COMPARISON OF THE OPTIMAL REFRIGERANT CHARGE WITH R134A AND R1234YF ON THE AUTOMOTIVE AIR CONDITIONING OFF-ROAD AGRICULTURAL VEHICLES

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F6 - Materials and Refrigerants

Abstract. *Automotive air conditioning for agricultural machinery presents some particular characteristics compared with others air conditioning systems for vehicles: operates under highly transient climatic conditions, relatively high cooling capacity to provide a rapid cool down. In this work, experiments were carried out to compare the performance of a typical AC for off-road vehicles using R-134a in a drop-in to R-1234yf. This system was installed in an experimental setup and the main parameters were measured. The optimum refrigerant charge for each refrigerant was established and the cycle parameters like evaporator capacity, COP, compressor power, compressor discharge pressure, degree of subcooling and superheat was compared. The experimental results indicate that for air inlet conditions in the evaporator and condenser at 40 ºC and a relative humidity of the 43%, the system running with the R-1234yf, has slightly lower values of: evaporator capacity, COP and a higher superheat in the evaporator, reach the optimal refrigerant charge with 50g more than with R-134a.*

Keywords: *R-1234yf; Automotive air conditioning; Drop-in; Off-Road Agricultural Machinery; Tractors*

1. INTRODUCTION

The automotive air conditioners manufactured before 1995 used R-12 as refrigerant, and after implementation of the Montreal Protocol (1987), virtually all new systems using the refrigerant R-134a. In 1997, the Kyoto Protocol (1997) established a gradual reduction of gases emission that contribute to the greenhouse effect, such as HFCs, including the R-134a.

According to the report RTOC (2014), the European Union and the United States already have regulations that influence the projects of the Mobile Air Conditioning (MAC), while other countries with a high density of vehicles are working in the same direction. The Directive of European Community 2006/40/EC (EU DIRECTIVE, 2006) also known as F-gas, applies to light passenger vehicles and commercial, specifying that from 2013 all new vehicle models with MAC not can be manufactured with fluorinated refrigerants with GWP higher than 150, extending to all vehicles produced from 2017. Also, the United States published a regulation EPA SNAP (RTOC, 2014), prohibiting the use of R-134a in cars from 2021.

Some refrigerants meet the GWP index and are being considered as possible replacements to apply in MAC systems, like R-744 (CO_2), and R1234yf. The challenge is to develop systems that meet legislation with similar energy efficiencies to systems with R-134a, and do not require major changes in the design of system components in relation to the currently used systems. The R-1234yf refrigerant has a GWP of 4 and a lower boiling temperature than R134a, approximately 3.7 °C less.

Optimum refrigerant charge and its effects on the performance of the refrigerating system had receiving a considerable attention in the last ten years mainly caused by the charge minimization studies to develop refrigeration systems with low environmental impact and the effects of refrigerant leakage in both system performance and the environment.

The refrigerant charge in an automotive air conditioning can change during its time of operation. This can happen due to small leaks in permeable seals and hoses in the refrigerant lines. A study by Clodic et al. (2007) demonstrated that refrigerant leakage in automotive a/c systems might be about 10 g/year. These leaks, in addition to the associated environmental problems, negatively affect the system performance, its stability and durability. A study realized by Tanino et al. (1988) showed that a reduction in refrigerant charge will cause reduced cooling capacity, reduced liquid line sub cooling, increased suction line superheat, increased compressor inlet temperature, increased compressor outlet temperature and decreased compressor outlet pressure. Apart from the problems caused by refrigerant leakage during operation of the system, the optimal charge setting can vary from one manufacturer to another. Accordingly Collins and Miller (1996), a typical automotive a/c test specification defines the optimum charge as the charge for which the refrigerant temperatures at the inlet and the outlet of the evaporator first "cross over". This method provides an indication that the refrigerant is in a saturated state for the entire length of the evaporator and heat is being transferred from the airflow, according to the specific heat of vaporization of the refrigerant. The charge determination based on refrigerant weight can be applied to any system, through equipment manufacturer information but must be adjusted for different refrigerant line lengths. Superheating method is particularly recommended for systems with expansion devices with fixed orifice and the sub cooling method for systems with expansion valves. Anyway, the system charge is always defined from a set of fixed operating conditions, internal and external and, particularly for automotive a/c systems, a given compressor speed. Optimum refrigerant charge and its effects on the performance of the refrigerating system had receiving a considerable interest in the last ten years mainly caused by the charge minimization studies to develop refrigeration systems with low environmental impact and the effects of refrigerant leakage in both system performance and the environment.

Farzad and O´Neal (1991) reported the effect of various refrigerant charges on the performance of residential air conditioner systems with capillary tube expansion showing that the degradation of performance is larger for undercharging than for overcharging. Kaynakli and Horuz (2003) experimentally investigated the performance of an automotive a/c summited to different condenser and evaporator inlet temperatures and compressor speed. It was found that condenser temperature increase with increasing ambient temperature and cooling capacity increases with condenser temperature increase, reaching a maximum when reduces again.

By increasing compressor speed, there is an increased in the cooling capacity, and compressor power resulting, however, in a decrease in the COP. Huyghe (2011) tested an automotive a/c using R134a in a range of 25 to 100 % of nominal system charge, accordingly to SAE J2765 (2008) for fixed displacement and external variable displacement compressors on a modern light-duty MAC system, with a cross charge thermostatic expansion valve. The main findings of the experiments are: the cooling capacity increases, the outlet air temperature of the panel decreases and the compressor power consumption increases from the condition of low charge to the rated charge. It should be mentioned that since 75 % up to the nominal charge, its effect on the system performance is negligible for both compressors.

Wang and Gu (2004) experimentally investigated the performance of an automotive air conditioning system through measurements of two-phase flow. The main results were: the total mass flow rate increases with increased refrigerant charge and the rise of the temperatures on the air side in the evaporator and condenser. The cooling capacity does not vary with change in refrigerant charge but increases with the evaporator air side temperature and decreases with increasing temperature in the condenser. The coefficient of performance decreases with increased refrigerant charge.

The purpose of this work is to investigate the influence of the refrigerant charge on the steady-state operation of an automotive air conditioning. An automotive a/c of 6.4 kW nominal capacity, for off-road vehicle, originally developed to work with R-134a refrigerant, in a direct replacement for refrigerant R-1234yf, was installed in an experimental setup and the main parameters of operation were measured. Compare the maximum capacity cooling system, COP, power consumed, to the system working with each refrigerants, and establishing the optimum refrigerant charge for each of the two refrigerants.

2. EXPERIMENTAL SETUP AND METHODOLOGY

Modern tractors are complex and highly efficient systems, cabins with air conditioned are increasingly employed in this type of machines. On the other hand a closed tractor cabin, can become unpleasant, unbearable and even dangerous (Ružić and Časnji, 2011), for this reason is very important a high performance of the air conditioning system. Fig. 1a shows the main thermal loads involved in the vicinity of a typical tractor cab.

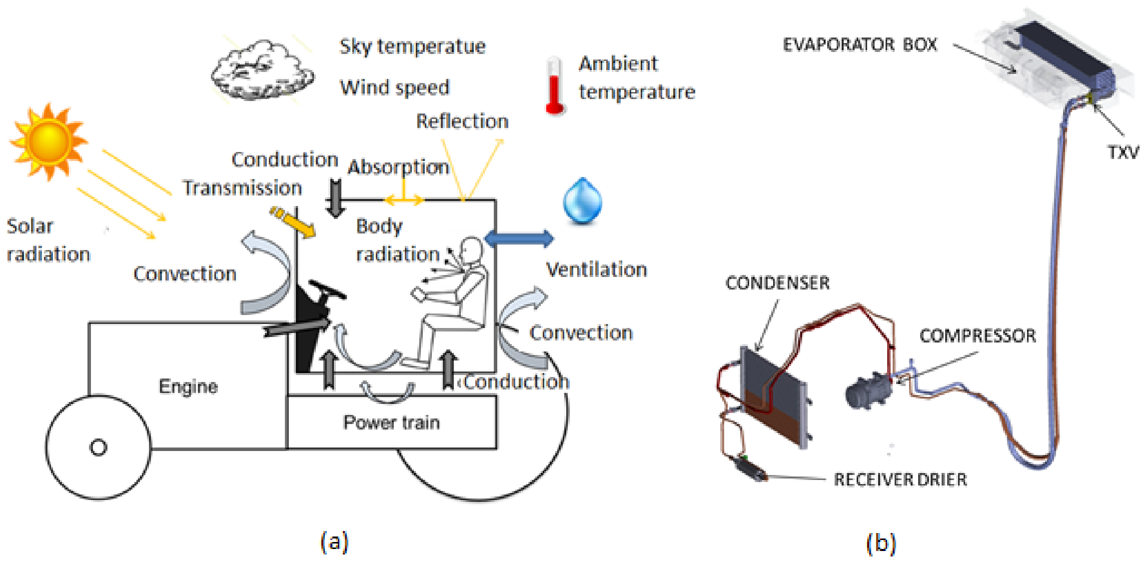


Figure 1.Thermal loads in a typical agricultural tractor cab (a), and a typical tractor air conditioning loop (b).

The test bench, according the scheme of Fig. 2, consists of a wind tunnel and a calorimeter, built according to ANSI/ASHRAE 51-1999 (2001), for evaluation of condensers and evaporators, respectively, and a typical automotive air conditioning system. Automotive swash plate compressor (Sanden SD7H15) with a fixed displacement of 154.9 cc/rev performs the mechanical compression work. This type of compressor is directly driven by a belt. As an alternative to combustion engine, a three-phase electric motor was used, with nominal power of 11kW. The motor is connected to a frequency inverter (Siemens MIDI Master vector), to vary the speed of the tests, simulating its normal working condition in vehicles.

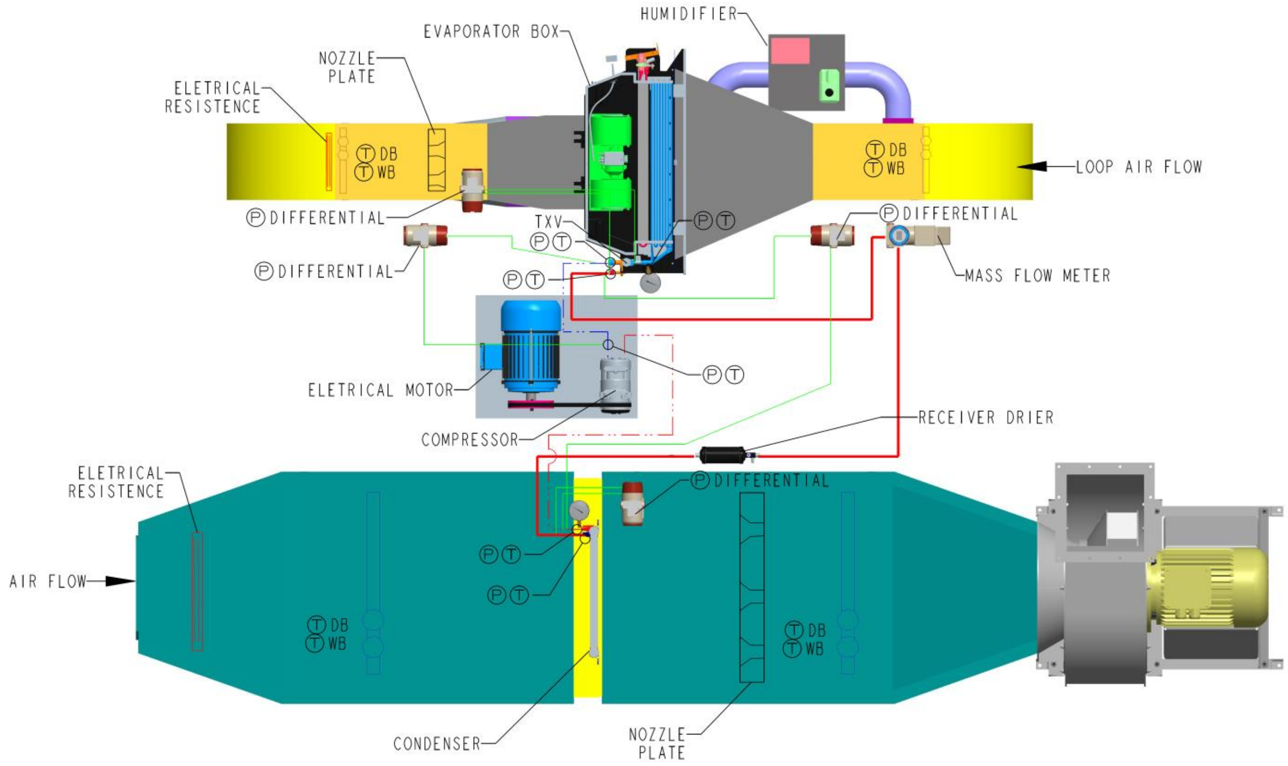


Figure 2.Schematic diagram of the experimental bench

The condenser is a multiport aluminum and flat tubes, with microchannels and parallel flow. A face area of 0.1501 m², and maximum capacity heat rejection of 10.4 kW. The refrigerant flows through the channels. The number of passes through tubes is variable, the first pass has the largest number of pipes decreasing in subsequent passes, and this feature reduces load loss resulting from the refrigerant inflow condition: superheated steam with a high specific volume. Since this condenser has a smaller frontal area relative to the airflow, compared to the finned-tube condensers, it has a low pressure loss on the air side (Copetti et al., 2008). In addition, it takes some advantages such as lower weight, high efficiency and lower volume (Liang et al., 2010).

The thermostatic expansion valve (TXV), also known as block valve, is widely used by the automotive industry. This valve model was developed to work with R-134a refrigerant, your body is made of anodized aluminum, O-rings and other sealing internal components are of NBR rubber and internal moving parts such as spring, pass ball, shaft and diaphragm are all stainless steel. The nominal capacity of the valve is 1.5 TR.

The evaporator box used in this study is a plastic case manufactured by injection molding process, inside the box are mounted evaporator, the centrifugal fan (Sirroco) double shaft with a blower on each axle and auxiliary components as resistance of the fan speeds, and anti-freezing thermostat. The maximum cooling capacity is 6.77 kW at an air flow of 650.4 kg/h, the evaporator is a finned-tube, the face area of 0.062 m².

The condenser is placed in the wind tunnel and the evaporator box in the calorimeter, which provide controlled ambient temperature, relative humidity and air flow rate. The air blower in the evaporator side was controlled by a DC Kepco BOP 20-20M Power Supply and in the condenser side by a Danfoss frequency inverter.

The temperatures measurements were made by sensors type PT 100, located as described in Fig. 2. The absolute pressure in the inlet of the evaporator and the condenser was made by two Keller pressure transmitters, model PA 33X, and the measures the pressure difference between the inlet and outlet of these two heat exchangers were made with two differential pressure transmitter ABB 600T and two differential pressure transmitter Yokogauwa. The refrigerant mass flow rate was measured by an Emerson Micro Motion Coriolis transmitter, located in the liquid line, after the receiver drier. The airside volumetric flow rate, and consequently the air velocity, was measured using two plate nozzles and two pressure transmitters: an Omega differential pressure transmitter in the evaporator side and a Dwyer pressure transmitter in the condenser side. The constants for the correlation equation between pressure drop on the nozzle plates and volumetric flow rate were adjusted from measurements of air velocity before the nozzles plates with a hot wire anemometer.

The tunnel and recirculation calorimeter were constructed according to ANSI/ASHRAE 41.2-1987 (1992), and the temperatures in the airside are made accordingly ANSI/ASHRAE 41.1-1986 (2006).

The procedure for comparative experimental tests of the MAC system was developed to represent the actual conditions of application of the system to the thermal comfort of tractor cabins. Reference standards for thermal comfort approval criteria in cabins of agricultural machinery and trucks were used. As a field data directly to the tractor model with a/c system studied in this work.

The standards ISO 10263-4 (2009) and SAE J1503 (1998) define approval criteria for a/c off-road machine systems (tractors, grain harvesters and agricultural machinery self-propelled generally), both specify a room temperature at least test 38 °C and 40% of humidity. Moreover, they specify a temperature drop at least 25°C inside the cabin, in the maximum interval of one hour from the a/c activation, the standard recommends too the heat load simulation of solar radiation of 950 W/m².

SAE J2646 (2011) shows the approval requirements for the a/c system a little more demanding thermal comfort criterion compared to the previous ones. It uses a value to room temperature of 43.3 °C at 40% of humidity, specifying a temperature drop up at least 23.9 °C at the end of a test time. These three standards to thermal comfort criteria for agricultural machinery and truck manufacturers, can also be used as standards for regulators to define conditions of the operator workplace, were taken as a reference in setting the temperature and humidity conditions for analysis of the studied system.

Tests with both refrigerants were performed in the same conditions of temperature and air flow at the inlet evaporator and condenser, and also for compressor speeds. The system with R-134a was first evaluated and is the standard of comparison. To find the optimal charge of each refrigerant studied, we used the model proposed by SAE J2765 (2008). This standard applies to automotive a/c systems and also for systems using compressors electrically driven, similar to those applied in hybrid vehicles. The temperature of the room air should be stabilized at 28 ± 2 °C. The compressor should operate at its maximum volumetric displacement, and this value is achieved with speed higher than 3500 RPM. The inlet temperatures to the condenser and evaporator are fixed at 40 ± 0.5 °C, the air velocity at 6 ± 0.18 m/s, and humidity on the evaporator inlet of 43 ± 3%.

The initial refrigerant charge was 700g and the system worked for at least 10 minutes under the above conditions before starting the measurements to ensure the steady-state operation (temperatures of air in the inlet of the condenser and the evaporator should remain within ± 0.5 °C) during the determination of charge.

The optimum refrigerant charge was found for a minimum charge able to reach the higher cooling capacity and lowest outlet air temperature of the evaporator, with the compressor discharge pressure less than 1800 kPa and a minimum subcooling of 5°C.

The evaporator cooling capacity, Q_E, and the compressor power consumption, W_C, were calculated by Eqs. (1) and (2),

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

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

where $\dot{m}$ is the refrigerant mass flow rate, $h_{e,i}$ and $h_{e,o}$ are the inlet and outlet evaporator enthalpies, $h_{c,i}$ and $h_{c,o}$ are the inlet and outlet compressor enthalpies.

Then, the *COP* of the system was calculated accordingly Eq. (3),

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

The subcooling temperature, ΔT_{sc} evaluated at the condenser outlet and superheating temperature ΔT_{sh} , evaluated at the evaporator outlet are defined accordingly Eqs (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 saturation temperature at condensing pressure, T_{liq} is the refrigerant temperature in the liquid line, $T_{sat,e}$ is the saturation temperature at evaporating pressure and T_{vap} is the refrigerant temperature in the suction line, the thermodynamic properties of R-134a and R-1234yf were obtained from REFPROP (Lemmon and McLinden, 2007).

3. EXPERIMENTAL RESULTS AND DISCUSSION

In order to obtain experimental data to compare the a/c cycle performance with both refrigerants, more than 80 steady-state tests were carried out with different refrigerant charges.

Figure 3a shows the variation of the evaporator cooling capacity as function of the refrigerant charge in the range of 700 to 1450 g, for both refrigerants. The cooling capacity, Q_e, increases with the increase of refrigerant charge, remaining almost constant for refrigerant charge higher than 1100 g and 1150 g for R-134a and R-1234yf, respectively. It can also be seen that maximum cooling capacity remained fairly constant around 5.2 kW for R-134a and 4.8 kW for R-1234yf, a capacity 7.7% less than the base line.

The result is similar that found by Zilio et al. (2011), which analyzed the drop-in of the R-1234yf in a MAC of a European compact automobile. The result showed a smaller cooling capacity for ambient temperature of the 35°C. Navarro et al. (2013), in your experimental analysis comparing the cooling performance from R-1234yf as R-134a, also found a smaller performance around 9%.

The compressor power consumption, WC, for each refrigerant charge is showed in Fig. 3b. The power consumption increases proportionally to both refrigerants with the increase of the refrigerant charge, and it is slightly smaller with R-1234yf until reach 1100 g. At this point, for R-134a the compressor power consumption stabilizes, but it keeps increasing for R-1234yf, until stabilizes in 1150 g.

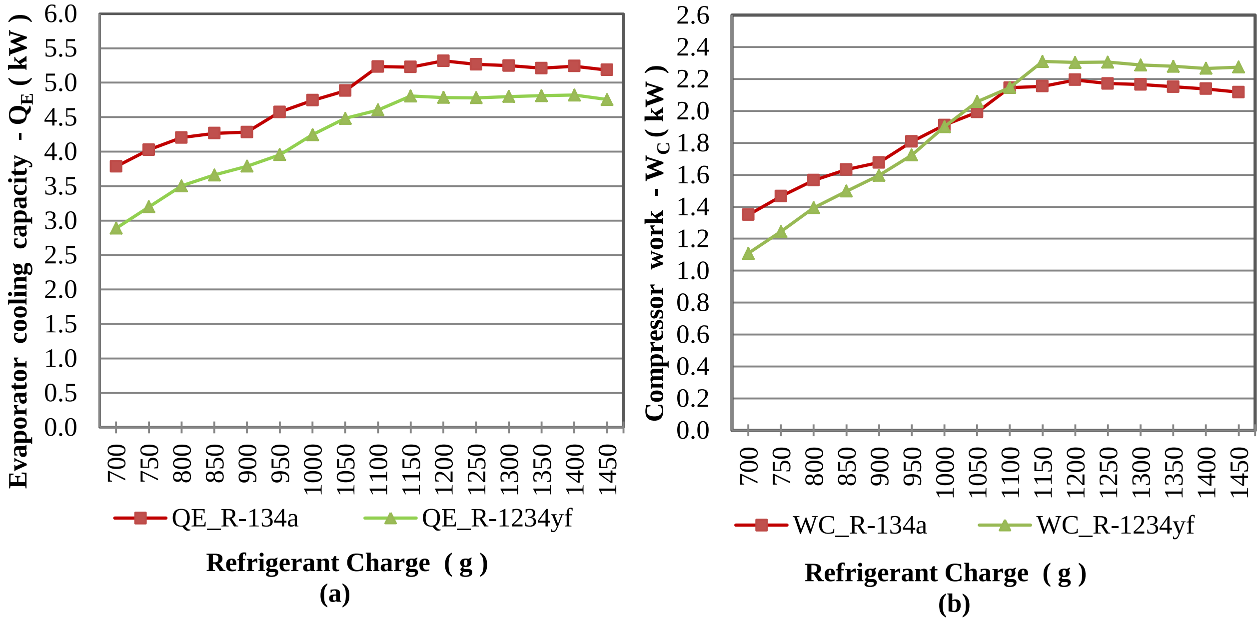


Figure 3 Evaporator cooling capacity, Q_E, (a), and Compressor work, W_C, (b) for different refrigerant charge.

Now turn your attention again to Fig. 3a and note that the cooling performance with R-134a also stabilizes with the refrigerant charge of the 1100 g. This is due the TXV setting, which is adjusted for pressures drop in the system that originally works with this refrigerant. Zilio et al. (2011), in your work proposed a tuning in the TXV to improve the cooling performance and COP of the system working in a drop-in application with R-1234yf and consequently reduced the compressor power consumption.

The COP for both refrigerants for different refrigerant charges is showed in Figure 4. The results show that there is a decrease in the COP of the system as increasing refrigerant charge. The system with R1234yf showed a COP slightly lower, between 7 and 14%. This result is consistent with the value found by Cho et al. (2013), that analyze an a/c system, working under different compressor speeds. They also found a lower COP, around of 4.5%, as a proposal to improving the system the authors tested the system with an additional internal heat exchanger. This change improved, and the COP was lower just 2.9% that an original system with R-134a.

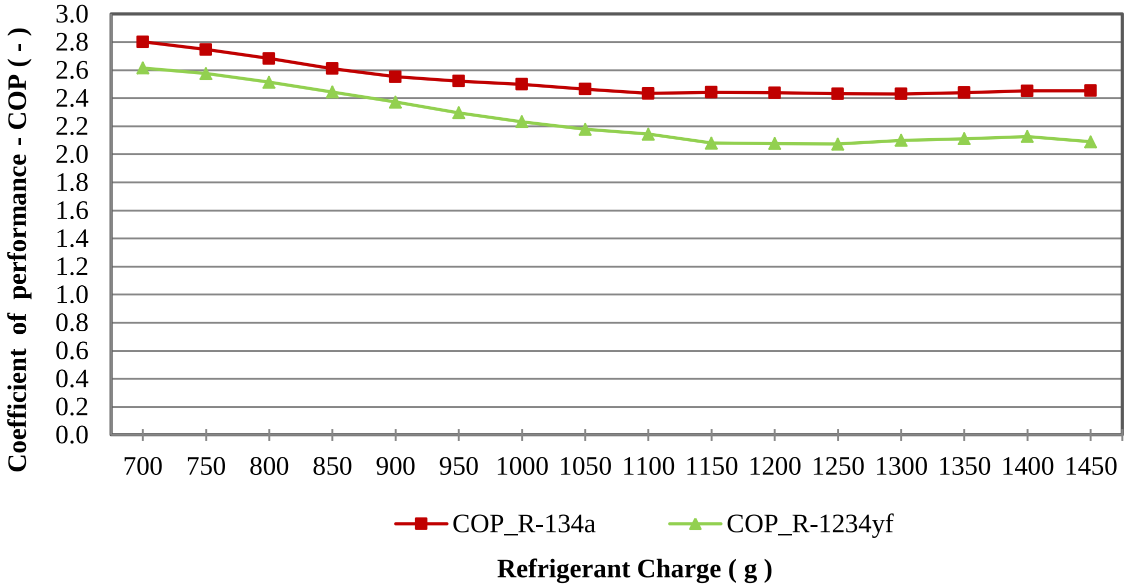


Figure 4 Coefficient of performance, COP, for different refrigerant charges

The effect of the refrigerant charge in the air-side evaporator outlet temperature is shown in Fig. 5. With increasing the refrigerant charge, gradually the temperature of the air at the evaporator outlet decreases, until stabilize for refrigerant charges higher than 1100 g.

Figure 6a shows the subcooling temperature at condenser exit as a function of refrigerant charge. The subcooling increases with increasing of refrigerant charge. For refrigerant charge until 1100 g the subcooling temperature for R-1234yf it is slightly higher, and it inverting after this point. Figure 6b shows the refrigerant superheating temperature behavior. The superheating decreases due to the increasing of refrigerant charge. The stabilization of superheating temperature occurs for the same value of refrigerant charge where stabilizes the subcooling. As in Fig. 6b show, the degree of superheating for R-134a stabilizes at approximately 4 °C, while for the R-1234yf at around 6 ºC. This difference occurs due to the TXV to be adjusted to work with R134a.

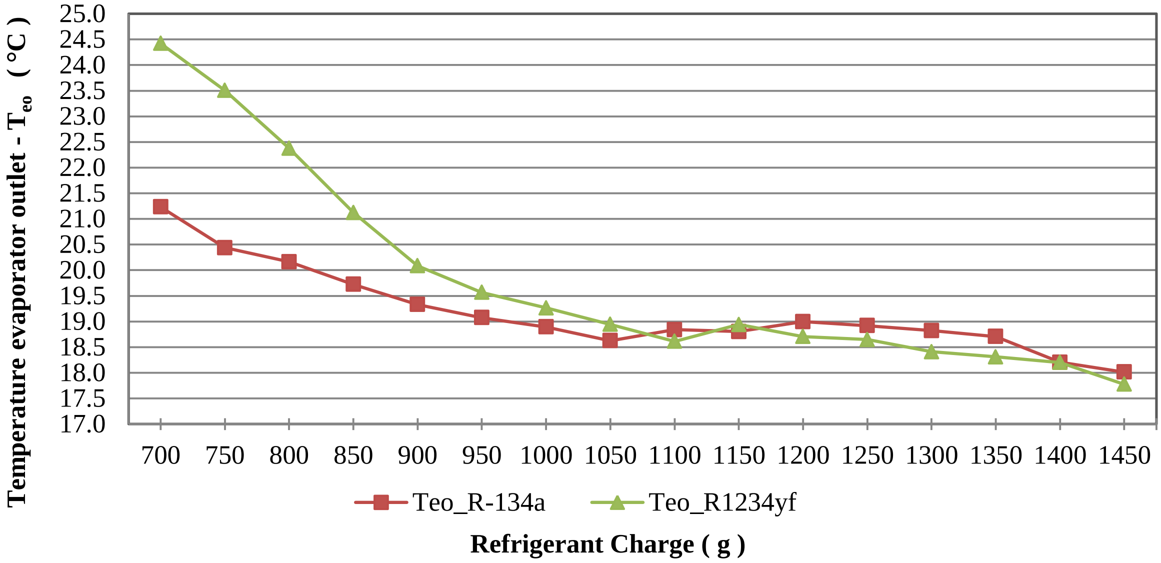


Figure 5 Evaporator outlet dry bulb temperature – T_{eo}, for different refrigerant charge.

In your work Collins and Miller (1996), show the definition from the automotive air conditioning industry to define the optimum charge as the refrigerant charge as which the refrigerant temperatures at the inlet and the outlet of the evaporator first "cross over." The "cross over" point is an indication that evaporation heat transfer is occurring throughout the evaporator, and heat is being transferred from the air stream according to the specific heat of vaporization of the refrigerant.

Although the COP is not optimized by this procedure, the cooling capacity, Q_E, is very nearly maximized. Analyzing the data from the system working with R-134a and shown in Fig 7a, can be defined your optimal refrigerant charge with 1100 g, this is the value of the maximum Q_E and COP, reached with a minimum refrigerant charge. The parameters values set for allowable compressor discharge pressure and refrigerant subcooling temperature also can be attended. In Fig 7b, can be saw that the optimal refrigerant charge for the system, working with R-1234yf is reached with a refrigerant charge of the 1150g, slightly larger than with R-134a. However the compressor discharge pressure is lower. The air side temperature in the evaporator outlet is approximately the same.

Cho et al. (2013) in your study of a/c system performance based on the optimum charge with R-134a and R-1234yf found a different result. The optimum charge in each system was based on the charging amount in which the system COP is the greatest. The comparison of the respective optimum charging amount revealed that the optimum charge amount of the R-134a refrigeration system was 750 g, and with R-1234yf refrigeration system was 675 g which is smaller than R134a by 10%.

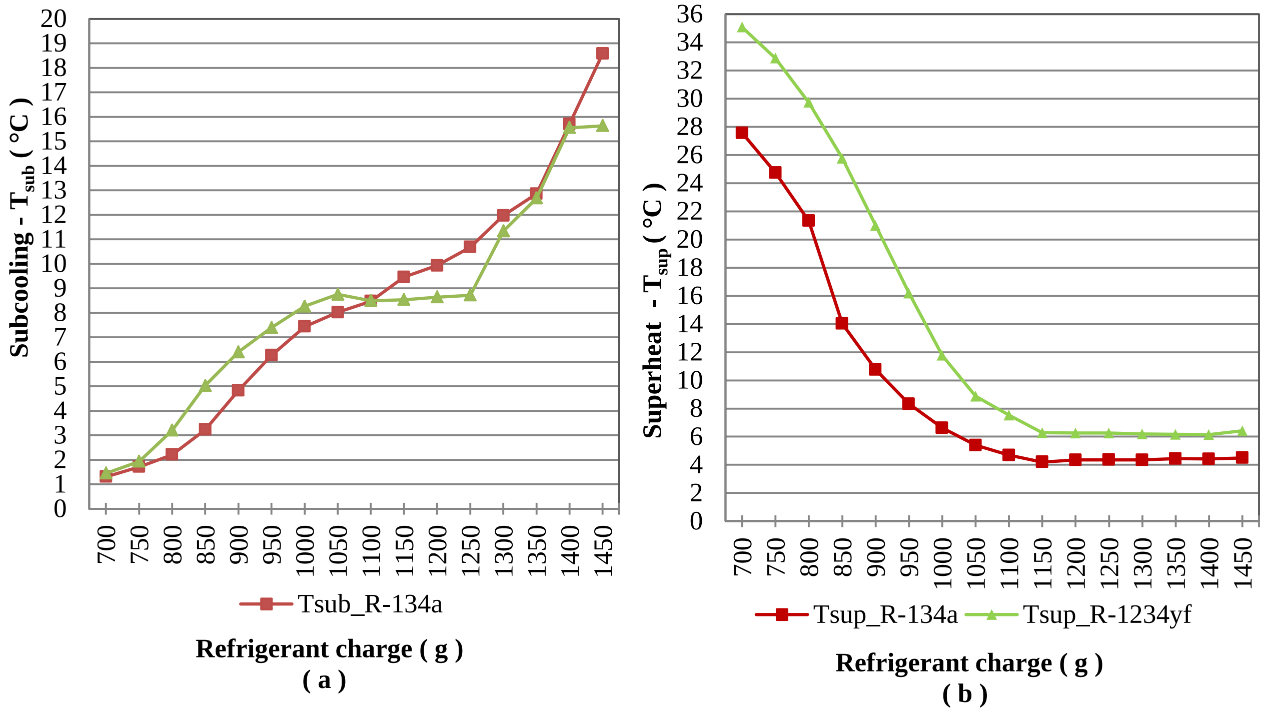
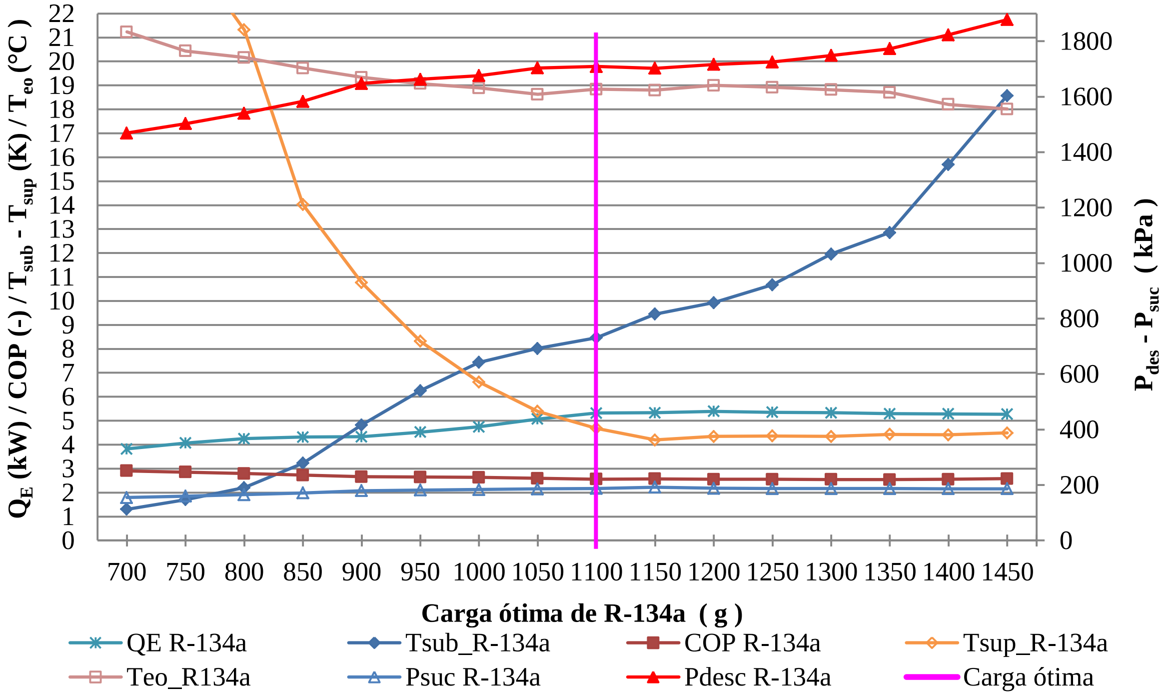
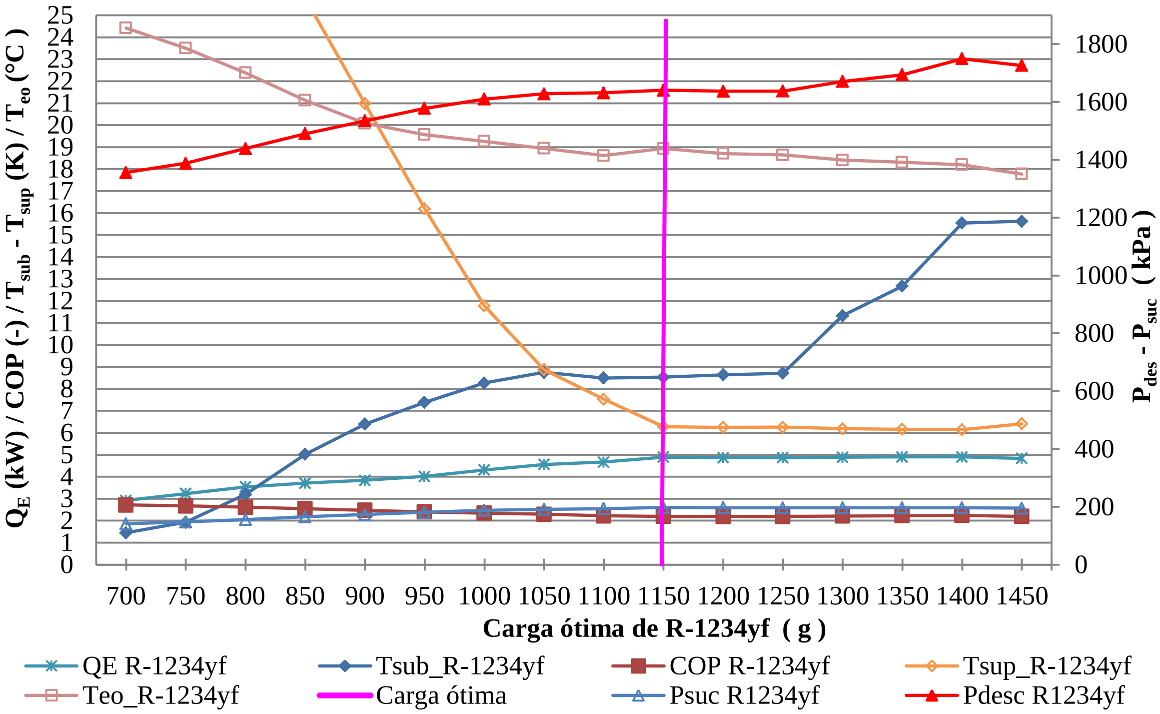


Figure 6 Subcooling temperature (a), superheating temperature (b) for different refrigerant charge.



(a)



(b)

Figure 7 Behavior of some a/c system parameters in a refrigerant charge determination procedure for R-134a (a) and R-1234yf (b).

4 CONCLUSIONS

In this study a typical automotive air conditioning system used in agricultural tractors was analyzed experimentally to compare the main parameters of the cooling system efficiency.

The system was evaluated in extreme application conditions at temperatures of 40 °C and 43% humidity, firstly with R-134a, originally used in this kind of system and after in a drop-in replacement to R-1234yf. The refrigerant charges were varied in the range of 700 to 1400 g.

The cooling capacity, compressor power consumption and the COP were slightly favorable to the system working with refrigerant R-134a, the air temperature at the evaporator outlet showed very close results for both refrigerants.

The subcooling temperature was slightly higher in the system working with R-1234yf, with charge smaller than 1100 g, and less than the system with R-134a refrigerant with more charge.

The superheating temperature of the system working with R1234yf was higher that the values for R-134a to all charge amount tested.

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