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NUMERICAL SIMULATION OF A SUPERHEATED CYCLE FOR TESTING OF REFRIGERATION COMPRESSOR.

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Summary. *This paper considers the theoretical simulation of a new cycle that only works in the superheated vapor phase. Such a cycle is being considered as a new improved method for compressor performance rating, allowing to get a faster transient change between steady state regimes. In order to verify the influence of the control parameters on the adjust of operating conditions, semi-empiric mathematical models were developed for each component in the cycle. Preliminary results obtained were validated by comparison against experimental data showing the feasibility of the numerical simulation.*

Password: Simulation, Compressors, Refrigeration. Testing.

1. INTRODUCTION

Air-conditioning and refrigeration systems have a significant participation in the overall energy consumption of a country. In Brazil, electric consumption due to air-conditioning and refrigeration applications is continuously growing. According to a governmental program to energy efficiency improvement, the contribution of such applications in the total consumption in the residential, commercial and industrial sectors is about 32%, 20% e 6% respectively, which represents about 15 % of the total electric energy production.

In air-conditioning and refrigeration applications, the vapour compression cycle is still one of the most important ones, where the compressor is the main component with respect to the energy efficiency usage.

Standards for compressor performance testing, is the scope of codes such as ISO 917 (ISO, 1989) e ASHRAE 23-1978R (ASHRAE, 1978) where several methods for performance rating are proposed. In these standards different limitations are imposed with respect to the droplet effect and steady state condition. In general, most of the proposed tests require a relatively complex apparatus, a long period of time for steady state regime, and present some difficulty to adjust test conditions.

In order to avoid such problems, Dirlea *et al.* (1996), have considered a testing cycle with all the refrigerant processes taking place in the superheated region only. A preliminary analysis of the feasibility of such a cycle was conducted in a small scale experimental apparatus, allowing to observe certain advantages to the superheated cycle, as for instance:

low energy consumption, easy adjust of testing conditions, simple layout of the test apparatus, small refrigerant charge, and quick transient change between steady state test regimes.

The application of the superheated cycle for compressor testing was later studied by Duarte, *et al.* (1998) using an experimental testing apparatus as shown schematically in (Fig. 1). Four main components are included: the compressor under test, a desuperheater heat exchanger, an expansion valve and a refrigerant reservoir.

The refrigerant transformation are as follows: After compression (1→2), the heat absorbed by the refrigerant (corresponding to the compression power) is delivered to the cooling water at the desuperheater heat exchanger (2→3). An isenthalpic expansion is then carried out trough an expansion device (3→1), closing the cycle.

The different operating conditions are attained by adjusting the following three parameters: desuperheater water flow rate ($\dot{m}_w$), expansion valve pressure drop (ΔP,by displacing the valve needle *"x"* in order to modify the flow passage area) and refrigerant system charge (M_t, by using the refrigerant tank).

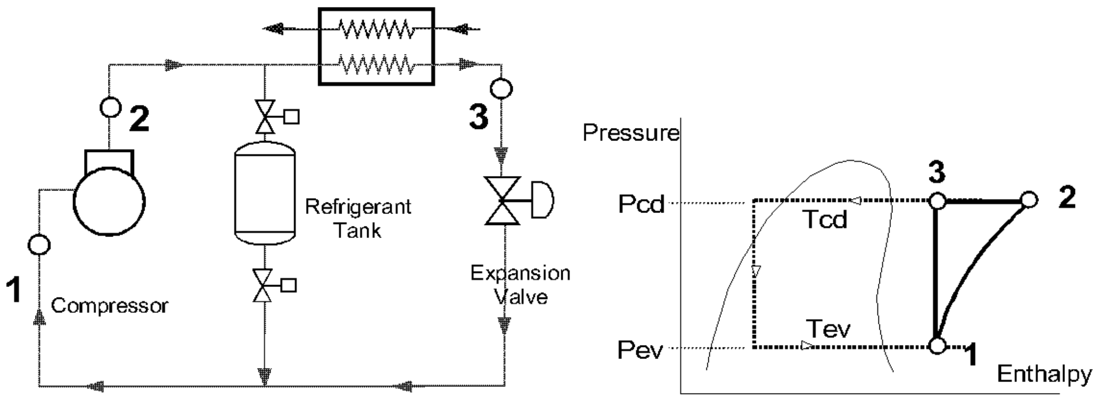

Figure 1 – Schematic apparatus and cycle.

2. MODELLING AND SIMULATING OF THE CYCLE

The simulation of the superheated test cycle is carried out on the basis of component models developed to the compressor, the desuperheater, the expansion valve and a refrigerant charge inventory.

The main goal of the simulation is to obtain, for different operational conditions, the adjusts required to the control of the cycle, i.e., being imposed refrigerant charge, water flow rate, and valve opening, which values must be adopted to the supply/exhaust pressures and the compressor inlet temperature (Fig. 2).

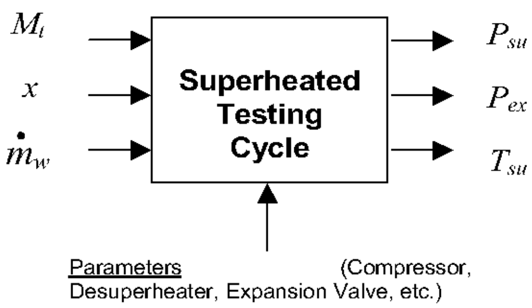

Figure 2 – Simulation of the superheated cycle.

2.1 Compressor

The compressor is modeled in a semi-empirical way (Bourdouxhe *et al.*, 1994; Oliveira *et al.*, 1998), and consists basically in the representation of the refrigerant flow rate, as:

$$\dot{V} = \dot{V}_S\left[1 + C_f - C_f\left(\frac{P_{ev}}{P_{cd}}\right)^{\frac{1}{\gamma}}\right] \tag{1}$$

and the power consumption by means of:

$$\dot{W}_{cp} = \dot{W}_{lo} + (1+\alpha)\dot{W}_{is} \tag{2}$$

Additionaly, the compressor model involves, the following equations:

$$\dot{W}_{is} = \dot{m}_{ref} \cdot P_{ev} \cdot v_1' \cdot \left((P_{cd}/P_{ev})^{\gamma/\gamma-1} - 1\right) \cdot \gamma/\gamma - 1 \tag{3}$$

$$\dot{m}_{ref} = \frac{\varepsilon_v \cdot \dot{V}_S}{v_1'} \tag{4}$$

$$\dot{W}_{cp} = (h_2 - h_1) \cdot \dot{m}_{ref} \tag{5}$$

$$\varepsilon_v = 1 + Cf - Cf \cdot (P_{cd}/P_{ev})^{\frac{1}{\gamma}} \tag{6}$$

$$T_1' = T_1 + \left(\frac{\dot{W}_{lo} + \alpha\,\dot{\mathrm{W}}_{\mathrm{IS}}}{\dot{m}_{ref}\,\mathrm{Cp}_{vap}}\right) \tag{7}$$

where,

ε_v	volumetric efficiency	[-]
$\dot{m}_{ref}$	refrigerant flow rate	[kg/s]
$\dot{V}$	volumetric refrigerant flow rate	[m^3/s]
C_f	clearance factor	[-]
$\dot{V}_S$	swept volume	[m^3/s]
P_{ev}	evaporating absolute pressure	[kPa]
P_{cd}	condensing absolute pressure	[kPa]
$\dot{W}_{cp}$	compressor power consumption	[W]
$\dot{W}_{is}$	isentropic compression power	[W]
$\dot{W}_{lo}$	constant electromagnetic losses	[W]
α	losses factor	[-]
T_1	temperature at the evaporator exhaust	[K]
T_1'	temperature after heating-up	[K]
v_1'	specific volume after heating-up	[m^3/kg]
Cp_{vap}	Mean specific heat at constant pressure in superheated vapour state	[J/kg K]

In such a compressor model the surroundings heat losses are neglected, and the compression process is assumed as isentropic. However, such simplifications does not imply in significant loss in model accuracy, as shown by Oliveira, *et al.* (1998).

The four parameters described above (C_f, $\dot{V}_s$, $\dot{W}_{lo}$ and α) are characteristic of the compressor and must be identified prior to the simulation. This was made by using both catalogue and experimental data, to which a minimisation procedure based in the least square technique is applied.

2.2 Refrigerant Charge Inventory

One important variable in the simulation of the superheated cycle is the charge refrigerant in the system (M_t). In order to simplify the modelling of the cycle as a whole, it is considered that the total mass of refrigerant in the circuit is the sum of the masses at low and high-pressure sides, as given from the following equation:

$$M_{\mathrm{t}} = \rho_{lp} V_{\mathrm{lp}} + \rho_{\mathrm{hp}} V_{\mathrm{hp}} \tag{8}$$

where,

ρ_{lp} , ρ_{hp} Refrigerant average density at low and high-pressure circuits, respect. [kg/m^3]

V_{lp}, V_{hp} Low and high pressure circuits volume, respect. [m^3]

The values of V_{lp} and V_{hp}, where previously determined from known geometric characteristics of the refrigerant circuit.

2.3 Expansion Valve

The refrigerant mass flow rate trough the expansion device is modelled by means of the following equations (Tamainot-Telto *et al.,* 1994 and Manzi *et al.,* 1998):

$$k_v = C\left(\frac{\dot{m}_{ref}}{\rho_{su}}\right)^n \tag{9}$$

$$\Omega = b_0 x + b_1 x^2 \tag{10}$$

$$\dot{m}_{ref} = k_v \Omega \sqrt{2\,\rho\,\Delta P} \tag{11}$$

$$\Delta P = P_{cd} - P_{ev} \tag{12}$$

where, k_v and Ω, are the flow coefficient and the flow passage area respectively, and, C, n, b_0 e b_1, are constant parameters identified for the valve, and x is the needle displacement.

2.4 Desuperheater

The desuperheater is modeled as a classical heat exchanger. To the known dimensions, geometry and fluids arrangement, the overall heat exchanger coefficient (U), is obtained from:

$$U = \left(1/h_{ref} + R_{wall} + 1/h_w\right)^{-1} \tag{13}$$

where,

h_{ref} internal convective heat transfer coefficient [W/m^{2}K]

R_{wall} conductive wall resistance [m^{2}K /W]

h_{ref} external convective heat transfer coefficient [W/m^{2}K]
with the heat transfer coefficients both at the water and refrigerant side computed from the classical Dittus-Boelter equation (Ashrae, 1993). Moreover, the total heat power exchanged is represented by:

$$\dot{Q}_{dsh} = \dot{m}_w c_{p_w} \Delta T_w \tag{14}$$

$$\dot{Q}_{dsh} = \dot{m}_{ref} \Delta h_{ref} \tag{15}$$

$$\dot{Q}_{dsh} = UA \cdot \Delta T_{LMTD} \tag{16}$$

$$\Delta T_{LMTD} = \frac{\left(T_3 - T_{w,su}\right) \cdot \left(T_2 - T_{w,ex}\right)}{\ln\left(\dfrac{T_3 - T_{w,su}}{T_2 - T_{,ex}}\right)} \tag{17}$$

where,

$\dot{m}_w$	water mass flow rate	[kg/s]
ΔT_w	water temperature increase trough the desuperheater	[K]
UA	overall heat transfer coefficient	[W/K]
ΔT_{LMTD}	log-mean temperature difference	[K]

2.5 Fluid Properties

In order to get more realistic results from the simulation, fluid properties were modelled considering real fluid behaviour, by means of a specific subprogram written as an adaptation from an available TRNSYS (TRNSYS, 1994) computational code. Then, from a given combinations of the thermodynamic properties input (temperature, pressure, quality, etc.) the thermodynamic state was easily obtained.

2.6 Simulation Procedure

The simulation of the superheated vapour cycle is based on the use of each component model separately. Knowing the technical-constructive characteristic and identified parameters of each component, the simulation of the cycle is made as shown in the information flow diagram bellow (Fig. 3).

Initially a operating condition is imposed by specifying the numerical values of: refrigerant charge (M_t); water flow rate trough the desuperheater ($\dot{m}_w$); opening position of the expansion valve (x) ; and water inlet temperature in the desuperheater ($T_{w,su}$ considered constant). Additionally, the previously identified parameters of compressor, expansion valve; desuperheater and the circuit volumes are considered. Then, an initial guess of the evaporation and pressures, as well as, compressor supply temperature is made.

First, the compressor model is called with the initial values of T_1, P_{ev} and P_{cd} and, after convergence to the heating up temperature (T_1'), we get a first estimate for the compressor power consumption ($\dot{W}_{cp}$) and for the refrigerant flow rate. Then, from Eq. 7, it is possible to get a first convergence value for the evaporating pressure (P_{ev}).

With the parameters of the expansion valve and the equations from 9 to 12 the convergence is determined around P_{cd}. Finally, with the model of the heat exchange, Eq. 13 and Eq. 17, the convergence for T_1 is reached.

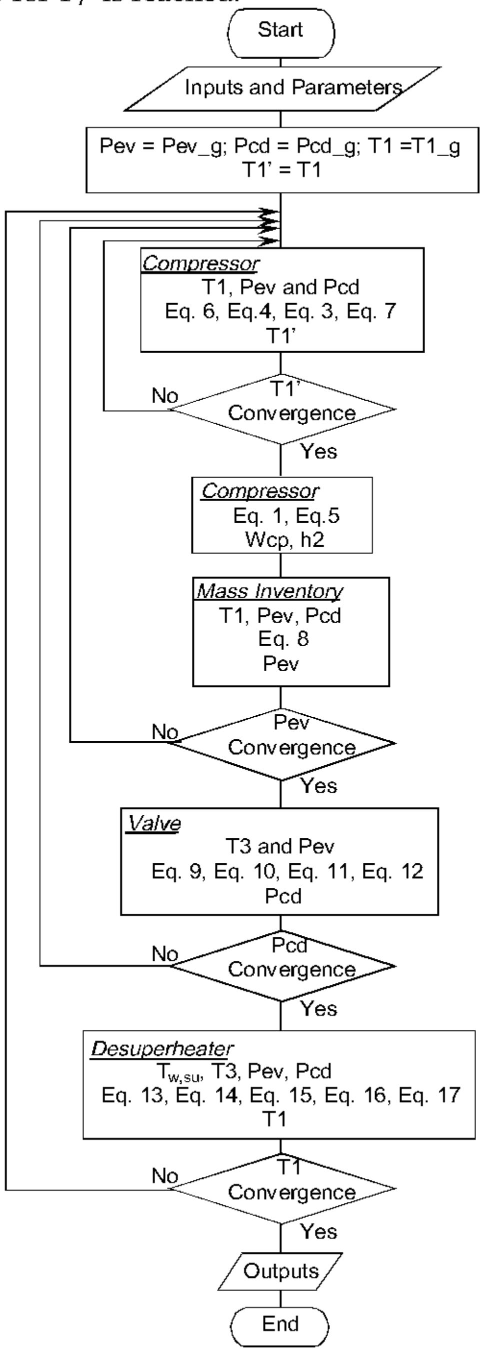

Figure 3 – Schematic view of simulation

3. SIMULATION RESULTS

Figure 4 bellow shows the influence of the refrigerant load on the operation conditions of the compressor, being maintained constant the other variables (needle position in expansion valve and water flow rate trough the heat exchanger). It can be seem that as the refrigerant load increases, there is a displacement of the cycle upward and to the right, i.e., both superheating and desuperheating increases. Besides, we can remark that an increasing of the system pressure will occur.

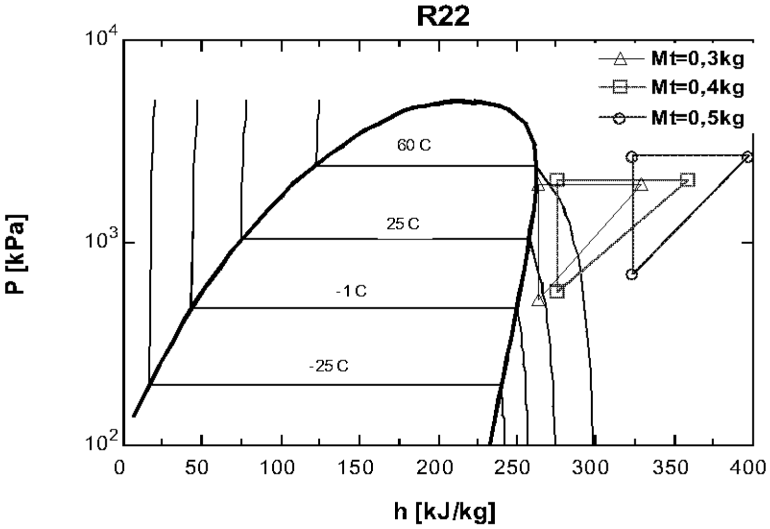

Figure 4 - Influence of the refrigerant load on the operation conditions of the compressor.

Figure 5 bellow shows the influence of the expansion valve opening, being maintained constant the other variables (water flow rate through heat exchange and the refrigerant load - 0,3 kg in this case). As the expansion valve is closed, we observe an increase in the compression rate, because the discharge pressure increases while the suction pressure decreases. The enthalpy in the compressor supply remains almost constant with the opening of the expansion valve, however, the enthalpy at the compressor exhaust decreases significantly. Great variations in the superheating degree does not occurs with expansion device opening.

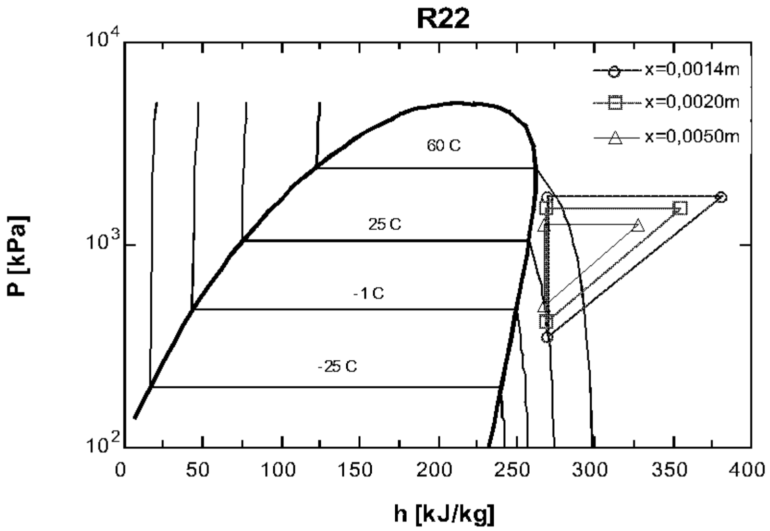

Figure 5 - Opening position of the expansion valve influence on the operation conditions of the compressor.

The influence of the water flow rate is shown in Figure 6. The increase of the water flow rate in the heat exchange causes a displacement of the cycle to the left, decreasing both the superheating and the desuperheating. Water flow rate increase will also cause a decrease in system pressure.

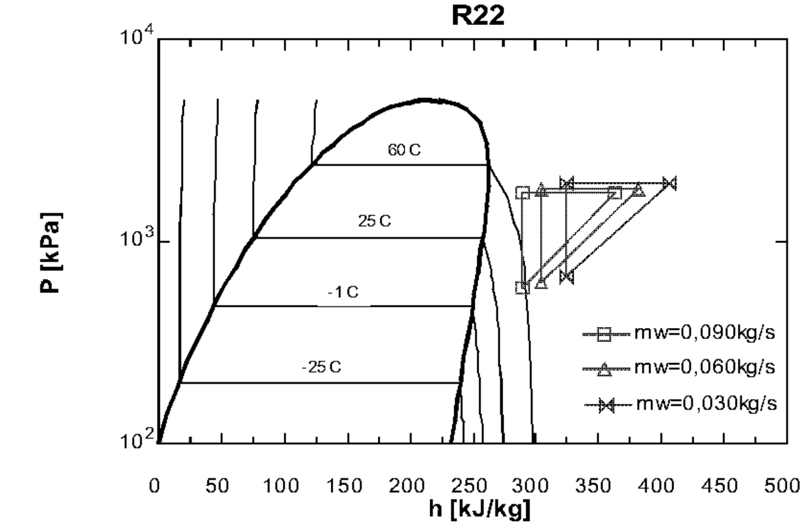

Figure 6 – Water flow rate influence on the change operation conditions of the compressor.

4. VALIDATION OF THE SIMULATION RESULTS

An existing general-purpose test bench for experimental study of refrigeration systems, was adapted for operation according the proposed testing cycle. The testing apparatus was described by Duarte et all. (1998).

Preliminary tests were carried out for a hermetic compressor. The main goal of these tests was to evaluate the influence of the control parameters on the adjust of operating conditions.

Figure 7 corresponds to a comparison between experimental (dots) and theoretical (lines) results, for a constant water flow rate at the desuperheater, showing the influence of both the refrigerant system charge and the expansion valve opening.

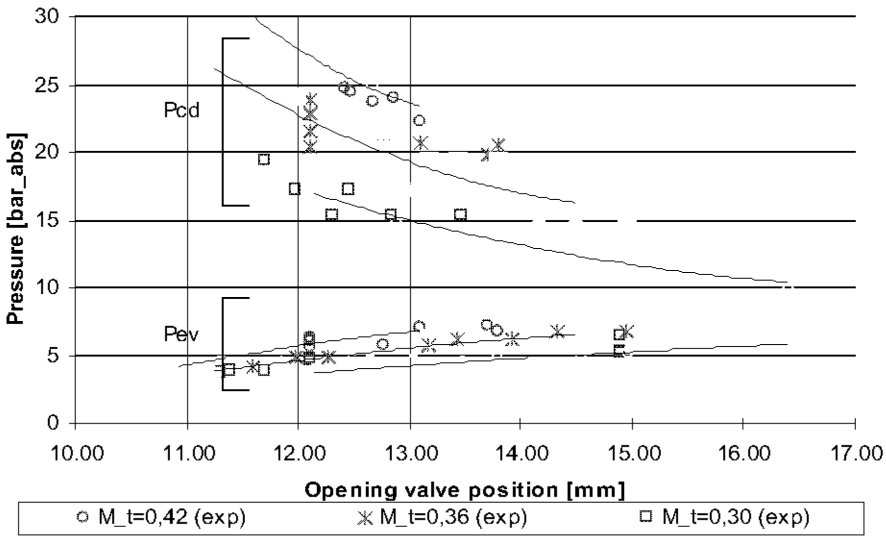

Figure 7 – R22 charge and expansion valve influence on the condensing and evaporating conditions (water flow rate = 2.0 kg/s).

For a given refrigerant charge in the system, different condensing and evaporating pressures can be imposed to the compressor by means of the expansion valve opening change. However, the change in the valve opening alone does not allow to adjust all the different combinations of condensing and evaporating required to the test of the compressors, being in this case necessary to change the refrigerant charge in the system.

With respect to the desuperheater water flow rate, this will have an influence on the compressor supply temperature. Then, to a given refrigerant charge and expansion device opening it is possible to adjust the water flow rate trough desuperheater in order to maintain a constant temperature at compressor supply.

5. CONCLUSIONS

The simplified modelling of a superheated cycle is developed in this paper, in order to predict system behaviour before experimental tests. Preliminary simulation allow verifying the influence of the change in the three control parameters of the cycle on the operating conditions. Moreover, analysis on the simulation results has allow to consider different aspects such as: control of the condensation occurrence, cooling requirements at compressor supply and installation of a chamber calorimeter to improve the compressor heat balance. A priori comparison of simulation results against experimental data shows good agreement.

Next activities in this work will consider additional tests in order to attain a better comprehension of the control of the new cycle.

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