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Jackson Braz Marcinichen  
*Federal University of Santa Catarina*

Leonardo C. Schurt  
*Federal University of Santa Catarina*

Claudio Melo  
*Federal University of Santa Catarina*

Luis A. T. Vieira  
*Federal University of Santa Catarina*

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# PERFORMANCE EVALUATION OF A PLUG-IN REFRIGERATION SYSTEM RUNNING UNDER THE SIMULTANEOUS CONTROL OF COMPRESSOR SPEED AND EXPANSION VALVE OPENING

Jackson B. MARCINICHEN, Leonardo C. SCHURT, Cláudio MELO, Luis A. T. VIEIRA  
 POLO Research Laboratories for Emerging Technologies in Cooling and Thermophysics  
 Department of Mechanical Engineering, Federal University of Santa Catarina  
 P.O. Box 476, 88040-900, Florianópolis, SC, BRAZIL  
 Phone: +55 48 3234 5691, e-mail: [melo@polo.ufsc.br](mailto:melo@polo.ufsc.br)

## ABSTRACT

This paper assesses the simultaneous influence of compressor speed and expansion valve opening on the thermodynamic performance of a plug-in (cassette) refrigeration system, running under different control strategies. Two different system configurations were investigated: (i) variable speed compressor (VSC) and capillary tube (CT); and (ii) VSC and electric expansion valve (EEV). A dual SISO (*Single-Input, Single-Output*) control strategy was devised based on two PI controllers, one for the compressor speed and another for the expansion valve opening. The former senses the return air temperature, while the latter senses the superheating degree at the evaporator exit. It was found that the average energy consumption of the VSC/EEV when compared to the VSC/CT configuration decreased by as much as 10.1%, depending on the cassette operation regime and ambient temperature.

## 1. INTRODUCTION

Refrigeration systems for light commercial applications are usually assembled with capillary tubes or thermostatic expansion valves and single speed compressors. More recently plug-in or cassette refrigeration systems have been introduced into the market. Although innovative in several aspects these systems are very conservative from a thermodynamic point of view. A new scenario for such systems might be the use of variable speed compressors (VSC) and electric expansion valves (EEV), both working together and under a logic provided by a smart control strategy. In this respect both the EEV and VSC would adjust themselves to allow a proper matching between the cooling load and the cooling capacity and also to guarantee proper feeding of the evaporator.

Cho *et al.* (2006) evaluated the performance of an air conditioning system varying the refrigerant charge, the compressor speed, the expansion valve opening and the internal heat exchanger (IHX) length. Carbon dioxide (CO<sub>2</sub>) was used as the working fluid. It was shown that any increase in the compressor speed reduced the system coefficient of performance (*COP*), independently of refrigerant charge. The authors also showed that any increase in the compressor speed increased the optimum valve opening and that the system optimum discharge pressure could be attained by the simultaneous adjustment of valve opening and compressor speed. Finally, they showed that the IHX increased the cooling capacity by 6.2% and 11.9% and *COP* by 7.1% and 9.1% at compressors speeds of 40Hz and 60Hz, respectively. All tests were carried out under steady-state conditions without the aid of any type of controller.

Pöttker and Melo (2006) studied the effect of compressor speed, valve opening and refrigerant charge on the thermodynamic performance of a HFC-134a refrigeration system. They showed that for a fixed refrigerant charge and under steady-state conditions there was a proper combination of the pair compressor speed and valve opening that maximized the *COP* for a given cooling load.

Lin and Yeh (2007) implemented a MIMO (*Multi-Input, Multi-Output*) control strategy in an air conditioning system comprised of a VSC (0-100Hz) and a stepper motor EEV (480 pulses). It was shown that the time variation of the conditioned ambient temperature was well reproduced by the control action and that the system *COP* was improved. Finally, the authors claimed that with a proper control of the evaporator and condenser air flow rates higher *COP* values would be expected.

Marcinichen *et al.* (2007) evaluated the energy consumption of a plug-in refrigeration system assembled with a VSC, running firstly at a fixed speed of 1800rpm and secondly with a variable speed to control the evaporating pressure. In both cases the EEV controlled the superheating at the evaporator exit. In the first case a drop of 8.7% in the energy consumption was found for the VSC/EEV when compared to the VSC/CT configuration. In the second case the energy consumption of the VSC/EEV was always higher than that of the VSC/CT combination. This gave a clear indication that the evaporating pressure was not a suitable parameter for use in control strategies focused on system performance.

In this study the same plug-in unit tested by Marcinichen *et al.* (2007) was used to explore a new compressor speed control strategy, based on the cassette return air temperature, which is an indirect way of balancing cooling capacity and cooling load. The VSC and EEV controllers were developed using the PI (*Proportional-Integral*) structure. The tests were carried out using a purpose-built calorimeter for plug-in systems, under three ambient conditions (24.0°C/45%, 32.2°C/65% and 40.5°C/75%).

## 2. EXPERIMENTAL FACILITY

### 2.1 Calorimeter

The calorimeter is basically composed of the following parts: i) 150mm thick wall polyurethane chamber, ii) cassette-calorimeter coupling, and iii) supply air duct. An electric heater is installed in the supply air duct to control the internal air temperature. A variable speed fan and a damper are also installed in this duct to control the pressure drop in the air loop (see Figure 1).

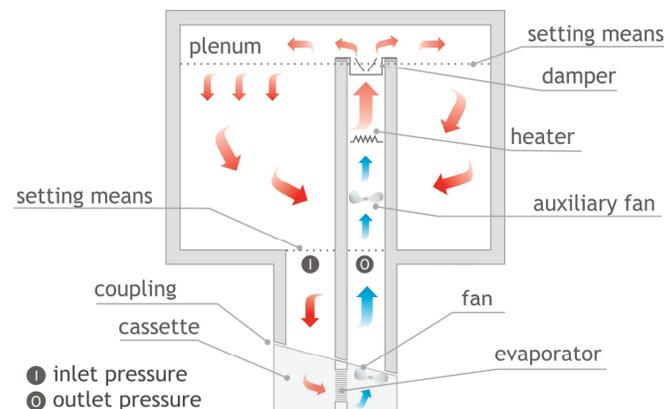


Figure 1: Schematic of the air loop

All variables present in a balance of energy involving the cassette itself and the calorimeter, are controlled and measured by the experimental set-up, both under steady-state and transient conditions. The cassette cooling capacity ( $\dot{Q}_e$ ) can be calculated in three distinct ways, as follows:

*i) Energy balance on the refrigerant side:* In this case the refrigerant mass flow rate ( $\dot{m}$ ) and the refrigerant enthalpy difference between the inlet of the evaporator and the exit of the cooled compartment ( $\Delta h$ ) should be evaluated (see Equation 1 and Figure 2),

$$\dot{Q}_e = \dot{m}\Delta h \quad (1)$$

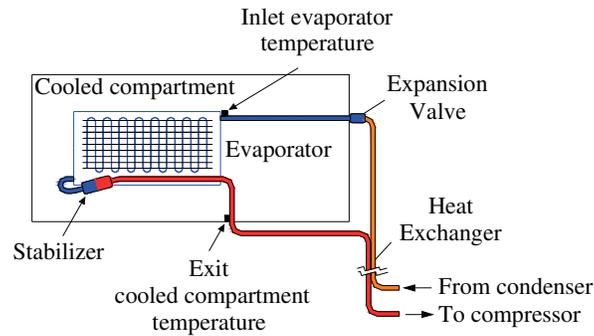


Figure 2: Schematic of the refrigerant loop

ii) *Energy balance on the air side:* In this case the air mass flow rate ( $\dot{m}_{air}$ ) and the air temperature ( $\Delta T_{air}$ ) and humidity ratio ( $\Delta W_{air}$ ) differences at the inlet and outlet of the evaporator should be measured (Equation 2),

$$\dot{Q}_e = \dot{m}_{air} c_{p,air} \Delta T_{air} + \dot{m}_{air} \Delta W_{air} h_w \quad (2)$$

where the symbols  $c_{p,air}$  and  $h_w$  correspond to the constant pressure specific heat and to the saturated water vapor enthalpy at 0°C, respectively.

iii) *Energy balance in the control volume involving the calorimeter and the cassette:* In this case all the energy forms flowing through the control surface should be taken into account (Equation 3),

$$\dot{W}_f + \dot{W}_h + \dot{Q}_w - \dot{Q}_e = \dot{m}_{air} c_{p,air} \frac{dT}{dt} \quad (3)$$

where  $\dot{W}_h$  is the heater power,  $\dot{W}_f$  is the fan power and  $\dot{Q}_w$  is the heat transfer rate through the chamber walls.

Although any of the introduced procedures may be used to evaluate the cassette cooling capacity, the option chosen in this study was the energy balance on the refrigerant side.

## 2.2 Cassette

The baseline cassette is comprised of a tube-fin evaporator, a helical condenser, a capillary tube-suction line heat exchanger and a variable speed compressor. The refrigerant fluid is R-134a.

The modified cassette incorporates the following changes with respect to the baseline version: i) the CT was replaced by a PWM (*Pulse Width Modulated*) EEV and ii) the capillary tube-suction line heat exchanger was replaced by a liquid line-suction line heat exchanger. The length of the internal heat exchanger was kept equal to that of the baseline system (83cm), but the liquid line diameter was decreased in order to keep the internal volume and consequently the refrigerant charge of both systems equal.

## 2.3 Instrumentation

A Coriolis mass flow meter was installed in the discharge line. The suction and discharge pressures were measured by absolute pressure transducers and the temperatures by T-type thermocouples. Voltage, current and power were measured by specific transducers. Specific electronic circuits, capable of converting digital into analogical signals, were developed and used to control both the VSC and the EEV. The measurement uncertainties are given in Table 1.

Table 1: Measurement uncertainties

Transducer	Uncertainty	Transducer	Uncertainty
Temperature	$\pm 0.2$ °C	Power	$\pm 1.5$ W
Pressure	$\pm 0.02$ bar	Energy consumption	$\pm 0.0015$ kWh/month
Voltage	$\pm 0.5$ V	Mass flow rate	$\pm 0.03$ kg/h
Current	$\pm 0.01$ A	Mass	$\pm 0.02$ g

## 2.4 Controllers

Two PI (*Proportional-Integral*) control structures were developed to establish the cassette return air temperature and the superheating degree. These structures were implemented in a dual SISO control strategy to allow variations in the compressor speed and valve opening. It should be noted that the superheating degree was calculated from the difference between the refrigerant temperature at the exit of the cooled compartment and the refrigerant temperature at the inlet of the evaporator (see Figure 2).

The controllers were designed by firstly running experiments to identify the system. In such experiments the behavior of the controlled variables was examined against stepwise changes in the compressor speed and valve opening (see examples in the Figures 3 and 4), under ambient conditions of 32.2°C/65%.

The models of the superheating degree as a function of valve opening and of the return air temperature as a function of compressor speed were developed based on the studies by Hong *et al.* (1992), Outtagarts *et al.* (1997) and Aprea and Renno (2001).

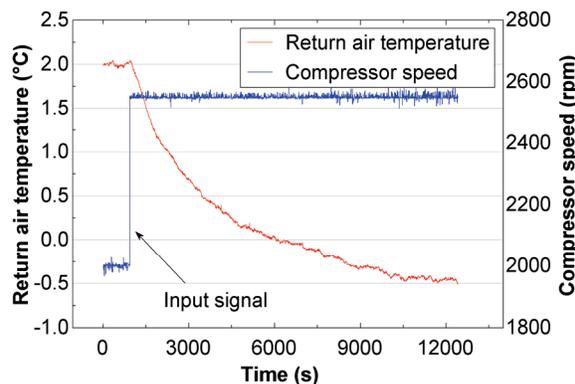


Figure 3: Return air temperature vs. compressor speed

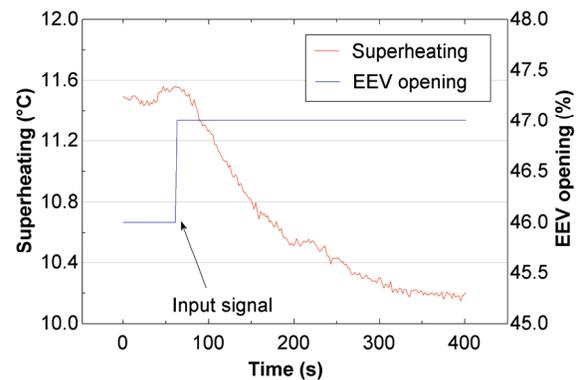


Figure 4: Superheating vs. valve opening

The first-order linear models developed are represented by equation (4) while the model parameters are given in Table 2.

$$G(s) = \frac{y(s)}{u(s)} = \frac{K_p}{\tau s + 1} \quad (4)$$

Where:

$G(s)$ : process transfer function

$y(s)$ : Laplace transform of the superheating degree ( $\Delta T$ ) or of the return air temperature ( $T_{r,air}$ ), °C

$u(s)$ : Laplace transform of the EEV opening ( $A_{EEV}$ ) or of the compressor speed ( $N$ ), % or rpm

$K_p$ : static gain, °C/% or °C/rpm

$\tau$ : time constant, s

Table 2: Identification test results

Parameter	$\Delta T$ ( $A_{EEV}$ )	$T_{r\_air}$ (N)
$K_p$	-2°C/%	-0.0044°C/rpm
$\tau$	150s	2800s

The PI controllers, represented by equation (5), were designed based on the above mathematical models. The PI controller parameters are shown in Table 3.

$$C(s) = \frac{u(s)}{e(s)} = K_c \left( 1 + \frac{1}{T_I s} \right) \quad (5)$$

Where:

$C(s)$ : controller transfer function

$u(s)$ : Laplace transform of the EEV opening or of the compressor speed, % or rpm

$e(s)$ : Laplace transform of the error in the superheating degree or in the return air temperature, °C

$K_c$ : controller proportional gain, %/°C or rpm/°C

$T_I$ : controller integral time, s

Table 3: PI controller parameters

PI Controller	$K_c$	$T_I$
$T_{r\_air}$	-680rpm/°C	750s
$\Delta T$	-1.5%/°C	53s

### 3. CASSETTE EVALUATION

#### 3.1 Charge determination

The refrigerant charge of the VSC/CT baseline system was firstly optimized according to two distinct testing procedures: i) the optimum charge was that which minimized the energy consumption under cyclic operation conditions and ii) the optimum charge was that which minimized the supply air temperature under continuous operation conditions, with a return air temperature of 2°C. The tests were carried out in an environmental chamber with the air temperature controlled at 32.2°C and the relative humidity at 65%. The compressor speed and the air loop pressure drop were maintained at 3600rpm and 25Pa, respectively.

Figures 5 and 6 show the results supplied by the first test procedure. Tests were carried out with a heater power of 0W. It can be seen that the optimum refrigerant charge was approximately 260g, a value that minimized the energy consumption and produced a superheating degree of 2°C.

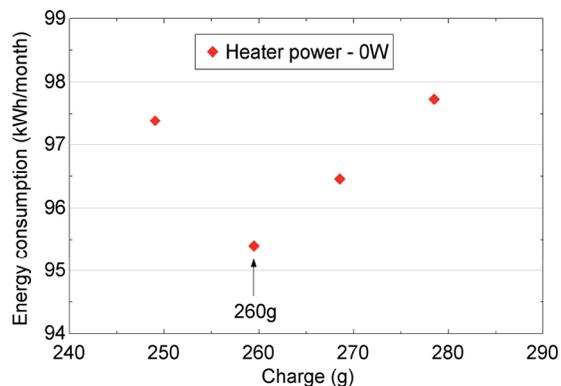


Figure 5: Energy consumption vs. refrigerant charge

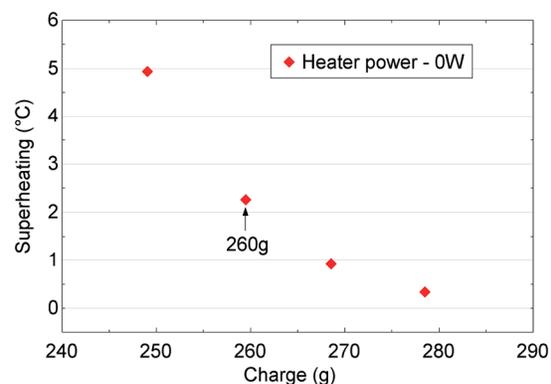


Figure 6: Superheating vs. refrigerant charge

The results obtained with the second test procedure are shown in Figure 7. In this case the optimum refrigerant charge corresponded to a value of 255g, a value that minimized the supply air temperature and produced a superheating degree of 1.7°C

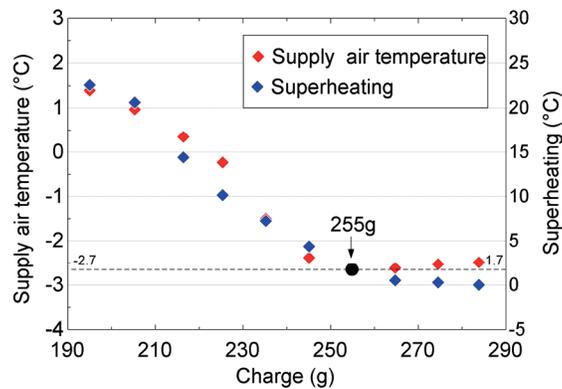


Figure 7: Superheating and supply air temperature vs. refrigerant charge

The optimum refrigerant charge was therefore 30g higher than that specified by the cassette manufacturer (230g). In all the following tests a refrigerant charge of 260g was employed.

### 3.2 Energy consumption

Energy consumption tests were carried out with the two cassette configurations, under three pre-established conditions: i) cooling load supplied by the electric heaters, ii) air loop pressure drop kept at 25Pa and iii) compressor on and off temperatures of 5.6°C and 1.7°C, respectively. Both the cassette and the calorimeter were mounted in an environmental test chamber, where tests were carried out under the following conditions: 24°C/45%, 32.2°C/65% and 40.5°C/75%. The tests were carried out using the compressor speed to control the return air temperature at 3°C and the valve opening to control the superheating degree at 6°C.

Table 4 shows the energy consumption and the runtime ratio (*RTR*) of the two cassette configurations running under three distinct ambient conditions. It is worth noted that the energy consumption refers only to the compressor and not to the other system components.

Table 4: Energy consumption and runtime ratio (*RTR*)

Condition	Heater power (W)	Compressor energy consumption (kWh/month) / <i>RTR</i>		Difference (%)
		VSC/CT	VSC/EEV	
24°C / 45%	40	64.27 / 59.65	67.59 / 57.76	5.17
	160	88.18 / 100	84.23 / 100	-4.48
32.2°C / 65%	30	78.65 / 83.28	83.12 / 77.35	5.68
	100	98.50 / 100	86.26 / 100	-12.43
40.5°C / 75%	20	100.04 / 100	87.31 / 100	-12.72

It can be observed that the energy consumption of the VSC/EEV was greater than that of the VSC/CT configuration only when the compressor switched on and off (*RTR* smaller than 100%). Under this condition, even at the lowest compressor speed, there is no balance between cooling load and cooling capacity, which decreases the return air temperature to values lower than the set point of 3°C. When a proper balance between cooling load and cooling capacity is attained, the compressor works continuously and the energy consumption of the VSC/EEV is always lower than that of the VSC/CT configuration, with the difference increasing with the ambient temperature.

Figures 8 to 11 show the time variation of the superheating degree, cooling capacity, compressor speed and power during a compressor cycle resulting from a heater power of 40W under ambient conditions of 24°C/45%.

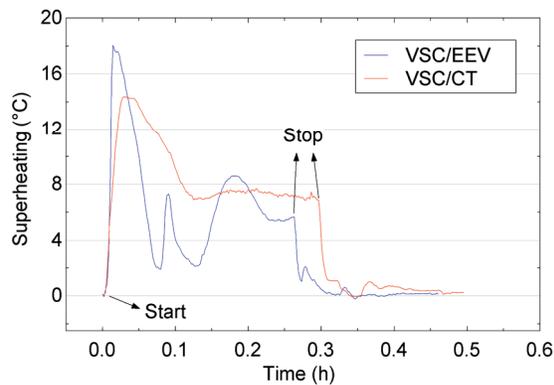


Figure 8: Superheating vs. time

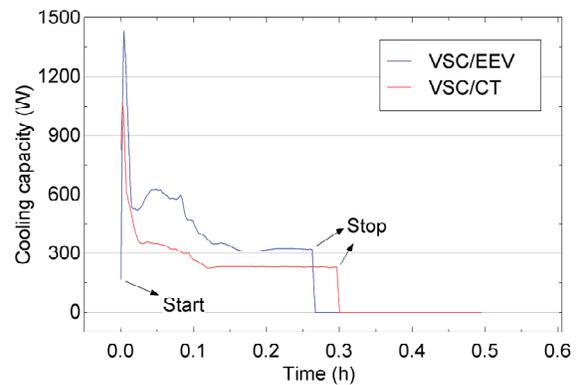


Figure 9: Cooling capacity vs. time

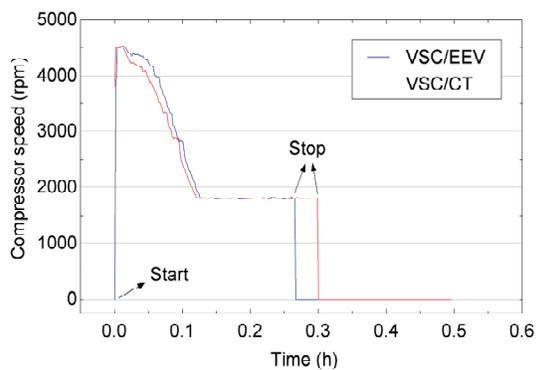


Figure 10: Compressor speed vs. time

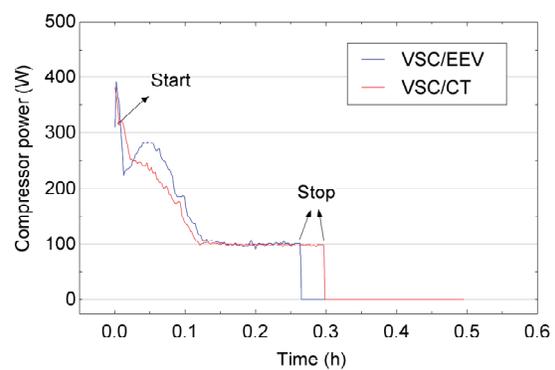


Figure 11: Compressor power vs. time

It can be seen that the VSC/EEV configuration operated with an average superheating of  $6.7^{\circ}\text{C}$ , a value lower than that obtained with the VSC/CT configuration ( $8.5^{\circ}\text{C}$ ). Furthermore the VSC/EEV produced a higher cooling capacity and operated with a smaller runtime ratio than that of the baseline system. However, the VSC/EEV compressor speed and power values were higher than those of the VSC/CT system. The net result was an increase in the energy consumption showing that the effect of a power increase surpassed the effect of a runtime ratio decrease.

Table 5 shows the compressor speed and power ( $\dot{W}_c$ ), the cooling capacity and the system coefficient of performance for the two cassette configurations running under ambient conditions of  $32.2^{\circ}\text{C}/65\%$ , and with a heater power of 100W. It is worth noting that the VSC/EEV configuration operated with a lower compressor speed and power, with a higher cooling capacity and consequently with a higher coefficient of performance than that of the baseline system.

Table 5: Cassette performance

Configuration	$N$ (rpm)	$\dot{W}_c$ (W)	$\dot{Q}_c$ (W)	$COP$
VSC/CT	2193.7	136.8	304.3	2.22
VSC/EEV	1918.5	119.9	329.1	2.75

#### 4. CONCLUDING REMARKS

The energy consumption of the VSC/EEV was higher than that of the VSC/CT configuration, when the compressor switched on and off. In this case the cooling capacity and the cooling load are not balanced against each other, which decreases the return air temperature to values lower than the set point of  $3^{\circ}\text{C}$ . When there is a proper balance between the cooling load and the cooling capacity, the compressor runs continuously and the energy consumption of the VSC/EEV is always lower than that of the VSC/CT configuration. A saving of up to 12.1% in energy

consumption was found for the VSC/EEV when compared to the VSC/CT configuration, depending on the ambient temperature.

Finally, it should be mentioned that the PI controllers were designed for ambient conditions of 32.2°C/65%, but responded quite well to the extrapolated conditions of 24°C/45% and 40.5°C/75%.

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