24th ABCM International Congress of Mechanical Engineering
December 3-8, 2017, Curitiba, PR, Brazil

COBEM-2017-1821
EXPERIMENTAL ANALYSIS OF AN AIR CONDITIONING SYSTEM WITH HFO-R1234yf AND THE INFLUENCE OF AIR INLET TEMPERATURES

Anselmo Gourlart Gonçalves
Jacqueline Biancon Copetti
Mario Henrique Macagnan
Universidade do Vale do Rio dos Sinos - Unisinos, Mechanical Engineering Graduate Program, LETEF
Laboratory of Thermal and Fluid Dynamic Studies, São Leopoldo, Brazil
anselmo_agg@hotmail.com, jcopetti@unisinos.br,mhmac@unisinos.br

Juliano Noetzold
AGCO – Agriculture Company, Av. Guilherme Schell, 10260, Canoas, RS – Brazil
juliano.noetzold@agcocorp.com

Abstract. *An air conditioning system with refrigerant R1234yf for off-road vehicles is experimentally evaluated. For an ideal load, previously established, the effect of the variation in air inlet temperatures of evaporator and condenser and the compressor speed were verified on the main performance parameters. As the evaporator air temperature increases, the refrigeration capacity, compressor power and COP also increase, but the latter less significantly. In condenser, the increases in the air temperature had the opposite effect, in addition, it required an increase the mass flow rate and subcooling and superheating temperatures decreased.*

Keywords: *R1234yf; Automotive air conditioning; Drop-in.*

1. INTRODUCTION

The natural refrigerant hydrofluoroolefin HFO-R1234yf has been experimentally studied as a promising substitute for R134a in automotive AC applications. It has zero ODP, low GWP, similar thermophysical properties and it can work at similar operating conditions (Tanaka and Higashi, 2010; Rotchana and Suen, 2012; McLinden et al., 2013), showing good drop-in performance (Spatz and Minor, 2008; Zilio et al., 2011; Lee and Jung, 2012).

Experimental and theoretical studies have compared the mobile air conditioning (MAC) cycle performance of HFO-1234yf with that of R134a using drop-in replacement. Zilio et al. (2011) verified that the R1234yf cooling capacity and COP in a MAC are considerably lower than those obtained with R134a, and they suggest some hardware modifications in order to reduce the different between both refrigerants. Jarall (2012) theoretically and experimentally compared the cycle performance of HFO-1234yf with that of R134a using drop-in replacement. The results shown that superheating and subcooling temperature play more important roles than that of R134a system. The performance of R1234yf system was less than that of R134a system in terms of cooling capacity, COP and compressor efficiency.

Lee and Jung (2012) obtained COP and cooling capacity of R1234yf up to 2.7% and 4.0% lower when compared to R134a. Cho et al. (2013) observed that the addition of an IHX (internal heat exchanger) in MAC can approach the R1234yf performance to those of R134a. Alternatively, this substitution has been proposed in refrigeration applications. Qi (2013) demonstrated that although R1234yf performs worse than R134a in laminated plate evaporator, the R1234yf cooling capacity is comparable and/or larger in microchannel parallel flow evaporator. In another work, this author (Qi, 2015) performed a thermodynamic analysis for R1234yf MAC system under various operation conditions. It was found that superheat was few benefits for both COP and cooling capacity. However, increasing subcooling, these parameters could be improved by 15% if compressor consumption power was fixed. Navarro- Esbrí et al. (2013a) studied R1234yf performance in a vapor compression system varying a wide range of condition, concluding that the cooling capacity and COP for R1234yf are about 9% and 19% lower than those obtained using R134a. The same authors (Navarro-Esbrí et al., 2013b) compared the influence of an IHX in systems with R134a and R1234yf. The results for energy performance were better using R1234yf. Lastly, in the case of the two-phase heat transfer, great similarities were found between R134a and R1234yf, in a review made by Wang (2013). The greatest differences took place for in-tube condensation, being heat transfer coefficient of R1234yf lower to those of R134a.

A previous experimental study (Noetzold *et al.*, 2016) was carried out to verify the feasibility of direct substitution of R134a of a typical AC for off-road vehicles (6.4 kW nominal capacity) with minimum modifications. The optimum load and performance of this refrigerant has been established for this application, which has a very high refrigeration capacity to provide a cool down. The initial results shown, for air inlet temperature conditions at 40 °C and relative humidity of 43%, that the system with the R1234yf has slightly lower values of cooling capacity and COP, and a higher superheating temperature, and the system reaches the optimal refrigerant charge with 50 g more than with R134a.

In this work, the analysis of system with R1234yf is extended by investigating experimentally the effect of the air temperature at the evaporator and condenser inlet, considering the results of optimum charge of previous work.

2. EXPERIMENTAL PROCEDURE AND DATA REDUCTION

The experimental setup utilized for automotive AC performance measurements is shown in Fig. 1, and consists a wind tunnel and a calorimeter, for evaluation of condensers and evaporators, respectively, which provide controlled temperature, relative humidity and airflow rate. The system is composed by the typical automotive AC components like a belt driven compressor, condenser, thermostatic expansion valve and an evaporator box. In the experiment were measure temperatures in different points of the refrigerant circuit and the air, absolute pressure at the inlet of evaporator and condenser, and differential pressure in these heat exchangers. In the liquid line, after the filter drier, a Coriolis mass transmitter measure the refrigerant mass flow rate. The air side volumetric flow rate, and consequently the air velocity, was measured using two plate nozzles and two pressure transmitters.

During the experiments, the major operating parameters were monitored graphically and numerically in real time by an Agilent 34980A data acquisition system, controlled by a personal computer. All data were stored for later analysis and graphical representation.

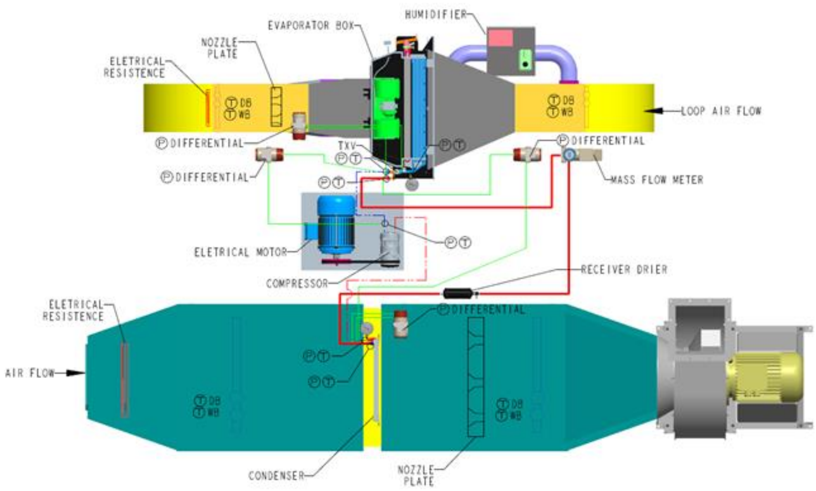


Figure 1. The schematic diagram of the experimental setup.

The charge of new refrigerant R1234yf was previously established with a study comparative with R134a to evaluate the possibility of direct drop-in replacement (Noetzold *et al.*, 2016). Then, considering the charge of 1150 g, the analysis of the effect of air temperatures at the inlet of the evaporator and the condenser, was developed. The evaporator inlet air temperature ranged from 15 to 40 °C, while in the condenser ranged from 25 to 45 °C.

For the studies at the condenser, the evaporator air inlet temperature was kept at 28 °C and relative humidity was set 50 % in order to reflect the driving conditions in summer season (Benouali and Clodic, 2003). Similarly, to the study of the evaporator, the condenser inlet air temperature was kept at 38 °C accordingly SAE J1503 that outlines the tests procedures for off-road self-propelled work machines, like general purpose industrial, agricultural and forestry machines. The compressor speed was set at 2000 rpm. For each experiment the operating conditions of the air conditioning were considered in steady state when the air side inlet temperatures at the condenser and the evaporator remained between ± 0,5 ºC.

The evaporator cooling capacity, Q_E, the condenser capacity, Q_C, and compressor power, W_C, were calculated by energy balances in the components, considering the enthalpies in the temperature and pressure of each thermodynamic state of the cycle, like the following equations:

$$Q_E = \dot{m}_r (h_{e,o} - h_{e,i}) \tag{1}$$

$$Q_C = \dot{m}_r (h_{c,i} - h_{c,o}) \tag{2}$$

where $\dot{m}_r$ is the refrigerant mass flow rate, $h_{e,i}$ and $h_{e,o}$ are the inlet and outlet evaporator enthalpies and $h_{c,i}$ and $h_{c,o}$ are the inlet and outlet condenser enthalpies. The thermodynamic properties of R1234yf were obtained from REFPROP software (Lemmon and McLinden, 2009). The performance of the system, *COP*, was calculated accordingly:

$$COP = \frac{Q_E}{W_C} \tag{3}$$

Subcooling, evaluated at the condenser outlet and superheat evaluated at the compressor inlet are defined accordingly Eq. (4) and (5):

$$\Delta T_{sc} = T_{sat,c} - T_{liq} \tag{4}$$

$$\Delta T_{sh} = T_{vap} - T_{sat,e} \tag{5}$$

where $T_{sat,c}$ is the saturated liquid temperature at condensing pressure, T_{liq} is the refrigerant temperature in the liquid line, $T_{sat,e}$ is the saturated vapor temperature at the evaporator pressure and T_{vap} is the refrigerant temperature in the suction line.

3. RESULTS

Table 1 presents some basic properties of the two refrigerants used in this work. R1234yf and R134a have similar values for molar mass, critical temperature, reduced pressure and specific heat. The heat of vaporization of R1234yf is smaller, about 21% the value of R134a.

Table 1. Characteristics and properties of R1234yf and comparison with R134a for 20ºC.

	R1234yf	R134a
GWP (100 anos)	< 4	1430
ODP	0	0
Molar mass (kg/kmol)	114	102
Critical temperature and pressure (ºC/MPa)	94.7/3.382	101/4.059
Saturation pressure$_{20\,ºC}$ (kPa); Reduced pressure (-)	591.7/0.175	571.7/0.14
Density vapor/liquid (kg/m³)	32.84/1110	27.8/1225
Specific volume vapor/liquid (m^3/kg)	0.03045/0.0009008	0.03597/0.0008163
Liquid thermal conductivity (mW/mK)	64.77	85.62
Specific heat liquid/vapor (kJ/kgK)	1.33/1.024	1.4/1.0
Liquid viscosity (μPa-s)	171.2	206.8
Heat of vaporization (kJ/kg)	150.2	182.27

3.1. R1234yf and R134a comparison

In Figure 2 are presented in a pressure-enthalpy diagram the cycle operating conditions using R1234yf, for air temperatures at the condenser and evaporator inlet equal to 40 ºC. Under these conditions, the cycle has a compressor discharge pressure of 1576 kPa, corresponding to a saturation temperature of 58.2 ° C. The subcooling at condenser outlet is 8.5 K. The refrigerant quality at the evaporator inlet is 28.1% and the superheating at evaporator exit is 6 K. The enthalpy change in the evaporator is 88 kJ/kg. When operating with R134a, the same parameters were measured, resulting in: compressor discharge pressure of 1730 kPa, corresponding to a saturation temperature of 60.5 °C, subcooling is 8.5 K, refrigerant quality 26,3% and the 3.5 K of superheating, enthalpy change in the evaporator is 137 kJ/kg, approximately 55% higher.

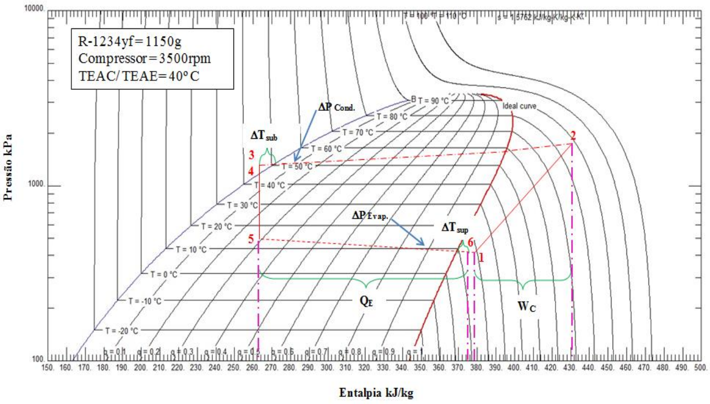


Figure 2. R1234yf cycle representation.

In Figure 3a the results of the tests are presented in terms of evaporator capacity. For air temperatures at the evaporator inlet at 35 and 30 °C, the R1234yf evaporator capacity was lower, between 25 and 15% and for the temperature of 20 °C the capacity reduction was more pronounced, between 30 and 41%. The results are similar to those by Zilio et al. (2011), who found lower evaporator capacities for the R1234yf, ranging from 12 to 20% for air temperatures of 25 °C, and 15 to 24% for air temperatures of 35 °C. Jarall (2012) also experimentally analyzed the direct replacement of R134a by R1234yf in a stationary AC system with a hermetic compressor rated at 550 W and operating at refrigerant saturation temperatures in the condenser at 40 and 45 °C. In the tests, the saturation temperature in the evaporator varied between -5 and 15 °C. The results showed lower capacities in the evaporator, between 3.4 and 11.15% and of 7.6 and 13.7% for condenser temperatures of 40 and 45 ºC, respectively.

Although the compression powers of the cycle with R1234yf are smaller, its lower evaporator capacity results in lower COP (Figure 3b), between 15 and 7%, for all the test conditions performed. According to the study of Navarro et al. (2013), the COP for the system using R1234yf were 11 to 24% lower than the system operating with R134a.

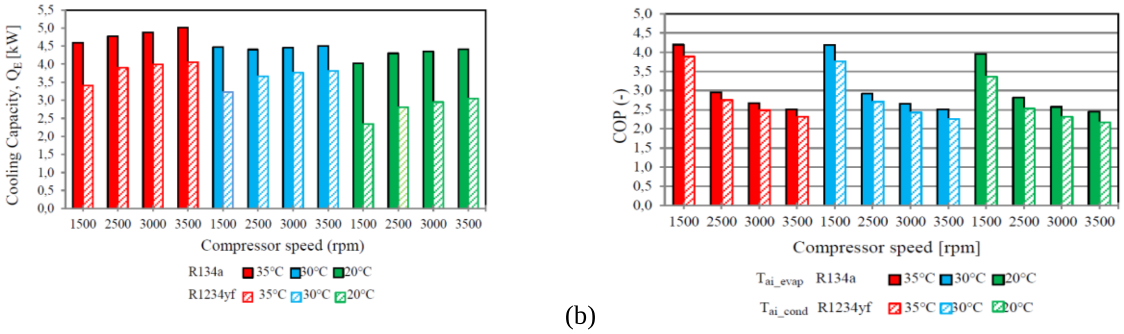
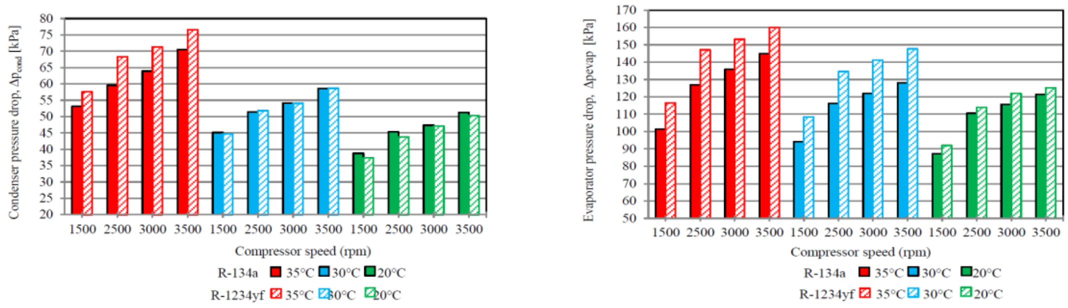


Figure 3. Cooling capacity (a) and COP (b) for different compressor speed and refrigerants R134a and R1234yf.

The pressure drop in the condenser is higher in the system operating with R1234yf with evaporator air inlet temperature of 35 ° C, as shown in Figure 4a. With air entering at 30 °C, the pressure loss was practically identical for both refrigerants, but at 20 °C, the trend reversed and R134a showed a greater pressure loss. Moreover, the tendency is to increase the pressure drop as compressor speed increases.

The pressure drop in the evaporator, shown in Fig. 4b, has the same behavior, is higher for R1234yf, increasing with both, evaporator air inlet temperature and compressor speed. The highest pressure drop values for R134a and R1234yf were 145 and 160 kPa, which represents a variation in saturation temperature of 12.1 and 13.5 ºC, respectively, operating at 35 °C and 3500 rpm. The smallest were 87 and 92 kPa, representing a variation in saturation temperature of 7.7 and 8.7 ºC, operating at 20 ° C and 1500 rpm.

Cho *et al.* (2013) performed an experimental study of an MAC system in direct substitution of R134a by R1234yf, focusing on heat exchangers and pressure drop in these components and found higher values of pressure drop in the system with R1234yf.



(a) (b)

Figure 4. Pressure drop in condenser (a) and evaporator (b) for refrigerants R134a and R1234yf at different evaporator air inlet temperature and compressor speed.

The pressure drop in the suction line between the evaporator outlet and the compressor inlet, shown in Fig. 5, is higher for R1234yf for all compressor speed range and evaporator air inlet temperature. Spatz and Minor (2008) recommend and improvement increasing the suction pipe diameter in the suction line of MCA with R1234yf. In this way, it is possible to reduce this pressure drop and thus optimize the superheat at the compressor inlet.

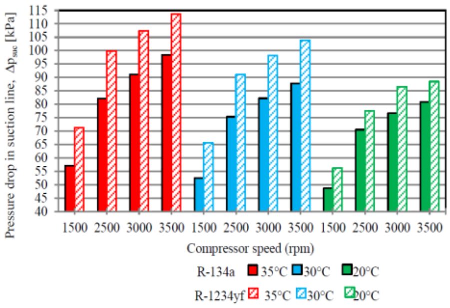
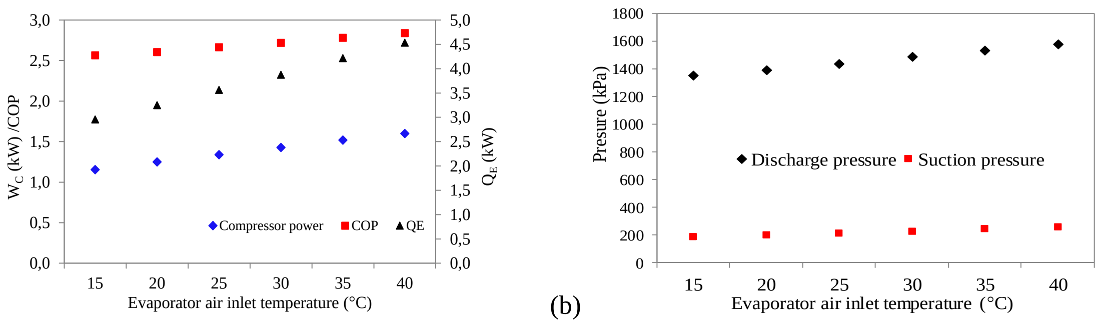


Figure 5. Pressure drop in the suction line for refrigerants R134a and R1234yf at different evaporator air inlet temperature and compressor speed.

3.2. Effect of air inlet temperature at evaporator and condenser for R1234yf system

In Figure 6a are presented the evaporator capacity, Q_E, the compressor input power, W_C, and the *COP* as function of the evaporator inlet air temperatures. As the air temperature rises, the temperature difference between the air and the refrigerant increases, increasing the capacity of the evaporator. Consequently, there is also an increase in compressor power. The COP also increases, but less significantly. Similarly, it can be observed that the increasing air temperature induce an increase of both the discharge and suction pressures, as shown in Figure 6b.



(a) (b)

Figure 6. a) Effect of evaporator inlet air temperatures on compressor power, evaporator capacity and COP, and b) in suction and discharge pressures for condenser air inlet temperature at 38ºC.

In Fig. 7 one can observe the variation of different parameters. The increase of the mass flow rate, $\dot{m}_r$, with the air temperature in the evaporator, due to the higher vaporization rate, also the increase of refrigerant temperature, $T_{e,i}$, at the evaporator inlet, and the outlet pressure (Fig. 7a). This causes an increase in the specific volume at the compressor inlet, increasing its capacity. The increase in evaporator capacity is much more sensitive to increasing the mass flow rate than the enthalpy change between inlet and outlet (Fig. 7b). Regardless of this, the increase in temperature ($T_{c,i}$) and pressure in the discharge of the compressor is very small.

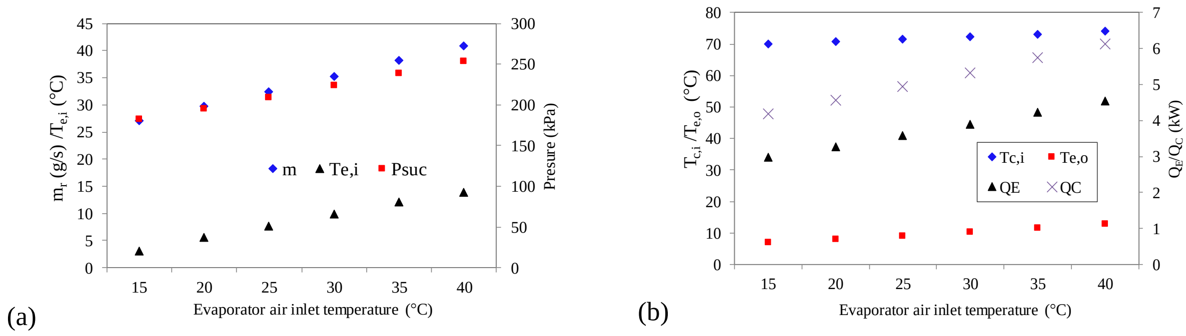


Figure 7. Effect of evaporator inlet air temperatures a) on refrigerant mass flow rate, refrigerant temperature at the evaporator inlet and the suction pressure b) on condenser inlet temperature, evaporator outlet temperature and capacity of evaporator and condenser.

The increase air temperature at the condenser inlet is responsible for increases in the compressor power and a decrease in evaporator capacity, as shown in Fig. 8a. As a result, the COP of the system decreases. In Fig. 8b it is possible to observe the increase in the discharge pressure with the condenser air inlet temperature and the increasing pressure ratio. The suction pressure also increases, but significantly less than the discharge pressure.

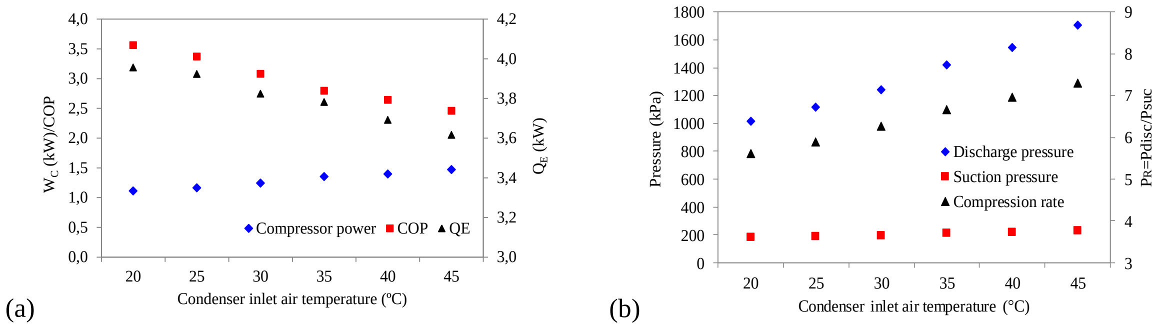


Figure 8. a) Effect of condenser inlet air temperatures on compressor power, evaporator capacity and COP and b) in suction and discharge pressures for evaporator air inlet temperature at 28ºC.

Figure 9 shows the effect of evaporator and condenser air inlet temperatures, *Tai_Evap* and *Tai_Cond*, respectively, on subcooling and superheating. As the average saturation temperature in the evaporator increases, the degree of superheating decreases slightly with increasing inlet air temperature and as the capacity of the condenser increases, the subcooling tends to increase. In relation to the increase of the condenser air inlet temperature, the capacity of thermal exchange in the condenser decreases slightly and therefore the subcooling. In the evaporator the saturation pressure increases, decreasing the capacity of the evaporator, and affecting the superheating.

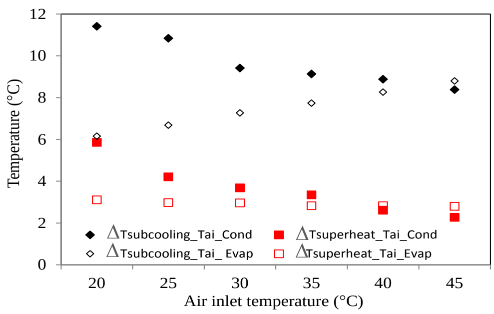


Figure 9. Effect of inlet air temperatures of condenser and evaporator on subcooling and superheating temperatures.

4. CONCLUSIONS

In this experimental work on a test rig using an automotive AC operating with R1234yf and thermostatic expansion valve were analyzed the influence of evaporator and condenser air inlet temperatures in the behavior of different system performance parameters. For a fixed refrigerant charge and speed compressor, the temperature of the air at the evaporator inlet and condenser influence the system performance.

5. ACKNOWLEGMENTS

The authors gratefully acknowledge support of this work by Capes and CNPq for graduate and undergraduate scholarships.

6. REFERENCES

Benouali, J. and Clodic, D., 2003. Fuel consumption of mobile air conditioning: method of testing and results. In: The Earth Technology Forum, Washington, USA, pp. 1-10.

Cho H., Lee H., Park, C., 2013. “Performance characteristics of an automobile air conditioning system with internal heat exchanger using refrigerant R1234yf”. *Applied Thermal Engineering*. Vol. 61, p.563 – 569.

Jarall, S. 2012. “Study of refrigeration system with HFO-1234yf as a working fluid”. *International Journal of Refrigeration*. Vol. 35, p.1668-1677.

Lee, Y., Jung, D. 2012. “A brief performance comparison of R1234yf and R134a in a bench tester for automobile applications”. *Applied Thermal Engineering*. Vol 35, p. 240 - 242.

Lemmon, E.W., Huler, M.L., McLinden, M.O. 2013. NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties-REFPROP. Version 9.1

McLinden, M.O., Kazakov, A.F., Brown, J.S., Domanski, P.A. 2013. “A thermodynamic analysis of refrigerants: possibilities and tradeoffs for low-GWP refrigerants”. *International Journal of Refrigeration*. Vol. 38, p. 80 - 92.

Mota-Babiloni, A., Navarro-Esbri, J., Barragan-Cervera, A., Moles, F., Peris, B., 2014. “Drop-in energy performace evaluation of R1234yf and R1234ze(E) in a vapor compression system as R134a replacements”. *Applied Thermal Engineering*. Vol. 71, p.259-265.

Navarro-Esbri, J., Mendoza-Miranda, J.M., Mota-Babiloni, A., Barragan-Cervera, A., Belman-Flores, J.M. 2013a. “Experimental analysis of R1234yf as a drop-in for R134a in a vapor compression system”. *International Journal of Refrigeration*. Vol. 36, p. 870-880.

Navarro-Esbrí, J., Moles, F., Barragan-Cervera A., 2013b. “Experimental analysis of the internal heat exchanger influence on a vapour compression system performace working with R1234yf as a drop-in replacement for R134a. Experimental analysis using R134a and R1234yf”. *Applied Thermal Engineering,* Vol. 59, p.153-161.

Noetzold, J., Copetti, J.B., Macagnan, M.H. 2016. “Comparison of the optimal refrigerant charge with R134a and R1234yf on the automotive air conditioning off-road agricultural vehicles”. In *Proceedings of 10º Congresso Internacional de Ar Condicionado, Refrigeração, Aquecimento e Ventilação – Mercofrio2016*. Porto Alegre, RS.

Qi, Z., 2013.”Experimental study on evaporator performance in mobile air conditioning system using HFO-1234yf as working fluid”. *Applied Thermal Engineering*. Vol. 53, p. 124- 130.

Qi, Z., 2015”Performance improvement potentials of R1234yf mobile air conditioning system”. *International Journal of Refrigeration*. Vol. 58, p.35-40.

Rotchana, P., Suen, K.O. 2012. “Effects of refrigerant properties on refrigerant performance comparison: A review”. *International Journal of Engineering Research and Applications (IJERA*), Vol. 2 (4), p. 486-493.

SAE J1503, 1998. Surface vehicle standard: performance test for air-conditioning, heated and ventilated off-road, selfpropelled work machines. SAE, Warrendale, USA.

Spatz, M. Minor, B. 2008. "HFO-1234yf low GWP refrigerant: a global sustainable solution for mobile air conditioning". *In Proceedings of SAE 2008-Alternate Refrigeration Systems Symposium*. Scottsdale, AZ.

Tanaka, K., Higashi, Y. 2010. "Thermodynamic properties of HFO-1234yf". *International Journal of Refrigeration*. Vol. 33, p. 474 - 479.

Zilio, C., Brown, J.S., Schiochet, S., Cavallini, A. 2011. "The refrigerant R1234yf in air conditioning systems". *Energy*. Vol. 36, p. 6110 - 6120.

Wang, C.C., 2013. "An overview for the heat transfer performance of HFO-1234yf". *Renewable Sustain. Energy Rev.* Vol. 19, p. 444 - 453.

7. RESPONSIBILITY NOTICE

The author(s) is (are) the only responsible for the printed material included in this paper.