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SEMI-EMPIRICAL MODELLING AND SIMULATION OF A CYCLE FOR REFRIGERATION COMPRESSORS TESTING IN SUPERHEATED REGION ONLY.

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ABSTRACT

This paper considers the development of a new compressor testing method based on a thermodynamic cycle taking place in the superheated region only. More than an alternative to standard testing methods the proposed superheated cycle would present significant advantages as for example a faster transient change between steady state test conditions. In order to carry out an investigation of the adjusting of operational conditions, the new cycle was modelled and simulated. Preliminary results allow to verify the system behaviour according the change in three control parameters : refrigerant system charge, expansion valve opening, and desuperheater water flow rate. Next steps in this research will consider an experimental implementation of the new testing method.

NOMENCLATURE

C_f	compressor clearance factor	[-]
$\dot{V}_S$	compressor swept volume	[m³/h]
P_{ev} , P_{cd}	evaporating and condensing pressures respect.	[bar]
W_{is}	isentropic compression power	[W]
W_{lo}	constant part of the electromagnetic losses	[W]
α	losses factor	[-]
h_{ref} , h_{ref}	internal and external convective heat transfer coefficients	[W/m²K]
R_{wall}	conductive wall resistance	[m²K /W]
Nu	Nusselt number	[-]
Re	Reynolds number	[-]
Pr	Prandtl number	[-]
ΔT_w	water temperature rise	[K]
Δh	specific enthalpy change	[kJ/kg]
$\dot{m}_w$, $\dot{m}_{ref}$	water and refrigerant mass flow rate respect.	[kg/s]

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 (Procel, 1998), 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. In such cycle, the compressor is the most important component with respect to the energy efficiency use. Although different compressor designs (scroll, screw, etc.) has gathered important market segments, the reciprocating type still being largely used due to its large field of application.

Standardization of compressor performance testing, is the scope of codes such as ISO 917 (ISO, 1989) and 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 all (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.

In the present paper, the superheated compressor testing cycle is analysed in more details. Emphasis is given to the steady state modelling and simulation of the superheated cycle, in order to verify the change in operating conditions with the parameters of adjustment of the cycle.

STANDARD TESTS vs. SUPERHEATED CYCLE

The final goal of a compressor test is to verify how well the actual performance matches the design compromises at different operating conditions. For that, during the test, the compressor is submitted to a certain operating condition, to which its behaviour is evaluated (figure 1). Such test conditions are defined by means of the following variables (considering constants rotation speed, ambient temperature, etc.): pressure at compressor supply (P_{su}); pressure at compressor exhaust (P_{ex}), and temperature at compressor supply (T_{su}). To these test conditions the compressor will response with a refrigerant flow rate ($\dot{m}_{ref}$), and some power consumption ($\dot{W}_{el}$). From this, other important performance parameters can be obtained, such as :refrigerating capacity ($\dot{Q}_{ref}$),volumetric (ε_{vol}) and isentropic (ε_{is}) effectiveness and coefficient of performance (COP).

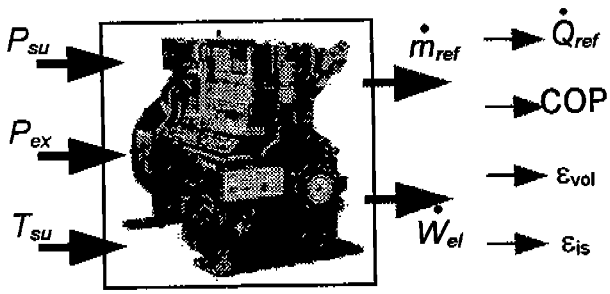


Figure 1 – Compressor performance testing concept

The ISO917 standard code (ISO, 1989) presents a set of nine different methods for the performance testing of reciprocating compressors. Basically, these methods can be classified according the technique used to obtain the refrigerant flow rate as direct (when a flow meter device is used) or indirect (when refrigerant flow rate is obtained from energy balances).

Along the ISO917 standard code, several restrictions are made with respect to the influence of oil-refrigerant mixing, droplet, and steady state conditions.

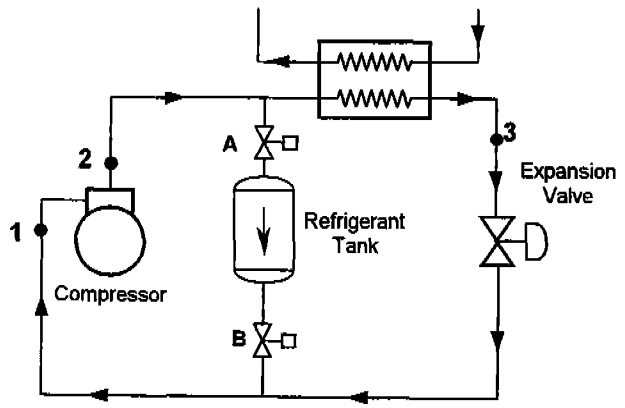
Concerning the oil-refrigerant mixing, the standard imposes that the oil pumping rate must be less than 1.5 % of the refrigerant mass flow rate and recommends the use of efficient oil separation devices in the circuit. Following these limits no correction is required by the standard to the effect of the lubricating oil concentration.

Minimum superheating and subcooling degrees are imposed in the standard in order to avoid the effect of incomplete condensation at condenser exhaust, and occurrence of liquid drops in the refrigerant leaving the evaporator. According the standard, the design of the testing apparatus shall be such that the outgoing vapour will not entrain droplets of refrigerant and is superheated by at least 8 K. The steady state conditions for the test are specified in the standard on the basis of a maximum acceptable change on the variables measured during the test inside a 1 h time interval

A schematic view of the apparatus for compressor testing in the superheated region only is shown in figure 2 bellow. Four main components are included : the compressor to be tested, 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, expansion valve pressure drop, and; refrigerant system charge. The water flow rate can be easily changed by modifying the opening of a globe valve at water inlet. The water inlet temperature is quite constant varying only according the ambient conditions. The pressure drop of the refrigerant trough the expansion valve is changed by displacing the needle of the valve in order to modify the flow passage area.

The refrigerant system charge is modified by means of the refrigerant reservoir and solenoid valves A and B (normally closed). If it is desired to decrease the system charge, solenoid valve A is open and part of the compressor exhaust gas will be bypassed to the reservoir. On the other hand, if it is needed to increase the system charge, valve B is open and some amount of gas will be added to the system.



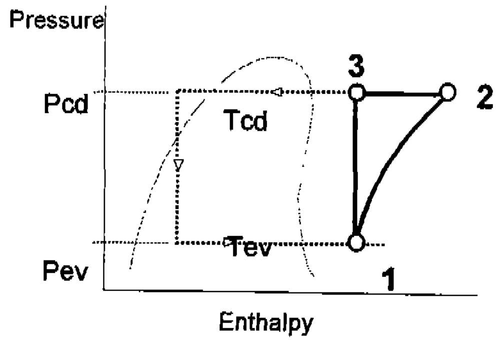


Figure 2 – Schematic view of apparatus and cycle for the new test.

With respect to the standard methods described before, the new proposed method may present the following advantages :

- The test apparatus is simpler, not requiring heat exchangers for condensation and evaporation;
- Steady state test conditions can be reached faster because of the smaller thermal capacitance of the system;
- The influence of droplet can be minimised or even avoided completely;
- Smaller refrigerant charge.

MODELLING AND SIMULATING THE NEW CYCLE

The simulation of the superheated test cycle is carried out on the basis of component models developed for 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 supply/exhaust pressures and the compressor inlet temperature, which values must be adopted to the refrigerant charge, water flow rate, and valve opening (figure 3).

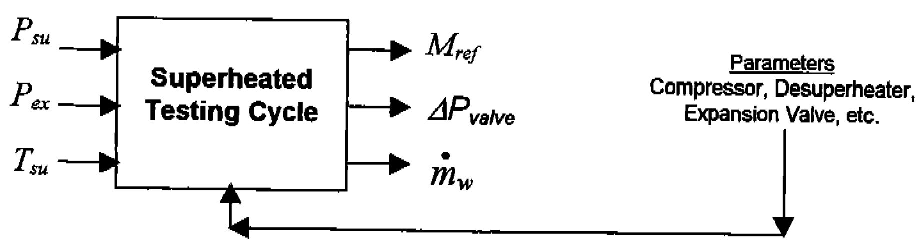


Figure 3 – Scheme for the simulation of the new cycle.

Compressor

The compressor is modelled in a semi-empirical way (Bourdouxhe et all., 1995), and consists basically in the representation of the refrigerant flow rate($\dot{V}$), as,

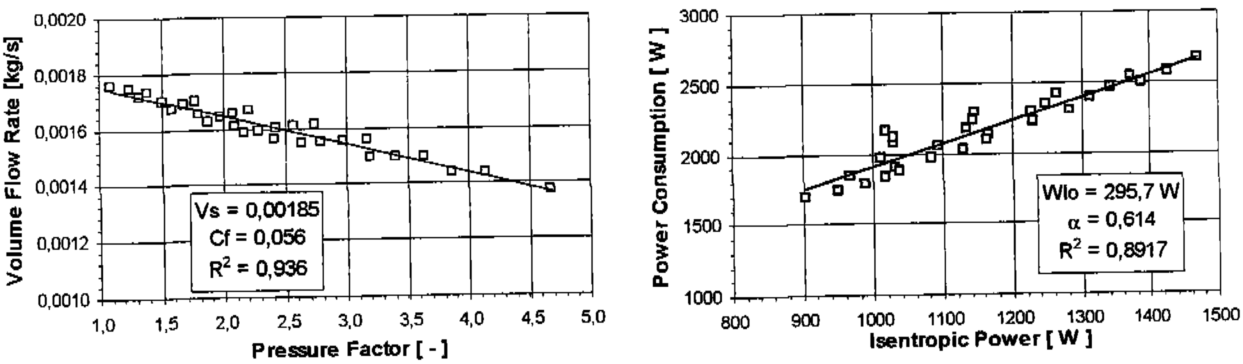
$$\dot{V} = \dot{V}_S \left[1 + C_f - C_f \left(P_{ex}/P_{su} \right)^{\frac{1}{\gamma}} \right] \tag{1}$$

and the power consumption ($\dot{W}_s$) by means of,

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

The main assumptions of the compressor model are related with the surroundings heat losses, which are neglected, and the compression process, which is assumed as being isentropic.

The four parameters described above are characteristic of the compressor and must be identified prior to the simulation. This was made using catalogue data, to which a minimisation procedure based in the least square technique is applied. Figure 4 shows the identification results.



(a) (b)

Figure 4 – Compressor identification results : (a) volume flow rate model; (b) power consumption model.

Desuperheater Heat Exchanger

The desuperheater is modelled as a classical heat exchanger. To the known dimensions, geometry and fluids arrangement, the overall heat exchanger coefficient (U), was obtained by means of :

$$U = \left(1/h_{ref} + R_{wall} + 1/h_w \right)^{-1} \tag{3}$$

where the heat transfer coefficients both at the water and at the refrigerant side were computed from the classical Dittus-Boelter equation (Ashrae, 1993):

$$Nu = 0{,}023\left[\mathrm{Re}^{0,8}\,\mathrm{Pr}^{0,4} \right] \tag{4}$$

The total heat exchanged in the desuperheater ($\dot{Q}$), neglecting heat losses to surroundings is given, by :

$$\dot{Q} = \dot{m}_w c_P \Delta T_w = \dot{m}_{ref} \Delta h_{ref} = UA\ LMTD \tag{5}$$

Equations (3) to (5) are used mainly to obtain the refrigerant temperature at the desuperheater outlet, which corresponds to the expansion device inlet.

Expansion Valve

The refrigerant mass flow rate trough an expansion device may be modelled by means of the following equation (Tamainot-Telto et all., 199?):

$$\dot{m}_{ref} = k_v \Omega \sqrt{2\,\rho \Delta p} \tag{7}$$

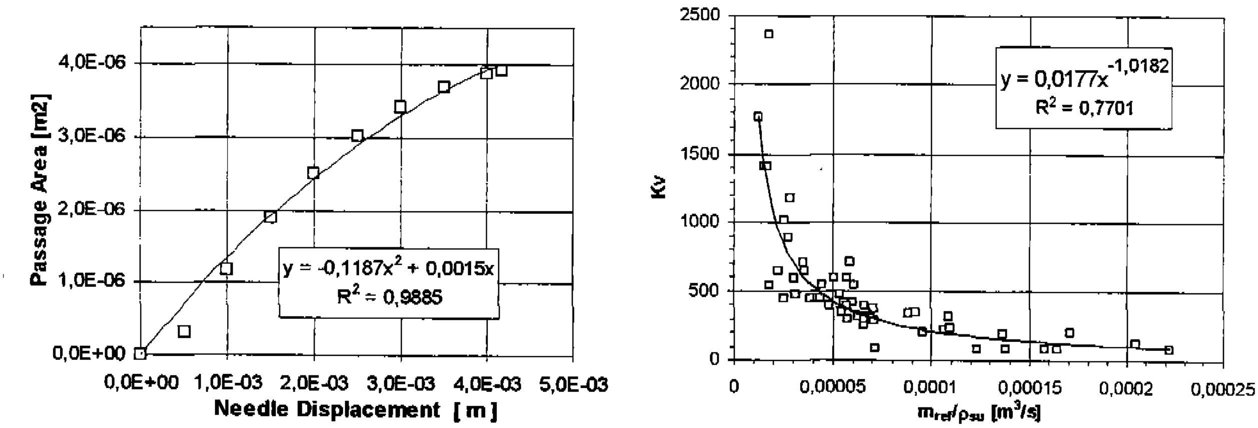
where, k_v and Ω, are the flow coefficient and the flow passage area respectively, which can be defined by,

$$k_v = C\left(\frac{\dot{m}_{ref}}{\rho_{su}}\right)^n \tag{8}$$

$$\Omega = b_0 x + b_1 x^2 \tag{9}$$

with C, n, b_0 e b_1, constant parameters of the valve, and x the needle displacement of the valve.

Knowing the main dimensions of the valve needle and passage hole, allow to obtain parameters b_0 and b_1 as being 0.0015 and –0.119 respectivelly. In order to determine C and n, the expansion valve was tested using both nitrogen and air. The pressure drop and flow rate through the valve, as well as inlet pressure and temperature, were measured for different valve openings and inlet pressures. Then, we obtain C = 0.018 and n = -1.018. Figure 5 show the identification results.



(a) (b)

Figure 5 – Expansion valve identification results. (a) Change in expansion valve passage area with needle displacement – equation (9) ; (b) Identification of flow parameters - equation (8).

Refrigerant Charge Inventory

One important variable in the simulation of the superheated cycle is the charge of the refrigerant in the system (M_t). It is considered that the total mass of refrigerant can be obtained by considering the sum of the masses at low and high-pressure sides :

$$M_t = \rho_{LP} V_{LP} + \rho_{HP} V_{HP} \tag{10}$$

where,

ρ_{LP}, ρ_{HP} refrigerant average density at low and high-pressure circuits, respectively [m³/kg]

V_{LP}, V_{HP} Low and high pressure circuits volume, respectively [m³]

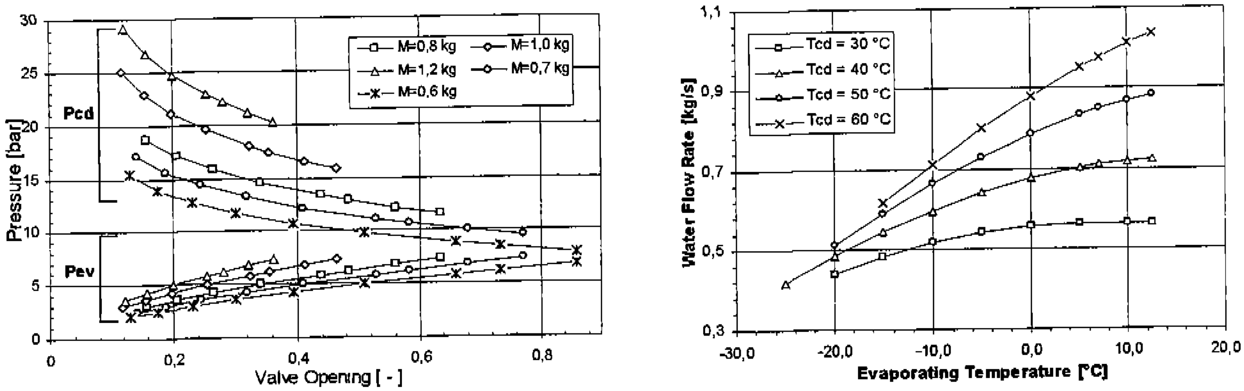
Simulation of the Cycle.

The governing equations defined above were solved simultaneously, to the known parameters, using the EES software (F-Chart, 1987). Thermodynamic fluid properties were calculated by built-in functions available with that program. The simulation was carried out for 30 operating points inside the application range of the compressor.

Figure 6(a) corresponds to simulation results, for a constant water flow rate at the desuperheater, showing the influence of both the refrigerant system charge and the expansion valve opening.

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.



(a) (b)

Figure 6 – (a) Refrigerant charge and expansion valve opening influence on the condensing and evaporating test conditions (water flow rate constant = 2.0 kg/s); (b) Water flow rate influence for fixed refrigerant charge and expansion valve opening.

CONCLUSIONS

The proposed superheated cycle seems to be a good alternative for the compressor performance rating. The main advantages are related with both the fast transient change between steady state test conditions and the installation simplicity. So, the new testing method would be specifically adequate for tasks like quality control, lifetime test, etc. The simulation carried out for the testing cycle, allow to verify the required change in the three control parameters, in order to impose to the compressor a well-defined operating condition. Next activities on this research will consider the experimental investigation of the new compressor testing method.

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