PERFOMANCE ANALYSES OF AN AUTOMOTIBILE AIR CONDITIONING SYSTEM WITH A VARIABLE VOLUMETRIC CAPACITY COMPRESSOR (SWASH PLATE)

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Abstract. *An experimental bench was modified to evaluate the main automotive air conditioning parameters using a variable volumetric capacity compressor – swash plate - and HFC134a refrigerant. The physical parameters taking in account are: amount of refrigerant gas, and the compressor revolutions. Measurements of the refrigerant flow, the air flow and the heat loss allow to determine the energy performance.*

Keywords*: automotive air conditioning, air conditioning systems, refrigeration, HVAC.*

1. INTRODUCTION

The automobile industry is always in constant evolution. Luxurious items considered of high cost had become accessible to the consumer. An evidence of this fact is the increase of car air conditioners equipment in Brazilian market. The automotive air conditioning cost is not the only criteria selection by the industries. The automobile industry technology departments develop constantly components to reduce weight, making possible to manufacture economic vehicles. At the same time, they search to get products more energetically efficient, besides minimizing the power consumed by refrigerating compressor, making possible the automobile design evolution that normally requires a reduction of the frontal air inlet. A vehicle with 3 m³ interior volume requires approximately 5 kW of power input, while that a residential environment of 30 m³ is necessary half of this power (Petroski *et al.*, 2006).

Kaynakli and Horuz (2003) presented an experimental analysis of automotive air conditioning. They analyzed the parameters that change the system efficiency, like the compressor temperature, refrigerant temperatures and the compressor speed. Jabardo *et al.* (2001) presented a model and an experimental evaluation for an automotive air conditioning system with a variable capacity compressor. Besides the validation of the simulation model, the speed variation compressor effect, the gas amount in the system and the condenser and the evaporator air temperature are analyzed. Li *et al.* (2003) used an electronic valve expansion to compare a conventional controller PID and a Fuzzy PID. The Fuzzy PID presented good results in the gas flow control. Petroski *et al.* (2006) presented an experimental bench developed to evaluate the main automotive air conditioning parameters. The physical parameters measured are: refrigerant gas amount, the evaporation capacity, the performance coefficient and the influence of the environment temperature.

In this work is presented the experimental bench changes on the apparatus presented by Petroski *et al.* (2006). Measurements of the refrigerant flow, the air flow and the heat loss allow comparing the energy conservation law by three different methodologies. The results are present to a variable volumetric capacity compressor - swash plate - and HFC134a refrigerant.

1. EXPERIMENTAL EQUIPMENT AND SYSTEM MODIFICATIONS

Petroski *et al.* (2006) presented a system to simulate the real conditions operation of an automotive air conditioning system, Fig. 1. Parameters are controlled to submit the system to diverse operation situations. First, an AC frequency inverter is used to control the rotation of an electric engine and, consequently, the compressor rotation. The electric engine is connected to a load cell to determine the mechanic power required by the refrigerating compressor. The evaporator is placed inside a calorimeter developed by Piske *et al*. (2004 and 2005). The calorimeter is an adapted horizontal household freezer which is basically a thermally insulated box. The inside airflow is powered by a fan. In order to assure the thermal control, an electrical heater is driven by a closed-loop control. The thermostatic expansion valve adjusts the evaporation temperature assuring no liquid goes to the compressor inlet. The calorimeter system is placed within a climatic chamber, which assures constant temperatures at the external envelope surfaces. The refrigerating circuit is instrumented for measure temperatures with type T thermocouples and pressures with pressure transducers. In this work a Coriolis flowmeter was used to determine the refrigerant mass flow and the specific mass.

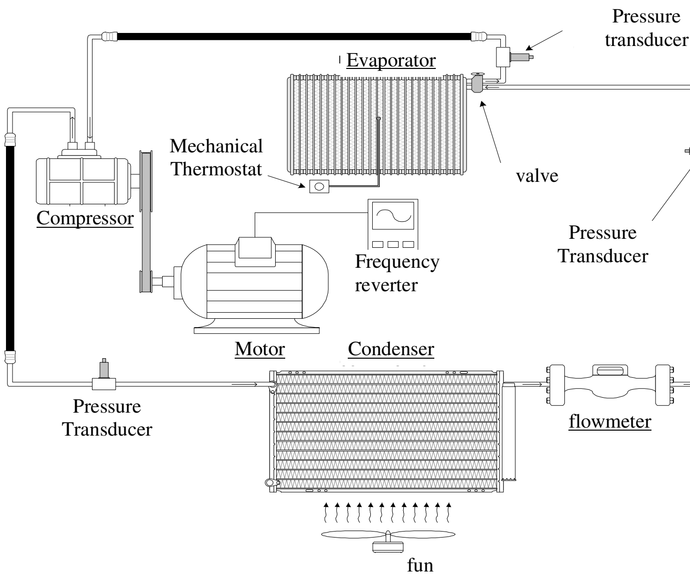

Figure 1. Sketch of refrigerating circuit (Petroski *et al.*, 2006)

When the electric motor is trigged off, the whole test bench begins to vibrate, the system of measurement of the torque is interdependent on the ground of the room, and consequently these vibrations also affect the measures taken by the load cell. The system was mechanical isolated from the ground to avoid this effect and Fig. 2 presents these results. The minimum vibration amplitude without mechanical isolation was measured to the frequency of 1508. The amplitude of this vibration, between 2500 and 2000 rpm decrease more than 80 %. Without the mechanical isolation system the vibration average amplitude was about 30 W on the compressor power measuring, using the isolation this amplitude decrease approximately to 3.5 W.

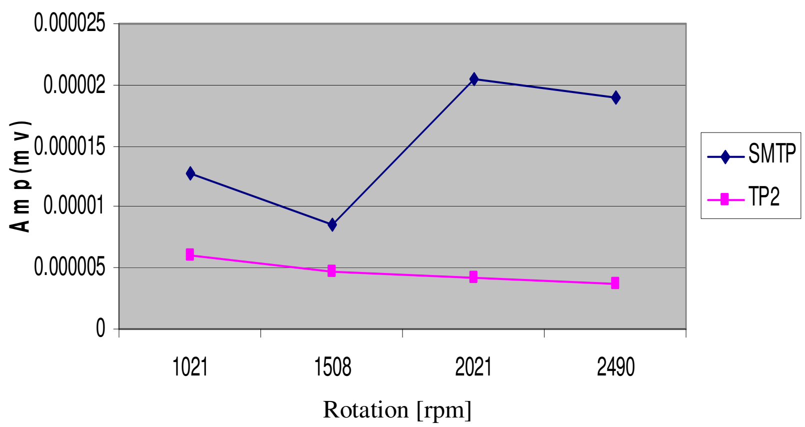

Figure 2. Vibratory amplitudes without the mechanical isolation system (SMTP) and using it (TP2)

2. FORMULATION

The heat transfer rate absorbed by the test evaporator can be measure by three different methods: *i*) Overall heat transfer rate colorimeter evaluation; *ii*) Enthalpic evaluation on the refrigeration flow; *iii*) Enthalpic evaluation on the air flow.

2.1 Method 1: overall heat transfer rate evaluation

This analysis is based on the calorimeter principle (Piske *et al.*, 2004 and 2005). The evaporator is placed on a horizontal household freezer which is basically a thermally insulated box, Fig. 3. The heat transfer by the walls, Q_{wall} , of the calorimeter was previously measured in a climatic chamber as a function of the internal and external temperature difference, $\overline{T}int$, $\overline{T}ext$, respectively , under similar convection heat transfer conditions and steady state. The overall heat transfer coefficient of the calorimeter boundaries is done by:

$$UA_{wall} = \frac{Q_{wall}}{\left(\overline{T}int - \overline{T}ext\right)} \tag{1}$$

The calorimeter is connected to an HFC134a refrigerating system which feeds continuously the evaporator under test. Then, when the hygrothermal domain of the calorimeter reaches steady-state conditions, it means the heat transfer rate at the evaporator surface is the sum of the heat dissipated by the heater, Q_{heater}, the heat dissipated by the fan, Q_{fan}, and the conduction heat transfer rate through the calorimeter envelope, Q_{wall}, Eqs. (2) and (3):

$$Q_{wall} = UA_{wall}(T_{wall,ext} - T_{wall,\text{int}}) \tag{2}$$

$$Q_{fan} = VI \tag{3}$$

where $T_{wall,ext}, T_{wall,\text{int}}$ are the average temperatures taking over the external and internal surface of the calorimeter walls, respectively. The electric potential is V and the electric current is I. The heat transfer rate absorbed by the evaporator is obtained by a lumped approach using the energy conservation equation applied to the calorimeter domain, Fig. 3, which gives:

$$Q_{evap} = Q_{heater} + Q_{fan} + Q_{wall} \tag{4}$$

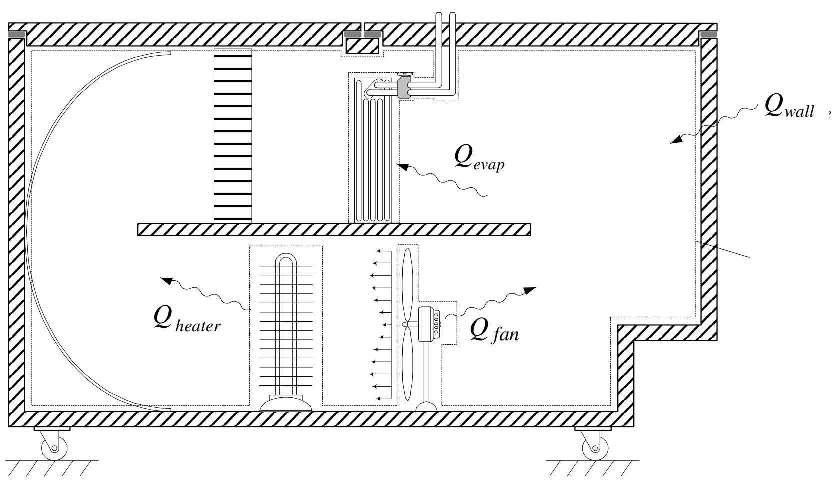

Figure 3. Diagram of the heat transfer rate exchanged in the calorimeter

2.2 Method 2: Enthalpy evaluation from the refrigeration flow

The heat transfer rate absorbed by the evaporator can be obtained by the energy conservation equation applied to the internal evaporator domain. In this case the refrigerant mass flow rate, $\dot{m}_{refr}$, must be measure and the inlet and outlet enthalpies, $h_{refr,outlet}$, $h_{refr,inlet}$, are determined by the temperature and pressure measurements of the refrigerant circuit. The heat transfer rate absorbed by the evaporator is calculated by:

$$Q_{evap} = \dot{m}_{refr}(h_{refr,outlet} - h_{refr,inlet}) \tag{5}$$

2.3 Method 3: Enthalpy evaluation from the air flow

Alternatively, the heat transfer rate absorbed by the evaporator can be obtained from the energy conservation equation applied to the external evaporator domain. In this case, the air mass flow rate, $\dot{m}_{air}$, must be measure and the air inlet and outlet enthalpies, $h_{air,outlet}$, $h_{air,inlet}$, are determined by the temperature and pressure measurements of the air flow. The heat transfer rate absorbed by the evaporator is calculated by:

$$Q_{evap} = \dot{m}_{air}\left(h_{air,inlet} - h_{air,outlet}\right) \tag{5}$$

2.4 Compressor power

The compressor power is determined measuring the momentum using a cell charge and a special structure supporting the compressor. The resulting equation is:

$$W_{comp} = FL\frac{n\pi}{30} \tag{6}$$

where F is the force measured by the cell charge, L is the distance between the rotation axe of the compressor and the application point of this force, n is the rotation. This method takes in account all the irreversibility in the process.

2.5 Coefficient of performance

The coefficient of performance (*COP*) of a vapor-compression refrigeration cycle represents the refrigeration effect per unit of compressor power required and is expressed by:

$$COP = Q_{evap} / W_{comp} \tag{7}$$

3. RESULTS

The results were obtained to a variable volumetric capacity compressor - swash plate - with HFC134a refrigerant. The gas amount was determined to define the optimal operation of the system. The experimental conditions are set to avoid any phenomenon of cycling; to assure the steady state conditions. The ambient temperature was set to 40°C and the compressor speed was fixed to 1000 rpm. The compressor capacity is controlled by a PWM signal to modulate the angular position of the interdependent plate of the piston. This plate movement allows changing the volumetric capacity of the system. When the signal amplitude is on the maximum value, the angle is on the maximum position. The volumetric capacity modify the compression ratio, in addition this rate change the according to the couple transmitted by the electric motor (number of revolutions). This system allows the regulation of the compressor consumption. The PMW signal is controlled by a PID system, resulting on a consumption optimization.

When the system is imputed by a thermal power and the volumetric capacity is decreased, the mass flow falls suddenly, following by a continuous increase until to the maximum volumetric capacity. The reduction of the volumetric capacity decreases the refrigerant flow, the pressure and temperature differences in the circuit, Fig. 4. Consequently reducing the enthalpy differences will reduce the refrigerating power. In order to limit the increase of outlet evaporator temperature, the regulation valve allows passing more fluid in the evaporator; it follows an increase in the inlet compressor pressure. The upstream temperatures and pressures of the compression influence directly the outlet conditions (polytrophic compression). The outlet condenser enthalpy and inlet compressor enthalpy keep fixed because the evaporator enthalpy difference does not change. This explains that the outlet compressor pressure and outlet evaporate pressure are fixed. The heat capacity of the evaporator is equal but it is carried out to higher pressures and temperatures conditions.

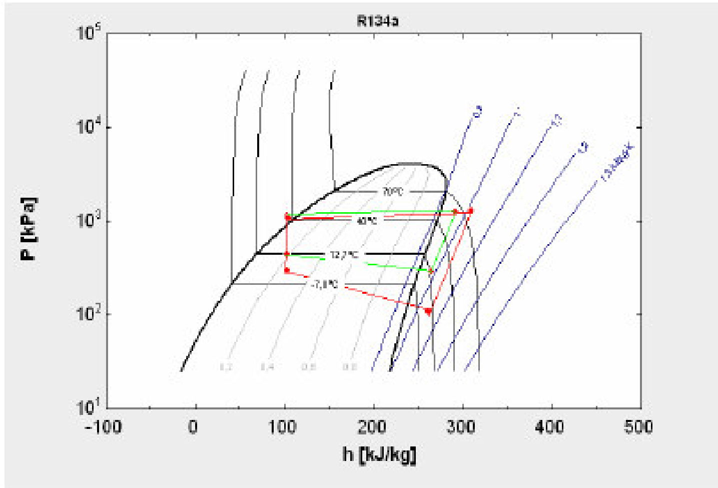

Figure 4. Pressure-Enthalpy diagram to HFC134a to the operation conditions. Red line is the refrigerating cycle with the maximum volumetric capacity compressor. Green line is the refrigerating cycle with the minimum volumetric capacity compressor

Figure 5 presents the difference between the inlet and the condenser outlet pressure, representing the refrigerant pressure losses function of the gas amount. The pressure drop is proportional to the square value of the refrigerant mass flow.

The outlet compressor pressure increases strongly with the gas amount. The pressure limit to this compressor is to values 30 bar. Over this the compressor can be damaged. Between a gas amount of 400 g and the 600 g the pressure values are practically constant.

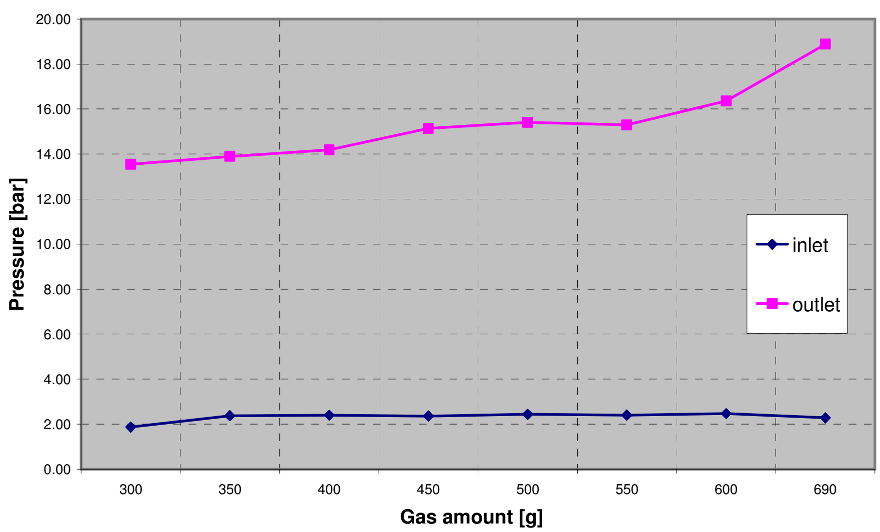

Figure 5. The inlet and the outlet pressure condenser function of the gas amount

Figure 6 shows that the flow rate decreases considerably with each gas increment. The expansion valve, whose flow rate is influenced by the outlet evaporator temperature, opens to the maximum, to enhance the refrigerant mass flow rate and thus the refrigerating power. With a constant refrigerant flow rate, increasing the gas amount, the refrigeration powerful will be also increase. The increase of refrigerant mass flow increases the energy necessary to the phase transition. This induces low temperatures in the outlet side of the evaporator; consequently the valve reduces this flow. The operation range is still highlighted between 450 g and 600 g.

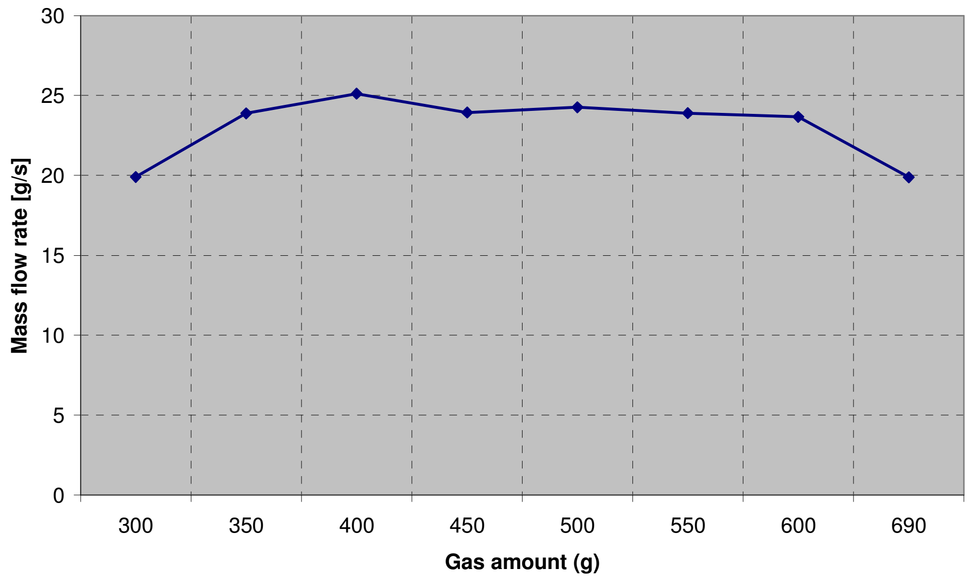

Figure 6.The mass flow rate function of the gas amount

Figure 7 shows the compressor power and the evaporator heat transfer rate function the gas amount. The compressor power and evaporator heat transfer rate increase until 400 g, and still constant until 600 g, decreasing to values over 600 g. The variations of the compression power are not in the same range as the evaporator heat transfer rate, which means that the efficiency will be affected by the evaporator heat transfer rate variations.

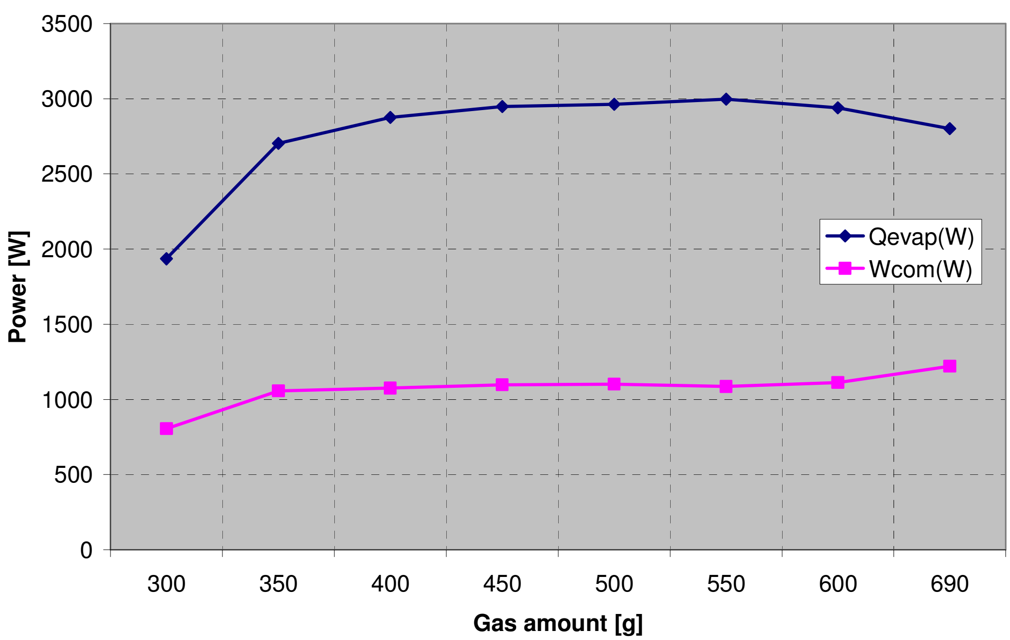

Figure 7. The compressor power and evaporator heat transfer rate function the gas amount

The COP is analyzed function of the compressor revolutions, Fig 8. The COP reaches the maximum for 500 rpm fixing the optimum refrigerant gas quantity. This can be explained by the variations of Q_{evap}, Fig. 7. The evaporator heat transfer rate represents the product between evaporator enthalpy variations by the refrigerant mass flow rate. These both variables are opposite drift, explaining the parabolic variation form of the evaporator heat transfer rate.

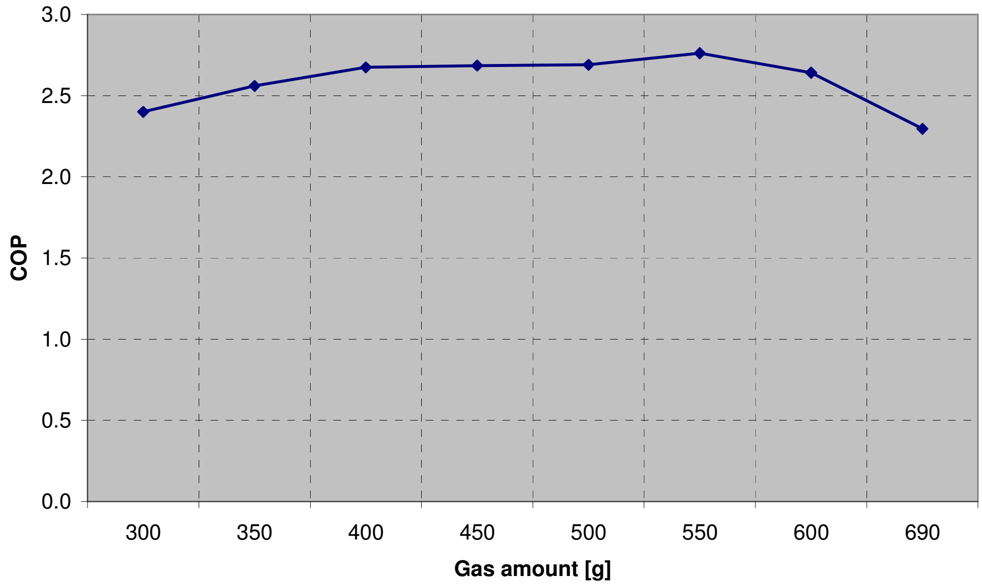

Figure 8. The compressor work and evaporator heat transfer rate function of the gas amount

One of the most important parameter on the system effectiveness is the number of revolutions of the compressor. In a automobile, its speed is very variable especially in downtown, acceleration, deceleration due to circulation (red lights, etc). The AC frequency inverter makes possible to fix the compressor rotation, a portable tachometer is used in order to measure the compressor rotation value. In order to reduce the cycling of the system it is fixed to 35°C the temperature of the climatic chamber and to 20°C the inlet evaporator temperature. Figure 9 presents the compressor power function the compressor rotation. The compressor power increase affecting the efficiency compressor. Otherwise, where the refrigerating power is too high and reduce too much the temperature, the thermostat valve control the compressor power in order to preserve the temperature conditions, inducing the cycling phenomenon. The period of the connection and disconnection cycles are decreasing at each rotation increment. Between 765 rpm and 2250 rpm, the period decrease by 50%, Fig. 10.

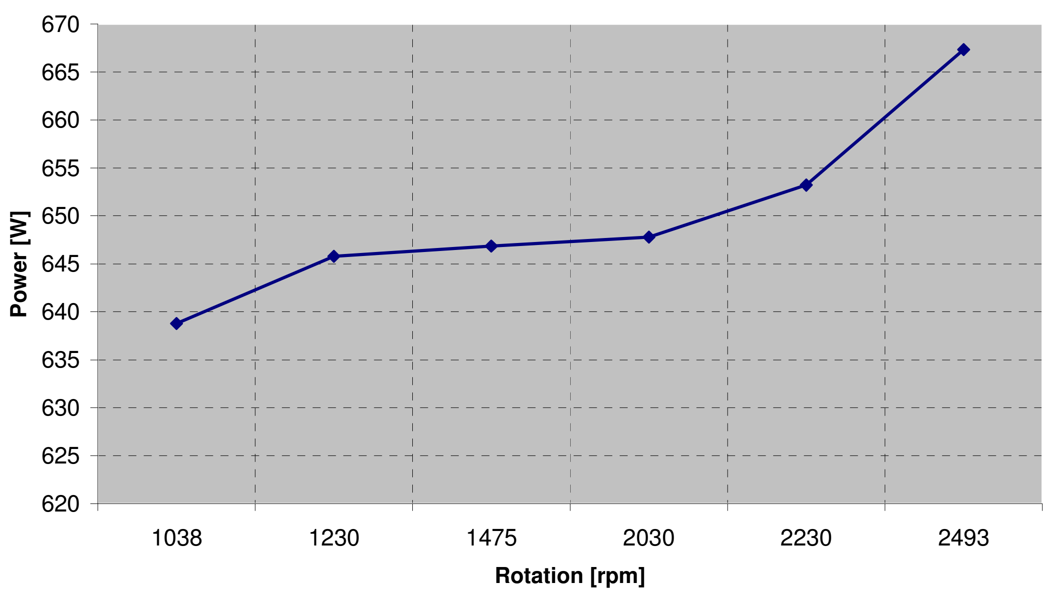

Figure 9. The compressor power function of the compressor rotation

The duty cycle between 765 rpm and 2250 rpm reduce from 72% to 45%, it is obvious that the refrigerant mass flow rate will increase when the system is trigged off, thus it can be observed a rise of 120 %. The compressor rotation increase and consequently, the refrigerant mass flow increase, aiding the latent heat exchange, increasing strongly refrigerant capacity and making it disproportional when it is compared to the quantity of absorbed heat transfer rate by the evaporator. The outlet evaporator temperature decreases too quickly what explains the increase of the duty cycles, Fig. 10.

Figure 11 shows that the average pressure values keep constant, like the refrigerant mass flow rate. The aspiration pressure is controlled by the expansion valve.

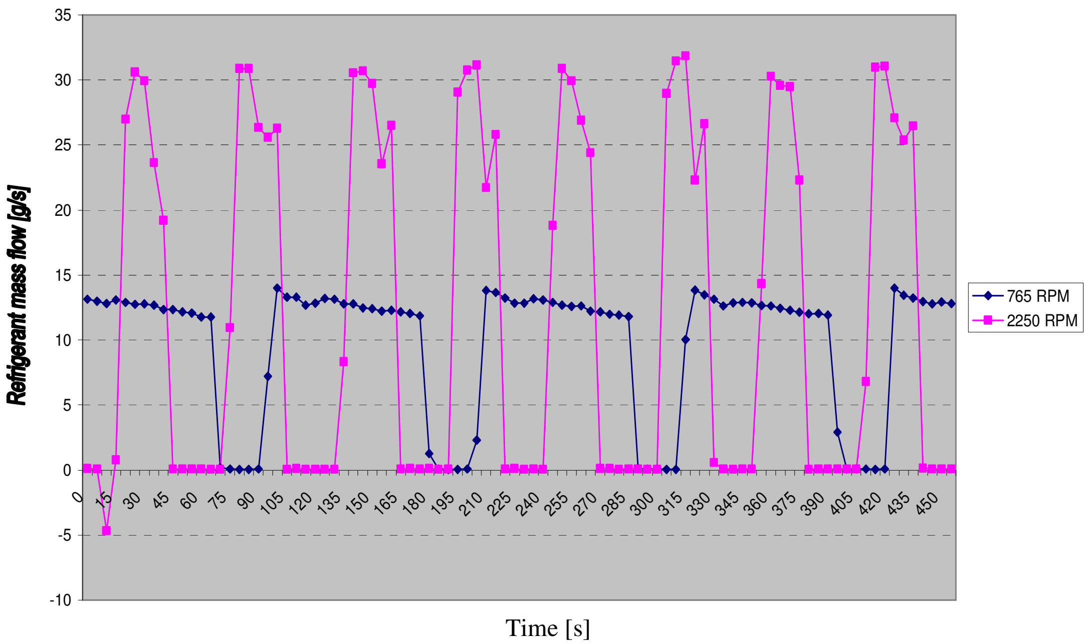

Figure 10.Cycling phenomenon

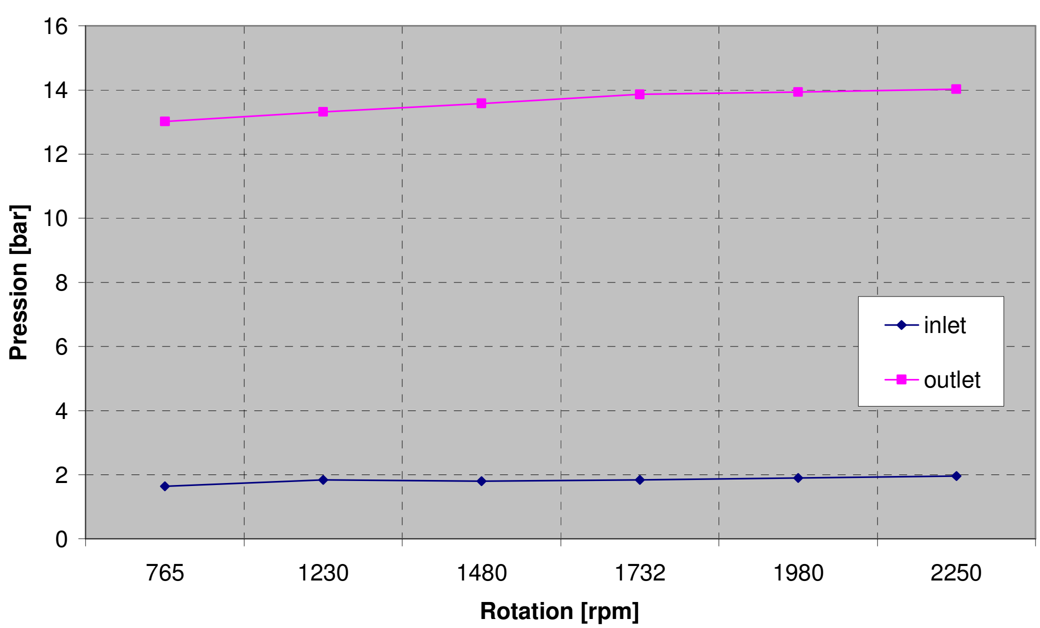

Figure 11. The average pressures function of the compressor rotation

3.1 Comparative study between overall heat transfer rate evaluation (Method 1) and enthalpic evaluation on the refrigeration flow (Method 2)

Figure 12 presents the comparison between measures using the refrigerant mass flowmeter - enthalpic calculations – and the overall heat transfer rate evaluation. To the enthalpic evaluation were used this follow assumptions*: i*) polytropic compression; *ii*) expansion of the valve is isenthalpic; *iii*) steady state conditions.

The evaporator heat transfer rate calculation presents an average difference of 2%, to this test conditions. The compression power calculated by enthalpic evaluation is over 25% lower compared to the overall heat transfer rate evaluation. It is necessary to taking in account the existence of the compressor efficiency, the measurements uncertainties, and the duty cycle.

The results presented previously allow making a coherent analysis with the measurements. In addition, applying the first law of thermodynamics in the system, it is obtained a lacking value close to 100W. This could represent the losses in the pipes of the circuit and the measurements uncertainties.

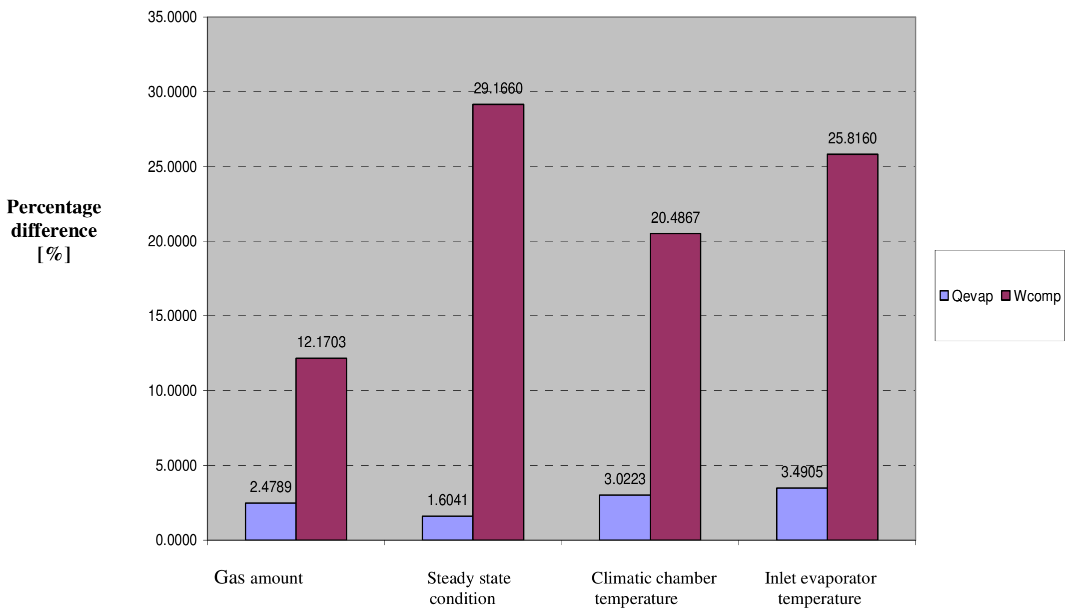

Figure 12. The comparison between measures using the refrigerant mass flowmeter - enthalpic calculations – and the overall heat transfer rate evaluation

4. CONCLUSION

This work goal was to evaluate the performances of a variable volumetric capacity compressor - swash plate. The refrigerant mass flow measurements make possible to study some physical phenomena, especially during the variation of the rotation compressor. For a constant evaporator heat transfer rate, when the rotation is increase, to keep the compressor power constant it is necessary that the torque decreases. However it proves that as the refrigerant flow is increasing too rapidly, the expansion valve limits this flow in order to keep the suction pressure constant. It is obvious that if the valve limits this flow, the pressure in upstream of the valve will be increasing in all the circuit (except in the low pressure zone from the evaporator inlet to the compressor inlet). The compression ratio thus will increase the torque. This phenomenon limits the torque decrease and consequently, it is not enough large to follow the rotation increase. This results on a compressor power increase and a performance reduction. This is one of greatest swash plate compressor advantage, the torque and the rotation being proportional and inversely proportional to the volumetric capacity, when one of both is increasing (for the same evaporator heat transfer rate), it is enough to reduce the volumetric capacity in order to maintain a good efficiency.

5. ACKNOWLEDGEMENTS

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