Exergy destruction must be reduced.

Although the condenser is an eminently dissipative area with the role of transferring to the environment the energy generated and transferred in the system, and therefore without an exergetic product, by specifying the condensation temperature it has an important role in establishing the efficiency in operation.

Table 8. The results obtained for $\Delta T_{minV}=3 K$; $\Delta T_{minB}=10 K$; $\eta_{sCp}=0.8$; $\eta_{sP}=0.8$ when $\Delta T_{minCd}=3 K$ is imposed

Zone	η_{ex} [%]	I [kW]	Ψ [%]	$\dot{Z}$ [EUR/h]	CI [EUR/kJ]	$\dot{CI} = CI \cdot \dot{I}$ [EUR/h]	$\dot{CI} + \dot{Z}$ [EUR/h]	f
Boiler	33.91	3.93	34.68	0.01117	$4.07 \cdot 10^{-5}$	0.5758	0.5869	0.01904
Expander	82.26	0.5872	5.182	0.01432	$1.615 \cdot 10^{-4}$	0.3415	0.3558	0.04024
Compressor	81.33	0.4837	4.269	0.004574	$1.979 \cdot 10^{-4}$	0.3445	0.3491	0.0131
Condenser		0.9541	12.4	0.01742	$2.147 \cdot 10^{-4}$	0.7374	0.7548	0.02307
MRC Evaporator	56.38	0.4878	4.305	0.007549	$4.258 \cdot 10^{-4}$	0.7476	0.7552	0.009996
ORC Pump	80.73	0.02535	0.2237	0.0053	$1.979 \cdot 10^{-4}$	0.01805	0.02335	0.2269

The cost of one unit of exergy product becomes $c_R = 7.585 \cdot 10^{-4} [EUR/kJex]$.

The cost of exergy destruction in the Pump, as well as the cost of amortizing the purchase of the pump, are the lowest in the system (table 8).

The highest value of the exergoeconomic factor of the pump area indicates the possibility of reducing the quality of the pump by reducing its isentropic efficiency η_{sP} , but the effect of this reduction on the thermodynamic parameters after pumping does not bring a decrease in the unit cost of the global system product represented by the cold obtained in the Evaporator.

The compressor has a low acquisition cost but induces a high cost of exergy destruction, which suggests the choice of a more expensive compressor with a higher isentropic compression efficiency as can be seen in the processed data in table 8.

Table 9. The results obtained for $\Delta T_{minCd}=3 K$; $\Delta T_{minV}=3 K$; $\Delta T_{minB}=10 K$; $\eta_{sP}=0.8$; when $\eta_{sCp}=0.85$ is imposed

Zone	η_{ex} [%]	I [kW]	Ψ [%]	$\dot{Z}$ [EUR/h]	CI [EUR/kJ]	$\dot{CI} = CI \cdot \dot{I}$ [EUR/h]	$\dot{CI} + \dot{Z}$ [EUR/h]	f
Boiler	33.91	3.93	34.68	0.01117	$4.07 \cdot 10^{-5}$	0.5758	0.5869	0.01904
Expander	82.26	0.5872	8.182	0.01432	$1.583 \cdot 10^{-4}$	0.3347	0.349	0.04102
Compressor	85.97	0.3636	3.209	0.009721	$1.939 \cdot 10^{-4}$	0.2538	0.2635	0.03688
Condenser		0.972	12.7	0.01801	$2.04 \cdot 10^{-4}$	0.7139	0.7319	0.02461
MRC Evaporator	56.38	0.5183	4.574	0.008021	$4.021 \cdot 10^{-4}$	0.7501	0.7582	0.01058
ORC Pump	80.73	0.02535	0.2237	0.0053	$1.939 \cdot 10^{-4}$	0.0177	0.023	0.2305

It results $c_R = 7.165 \cdot 10^{-4}$ [EUR/kJex], and if one considers $\eta_{sCp}=0.89$, the following results are obtained (Table 10).

Table 10. The results obtained for $\Delta T_{minCd}=3$ K; $\Delta T_{minV}=3$ K; $\Delta T_{minB}=10$ K; $\eta_{sE}=0.8$; when $\eta_{sCp}=0.89$ is imposed.

Zone	η_{ex} [%]	I [kW]	Ψ [%]	$\dot{Z}$ [EUR/h]	CI [EUR/kJ]	$\dot{CI} = CI \cdot \dot{I}$ [EUR/h]	$\dot{CI} + \dot{Z}$ [EUR/h]	f
Boiler	33.91	3.93	34.68	0.01117	$4.07 \cdot 10^{-5}$	0.5758	0.5869	0.01904
Expander	82.26	0.5872	5.182	0.01432	$1.571 \cdot 10^{-4}$	0.3322	0.3465	0.04132
Compressor	89.69	0.2761	2.357	0.05089	$1.925 \cdot 10^{-4}$	0.185	0.2359	0.2157
Condenser		0.9868	12.94	0.01849	$2.005 \cdot 10^{-4}$	0.7123	0.7308	0.0253
MRC Evaporator	56.38	0.5426	4.789	0.008398	$3.933 \cdot 10^{-4}$	0.7683	0.7767	0.01081
ORC Pump	80.73	0.02535	0.2237	0.0053	$1.925 \cdot 10^{-4}$	0.01756	0.02286	0.2318

It results for one unit of exergy of product a cost of $c_R = 7.009 \cdot 10^{-4}$ [EUR/kJex]

In the area of the expander, both the cost of amortization of the acquisition expense and the cost of exergy destruction are high, which indicates the possibility of reducing the expense, with exergy destruction by choosing a more efficient expander, a trend tempered however by the expander's cost being higher than, for example, the cost of the compressor.

Table 11. The results obtained for $\Delta T_{minCd}=3$ K; $\Delta T_{minV}=3$ K; $\Delta T_{minB}=10$ K; $\eta_{sCp}=0.89$; when $\eta_{sE}=0.85$ is imposed.

Zone	η_{ex} [%]	I [kW]	Ψ [%]	$\dot{Z}$ [EUR/h]	CI [EUR/kJ]	$\dot{CI} = CI \cdot \dot{I}$ [EUR/h]	$\dot{CI} + \dot{Z}$ [EUR/h]	f
Boiler	33.91	3.93	34.68	0.01117	$4.07 \cdot 10^{-5}$	0.5758	0.5869	0.01904
Expander	86.74	0.4422	3.903	0.02454	$1.54 \cdot 10^{-4}$	0.2451	0.2697	0.09101
Compressor	89.69	0.2846	2.512	0.05423	$1.799 \cdot 10^{-4}$	0.1843	0.2385	0.2274
Condenser		1.009	13.3	0.01919	$1.908 \cdot 10^{-4}$	0.6934	0.7125	0.02693
MRC Evaporator	56.38	0.5783	5.104	0.008949	$3.72 \cdot 10^{-4}$	0.7745	0.7835	0.01142
ORC Pump	80.73	0.02535	0.2237	0.0053	$1.799 \cdot 10^{-4}$	0.01641	0.02171	0.2441

In this case one unit of product exergy costs $c_R = 6.632 \cdot 10^{-4}$ [EUR/kJex]

With a more performant expander ($\eta_{sE}=0.9$) one gets (Table 12) :

Table 12. The results obtained for $\Delta T_{minCd}=3$ K; $\Delta T_{minV}=3$ K; $\Delta T_{minB}=10$ K; $\eta_{sCp}=0.89$; when $\eta_{sE}=0.9$ is imposed.

Zone	η_{ex} [%]	I [kW]	Ψ [%]	$\dot{Z}$ [EUR/h]	CI [EUR/kJ]	$\dot{CI} = CI \cdot \dot{I}$ [EUR/h]	$\dot{CI} + \dot{Z}$ [EUR/h]	f
Boiler	33.91	3.93	34.68	0.01117	$4.07 \cdot 10^{-5}$	0.5758	0.5869	0.01904
Expander	91.19	0.2961	2.613	0.0859	$1.527 \cdot 10^{-4}$	0.1627	0.2486	0.3455

Compressor	89.69	0.3021	2.667	0.05757	$1.753 \cdot 10^{-4}$	0.1906	0.2482	0.232
Condenser		1.033	13.66	0.01989	$1.874 \cdot 10^{-4}$	0.6967	0.7166	0.02775
MRC Evaporator	56.38	0.3109	5.418	0.009501	$3.634 \cdot 10^{-4}$	0.8031	0.8126	0.01169
ORC Pump	80.73	0.02535	0.2237	0.0053	$1.753 \cdot 10^{-4}$	0.01599	0.02129	0.2489

conditions in which the cost of the product unit becomes $c_R = 6.478 \cdot 10^{-4}$ [EUR/kJex].

Following the exergoeconomic optimization procedure (Tables 6–12), the optimal functional and constructive solution was obtained (Table 12). The unit cost of cold in this case ($c_R = 6.478 \cdot 10^{-4}$ [EUR/kJex]), decreased by half compared to the initial situation (Table 6) ($c_R = 13.15 \cdot 10^{-4}$ [EUR/kJex]).

The impact of the fuel cost of the global ORC-MCR system (Diesel fuel) on the cost of the cold unit is presented in Figure 10.

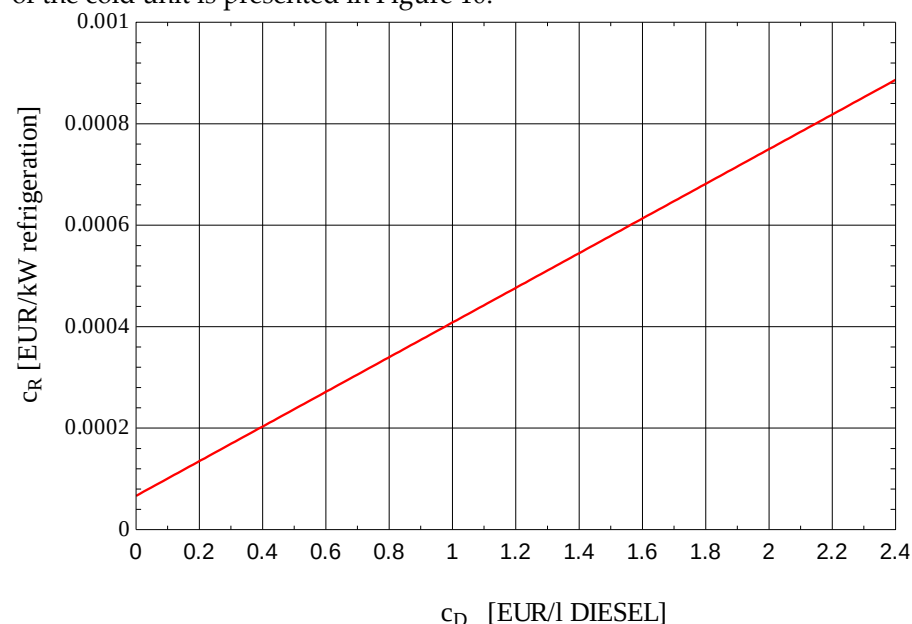


Figure 10. Influence of the Diesel fuel unitary cost on the unitary cost of the refrigeration $\Delta T_{minCd} = 3$ K; $\Delta T_{minV} = 3$ K; $\Delta T_{minB} = 10$ K; $\eta_{sCp} = 0.89$; $\eta_{sD} = 0.9$; $\eta_{sP} = 0.8$

4. Discussion

The presented study refers to the heat recovery of the combustion gases of a stationary internal combustion engine through cogeneration of mechanical power and cold.

Heat recovery was considered as a theme of the project, for which the combustion gases were considered as an unused part of the fuel used in the engine to produce mechanical power and represent the fuel for the refrigeration plant driven by the ORC system.

If the project did not require the recovery of the energy of the combustion gases, then this would be considered an accepted loss, a situation in which the combustion gases are assigned a zero cost ($c_{gMAI} = 0$).

It is interesting to note in this case, that contrary to the impression that if the cost of fuel does not matter, the optimal economic solution is obtained for the cheapest (and therefore non-performing equipment), things are not like that because the lower performance of the equipment affects the product of the system, i.e. the cold product.

This is due to the conditions of the case, namely that the potential of the recoverable energy from the combustion gases, although free, is limited.

The optimization problem is to obtain the maximum product (minimum unit cost of the product) from a limited resource with minimum monetary cost.

To find the optimal operating and design conditions, the exergoeconomic analysis and optimization method was used, the only method that looks for the optimal solution in sight, offering the user clear rules for improving the studied system, the effects of which can be followed step by step.

Unlike exergoeconomic analysis, any other optimization procedure based on statistical, evolutionary or mathematical algorithms represents black boxes for the user and the results must be accepted on trust. In addition, any mathematical optimization method performs the search within the limits of the specified scheme without providing any hints on its structural improvement.

As a proof of the power of the exergy analysis to suggest structural changes in the system, aiming to reduce the internal consumption of usable energy (reduction of exergy destruction), an exergy analysis is presented that highlights the high exergy destruction in the ORC boiler and suggests, to reduce it, using an internal heat exchanger.

But the value of exergy destruction quantified only from a thermodynamic point of view is not defining in the conditions where, in addition to the interaction with the physical environment, the interaction of the system with its economic environment must also be considered. For this purpose, in order to assign an economic cost to each exergy destruction, a strategy was followed to find the unit cost of each substance and energy stream as part of the cost of the resources purchased from outside the system and the cost of amortization of the component equipment.

The system was divided into functional areas for which the exergetic resource (generically the fuel), the product and the destruction were highlighted for each.

For each piece of equipment, the exergoeconomic correlation was built that provides the connection between the purchase cost, the size of the exergetic product and the exergetic performance parameter.

Unlike thermoeconomic correlations that calculate costs based on material consumption, exergoeconomic correlations provide an image of the sensitivity of the monetary cost of an equipment depending on its exergetic performance (that is, the exergy destruction induced due to the magnitude of the irreversibility of internal processes).

The modification of the exergetic and exergoeconomic performance coefficients to the changes made in the system guided the optimization procedure.

Due to the high cost of exergy destruction associated with the processes in the system, the exergoeconomic optimization procedure seeks to reduce them by increasing the cost of investment in more efficient performing equipment.

It can be seen that the cost of the fuel for driving the thermal engine (the fuel of the global system) has a great influence on the cost of the cold unit.

5. Conclusions

The exergoeconomic optimization procedure takes into account the interaction of the system with its physical and economic environment, looking for the functional and constructive conditions for which the unit cost of the product is minimal. The total cost of the system for a period of time is taken into account, calculated as the sum of the operating cost and the amortization cost of the invested capital.

The analysis proposes that the optimal solution is to invest in high-performance equipment that will reduce exergy consumption in the system.

The exergoeconomic analysis is the only one that offers the research engineer a methodology to search for the optimal step by step that shows the immediate effects of the functional and performance changes of the equipment on the final product. In this way, the level of understanding of the processes that take place in the system and the connections between them and the design of the equipment increases.

Author Contributions: Conceptualization, D.T., C.D. and A.D.; methodology, L.G., M.B. and V.A.; software, D.T., H.P. and A.D.; validation, V.A., C.D. and H.P.; formal analysis, M.B. and V.A.; investigation, D.T. and C.D.; resources, H.P.; data curation, M.B.; writing—original draft

preparation, D.T., C.D. and A.D.; writing—review and editing, L.G. and C.D.; visualization, D.T. and H.P.; supervision, L.G. and A.D.; project administration, V.A.; funding acquisition, C.D. All authors have read and agreed to the published version of the manuscript.

Funding: This research was funded by the Romanian Ministry of Education and National University of Science and Technology Politehnica Bucharest through the PubArt programme.

Conflicts of Interest: The authors declare no conflict of interest.

Nomenclature

B	ORC boiler
$\dot{C}$	current of monetary cost, EUR/h
c	unitary monetary cost, EUR/unit
Cd	condenser
CI	xergy destruction unit cost, EUR/kJ
$\dot{C}I$	current of exergy destruction cost, EUR/h
Cp	compressor
E	expander
$\dot{E}x$	current of exergy, kW
Ev	MRC evaporator
h	mass enthalpy, kJ/kg
$\dot{I}$	current of exergy destruction due to internal irreversibility, kW
ICE	internal combustion engine
$\dot{L}$	current of exergy loss, kW
$\dot{m}$	mass flow rate, kg/s
$\dot{Q}$	current of heat, kW
P	pump, exergetic product, kW
s	mass entropy, kJ/(kg K)
T	temperature, K
TV	throttling valve
$\dot{Z}$	Rate of amortization of the investment cost, EUR/h
$\dot{W}$	mechanical power, kW

Subscripts

0	environment, in equilibrium with the environment
B	ORC boiler
Cd	condenser
Cp	compressor
c	condensation
D	Diesel fuel
E	expander
Ev	MRC evaporator
h	heating zone of the boiler
i	inlet
L	exergy loss
m	mean temperature difference
min	minimum temperature difference
MRC	mechanical refrigeration cycle
o	outlet
oh	overheating
ORC	ORC fluid
P	pump
Q	heat
R	refrigeration

TV throttling valve

v evaporation in the MRC evaporator

vb evaporation in the ORC boiler

w water

Superscript

T thermodynamic temperature, K

Greek Symbols

Ψ share of an exergetic loss or destruction in the fuel consumption

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