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

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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.

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.

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 °C 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 °C. Daviran

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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_c). 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].

3- Thermodynamic Analysis

The evaporation temperature (T_e) is assumed to be 10-15 °C lower than and the condensation temperature (T_c) is 10-15 °C higher than the inlet temperature of the cooling fluid. \( \Delta T_{sc} \) and \( \Delta T_{sh} \) are showing the subcooling and superheating
In this system, the mass flow rate can be determined as:

\[ h_2 = h_1 + \left( h_3 - h_1 \right) / \eta_{pi} \]  

(1)

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

\[ h_{14} = h_{12} - \eta_p \left( h_{12} - h_{13} \right) \]  

(3)

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

\[ h_{26} = h_b \]  

(4)

The produced work by the turbine is:

\[ W_h = \dot{m}_i \left( h_{12} - h_{14} \right) \eta_{tm} \]  

(5)

The compressor consumed power is:

\[ W_c = \dot{m}_i \left( h_2 - h_i \right) \eta_{cp} \]  

(6)

Similarly, the work needed by the pump is:

\[ W_p = \dot{m}_i \left( h_{11} - h_b \right) \eta_{pm} \]  

(7)

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

\[ W_c = W_h + W_p \]  

(8)

The values of mechanical efficiency for the turbine, compressor and pump are \( \eta_{tm} \), \( \eta_{cp} \), and \( \eta_{pm} \), respectively, and their values are given in Table 1 [3].

\( w \) is the ratio of the mass flow rates:

\[ w = \frac{m_i}{\dot{m}_i} \]  

(9)

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

\[ w = \frac{\left( h_{12} - h_{14} \right) \eta_{tm} \eta_{cp} - \left( h_{11} - h_b \right) \left( \eta_{pm} / \eta_{tm} \right)}{h_{26} - h_1} \]  

(10)

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

\[ h_{15} = \frac{h_{14} + w \cdot h_3}{1 + w} \]  

(11)

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

\[ COPH = \frac{w \left( h_1 - h_7 \right)}{h_{12} - h_1} \]  

(12)

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

\[ COPW = \frac{w \left( h_1 - h_7 \right) \eta_{pm}}{h_{11} - h_b} \]  

(13)

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

\[ COP = \frac{\dot{Q}_{eva}}{W_{\text{compressor}}} \]  

(14)

### Table 1. Turbine, compressor and pump efficiencies

<table>
<thead>
<tr>
<th>( \eta_{tm} )</th>
<th>( \eta_{pi} )</th>
<th>( \eta_{cp} )</th>
<th>( \eta_{pm} )</th>
<th>( \eta_{pm} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.8</td>
<td>0.75</td>
<td>0.97</td>
<td>0.97</td>
<td>0.97</td>
</tr>
</tbody>
</table>

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]:

\[ \dot{Q} = U F \Delta T_m \]  

(15)

\[ \dot{Q} = \dot{m} \Delta h_{te} \]  

(16)

\[ \dot{Q} = \dot{m} \Delta h_{te} \]  

(17)

\( \dot{Q}, UF, \) and \( \Delta T_m \) 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 °C. [24, 25]. The outside temperature of the bus can be assumed as 35 °C [6]. To dehumidify the air, evaporation temperature can be selected at 10 °C. Outside highest temperature can be supposed as 42 °C [25]. Therefore, selecting condensation temperature as 50 °C, would be appropriate. Subcooling and superheating temperature differences are assumed as \( \Delta T_{sc} = \Delta T_{sh} = 3^\circ \text{C} \).

Mastrullo et al. [26], selected diesel engine exhaust gases average composition as follows:

\[ N_2 = 0.7304 \cdot H_2O = 0.0537 \cdot O_2 = 0.0649 \cdot CO_2 = 0.1510 \]

Gasoline engine, exhaust gases average composition are selected as follows:

\[ N_2 = 0.7192 \cdot H_2O = 0.08854 \cdot CO_2 = 0.1923 \]

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°C. Santiago et al. [27] utilized thermolin-vp1 as heat transfer
4- Exergy Analysis

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

\[ s_{ex}(T, P) = X_{CO_2} s_{CO_2}(T, P_{CO_2}) + X_{H_2O} s_{H_2O}(T, P_{H_2O}) + X_{N_2} s_{N_2}(T, P_{N_2}) \]  

(18)

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

\[ \sum m_i = \sum m_e \]  

(19)

\[ \dot{Q} + \sum m_i h_i = \dot{W} + \sum m_e h_e \]  

(20)

\[ Ex_Q + \sum m_i ex_i = Ex_w + \sum m_e ex_e + Ex_D \]  

(21)

The inputs and outputs of the control volume are represented by \( i \) and \( e \). \( Ex_{ph} \) is the exergy destruction rate. \( Ex_e \) is the heat transfer exergy rate, \( \dot{Ex}_w = (1 - (T_i / T)) \dot{Q} \) and \( Ex_w \) is the mechanical power exergy rate, \( \dot{Ex}_w = \dot{W} \). 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]:

\[ ex_{ph} = (h - h_0) - T_0 (s - s_0) \]  

(22)

\[ Ex = \dot{m} \left[ (h - h_0) - T_0 (s - s_0) \right] \]  

(23)

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

\[ \eta_h = \frac{ex_{product}}{ex_{total input}} \]  

(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 \( P_{ij} \) and \( T_{ij} \) to \( COPH \) will be investigated. In order to find optimum value of \( COPH \) for each refrigerant at \( P_{ij} \) and \( T_{ij} \), 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 \( T_{ij} \) and \( P_{ij} \).

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 \( P_{ij} \), 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 °C, increases the heat transfer fluid temperature to 281.55 °C. 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.
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)

<table>
<thead>
<tr>
<th>Points in cycles</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
<th>9</th>
<th>10</th>
<th>11</th>
<th>12</th>
<th>13</th>
<th>14</th>
<th>15</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature (°C)</td>
<td>13</td>
<td>56.09</td>
<td>50</td>
<td>50.0</td>
<td>50.0</td>
<td>47.0</td>
<td>10.0</td>
<td>10.0</td>
<td>10.0</td>
<td>10.0</td>
<td>49.82</td>
<td>50.81</td>
<td>200</td>
<td>93.59</td>
<td>104</td>
</tr>
<tr>
<td>Pressure (kPa)</td>
<td>1148.7</td>
<td>4496.6</td>
<td>4496.6</td>
<td>496.6</td>
<td>4496.6</td>
<td>4496.6</td>
<td>148.7</td>
<td>1148.7</td>
<td>1148.7</td>
<td>6000</td>
<td>66000</td>
<td>6000</td>
<td>4496.6</td>
<td>4496.6</td>
<td>4496.6</td>
</tr>
<tr>
<td>Enthalpy (kJ/kg)</td>
<td>603.9</td>
<td>667.8</td>
<td>651.8</td>
<td>655.5</td>
<td>322.1</td>
<td>314.4</td>
<td>314.4</td>
<td>598.8</td>
<td>223.5</td>
<td>324.4</td>
<td>326.9</td>
<td>854.9</td>
<td>744.5</td>
<td>766.6</td>
<td>714.7</td>
</tr>
<tr>
<td>m (kg/s)</td>
<td>0.119</td>
<td>0.119</td>
<td>0.119</td>
<td>0.226</td>
<td>0.226</td>
<td>0.226</td>
<td>0.119</td>
<td>0.119</td>
<td>0.119</td>
<td>0.107</td>
<td>0.107</td>
<td>0.107</td>
<td>0.107</td>
<td>0.107</td>
<td>0.107</td>
</tr>
<tr>
<td>$e_p$ (kJ/kg)</td>
<td>118.35</td>
<td>667.09</td>
<td>666.24</td>
<td>66.41</td>
<td>550.91</td>
<td>550.58</td>
<td>43.9</td>
<td>118.77</td>
<td>551.93</td>
<td>60.58</td>
<td>660.7</td>
<td>186.2</td>
<td>775.84</td>
<td>779.62</td>
<td>771.58</td>
</tr>
<tr>
<td>$\dot{E}$ (kW)</td>
<td>22.183</td>
<td>77.983</td>
<td>77.882</td>
<td>15</td>
<td>111.50</td>
<td>111.43</td>
<td>5.224</td>
<td>22.233</td>
<td>36.179</td>
<td>6.482</td>
<td>66.494</td>
<td>19.92</td>
<td>88.114</td>
<td>88.519</td>
<td>116.17</td>
</tr>
</tbody>
</table>

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)

<table>
<thead>
<tr>
<th>Component</th>
<th>$\dot{E}_{X_f}$ (kW)</th>
<th>$\dot{E}_{X_p}$ (kW)</th>
<th>$\dot{E}_{X_D}$ (kW)</th>
<th>$\eta_{II}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gas – Oil Heat exchanger</td>
<td>21.95</td>
<td>17.30</td>
<td>4.65</td>
<td>78.81%</td>
</tr>
<tr>
<td>Oil – Refrigerant Heat exchanger</td>
<td>17.30</td>
<td>13.55</td>
<td>3.75</td>
<td>78.32%</td>
</tr>
<tr>
<td>Refrigerant pump</td>
<td>1.39</td>
<td>1.09</td>
<td>0.3</td>
<td>78.41%</td>
</tr>
<tr>
<td>Turbine</td>
<td>11.51</td>
<td>9.24</td>
<td>2.27</td>
<td>80.27%</td>
</tr>
<tr>
<td>Compressor</td>
<td>7.85</td>
<td>5.81</td>
<td>2.04</td>
<td>74.01%</td>
</tr>
<tr>
<td>Condenser</td>
<td>5.58</td>
<td>1.44</td>
<td>4.14</td>
<td>25.80%</td>
</tr>
<tr>
<td>Evaporator</td>
<td>3.04</td>
<td>1.74</td>
<td>1.3</td>
<td>57.23%</td>
</tr>
</tbody>
</table>

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

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

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

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

### 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


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