AUT Journal of Mechanical Engineering

AUT J. Mech. Eng., 2(2) (2018) 253-262
DOI: 10.22060/ajme.2018.13885.5694

Thermodynamic Analysis and Feasibility Study of Internal Combustion Engine Waste Heat Recovery to Run its Refrigeration System

T. Ghorbani, M. Yari, F. Mohammadkhani*

Department of Mechanical Engineering, Mechanical Engineering Faculty, University of Tabriz, Tabriz, Iran

ABSTRACT: Automobiles refrigeration systems are mainly vapor compression refrigeration systems, and they use high power which is taken directly from the engine. The use of these systems will increase fuel consumption, and this fuel consumption will increase up to 15%. By considering the importance of fuel saving, optimum use of fuel will be necessary. One of the effective ways, is the waste heat recovery from the engine exhaust gas. The purpose of this study is the thermodynamic analysis of a new cogeneration system based on internal combustion engine. In fact, the system will generate power using heat recovery from exhaust the engine, and then the power will be used to run the refrigeration system. The system is used in the actual operating modes of gasoline and diesel engines. Different refrigerants are used in the system. Results show that the system can generate required refrigeration capacities of both automobiles and buses. Furthermore, additional refrigeration capacities will also be available. R245fa and R600 refrigerants have better performances in the system. Maximum refrigeration capacity generated by the system is 20 kW when using gasoline engine exhaust gases waste heat recovery, and 130 kW when using diesel engine exhaust gases waste heat recovery.

Review History:

Received: 31 December 2017
Revised: 11 April 2018
Accepted: 29 April 2018
Available Online: 20 May 2018

Keywords:

Engine waste heat recovery
Exhaust gas
Exergy analysis
Refrigeration capacity
Coefficient of performance

1- Introduction

In internal combustion engines, a large portion of the fuel energy is wasted by the exhaust gases. Energy balance of fuel combustion in an automobile engine indicates that the portion of the fuel energy which is converted to the useful work is about on third, while the fuel remaining portion is wasted mainly through the radiator as well as the exhaust system [1]. Air condition systems in automobiles and buses are necessary for comfort of driver and passengers. Appropriate air condition system increases thermal comfort and improves drivers' performance as well as safety at high ambient temperatures [2].

Application of absorption refrigeration systems may not be appropriate in automobiles because they operate in two phases .The working conditions of these systems increase the need for maintenance and therefore, these systems will not be suitable for cars [3], or they may have low Coefficient of Performance (COP) [1].

Other refrigeration systems are organic Rankine vapor compression refrigeration systems and ejector refrigeration systems. To achieve higher coefficient of performances in these systems, high temperature heat sources are needed. Selecting appropriate refrigerants in these systems are important [3].

Automobiles air conditioning systems are mainly vapor compression refrigeration systems and they consume high axial power which is directly taken from engine. The use of these systems will increase fuel consumption. For both economic and environmental purposes, optimal use of fuel is necessary. One of the effective ways that can be suggested, is the engine exhaust gases waste heat recovery.

Corresponding author, E-mail: f.mohammadkhani@tabrizu.ac.ir

Little and garimella [4] investigated the various ways of generating refrigeration capacity by using engine exhaust gases waste heat recovery. Wang et al. [5] examined a vapor compression refrigeration system with R245fa refrigerant, in which refrigeration and power cycles work independently. Wang et al. [6] studied the effects of different parameters on the coefficient performance of a refrigeration system with R245fa refrigerant. Jeong and Kang [7] studied an organic Rankine vapor compression refrigeration system, in which the needed power of the compressor is supplied by the Rankine cycle turbine. The results indicated that R245fa refrigerant had the best performance in system. Li et al. [8] investigated the effects of different refrigerants on the organic Rankine vapor compression refrigeration system and determined coefficient of performances on the basis of temperature changes in the heat exchangers. Wang et al. [9] reviewed application of the Rankine cycles to utilizing the engine exhaust gas. Yu et al. [10] utilized organic Rankine cycle for recovering the waste heat from diesel engine exhaust gas, and stated that by changing evaporating pressure, thermal efficiency of system can be increased up to 5.8%. Salek et al. [11] investigated on diesel engine exhaust gases waste heat recovery. They analyzed Rankine cycle and ammonia absorption refrigeration cycle which were coupled to the diesel engine. The results show the recovery of 10% of the engine power. Velez et al. [12] studied utilizing the organic Rankine cycle for producing power from the low temperature heat sources examining various working fluids. Tchanche et al. [13] reviewed various applications of organic Rankine cycles and assumed heat sources with temperatures lower than 230 ºC as low temperature heat sources. Shu et al. [14] stated that decomposition temperature of organic fluids in low temperature heat sources are about 200 - 300 ºC. Daviran

et al. [15] performed a comparative study for an automotive air conditioning system considering R1234yf and R134a as working fluids.
One of the effective ways to prevent organic fluids from decomposition in high temperature heat sources like engine exhaust gas is transferring the exhaust gas energy to the working fluid using a thermal oil circuit [16, 17].
Alper Yilmaz [3] investigated the waste heat recovery from a diesel engine exhaust gas using an organic Rankine vapor compression refrigeration system. But the amount power required to run the pump was not evaluated. For instance about 1.61 kW mechanical power is required to run the pump with R134a refrigerant to create 30 kW cooling load.

In this paper, a new Transcritical Organic Rankine Vapor Compression (T-ORVC) refrigeration system will be investigated. In this system, turbine will provide the needed power of both pump and compressor. In this case, the system only needs heat energy of exhaust gases to create refrigeration effect. This is the system advantage, and with this feature the system can be compared with absorption refrigeration systems, as it only needs heat energy.
Several types of refrigerants have been used. Some researchers used hydrocarbons as refrigerants .Bayrakci and Ozgur [18] used R290, R600, R1270 hydrocarbons as refrigerants in a vapor compression refrigeration system. Investigations of Ahamed et al. [19] and Chang et al. [20] can be used as examples for the use of hydrocarbons in the vapor compression refrigeration systems. Suggested refrigerants for use in the system are R134a, R245fa, R290, R600, R1234yf and R1270.
To study the feasibility of using this system in automobiles and buses, the generated refrigeration capacities by the system will be compared with the refrigeration capacities generated by the conventional refrigeration systems of these vehicles. SD6V12 compressor can be used in refrigeration system of gasoline passenger automobiles and FK40-560 compressor can be used in busses refrigeration systems. Both of the compressors have their own performance table, in which their refrigeration capacities are illustrated.
The advantage of this system compared to the conventional refrigeration systems in vehicles is that the system will provide its required power from the engine exhaust gases waste heat recovery which cause fuel consumption reduction. The system can generate the required refrigeration capacities of vehicles, and from the thermodynamic viewpoint, it can be substituted with the conventional refrigeration systems of vehicles.

2- Refrigeration System

Little and Garimella [4] investigated waste heat recovery cycles and explained vapor compression refrigeration cycle. In the organic Rankine vapor compression cycle, if the refrigerant pressure after the pump is higher than its critical pressure, the cycle will be transcritical. Fig. 1 illustrates the cycle design. Fig. 2 shows cycle processes in *P-h* diagram. In the transcritical cycle, the higher pressure of the cycle is higher than the refrigerant critical pressure (P_{cr}). Through an isentropic compression process the refrigerant can reach Point 10. Oil pump circulates the oil between the heat exchangers, and thus the temperature of T_{12} can be obtained. The high pressure refrigerant with high temperature enters the turbine, and after producing power reach to the condensation pressure (P_{14}). Point 13 (P_{13}) is a state that the refrigerant reach to the condenser pressure (P_4) through an isentropic expansion. The turbine drives the compressor. The compressor increases the working fluid pressure to the P_2. Similarly, Point 3 indicates a state after an isentropic process. The working fluids at the compressor and turbine exhaust are mixed and form state 15. Point 5 is condensation point and at point 6 the refrigerant is assumed to be subcooled. The refrigerant leaves expansion valve at point 7. The process in the expansion valve is isenthalpic. The other part of the refrigerant at the condenser exhaust pressurized in the pump to complete the cycle [3].
There is no need for recuperator in this system since the working fluid in the turbine exhaust is near to the saturation state [21].

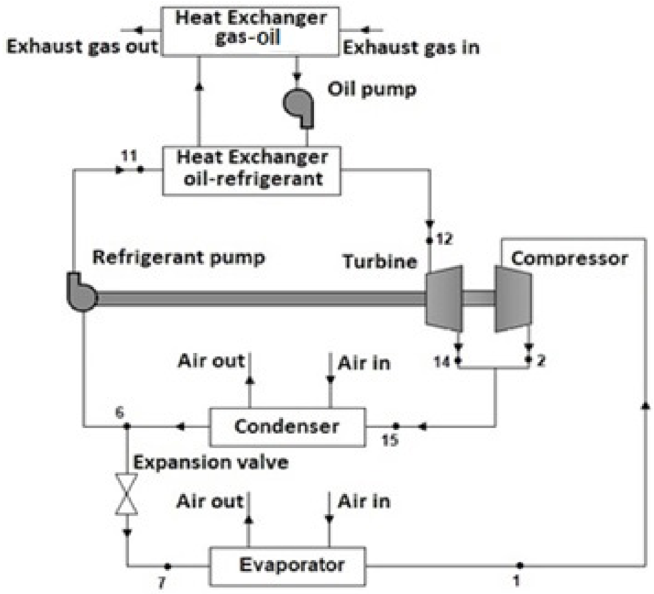

Fig. 1. Schematic scheme of refrigeration system.

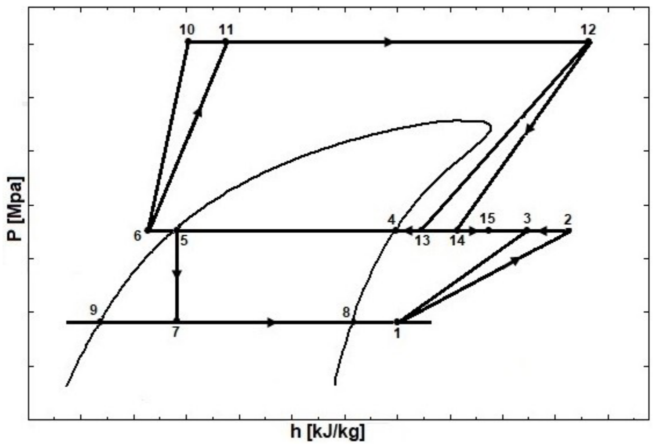

Fig. 2. Schematic display of the refrigeration system in *P-h* diagram for R245fa refrigerant

3- Thermodynamic Analysis

The evaporation temperature (T_e) is assumed to be 10-15 ºC lower than and the condensation temperature (T_c) is 10-15 ºC higher than the inlet temperature of the cooling fluid. ΔT_{SC} and ΔT_{SH} are showing the subcooling and superheating

degrees in the condenser and evaporator, respectively. The values of P_{12} and T_{12} are calculated in a way that their effects on the system coefficient of performance are optimal [3].
Because ΔT_{SC} and ΔT_{SH} are also given, inlet temperature of compressor and outlet temperature condenser can be calculated.
Using the assumed isentropic efficiency for the pump and compressor, points 3 and 10 enthalpies can be determined. Points 2 and 11 enthalpies are obtained from the following statements [3, 22]:

$$h_2 = h_1 + (h_3 - h_1)/\eta_{ci} \tag{1}$$

$$h_{11} = h_6 + (h_{10} - h_6)/\eta_{pi} \tag{2}$$

Using the condenser pressure, the enthalpy of point 13 can be determined. Point 14 enthalpy can be determined as:

$$h_{14} = h_{12} - \eta_{ti}(h_{12} - h_{13}) \tag{3}$$

Considering that the enthalpy of the working fluid remains constant in an expansion process in a valve, one can write:

$$h_7 = h_6 \tag{4}$$

The produced work by the turbine is:

$$\dot{W}_t = \dot{m}_t (h_{12} - h_{14}) \eta_{tm} \tag{5}$$

The compressor consumed power is:

$$\dot{W}_c = \dot{m}_c (h_2 - h_1) \eta_{cm} \tag{6}$$

Similarly, the work needed by the pump is:

$$\dot{W}_p = \dot{m}_t (h_{11} - h_6) \eta_{pm} \tag{7}$$

The compressor and pump work is provided by the turbine, therefore the following expression can be written:

$$\dot{W}_t = \dot{W}_c + \dot{W}_p \tag{8}$$

The values of mechanical efficiency for the turbine, compressor and pump are η_{tm}, η_{cm} and η_{pm}, respectively, and their values are given in Table 1 [3].
w is the ratio of the mass flow rates:

$$w = \frac{\dot{m}_c}{\dot{m}_t} \tag{9}$$

In this system, the mass flow rate can be determined as follows:

$$w = \frac{(h_{12} - h_{14})\eta_{tm}\eta_{cm} - (h_{11} - h_6)(\eta_{cm}/\eta_{pm})}{h_2 - h_1} \tag{10}$$

Enthalpy of the refrigerant at point 15 be determined as [3]:

$$h_{15} = \frac{h_{14} + w \cdot h_2}{1 + w} \tag{11}$$

The ratio of refrigeration capacity to the entering heat to the system in oil-refrigerant heat exchanger is defined as:

$$COPH = \frac{w(h_1 - h_7)}{h_{12} - h_{11}} \tag{12}$$

The following equation shows the ratio of refrigeration capacity to the pump's required power [3, 22]:

$$COPW = \frac{w(h_1 - h_7)\eta_{pm}}{h_{11} - h_6} \tag{13}$$

η_{pm} is mechanical efficiency of the pump. Following equation for COP represents the work needed for compressor to refrigeration capacity [3, 22]:

$$COP = \frac{\dot{Q}_{eva}}{\dot{W}_{compressor}} \tag{14}$$

Table 1. Turbine, compressor and pump efficiencies

η_{ti}	η_{ci}	η_{pi}	η_{tm}	η_{cm}	η_{pm}
0. 8	0.75	0. 8	0.97	0.97	0.97

3- 1- Heat exchangers analysis

Heat exchangers of the system are assumed to be the counter flow type. Fig. 3 shows the temperature distributions in the heat exchangers.
Heat transfer in all heat exchangers can be determined from following equations [23]:

$$\dot{Q} = UF\Delta T_m \tag{15}$$

$$\dot{Q} = \dot{m} c_p \Delta T_{ie} \tag{16}$$

$$\dot{Q} = \dot{m}\Delta h_{ie} \tag{17}$$

$\dot{Q}$, UF, and ΔT_m are heat transfer rate, overall heat transfer coefficient and Logarithmic Mean Temperature Difference (LMTD).
In the summer for bus air conditioning, the temperature inside the bus can be adjusted at 25 ºC. [24, 25]. The outside temperature of the bus can be assumed as 35 ºC [6]. To dehumidify the air, evaporation temperature can be selected at 10 ºC. Outside highest temperature can be supposed as 42 ºC [25]. Therefore, selecting condensation temperature as 50 ºC, would be appropriate. Subcooling and superheating temperature differences are assumed as $\Delta T_{SC} = \Delta T_{SH} = 3$ºC.
Mastrullo et al. [26], selected diesel engine exhaust gases average composition as follows:
$N_2 = 0.7304$ - $H_2O = 0.0537$ - $O_2 = 0.0649$ - $CO_2 = 0.1510$
Gasoline engine, exhaust gases average composition are selected as follows:
$N_2 = 0.7192$ - $H_2O = 0.08854$ - $CO_2 = 0.1923$ [26].
To prevent corrosion incident in gas – oil heat exchanger because the risk of H_2O condensation, exhaust gases temperature after heat recovery is assumed to be 110 ºC.
Santiago et al. [27] utilized therminol-vp1 as heat transfer

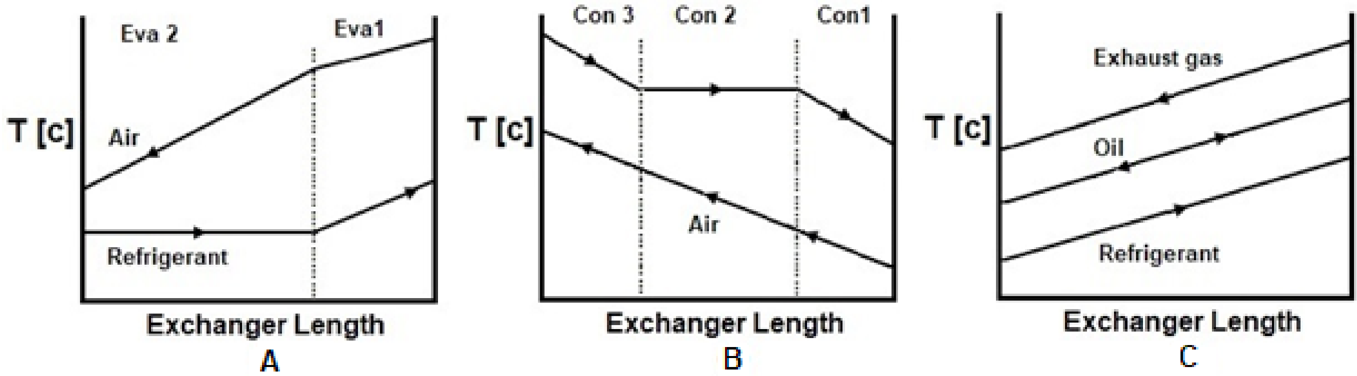

Fig. 3. (A) Evaporator, (B) Condenser, (C) heat exchanger of the exhaust gas

fluid, and in this study it has been used as the heat transfer fluid. Fig. 2 shows the specifications of the engine exhaust gases in operational conditions.

4- Exergy Analysis

The entropy of exhaust gases can be calculated as follows [28]:

$$\begin{aligned} s_{\text{üüüü}}(T,P) &= X_{\ \,2}\, s_{\ \,2}\left(T,P_{\ \,2}\right) \\ &+ X_{\mathrm{O}_{\text{üüüü}}} s_{\mathrm{O}}\left(T,P_{\mathrm{O}}\right) + X_{\mathrm{H\,O}} s_{\mathrm{H\,O}}\left(T,P_{\mathrm{H\,O}}\right) + X_{\mathrm{N}} s_{\mathrm{N}}\left(T,P_{\mathrm{N}}\right) \end{aligned} \quad (18)$$

The following equations represent mass, energy and exergy balance [22]:

$$\sum \dot{m}_{\mathrm{i}} = \sum \dot{m}_{\mathrm{e}} \quad (19)$$

$$\dot{Q} + \sum \dot{m}_{\mathrm{i}} h_{\mathrm{i}} = \dot{W} + \sum \dot{m}_{\mathrm{e}} h_{\mathrm{e}} \quad (20)$$

$$Ex_{\mathrm{Q}} + \sum \dot{m}_{\mathrm{i}} ex_{\mathrm{i}} = Ex_{\mathrm{W}} + \sum \dot{m}_{\mathrm{e}} ex_{\mathrm{e}} + Ex_{\mathrm{D}} \quad (21)$$

The inputs and outputs of the control volume are represented by i and e. Ex_D is the exergy destruction rate. Ex_C is the heat transfer exergy rate, $\dot{E}x_Q=(1-(T_0/T_i))\dot{Q}_i$ and Ex_W is the mechanical power exergy rate, $\dot{E}x_W=\dot{W}$. Here $\dot{Q}$ is the heat as well as $\dot{W}$ is the work rates while T_i is the average thermodynamic temperature.
The physical exergy can be calculated as follows [29, 30]:

$$ex_{\mathrm{ph}} = \left(h - h_0\right) - T_0\left(s - s_0\right) \quad (22)$$

$$\dot{E}x = \dot{m}\left[\left(h - h_0\right) - T_0\left(s - s_0\right)\right] \quad (23)$$

For each component the exergy efficiency is calculated as [31, 32]:

$$\eta_{\mathrm{II}} = \frac{\text{exergy in product}}{\text{total exergy input}} \quad (24)$$

4- 1- Fuel and product

The definition of the exergy of fuel and product for the refrigeration system components are as given in Table 3.

5- Results and Discussions

Alper Yilmaz [3] selected optimum values of *COP* for both R134a and R245fa refrigerants. In the system, turbine provides power of the pump and the compressor. Therefore, the important factor for the system performance is *COPH*, thus the effects of P_{12} and T_{12} to *COPH* will be investigated. In order to find optimum value of *COPH* for each refrigerant at P_{12} and T_{12}, thermodynamic analyzes have been performed by using Engineering Equation Solver (EES) software.
To calculate cycle point's thermodynamic properties for refrigerants R134a, R245fa, R290, R600, R1234yf and R1270 finding optimum value of *COPH* is necessary. Variation of *COPH* for the refrigerants R290, R600, R1234yf and R1270 are shown in Figs. 4 to 9. In this study, optimum values of *COPH* are selected for each refrigerant regarding to T_{12} and P_{12}.
In the system importance of *COPH* is more than the other coefficient of performances, because it indicates the ratio of heat energy received from exhaust gases to refrigeration capacity. With increasing of P_{12}, the power needed to run the pump also will be increased, therefore it will be important to select optimum value for *COPH*.
Figs. 4 to 9 are showing that R245fa and R600 refrigerants have better performances and have high refrigeration capacities and *COPH* in comparison with other used refrigerants. Optimum COPH value is 0.614 for refrigerant R245fa and 0.606 for R600.
Thermodynamic properties of refrigeration system running with diesel engine exhaust gases heat energy at 1000 rpm and 25% load with R600 refrigerant are given in Table 4 and the system exergy analysis is demonstrated in Table 5.
From Table 5 it can be inferred that Gas – Oil heat exchanger has the most exergy destruction. In order to avoid decomposition of refrigerant, between exhaust gases and working fluids, a heat transfer fluid is added. High energy exhaust gases with temperature of 363.1 ºC, increases the heat transfer fluid temperature to 281.55 ºC. Because all possible heating potential of the exhaust gases are not used, exergy loss of exhaust gases is predictable.
Figs. 10 to 21 are showing the refrigeration capacities that generated by system using exhaust gases energies of both gasoline and diesel engines.

Table 2. Engines exhaust gases properties [26]

		Engine speed (rpm)					
		1000	**2000**	**3000**	**4000**	**5000**	**6000**
Mass flow rate of Gasoline engine exhaust gas (kg/s) Torque (Nm)	225			0.080	0.110	0.143	
	200		0.046	0.068	0.090	0.118	0.148
	150	0.018	0.033	0.055	0.071	0.093	0.110
	100	0.015	0.0262	0.038	0.053	0.070	0.083
	50	0.01	0.017	0.023	0.033	0.043	0.054
		Engine speed (rpm)					
		1000	**2000**	**3000**	**4000**	**5000**	**6000**
Gasoline exhaust gas temperature (kg/s) Torque (Nm)	240			692	747	772	
	225		597	682	742	797	817
	200		572	662	727	787	812
	150	327	502	627	702	772	802
	100	307	447	527	627	702	777
	50	352	417	502	567	607	657

		Engine speed (rpm)		
		1300	**1700**	**2100**
Mass flow rate of Diesel engine exhaust gas (kg/s) Load (%)	100	0.490	0.512	0.439
	75	0.395	0.441	0.410
	50	0.306	0.368	0.352
	25	0.205	0.246	0.269
		Engine speed (rpm)		
		1300	**1700**	**2100**
Diesel exhaust gas temperature Load (%)	100	503.5	483.7	435.2
	75	459.3	423.3	382.4
	50	394.2	353.6	317.6
	25	363.1	334.5	291.0

Table 3. Fuel and product exergy definitions for refrigeration system.

Component	Fuel exergy rate (kW)	product exergy rate (kW)
Heat exchanger gas - oil	$\dot{E}x_{exhaust\ gas\ in} - \dot{E}x_{exhaust\ gas\ out}$	$\dot{E}x_{oil\ in} - \dot{E}x_{oil\ out}$
Heat exchanger oil - refrigerant	$\dot{E}x_{oil\ in} - \dot{E}x_{oil\ out}$	$\dot{E}x_{12} - \dot{E}x_{11}$
Refrigerant pump	$\dot{W}_p$	$\dot{E}x_{11} - \dot{E}x_6$
Turbine	$\dot{E}x_{12} - \dot{E}x_{14}$	$\dot{W}_t$
Compressor	$\dot{W}_c$	$\dot{E}x_2 - \dot{E}x_1$
Condenser	$\dot{E}x_{air\ out} - \dot{E}x_{air\ in}$	$\dot{E}x_{15} - \dot{E}x_6$
Expansion valve	$\dot{E}x_6$	$\dot{E}x_7$
Evaporator	$\dot{E}x_1 - \dot{E}x_7$	$\dot{E}x_{air\ out} - \dot{E}x_{air\ in}$

In the gasoline engine for each torque with increasing of engine speed, temperature of exhaust gases and its mass flow rate will be increased. Therefore, the heat energy of the exhaust gases entering the system as well as the refrigeration capacity will be increased. In the diesel engine for each engine load with increasing of engine speed from 1300 to 2100 rpm, exhaust gases temperature decreases, and except the 25% engine load, mass flow rate of exhaust gases increases from 1300 to 1700 rpm, and then decreases from 1700 to 2100 rpm. Changes in engine exhaust gases mass flow rate and temperature with variation of engine load and speed, cause changes in heat energy input to the system. Accordingly, system refrigeration capacities will vary with diesel engine speed and loads.

Figs. 22 and 23 are showing the performance table of SD6V12 and FK40-560 compressors, in which the SD6V12 compressor can be used in passenger cars and FK40-560 compressor can be used in buses.

Table 4. Thermodynamic properties of system with R600 refrigerant (system is running with the heat energy of exhaust gases at 1000 rpm and 25% load of diesel engine)

Points in cycles	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Temperature (℃)	13	56.09	50	50.0	50.0	47.0	10.0	10.0	10.0	49.82	50.81	200	93.59	104	79.23
Pressure (kPa)	1148.7	4496.6	4496.6	496.6	4496.6	4496.6	148.7	1148.7	1148.7	6000	66000	6000	4496.6	4496.6	4496.6
Enthalpy (kJ/kg)	603.9	667.8	651.8	655.5	322.1	314.4	314.4	598.8	223.5	324.4	326.9	854.9	744.5	766.6	714.7
Entropy (kJ/kg.K)	22.428	22.477	22.428	2.439	11.408	11.383	1.405	22.41	11.084	1.383	11.391	2.697	22.697	22.757	22.614
$\dot{m}$ (kg/s)	00.119	00.119	00.119	0.226	00.226	00.226	0.119	00.119	00.119	0.107	00.107	0.107	00.107	00.107	00.226
e_{ph} (kJ/kg)	118.35	667.09	666.24	66.41	550.91	550.58	43.9	118.77	551.93	60.58	660.7	186.2	775.84	779.62	771.58
$\dot{E}$ (kW)	22.183	77.983	77.882	15	111.50	111.43	5.224	22.233	36.179	6.482	66.494	19.92	88.114	88.519	116.17

Table 5. Exergy results for the refrigeration system with R600 refrigerant (system is running with the heat energy of exhaust gases at 1000 rpm and 25% load of diesel engine)

Component	$\dot{E}x_F$ (kW)	$\dot{E}x_P$ (kW)	$\dot{E}x_D$ (kW)	η_{II}
Gas – Oil Heat exchanger	21.95	17.30	4.65	78.81%
Oil – Refrigerant Heat exchanger	17.30	13.55	3.75	78.32%
Refrigerant pump	1.39	1.09	0.3	78.41%
Turbine	11.51	9.24	2.27	80.27%
Compressor	7.85	5.81	2.04	74.01%
Condenser	5.58	1.44	4.14	25.80%
Evaporator	3.04	1.74	1.3	57.23%

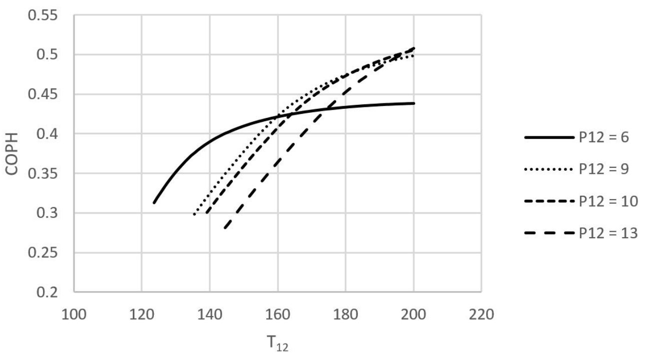

Fig. 4. ***COPH*** **versus temperature and pressure for R134a refrigerant. Optimum value for** ***COPH*** **is at** P_{12} **= 9 MPa,** T_{12} **= 180 °C , (*COPH* = 0.4742 – *COPW* = 12.55 – *COP* = 4.098)**

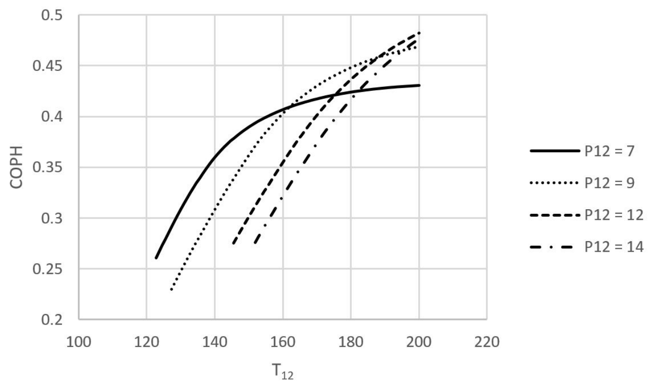

Fig. 6. ***COPH*** **versus temperature and pressure for R600. Optimum value for** ***COPH*** **is at** P_{12} **= 12 MPa,** T_{12} **= 200 °C , (*COPH* = 0.4824 – *COPW* = 8.142 – *COP* = 3.997)**

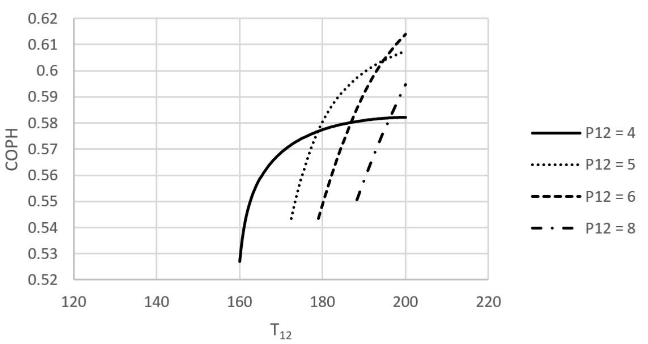

Fig. 5. ***COPH*** **versus temperature and pressure for R245fa. Optimum value for** ***COPH*** **is at** P_{12} **= 6 MPa,** T_{12} **= 200 °C , (*COPH* = 0.614 – *COPW* = 29.16 – *COP* = 4.339)**

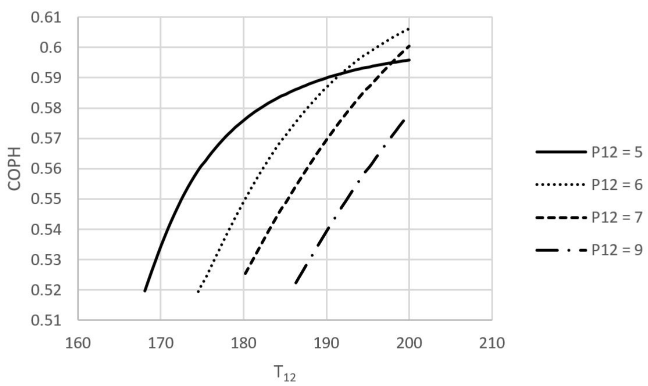

Fig. 7. ***COPH*** **versus temperature and pressure for R290. Optimum value for** ***COPH*** **is at** P_{12} **= 6 MPa,** T_{12} **= 200 °C , (*COPH* = 0.6062 – *COPW* = 24.84 – *COP* = 4.32)**

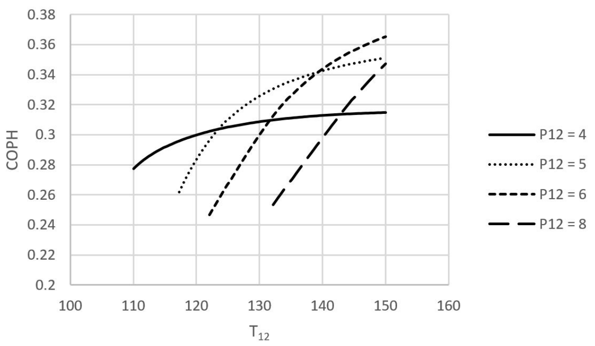

Fig. 8. ***COPH*** **versus temperature and pressure for R1234yf. Optimum value for** ***COPH*** **is at** P_{12} **= 6 MPa,** T_{12} **= 150 °C , (*COPH* = 0.3652 – *COPW* = 11.59 – *COP* = 3.903)**

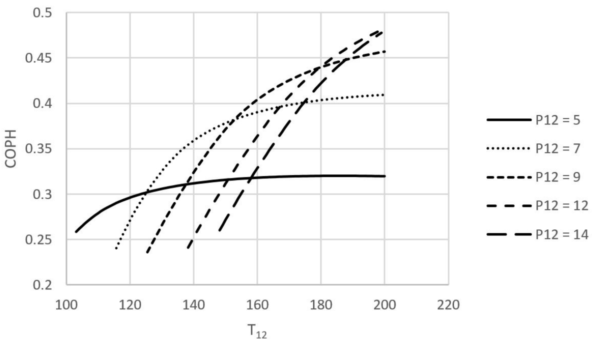

Fig. 9. ***COPH*** **versus temperature and pressure for R1270. Optimum value for** ***COPH*** **is at** P_{12} **= 12 MPa,** T_{12} **= 200 °C , (*COPH* = 0.4819 – *COPW* = 8.365 – *COP* = 3.971)**

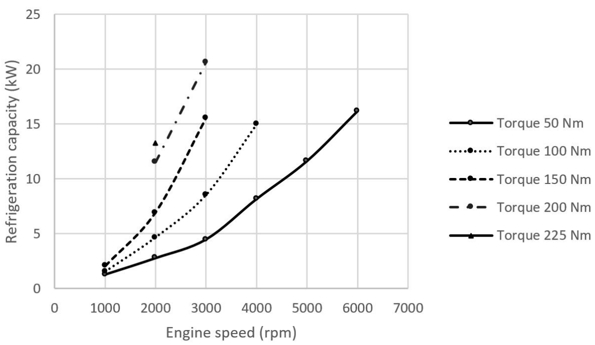

Fig. 10. Generated refrigeration capacities by system with refrigerant R134a in different speeds and torques of gasoline engine

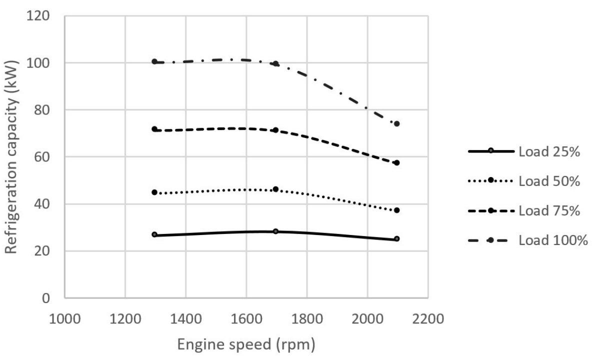

Fig. 11. Generated refrigeration capacities by system with refrigerant R134a in different speeds and loads of diesel engine

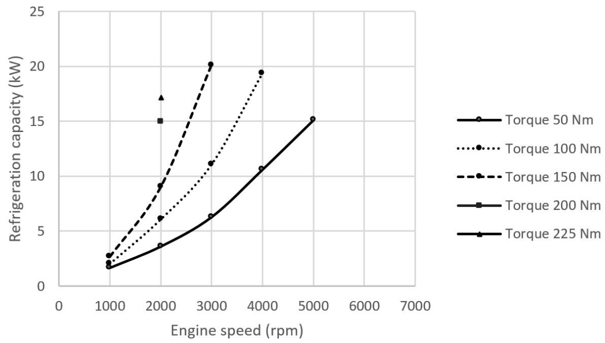

Fig. 12. Generated refrigeration capacities by system with refrigerant R245fa in different speeds and torques of gasoline engine

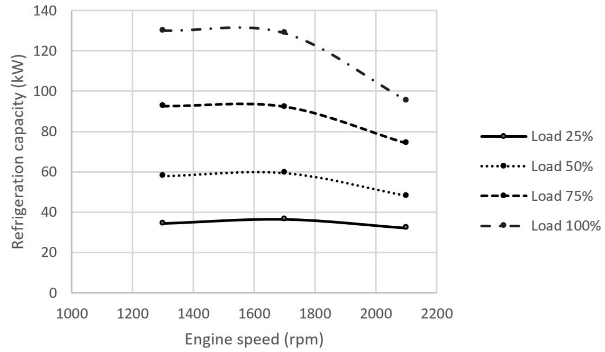

Fig. 13. Generated refrigeration capacities by system with refrigerant R245fa in different speeds and loads of diesel engine

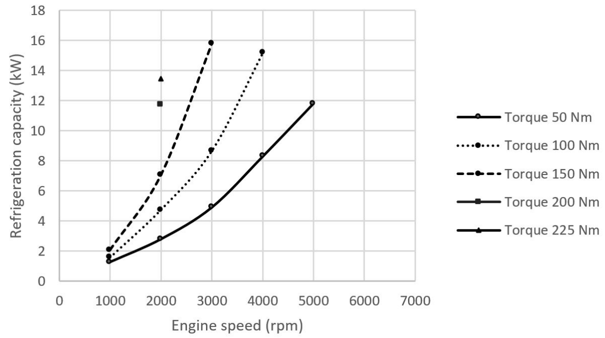

Fig. 14. Generated refrigeration capacities by system with refrigerant R290 in different speeds and torques of gasoline engine

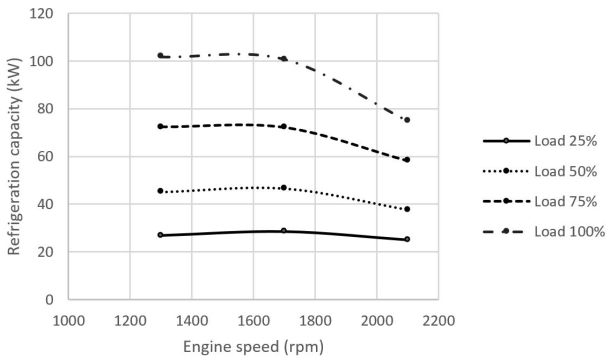

Fig. 15. Generated refrigeration capacities by system with refrigerant R290 in different speeds and loads of diesel engine

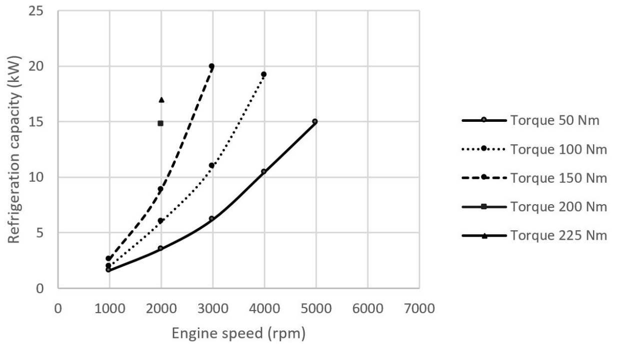

Fig. 16. Generated refrigeration capacities by system with refrigerant R600 in different speeds and torques of gasoline engine

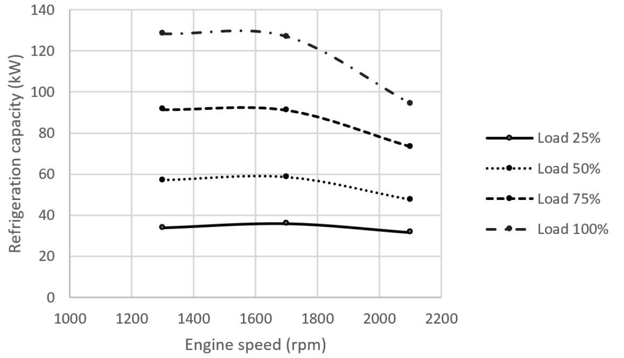

Fig. 17. Generated refrigeration capacities by system with refrigerant R600 in different speeds and loads of diesel engine

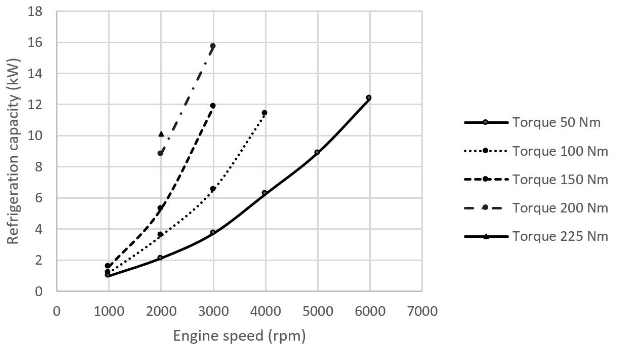

Fig. 18. Generated refrigeration capacities by system with refrigerant R1234yf in different speeds and torques of gasoline engine

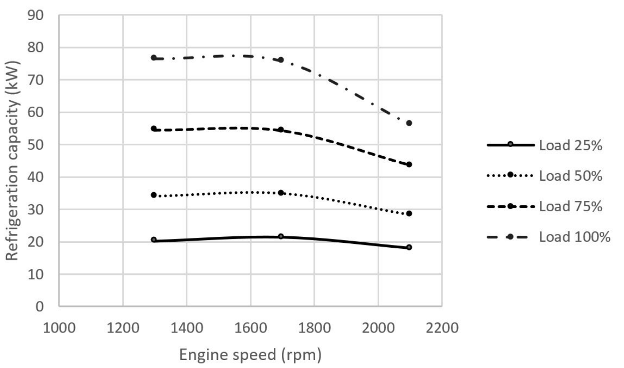

Fig. 19. Generated refrigeration capacities by system with refrigerant R1234yf in different speeds and loads of diesel engine

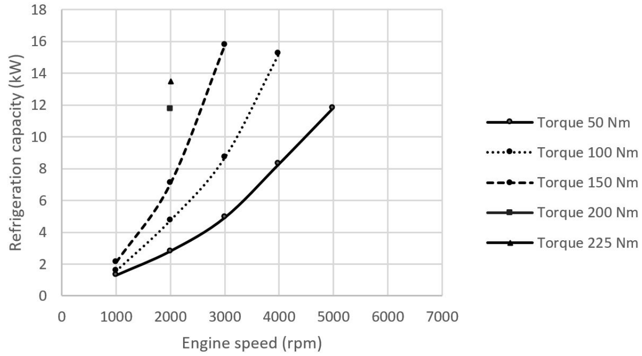

Fig. 20. Generated refrigeration capacities by system with refrigerant R1270 in different speeds and torques of gasoline engine

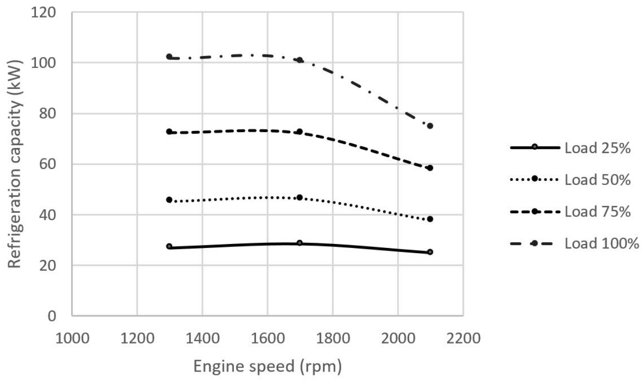

Fig. 21. Generated refrigeration capacities by system with refrigerant R1270 in different speeds and loads of diesel engine

If created refrigeration capacities in Figs. 10 to 21 compared with the refrigeration capacities of compressors in Figs. 22 and 23, it will be inferred that system can provide required refrigeration capacities of both passenger cars and buses, and also extra refrigeration capacities will be available.

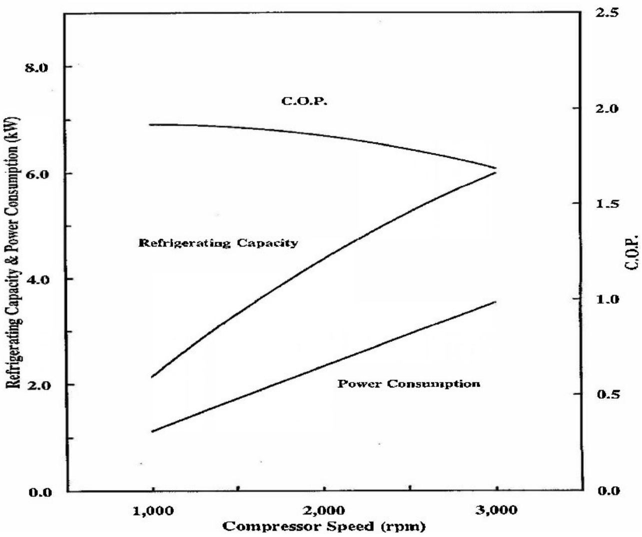

Fig. 22. SD6V12 performance diagram [33]

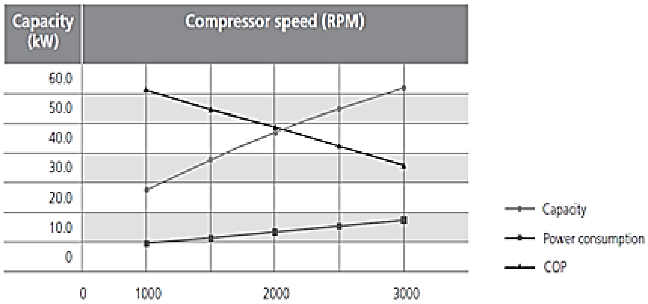

Fig. 23. FK40/560 performance diagram [34]

6- Conclusions

In this study, thermodynamics analysis of a new cogeneration system has been investigated, and the results showed that system is capable to create required refrigeration capacities of conventional automobiles and buses.

- Maximum recoverable heat energy from exhaust gases of diesel engine at 1000 rpm and 25% load is about 56.99 kW.
- Different refrigerants are used in the system. Results show that the system can generate required refrigeration capacities of both automobiles and buses. Furthermore, additional refrigeration capacities will also be available.
- R245fa and R600 refrigerants have better performances in the system.
- Generated refrigeration capacity with that amount of heat energy by system with R600 refrigerant is about 33.93 kW. This amount of refrigeration capacity is within the range of FK40/560 compressor refrigeration capacity.
- A parametric study is performed to reveal the influences of engine speed on the refrigeration capacity for different conditions.
- Based on thermodynamic analysis, the proposed system has the ability to be replaced with vehicles conventional refrigeration systems.

References

[1] G. Vicatos, J. Gryzagoridis, S.J.J.o.E.i.S.A. Wang, A car air-conditioning system based on an absorption refrigeration cycle using energy from exhaust gas of an internal combustion engine. *Journal of Energy in Southern Africa*, 19(4) (2008) 6-11.

[2] H.A. Daanen, E. Van De Vliert, X.J.A.e. Huang, Driving performance in cold, warm, and thermoneutral environments. *Applied Ergonomics*, 34(6) (2003) 597-602.

[3] A.J.E. Yılmaz, Transcritical organic Rankine vapor compression refrigeration system for intercity bus air-conditioning using engine exhaust heat. *Energy*, 82 (2015) 1047-1056.

[4] A.B. Little, S.J.E. Garimella, Comparative assessment of alternative cycles for waste heat recovery and upgrade. *Energy*, 36(7) (2011) 4492-4504.

[5] H. Wang, R. Peterson, K. Harada, E. Miller, R. Ingram-Goble, L. Fisher, J. Yih, C.J.E. Ward, Performance of a combined organic Rankine cycle and vapor compression cycle for heat activated cooling. *Energy*, 36(1) (2011) 447-458.

[6] H. Wang, R. Peterson, T.J.E. Herron, Design study of configurations on system COP for a combined ORC (organic Rankine cycle) and VCC (vapor compression cycle). *Energy*, 36(8) (2011) 4809-4820.

[7] J. Jeong, Y.T.J.I.j.o.r. Kang, Analysis of a refrigeration cycle driven by refrigerant steam turbine. *International Journal of Refrigeration*, 27(1) (2004) 33-41.

[8] H. Li, X. Bu, L. Wang, Z. Long, Y.J.E. Lian, buildings, Hydrocarbon working fluids for a Rankine cycle powered vapor compression refrigeration system using low-grade thermal energy. *Energy and Buildings*, 65 (2013) 167-172.

[9] T. Wang, Y. Zhang, Z. Peng, G.J.R. Shu, s.e. reviews, A review of researches on thermal exhaust heat recovery with Rankine cycle. *Renewable and Sustainable Energy Reviews*, 15(6) (2011) 2862-2871.

[10] G. Yu, G. Shu, H. Tian, H. Wei, L.J.E. Liu, Simulation and thermodynamic analysis of a bottoming Organic Rankine Cycle (ORC) of diesel engine (DE). *Energy*, 51 (2013) 281-290.

[11] F. Salek, A.N. Moghaddam, M.M.J.E.C. Naserian, Management, Thermodynamic analysis of diesel engine coupled with ORC and absorption refrigeration cycle. *Energy Conversion and Management*, 140 (2017) 240-246.

[12] F. Velez, J.J. Segovia, M.C. Martín, G. Antolin, F. Chejne, A.J.F.P.T. Quijano, Comparative study of working fluids for a Rankine cycle operating at low temperature. *Fuel Processing Technology*, 103 (2012) 71-77.

[13] B.F. Tchanche, G. Lambrinos, A. Frangoudakis, G.J.R. Papadakis, S.E. Reviews, Low-grade heat conversion into power using organic Rankine cycles–A review of various applications. *Renewable and Sustainable Energy Reviews*, 15(8) (2011) 3963-3979.

[14] G. Shu, L. Liu, H. Tian, H. Wei, G.J.A.E. Yu, Parametric and working fluid analysis of a dual-loop organic Rankine cycle (DORC) used in engine waste heat recovery. *Applied Energy*, 113 (2014) 1188-1198.

[15] S. Daviran, A. Kasaeian, S. Golzari, O. Mahian, S. Nasirivatan, S.J.A.T.E. Wongwises, A comparative study on the performance of HFO-1234yf and HFC-134a as an alternative in automotive air conditioning systems. *Applied Thermal Engineering*, 110 (2017) 1091-1100.

[16] D. Gewald, K. Siokos, S. Karellas, H.J.R. Spliethoff, S.E. Reviews, Waste heat recovery from a landfill gas-fired power plant. *Renewable and Sustainable Energy Reviews*, 16(4) (2012) 1779-1789.

[17] H.C. Bayrakçi, A.E.J.I.J.o.E.R. Özgür, Energy and exergy analysis of vapor compression refrigeration system using pure hydrocarbon refrigerants. *International Journal of Energy Research*, 33(12) (2009) 1070-1075.

[18] J.U. Ahamed, R. Saidur, H.H. Masjuki, M.J.I.j.o.G.e. Sattar, An analysis of energy, exergy, and sustainable development of a vapor compression refrigeration system using hydrocarbon. *International Journal of Green Energy*, 9(7) (2012) 702-717.

[19] Y. Chang, M. Kim, S.J.I.j.o.r. Ro, Performance and heat transfer characteristics of hydrocarbon refrigerants in a heat pump system. *International Journal of Refrigeration*, 23(3) (2000) 232-242.

[20] A. Schuster, S. Karellas, E. Kakaras, H.J.A.t.e. Spliethoff, Energetic and economic investigation of Organic Rankine Cycle applications. *Applied Thermal Engineering*, 29(8-9) (2009) 1809-1817.

[21] Y.A. Cengel, M.A. Boles, Thermodynamics: An Engineering Approach 5th ed., McGraw-Hill, New York, 2006.

[22] J. P.Holman, *Heat Transfer*, 10th ed., McGraw-Hill, New York, 2010.

[23] Automobile and Mass Transport ASHRAE handbook-HVAC applications, American Society of Heating, Refrigerating and Air Conditioning Engineers, 2007.

[24] R.C. Arora, *Refrigeration and air conditioning*, PHI Learning Pvt. Ltd., 2012.

[25] R. Mastrullo, A.W. Mauro, R. Revellin, L.J.E.C. Viscito, Management, Modeling and optimization of a shell and louvered fin mini-tubes heat exchanger in an ORC powered by an internal combustion engine. *Energy Conversion and Management*, 101 (2015) 697-712.

[26] S.S. de la Fuente, D. Roberge, A.R.J.M.P. Greig, Safety and CO2 emissions: Implications of using organic fluids in a ship's waste heat recovery system. *Marine Policy*, 75 (2017) 191-203.

[27] B.E. Poling, J.M. Prausnitz, J.P. O'connell, The properties of gases and liquids, Mcgraw-hill New York, 2001.

[28] E.F. Kreith, Moran, MJ, Tsatsaronis, G., *Engineering Thermodynamics*. The CRC Handbook of Thermal Engineering. Ed. Frank Kreith Boca Raton: CRC Press LLC, 2000, (2000).

[29] I. Dincer, M.A. Rosen, Exergy: energy, environment and sustainable development, Elsevier, New York, 2012.

[30] M. Kanoglu, A. Ayanoglu, A.J.E. Abusoglu, Exergoeconomic assessment of a geothermal assisted high temperature steam electrolysis system. *Energy*, 36(7) (2011) 4422-4433.

[31] M. Yari, S.J.A.T.E. Mahmoudi, Utilization of waste heat from GT-MHR for power generation in organic Rankine cycles. *Applied Thermal Engineering*, 30(4) (2010) 366-375.

[32] http://www.sanden.com/objects/SD6V12_Performance.pdf

[33] https://www.webasto.com/fileadmin/webasto_files/documents/international/hd/catalogues/heavy-duty-air-conditioning-accessories-catalog.pdf

Please cite this article using:

T. Ghorbani, M. Yari, F. Mohammadkhani, Thermodynamic Analysis and Feasibility Study of Internal Combustion Engine Waste Heat Recovery to Run its Refrigeration System, *AUT J. Mech. Eng.*, 2(2) (2018) 253-262.

DOI: 10.22060/ajme.2018.13885.5694