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ANALYSIS OF EVAPORATOR EFFECTIVENESS ON 1/2 CYCLE REFRIGERATION SYSTEMS: A CASE STUDY ON LPG FUELED VEHICLES

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Graphical abstract
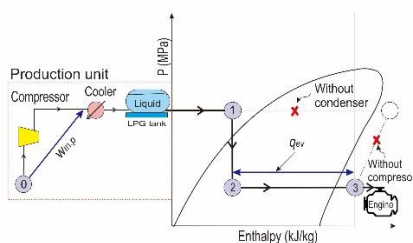
Abstract

The ½ cycle refrigeration system on LPG fueled vehicles has a significant cooling effect. However, the cooling is very dependent on the heat exchange process in the evaporator. Therefore, this paper analyses the deviation of the actual cooling curve from the ideal scenario carried out on a laboratory scale. The analytical method used is the calculation of the effectiveness of the evaporator, which compares the actual to the potential heat transfer capacity. The LPG flow rate was varied from 1-6 g/s, while the evaporation pressure ranged between 0.05, 0.10, and 0.15 MPa, which applied to compact type evaporators with dimensions of 262 x 200 mm, with a thickness of 65 mm. The research results confirm that the higher the LPG mass flow rate, the lower the heat transfer effectiveness. At the higher LPG mass flow rate, heat transfer occurs less optimally, due to incomplete evaporation of LPG in the evaporator.

Keywords: LPG fueled vehicle, ½ cycle refrigeration, COP, evaporator effectiveness

1.0 INTRODUCTION

The Air Conditioning (AC) system has, for long, been used as a supporting mechanism for a passenger car. Before 1940, adjusting the vent or opening the side glasses was the only way one could be comfortable in a car [1]. However, Packard became the first car manufacturer that complements its products with an AC system in 1940. The cooling system equipment is located in the rear trunk, unlike the current AC mechanism, which is compactly designed in the engine chamber and inside the dashboard. Then, more than half of the new cars were equipped with AC systems that could be operated easily in 1969 [2].

For many years, the AC system of passenger cars utilized a vapor compression refrigeration mechanism. The working principle of the main components is less developed, except for compressors and refrigerants due to the demands of efficiency and the environment effect [3-4]. The main components and refrigerant cycle of the vapor-compression AC system are shown in Figure 1, where 1→2 represents the isentropic compression process, 2→3 condensation process at constant pressure, 3→4 isenthalpic expansion process, and 4→1 represent evaporation process or harvesting cooling effect.

Along with development of technology, at present, the vehicle's AC system has been able to regulate automatically based on internal temperatures, but have considered the effects of solar radiation, vehicle speed, and external temperatures [6]. The mechanism of controlling the compressor on the AC system has also been developed, from manual, electric, to virtual controlling [7-8].
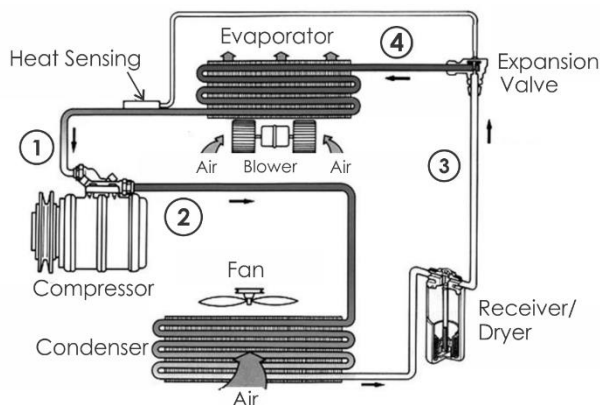
Figure 1 Basic component and refrigerant cycle of vapor-compression AC system applied to passenger cars [5]

1.1. Some Alternative Technologies to Replace Conventional AC System

Generally, the car AC system works based on the vapor compression cycle. However, due to the demands of engine efficiency and environmental issues, and absorption system has been developed for several years, for exemple, the one examined by Chandrakar [9]. Thermal energy from the exhaust gas is used in the absorption refrigeration systems. From theoretical designs, it is verified in the laboratory and road tests. This system has been utilized for long, though it only produces COP ranging from 0.8 to 0.9. It was also researched and applied to the truck cabin, where the developed absorption system produces a cooling capacity of up to 1.2 kW, with a COP ≤ 0.45 [10].

Another absorption system is developed by heat supply from the loop of the engine coolant [11]. A test was carried out by installing the adsorption chiller into a prototype of the truck cabin, producing an air temperature of 9°C with a cooling capacity estimated at 2 kW. For long, the absorption system has been perceived as a better choice for replacing the vapor compression mechanism, even though its performance is still relatively low [12-15].

In terms of material optimization, a simulation of the absorption refrigeration system with energy from the exhaust gas is developed, especially for evaporator and condenser [16]. All parts needed for absorption refrigeration systems are designed and modeled in the Pro/Engineer software and analyzed by ANSYS. The results of the simulation show that the evaporator and condenser made of aluminum alloy have a better performance than copper.

Apart from the absorption system, a compressor with external power is also developed. A biogas engine with constant rotation is used to drive an AC compressor [17]. With this system, it produces 1 Ton of Refrigeration (TR), which is equivalent to the cooling load of passenger cars. This study refers to Damrongsak [18], who removed the AC system from the car to be tested with small-sized biogas or compact modular engine. It produces a cooling

effect of 3.5 kW on the compressor rotation of 1000 rpm. However, the AC system with an external mechanical drivehas only been studied on a laboratory scale. Application to vehicles is more complicated since it requires two engines and fuel systems.

A turbo system develops a new method of driving an AC compressor to reduce engine load. Primarily, kinetic energy and exhaust gas pressure are used to rotate the turbo and the AC compressor with a magnetic gear connector [19]. One of the main advantages of this concept is that it can be easily applied for low-capacity engines. It offers better utilization of exhaust gas energy and reduces fuel consumption. Although this study is still in the proposed model, it might be applied on a special scale by the rapid advancement of technology in the thermo-mechanical field.

Although absorption refrigeration and other systems cannot yet replace the vapor compression mechanism, it is necessary to develop an alternative option for passenger vehicles that low in energy input. This because a compressed vapor AC system requires much energy serves as an engine load. The fuel consumption due to the AC system is diverse depending on technology and heat load, though almost all researchers agree that it is significant and must be reduced [20-24].

1.2. Experiences from Previous Research: Performance of 1/2 Cycle AC Systems in LPG Vehicles

In previous research, laboratory-scale tests on ½ cycle AC systems applied to LPG fueled vehicles were examined. The phase change of LPG from liquid to vapor on the fuel line requires evaporation heat. The fuel system was modified to replace the loop engine coolant in the evaporator with air. LPG absorbs heat from the air that crosses the evaporator and resulting in cold air moves to the cabin to produce a cooling effect. Numerical studies are conducted with data obtained from LPG properties as well as specific state points in preliminary research [25-26]. As a result, the ½ cycle system provides a promising cooling effect. In general, the cooling effect of more than 1.2 kW at a mass flow rate of LPG 3 g/s or higher can be harvested without using the compressor [27]. The highest COP value of this study was 6.27, which was obtained at an LPG mass flow rate of 1 g/s and evaporation pressure of 0.15 MPa.

However, the cooling effect produced is not linear, with an increase in LPG mass flow rate. Previous studies have not addressed this weakness. Moreover, previous works attribute this problem to LPG evaporation at a mass flow rate above 3 g/s, which is imperfect, as shown in Figure 2. Some LPG leaves evaporator in the form of a mixture of vapor and liquid. Also, heat exchange occurs optimally in case LPG exits the evaporator in the form of saturated or superheated vapor. Therefore, this paper analyzes heat transfer in LPG evaporators in terms of effectiveness (ϵ).
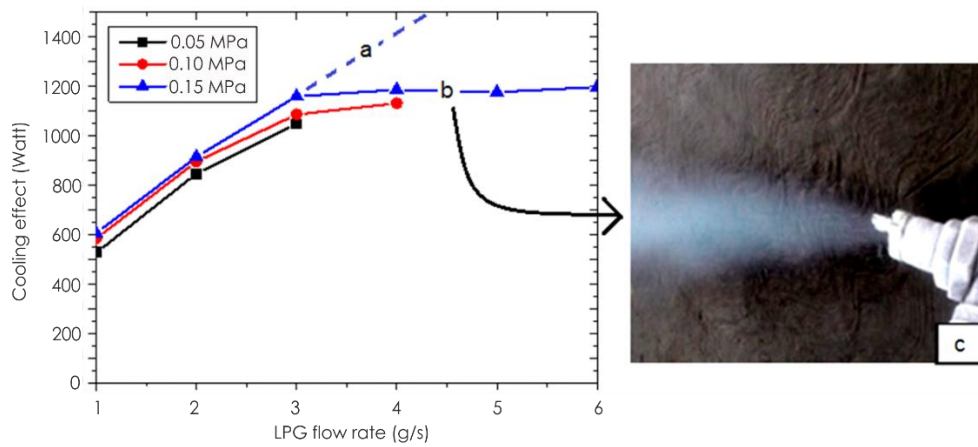
Figure 2 Problems in previous research: (a) ideal cooling effect curve, (b) cooling effect curve of test results (actual), and (c) incomplete evaporation of LPG that comes out of the evaporator [27]

2.0   METHODOLOGY

2.1   Experiment Set up and Limitation

The ½ cycle refrigeration system involves working of the compressor and condenser like a full cycle system. The cooling effect is obtained only by dropping LPG pressure on the expansion valve and harvesting the cooling effect on the evaporator. Pressurized LPG is supplied from the tank. The concept and specific state points of the ½ cycle refrigeration system are presented in Figure 3, where 1→2 represents the isenthalpic expansion, while 2→3 is the evaporation. There is no process from 3 to 1 since the LPG from point 3 is fed to the engine as fuel. 0→1 represents a process outside the fuel system in the vehicle, which describes how LPG is produced into pressurized commercial products in the tank.

In this study, LPG is maintained in the form of liquid in the tank by monitoring the pressure and temperature. Its mass flow rate at point 1 is varied at 1-6 g/s, which represents the fuel consumption of passenger vehicles [28]. Besides, evaporation pressure at 2-3 is set at 0.05, 0.10, and 0.15 MPa for each mass flow rate. Setting the mass flow rate and pressure is conducted on a specially designed expansion valve made of Teflon, which has a low thermal conductivity for more benefits. During testing, pressure and temperature in each specific state point were observed with RTD thermocouple and PSAN pressure transducer connected to the computer through the module. DAQ Master software supported by Autonics conducts the monitor screen reading.
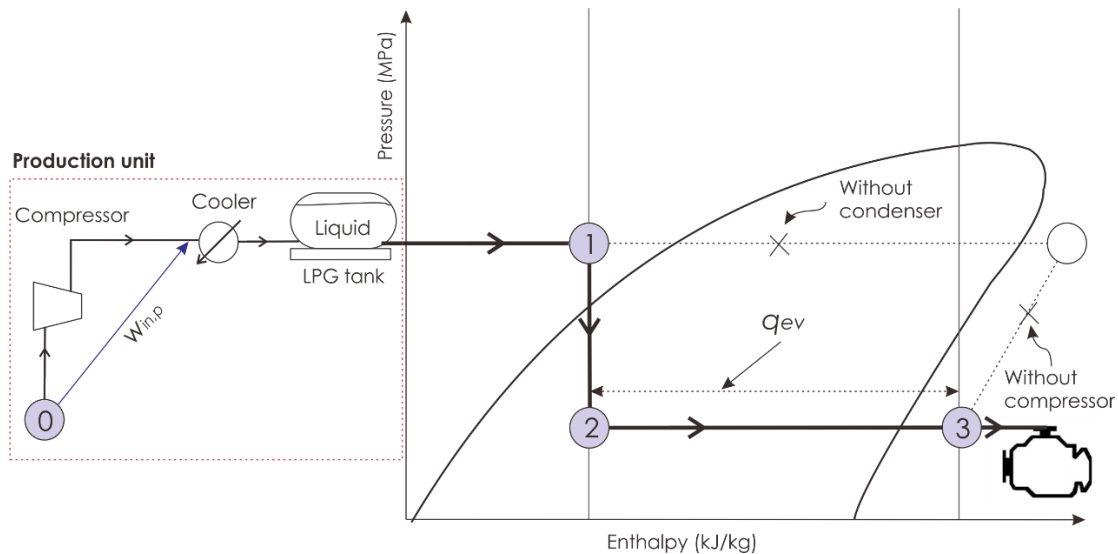

Figure 3 Concept and specific state points of the ½ cycle refrigeration system [27]

2.2   Analysis Method

In this study, LPG in the tank is in the form of pressurized liquid. For this reason, work in the system is calculated as the work of compressors to increase pressure in the production process before LPG is supplied to dispensers at a gas station. Assuming there is no heat entering and exiting the system during compression which takes place in an isentropic manner, there is no change in entropy. Figure 4 shows the boundary system and *T-s* diagram for calculating the work input.
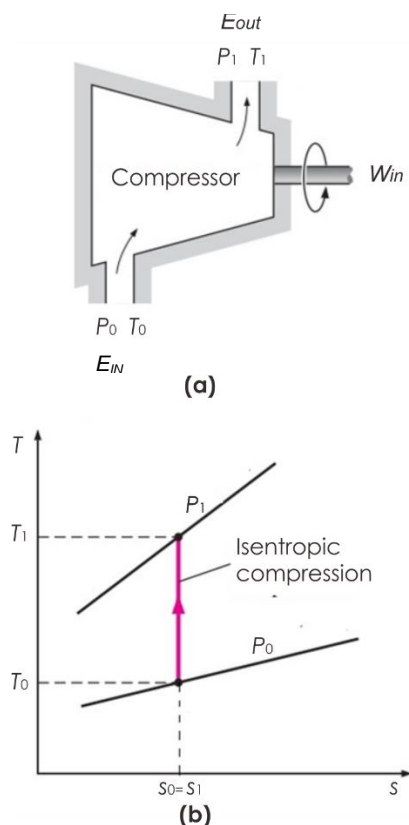

(a)

(b)

Figure 4 Isentropic compression process: (a) Boundary system and (b) *T-s* diagram

From Figure 3 and Figure 4, relationship between temperature, enthalpy and pressure on compressor work is given in the Equation (1) to Equation (4) as follows [30].

$$E_{in} = E_{out} \tag{1}$$

Assuming that kinetic energy and potential energy are ignored because change in speed is very small and there is no change in altitude, then:

$$w_{in} + h_0(P_0, T_0) = h_1(P_1, T_1) \tag{2}$$

$$w_{in} = h_1(P_1, T_1) - h_0(P_0, T_0) \tag{3}$$

Assuming that compression is isentropic ($ds = 0$) then $Tds = 0$. Thus, the work of compressors to produce pressurized liquid LPG is stated below.

$$w_{in,p} = \int_0^1 dh = h_1 - h_0 \tag{4}$$

Apart from the COP (general method for measuring the performance of an AC system), which was generally utilized to analyze cooling performance (2→3), this study uses the calculation of the effectiveness of heat exchange in the evaporator. The formula for heat transfer effectiveness is given in Equation (5) [29].

$$\epsilon = \frac{T_{1,o} - T_{1,i}}{T_{2,i} - T_{1,i}} \tag{5}$$

In this case, $T_{1,i}$ and $T_{1,o}$ are temperatures of LPG entering and leaving the evaporator, respectively. Also, $T_{2,i}$, and $T_{2,o}$ are the temperature of the air entering and exiting across the evaporator, as presented in Figure 5.
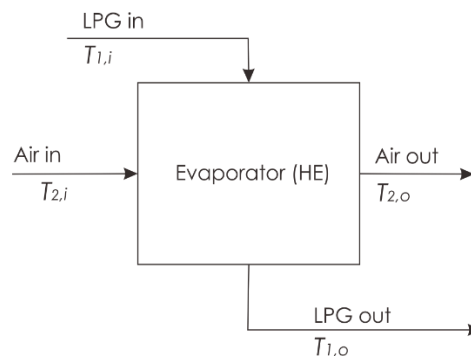

Figure 5 Sketch of heat transfer in the evaporator

3.0   RESULTS AND DISCUSSION

3.1   Cooling Effect and COP

This test results showed that the ½ cycle refrigeration system produces a cooling effect of up to 1.2 kW, with data shown in Table 1.

Table 1 Cooling effect ($\dot{Q}_{ev}$) at variations of LPG mass flow rate and evaporation pressure

LPG flow rate (g/s)	$\dot{Q}_{ev}$ (Watt) at evaporation pressure of		
	0,05 MPa	0,10 MPa	0.15 MPa
1	527	584	606
2	846	896	925
3	1048	1087	1125
4	n/a	1132	1186
5	n/a	n/a	1177
6	n/a	n/a	1198

Furthermore, the input work (w_{in}) is calculated based on the compressing LPG from 24 °C and 0 MPa to 0.7 MPa. In this case, the input produces pressurized liquid LPG ($w_{in,p}$). Assuming that the process of compression is isentropic, using NIST REFPROP, data on density, enthalpy, and entropy are obtained, as shown in Figure 6. Data on the first and second lines represent the condition of LPG entering and exiting the compressor, respectively.

Figure 6 LPG property when entering and exiting the compressor (in production unit)

Unlike the full cycle refrigeration system, where compressor works is not constant, in this study the work of the compressor is constant for all evaporated LPG mass flow rates. By Equation (5) and data from Figure 5, enthalpy LPG value when entering the compressor is 480.2 KJ/kg and enthalpy LPG out of compressor is 576.80 KJ/kg. Thus, the compressor's work is:

$$w_{in,p} = \int_0^1 dh$$

$$w_{in,p} = 576,80 - 480,2 \quad [\text{kJ/kg}]$$

$$w_{in,p} = 96,6 \text{ KJ/kg}$$

Then, by comparing the cooling effect ($\dot{Q}_{ev}$) generated by work input of compressor ($\dot{W}_{in,p}$), the Coefficient of Performance (COP) is shown in Figure 7 as follows.
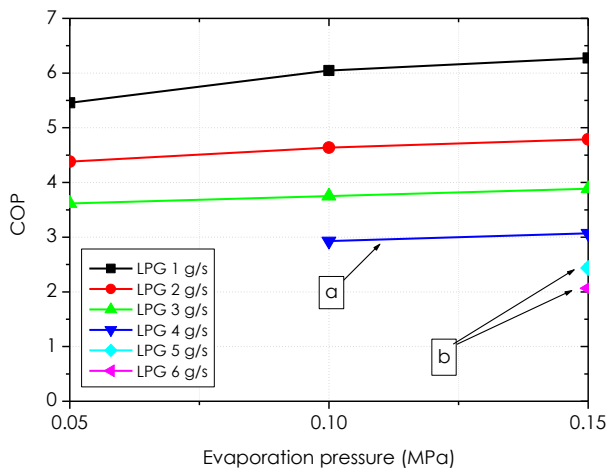

Figure 7 COP at variations of LPG mass flow rate and evaporation pressure: (a) LPG 4 g/s can only be flowed by evaporation pressures of 0.10 and 0.15 MPa, and (b) LPG 5-6 g/s can only be flowed by evaporation pressures of 0.15 MPa

From Figure 7, two interesting phenomena need to be discussed. First, increasing evaporation pressure affect COP. At a pressure of 0.15 MPa, there is a transfer of latent heat as long as LPG in the evaporator is longer than sensible heat transfer. However, the difference in COP due to variations in evaporation pressure is smaller in case the mass flow rate is enlarged, as shown in Figure 8. Second, the higher the LPG mass flow rate, the smaller the COP produced. This is attributed to the increase in the cooling effect produced ($\dot{Q}_{ev}$) not being linear with the increasing mass flow rate. Generally, increasing LPG mass flow rate also results in incomplete evaporation, as presented visually in Figure 2 [27]. Although the increase in mass flow rate increases the cooling effect ($\dot{Q}_{ev}$), it may cause a decreasing in LPG temperature when exiting the evaporator. Since enthalpy values depend on temperature, the COP tends to decrease when the LPG mass flow rate increases.
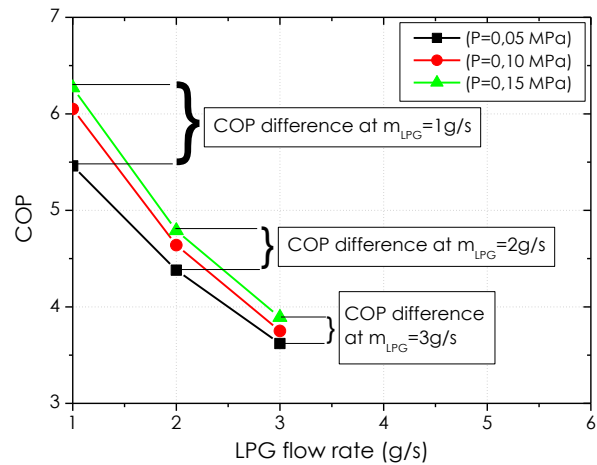

Figure 8 Effect of mass flow rate and evaporation pressure on the COP

3.2 Evaporator Effectiveness

The evaporator in LPG fuel systems is additional equipment attached to the fuel line, which serves to vaporize LPG. The heat to vaporize LPG is obtained from the air crossing fins, which are driven by electric blowers. Evaporator has dimensions of 262 x 200 mm, with a thickness of 65 mm. From $T_{1,o} - T_{1,i}$ as the difference in actual temperature and $T_{2,i} - T_{1,i}$ the difference in maximum temperature (potential), and using Eq. (1), the effectiveness of evaporator on various evaporation pressures and mass flow rates is presented in Figure 9.
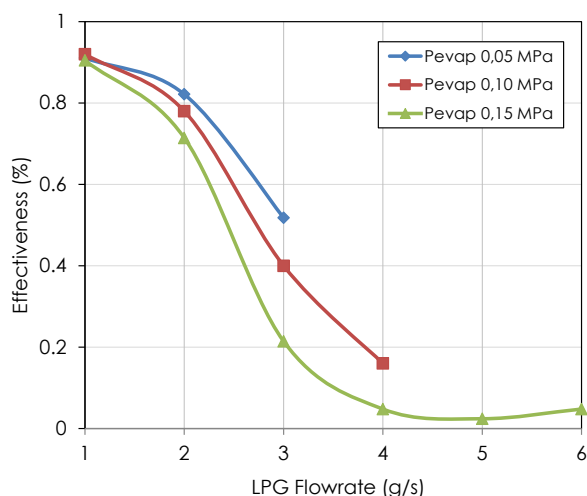
Figure 9 Effectiveness of heat transfer on the evaporator on variations LPG mass flow rate and evaporation pressure

Figure 9 shows the greater the LPG mass flow rate, the lower the heat transfer effectiveness. At a higher mass flow rate, heat transfer occurs less optimally, where the temperature on the evaporator exit side is lower. Finally, the COP curve and the cooling effect curve for mass flow rates are presented in Figure 10.
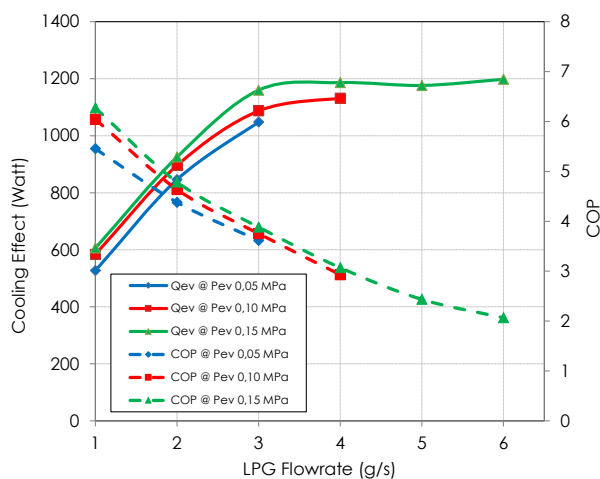

Figure 10 COP curve and cooling effect curve for LPG mass flow rate

4.0   CONCLUSION

From the analysis, the deviation of the actual cooling curve from an ideal cooling arc is due to the effectiveness of heat transfer in the evaporator. At a small LPG flow rate (<3 g/s), transfers heat from the air to LPG for evaporation to be perfect, coming out as vapor or superheated vapor. However, in case the mass flow rate is enlarged, the evaporator cannot transfer heat properly due to the limitations of an effective area. At the flow rate greater than 3 g/s, LPG comes out of evaporator as a mixture of vapor and liquid. From the results of this analysis, various types of

evaporators with greater heat transfer capacity still needs to be worked on.

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