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Performance Investigation of an Automotive Hybrid Air-Conditioning System without and with an Internal Heat Exchanger (IHX) using R1234ze(E) as Substitute for R134a

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Research Article

Abstract

In recent years, there is a growing attention drawn to the area of the technology of combined power/refrigeration cycle, due to its high capability in energy saving. In this study, tow automotive hybrid air-conditioning systems without (ORC–ACS) and with internal heat exchanger (ORC–ACS/IHX) driven by mechanical power of organic Rankine cycle (ORC) from waste heat of engine coolant are proposed to generate cooling by using R134a and R1234ze(E) as working fluids in sub-cycles (ACS and ACS/IHX).

A computer code was developed and implemented in MATLAB environment for solving engineering equations to calculate performance parameters of systems such as : coefficient of performances (COP_{ACS} and $COP_{ACS/IHX}$), compressor works input ($W_{comp(ACS)}$ and $W_{comp(ACS/IHX)}$) of sub-cycles (ACS and ACS/IHX), and overall performance (COP_{oval}) of combined cycles (ORC–ACS and ORC–ACS/IHX).

Assuming 5.0 kW cooling capacity needed, the results of current analysis highlight that (ORC–ACS/IHX) system working with (R134a and R1234ze(E)) produces an increment of the coefficient of performance and lower compressor work input compared to the (ORC–ACS) system. As a result of the study, when T_{clnt} reaches 150 °C, the $W_{comp(ACS/IHX)}$ of the (ORC–ACS/IHX) system with R1234ze(E) reduced by 7.35 % compared to the (ORC–ACS) system with R1234ze(E). Moreover, the $COP_{ACS/IHX}$ and COP_{oval} values were enhanced by 7.94 % and 7.92 %, respectively. Overall, the results confirm that (ORC–ACS/IHX) system with R1234ze(E) can be useful for automotive air conditioning applications.

Keywords: Automotive air-conditioning system; Engine coolant; IHX; Performance parameters; R1234ze(E)

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1. Introduction

In recent years, automotive air-conditioning application is rapidly growing around the world due to the increased driver comfortable environment need in the driving cabin, especially in hot weather, where automotive air-conditioning system (ACS) is now one of the leading systems used in vehicles, which supplied by the vapor compression cycle (VCC) that powered by the engine driven compressor and use refrigerant gas to cool the air in the cabin [1].Therefore, automobile manufacturers equip the majority of models with an ACS.

Automotive engines exemplify the principal driver of vapor compression cycle, which convert the mechanical energy generated from the fuels into vital air-conditioning, heating, and ventilation. Consequently, this has a considerable effect on fuel consumption [2].

The operation of an automotive air-conditioning system imposes a lot of load on the engine due to the compressor of VCC connected to it, where the increase of the input power of the compressor leads to the increase of engine speed. As a result, it increases fuel consumption of the car. Therefore, the automotive air conditioning system is the most significant auxiliary load in a vehicle [3, 4]. The dependence on automotive air-conditioning on mechanical energy generated from fuels is a biggest problem, which spurred the exploration of alternative technologies in order to save the energy and reduce power consumption during ACS operation. In this respect, thermally activated cooling technologies have gained considerable interest [5].

The thermally activated cooling can be fulfilled by sorption (adsorption and absorption), thermoelectric and (ORC–VCC) system denoted as the organic Rankine cycle (ORC) powered vapor compression cycle (VCC).

In comparison to the others, the last one exhibits reduced size and flexibility in relation to the mechanical power provided by an expander, enabling the system to perpetually harness energy [6]. These features may facilitate the acceptance of this technology in the field of automotive air conditioning systems.

Recently, many scientists have employed an (ORC) to generate cooling as an energy source for the systems, which working with the (VCC).

Li et al et al. [7] analyzed and evaluated the (ORC–VCR) system using various working fluids, revealing that the optimal refrigerant for the (ORC–VCR) system is R600, with a boiler exit temperature ranging from (60 to 90 °C), a condenser temperature between (30 and 55 °C), and an evaporation temperature spanning from (-15 to 15 °C).

Lizarte at al. [8] examined the performance of a stand-alone refrigeration system comprising a combined organic Rankine cycle and a cascade refrigeration system utilizing toluene for the organic Rankine cycle and NH3/CO2 for the cascade refrigeration system. The maximum calculated value of overall system coefficient of performance was 0.70.

Cihan, [9] studied the use of R600a, R600, R601, and R245fa as working fluids in an (ORC–VCR) system powered by a low-grade heat source. The results indicated that R601 is the most suitable fluid for the system.

Aphornratana and Sriveerakul, [10] examined an (ORC–VCR) system utilizing low-grade thermal energy. Their conclusion indicated that the system utilizing R22 exhibited a superior coefficient of performance compared to the system employing R134a.

Küçük and Kılıç, [11] investigated a hybrid (ORC–VCR) system functioning under various conditions to generate power and cooling. The analysis employs the working fluids R114, R123, R600, R600a, and R245fa in the ORC system, and R141b, R600a, R290, R134a, R123, R245fa, and R143a in the VCRC subsystem.

The results indicated the R123-R141b fluid pair yields the optimal values for energy utilization factor, exergy efficiency, system coefficient of performance, and net power.

Jeong and Kang, [12] evaluated three different refrigerants: R123, R134a and R245ca, for finding the best candidate for the (ORC–VCR) system. It is found that the R123 case gives the highest thermal efficiency.

Bu et al. [13] looked into six working fluids (R134a, R123, R245fa, R290, R600a, and R600) to find the best working fluids for an ORC–VCR system that is powered by geothermal energy. They concluded that R600a is the best option. Nevertheless, enough attention should be paid to R600a's flammability.

Wang et al. [14] studied an (ORC–VCR) system using two different working fluids for the organic Rankine cycle and conventional vapor compression cycle, namely R245fa and R134a, respectively. The overall system coefficient of performance reached nearly 0.50.

In other study, Egrican and Karakas, [15] examined an ORC–VCR system based on thermodynamics (energy and exergy) using the working fluids R114 for the Rankine cycle and R22 for the vapor compression cycle. They concluded that it is critical to have the least amount of irreversibility possible in the system to complete the task more cheaply and with a more economical use of natural resources.

Using five working fluids (R22, R141b, R236ea, R218, and R601), Bing et al. [16] analyzed and evaluated the combined (ORC–VCR) system for ship air conditioning to transfer the heat from flue gas waste and effectively use cooling water. It was determined through calculations that R601 was the best working fluid.

Zhar et al. [17] investigated the performance of (ORC–VCR) system driven from waste heat source using R123, R11, and R113 as working fluids. The simulation model of the (ORC–VCR) system is developed under EES software. Their findings indicate that the R123 is the best working fluid where the highest energy and exergy efficiencies using R123 as working fluid are over 1.02 and 0.53 respectively.

Al-Sayyab et al. [18] looked at the working fluid selection and performance of modified (ORC–VCR) system using ultra-low GWP working fluids (R1234ze(E), R1243zf, and R1234yf). The system can be configured to three operational modes, contingent upon the ground source temperature, which ranges from 55 to 90 °C: power-cooling, power-heat pump heating, and power-ground source heating. The findings demonstrate that this system significantly enhances the overall efficacy of all examined refrigerants. In comparison to traditional organic Rankine and vapor compression cycles (ORC and VCC), the R1234ze(E) power-cooling mode exhibits the most significant increase in coefficient of performance (COP), at 18 %. Besides, including a recapture heat exchanger for condenser waste heat recovery can increase power generation by 58 %. At ground source temperatures up to 65 ℃, thermal efficiency and power generation increased in the power-heating mode due to the absence of the compressor power consumption.

Maalem and Madani, [19] studied the use of five hydrocarbons (butane, isobutene, propane, propylene, and cyclopentane) as working fluids in an (ORC–VCR) system powered by a low-grade heat source. Their results showed that cyclopentane gas could be a promising working fluid in terms of performance indicators for the traditional hydrocarbons.

In the study of Wang et al. [20], authors experimented the performance of the (ORC–VCC) system with a zeotropic mixture of R245fa/R134a (0.9/0.1) at various evaporation temperatures and cooling conditions. In addition, the coupling effect of cooling water temperature and flow rate on the performance of the (ORC–VCC) system are investigated. They concluded that the cooling water temperature has a greater impact on system operational characteristics than cooling water flow rate. As the cooling water temperature, decreases and its flow rate increases, the cooling capacity of the system increases, while the coefficient of performance changes little.

Xia at al. [21] examined the performance of combined system typically includes single-fluid configuration with a common condenser and double-fluid configuration with two condensers. They concluded that the double-fluid system configuration has better thermodynamic performance than the single-fluid system configuration. In second study of Xia at al. [22], authors examined the multi-layer performance optimization of operation parameter-working fluid-heat source for the (ORC–VCR) system. Their results showed that among the candidate working fluids, HCs have the best comprehensive performance and R602 is almost the optimal working fluid under all heat resource temperature. In third study of Xia at al. [23], authors investigated a hybrid (ORC–VCR) system functioning under various conditions to generate power and cooling. The analysis employs 12 zeotropic mixtures as working fluids. The results indicated that the most suitable working fluid is R245fa/R1234yf (0.55/0.45).

On the other hand, the R134a is one of the most frequently utilized refrigerants for automotive air conditioning systems, serving as a substitute for R12 as the working fluid.

The R12 has been identified as a contributor to ozone depletion and global warming due to the presence of chlorine atoms in R12. Consequently, R12 was substituted with R134a. Nevertheless, the annual leakage rate of R134a in automotive air conditioners can attain 6 %–9 % of the total charge, significantly contributing to global warming, with a global warming potential (GWP) of 1430 [24].This will lead to the discontinuation of R134a by the year 2030.

The Paris climate convention, adopted on 4 November 2016, designates the development of sustainable refrigerants that address environmental concerns as an urgent priority. Currently, new generation fluids (hydrofluoroolefin (HFO)) which characterized by a low GWP are attracting the most attention and appear to be the most appropriate alternatives [25-26]. In this context, in response to the need of implement low-GWP fluids in the refrigeration systems, the working fluid R1234ze(E) of the new generation HFO, which has GWP=6 is proposed in the present work as substitute for R134a.

From the literature survey about the technology of (ORC–VCC) system cited above, it was noted that many research works are devoted to the performance evaluation using different type of working fluids and most of the investigations have been concentrating on the application of technology in different field of engineering. However, application of technology using new fluid such as HFO in the field of air-conditioning systems (ACS) of automobiles is not investigated in the published literature.

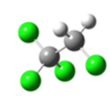
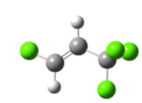
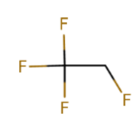
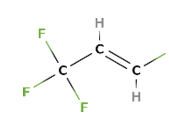
To the best of the author's knowledge, there is no work done in the area of (ORC–VCC) systems in automotive scale with the fluids of HFO such as R1234ze(E). Therefore, this study is conducted for this purpose.

In this research article, an automotive air-conditioning system without (ORC–ACS) and with internal heat exchanger (ACS/IHX) driven by mechanical power from waste heat of engine coolant is proposed in order to study and explore the potential of the both systems. In addition, a comparative examination between the phase-out R134a and the new ecofriendly R1234ze(E) in air -conditioning systems.

To address the comparison, several tests are carried out varying different parameters obtaining a wide range of operation. The following parameters are analyzed: coefficient of performances (COP_{ACS} and $COP_{ACS/IHX}$), compressor works input ($W_{comp(ACS)}$ and $W_{comp(ACS/IHX)}$) of sub-cycles (ACS and ACS/IHX), and overall performance (COP_{oval}) of combined cycles (ORC–ACS and ORC–ACS/IHX).

The thermodynamic and environmental properties of R134a, and R1234ze(E) are briefly being compared in Table 1 [27-28].

Table 1. Main properties of examined refrigerants.

Refrigerants	R134a	R1234ze(E)
Cas Nº	811-97-2	29118-24-9
Category	Hydrofluorocarbons	Hydrofluoroolefin
Chemical Name	1,1,1,2-Tetrafluoroethane	Trans-1, 3, 3, 3-Tetrafluoropropene
Molecular formula	$C_2H_2F_4$	$CHF=CHCF_3$
Molecular structure		
Chemical structure		
Molar mass (kg/kmol)	102.03	114.04
Critical temperature (K)	374.21	382.51
Critical pressure (MPa)	4.0593	3.6349
Normal boiling point (K)	247.08	254.18
ASHRAE safety group	A1	A2L
ODP	0	0
GWP	1430	6

2. Description of Air-Conditioning Systems

2.1. Technology of (ORC–ACS)

The first technology of air-conditioning system considered for the investigated is a hybrid system comprises two sub-cycles (ORC and ACS), which together form the air-conditioning system for automotive powered by waste heat of engine coolant, as illustrated in Figure 1. The technology of (ORC–ACS) is a combined organic Rankine cycle (ORC) and air conditioning system (ACS). The first sub-cycle (ORC) is identified as (1→2→3→4→1) with R600a as the working fluid. It includes four components: pump, generator, expander, and condenser [7].

The second sub-cycle (ACS) is identified as (5→6→7→8→5) with (R134a and R1234ze(E)) as the working fluids. It includes also four components: throttle valve, evaporator, compressor and condenser.

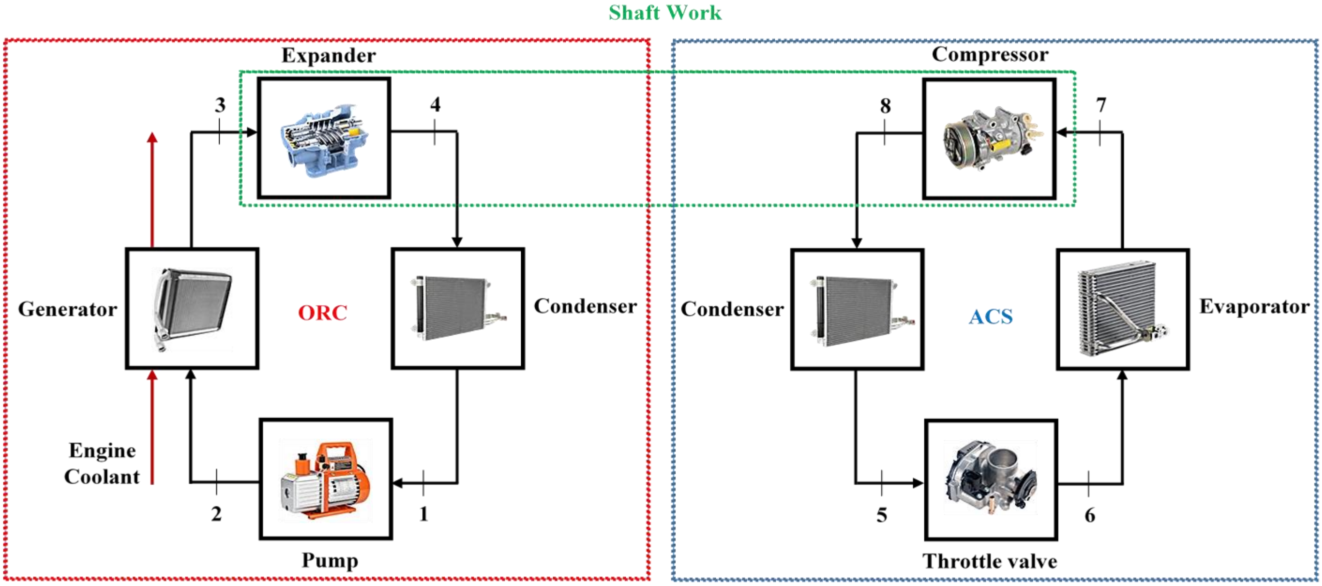


Figure 1. Schematic of the technology of (ORC–ACS)

Figure 2 presents the (T-s) diagram of the technology of the (ORC–ACS). Each sub-cycle of the technology consists of four phases.

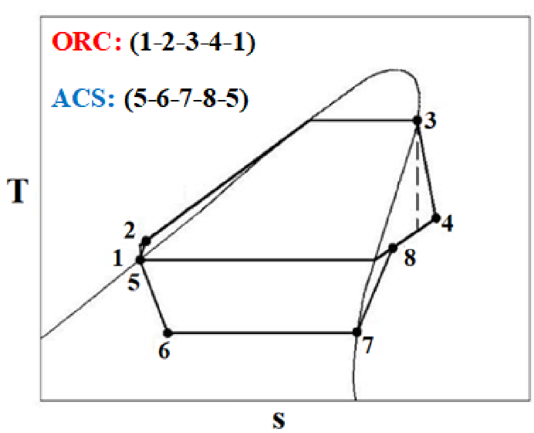


Figure 2. (T-s) diagram of the technology of (ORC–ACS)

The various processes of the first sub-cycle (ORC) are presented as follows [7]:

- (1→2): actual pumping work;
- (2→3): heat addition in the generator;
- (3→4): actual expansion in the expander;
- (4→1): heat rejection in the condenser of the ORC.

The various processes of the second sub-cycle (ACS) are presented as follows:

- (5→6): isenthalpic expansion in the throttle valve of the ACS;
- (6→7): heat absorption in the evaporator of the ACS;
- (7→8): actual compression in the compressor of ACS;
- (8→5): heat rejection in the condenser of the ACS.

In the first sub-cycle (ORC), the condensed R600a (state 1) is pressurized by the pump and subsequently enters the generator (state 2), where it is heated by the waste heat of engine coolant. The vapor of R600a generated in the generator (state 3) subsequently flows into the expander unit, which generates mechanical work to operate the pump in the first sub-cycle (ORC) and the compressor unit in the second sub-cycle (ACS). Subsequently, R600a returns to the condenser unit (state 4), concluding the ORC process.

In the second sub-cycle (ACS), the liquid (R134a or R1224ze(E)) exits the condenser unit (state 5), passes through the throttle valve, and enters the evaporator unit (state 6), where the low-pressure, low-temperature (R134a or R1224ze(E)) vaporizes and cools the air circulated inside the automotive. The vapor of (R134a or R1224ze(E)) is drawn into the compressor (state 7), where it is pressurized and subsequently expelled into the condenser (state 8) to finalize the second sub-cycle.

2.2. Technology of (ORC–ACS/IHX)

The second technology of air-conditioning system considered for the investigated is a hybrid system comprises two sub-cycles (ORC and ACS/IHX), which together form the air-conditioning system for automotive powered by waste heat of engine coolant, as illustrated in Figure 3.

The technology of (ORC–ACS/IHX) is a combined organic Rankine cycle (ORC) and air-conditioning system with an internal heat exchanger (ACS/IHX). The first sub-cycle (ORC) is identified as (1→2→3→4→1) with R600a as the working fluid. It includes four components: pump, generator, expander, and condenser [7]. The second sub-cycle (ACS/IHX) is identified as (5→6→7→8→9→10→5) with (R134a and R1234ze(E)) as the working fluids. It includes five components: throttle valve, evaporator, compressor, internal heat exchanger, and condenser.

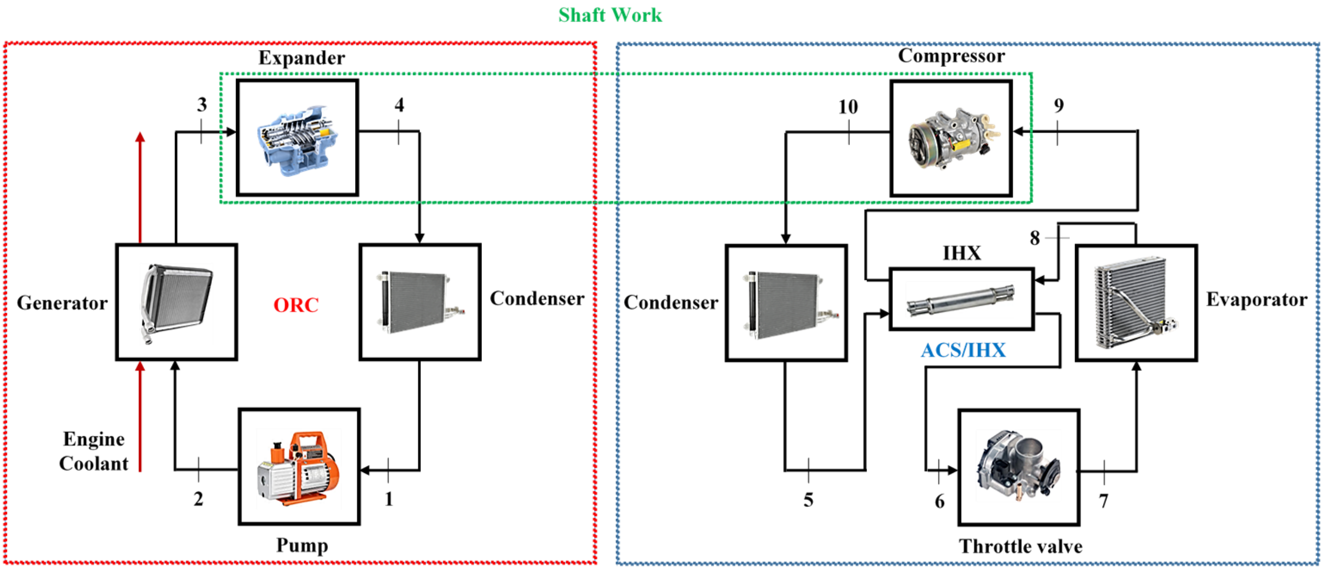


Figure 3. Schematic of the technology of (ORC–ACS/IHX)

Figure 4 presents the (T-s) diagram of the technology of the (ORC– ACS/IHX). The first sub-cycle (ORC) of the technology consists of four phases. However, the second sub-cycle (ACS/IHX) consists of six phases.

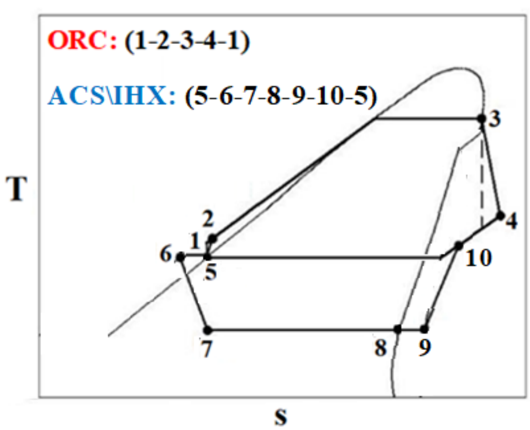


Figure 4. (T-s) diagram of the technology of (ORC–ACS/IHX)

The various processes of the first sub-cycle (ORC) are presented as follows [7]:

- (1→2): actual pumping work;
- (2→3): heat addition in the generator;
- (3→4): actual expansion in the expander;
- (4→1): heat rejection in the condenser of the ORC.

The various processes of the second sub-cycle (ACS) are presented as follows:

- (5→6): sub-cooled in the internal heat exchanger unit;
- (6→7): isenthalpic expansion in the throttle valve of the ACS/IHX;
- (7→8): heat absorption in the evaporator of the ACS/IHX;
- (8→9): superheated in the internal heat exchanger unit;
- (9→10): actual compression in the compressor of the ACS/IHX;
- (10→5): heat rejection in the condenser of the ACS/IHX.

The operating principle of the first sub-cycle (ORC) of the technology of (ORC–ACS/IHX) is identical to the first sub-cycle (ORC) of the technology of (ORC–ACS).

In the second sub-cycle (ACS/IHX), the liquid (R134a or R1224ze(E)) exits the condenser unit (state 5), get sub-cooled from state 5 to state 6 in the internal heat exchanger unit (IHX) and from there it is expanded in the throttle valve, and enters the evaporator unit (state 7), where the low-pressure, low-temperature (R134a or R1224ze(E)) vaporizes and cools the air circulated inside the automotive. Then, saturated vapor of (R134a or R1224ze(E)) from state 8 get superheated to state 9 in the internal heat exchanger unit. After this process, the superheated fluid is drawn into the compressor (state 9), where it is pressurized and subsequently expelled into the condenser (state 10) to finalize the second sub-cycle.

3. Thermodynamic Modelling

In order to analyze and investigate the combined cycles using R134a and R1234ze(E), the energy models based on the first law thermodynamics are established.

3.1. Assumptions for the analysis

For simplicity, the following assumption were made [7, 19]:

- The system operates at a steady state;
- The kinetic and potential energy changes are negligible;

- Ignore the pressure loss in the heat exchangers and the pipes;
- The isenthalpic process is considered in the throttle valve;
- Working fluid at the outlet of the generator and evaporator is saturated steam;
- Working fluid leaves the condenser as a saturated liquid;
- The isentropic efficiency of the expander, pump, and compressor are considered 0.80, 0.75, and 0.75, respectively;
- The internal heat exchanger effectiveness (ε) is taken to be 0.8 ;
- The cooling capacity is taken to be 5.0 kW.

3.2. Mathematical model

Based on these assumptions, the governing equations of the air-conditioning systems are developed as follows:

Technology of (ORC– ACS):

The power generated from the ORC expander can be calculated as [7]:

$$\dot{W}_{\exp} = \dot{m}_{ORC}\left(h_3 - h_{4s}\right)\eta_{\exp} \tag{1}$$

The power consumed by the pump can be calculated as:

$$\dot{W}_{pump} = \frac{\dot{m}_{ORC}\left(h_{2s} - h_1\right)}{\eta_{pump}} \tag{2}$$

The power consumed by the compressor can be calculated as:

$$\dot{W}_{comp} = \frac{\dot{m}_{ACS}\left(h_{8s} - h_7\right)}{\eta_{comp}} \tag{3}$$

Assuming a given cooling capacity ($\dot{Q}_{evap}$, kW), the mass flow rate in the ACS can be determined as:

$$\dot{m}_{ACS} = \frac{\dot{Q}_{evap}}{\left(h_7 - h_6\right)} \tag{4}$$

Then, assuming that the mechanical work delivered by the expander is used to drive the pump in the ORC and the compressor in the ACS, the mass flow rate in the ORC is calculated as:

$$\dot{m}_{ORC} = \frac{\dot{W}_{comp}}{\left[\left(h_3 - h_{4s}\right)\eta_{\exp} - \left(h_{2s} - h_1\right)/\eta_{pump}\right]} \tag{5}$$

The heat requirement in the generator of ORC for producing the given cooling capacity is defined as [19]:

$$\dot{Q}_{gen} = \dot{m}_{ORC}\left(h_3 - h_2\right) \tag{6}$$

The thermal efficiency (η_{ORC}) of the ORC depends on the net power output ($\dot{W}_{net}$) and heat input ($\dot{Q}_{gen}$) to the system.
Is defined as:

$$\eta_{ORC} = \frac{\dot{W}_{net}}{\dot{Q}_{gen}} = \frac{\left(\dot{W}_{\exp} - \dot{W}_{pump}\right)}{\dot{Q}_{gen}} \tag{7}$$

The COP_{ACS} of the ACS is a function of the evaporator capacity and compressor power input to the system, as follows:

$$COP_{ACS} = \frac{\dot{Q}_{evap}}{\dot{W}_{comp}} \tag{8}$$

The overall COP (COP_{oval}) of the (ORC–ACS) system can be calculated as:

$$COP_{oval} = \eta_{ORC} COP_{ACS} \tag{9}$$

Technology of (ORC– ACS/IHX):

The power consumed by the compressor can be calculated as:

$$\dot{W}_{comp} = \frac{\dot{m}_{ACS/IHX}\left(h_{10s} - h_9\right)}{\eta_{comp}} \tag{10}$$

Assuming a given cooling capacity ($\dot{Q}_{evap}$, kW), the mass flow rate in the ACS/IHX can be determined as:

$$\dot{m}_{ACS/IHX} = \frac{\dot{Q}_{evap}}{\left(h_8 - h_7\right)} \tag{11}$$

Then, assuming that the mechanical work delivered by the expander is used to drive the pump in the ORC and the compressor in the ACS/IHX, the mass flow rate in the ORC is calculated as:

$$\dot{m}_{ORC} = \frac{\dot{W}_{comp}}{\left[\left(h_3 - h_{4s}\right)\eta_{\exp} - \left(h_{2s} - h_1\right)/\eta_{pump}\right]} \tag{12}$$

The effectiveness (ε) of the IHX is given by:

$$\varepsilon = \frac{T_6 - T_5}{T_8 - T_5} \tag{13}$$

The $COP_{ACS/IHX}$ of the ACS/IHX is a function of the evaporator capacity and compressor power input to the system, as follows:

$$COP_{ACS/IHX} = \frac{\dot{Q}_{evap}}{\dot{W}_{comp}} \tag{14}$$

The overall *COP* (COP_{oval}) of the (ORC–ACS/IHX) system can be calculated as:

$$COP_{oval} = \eta_{ORC} COP_{ACS/IHX} \tag{15}$$

Based on the mathematical model built above, a computer program was developed in MATLAB and the refrigerants thermodynamic properties were obtained using REFPROP Version 9.0 to investigate the performance potential of the tow automotive hybrid air-conditioning systems without (ORC–ACS) and with internal heat exchanger (ORC–ACS/IHX) in a wide range of working conditions using the working fluids R1234ze(E) and R134a. Table 2 represents the input parameters used in the thermodynamic modeling.

Table 2. The input parameters.

Parameters	Typical value	Range
Coolant temperature, T_{clnt}	90 °C	85-105 °C
Ambient air temperature, T_{air}	35 °C	25-45 °C
Evaporation temperature, T_{evap}	5 °C	0-7 °C
Compressor isentropic efficiency, η_{comp}	75 %	-
Expander isentropic efficiency, η_{exp}	8 %	-
Pump isentropic efficiency, η_{pump}	75 %	-
The IHX effectiveness, ε	80 %	-
Cooling capacity, $\dot{Q}_{evap}$	5.0 kW	-

4. Results and Discussion

In the present study, the performances of the technologies of (ORC–ACS) system and (ORC–ACS/IHX) system are analyzed using R134a and R1234ze(E) as working fluids in sub-cycles (ACS and ACS/IHX) and R600a as working fluid in ORC.

4.1. Validation model

Prior to presenting the results of the thermodynamic analysis, it is pertinent to briefly discuss the validation of the current computational program for system simulation.

Data from Li et al [7] are utilized as a reference for comparisons owing to their analogous integrated cycle configuration.

The coefficient of performance for the (ORC–VCC) system is utilized as the comparative index under identical operating conditions (m_{ORC}=1 kg/s, evaporation temperature of 5°C, condensation temperature of 40°C, boiler temperature range of 60-90°C, η_{exp}=0.80, η_{com}=0.75, and η_{pump}=0.75) and working fluids (R290, R600, R600a, and R1270).

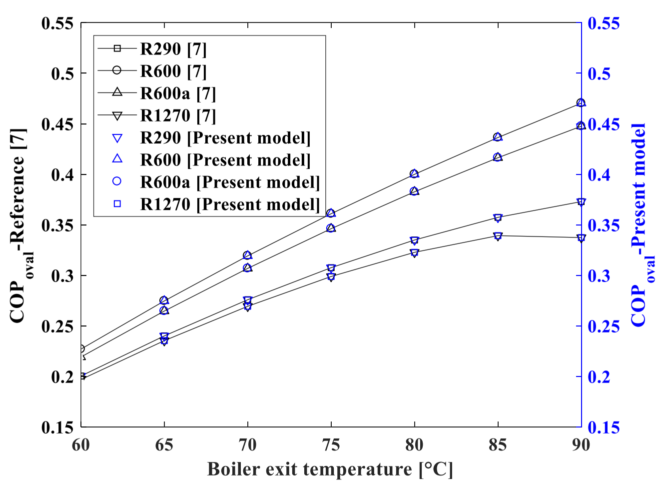


Figure 5. Modeling verification with [7]

Based on different boiler temperatures, figure 5 compares the product of the coefficient of performance for the ORC–VCC system between the current work and the reference.

The comparison results indicates a very good agreement between the results, which confirms the validity of our simulation model.

4.1. Thermodynamic performances comparison

The following part presents the simulation results of the thermodynamic performances of the proposed (ORC–ACS and ORC–ACS/IHX) systems and the effect of operating temperatures variation on the thermodynamic performances of the both systems.

Figure 6 displays the influence of the engine coolant temperature (85 to 105 °C) on the maximum thermal efficiency (η_{ORC}) of the power system (ORC) and coefficient of performances (COP_{ACS} and $COP_{ACS/IHX}$) of the both investigated air conditioning systems (ACS and ACS/IHX), respectively for automotive, where the hydrocarbon R600a (isobutene) is selected as working fluid for the ORC and the both fluids (phase-out R134a and eco-friendly R1234ze(E)) are selected as working fluids for the ACS and ACS/IHX.

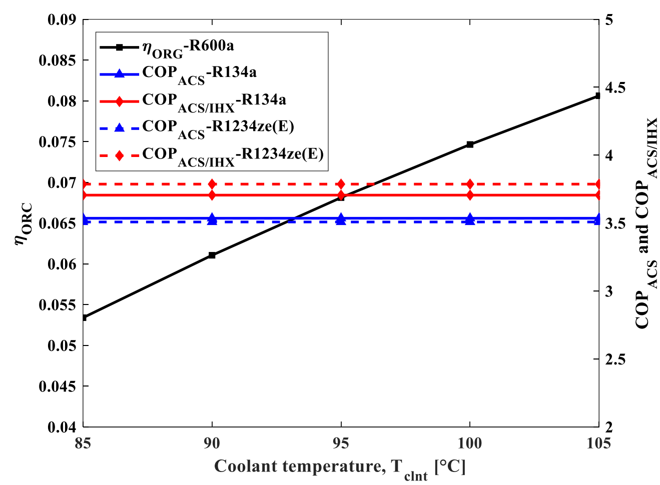


Figure 6. Variation of maximum η_{ORC}, COP_{ACS} and $COP_{ACS/IHX}$ at different coolant temperatures

From the curves of the variation of the thermal efficiency and coefficient of performances, it was noticed that as the engine coolant temperature increases, the thermal efficiency of the ORC increases from (0.0534 to 0.0806), while the coefficient of performances of the ACS and ACS/IHX keeps constant.

By analyzing the performance of both refrigerants (R134a and R1234ze(E)) in the both air conditioning systems (ACS and ACS/IHX), it was noticed that the values of coefficient of performance of the ACS working with the traditional R134a refrigerant is higher than that obtained with the eco-friendly R1234ze(E), while for the ACS/IHX, the values of coefficient of performance of the R134a is lower than the obtained with R1234ze (E).

As the engine coolant temperature increases from (85 to 105 °C), the COP_{ACS} values calculated of the both refrigerants (R134a and R1234ze(E)) in the ACS keeps constant at 3.5359 and 3.5084, respectively. However, the $COP_{ACS/IHX}$ values calculated of the same refrigerants in the ACS/IHX keeps constant at 3.7060 and 3.7870, respectively. The results indicate that at the interval of the coolant temperature [85 °C; 105 °C], the $COP_{ACS/IHX}$ of R134a in ACS/IHX increase by 4.81 % compared to the COP_{ACS} of R134a in ACS, while the $COP_{ACS/IHX}$ of R1234ze(E) in ACS/IHX increase by 7.94 % com-pared to the COP_{ACS} of R1234ze(E) in ACS.

Upon examination, it is found that the configuration of air conditioning system with internal heat exchanger (ACS/IHX) working with the both investigated refrigerants (R134a and R1234ze(E)) produces an increment of the coefficient of performance than the obtained with the configuration of air conditioning system without internal heat exchanger (ACS) for all engine coolant temperatures.

Figure 7 illustrates the effect of the engine coolant temperature (85 to 105 °C) on the compressor work input of the both air conditioning systems (ACS and ACS/IHX), respectively for automotive, where the both fluids (phase-out R134a and eco-friendly R1234ze(E)) are selected as working fluids for ACS and ACS/IHX.

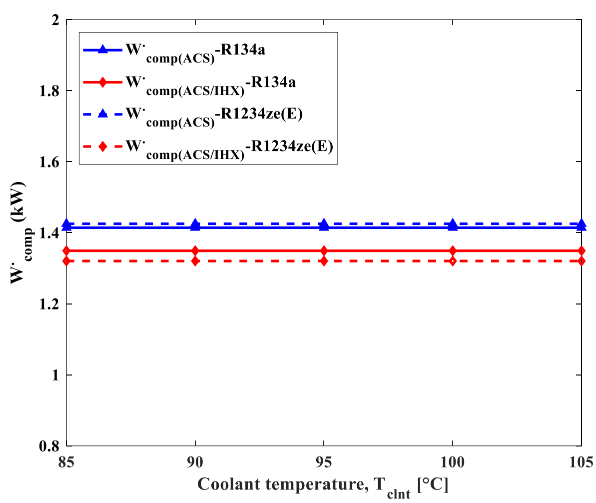


Figure 7. Variation of maximum W_{comp} of ACS and ACS/IHX at different coolant temperatures

Upon examination, it is found that like the effect of the engine coolant temperature on the coefficient of performances (COP_{ACS} and $COP_{ACS/IHX}$), similar trends were observed in the curve profiles of the variation of the compressor work input, where the input work of ACS and ACS/IHX keeps constant with the increase of engine coolant temperature. The results also show that the configuration of air conditioning system with internal heat exchanger (ACS/IHX) working with the both investigated refrigerants (R134a and R1234ze(E)) consumes less than the obtained with the configuration of air conditioning system without internal heat exchanger (ACS) for all engine coolant temperatures.

By analyzing the compressor work input among refrigerants in the both air conditioning systems (ACS and ACS/IHX), it was noticed that in the case of ACS/IHX, when ACS/IHX working with the R1234ze(E) fluid, the ACS/IHX does not consume much energy at the compressor work input compared to ACS/IHX working with the R134a fluid over the interval of the coolant temperature [85 °C; 105 °C]. However, in the case of ACS, when ACS working with the R1234ze(E) fluid, the ACS consume much energy at the compressor work input compared to ACS working with the R134a fluid over the interval of the coolant temperature [85 °C; 105 °C].

As the engine coolant temperature increases from (85 to 105 °C), the compressor work input values calculated of the both refrigerants (R134a and R1234ze(E)) in the ACS keeps constant at 1.4141 and 1.4251 kW, respectively. However, the compressor work input values calculated of the same refrigerants in the ACS/IHX keeps constant at 1.3492 and 1.3203 kW, respectively.

The results indicate that at the interval of the coolant temperature [85 °C; 105 °C], the compressor work input of R134a in ACS/IHX reduced by 4.59 % compared to the compressor work input of R134a in ACS, while the compressor work input of eco-friendly R1234ze(E) in ACS/IHX reduced by 7.35 % compared to the COP_{ACS} of R1 eco-friendly R1234ze(E) in ACS.

When the ACS/IHX working with the eco-friendly R1234ze (E) fluid, the system does not consume much energy at the compressor unit and give a good coefficient of performance ($COP_{ACS/IHX}$), which confirms that it could be a good working fluid for the ACS/IHX compared to R134a fluid.

Figure 8 shows the effect of different engine coolant temperatures on the overall performance (COP_{oval}) of both combined technologies (ORC–ACS and ORC–ACS/IHX). The COP_{oval} can be measured as an energy efficiency index of the technology of combined systems.

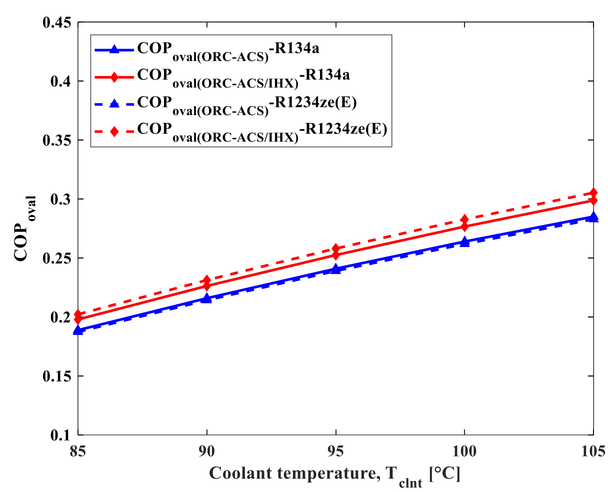


Figure 8. Variation of maximum COP_{oval} of ORC–ACS and ORC–ACS/IHX at different coolant temperatures

From the curves of the variation of the COP_{oval}, it was noticed that as the engine coolant temperature increases from 85 to

105 °C, the COP_{oval} of the systems (ORC–ACS and ORC–ACS/IHX) increases for all studied working fluids.

By comparing the simulation results obtained for all the temperature range studied, the results showed that the COP_{oval} values calculated of the both refrigerants (R134a and R1234ze(E)) in the first combined system (ORC–ACS) increases from (0.1888 to 0.2851) and from (0.1873 to 0.2829), respectively. However, the COP_{oval} values calculated of the both refrigerants (R134a and R1234ze(E)) in the second combined system (ORC–ACS/IHX) increases from (0.1979 to 0.2988) and from (0.2022 to 0.3053), respectively.

The results indicate that when engine coolant temperature reaches 105 °C, the COP_{oval} of the combined system (ORC–ACS/IHX), which use the R134a as working fluid in ACS/IHX increase by 4.81 % compared to the COP_{oval} of the combined system (ORC–ACS), which use the same working fluid in the ACS. However, the COP_{oval} of the combined system (ORC–ACS/IHX), which use the R1234ze(E) as working fluid in ACS/IHX increase by 7.92 % compared to the COP_{oval} of the combined system (ORC–ACS), which use the same working fluid in the ACS.

The effect of the ambient air temperatures (25 to 45 °C) on the maximum thermal efficiency (η_{ORC}) of the power system (ORC) and coefficient of performances (COP_{ACS} and $COP_{ACS/IHX}$) of the both air conditioning systems (ACS and ACS/IHX), respectively for automotive, where the hydrocarbon R600a is selected as working fluid for the ORC and the both fluids (phase-out R134a and eco-friendly R1234ze(E)) are selected as working fluids for ACS and ACS/IHX is indicated in Figure 9.

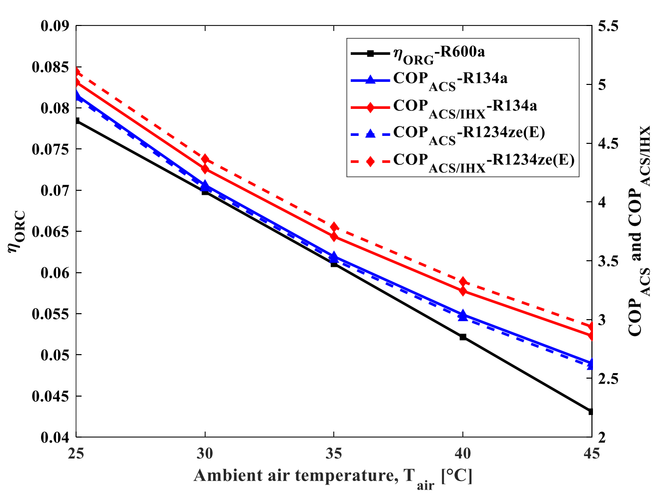


Figure 9. Variation of maximum η_{ORC}, COP_{ACS} and $COP_{ACS/IHX}$ at different ambient air temperatures

From the curves of the variation of the η_{ORC} of the ORC and the COP_{ACS} and $COP_{ACS/IHX}$ of the ACS and ACS/IHX, it is noticed that the three performance indicators (η_{ORC}, COP_{ACS} and $COP_{ACS/IHX}$) decreases with increase in ambient air temperature for all the considered working fluids.

Upon examining the graph, it can be found that, for all operating conditions, the $COP_{ACS/IHX}$ of the ACS/IHX is higher than that of the COP_{ACS} of standard system (ACS).

It is found also that, for all ambient air temperatures, the values of coefficient of performance of ACS working with the traditional R134a refrigerant is higher than that obtained with the eco-friendly R1234ze(E), while for the ACS/IHX, the values of coefficient of performance of the R134a is lower than the obtained with R1234ze(E).

As the ambient air temperature increases from (25 to 45 °C), the η_{ORC} values calculated of the R600a in the ORC decreases from (0.0784 to 0.0431) and the COP_{ACS} values calculated of the both refrigerants (phase-out R134a and eco-friendly R1234ze(E)) in the ACS decreases from (4.9105 to 2.6273) and (4.8875 to 2.5977), respectively. However, the $COP_{ACS/IHX}$ values calculated of the same refrigerants in the ACS/IHX decreases from (5.0191 to 2.8617) and (5.1071 to 2.9389), respectively.

The results indicate that at (25 °C), the $COP_{ACS/IHX}$ of R134a in ACS/IHX increase by 2.21 % compared to the COP_{ACS} of R134a in ACS, while the $COP_{ACS/IHX}$ of R1234ze(E) in ACS/IHX increase by 4.49 % compared to the COP_{ACS} of R1234ze(E) in ACS.

Figure 10 displays the influence of the ambient air temperature (25 to 45 °C) on the compressor work input of the both air conditioning systems (ACS and ACS/IHX) for automotive, where the both fluids (phase-out R134a and eco-friendly R1234ze(E)) are selected as working fluids for ACS and ACS/IHX.

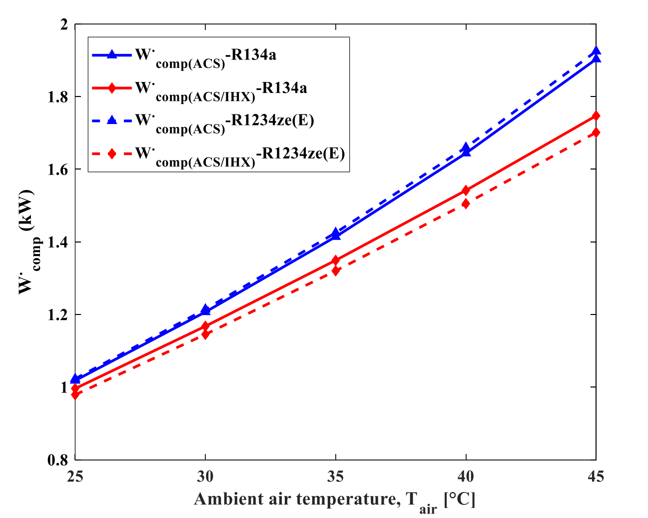


Figure 10. Variation of maximum W_{comp} of ACS and ACS/IHX at different ambient air temperatures

The results show that the compressor work input consumed by the compressor increases with the increase in the ambient air temperature of each air conditioning system.

As the ambient air temperature increases from (25 to 45 °C), the compressor work input values calculated of the both refrigerants (phase-out R134a and eco-friendly R1234ze(E)) in the ACS increases from (1.0182 to 1.9031) kW and from (1.0230 to 1.9248) kW, respectively. However, the compressor work input values calculated of the same refrigerants in the ACS/IHX increases from (0.9962 to 1.7472) kW and from (0.9790 to 1.7013) kW, respectively.

The results indicate that at (25 °C), the compressor work input of R134a in ACS/IHX reduced by 2.21 % compared to the compressor work input of R134a in ACS, while the compressor work input of eco-friendly R1234ze(E) in ACS/IHX reduced by 4.49 % compared to the COP_{ACS} of eco-friendly R1234ze(E) in ACS.

Figure 11 shows the effect of different ambient air temperatures (25 to 45 °C) on the COP_{oval} of both combined technologies (ORC–ACS and ORC–ACS/IHX).

From the curves of the variation of the COP_{oval}, it was noticed that as the ambient air temperature increases from 25 to 45 °C, the COP_{oval} of the systems (ORC–ACS and ORC–ACS/IHX) decreases for all studied working fluids.

By comparing the simulation results obtained for all the temperature range studied, the results showed that the COP_{oval} values calculated of the both refrigerants (phase-out R134a and eco-friendly R1234ze(E)) in the first combined system (ORC–ACS) decreases from (0.3852 to 0.1132) and from (0.3834 to 0.1119), respectively. However, the COP_{oval} values calculated of the both refrigerants (phase-out R134a and eco-friendly R1234ze(E)) in the second combined system (ORC–ACS/IHX) decreases from (0.3937 to 0.1233) and from (0.4006 to 0.1266), respectively.

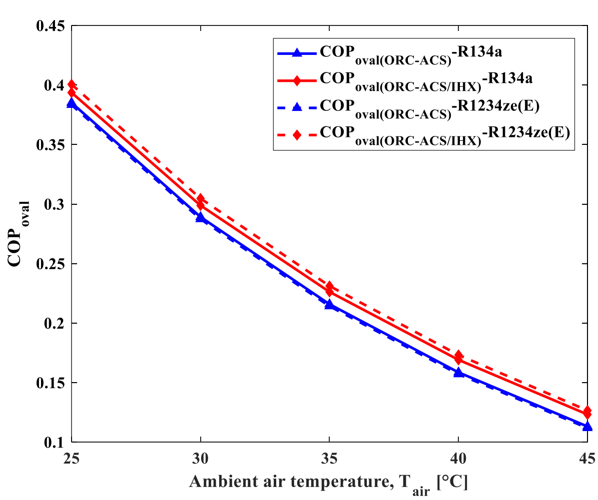


Figure 11. Variation of maximum COP_{oval} of ORC–ACS and ORC–ACS/IHX at different ambient air temperatures

The results indicate that when engine coolant temperature reaches 25 °C, the COP_{oval} of the combined system (ORC–ACS/IHX), which use the R134a as working fluid in ACS/IHX increase by 2.21 % compared to the COP_{oval} of the combined system (ORC–ACS), which use the same working fluid in the ACS. However, the COP_{oval} of the combined system (ORC–ACS/IHX), which use the eco-friendly R1234ze(E) as working fluid in ACS/IHX increase by 4.49 % compared to the COP_{oval} of the combined system (ORC–ACS), which use the same working fluid in the ACS.

Figure 12 presents the variation of the maximum thermal efficiency (η_{ORC}) of the power system (ORC) using R600a and coefficient of performances (COP_{ACS} and $COP_{ACS/IHX}$) of the both air conditioning systems (ACS and ACS/IHX), respectively for automotive by using HFC fluid (R134a) and HFO fluid (R1234ze(E)) at different evaporator temperatures.

From the curves of the variation of the thermal efficiency and coefficient of performances, it was noticed that as the evaporation temperature increases, the thermal efficiency of the ORC keeps constant at 0.0611, while the coefficient of performances of ACS and ACS/IHX increases over the interval [0 °C; 7 °C] for all fluids studied.

As the evaporator temperature increases from (0 to 7 °C), the COP_{ACS} values calculated of the both refrigerants (phase-out R134a and eco-friendly R1234ze(E)) in the ACS increases from (3.0430 to 3.7660) and (3.0089 to 3.7416), respectively. However, the $COP_{ACS/IHX}$ values calculated of the same refrigerants in the ACS/IHX increases from (3.2199 to 3.9329) and (3.2919 to 4.0181), respectively.

An important discovery on the results showed that the configuration of air conditioning system with internal heat exchanger (ACS/IHX) gives good coefficient of performance compared to the configuration of air conditioning system without internal heat exchanger (ACS) for all evaporation temperatures.

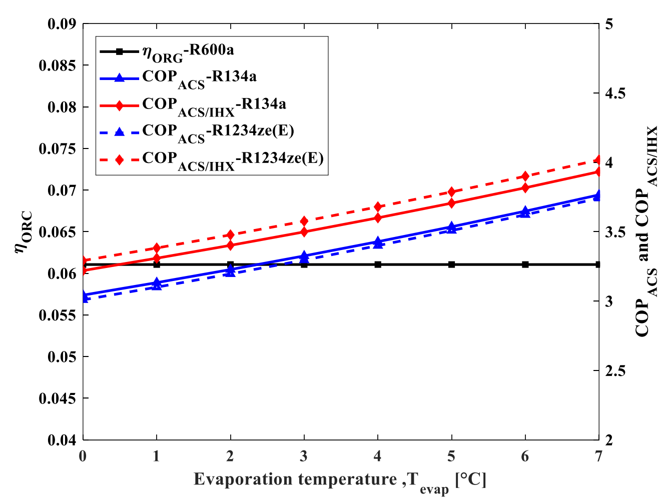


Figure 12. Variation of maximum η_{ORC}, COP_{ACS} and $COP_{ACS/IHX}$ at different evaporation temperatures

The results indicate that when evaporator temperature reaches 7 °C, the $COP_{ACS/IHX}$ of phase-out R134a in ACS/IHX increase by 4.43 % compared to the COP_{ACS} of R134a in ACS, while the $COP_{ACS/IHX}$ of eco-friendly R1234ze(E) in ACS/IHX increase by 7.39 % compared to the COP_{ACS} of eco-friendly R1234ze(E) in ACS.

The effect of the evaporation temperatures (0 to 7 °C) on the compressor work input of the both air conditioning systems (ACS and ACS/IHX) for automotive, where the both fluids (phase-out R134a and eco-friendly R1234ze(E)) are selected as working fluids for ACS and ACS/IHX is indicated in Figure 13.

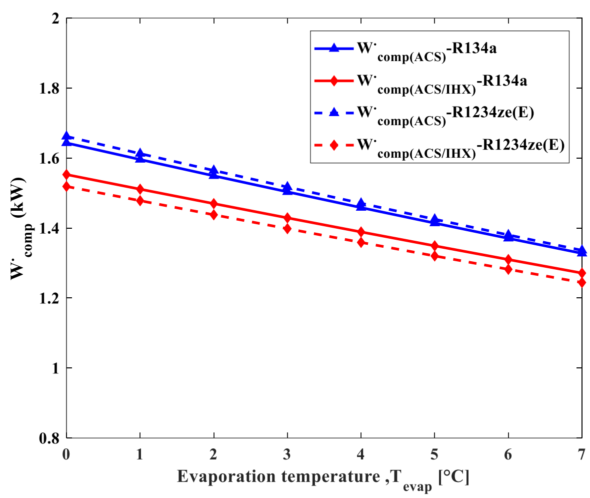


Figure 13. Variation of maximum W_{comp} of ACS and ACS/IHX at different evaporation temperatures

The results show that the compressor work input consumed by the compressor decreases with the increase in the evaporation temperature of each air conditioning system. An important discovery on the results showed that ACS/IHX consumes less compressor work input compared to ACS.

As the evaporation temperature increases from (0 to 7 °C), the compressor work input values calculated of the both refrigerants (phase-out R134a and eco-friendly R1234ze(E)) in the ACS decreases from (1.6431 to 1.3277) kW and from (1.6617 to 1.3363) kW, respectively. However, the compressor work input values calculated of the same refrigerants in the ACS/IHX decreases from (1.5528 to 1.2713) kW and from (1.5189 to 1.2444) kW, respectively.

The results indicate that at (7 °C), the compressor work input of phase-out R134a in ACS/IHX reduced by 4.44 % compared to the compressor work input of phase-out R134a in ACS, while the compressor work input of eco-friendly R1234ze(E) in ACS/IHX reduced by 7.39 % compared to the COP_{ACS} of eco-friendly R1234ze(E) in ACS.

Figure 14 presents the effect of different evaporation temperatures on the COP_{oval} of both combined technologies (ORC–ACS and ORC–ACS/IHX).

From the curves of the variation of the COP_{oval}, it was noticed that as the evaporator temperature increases from (0 to 7 °C), the COP_{oval} of the systems (ORC–ACS and ORC–ACS/IHX) increases for all studied working fluids.

By comparing the simulation results obtained for all the evaporation temperature range [0 °C; 7 °C] studied, the results showed that the COP_{oval} values calculated of the both refrigerants (phase-out R134a and eco-friendly R1234ze(E)) in the first combined system (ORC–ACS) increases from (0.1858 to 0.2299) and from (0.1837 to 0.2285), respectively. However, the COP_{oval} values calculated of the both refrigerants (phase-out R134a and eco-friendly R1234ze(E)) in the second combined system (ORC–ACS/IHX) increases from (0.1966 to 0.2401) and from (0.2010 to 0.2453), respectively.

The results indicate that when evaporator temperature reaches 7 °C, the COP_{oval} of the second combined system (ORC–ACS/IHX), which use the phase-out R134a as working fluid in ACS/IHX increase by 4.44 % compared to the COP_{oval} of the first combined system (ORC–ACS), which use the same working fluid in the ACS. However, the COP_{oval} of the second combined system (ORC–ACS/IHX), which use the eco-friendly R1234ze(E) as working fluid in ACS/IHX increase by 7.35 % compared to the COP_{oval} of the first combined system (ORC–ACS), which use the same working fluid in the ACS.

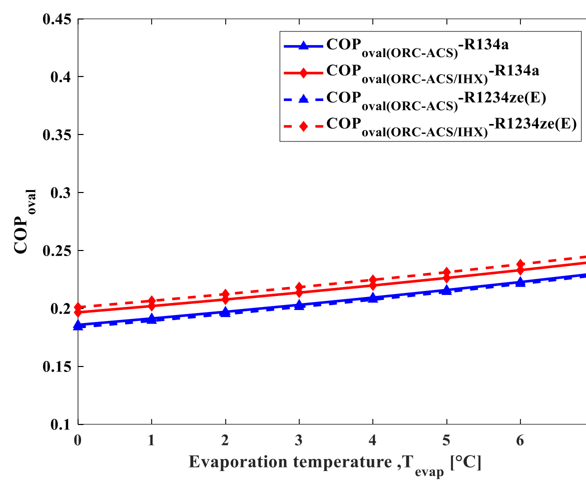


Figure 14. Variation of maximum COP_{oval} of ORC–ACS and ORC–ACS/IHX at different evaporation temperatures

5. Conclusion

In this research, tow automotive hybrid air-conditioning systems without (ORC–ACS) and with internal heat exchanger (ORC–ACS/IHX) consisting of sub-cooling cycles (ACS and ACS/IHX) and power cycle (ORC) are presented. The both systems operate with the waste heat of the engine coolant, which delivered to the (ORC–ACS and ORC–ACS/IHX) to produce refrigeration.

The performance parameters investigation was performed of the tow automotive hybrid air-conditioning systems operating with phase-out R134a and eco-friendly R1234ze(E) working fluids, where a computer code was developed and implemented in MATLAB environment for solving engineering equations to calculate performance parameters of investigated systems such as : coefficient of performances (COP_{ACS} and $COP_{ACS/IHX}$), compressor works input ($W_{comp(ACS)}$ and $W_{comp(ACS/IHX)}$) of sub-cycles (ACS and ACS/IHX), and overall performance (COP_{oval}) of combined cycles (ORC–ACS and ORC–ACS/IHX).

The comparison of the performance parameters of both investigated working fluids (phase-out R134a and eco-friendly R1234ze(E)) in the (ORC–ACS) system and (ORC–ACS/IHX) system was carried out under the same air conditioning operating conditions for coolant temperature selected at (85 to 105 °C), ambient air temperature selected at (25 to 45 °C), and evaporation temperatures ranged between (0 to 7 °C).

Below, the most valuable outcomes are summarized:

- The variation in the coolant temperature has a significant impact on the thermal efficiency (η_{ORC}) of the ORC;
- As the coolant temperature rises, the thermal efficiency of the ORC go up, however, as the ambient air temperature rises, the thermal efficiency of the ORC go down;
- The coefficient of performances (COP_{ACS} and $COP_{ACS/IHX}$) and compressor works input ($W_{comp(ACS)}$ and $W_{comp(ACS/IHX)}$) of the sub-cycles (ACS and ACS/IHX) studied is unaffected by variations in the coolant temperature;
- The COP_{ACS} and $COP_{ACS/IHX}$ of the sub-cycles (ACS and ACS/IHX) studied under study rises as the evaporation temperature rises and falls as the ambient air temperature falls;
- The $W_{comp(ACS)}$ and $W_{comp(ACS/IHX)}$ of the sub-cycles (ACS and ACS/IHX) studied under study rises as the ambient air temperature rises and falls as the evaporation temperature falls;
- At the same operating temperatures, the $W_{comp(ACS/IHX)}$ of the ACS/IHX is lower than that of the $W_{comp(ACS)}$ of the ACS;
- At the same operating temperatures, the $COP_{ACS/IHX}$ of the ACS/IHX is higher than that of the COP_{ACS} of the ACS;
- The variation in the coolant and evaporation temperature have a significant impact on the overall coefficient of performance of the investigated combined cycles (ORC–ACS and ORC–ACS/IHX);
- For the both investigated combined cycles (ORC–ACS and ORC–ACS/IHX), the overall coefficient of performance increase with the increase of the coolant and evaporation temperatures and decreases with the ambient air temperature;
- At the same operating temperatures, the overall coefficient of performance of the (ORC–ACS/IHX) system is higher than that of the (ORC–ACS) system;
- The employing of the internal heat exchanger (IHX) unit is favourable in the ACS.

Conflict of Interest Statement

The authors declare that there is no conflict of interest in the study.

CRediT Author Statement

Youcef Maalem: Conceptualization, Writing-original draft,
Hakim Madani: Conceptualization, Validation, Supervision

Nomenclature

Symbols

COP_{ACS}	: coefficient of performance of ACS
$COP_{ACS/IHX}$	: coefficient of performance of ACS/IHX
COP_{oval}	: overall coefficient of performance
h	: specific enthalpy (kJ kg^{-1})
$\dot{m}$	: mass flow rate (kg s^{-1})
T	: temperature (°C)
$\dot{Q}_{gen}$	: generator heat input (kW)
$\dot{Q}_{evap}$	: evaporator cooling capacity (kW)
$\dot{W}_{comp}$	: compressor work input (kW)
$\dot{W}_{\exp}$	: expander work output (kW)
$\dot{W}_{net}$	: net work output (kW)
$\dot{W}_{pump}$	: pump power consumption (kW)

Greek symbols

ε	: internal heat exchanger effectiveness
η_{comp}	: isentropic efficiency of compressor
$\eta_{\exp}$	: isentropic efficiency of expander
η_{pump}	: isentropic efficiency of pump
η_{ORC}	: power cycle thermal efficiency

Subscripts

evap	: evaporator
clnt	: coolant
air	: ambient air
s	: isentropic process
ORC	: organic Rankine cycle
ACS	: air-conditioning system without IHX
ACS / IHX	: air-conditioning system with IHX
IHX	: internal heat exchanger
1,...,10	: numbering of states in systems

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