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ANALYSIS OF THE INFLUENCE OF REFRIGERANT CHARGE AND COMPRESSOR DUTY CYCLE IN AN AUTOMOTIVE AIR CONDITIONING SYSTEM

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Abstract. *An automotive air conditioning system fully equipped with a compressor, condenser, evaporator, a box type thermostatic expansion valve and a filter drier receiver is investigated experimentally, using R-134a as refrigerant. The evaporator cooling capacity, coefficient of performance, compressor power consumption, mass flow rate, pressures and temperatures at condenser and evaporator, evaporator refrigerant inlet quality and evaporator pressure drop are measured and analyzed to quantify the influence of the refrigerant charge and compressor speed on the steady-state operation, simulating realistic conditions for off-road vehicles. The compressor speed proved to be the most important parameter on system performance. The influence of the refrigerant charge in the system, in the range used on the experiments, not showed conclusive results. Anyway, the charge of 1450 g refrigerant currently used in actual systems might be considered appropriate for this application.*

Keywords: *automotive AC, refrigerant charge, compressor duty cycle, system capacity*

1. INTRODUCTION

Automotive air conditioning presents some particular characteristics when compared with others stationary a/c systems: adjustable air velocity and temperature over a wide range of conditions; relatively high cooling capacity to meet high thermal loads and provide a rapid cool down of the passenger compartment; operates under highly transient climatic conditions; the compressor duty cycle is directly related to the vehicle speed; operates in an environment subject to severe vibration and the connection between the parts of the system is performed through hoses. These operating conditions are even more severe for air conditioning systems in off-road agricultural machinery, like tractors and combine harvesters.

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 on the order of 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 the following changes in a/c systems performance: reduced cooling capacity, reduced liquid line subcooling, 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 to 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. Other methods of refrigerant charge have been described by Houcek and Thedford

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(1984), like: weighing the refrigerant during the charge, controlling the superheating in the evaporator outlet, controlling the subcooling in the condenser exit, etc. 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 subcooling 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 air conditioning systems, a given compressor speed (Temple, 2004).

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. Ratts and Brown (2000) quantified thermodynamic losses in an automotive refrigeration system as function of refrigerant charge level. This study showed that the system is more efficient as the refrigerant charge level decreases at the expense of increased refrigeration temperature and decreased refrigeration capacity. Experimentally was also observed that the compressor cycling increases, the condenser exit temperature decreases and the superheat increases as the refrigerant charge level decreases.

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 verified that the cooling capacity increases with increasing inlet temperature of the air in the condenser and the compressor speed. However, with the increasing speed of the compressor also increases their power consumption, decreasing the COP of the system.

Huyghe (2011) tested an automotive a/c using R134a in a range of 25 to 100 % of nominal system charge, accordingly to SAE J2765 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 % 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.

Some researchers have proposed analytical methods to evaluate the influence of the refrigerant charge in the system, mainly in refrigeration and heat pumps working with constant speed compressor, especially the works of Vjacheslav et al. (2001) and Corberán et al. (2011). According to these researchers, the results showed good agreement with experimental data.

The purpose of this work was to investigate the influence of the refrigerant charge and the compressor duty cycle on the steady-state operation of an automotive air conditioning. An automotive a/c of 6.4 kW nominal capacity, using R134a, was installed in an experimental setup and the main parameters of operation were measured. The refrigerant charge was varied in the range of -45 % to + 28 % of the manufacturer charge and the compressor velocity varied between 1500 to 3500 rpm in steps of 500 rpm. The effects of varying these two parameters on the evaporator capacity, mass flow rate, evaporator superheating, condenser subcooling, compressor power, COP, condenser and evaporator pressure and evaporator e condenser air side temperatures were determined.

2. METHODOLOGY AND EXPERIMENTAL SETUP

The schematic diagram of the experimental setup utilized for the measurements of the automotive a/c performance is showed in Fig. 1. The test setup consisted of a belt driven compressor, a thermostatic, box type, expansion valve, a finned coil evaporator and condenser and a filter drier receiver in the liquid line, operating with R134a. The belt driven compressor was coupled to a three phase electrical traction motor, with nominal power of 11 kW and nominal speed of 1750 rpm, controlled by a Siemens Midimaster Vector frequency inverter, to simulate compressor speeds between 1500 to 3500 rpm. The motor input power was measured by a Fluke 43B Power Quality Analyzer, controlled by the Fluke View proprietary software and the experiments data were stored on a personal computer.

The two heat exchangers were placed into two tunnel-type calorimeters, which provide precontrolled 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.

All the temperatures measurements were made by sensors type PT 100, located as described in Fig. 1. The absolute pressure in the inlet of the evaporator and the condenser was made by two Keller pressure transmitters, model PA 33X

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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.

The refrigerant mass flow rate was measured by an Emerson Micro Motion Coriolis type mass transmitter, located in the liquid line, after the filter drier receiver.

The air side 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 two recirculation calorimeters tunnels were constructed according to ANSI/ASHRAE 41.2 (1992) and the temperatures in the air side are made accordingly ANSI/ASHRAE 41.1-1986 (2006) and fully described in Souza (2011).

The refrigerant mass charge, for each experiment, was measured by a Lax Lx36575 digital electronic scale.

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.

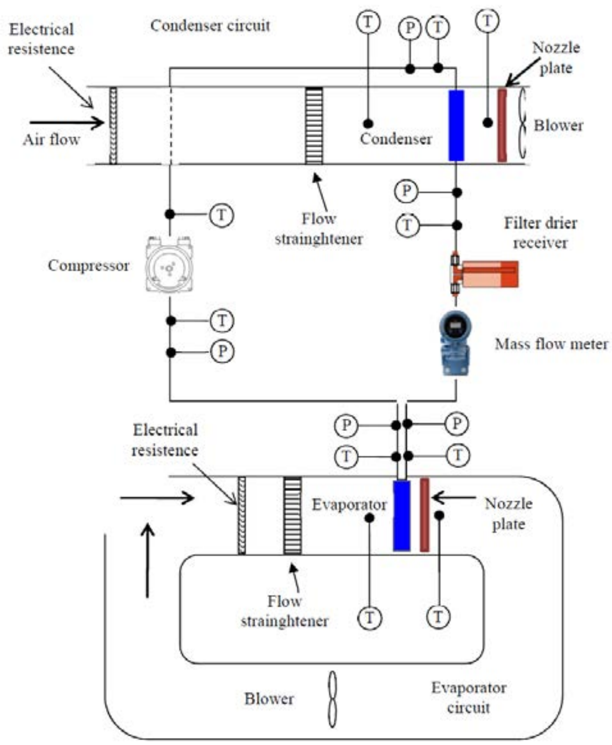

Figure 1. Schematic diagram of the experimental setup

2.1 Methodology

In order to reflect the driving conditions in summer season, the evaporator air side inlet temperature and relative humidity were set to 28 °C and 50 %, respectively (Benouali and Clodic, 2003). In the condenser air side, the temperature was set to 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 nominal refrigerant charge of the air conditioning system was 1450 g, slightly modified by the presence of the mass flow meter in the experimental setup and the additional length of the hoses in the liquid line. The initial refrigerant charge was 950 g with increments of 100 g in each experiment, rising until 1850 g. This means a range from about 65 % to about 128 % of the rated refrigerant charge. The compressor speed, measured by an optical tachometer, was varied from 1500 rpm to 3500 rpm, at intervals of 500 rpm. The above tests conditions reflect the actual operating conditions for off-road vehicles, particularly agricultural and forestry machines.

For each experiment, the air loops were started up and the evaporator air loop was warmed up, but the condenser air loop was not warmed until de refrigeration system was in operation and the compressor clutch engaged. The operating conditions of the air conditioning were considered in steady state when, during 20 minutes, the air side temperatures on the condenser and in the evaporator inlet remained between ± 0.5 °C

The evaporator heat capacity, $\dot{Q}$, and the condenser capacity were calculated by the refrigerant enthalpy method, described in Eq. (1) and (2):

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

$$\dot{Q}_c = \dot{m}_r \left(h_{c,i} - h_{c,o}\right) \tag{2}$$

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 R134a were obtained from REFPROP software (Lemmon and McLinden, 2007).

The refrigeration system COP was calculated accordingly Eq. (3):

$$COP = \frac{\dot{Q}}{P_m} \tag{3}$$

where P_m is the electric motor power consumption. There was no correction applied to the measured power at the compressor inlet due to its efficiency. 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

The experiments were carried out at different refrigerant charges and compressor speed. During the experiments, the evaporator and condenser inlet air-side temperatures were maintained constant at 28 °C and 38 °C, respectively. Figure 2 shows the variation of the evaporator cooling capacity as function of the refrigerant charge and compressor speed. The average capacity of the evaporator was around 3.5 kW showing little influence due to the refrigerant charge and a slight increment for each charge due to the increase of compressor speed.

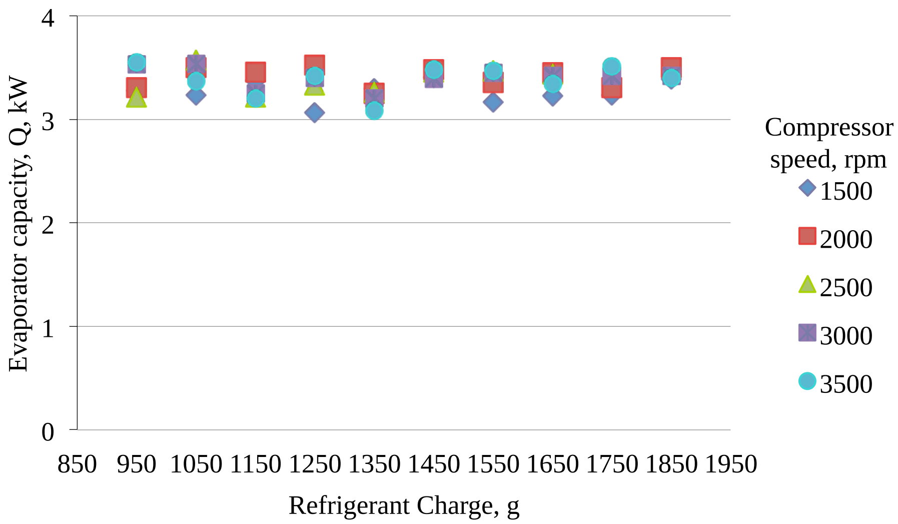

Figure 2. Evaporator cooling capacity for different refrigerant charge and compressor speed

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As shown by Kim and Braun (2010), the thermostatic expansion valve associated with the presence of the filter drier receiver, adjust the mass flow rate in response to changes in refrigerant charge. Also, the evaporator cooling capacity is limited by the heat transfer on the air side, operating at constant volumetric flow rate and with a temperature difference between input and output relatively low, as shown in Fig 10. In these range of refrigerant charges was not verified the presence of two-phase refrigerant from condenser to the expansion valve.

The compressor power consumption for each refrigerant charge and compressor speed is showed in Fig. 3. The power consumption of the compressor is significantly affected by its speed, increasing approximately 78% from lower to higher speed, but does not vary significantly as a function of refrigerant charge for the range of charges used in the experiments.

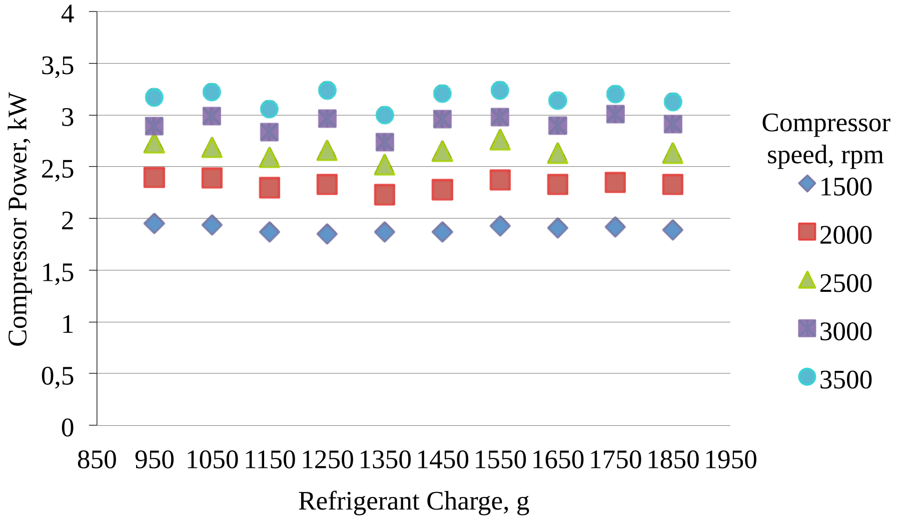

Figure 3. Compressor power consumption for different refrigerant charge and compressor speed

The behavior of the refrigerant mass rate as a function of system parameters is shown in Fig 4. It can be noted that the refrigerant mass flow rate increases with the speed of the compressor for each charge interval. Analyzing Fig. 4 is not evident the existence of a pattern of variation in the mass flow rate as a function of refrigerant charge. In the literature (for example: Kaynakli and Horuz, 2003; Choi and Kim, 2002), the effect of mass flow rate is usually represented as a function of the pressure in the condenser or in the evaporator, and not as a function of refrigerant charge. The mass flow rate in the evaporator is controlled by the superheat at the evaporator outlet, means that there is a cycling of this parameter in the operation of the system, accordingly to Fig 5, and the calculated average values as shown above, lead to misinterpretations. In the paper of Wang and Gu (2004), which were measured separately the liquid and vapor phases in the refrigerant flow, it was noted just a small increase in the mass flow rate for increasing the refrigerant charge for a constant speed compressor.

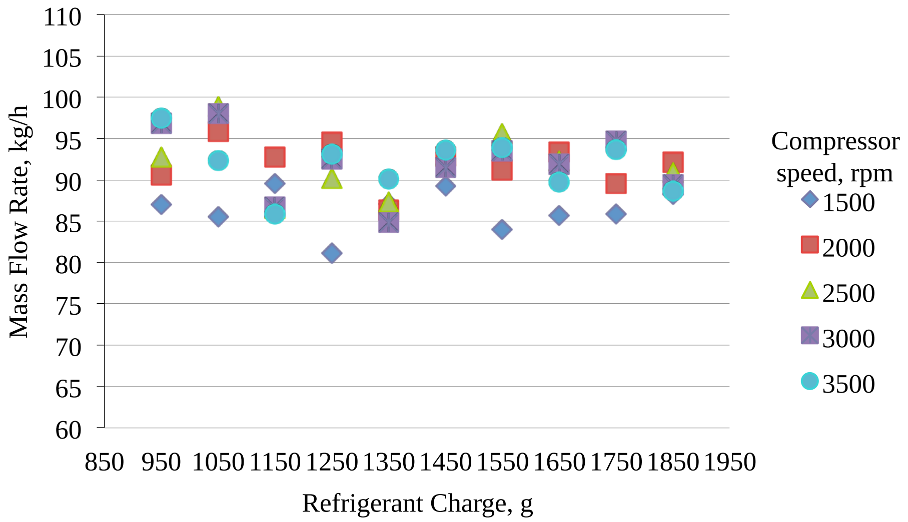

Figure 4. System refrigerant mass flow rate for different refrigerant charge and compressor speed

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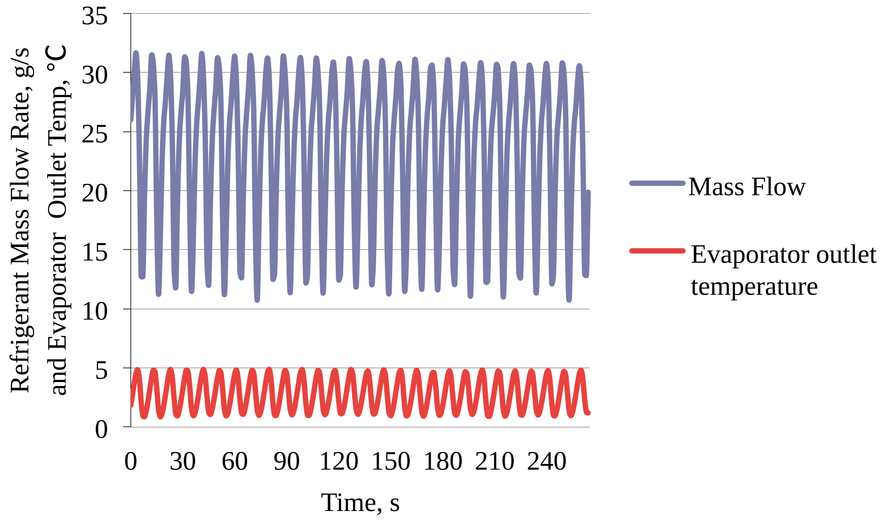

Figure 5. Cycling of the refrigerant mass flow rate and exit temperature in the evaporator

The compressor discharge temperatures, for each operational condition, are showed in Fig. 6. The refrigerant charge showed little influence to the compressor discharge temperature, showing a slight upward trend for larger charges, but that is essentially affected by the compressor velocity. This increase in temperature occurs by the increased work supplied to the compressor, as its speed increases, as shown in Fig 3.

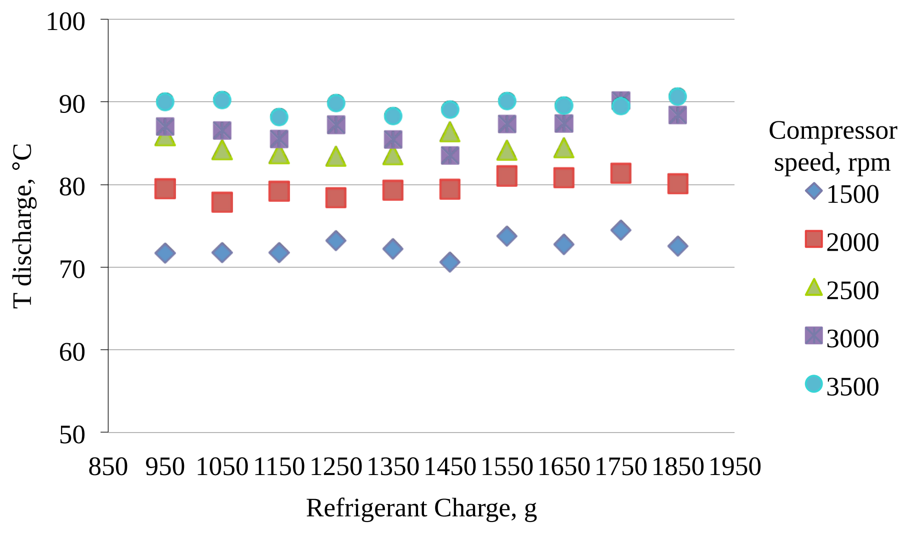

Figure 6. Compressor discharge temperature for different refrigerant charge and compressor speed

The COP of the refrigeration system, for each operational condition, is showed in Fig. 7. The results show that there is a significant decrease in the COP of the system as the compressor speed increases. This is because the power consumption of the compressor increases with its speed while the cooling capacity is little affected by this parameter. If added a trend line on the data of Fig 7 may be seen that there is a slight increase in the COP by increasing the refrigerant charge. This observation would be consistent with the work of Atik and Aktas (2011), but contrary to the work of Wang and Gu (2004), both using a capillary tube as the expansion device and R134a as refrigerant.

The absolute pressures at the condenser and evaporator inlet are shown in Fig 8 and 9. In the condenser the pressure increases with increasing speed of the compressor and in the evaporator case, the opposite effect. In the condenser there is a slight pressure increase with increased refrigerant charge and in the evaporator the inlet pressure decrease, agreeing with the data observed by Wang and Gu (2004).

The effect of the compressor speed and refrigerant charge in the air-side evaporator outlet temperature is shown in Fig 10. As the compressor speed increases, the outlet evaporator air-side temperature decreases, increasing slightly the capacity of the evaporator, as shown in Figure 2.

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Increasing the refrigerant charge gradually, the temperature of the air at the evaporator outlet increases, reaching a maximum value corresponding to the charge of 1350 g and then decrease to minimum values for larger charges. For a charge greater than 1850 g, ice formation on the external surface of the evaporator was verified.

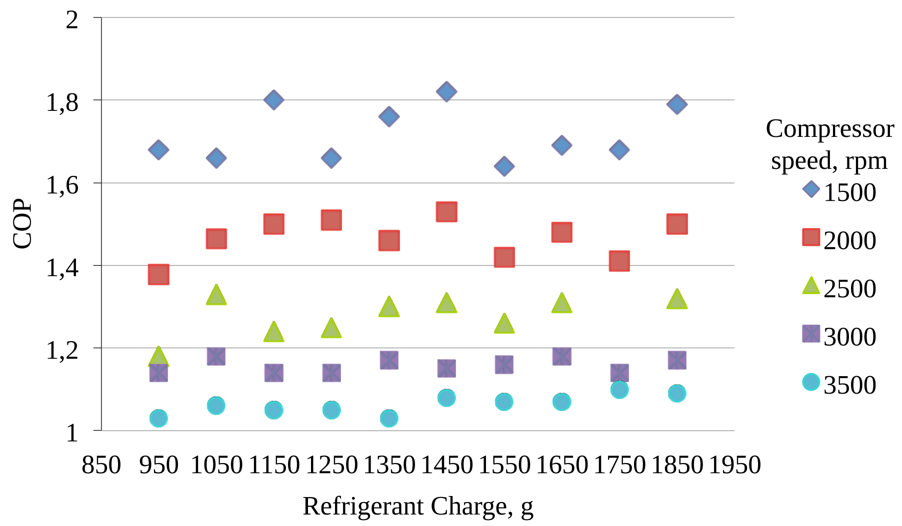

Figure 7. System COP for different refrigerant charge and compressor speed

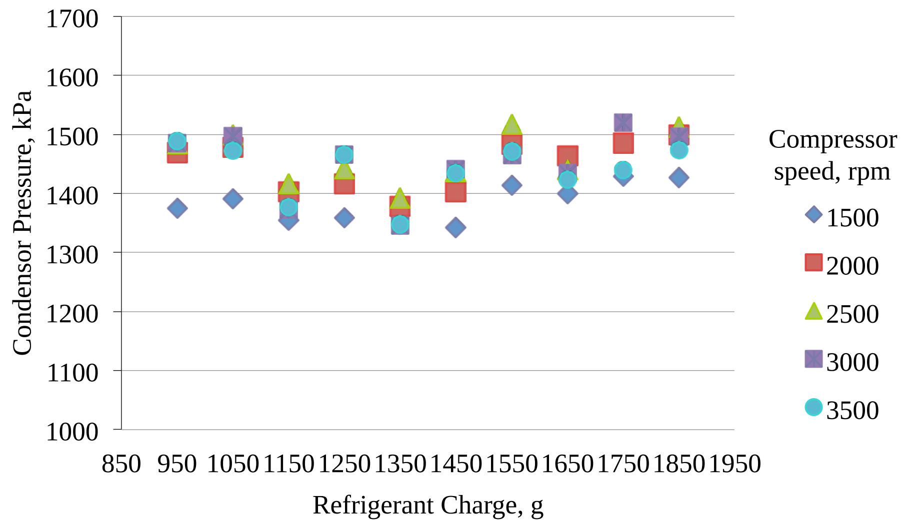

Figure 8. Condenser inlet absolute pressure for different refrigerant charge and compressor speed

The pressure drop in the refrigerant side of the evaporator is showed in Fig. 11. As shown in Figure 4, increasing the speed of the compressor increases refrigerant mass flow rate and as a result, increases the pressure drop in the evaporator. The pressure drop also increases with increased refrigerant charge.

Figure 12 shows the degree of subcooling at condenser exit as a function of refrigerant charge and compressor velocity. The degree of subcooling increases with the speed of the compressor and the refrigerant charge. As the refrigerant charge increased, the condenser pressure increased, accordingly Fig. 8, due to an accumulation of refrigerant in the high pressure side, increasing the subcooling. This behavior was also observed by Choi and Kim (2002) for the case of a refrigerating system operating with electronic expansion valve and by Farzad and O'Neal (1988) for a system using capillary tube.

Figure 12 also shows that there was no saturated vapor coming out the condenser outlet. This means there was no loss of capacity in the condenser for the overcharging.

The degree of superheating in the evaporator exit as a function of refrigerant charge and compressor velocity is showed in Fig. 13. The superheat decreases with increasing compressor velocity, due to an increase in the mass flow rate of the refrigerant but increase with the refrigerant charge. This behavior contradicts the observation of Choi and Kim (2002) for a system with capillary tube.

The refrigerant quality at the evaporator inlet is calculated considering an expansion process with constant enthalpy and the pressure and temperature at the evaporator inlet. These values are shown in Fig 14.

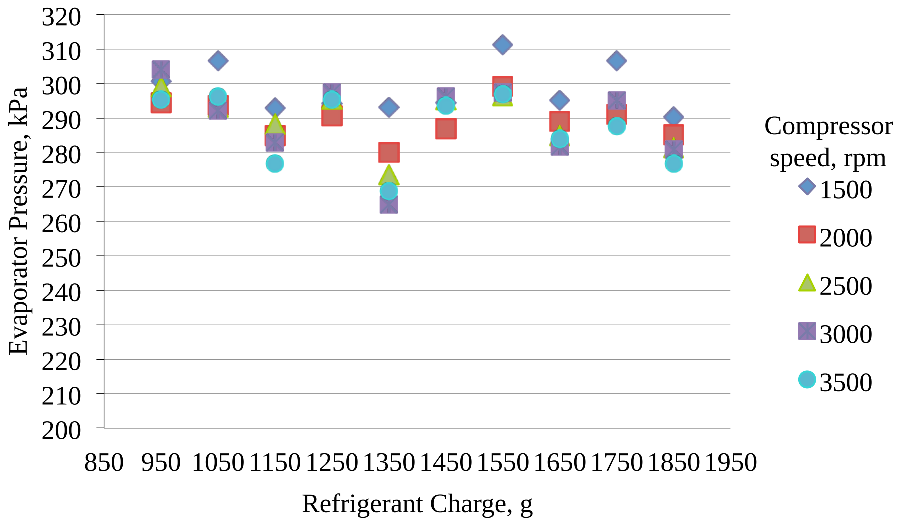

Figure 9. Evaporator inlet absolute pressure for different refrigerant charge and compressor speed

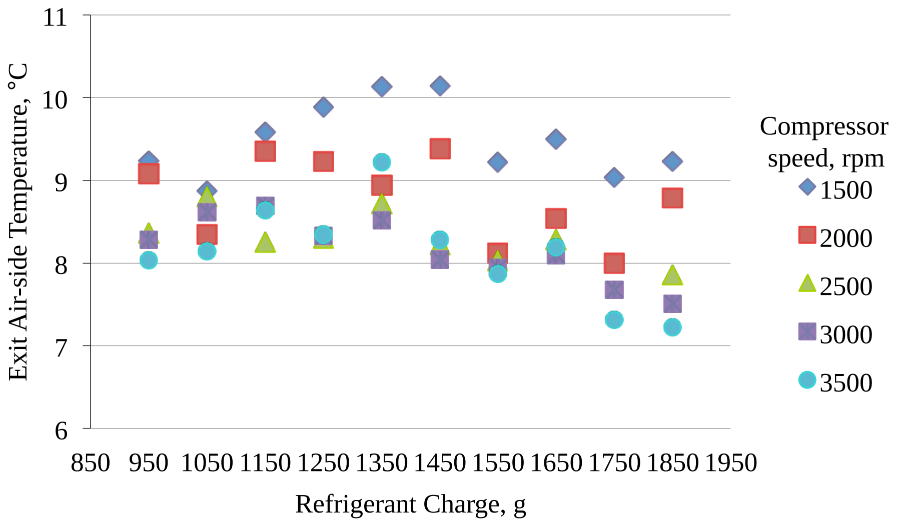

Figure 10. Evaporator air-side outlet temperature for different refrigerant charge and compressor speed

By analyzing this figure it can be noted that there is a decrease of the refrigerant quality at the evaporator inlet as the refrigerant charge increases which reduces the throttling losses in the expansion process. It can also be observed that the refrigerant quality is smaller for lower rotational speeds of the compressor.

The low refrigerant quality at the evaporator inlet associated with the lower power consumption of the compressor causes the COP of the refrigeration system, operating at low compressor velocity, to be higher than in other conditions. A similar observation was made by Ratts and Brown (2000) quantifying the thermodynamic losses of each device in a refrigeration system as a function of refrigerant charge level.

This study shows that the system is more efficient as the level of refrigerant charge decreases. However, this is accompanied by a decrease in the evaporator cooling capacity and an increase in refrigerant temperature.

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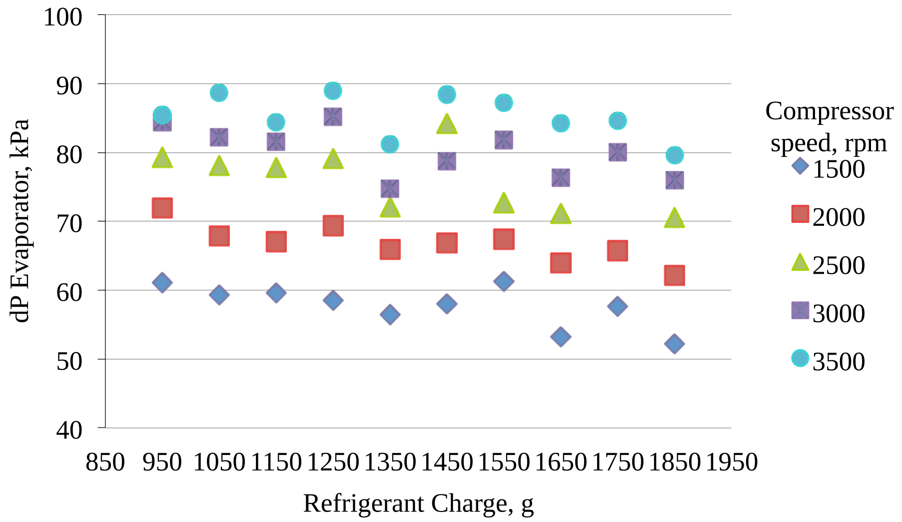

Figure 11. Evaporator pressure drop for different refrigerant charge and compressor speed

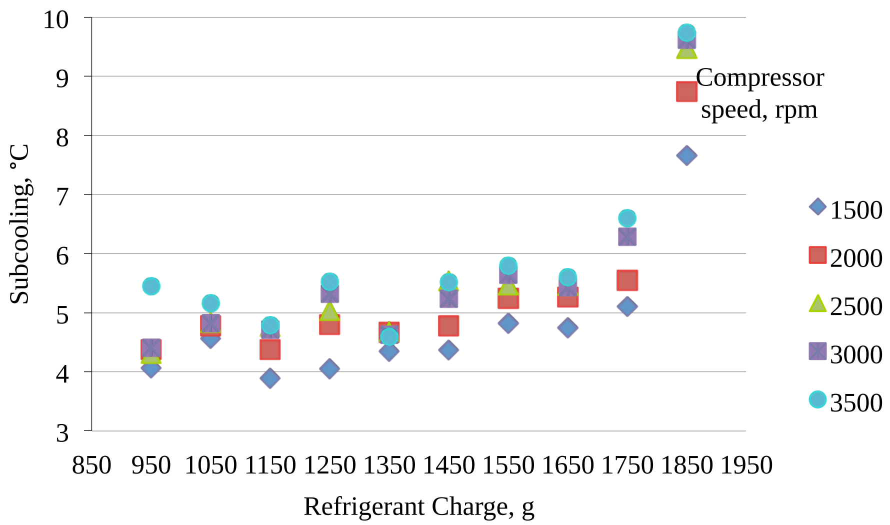

Figure 12. Liquid subcooling at condenser exit for different refrigerant charge and compressor speed

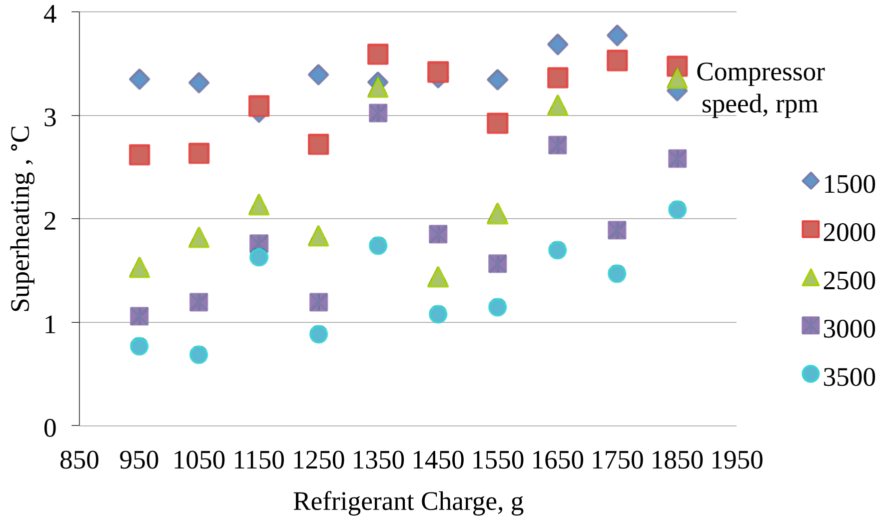

Figure 13. Vapor superheating at evaporator exit for different refrigerant charge and compressor speed

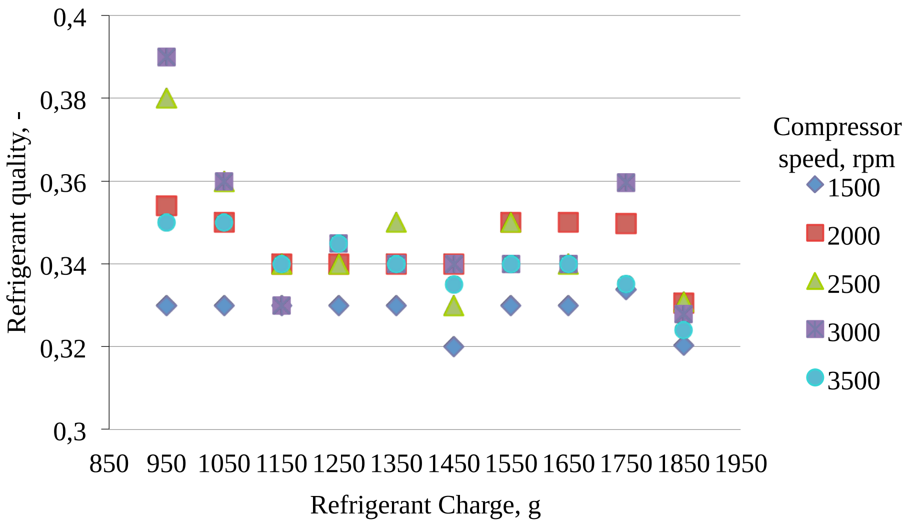

Figure 14. Refrigerant quality at evaporator inlet for different refrigerant charge and compressor speed

4. CONCLUSIONS

In this work were investigated the performance characteristics of an automotive air conditioning system, using R134a, designed for off-road applications. Several tests were performed to determine the operating parameters using an experimental setup. Charges of refrigerant between 950 and 1850 g and compressor velocities between 1500 and 3500 rpm were used. Below refrigerant charge of 950 g, bubbles were visible in the sight glass of the filter drier receiver. The main findings of this work were:

- The effects of the compressor velocity on the a/c performance is more pronounced than the refrigerant charge for the operating ranges used;
- At low compressor velocities, the power consumption is low, increasing the system COP;
- At low refrigerant charge, additional charging leads to a considerable increase in the sub cooling, while condensing pressure is hardly affected;
- For this system particularly, the COP is little influenced by the refrigerant charge. In all cases analyzed, a clear maximum in COP could not be appreciated. This may have happened by the presence of the liquid receiver before the expansion valve. Anyway, as can be seen in the literature discussed, the sensitivity to charge variation is very different for each studied systems.

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