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Energy and Exergy Analysis of Organic Rankine Cycle Driven by Solar and Waste Heat

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Abstract

The growth of the world population and rapid technological advancement, especially in the industrial sector, have considerably increased energy demand. Conventional power generation has caused many social and environmental problems. Two main approaches to solving these problems are to use renewable energy sources, such as solar energy, and to recover waste heat from various processes. Solar energy is a potential alternative to traditional fuels for electricity generation due to its low environmental impact and wide availability in many parts of the world. In contrast, waste-heat recovery is critical for reducing energy losses and improving energy management. One way to make use of these energy sources is the Organic Rankine Cycle (ORC). The ORC is similar to the traditional Rankine cycle, except that organic fluids are used as the working fluid. In this study, energy and exergy analyses of a solar/waste-heat-driven hybrid ORC are performed for four working fluids and four waste-heat temperature levels. Power output, thermal efficiency, and energy and exergy efficiencies are compared under different operating conditions. The results indicate that toluene provides the highest net power output, the highest first-law efficiency of the whole System, and the second-law first- and second-law efficiencies of the cycle among the considered working fluids. The performance of Cyclohexane, MDM, and n-pentane is comparable.

Keywords: Solar energy, Waste-heat recovery, Hybrid systems, Organic rankine cycles and energy and energy efficiency.

1 | Introduction

The increasing global demand for energy, the limited supply of fossil fuels, and the desire to reduce greenhouse gas emissions have made the development of high-efficiency energy conversion systems and the efficient use of renewable energy and waste heat important research areas. In internal combustion engines,

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power plants, and industry, a large part of the energy is lost as waste heat into the environment over a wide temperature range. However, solar energy, one of the most accessible renewable energy sources, has huge potential to supply the thermal energy required for power generation cycles. The integration of these two energy sources may improve the utilization of available energy and reduce the dependence on fossil fuels and associated emissions [1–4].

The Organic Rankine Cycle (ORC) is one of the best techniques for converting low- and medium-temperature heat sources into mechanical and electrical power. Organic fluids can be used as low-quality thermal resources because they boil and criticalize at lower temperatures than water [5–7]. ORC technology has been of great interest in recent years for many applications, including waste-heat recovery, biomass, solar energy, and geothermal energy [8–10]. In the study by Quoilin et al. [1], the ORC system is examined in terms of technical and financial issues, and it is shown that system performance is highly dependent on the proper selection of cycle components and thermal-source conditions. Tchanche et al. [2] have studied the use of ORCs for low-temperature heat sources and demonstrated that these cycles can convert low-grade thermal energy into useful electricity.

One of the most important aspects of the ORC design is the selection of the working fluid. Saleh et al. [11] investigated the performance of different working fluids for low-temperature ORCs. They showed that the thermodynamic properties of the working fluid play a major role in determining the operating pressures, evaporation temperature, and cycle performance. Moreover, Chen et al. [3] studied several cycles and working fluids and showed that the selection of the fluid should be compatible with the thermal characteristics of the heat source. In a review study, Bao and Zhao [6] examined the effects of expander and working fluid selection on ORC performance. They highlighted the importance of matching fluid properties to the cycle operating conditions. Papadopoulos et al. [12] proposed a systematic approach to working-fluid selection and showed that all features of the saturation curve, including the critical temperature and pressure, should be considered simultaneously. Tchanche et al. [7] examined various working fluids for a low-temperature solar ORC and demonstrated that the heat source temperature significantly affects working-fluid performance. Thus, the working-fluid selection cannot be considered separately from the thermal source characteristics.

The ORC is one of the promising techniques for generating electricity from solar thermal energy in terms of solar energy utilization. Quoilin et al. [13] have studied the performance and design of a low-cost solar ORC for power generation in remote locations and have shown that optimizing cycle components can significantly impact system performance. Later investigations focused on the use of concentrated solar collectors, particularly parabolic trough collectors, to supply the thermal energy required by ORCs. Loni et al. [14] presented a comprehensive review of the development of solar ORCs, covering issues such as working fluid selection, component design, operating conditions, and integration with solar energy systems. Arjunan [15] investigated the selection of working fluids for solar ORCs and demonstrated that performance, environmental, and thermodynamic factors must be considered simultaneously. In addition, Salem et al. [16] investigated the influence of thermal energy storage in solar ORCs. They demonstrated that thermal storage can reduce fluctuations in solar radiation and improve power-generation stability.

Conversely, waste-heat recovery via ORC is a valid method for improving the energy efficiency of internal combustion engines and industrial systems. Vaja and Gambarotta [17] studied the use of ORCs downstream of internal combustion engines, demonstrating that the heat in engine exhaust gases can be used to produce additional power. Larsen et al. [18] studied the design and optimization of ORCs for waste-heat recovery in maritime applications. They demonstrated the importance of matching the cycle operating conditions with the heat source properties. Song et al. [19] conducted a thermodynamic analysis and optimization of an ORC for the waste-heat recovery of marine diesel engines. They also studied the effects of operating parameters and working fluids on the System's performance. The study by Shu et al. [20] evaluated a Cascade-ORC system for the utilization of waste heat from diesel engines. It demonstrated that the use of heat sources at different temperature levels can be enhanced by employing multiple energy-conversion stages. Moreover, Scaccabarozzi et al. [21] studied various working fluids for waste-heat recovery from large internal combustion engines. They concluded that the choice of an optimal working fluid depends strongly on the heat source

temperature. These results highlight the importance of simultaneously considering the effects of waste-heat temperature and working-fluid type in ORC design.

Besides energy efficiency, exergy analysis is also important for evaluating ORC systems. The energy analysis cannot quantify the irreversibilities occurring in different cycle components, nor can it identify the energy quality. The calculation of exergy destruction and exergy efficiency enables one to identify the main causes of performance degradation. Nazari et al. [22] investigated the performance of a combined steam-ORC cycle for waste heat recovery through energy, exergy, and exergoeconomic analyses. The exergy analysis was shown to be a useful tool for identifying the locations of major irreversibilities within the System. The relationships between working-fluid properties and the thermodynamic and economic performance criteria of ORCs were studied theoretically and analytically by Stijepovic et al. [23]. Previous studies have shown that an increase in energy efficiency does not necessarily lead to an increase in exergy efficiency [22, 23]. Hence, both should be taken into account simultaneously in optimal system design.

Solar ORCs and waste-heat recovery ORCs have been widely studied. However, more research is needed to understand better the simultaneous use of these two thermal sources in a hybrid system. In a hybrid system, waste heat can be used as an auxiliary heat source, and solar energy as a renewable thermal source. Such a configuration can alleviate the disadvantages of relying on a single heat source and simultaneously enable more efficient use of thermal resources of varying quality. Bellos and Tzivanidis [24] investigated four different working fluids, toluene, cyclohexane, MDM, and n-pentane, for a hybrid ORC using solar and waste heat. Their results indicate that the type of working fluid strongly affects power production and system efficiency. Toluene performed well in the conditions studied. However, there remains a lack of systematic research on the effects of different waste-heat temperatures on the energy and exergy performance of various working fluids in a hybrid system.

In this respect, the present study investigates the energy and exergy performance of a hybrid ORC that simultaneously uses waste heat and solar energy. Waste-heat temperatures at four levels are investigated for four working fluids: toluene, cyclohexane, MDM, and n-pentane. The system performance is evaluated and compared in terms of net power production, energy efficiency, thermal efficiency, and exergy efficiency. The aim is to evaluate the influence of the type of working fluid and of the waste heat source temperature on the thermodynamic performances of the System and to identify the operating parameters for the optimal utilization of the two thermal sources. In this regard, the present study aims to fill the existing research gap concerning the simultaneous evaluation of the effects of waste-heat temperature level and the type of working fluid on the energy and exergy performance of a hybrid solar-waste-heat ORC.

2 | System Model and Assumptions

The present study deals with a hybrid ORC that can simultaneously exploit waste heat and solar energy. The objective is to improve waste-heat recovery and solar energy utilization by coupling two thermal sources at different temperatures. The System under investigation is schematically illustrated in the figure.

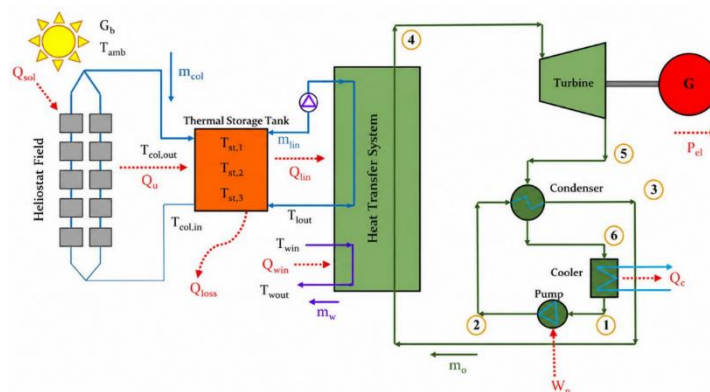


Fig. 1. Schematic diagram of the studied system.

The solar field consists of EuroTrough II parabolic-trough collectors. The heat-transfer fluid is Therminol VP-1. The thermal oil is heated by the solar energy collected by the collectors and is stored in a stratified thermal storage tank. Hot oil is drawn from the top of the storage tank by the heat recovery system and returned to the bottom of the tank after transferring energy to the organic working fluid.

Waste heat is believed to come from hot air, which is estimated to be composed of 21 % oxygen, 78 % nitrogen, and 1 % argon, under a pressure of 2 bar. This stream has an intake temperature of between 150 and 300°C. The solar thermal oil and the waste-air stream heat the organic working fluid of the heat-recovery System. The utilization of both heat sources reduces the waste stream output temperature and increases waste-energy recovery.

The working fluid, after leaving the ORC condenser, is pumped into the recuperator, where it is heated and evaporated in the economizer and evaporator. The resulting saturated vapor is then admitted to the turbine, expands, and delivers mechanical power, which is then converted to electrical power. Part of the turbine outlet stream's energy is transferred to the recuperator pump outlet stream before the stream enters the condenser. In the present work, the cycle is shown as a subcritical cycle without superheat

Four working fluids are selected to evaluate cycle performance: toluene, MDM (octamethyltrisiloxane), cyclohexane, and n-pentane. *Table 1* gives the critical pressures and temperatures of the working fluids.

Table 1. Principal properties of the working fluids.

Working Fluid	Critical Temperature (°C)	Critical Pressure (bar)	Molecular Weight (kg/kmol)
Toluene	318.6	41.26	92.14
MDM	290.9	14.15	236.5
Cyclohexane	280.5	40.75	84.16
n-Pentane	196.6	33.70	72.15

The main model parameters are: ambient temperature 25°C, condenser temperature 40°C, turbine isentropic efficiency 85%, pump efficiency 70%, electromechanical generator efficiency 98%, waste-heat-source mass flow rate 10 kg/s, solar irradiance 0.8 kW/m², and minimum heat-exchanger temperature difference 20 K. In the baseline analysis, these factors are held constant. When selecting the working fluids, their critical temperatures are considered, along with their compatibility with the temperature range of the available heat sources.

3 | Governing Equations

Mass, energy, and exergy balances for a control volume under steady-state conditions, and assuming negligible changes in kinetic and potential energies, are given by *Eqs. (1)-(3)*, respectively.

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \quad (1)$$

$$\dot{Q} - \dot{W} = \sum \dot{m}_{out} h_{out} - \sum \dot{m}_{in} h_{in} \quad (2)$$

$$\dot{E}_{heat} - \dot{W} = \sum \dot{E}_{out} - \sum \dot{E}_{in} + \dot{E}_{D; Total} \quad (3)$$

In the above equations, $\dot{m}$ is the mass flow rate of the working fluid; $\dot{Q}$ and $\dot{W}$ are the heat transfer into and the work output from the control-volume boundary, respectively; h is the enthalpy; $\dot{E}_D$ is the total exergy destruction in the components; and $\dot{E}$ is the exergy transfer or heat transfer. The subscripts in and out denote the inlet and outlet states, respectively. The relation gives the exergy transfer corresponding to heat transfer at temperature T :

$$\dot{E}_{heat} = \dot{Q} \left(1 - \frac{T_0}{T} \right) \quad (4)$$

The rate of exergy is given by

$$\dot{E}_x = \dot{m}ex. \quad (5)$$

$$ex = h - h_0 - T_0(s - s_0). \quad (6)$$

Eqs. (5) and (6), where s is the entropy. The percentage irreversibility of each component is calculated by dividing the exergy destruction of the component by the total exergy destruction:

$$IR = \frac{\dot{E}_{i,d}}{\dot{E}_{D,Total}} \times 100. \quad (7)$$

3.1| Solar Field Simulation

The solar radiation utilized by the parabolic-trough collector (Q_{sol}) is obtained by multiplying the collector aperture area (A_{col}) by the direct solar irradiance (G_b):

$$Q_{sol} = A_{col} \times G_b. \quad (8)$$

An energy balance on the thermal-oil stream is used to determine the useful heat generated by the parabolic-trough collector (Q_u):

$$Q_u = \dot{m}_{col} \times c_p (T_{col,out} - T_{col,in}). \quad (9)$$

The useful heat gain is calculated by the mass flow rate ($\dot{m}_{col}$), inlet temperature in the solar field ($T_{col,in}$), and outlet temperature from the solar field ($T_{col,out}$).

For this study, we choose the commercially available EuroTrough II parabolic-trough collector. The thermal efficiency of this collector at zero incidence angle is given by the following equation [30]:

$$\eta_{col} = 0.7408 - 0.0432 \times \left(\frac{T_{col,in} - T_{am}}{G_b} \right) - 0.000503 \times G_b \times \left(\frac{T_{col,in} - T_{am}}{G_b} \right)^2. \quad (10)$$

The useful exergy output of the collector ($\dot{E}_u$) is determined by applying the relation [26]:

$$E_u = \dot{m}_{col} \times c_p \times [(T_{col,out} - T_{col,in}) - T_{am} \times \ln(\frac{T_{col,out}}{T_{col,in}})]. \quad (11)$$

The exergy rate of solar radiation ($\dot{E}_{solar}$) is obtained from Eq. (11) [26]:

$$E_{solar} = Q_{solar} \times [1 - \frac{4}{3} \times \left(\frac{T_{am}}{T_{sun}} \right) + \frac{1}{3} \times \left(\frac{T_{am}}{T_{sun}} \right)^4]. \quad (12)$$

Thus, the exergy efficiency of the solar collector ($\eta_{ex,col}$) is calculated as [26]:

$$\eta_{ex,col} = \frac{E_u}{E_{solar}}. \quad (13)$$

The exergy efficiency of the solar field can be calculated as:

$$\eta_{ex,sol} = \frac{|Q_{lin} \times (1 - \frac{T_{am}}{T_{sol,hrs}})|}{E_{solar}}. \quad (14)$$

The present work is carried out at a zero incidence angle to highlight heat transfer in the heat-recovery System and the thermodynamic evaluation of the ORC.

3.2 | Modeling of Thermal Storage Tanks

The thermal storage tank is modeled using a thermal zone model. In this method, the storage tank is segmented into horizontal zones containing thermal oil at roughly the same temperatures, meaning each zone has a uniform temperature. Between the zones, heat and mass exchange occur, causing temperature stratification in the tank. The upper part of the storage tank contains the hotter thermal oil and the lower part contains the colder thermal oil. The present storage tank is divided into three well-mixed zones, a configuration adopted in previous studies [26], [27]. The use of well-mixed zones gives satisfactory results.

The energy balances for all mixing zones are given below as three differential equations describing the behavior of the storage tank [30]:

$$\begin{aligned} \frac{\rho V}{3} \times c_p \times \frac{\partial T_{st,1}}{\partial t} &= \dot{m}_{col} \times c_p \times (T_{col,out} - T_{am}) + \dot{m}_l \times c_p \times (T_{st,2} - T_{st,1}) \\ &\quad - U_T \times A_{st,1} \times (T_{st,1} - T_{am}). \end{aligned} \quad (15)$$

$$\begin{aligned} 3) \frac{\rho V}{3} \times c_p \times \frac{\partial T_{st,2}}{\partial t} &= \dot{m}_{col} \times c_p \times (T_{st,1} - T_{st,2}) + \dot{m}_l \times c_p \times (T_{st,3} - T_{st,2}) - U_T \times \\ &\quad A_{st,2} \times (T_{st,2} - T_{am}). \end{aligned} \quad (16)$$

$$\begin{aligned} \frac{\rho V}{3} \times c_p \times \frac{\partial T_{st,3}}{\partial t} &= \dot{m}_{col} \times c_p \times (T_{st,2} - T_{st,3}) + \dot{m}_l \times c_p \times (T_{st,out} - T_{st,3}) \\ &\quad - U_T \times A_{st,3} \times (T_{st,3} - T_{am}). \end{aligned} \quad (17)$$

The above equations take into account the density of storage fluid ρ , storage tank volume (V), storage tank heat loss coefficient (U_T), outlet area of storage tank (A_{st}), mass flow rate through collector loop $\dot{m}_{col}$, mass flow rate through oil loop $\dot{m}_l$, storage tank temperature levels (T_{st}), ambient temperature (T_{am}), solar field outlet temperature ($T_{col, out}$), and specific heat capacity (c_p).

It is assumed that the inlet temperature of the collector field ($T_{col, in}$) is equal to the temperature level in the lower part of the storage tank ($T_{st,3}$):

$$T_{colin} = T_{st3}. \quad (18)$$

Similarly, the inlet temperature of the heat-transfer oil to the recovery system, $T_{l, in}$, is assumed to be equal to the temperature of the upper region of the storage tank, $T_{st,1}$:

$$T_{lin} = T_{st1}. \quad (19)$$

The mass flow rate of the heat recovery system $\dot{m}_{hr}$, ml/s, is assumed to be equal to that of the collector field, $\dot{m}_{col}$. This assumption guarantees that the heat generated equals the heat supplied, thereby maintaining a highly stable system. The specific volume of the storage tank (V/A_{sc}) is chosen $0.125 \text{ m}^3/\text{m}^2$ as per Tzivanidis et al. [28].

The thermal loss efficiency of the storage tank [28] is defined as the relationship of the energy transferred to the fluid to the useful energy produced by the collectors:

$$\eta_{loss} = \frac{Q_{net}}{Q_u}. \quad (20)$$

The thermal losses of the storage tank can be calculated by the heat loss coefficient U_l using Eq. (21) [28]:

$$Q_{loss} = Q_u - Q_{lin} = A_{tank} \times U_l \times (T_{oil} - T_{am}). \quad (21)$$

Heat loss varies from one region to another because each region has its own temperature and external surface area. The external surface areas of various mixing regions, A_{st} , are given in Eqs. (22)-(24) [26], [27]. The storage tank is assumed to be a cylinder with diameter D_{st} and height L_{st} .

$$A_{st1} = \frac{\pi \cdot D_{st}^2}{4} + \frac{\pi \cdot D_{st} \cdot L_{st}}{3}. \quad (22)$$

$$A_{st,2} = \frac{\pi \cdot D_{st} \cdot L_{st}}{3}. \quad (23)$$

$$A_{st,3} = \frac{\pi \cdot D_{st}^2}{4} + \frac{\pi \cdot D_{st} \cdot L_{st}}{3}. \quad (24)$$

3.3 | Modeling of Heat Recovery System

In this System, the waste heat and heat from the heat-transfer oil are transferred to the organic working fluid. The heat recovery system is essentially a large heat exchanger which is split into latent and sensible sections. The general energy balance of the heat recovery system is written as:

$$Q_{in} = m_l \times C_{pl} \times (T_{lin} - T_{lout}) + m_w \times C_{pw} \times (T_{win} - T_{wout}) = m_o \times (h_4 - h_3). \quad (25)$$

The rate of exergy associated with the heat sources entering the heat recovery system is given by:

$$\dot{E}x_{HRS} = m_l(ex_4 - ex_3) + m_w(ex_4 - ex_3). \quad (26)$$

In addition, the heat-exchange efficiency η_{hex} of the waste-heat source should be specified. This parameter is defined as the ratio of the actual heat transferred to the maximum possible heat transfer when the outlet temperature of the heat source reaches the ambient temperature:

$$\eta_{hex} = \frac{T_{win} - T_{wout}}{T_{win} - T_{am}}. \quad (27)$$

3.4 | Organic Rankine Cycle Modeling

In this section, the basic equations related to the modeling of the ORC are presented. Additional information on the modeling methodology is available in Ref. [27]. These equations provide the energy and exergy balances for the various components of the cycle. The power generated by the electric generator (P_G) can be computed by the following formula [30]:

$$P_G = \eta_{mg} \times m_o(h_4 - h_5). \quad (28)$$

The net electrical power output, W_{net} , is the difference between the electrical power produced by the generator and the power used by the feed pump:

$$W_{net} = \eta_{mg} \times m_{orc} \times (h_4 - h_5) - \frac{m_{orc} \times W_{pump}}{\eta_{mot}}. \quad (29)$$

The following equations give the heat input from the heat sources to the ORC. The energy balances for the organic working fluid, heat-transfer oil, and waste-heat source are given by Eqs. (30)-(32) [28].

$$Q_{net} = m_{orc} \times (h_4 - h_3). \quad (30)$$

$$Q_{net} = m_l \times c_{pl} \times \Delta T_l. \quad (31)$$

$$Q_{\text{net}} = m_w \times cp_w \times \Delta T_w. \quad (32)$$

The turbine expansion is modeled with the isentropic efficiency (η_{is}) calculated by [28], [30]:

$$\eta_{\text{is}} = \frac{h_4 - h_5}{h_4 - h_{5\text{is}}}. \quad (33)$$

The state point 5_{is} has the same enthalpy relationship as state point 4 under isentropic expansion. Its pressure is equal to P_{low} . This low pressure equals the saturation pressure at the condensation temperature (T_c). The power consumption of the organic working-fluid pump (W_{pump}) is computed as in [30].

$$W_{\text{pump}} = \frac{m_0 \times (P_{\text{high}} - P_{\text{low}})}{\rho_0 \times \eta_{\text{pump}}}. \quad (34)$$

The present model includes the efficiencies of the electrical generator, η_{mg} , and the organic working-fluid pump, η_{pump} . The net work output W_{net} of the ORC is defined as the difference between the electrical power generated by the generator P_G and the power consumption of the pump W_{pump} :

$$W_{\text{net}} = P_G - W_{\text{pump}} \quad (35)$$

The net power is equal to the power generated minus the work done by the pump: The thermal efficiency of the ORC η_{orc} is given as [30]:

$$\eta_{\text{orc}} = \frac{W_{\text{net}}}{Q_{\text{in}}} \quad (36)$$

3.5 | Overall System

The thermal efficiency, or first-law efficiency, of the cycle can be expressed as follows [29]:

$$\eta_{\text{Icyc}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{cyc;in}}}, \quad (37)$$

where $\dot{Q}_{\text{cyc;in}}$ represents the rate of heat supplied to the cycle and is calculated as follows:

$$\dot{Q}_{\text{cyc;in}} = \dot{m}_l (h_{\text{lin}} - h_{\text{lout}}) + \dot{m}_w (h_{\text{win}} - h_{\text{wout}}) = \dot{m}_0 (h_4 - h_3). \quad (38)$$

The overall system efficiency is calculated as follows [30]:

$$\eta_{\text{sys}} = \frac{W_{\text{net}}}{Q_{\text{sol}} + m_w \times cp_w (T_{\text{win}} - T_{\text{wout}})}. \quad (39)$$

The second-law efficiency of the cycle is obtained from Eqs. (3–47) [29]:

$$\eta_{\text{II;cyc}} = \frac{\dot{W}_{\text{net}}}{\dot{E}x_{\text{cyc;in}}}, \quad (40)$$

where $\dot{E}x_{\text{cyc;in}}$ denotes the rate of exergy input to the cycle, which is calculated as follows:

$$\dot{E}_{x_{cyc;in}} = [\dot{m}_l (h_{lin} - h_{lout}) - T_0 (S_{lin} - S_{lout})] + [\dot{m}_w (h_{win} - h_{wout}) - T_0 (S_{win} - S_{wout})] = \dot{m}_0 (h_3 - h_4) - T_0 (S_3 - S_4). \quad (41)$$

4 | Results of Modeling

These are the results obtained from the energy and exergy analysis of the studied cycle. The model in this thesis was developed by Belos and Zivandis in 2018 [30] and described in the previous chapter. The accuracy of the present model has been verified by comparing selected results from the present study with those reported by Belos and Zivandis [30], as shown in *Table 2*. The results obtained in the present study are reasonably consistent with the reference study, as shown in *Table 2*, indicating the validity and reliability of the developed model. It should be noted that some differences can be seen between the two studies, as several assumptions and modeling details were not explicitly reported by Belos and Zivandis [30].

Table 2. Comparison and validation of the present results with those reported by Belos and Zivandis [30].

Working Fluid	Waste-Heat Source Temperature (°C)	Parameter	Belos and Zivandis [30]	Present Study
Toluene	150	Collector aperture area (m ²)	3443	3400
		Collector efficiency	0.7231	0.7185
		Heat-exchange efficiency	0.687	0.56
		Net power output (kW)	479.3	757.3
Cyclohexane	150	Collector aperture area (m ²)	2792	2790
		Collector efficiency	0.7231	0.7214
		Heat-exchange efficiency	0.6176	0.624
		Net power output (kW)	402	612.3
MDM	150	Collector aperture area (m ²)	1561	1561
		Collector efficiency	0.7231	0.7229
		Heat-exchange efficiency	0.4378	0.44
		Net power output (kW)	237.5	307.9
n-Pentane	150	Collector aperture area (m ²)	1810	1810
		Collector efficiency	0.7231	0.7272
		Heat-exchange efficiency	0.5843	0.60
		Net power output (kW)		

4.1 | Investigation into the Temperatures of the Waste-Heat Source at the Inlet and Outlet

Fig. 1 shows the outlet temperature of the waste-heat source from the heat recovery system. The outlet temperature of the heat recovery system increases with the inlet temperature of the waste-heat source. Toluene has a higher critical temperature than the other investigated working fluids and consequently a more subtle increase in the outlet temperature, especially at higher inlet temperatures. The outlet temperature of the heat recovery system increases only moderately for MDM and cyclohexane, which have the second- and third-highest critical temperatures, respectively. At higher temperatures, this rise tends to become almost constant. Conversely, the critical temperature of n-pentane is the lowest among the working fluids examined. Thus, the outlet temperature increases much more steeply at higher waste-heat source temperatures. As shown in *Fig. 2*, the contribution of the solar heat source to the total heat supplied to the heat recovery system decreases as the inlet temperature of the waste-heat source increases. The decrease is more pronounced at lower critical temperatures of the working fluid. This behavior can be explained by the higher-temperature waste-heat source's greater ability to meet the ORC's heat demand, thereby reducing the relative contribution required from the solar heat source.

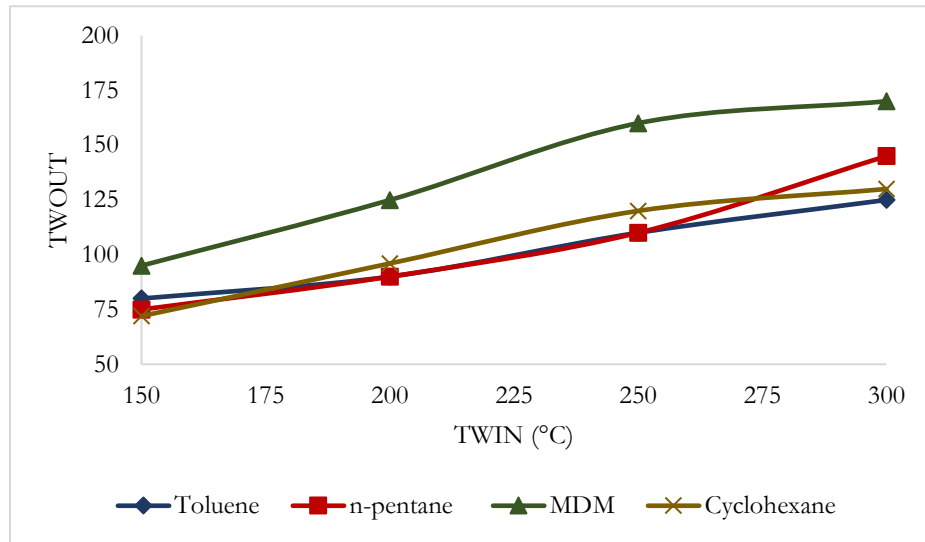


Fig. 1. Outlet temperature of the heat recovery system as a function of the inlet temperature of the waste-heat source for all investigated working fluids.

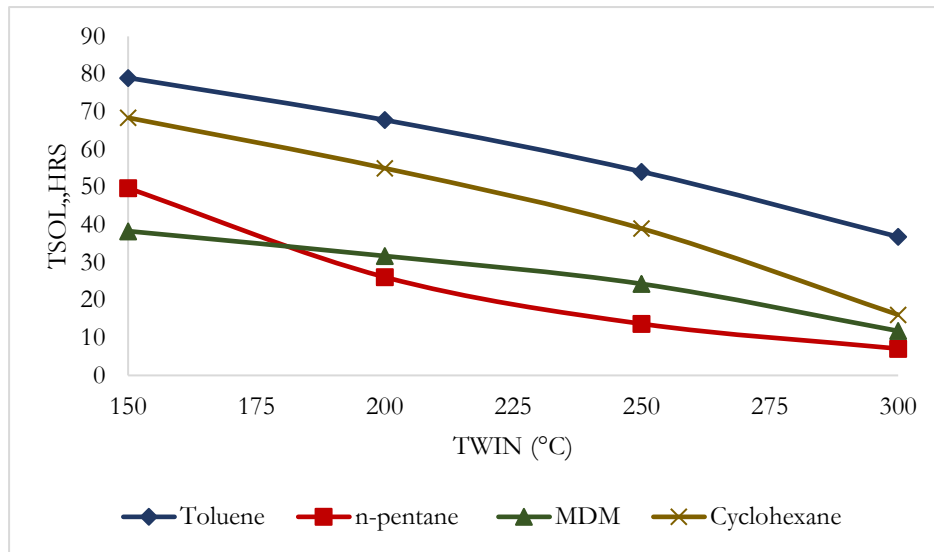


Fig. 2. Solar-section temperature of the heat recovery system as a function of the inlet temperature of the waste-heat source for all investigated working fluids.

4.2 | Investigation of Collector Area, Solar-Section Temperature, and Solar Exergy Efficiency

The effects of the collector area A_{col} on the storage-tank outlet temperature $T_{tank, out}$ and solar exergy efficiency $\eta_{s, sol}$ for the four investigated working fluids are presented in Fig. 3. In all cases, the storage-tank outlet temperature increases nearly linearly with A_{col} . This is mainly due to the increase in the area available for the collection of solar radiation, and hence the amount of thermal energy transferred to the working fluid.

In contrast, the exergy efficiency exhibits a non-linear trend and is strongly dependent on the working fluid. For toluene, the exergy efficiency initially increases, reaches a maximum at a collector area of about 2300 m², and then decreases slightly. For n-pentane, MDM, and cyclohexane, the exergy efficiency generally decreases with increasing collector area. However, the variations are rather limited once they reach a minimum value.

Thus, an increase in collector area does not necessarily improve the System's exergetic performance. Rather, each working fluid has an optimum collector-area range. This finding emphasizes the importance of simultaneously optimizing the working fluid and the solar field size in the design of hybrid solar-ORC systems.

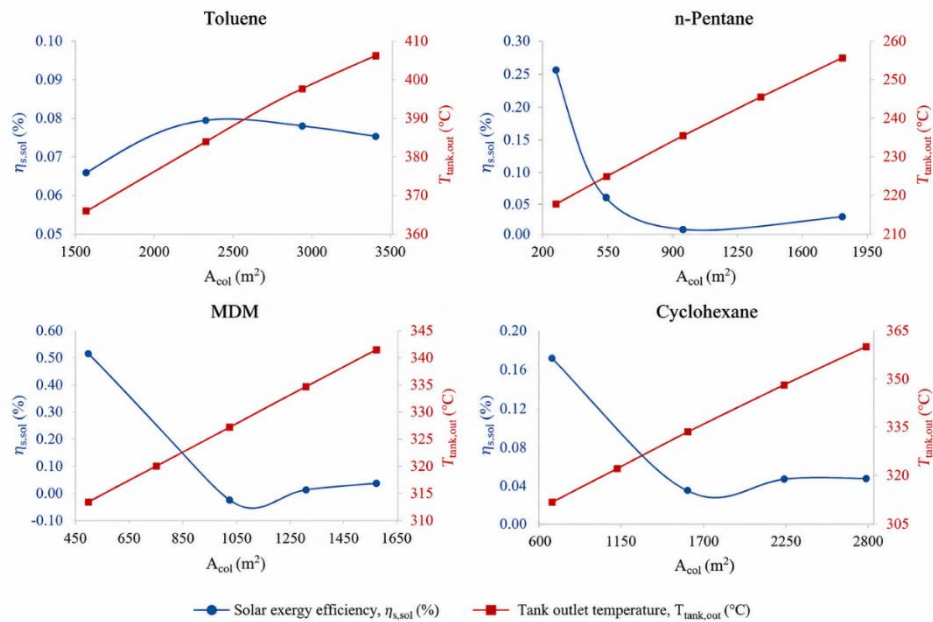


Fig. 3. Effect of solar collector area on storage tank outlet temperature and solar exergy efficiency for toluene, n-pentane, MDM and cyclohexane.

4.3 | Investigation of Collector Inlet Temperature and Collector Efficiencies

One of the important parameters affecting collector performance ($T_{col,in} - T_{am}/G_b$) is the inlet temperature. It can be seen from Fig. 4 that the collector thermal efficiency decreases distinctly and almost linearly with increasing ($T_{col,in} - T_{am}/G_b$) temperature for all the investigated working fluids, i.e., toluene, n-pentane, MDM, and cyclohexane.

As ($T_{col,in} - T_{am}/G_b$) the temperature increases, the temperature difference between the collector and the ambient environment increases. Therefore, convective and radiative thermal losses increase, and a smaller fraction of the absorbed solar energy is converted into useful thermal energy. This is responsible for the near-linear decrease in the collector's thermal efficiency.

Conversely, the collector exergy efficiency is relatively insensitive over the investigated range and remains essentially constant. Also, in some systems, a very slight upward trend is evident. This behavior suggests that variations ($T_{col,in} - T_{am}/G_b$) in have a strong influence on the amount of useful thermal energy, but not on its thermodynamic quality.

In general, the results indicate that increasing ($T_{col,in} - T_{am}/G_b$) has a significant adverse effect on the collector thermal efficiency but has a relatively small effect on the collector exergy efficiency. Therefore, these two performance indicators exhibit different sensitivities to operating conditions, underscoring the importance of considering both energy- and exergy-based criteria to assess the performance of solar collectors.

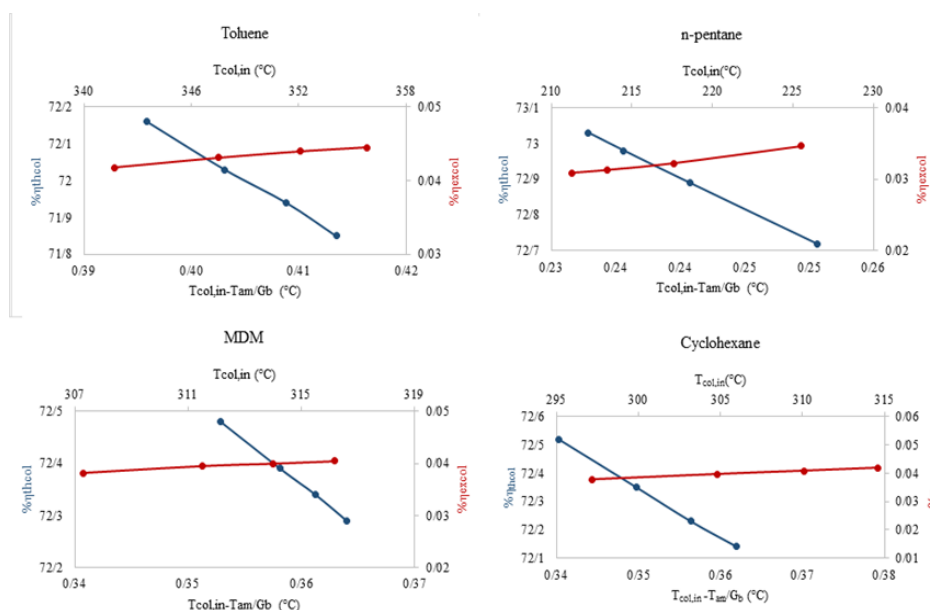


Fig. 4. Thermal and exergy efficiencies of the collector for the systems studied with toluene, n-pentane, MDM, and cyclohexane as working fluids.

4.4 | Study of Heat Source Temperature, Net Power Output and Power Consumption

Fig. 5 shows net power output as a function of the inlet temperature of the waste-heat source for all the systems and working fluids considered. The toluene-based System shows the highest net power output, exceeding 750 kW across the range of temperatures studied. This value is about 150 kW higher than that obtained with cyclohexane and about 400 kW higher than the corresponding values for MDM and n-pentane. Second is cyclohexane, with a net power output of about 536- 612 kW.

The net power output of the toluene and cyclohexane systems decreases slightly when the temperature of the waste-heat source exceeds 250 °C. In contrast, the net power output of the MDM and n-pentane systems remains almost unchanged. Since the critical temperature of cyclohexane ($\sim 280^{\circ}\text{C}$) is lower than that of toluene ($\sim 318^{\circ}\text{C}$), the reduction in the net power output is more pronounced for cyclohexane at temperatures higher than 250°C . This behavior underscores the importance of the heat-source temperature being close to the working fluid's critical temperature for the thermodynamic performance of the ORC.

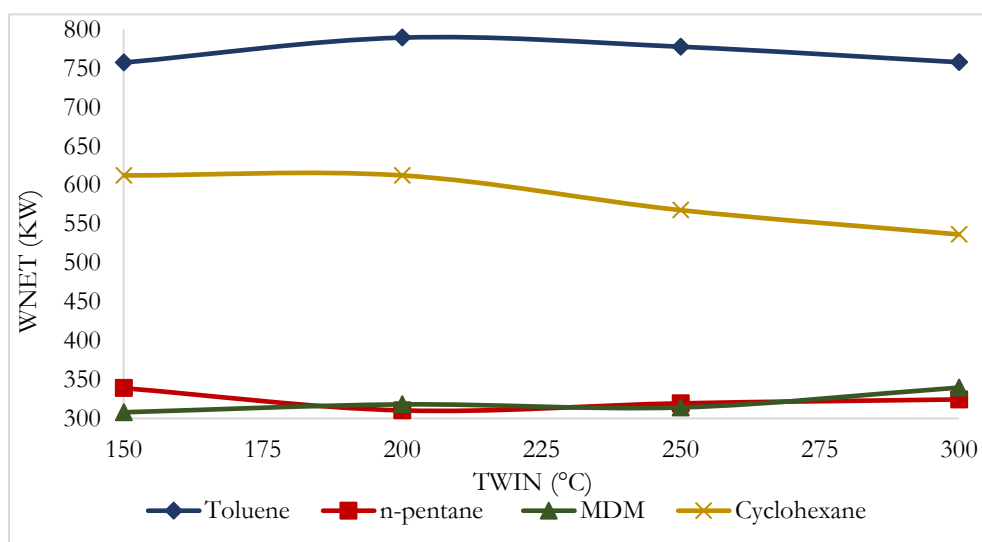


Fig. 5. Net power output vs. Inlet temperature of the waste-heat source for systems with different working fluids.

Fig. 6 shows the power consumption of the systems with different working fluids at different waste-heat source inlet temperatures. The System with MDM has the lowest power consumption. The power consumption values of the other working fluids are close to each other. The systems using toluene, MDM, and n-pentane exhibit only small changes in power consumption as the inlet temperature of the waste-heat source increases. By contrast, the power consumption of the cyclohexane-based System decreases as the inlet temperature of the waste-heat source increases. This reduction can be attributed to the changes in the thermodynamic properties of cyclohexane and the corresponding operating conditions of the cycle.

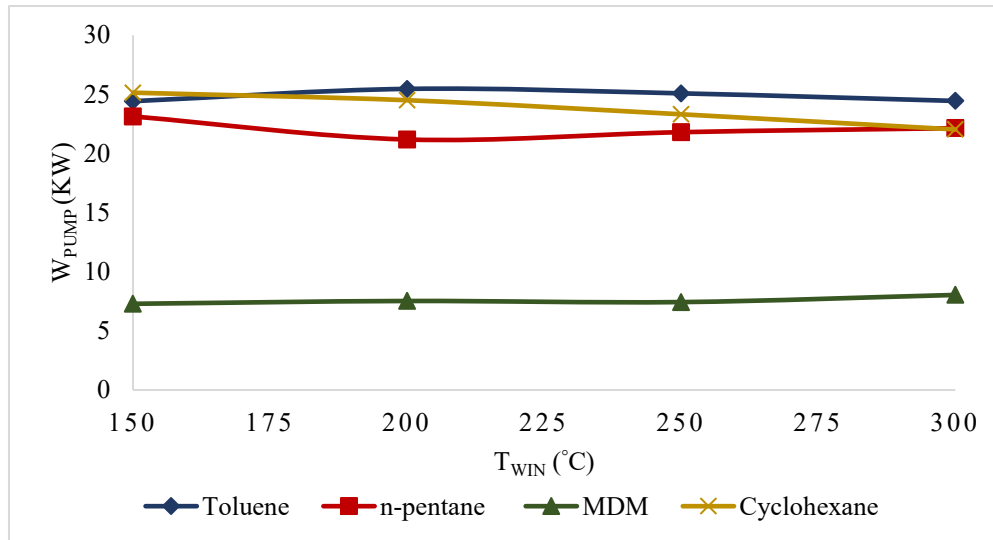


Fig. 6. Power consumption as a function of the inlet temperature of the waste-heat source for systems with different working fluids.

Table 2 presents the results for the useful heat generated by the collector, the heat losses from the storage tank, and the heat rejected by the condenser as a function of the working fluid, the collector aperture area, and the waste-heat source inlet temperature. As expected, reducing the collector aperture area reduces both the useful heat produced by the collector and the heat losses from the storage tank across all investigated systems. This behavior is mainly due to a decrease in the solar radiation collection area and, consequently, a decrease in the thermal energy supplied to the System. Thus, the collector aperture area is one of the main parameters that control the available thermal energy and the overall system performance.

Table 2. Selected simulation results of the System for different working fluids at different waste-heat source temperatures.

Working Fluid	T _{in} (°C)	A _c (m ²)	Q _u (kW)	Q _{loss} (kW)	Q _c (kW)
Toluene	150	3400	277473	399.3	1925
	200	2900	264844	391.2	2006
	250	2300	249642	381.3	1977
	300	1550	230564	368.8	1926
n-Pentane	150	1810	63307	250.1	1477
	200	940	39270	233.6	1352
	250	490	26780	224.9	1391
	300	255	20245	220.4	1413
MDM	150	1561	194210	341.8	1152
	200	1290	187267	337.1	1189
	250	305	179408	331.8	1175
	300	475	166018	322.8	1270
Cyclohexane	150	2790	209691	350.9	1796
	200	2230	195163	341.3	1750
	250	1570	177801	329.9	1665
	300	640	153219	313.6	1573

The first-law efficiency of the heat recovery system as a function of the inlet temperature of the waste-heat source for different working fluids is shown in *Fig. 7*. The efficiency of first-law heat recovery increases with temperature up to 150–200°C and then remains nearly constant up to ca. 300°C for the toluene-based System. At temperatures over 300°C, there is a little increase in efficiency.

The first-law efficiency of n-pentane increases from 150 to 200 °C and remains almost constant in the range of 200–250 °C. As the temperature of the waste-heat source is further increased, the efficiency decreases. The System based on cyclohexane shows an approximately opposite trend to that of n-pentane. However, the MDM-based System exhibits the lowest first-law heat-recovery efficiency. Efficiency drops until ~250°C and then starts to increase. Variations in thermodynamic and thermophysical properties, such as boiling temperature, vapor pressure, latent heat of vaporization, specific heat capacity, and the temperature dependence of density and viscosity, can explain the different behaviors of the working fluids. Thus, an increase in the waste-heat source temperature does not necessarily lead to a monotonic increase in heat-recovery efficiency, because the heat-transfer conditions and the temperature-level matching between the heat source and the working fluid change simultaneously.

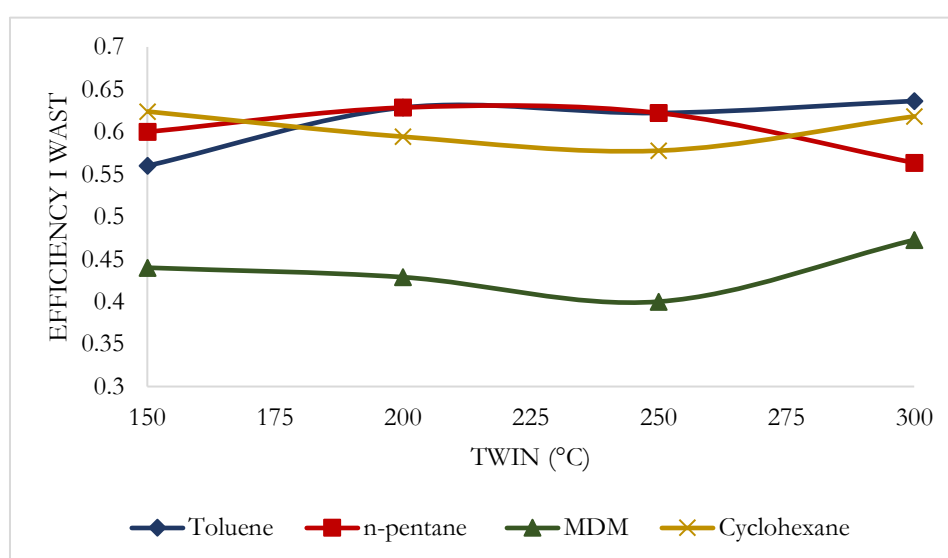


Fig. 7. First-law efficiency of the heat recovery system as a function of inlet temperature of the waste-heat source for the working fluids under investigation.

Fig. 8 shows the first-law efficiency of the collector for all the investigated systems. The collector first-law efficiency increases with increasing waste-heat source inlet temperature for all the systems, as shown. Raising the temperature of the waste-heat source increases the System's operating temperature and improves the conditions for energy transfer. This means more of the energy collected by the solar collector can be utilized as useful thermal energy. It is observed that a similar trend holds for all working fluids, indicating that the effect of waste-heat source temperature on collector performance is more dominant than the differences in the working fluids' thermophysical properties. This result highlights the strong influence of thermal operating conditions on the energy performance of the solar collection subsystem.

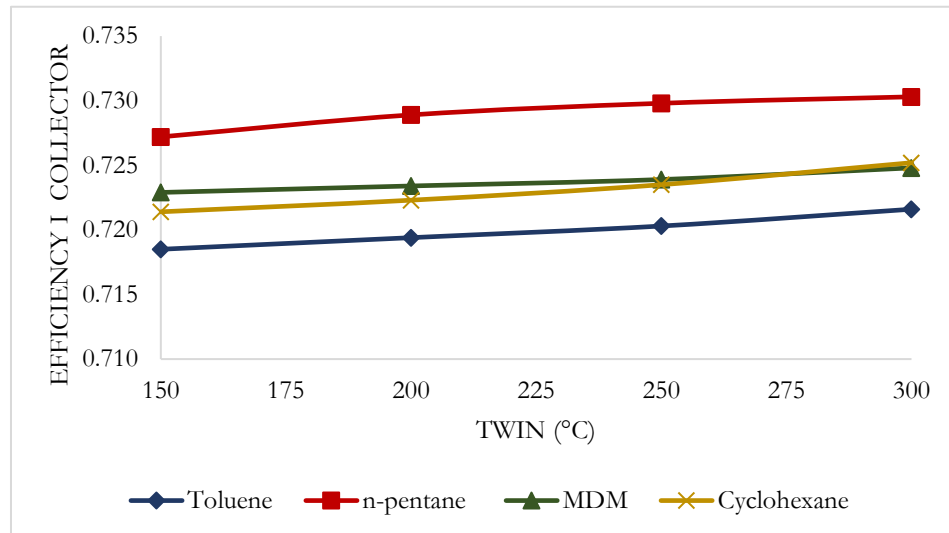


Fig. 8. First-law efficiency of the solar collector as a function of waste-heat source inlet temperature for the systems investigated.

Fig. 9 shows the first-law efficiency of the ORS for systems with various working fluids. The first-law efficiency of the ORC is nearly constant over the range of inlet temperatures studied, unlike the heat recovery and collector efficiencies. This behavior indicates that, within the considered temperature range, changes in the waste-heat source temperature do not significantly affect the ORC's energy conversion ratio. However, the thermodynamic properties of the working fluid and the cycle operating conditions have a stronger impact on the cycle performance.

Of the working fluids studied, the toluene-based System shows the highest first-law efficiency of the ORC, about 30%. Cyclohexane, MDM, and n-pentane are next in ORC efficiency to toluene. The better performance of toluene is attributed to its more favorable thermodynamic properties within the operating temperature range and its better temperature match with the available heat source. So, the higher the temperature of the waste-heat source, the more thermal energy can be recovered. However, the selection of a suitable working fluid is a more decisive factor in enhancing the energy conversion efficiency of the ORC.

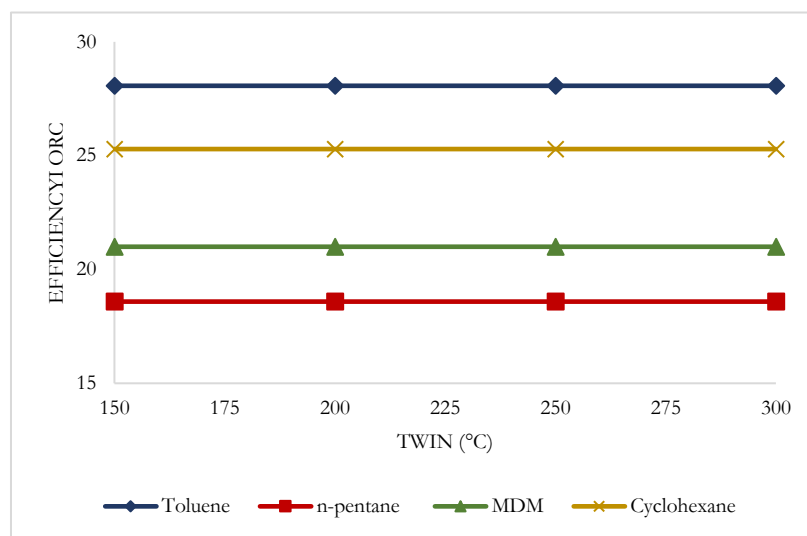


Fig. 9. First-law efficiency of the ORS versus the waste-heat source inlet temperature for the working fluids considered.

The results presented in this section demonstrate that the three main parameters, collector aperture area, waste-heat source temperature, and working-fluid type, have an impact on different subsystems of the proposed configuration. Increasing the collector area mainly increases the amount of solar thermal energy collected, while increasing the temperature of the waste-heat source generally increases the energy available for heat recovery. In contrast, the choice of the working fluid plays a more important role in the

thermodynamic performance of the ORC. Among the working fluids studied, toluene shows the best ORC first-law performance, with an efficiency of around 30% under the operating conditions studied. Hence, toluene can be considered a candidate working fluid for the investigated hybrid solar-waste-heat-driven ORC configuration. However, the optimum system design should not be based solely on ORC efficiency; collector performance, heat recovery efficiency, power consumption, net power output, and, finally, exergy performance should be considered together.

5 | Conclusion

This study used toluene, n-pentane, MDM, and cyclohexane as working fluids to examine the energy and exergy performance of a hybrid solar-waste-heat-driven ORC with thermal storage and heat recovery. The following are the primary conclusions:

- I. Although the usable solar heat and storage-tank output temperature increase with larger collector aperture area, exergy efficiency is not always improved by a larger collector area. Thus, depending on the working fluid, there is an ideal collector area.
- II. Although the working fluid and its critical temperature significantly affect heat recovery efficiency and net power output, raising the waste-heat source temperature often increases the recoverable energy available.
- III. With a net power output of about 750 kW and an ORC first-law efficiency of almost 30%, toluene was the most efficient. In terms of net power output, cyclohexane came in second.
- IV. Despite having a poorer total power-generation performance than toluene, MDM had the lowest power consumption.

The findings show that to achieve optimal system performance, the working fluid, collector area, and waste-heat source temperature must all be selected simultaneously.

Overall, toluene was shown to be the best working fluid for the hybrid solar-waste-heat ORC system under study due to its higher net power generation and ORC efficiency within the considered operating parameters.

Conflict of Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Data Availability

The data that support the findings of this study are available within the article. No additional data are associated with this publication.

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