M5-005 Study of an Ejector Refrigeration Cycle Implemented in Automobile Systems

C. Meng[1, 2], S. Chan [1, 2], I M. Astina[3], P. S. Darmanto[3]

[1] Department of Mechanical and Industrial Engineering, Institute of Technology of Cambodia, Cambodia
[2]Graduate student of Institut Teknologi Bandung, Indonesia

E-mail: mengchamnan@yahoo.com
[3] Faculty of Mechanical and Aerospace Engineering, Institut Teknologi Bandung, Jalan Ganesha 10, Bandung 40132 West Java, Indonesia

ABSTRACT

Conventional compression-refrigeration systems used in automobiles as air-conditioning system directly consume high grade mechanical energy of engine; therefore their operations contribute to higher fuel consumption or CO_2 generation. This causes negative impact on environment and expenses more costly. With compact size, ejector refrigeration system would be good alternative to create cooling effect in automobile by utilizing waste heats of engine. Even though the ejector system requires mechanical pump, its energy consumption is smaller than that required by compressor of vapor compression system. This study presents the theoretical analysis of ejector-refrigeration system performance with various environmental friendly refrigerants, HCs and HFCs, under the operating condition ranges suitable for automobiles cooling application. Engine mechanical energy can be saved ranged from 1.10 to 1.97 kW, depending on working fluids used. The saving is estimated to be equivalent to the decrease in fuel consumption of 15,894,040 liters annually which cost approximately Rp 71,523,179,067, in case all 10,667 tourist buses operated in Indonesia implement ejector system. Among the refrigerants studied, propane performs the highest performance.

Keywords: ejector refrigeration, cooling water heat, exhausts gas heat, generator, refrigerant, COP

Nomenclature

A	area, m^2
Ar	area ratio (A_{ca}/A_t)
LHV	lower heating value, $kJ \cdot kg^{-1}$
COP	coefficient of performance
D	diameter, m
Es	energy supply
Er	energy require
HSD	high speed diesel
$\dot{m}$	mass flow rate, $kg \cdot s^{-1}$
P	pressure, Pa
Rp	Indonesia currency
Q	heat rate, kW
T	temperature, ^oC

Greek letters

η	efficiency

Subscripts

ca	constant area section
c	condenser
e	evaporator
g	vapor generator
is	isentropic process
m	mixing
p	primary fluid
pm	primary flow at section A-A
s	secondary fluid
sm	secondary flow at section A-A
t	nozzle throat

1. Introduction

Generally, conventional compression refrigeration cycles are popular for air conditioners to get a high efficiency and compactness. The compressor is required for a large power to operate in the system. Compressor in automobile is directly driven by engine. Therefore, many researchers have greatly emphasized about this problem and development of non-mechanical refrigeration systems using geothermal, solar, or heat-waste as energy sources. Absorption system is the one type of non-mechanical refrigeration system. This system needs more space so that it is not suitable in applying for automobiles. More space and high initial investment and maintenance are rather specialized in specific application.

Ejector refrigeration system is also classified into non-mechanical refrigeration system. This system has advantage, i.e., more compact and driven by heat source. It should be notified that the cooling water from engine entering into the radiator is approximately 90^{o}C which is good enough to activate the ejector cycle. In case the heat supply is not enough, the exhaust gas heat which may reach 350^{o}C to 400^{o}C can be used.

Recently, Chan et al. [1] do the feasibility study on the ejector-bases air conditioning system using natural refrigerants. They used many refrigerants to simulate in the hybrid ejector system. According to the theoretical analysis results, propane has the highest performance, especially at higher generator temperature. However the application of the propane as refrigerant for automobile air conditioning system should be studied more carefully due to the flammability property. Salim [2] considered the thermally activated mobile ejector refrigeration system analysis. This investigation was focused for an automotive system which utilizes R-134a. In addition, the utilization of engine seems to be more practical and offers many advantages since the availability of exhaust gas and cooling water as heat source. Chen and Sun [3] conducted an experimental study on performance characteristics of a steam-ejector refrigeration system. They investigate the performance characteristics of the steam-ejector refrigeration cycle. A relatively small scale system was built and tested at various operating conditions. A 1-D analysis of ejector performance was developed by Huang et al. [4] to analyze entrance flow at choking condition. The constant-pressure mixing occurs inside the constant-area section of the ejector was assumed. Within the range of the variation of ejector area ratio and ejector entrainment ratio conducted in the experiment, the 1-D model proposed by Huang et al. using the empirical coefficients can accurately predict the performance of the ejectors.

Total equivalent warming impact (TEWI) is used to estimate how a system contributes to the green house effect [5]. TEWI of an automotive air-conditioning systems may be divided into two parts, i.e., a direct effect which results from leakage of refrigerant and an indirect effect inducted by the production of other gases, such as carbon dioxide. The absence of the vapor compressor system can reduce both the direct and indirect effects on TEWI. Waste-heat powered ejector- refrigeration systems are more interesting due to the freedom from vibration, high reliability, and low operation and maintenance costs [6].

Present study is to investigate theoretically the performance of an ejector refrigeration system using various natural refrigerants as working fluids with three different cooling loads applied on automotive. Theoretical analysis method is used for design and optimization of the system with respect to the each refrigerants and operating conditions. Furthermore, energy saving evaluation and comparison between supplied and required waste heat sources in the ejector cycle for various refrigerants will be presented in this paper.

2. Ejector Refrigeration Cycle

Ejector refrigeration cycle that was conducted in this study is shown in Fig. 1. This cycle utilizes an ejector and a pump for maintaining the pressure difference between evaporator and condenser. A combination system between power cycle (6)-(1)-(2)-(3)-(A)-(4)-(5)-(6) and a refrigeration cycle (A)-(5)-(6)-(7)-(8)-(A) is called as an ejector refrigeration system. Power cycle can be run by several kinds of heat to boil the refrigerants in generator, such as solar energy, electrical heater, waste heat and combustion gas. For automotive application, generator can be activated by exhaust gas and/or engine water coolant. The waste heats from engine are an excellent waste source [3].

Chan et al. [1], as shown in Figure 2, described that the high-pressure vapor known as a primary fluid, from generator (1-2), is expanded through converging-diverging nozzle (2-3) where its pressure energy is converted into kinetic energy. The fluid flow leaving the nozzle with the very low pressure is in supersonic velocity range. Due to the lower pressure inside of suction chamber, the refrigerant from evaporator could be sucked into suction chamber. In the constant-area section (3-4), the two fluids start to mix with high-speed in a mixing section (A) where a normal shock may happen, but it depends on the flow conditions and the physical design of ejector. Finally, the pressure is increased at diffuser section.

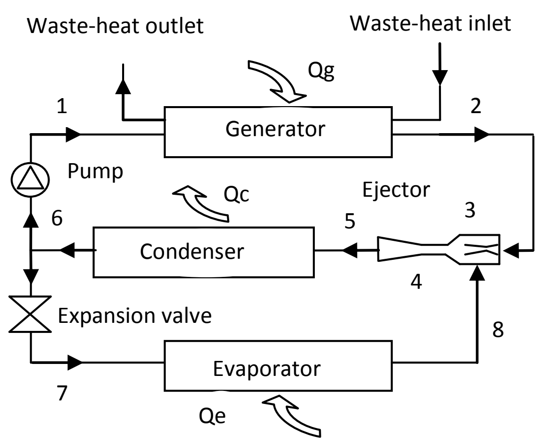

Fig. 1: Schematic of an ejector refrigeration cycle

Many researchers [1, 3, 4] have adopted the one-dimensional constant-area-ejector flow theory, in which the following assumptions were made in the ejector analysis:

1. Compressible one dimensional flow and steady state.
2. No heat exchange with the surroundings.
3. Secondary flow pressure drop and momentum at point (3) mentioned in Fig. 2, are very small compared to the primary flow.
4. When mixing occurs, there is no pressure drop between sections 3 and A.
5. No change in fluid characteristics at the diffuser throat.
6. The velocities at inlet of primary nozzle, suction chamber and the outlet of diffuser can be neglected.

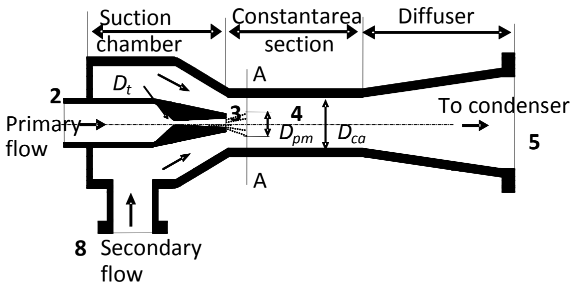

Fig. 2: Schematic diagram of ejector [1]

Calculation methods and simulation are adopted from Chan et al. [1] to solve the main equations, such as mass, energy, and momentum balance equations. Varying the operating conditions and working fluids are important to get the best performance of the cycle. The various generator temperatures and working fluids are set during simulation process of the ejector.

3. Simulation Results and Discussion

More completely data of cooling capacity of an automobile are represented in Ref [7, 8, 9]. There are three different cooling loads which correspond to the three different sizes of automobile as summarized in Table 1.

Chan et al. [1] were adopted the shock cycle model to analyze the ejector performance. The comparative results between COP and *Ar* are shown by assuming the value of coefficients η_p, η_s and η_m are 0.95, 0.83, and 0.87, respectively, with 2 kW of generator heat input. The ejector cycle using propane performs 72% of COP at the following operating conditions, i.e., 90^{o}C of generator temperature, 30^{o}C of condenser temperature, and 10^{o}C of evaporator temperature. Using the developed analysis method above in which the cycle operating with different fluids such as 1,1,1,2-Tetrafluoroethane (R-134a), Isobutane (R-600a), Butane (R-600), Propane (R-290), and R-142b at three values of cooling load are conducted in the present study as simulation parameters.

The results shown in Figs. 3 and 4 are obtained for the condenser temperature at T_c = 35^{o}C with two values of evaporator temperature (T_e), 7.5^{o}C and 12.5^{o}C, and at three values of the cooling load (Q_e), 14 kW, 20.9 kW, and 32.53 kW. As shown in Figs. 3 and 4, the simulation results present that when evaporator temperature T_e and generator temperature T_g increase the area ratio (*Ar*) and coefficient of performance (COP) also increase. On the other hand, the values of *Ar* and COP do not depend on the air cooling load Q_e whatever changing from 14 kW to 32.53 kW. Propane, as seen in the figures, can perform well simulation results. Its COP can reach till 52.50% while butane's COP is only up to 36% at T_e = 12.5^{o}C, T_g = 80^{o}C, and T_c = 35^{o}C.

Table 1: Specification of three different automobile sizes

Model 1	Model	FDG6700 (mini bus)
	L x W x H (mm)	6,990 x 2,050/2,125 x 2,680
	Seats	18 to 28
	Engine Model	CY4102BZQ
	Rated Power (PS)	120 (88.25 kW)
	Cooling load (kcal/h)	12,000 (14 kW)
Model 2	Model	JINQ6800 (public bus)
	L x W x H (mm)	7,990 x 2,450 x 3,200
	Seats	15 to 46
	Engine Model	CA4DF2-13
	Rated Power (kW)	96.00
	Cooling load (kcal/h)	18,000 (20.9 kW)
Model 3	Model	LCK6115H CNG Bus
	L x W x H (mm)	11,505 x 2,480 x 3,620
	Seats	45+1+1
	Engine Model	CG-250
	Rated Power (kW)	188.00
	Cooling load (kcal/h)	28,000 (32.53 kW)

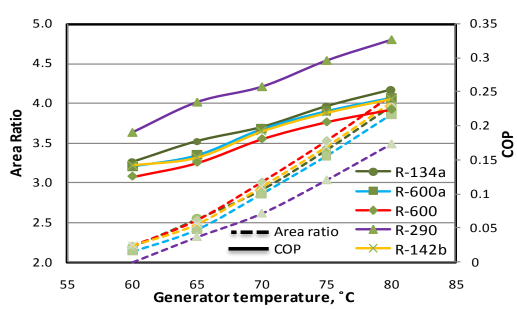

Fig. 3 Comparisons of *Ar* and COP at $T_c = 35^oC$ and $T_e = 7.5^oC$

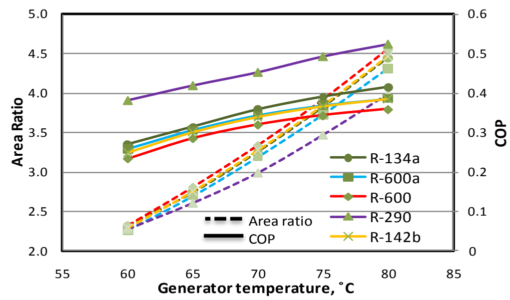

Fig. 4 Comparisons of *Ar* and COP at T_c = 35°C and T_e = 12.5°C

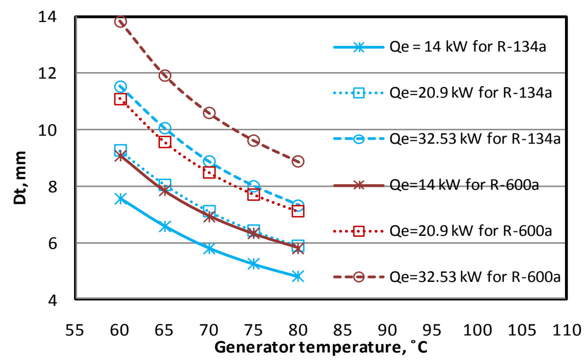

Fig. 5 Comparisons of D_t for R-134a and R-600a at T_c =35°C, T_e = 12.5°C, and various values of cooling load

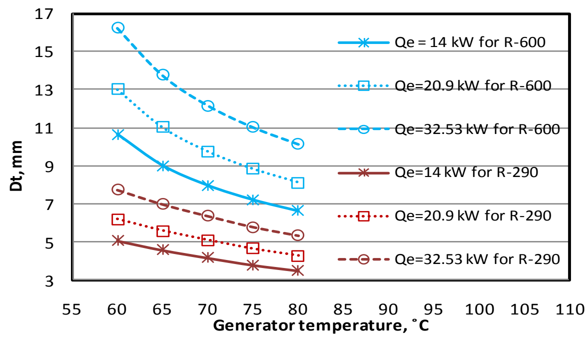

Fig. 6 Comparisons of D_t for R-600 and R-290 at T_c =35°C, T_e = 12.5°C, and various values of cooling load

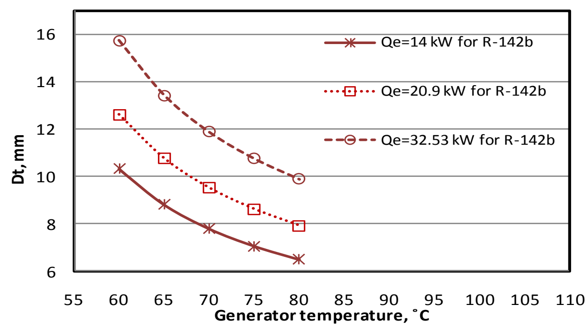

Fig. 7 Comparisons of D_t for R-142b at T_c =35°C, T_e = 12.5°C, and various values of cooling load

Figs. 5-7 show the performance characteristics at different throat diameters D_t for different fluids. As can be seen in the figures, when T_g decreases, D_t smoothly increases. Decreasing T_g causes decreasing of the vapor pressure and fluid flow rate of generator. The values of Q_e can mainly affect on the throat diameters D_t. Actually, an automobile has limited space for installing systems. It is the reason why correct D_t estimation value is very important for ejector system designer. As the results, the maximum D_t for butane is 16.24 mm at T_c = 35°C, T_e = 12.5°C, and T_g = 60°C for Q_e = 32.53 kW, while D_t for propane is 7.73 mm at the same conditions.

In general, the input energy from Diesel engine is distributed 38%, 24%, 35%, and 3% to the major areas of exhaust, coolant, output power, and radiation heat loss respectively. Three kinds of automobile engine output power are selected, i.e., 88.25 kW, 96 kW, 188 kW. Based on the above assumptions, the possibility of using energy-require (*Er*) to drive the generator and the energy-supply (*Es*) from heat loss can be computed. Figs. 8-13 graphically show the two different energy-supplies *Es* (water coolant and exhaust gas) comparing to the energy required *Er* for operating this proposed ejector air conditioner. Totally, the requirement energy *Er* from various fluids depends on the T_g and T_e. When its temperature decreases the *Er* increases. However, not only *Er* from exhaust gas heat can be used to for providing the high T_g, but also from cooling water heat. However, when T_g is smaller than 70°C and Q_e = 20.9 kW, cooling water heat *Es* could only supply 65.83 kW. As shown in Fig. 12 this energy is not enough to supply the system for all fluids except propane. In this condition the system requires additional heating source. It should be noted that utilization of cooling water as a heat resource is limited especially at start up condition because the temperature is still low [3].

Salim [3] has reported the compressor in a conventional air-conditioning system using R-134a requires 2.9-5.3 kW input power from engine. These values are equivalent to 5-30% of available engine power. An ejector system needs centrifugal pump only and low power consumption as around 0.08-0.2 kW.

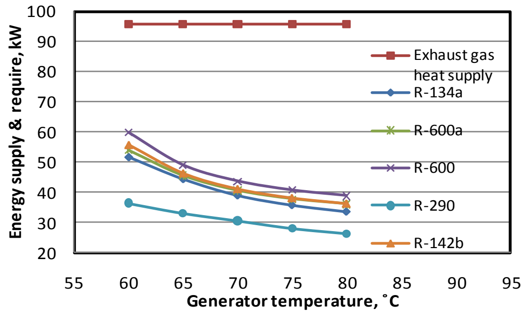

Fig. 8 Comparisons of *Es* (for exhaust gas heat) and *Er* at Tc = 35°C, Te = 12.5°C, and Qe = 14 kW for various refrigerants

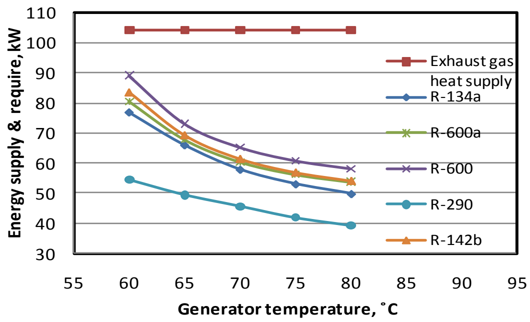

Fig. 9 Comparisons of *Es* (for exhaust gas heat) and *Er* at Tc = 35°C, Te = 12.5°C, and Qe = 20.9 kW for various refrigerants

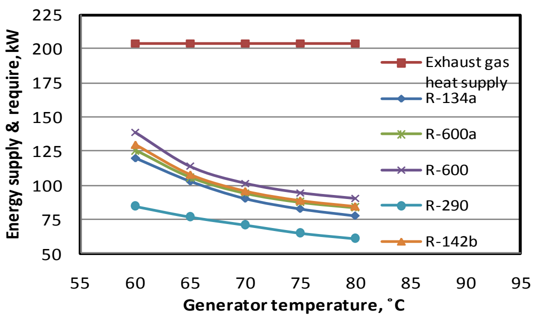

Fig. 10 Comparisons of *Es* (for exhaust gas heat) and *Er* at Tc = 35°C, Te = 12.5°C, and Qe = 32.53 kW for various refrigerants

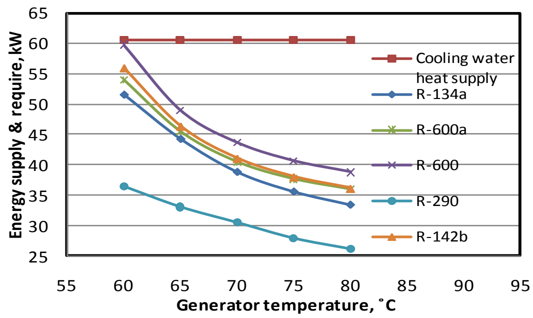

Fig. 11 Comparisons of *Es* (for cooling water heat) and *Er* at Tc = 35°C, Te = 12.5°C, and Qe = 14 kW for various refrigerants

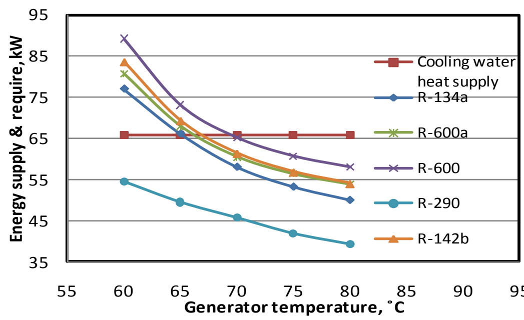

Fig. 12 Comparisons of *Es* (for cooling water heat) and *Er* at Tc = 35°C, Te = 12.5°C, and Qe = 20.9 kW for various refrigerants

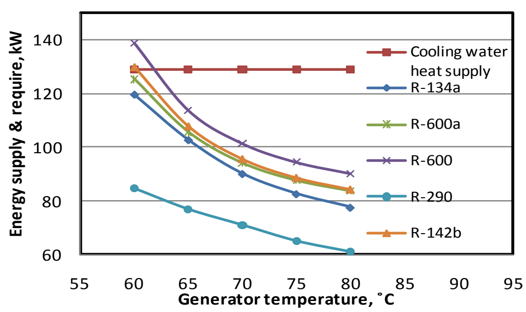

Fig. 13 Comparisons of *Es* (for cooling water heat) and *Er* at Tc = 35°C, Te = 12.5°C, and Qe = 32.53 kW for various refrigerants

The evaluations of compressor power in a conventional vapor compressor cycle have also conducted in this paper by modifying C++ program code of the compressor isentropic efficiency calculation in accordance with Brunin et al. [11] as Eg. (1):

$$\eta_{is} = 0.874 - 0.0135\frac{P_c}{P_e} \qquad (1)$$

According to the simulation results as presented in Figs. 14-16, for all working fluids with different air cooling loads, this proposed system can save the energy more than 1 kW. The increasing of T_g can save more energy due to T_g strongly affects on the vapor pressure and fluid flow rate in the generator. It means that when T_g increases, the vapor pressure and fluid flow rate also increase, while pump requires more power to activate the cycle. For R-290 and R-134 perform smaller energy saving than the other fluids at the same conditions. Not only the increasing of T_g affects on the energy saving, but it is also affected by Q_e and T_e change. Clearly, butane can provide the highest energy saving 1.97 kW at T_c = 35^{o}C, T_e = 12.5^{o}C, and for Q_e = 20.9 kW. At the same conditions, only 1.10 kW is provided by propane as energy saving. This value is the minimum among refrigerants being studied in this research as shown in Figs 14-16.

Currently, the HSD fuel saving per year can be figured out by using Eqs. (2) and (3). The energy saving is shown in Figs. 14-16 for Q_e = 32.53 kW at T_g = 70 ^{o}C, T_c = 35^{o}C, and T_e = 12.5^{o}C respectively. The engine efficiency, η_{engine} = 35%, is assumed as constant value during simulation. The lower heating value of fuel (LHV_{fuel}) is equal to 42,300 kJ·kg^{-1}, as refered to the Eurostat [10]. According to the statistical of transportation yearbook issued by the ministry of transportation of Indonesia in 2008 [12], there are about 10,667 tourist buses operated in Indonesia. In the HSD and financial saving calculation all buses are assumed to be operated of around 10 hours per day.

$$Q_{fuel} = \frac{Energy_{saving}}{\eta_{engine}} \qquad (2)$$

$$\dot{m}_{fuel} = \frac{Q_{fuel}}{LHV_{fuel}} \qquad (3)$$

As can be seen in Fig. 17, million liters of HSD fuel can be saved per year if the proposed ejector refrigeration cycle is applied, especially for using R-600 as a working fluid. Annually, HSD fuel is saved 20,605,447 liters by using R-600; 19,836,920 liters for R-142b; 19,680,986 liters with R-600a; 16,695,981 liters for R-134a; and 15,894,039 liters if R-290 is used. In average the fuel that can be saved is more than 1,500 liters per year per bus.

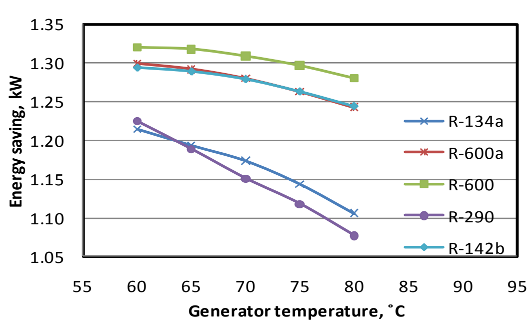

Fig. 14 Energy saving at T_c = 35^{o}C, T_e = 12.5^{o}C, and Q_e = 14 kW for various refrigerants

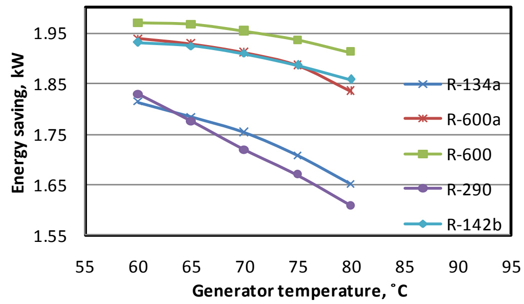

Fig. 15 Energy saving at T_c = 35°C, T_e = 12.5°C, and Q_e = 20.9 kW for various refrigerants

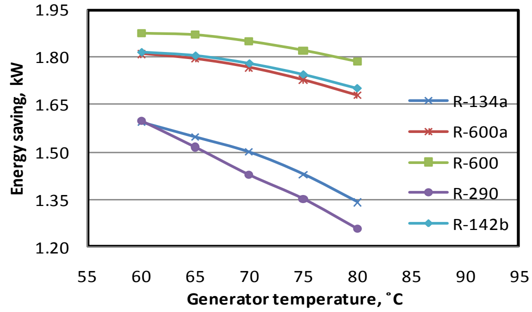

Fig. 16 Energy saving at T_c = 35°C, T_e = 12.5°C, and Q_e = 32.53 kW for various refrigerants

According to the recent price of the HSD fuel in Indonesia, the subsidized price of HSD fuel is Rp 4,500. The financial saving per year can be estimated and the result is shown in Fig. 18 for the same above operating conditions. Total tourist buses in Indonesia could save in average Rp 83,442,038,200 per year at T_g = 70 °C, T_c = 35°C, and T_e = 12.5°C for Q_e = 32.53 kW. In the case of ejector cycle, R-290 can perform highest COP. However, this refrigerant requires highest value of pump input-power to activate the cycle. In contrast, the lowest performance is provided by using R-600 for ejector cycle due to the lowest input-power of the pump to activate the cycle.

In contrary, for conventional system, propane has the lowest value of COP than the other refrigerants due to the highest compressor power input. The different input-power between compressor and pump is the advantage of the application of ejector system compared to the conventional one. Energy saving between both cycles affects on the HSD fuel and financial saving.

In the case of conventional vapor compression system, energy saving for R-290 is smaller than that of other refrigerants. Consequently, the HSD fuel and financial that could be saved per year are also lower than the other refrigerants. Meanwhile, R-600 can provide the highest COP and the smallest input-power for the compressor to drive the cycle. It means that the conventional system using R-600 consumes less HSD fuel and resulted in more cash saving.

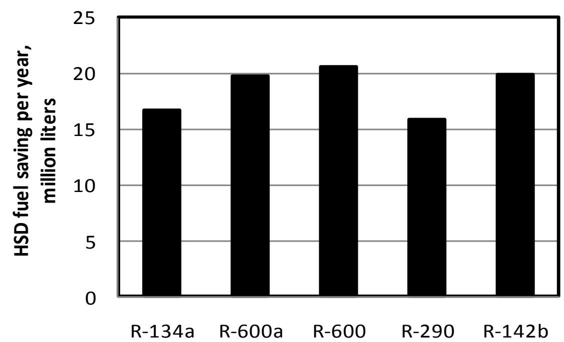

Fig. 17 HSD fuel saving per year at T_g = 70 ^{o}C, T_c = 35^{o}C, and T_e = 12.5^{o}C for Q_e = 32.53 kW

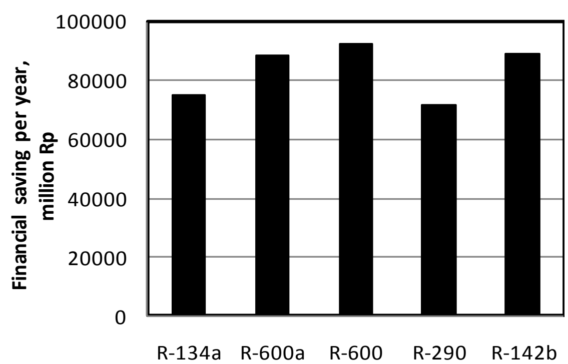

Fig. 18 Financial saving per year at T_g = 70 ^{o}C, T_c = 35^{o}C, and T_e = 12.5^{o}C for Q_e = 32.53 kW

4. Conclusion

The application of an ejector refrigeration system for automobile powered by the waste heat was studied. The results show that the highest performance is achieved when propane is used as refrigerant. For the different cooling loads at T_e = 12.5^{o}C, T_g = 80^{o}C, and T_c = 35^{o}C, the COP of ejector cycle using propane can reach 52.50%. For the same conditions, but T_e = 7.5^{o}C, its performance drops to 32.60%. In this case, propane is an excellence refrigerant comparing to other fluids such as butane where its maximum performance is 36% at the same operating conditions. Propane as refrigerant needs only 84.65 kW as maximal energy requirement for receiving 32.53 kW of cooling load. The other advantage of propane is that the ejector system can be operated using both exhaust gas waste and engine cooling water. Propane performs smaller energy saving than the other fluids such as butane. Energy saving of the propane is 1.10 kW and the butane is 1.97 kW for 20.9 kW of cooling load at T_e = 12.5^{o}C, T_g = 60 ^{o}C, and T_c = 35^{o}C.

The HSD fuel that can be saved is around 15,894,040 liters for R-290 and the maximum value is 20,605,448 liters for R-600 per year for total numbers of 10,667 tourist buses, 10 of daily operational hours. That is equivalent to the annual financial saving around Rp 71,523,179,067 and Rp 92,724,513,857 for R-290 and R-600, respectively.

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