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Research Paper

Comparative Analysis of Ejector Refrigeration System Powered with Engine Exhaust Heat using R134a and R245fa

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Abstract

An ejector refrigeration (ER) system using exhaust waste heat of a heavy vehicle engine is investigated. A program is developed using engineering equation solver software and it is used to make the calculations of the system. The system is taking all the efficiencies of system's components into account. Refrigerants R134a and R245fa are used for the comparative simulation of the system. The pressure at the exit of the pump is varied from 6 to 14 MPa and 3 to 10 MPa for R134a and R245fa, respectively. It can be concluded that COP (coefficient of performance) of the system gradually increases with the increase in pump exit pressure. Results show that, the performance of the system would be higher if R245fa is preferred rather than R134a with the given operating conditions.

Keywords: COP, ejector refrigeration, heavy vehicle, air conditioning.

1. INTRODUCTION

In recent years, air conditioning application is rapidly growing in automotive industry due to the increased human thermal comfort need. Required energy is supplied from the engine. Thus, it causes additional fuel consumption. On the other hand, almost 70% of the fuel energy is released to the atmosphere as a waste heat [1]. Waste heat driven ejector refrigeration system is one of the best promising way to obtain air conditioning. Therefore, many researchers are involved in searching the ways to improve the cooling performance of ER systems

Dong et al. performed an experimental analysis of ejector refrigeration system using low temperature heat sources. Results proved that, ejector refrigeration can successfully be carried out at the generator temperatures between 40 °C and 70 °C. Study also showed that lower temperatures give higher COP [2].

Chen et al. carried out a theoretical study on two stage compression cooling cycle using R717 as working fluid. First stage is realized by a mechanical compressor and second one is realized by an ejector. The effect of pressure and condensing/evaporating temperatures on the system performance is inspected. Results revealed that about 34.5% less power consumption can be obtained with the given two stage cycle as a substitute for vapor compression cycle [3].

Ünal et al. performed experimental and theoretical optimization study on two phase ejector refrigeration system for bus air conditioning using R134a as refrigerant. The influence of condensation and evaporation is inspected. According to results, 55% and 4% of heat transfer surface area can be reduced for evaporator and condenser, respectively [4].

Khaliq introduced a refrigeration system which is a combination of ejector expansion Joule–Thomson cooling cycle and ejector–absorption refrigeration cycle powered by waste heat [5].

Zhang et al. presented a new electronic expansion valve based ejector for domestic cooling applications. In order to determine the optimum geometric parameters of the model computational fluid dynamic simulations (CFD) were applied to the proposed cycle. At optimum conditions, area ratio and length of the fixed area mixing room, which are the two critical parameters, were determined as 7.29 and 5 mm, respectively [6].

In this paper, ejector refrigeration system is studied at moderate generator temperatures at the ranges of 130–210 °C. utilizing heavy vehicle waste exhaust gases. Wet refrigerant R134a and dry refrigerant R245fa are selected as working fluids. Both refrigerants have 0 ODP (ozone depletion potential), low toxicity and high resistance to flame. Ejector inlet temperature and pressure are selected so that vapor-liqu-

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id two-phase flow could not originate in the ejector.

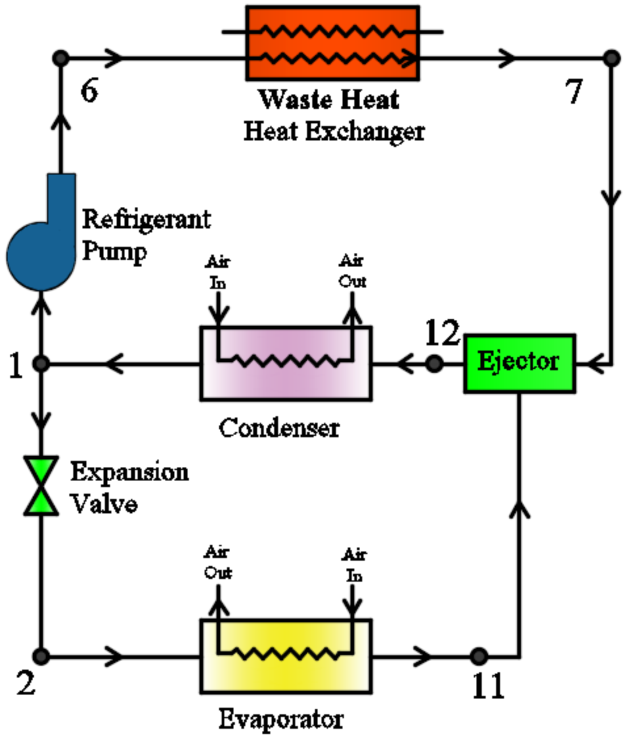

Figure 1. Schematical presentation of ejector refrigeration system

2. EJECTOR REFRIGERATION SYSTEM

Ejector refrigeration system and the ejector itself are schematically drawn in figure 1 and 2, respectively. The system cycle is demonstrated in a P-h diagram in figure 3. Refrigerant is pressurized from condensation pressure at point 1 to a pressure of P_6 which is higher than critical pressure. In case of isentropic compression, the state obtained after the pump is point 5. In the generator, refrigerant is heated up to temperature T_7 below the exhaust gas inlet temperature. High pressure and heated refrigerant enters the ejector's nozzle and leaves it at the evaporation pressure P_3 at point 9 (P_3= P_9). If isentropic expansion is assumed in the nozzle of the ejector, point 8 would be reached. This vapor at point 9 and the vapor leaving the evaporator at point 11 are mixed at point 10 and then elevated to the condenser pressure P_{14} at point 12 (P_{14}= P_{12}) in the diffuser. Assuming isentropic compression in the ejector's diffuser, state 13 would be reached. After condensation at point 1, certain portion of the refrigerant enters into the expansion valve and then into the evaporator to obtain cooling at the temperature of evaporator between the points 2 and 11. Rest of the refrigerant that leaves the condenser enters into the pump and the cycle is repeated. The power and the refrigeration cycles are investigated with the thermodynamic analysis below. Refrigerant is heated using the heat exchanger which transfer heat from engine exhaust waste gas.

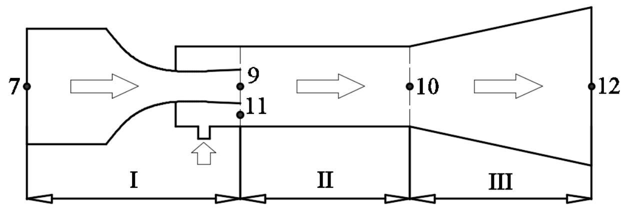

Figure 2. Ejector, I: Nozzle, II: Mixing Room, III: Diffuser.

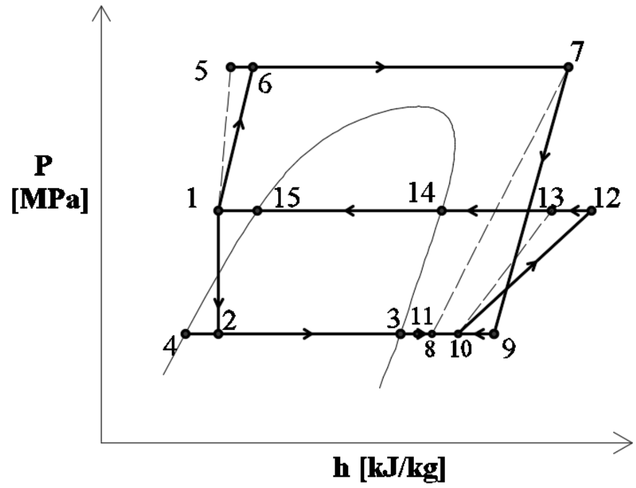

Figure 3. Pressure – enthalpy diagram of ejector refrigeration system

2.1 Thermodynamic Analysis

In air-conditioners, the main parameter that determines the evaporation temperature is the inlet temperature of air to be cooled. Working fluid temperature T_2 can be assumed to be 10-15 °C below air inlet temperature that is to be cooled dependent on the evaporator. By air-conditioning, dehumidifying must also be taken into consideration. T_2 is assigned taking these factors into account. Moreover, condensation temperature T_{15} is also given dependent upon inlet air temperature of the condenser.

For the analysis, isentropic efficiency of the pump η_{pi}, total mechanical, coupling and electromotor efficiencies of the pump η_{pme} must also be supplied. All those efficiencies are assumed as constant. Besides them, electricity production efficiency η_e from heat energy is needed. In the ejector, primary nozzle isentropic efficiency η_n and diffuser isentropic efficiency η_d are also required. It is also necessary to give the value of the mixing efficiency η_m in the ejector.

Table 1. Ejector and pump isentropic efficiency values

Ejector Efficiencies			Pump Isentropic Efficiency
η_n	η_m	η_d	η_{pi}
0.90	0.90	0.875	0.825

In different works, efficiencies of vapor ejectors are investigated [7-14]. Different primary nozzle isentropic efficiencies, diffuser isentropic efficiencies and mixing efficiencies are investigated and mean values of them are determined and given in table 1. Pump isentropic efficiencies [11,14-17] are also given in this table.

Condenser and evaporator pressures are obtained using condenser and evaporator temperatures that are given, Be-

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cause temperature differences for subcooling ΔT_{SC} and superheating ΔT_{ST} are given, condenser exit temperature T_6 and evaporator exit temperature T_{11} can be determined. After the pump, point 5 is stated for isentropic compression. This point can be determined from:

$$s_5 = s_1 \tag{1}$$

$$P_5 = P_6 \tag{2}$$

Pressure P6 is given as parameter for the calculations.

Afterwards, one can determine the real point 6 from the isentropic pump efficiency definition:

$$\eta_{pi} = \frac{h_5 - h_1}{h_6 - h_1} \tag{3}$$

Point 7 can be determined using the temperature after the heater. Heated and pressurized fluid at point 7 enters the ejector's nozzle. The refrigerant vapour after the evaporator at point 11 enters into the ejector at the suction part. These two streams are mixed at point 10 and then the refrigerant leaves the ejector at point 12 which is the inlet condition of the condenser.

At the nozzle's exit, pressure is assumed to the evaporator pressure:

$$P_8 = P_9 = P_{11} \tag{4}$$

The isentropic nozzle efficiency η_n is defined as follows:

$$\eta_n = \frac{h_7 - h_9}{h_7 - h_8} \tag{5}$$

h_{13} is the enthalpy at the exit of the nozzle in case of isentropic process. Therefore,

$$s_8 = s_7 \tag{6}$$

From these equations, one can find point 9, where the velocity is determined from energy equation between point 7 and 9 with the neglection of the kinetic energy at the nozzle inlet at point 7:

$$u_9 = \sqrt{2(h_7 - h_9)} \tag{7}$$

Energy equation at points 7, 11 and 10 yields:

$$h_7 + w \cdot h_{11} = (1+w)(h_{10} + \frac{u_{10}^2}{2}) \tag{8}$$

Here, the entrainment ratio is given by:

$$w = \frac{\dot{M}_{11}}{\dot{M}_7} \tag{9}$$

$\dot{M}_7$ and $\dot{M}_{11}$ are primary fluid's mass flow rates at point 7 from the heat exchanger and secondary fluid's at point 11 from the evaporator. Using the momentum equation between nozzle exit and the end of the mixing region (point 10), one obtains the following equation with introducing mixing efficiency η_m [14]:

$$u_{10} = \frac{u_9}{1+w}\sqrt{\eta_m} \tag{10}$$

In the mixing region, constant pressure and constant cross-sectional area are assumed. Isentropic diffuser efficiency is defined as follows:

$$\eta_d = \frac{h_{13} - h_{10}}{h_{12} - h_{10}} \tag{11}$$

Here, point 13 is isentropic compression point at the end of the diffuser. It can be found from:

$$s_{13} = s_{10} \tag{12}$$

$$P_{12} = P_{13} \tag{13}$$

From the energy equation between the points 10 and 12, it follows:

$$h_{12} = h_{10} + \frac{u_{10}^2}{2} \tag{14}$$

Expansion from 1 to 2 is assumed to occur at constant enthalpy and therefore, the following expression can be written:

$$h_2 = h_1 \tag{15}$$

Because superheating at the evaporator exit is given, point 11 can be determined from the given evaporator temperature.

$$COP = w\frac{h_{11} - h_2}{h_4 - h_6} \tag{16}$$

3. RESULTS AND DISCUSSION

In mobile systems, summer air-conditioning indoor and outdoor design temperatures are taken as 25 °C and 35 °C, respectively [19, 20]. Evaporation temperature of refrigerant is selected as 10 °C to compensate for the dehumidification duty of the air-conditioning unit. Moreover, condensation temperature of refrigerant is assumed to be 45 °C which would be appropriate under summer outdoor design temperature. On the other hand, degree of subcooling and superheating are taken as 1 °C.

In figures 4 and 5, variation of COP values with temperature are shown using R134a and R245fa as the working fluid, respectively. Efficiencies from literature are utilized for the calculations. Parameter in these figures is the exit pressure of the pump. It is seen that increasing pressure results in increasing COP values for both refrigerant. Besides, COP values decrease slowly with increasing temperature for R134a.

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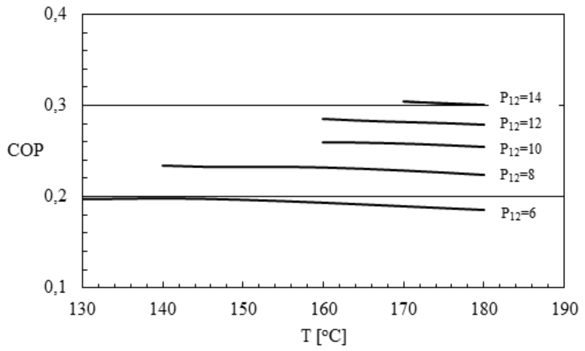

Figure 4. Variation of COP with temperature for different pressures for the refrigerant R134a

Results obtained using R245fa as the working fluid are similar to those obtained using R134a. Coefficient of performance values are a bit greater than those obtained utilizing R134a at the same operating conditions. However, unlike R134a, COP values tends to increase with increasing temperature at high pressures for refrigerant R245fa.

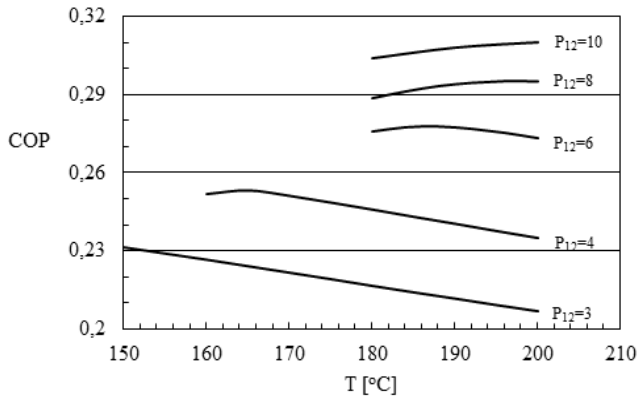

Figure 5. Variation of COP with temperature for different pressures for the refrigerant R245fa

Waste exhaust gas temperatures of heavy-duty diesel engines can be assumed in the range of 500-600 °C and corresponding waste heat energy amounts to till 400 kW [18, 21-23]. It is seen that, modelled and simulated ejector refrigeration system can be used for air-conditioning of heavy vehicles with the utilization of engine's waste heat energy in exhaust gases even at low pump exit pressures and high temperatures.

4. CONCLUSION

Ejector refrigeration system is studied utilizing exhaust waste heat source for heavy vehicles air-conditioning. In addition to primary nozzle isentropic efficiencies, diffuser isentropic efficiencies and mixing efficiencies of the ejector; isentropic efficiency of the pump, total mechanical, coupling and electromotor efficiencies of the pump are taken into account for the analysis, utilizing different works in literature. Pump exit pressure is selected as parameter. The following conclusions can be drawn from this study:

- COP values increase up to 10 MPa and 14 MPa pressure for both R245fa and R134a, respectively.
- For R245fa, coefficient of performance values are a bit greater than those obtained utilizing R134a at the same operating condition.
- At low pump exit pressures, COP values are decreased after a certain temperature for both refrigerants.
- COP values tends to increase with increasing temperature at high pump exit pressure.

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