International Scientific Journal
Journal of Environmental Science
http://environment.scientific-journal.com/

COMPARISON OF THE EXPERIMENTAL PERFORMANCE OF AN AUTOMOTIVE HEAT PUMP SYSTEM USING HFO1234YF AND HFC134A

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***Abstract*—In this study, performance characteristics of an experimental automotive heat pump (AHP) system charged with refrigerants HFO1234yf and HFC134a have been evaluated and compared with each other. For this aim, a bench-top experimental AHP system was made up from the original components of an air conditioning system belonging to a compact car. The AHP system was also equipped with some extra components to operate it in reverse direction as an air-source heat pump. The system had instruments for temperature, pressure, compressor speed and refrigerant mass flow rate measurements as well as data acquisition system. The system were tested at five different compressor speeds, namely 1000, 1500, 2000, 2500 and 3000 rpm, and for each compressor speed, the temperatures of the air streams at the inlets of the evaporator and condenser were maintained at two different sets, namely $T_{cond,ai}$=5°C – $T_{evap,ai}$=5°C and $T_{cond,ai}$=15°C – $T_{evap,ai}$=15°C. It was determined that the AHP system for both refrigerant cases provided enough heating capacity and conditioned air stream temperatures, and these two performance parameters increased with rising compressor speed. Furthermore, although HFO1234yf provided slightly lower heating capacity and lower coefficient of performance, it yielded comparable performance with HFC134a. This study shows that HFO1234yf is a major alternative to HFC134a for not only summer air conditioning but also winter comfort heating by using it in a heat pump.**

***Keywords*—heat pump; automotive; HFO1234yf; HFC134a; experimental**

I. Introduction

Thermal comfort of passengers is one of the important factors to be considered in designing a vehicle. The main intentions of vehicle thermal comfort are absolutely cooling, dehumidifying and heating. Although cooling, dehumidifying and heating using engine waste heat have been prevalently investigated and employed in automotive air conditioning (AAC) systems, the heating using an automotive heat pump (AHP) has been developed in some recent decades. In order to provide thermal comfort inside the passenger compartment of vehicles in winter season, a heating system using engine coolant as a heat source is commonly utilised. However, it is reported that modern diesel engines cannot supply sufficient amount of waste heat in a reasonable period of time after starting up the engine due to their high efficiency caused by the advances in the development of injection engines and turbocharger use [1]. Moreover, because electric vehicles do not employ internal combustion engines, they do not have a significant waste heat source for comfort heating of the passenger compartment [2, 3]. Therefore, the aforementioned vehicles should utilise another heat source to accomplish the thermal comfort demand in cold climate conditions. One of the alternative ways for providing comfort heating is to reverse the operation of the AAC system, thus operating it as a heat pump. Such a heat pump system may provide supplementary comfort heating in a vehicle with diesel engine, or it may supply all heating load in an electric vehicle.

When implementing the mandate of the Montreal Protocol in 1987, HFC134a, also known as R134a, was introduced as an alternative refrigerant to R12 for refrigeration and air conditioning industries as well as AAC systems due to its zero ozone depletion potential (ODP) [4]. Because HFC134a has a high global warming potential (GWP), namely a GWP-per-100-year of 1430, it was listed as one of the controlled greenhouse gases in the Kyoto Protocol [5]. Moreover, European Union's f-gas regulation bans the use of HFCs with a GWP above 150 in the air conditioning systems of all new vehicles placed in the EU market after January 1st, 2017 [6]. Based on these regulations, the search of alternative refrigerants to replace HFC134a has been performed. Among the potential candidates are CO_2, R152a, and HFO1234yf. The latter has been leading the way as the best potential candidate for AAC use, since CO_2 requires extremely high system pressures and has relatively low efficiency, and on the other hand, R152a has comparatively high flammability [7].

HFO1234yf has been ratified to be able to replace HFC134a in AAC systems for its ability of working in similar operating conditions [8] and having good drop-in performance [9]. Moreover, it is approved to be compatible with the current regulations for having zero ODP [10] and a GWP-per-100-year of 4 [11]. Even though there is no previous investigation on the use of HFO1234yf in AHP systems, some investigators compared the cooling performance of HFO1234yf in AAC systems with that of R134a. Mota-Babiloni et al. [12] determined that the coefficient of performance (COP) and

cooling capacity of an AAC system with HFO1234yf was up to 3–11% and 9% lower, respectively, compared to the system with HFC134a. Similarly, Lee and Jung [13] found that an AAC system with HFO1234yf had 2.7% lower COP and 4.0% lower cooling capacity in comparison to the system with HFC134a. Additionally, Navarro-Esbri et al [14] investigated the performance of HFO1234yf in a vapour compression refrigeration system under a wide range of operating conditions, concluding that the COP and cooling capacity of the HFO1234yf system are about 19% and 9% lower than that of the HFC134a system, respectively.

Although studies on the use of HFO1234yf in AHP systems have not been found in the open literature yet, there are numerous studies on the HFC134a AHP systems. Meyer et al. [15] developed an air-source HFC134a AHP by modifying a truck air conditioning system to measure heating curve. They discovered that the developed system had a better performance than that of the baseline. Hosoz and Direk [16] developed an experimental air-source AHP system with HFC134a, and tested it by changing the air inlet temperature and compressor speed. They determined that the AHP system provided significant heating capacity with a COP for heating higher than COP for cooling. Hosoz et al. [17] also compared the experimental performance of HFC134a AHP system using various heat sources, namely ambient air, exhaust gas and engine coolant, with the baseline heating system. Both transient and steady state performance of the bench-top AHP system driven by a Fiat Doblo JTD diesel engine was evaluated. When compared to the baseline heating system, it was revealed that the three considered AHP systems were beneficial in terms of their heating capacity both in idling and in the first minutes of high speed and torque operations.

As seen in the literature survey outlined above, a comprehensive performance analysis of an AHP system using HFO1234yf, the most promising refrigerant in the market today as alternative to HFC134a, has not been presented yet. This study investigates and compares the performance characteristics of an AHP system using HFO1234yf and HFC134a with in a broad range of operating conditions in a fully monitored bench-top AHP system. Using experimental data obtained from the tests, the evaluations of both HFO1234yf and HFC134a AHP systems have been carried out in terms of the refrigerant mass flow rate, heating capacity, compressor power, COP for heating, the conditioned air temperature, and compressor discharge temperature of the AHP system. Finally, the results have been presented in comparative graphics and discussed comprehensively.

II. Methodology

A. Apparatus

The photograph and schematic illustration of the experimental AHP system are shown in Figs. 1 and 2, respectively. It consists of a five-cylinder wobble-plate compressor, a reversing valve, a plate-and-fin laminated type indoor unit serving as condenser in heat pump operations, two thermostatic expansion valves (TXVs), a receiver, a filter/drier, and a parallel-flow micro-channel type outdoor unit serving as evaporator in heat pump operations. The components were connected to each other by using insulated copper tubing with appropriate diameter.

The compressor was driven at the required speeds by a four kW three phase electric motor with a nominal speed of 2850 rpm energised via a frequency inverter. The indoor and outdoor units were inserted into their one-meter-long air ducts with flow areas of 0.24×0.24 m^2 and 0.67×0.35 m^2, respectively. In order to provide the required air streams, a centrifugal fan was attached to the inlet of the indoor air duct providing an air flow rate of 0.112 m^3/s. On the other hand, an axial fan was attached to the inlet of the outdoor air duct providing an air flow rate of 0.182 m^3/s. Furthermore, to test the system at the required air inlet temperatures, both air ducts were equipped with electric heaters located upstream of the indoor and outdoor units. The electric heaters in the indoor and outdoor ducts could be energised in the ranges of 0-2 kW and 0-6 kW, respectively.

The instruments used for mechanical measurements in the AHP system are depicted in Fig. 2, whereas their specifications are described in Table 1. A Coriolis mass flow meter, installed in the liquid line of the system, was used to measure the refrigerant mass flow rate. The refrigerant pressures both at the inlet and discharge of the compressor were measured by both pressure transmitters and Bourdon gauges. On the other hand, refrigerant temperatures at the inlets and outlets of each essential component were measured by type-K thermocouples. Furthermore, the dry bulb and wet bulb temperatures of the air streams were measured at the locations shown in Fig. 2. All temperature and pressure data were acquired through a data acquisition system and quasi-automatically recorded into a computer. The data acquisition system has 16 bit-200 KHz of frequency with 56 input channels of thermocouple module and 8 channels of transducer interface module.

The AHP system works by simply reversing the direction of the refrigerant flow in the circuit components excluding the compressor. This reversing loop is accomplished by employing a reversing valve installed at the suction and discharge lines of the compressor. Furthermore, the experimental AHP system employs eight manual valves to provide refrigerant flow in the required direction. These manual valves are shown in Fig. 1 using ‘V’ abbreviation followed by numerical order from 1 to 8. In order to operate the system in air conditioning (AC) mode, it is necessary not to energise reversing valve, to open the valves V1, V2, V3 and V4, and to close the others. On the other hand, to operate the system in heat pump (HP) mode, the reversing valve is energised, the valves V5, V6, V7 and V8 are opened, while others are closed. The flow direction of the refrigerant in HP mode is shown by dash-and-dot-arrow lines to differentiate with the flow direction in AC mode.

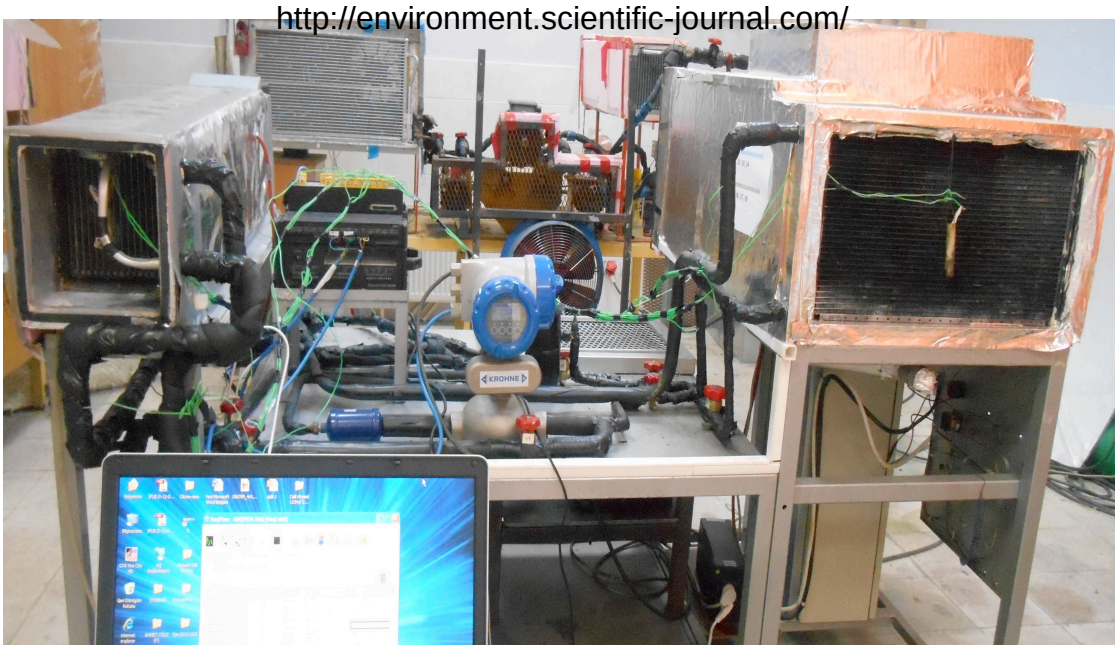

Fig.1. Photograph of the experimental automotive heat pump system

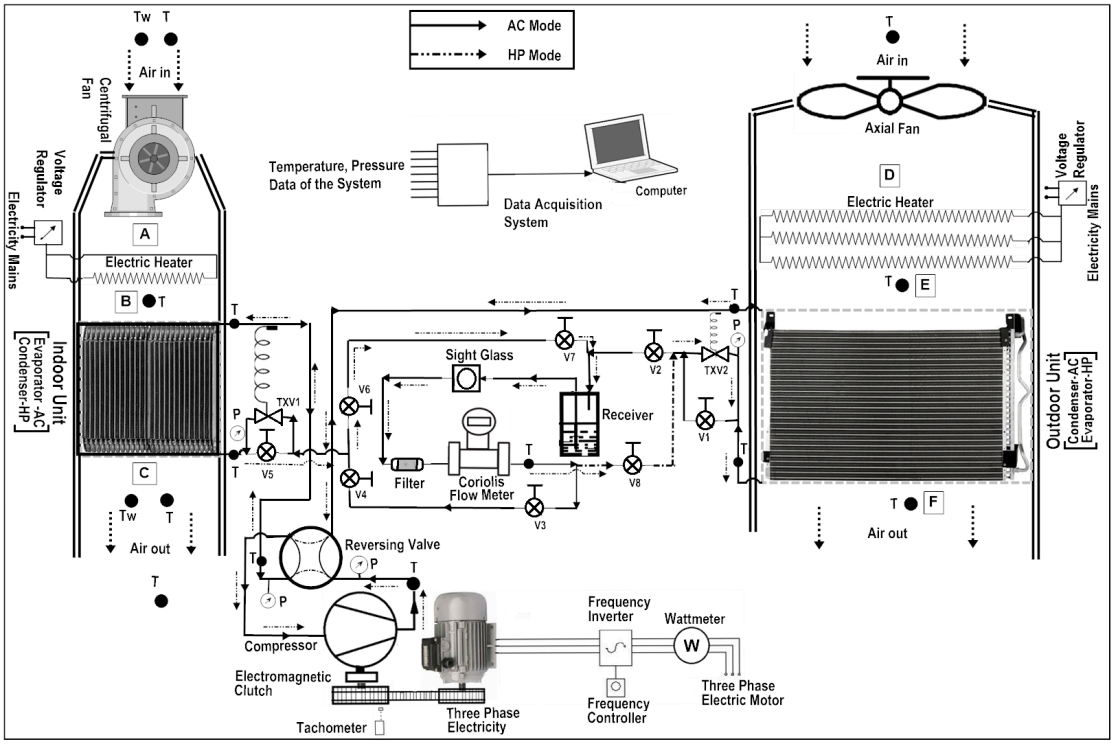

Fig. 2. Schematic diagram of the experimental automotive heat pump system

TABLE I. CHARACTERISTICS OF THE INSTRUMENTATION

Measured Variable	Instrument	Range	Uncertainty
Temperature	Type K thermocouple	-50 – 500 °C	± 0.3%
Pressure	Pressure transmitter Bourdon gauge	0 – 25 bar -1 – 10, 0 – 30 bar	± 0.2% 0.1, 0.5 bar
Air flow rate	Anemometer	0.1 – 15 m/s	± 3.0%
Refrigerant mass flow rate	Coriolis flow meter	0–350 kg/h	± 0.1%
Compressor speed	Photoelectric tachometer	10 – 100000 rpm	± 2%

After energising the reversing valve and opening/closing aforementioned valves for the HP operation, the high pressure superheated refrigerant leaving the compressor enters the reversing valve, which directs the refrigerant to the indoor unit serving as condenser in this HP mode. As the refrigerant passes through the indoor unit, it rejects heat to the indoor air stream, thus providing a conditioned air stream. After leaving the indoor unit as high pressure subcooled liquid, the refrigerant passes through V5, V6, V7 and enters the receiver, which holds the unneeded refrigerant when the TXV reduces the refrigerant mass flow rate at low outdoor temperature operations. After leaving the receiver, the liquid refrigerant passes through the sight glass, filter/drier and Coriolis flow meter. Because V3 is closed, the refrigerant flows through V8 directly to the thermostatic expansion valve 2 (TXV2), in which it undergoes pressure reduction and accompanying temperature drop before entering the evaporator. The refrigerant leaving the TXV2 as low pressure mixture of saturated liquid and vapour enters the outdoor unit serving as evaporator. By absorbing heat from the outdoor air stream, the refrigerant evaporates and leaves the evaporator as low pressure superheated vapour. The bulb of the TXV2 continually senses the temperature at the outlet of outdoor unit, and the TXV2 adjusts the refrigerant mass flow rate to keep the superheat at the evaporator outlet constant at a predetermined value. Then, the refrigerant goes to the reversing valve, which directs the refrigerant back to the compressor, thus completing the cycle.

The AHP system was first charged with 2.00 kg of HFO1234yf, and the first set of tests were conducted to evaluate the performance of the system. Then, the HFO1234yf was recovered and 2.20 kg of HFC134a was charged into the system for the second set of tests. As Wang [7] suggested,t about 10% less HFO1234yf charge was used mainly due to its 10% lower liquid density compared with HFC134a. The oil used in the system was approximately 300 cc of PAG oil.

The system were tested at five different compressor speeds, namely 1000, 1500, 2000, 2500, and 3000 rpm, and for each compressor speed, the temperatures of the air streams at the inlets of the condenser and evaporator were maintained at two different sets, namely $T_{cond,ai}$=5°C – $T_{evap,ai}$=5°C and $T_{cond,ai}$=15°C – $T_{evap,ai}$=15°C. Each test took 15 minutes, and the steady-state test data, which were gained after about 10-minute transient operation, were employed in the evaluations. It was assumed that the steady-state was achieved when the temperature deviations at the key points considered were within 0.5 °C for 3 minutes [18]. Jabardo et al. [19] proposed that the steady-state operation of an AHP system constitutes an adequate procedure for performance evaluation despite of the fact of rather unusual. In addition, after finishing each test, 10 minutes of interlude was given to the AAC system before starting the next test to ensure that the similar starting conditions existed.

In order to evaluate the energetic performance, the thermodynamic properties were obtained from REFPROP [20] using the data from the thermocouples and pressure transmitters located at the locations shown in Fig. 2. Eventually, using the equations presented in the next sections, the energetic performance parameters, i.e. the refrigerant mass flow rate, heating capacity, compressor power, COP for heating, air stream temperature at the outlet of condenser, and compressor discharge temperature were evaluated. The comparison of energetic performance parameters of the AHP system with both refrigerants as a function of the compressor speed and the set of $T_{cond,ai}$ – $T_{evap,ai}$ are discussed thoroughly.

B. Energetic Performance Analysis

The performance parameters of the AHP system can be evaluated by applying the first law of thermodynamics (FLT) to each component of the system by assuming that the system operates in steady state and the changes in kinetic and potential energies are negligible.

The heating capacity can be evaluated by applying the FLT to the indoor unit, i.e.

$$\dot{Q}_{cond} = \dot{m}_r\left(h_{cond,in} - h_{cond,out}\right) \quad (1)$$

where $\dot{m}_r$ refers to the refrigerant mass flow rate and h is the enthalpy of the refrigerant.

By applying the FLT to the outdoor unit, the evaporator load can be determined from

$$\dot{Q}_{evap} = \dot{m}_r\left(h_{evap,out} - h_{evap,in}\right) \quad (2)$$

Assuming that the compressor works adiabatically, the compressor power absorbed by the refrigerant during the compression process can be obtained from

$$W_{comp} = \dot{m}_r\left(\mathrm{h}_{\mathrm{comp,out}} - \mathrm{h}_{\mathrm{comp,in}}\right) \quad (3)$$

The energy performance of the AHP system is revealed by evaluating its COP for heating. For heat pump systems, it is defined as the ratio between the heating capacity and compressor power, i.e.

$$COP = \frac{Q_{cond}}{W_{comp}} \quad (4)$$

Finally, the numerical comparison of the performance parameters for both refrigerants can be exposed to relative deviation by taking HFC134a as baseline [21] in order to obtain a distinct evaluation, i.e.

$$\%\varepsilon = \left|\frac{\beta_{1234yf} - \beta_{134a}}{\beta_{134a}}\right| \times 100 \quad (5)$$

III. RESULTS AND DISCUSSION

The performance parameters of the experimental AHP system using HFO1234yf and HFC134a were evaluated and presented in Figs. 3-7 as a function of compressor speed for two different sets of air temperatures at the inlets of indoor and outdoor units, namely $T_{cond,ai}$=5°C – $T_{evap,ai}$=5°C and $T_{cond,ai}$=15°C – $T_{evap,ai}$=15°C.

Fig. 3 displays the comparison of the refrigerant mass flow rate ($\dot{m}_r$) circulating through the system as a function of the compressor speed. It was found that the system using HFO1234yf yielded on average 3.67% lower $\dot{m}_r$ values for the air inlet temperature sets of $T_{cond,ai}$=5°C – $T_{evap,ai}$=5°C, and 3.77% higher $\dot{m}_r$ values for the set of $T_{cond,ai}$=15°C – $T_{evap,ai}$=15°C compared with the use of HFC134a. It is seen that the $\dot{m}_r$ increases with rising compressor speed and the set of air inlet temperatures for both refrigerants.

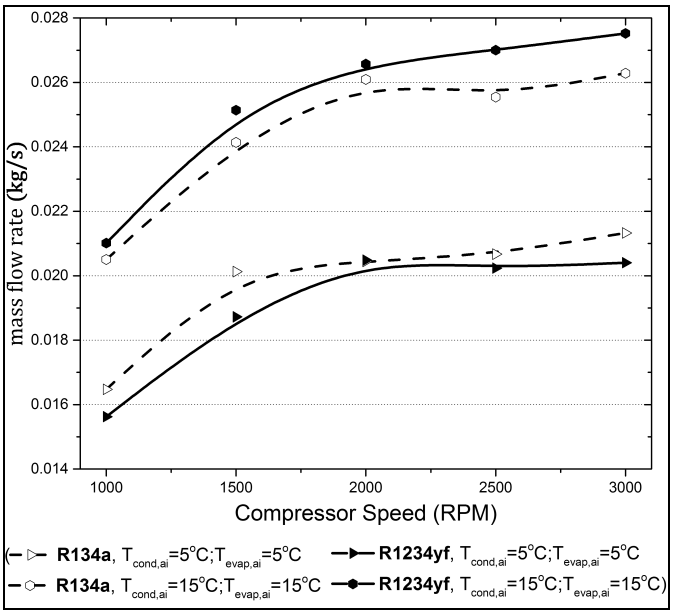

Fig. 3. Comparison of the refrigerant mass flow rate

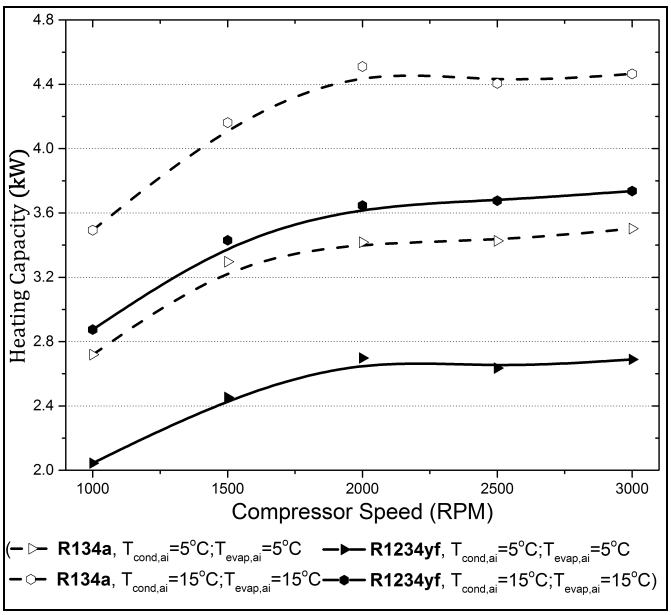

Fig. 4. Comparison of the heating capacity

Fig. 4 demonstrates the comparison of heating capacity as a function of compressor speed for two different sets of air temperatures at the inlets of indoor and outdoor units. The heating capacities for the two refrigerants increase on rising both the compressor speed and the set of $T_{cond,ai}$ – $T_{evap,ai}$. As the compressor speed increases, so does the refrigerant mass flow rate, thus increasing the heating capacity. As the air temperature entering the evaporator increases, the refrigerant can absorb more heat during evaporation, thus rejecting more heat when condensing, which consequently causing an increase in the heating capacity. On the other hand, the heating capacity of HFO1234yf was 21.05–25.57 and 16.32–17.57% lower than that of HFC134a for the sets of $T_{cond,ai}$=5°C – $T_{evap,ai}$=5°C and $T_{cond,ai}$=15°C – $T_{evap,ai}$=15°C, respectively. Despite of the fact that the refrigerant mass flow rate of HFO1234yf was higher than that of HFC134a for the operations at the set of $T_{cond,ai}$=15°C – $T_{evap,ai}$=15°C, the poor heating capacity of the system with HFO1234yf was due to the superior thermophysical properties of HFC134a compared to those of HFC1234yf.

The comparison of compressor power, i.e. power absorbed by refrigerant in the compressor, as a function of compressor speed is shown in the Fig. 5. The figure reveals that the power absorbed by refrigerant in the compressor for the two refrigerants increase on rising both the compressor speed and the set of $T_{cond,ai}$ – $T_{evap,ai}$. Increasing the compressor speed essentially lowers the volumetric efficiency, thus causing higher thermodynamic losses in the compression process. The power absorbed by the refrigerant in the compressor with HFO1234yf was 11.36-21.00 and 8.93-21.83% lower than that with HFC134a for the sets of $T_{cond,ai}$=5°C – $T_{evap,ai}$=5°C and $T_{cond,ai}$=15°C – $T_{evap,ai}$=15°C, respectively.

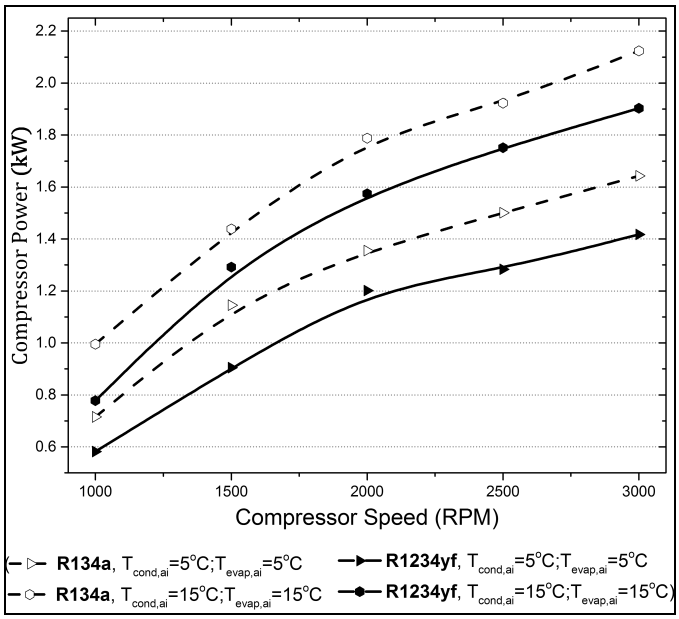

Fig. 5. Comparison of the compressor power

Fig. 6 indicates the comparison of the coefficient of performance for heating between HFO1234yf and HFC134a as a function of compressor speed. Recall that the COP for heating for an AHP system is the ratio of the heating capacity to the power absorbed by refrigerant in the compressor. Because the compressor power increases more sharply than the heating capacity does with the compressor speed, the COP for heating in both refrigerant cases drops with increasing compressor speed. Moreover, the COP for heating increase on rising the set of $T_{cond,ai}$ – $T_{evap,ai}$. Fig. 6 also reveals that the system with HFO1234yf had on average 9.08 and 5.23% lower COP values than that with HFC134a for the sets of $T_{cond,ai}$=5°C – $T_{evap,ai}$=5°C and $T_{cond,ai}$=15°C – $T_{evap,ai}$=15°C, respectively.

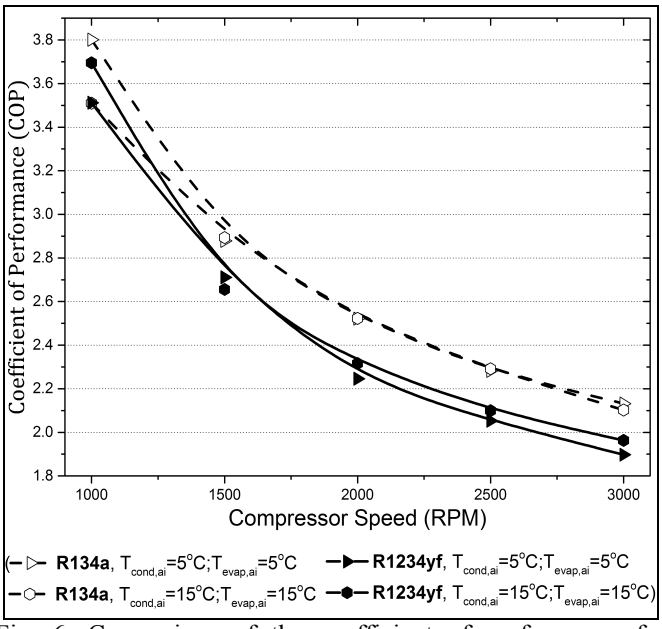

Fig. 6. Comparison of the coefficient of performance for heating

Fig. 7 depicts the comparison of dry bulb temperature of the conditioned air stream at the outlet of condenser as a function of compressor speed. Recall that condenser for an AHP system is the indoor unit, which provides the comfort heating for the passenger compartment. It was determined that the dry bulb temperature of the conditioned air stream at the outlet of condenser with HFO1234yf was 5.7–8.0°C and 5.6–7.4°C lower than that with HFC134a for the sets of $T_{cond,ai}$=5°C – $T_{evap,ai}$=5°C and $T_{cond,ai}$=15°C – $T_{evap,ai}$=15°C, respectively. These results reveal that HFO1234yf shows comparable performance with HFC134a in AHP systems in terms of both the conditioned air temperature and the heating capacity.

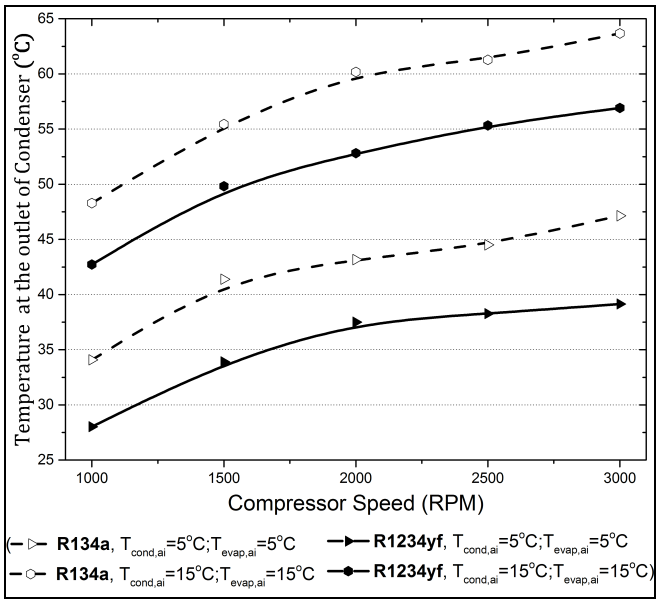

Fig. 7. Comparison of the conditioned air temperature

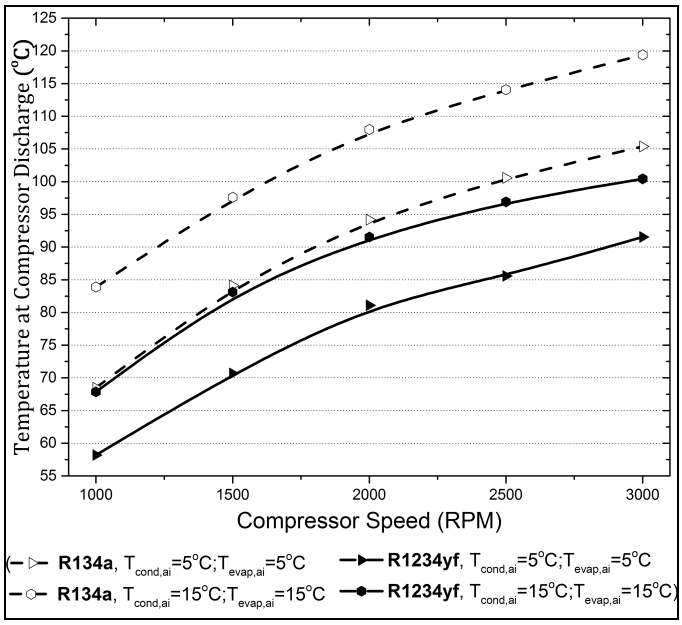

Fig. 8. Comparison of the compressor discharge temperature

The comparison of compressor discharge temperature, which is an essential parameter to be used as a limit criterion of usability, is shown in Fig. 8. This comparison is done also as a function of compressor speeds for both HFO1234yf and HFC134a. The higher the discharge temperature, the higher the risk of impairing lubrication, thus affecting the mechanical parts of the compressor and causing the deterioration of its durability [21]. It was determined that the compressor discharge temperature with HFO1234yf was 10.3–15.0°C and 14.5–18.9°C lower than that with HFC134a for the sets of $T_{cond,ai}$=5°C – $T_{evap,ai}$=5°C and $T_{cond,ai}$=15°C – $T_{evap,ai}$=15°C, respectively. As clearly depicted in the graph, the compressor discharge temperature increases on rising the compressor speed and increasing both $T_{cond,ai}$ – $T_{evap,ai}$ simultaneously. Since the compressor discharge temperature with HFO1234yf is significantly lower than that with HFC134a, HFO1234yf is considered as appropriate in terms of the durability of mechanical parts of the compressor for further designs.

IV. CONCLUSION

This study presents a comparison of the performance parameters of an automotive heat pump system using HFO1234yf and HFC134a as refrigerants. The following conclusions can be drawn from this investigation.

- The AHP system with HFO1234yf shows slightly poorer performance in terms of the heating capacity, compressor power and COP compared with HFC134a. However, because the global warming potential of HFO1234yf is just 4 and that of HFC134a is as high as 1430, HFO1234yf can still be regarded as a good candidate for HFC134a in not only summer air conditioning but also heat pump systems for automotive applications.
- Although the dry bulb temperature of the conditioned air stream at the outlet of condenser with HFO1234yf is 5.6–

8.0°C lower than that with HFC134a, the AHP system with HFO1234yf can still be used for comfort heating of automotive passenger compartments. However, increasing the capacities of refrigeration circuit components, namely the compressor, condenser, expansion device and evaporator, will yield higher heating capacities and conditioned air temperatures for the AHP system using HFO1234yf.

- Finally, in terms of the compressor durability, which can be assessed using the temperature at the discharge of the compressor, the system with HFO1234yf fits the requirement better due to its lower temperature compared to that of HFC134a.

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