

**MERCOFRIO 2016 - 10º CONGRESSO INTERNACIONAL DE
AR CONDICIONADO, REFRIGERAÇÃO, AQUECIMENTO E VENTILAÇÃO**

**PERFORMANCE CORRECTION FACTORS FOR REFRIGERATION AND HEAT PUMP SYSTEMS
TESTED WITH UNCONTROLLED CONDENSER CONDITIONS**

Guilherme Maia Nobre – gmaianobre@hotmail.com

Adriano Akel Vasconcelos – adriano.ake@gmail.com

José Alberto Reis Parise – parise@puc-rio.br

Pontifícia Universidade Católica do Rio de Janeiro, Departamento de Engenharia Mecânica, www.puc-rio.br

Abdul Orlando Cárdenas Gómez – orlandocardenas1589@gmail.com

Enio Pedone Bandarra Filho – bandarra@mecanica.ufu.br

Universidade Federal de Uberlândia, Escola de Engenharia, www.ufu.br

R2 - Componentes e Equipamentos

Abstract. *A method for experimental data adjustment consisting of correction equations for the performance parameters of the cycle, like, compressor work or refrigerant effect, when operation conditions departed from those established in testing standards, is here presented. Correction equations are proposed for evaporator capacity and compression power consumption, for situations when condensing pressure has not followed standards conditions or has not been controlled during experiments. The method was tested against experimental data where it was possible to control the variation of the measured variables. Good agreement was obtained.*

Keywords: *Refrigeration testing standards, heat pump testing standards, vapor compression cycle, correction factors*

1. INTRODUCTION

The testing of refrigeration and heat pump systems, either in laboratory or field tests, and depending on the experimental apparatus, may present uncontrollable variables like condensing pressure, degree of subcooling in the liquid line and evaporator degree of superheat, whose variations, along the experimental runs, will end up marring overall results.

The present paper presents mathematical expressions, based on a simple thermodynamic refrigeration model, that correct calculated variables, derived from measured variables that are bound to vary, or oscillate, between experimental runs, and would preferably remain unchanged.

A similar procedure, for example, has been the practice for the testing of internal combustion engines (Giacosa, 2000). Being heat engines that operate open to the atmosphere, their performance is largely affected by parameters like atmospheric pressure, as well as ambient temperature and humidity. The difficulty in controlling such variables during tests entails their participation in the correction factor formulae, particularly, for shaft power.

Concerning heat pumps, Klein (2012) presents a discussion on what has to be considered when comparing performance ratings of such devices. Factors to be included and testing procedures to be followed are highlighted. According to Klein (2012), the determination of the performance of a heat pump, in Europe, is largely based on the EN14511 Standard (CEN, 2007), which defines the standardized conditions for testing electric heat pumps. These conditions, for water cooled condensers, include four temperature levels at the condenser, namely, 35°C, 45°C, 55°C and 65°C, all related to the outlet water temperature. However, as recognized by Klein (2012), the performance of the heat pump depends on the condensing temperature, which means that water mass flow rate is also adjusted, so as to obtain a 5°C water temperature difference across the condenser. For air-cooled heat pumps, a uniform inlet temperature of 20°C at the condenser is specified, which, of course, results in runs with different condensing temperatures.

The ANSI/AHRI Standard 210/240 specifies test operating conditions of unitary air conditioning and air-source heat pump equipment. These conditions rely on outdoor coil entering temperatures of 35°C and 27.8°C for steady-state tests in the cooling mode and indoor coil (condenser) air entering temperatures of 21.1°C, for steady-state or cyclic heating mode tests. The information gathered from two sampled standards suggests that if the performance of air conditioning or heat pump system is to be determined, by means of the usual parameters (COP, etc.) as a function of the condensing pressure, no specific standard would be available. Yet, results from systems tested under the conditions set forward by the standards above could be adjusted so that a constant condensing pressure could be obtained.

2. MATHEMATICAL MODEL

The correction factors are based on a simple thermodynamic mathematical model for the vapor compression refrigeration cycles. The model will be applied based on the assumption that the variation of the measured variable will be small enough to hold valid, within that range of variation, all the assumptions established for the model. In the present work, expressions will be derived for three correction factors due to variations of one measured variable, the condensing pressure. Its effect on the evaporator capacity and compressor shaft power will be determined.

A simple vapor compression refrigeration cycle, comprising condenser, expansion device, evaporator and compressor, will be considered. For the set of measured and calculated variables previously defined, the following assumptions apply.

- a) steady-state operation;
- b) negligible pressure drop across heat exchangers;
- c) negligible exchange of heat between refrigerant and heat exchangers neighborhoods;
- d) constant compressor isentropic efficiency;
- e) compressor volumetric efficiency only dependent on the re-expansion of the high pressure gas trapped in the clearance volume, with all other losses having a fixed reduction on the volumetric efficiency;
- f) the polytropic exponent remains constant throughout the experiment;
- g) compressor overall efficiency is a function of shaft speed only;
- h) isenthalpic expansion through the expansion device.

It should be noted that assumptions (b) to (g) need to be valid only within the small range of variation of the measured and calculated variables. The refrigerant effect, q_{ev} , and the refrigeration capacity of the cycle, $\dot{Q}_{ev}$, are given by, respectively:

$$q_{ev} = (h_1 - h_4) \quad (1)$$

$$\dot{Q}_{ev} = \dot{m}(h_1 - h_4) \quad (2)$$

where h_4 and h_1 are the refrigerant specific enthalpies at inlet and outlet of the evaporator, respectively. The refrigerant mass flow rate, $\dot{m}$, is:

$$\dot{m} = \frac{V_c}{v_1} N \left\{ 1 - r \left[\left(\frac{P_{cd}}{P_{ev}} \right)^{\frac{1}{n}} - 1 \right] \right\} C_v \quad (3)$$

where P_{cd} and P_{ev} are condensing and evaporating pressures, respectively; compressor geometry is defined by the displacement volume, V_c , and clearance ratio, r ; compressor operational conditions, by rotational speed, N , volumetric coefficient, C_v , which takes into account all other flow losses within the compressor, the polytropic exponent, n , and v_1 , the refrigerant specific volume at compressor suction. Likewise, the enthalpy gain during polytropic compression, $(h_2 - h_1)$, is:

$$\frac{(h_2 - h_1)}{(h_{2s} - h_1)} = \left(\frac{\frac{n}{n-1}}{\frac{\gamma}{\gamma-1}} \right) \left\{ \frac{\left[\left(\frac{P_{cd}}{P_{ev}} \right)^{\frac{n-1}{n}} - 1 \right]}{\left[\left(\frac{P_{cd}}{P_{ev}} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right]} \right\} \quad (4)$$

where h_{2s} is the specific enthalpy of refrigerant at the discharge of an isentropic compression and γ , the specific heat ratio.

$$h_{2s} = \hat{h}(s = s_1; P = P_{cd}) \quad (5)$$

$$\gamma = \frac{c_p}{c_v} \quad (6)$$

Compressor shaft power, $\dot{W}_{cp}$, is calculated in terms of refrigerant mass flow rate, specific enthalpy increase and overall compressor efficiency, η_{cp} :

$$\dot{W}_{cp} = \frac{\dot{m}(h_2 - h_1)}{\eta_{cp}} \quad (7)$$

Refrigerant temperature at the evaporator outlet, T_1 , is:

$$T_1 = T_{ev} + \Delta T_{sh} \quad (8)$$

where ΔT_{sh} is the evaporator degree of superheat and T_{ev} is the evaporating pressure. From assumption (g), the specific enthalpies at both inlet and outlet of the expansion device, h_3 and h_4 , respectively, are equal.

$$h_4 = h_3 \quad (9)$$

$$T_3 = T_{cd} - \Delta T_{sc} \quad (10)$$

where T_{cd} is the condensing temperature and ΔT_{sc} , the degree of subcooling. Therefore,

$$h_3 = \hat{h}_L(T_{cd} - \Delta T_{sc}) \quad (11)$$

where h_3 is the specific enthalpy of the liquid at the condenser outlet and $\hat{h}_L$ is the liquid specific enthalpy equation, which is, approximately, a function of the temperature only.

3. CORRECTION FACTORS

According to the model above, a vapor compression system, working at a condensing pressure P_{cd}^* different from that actually measured, P_{cd} , for a given experimental run, will imply new values for mass flow rate, $\dot{m}^*$, and compressor discharge and condenser outlet specific enthalpies, h_2^* and h_3^* , respectively.

$$\dot{m}^* = \frac{V_c}{v_1} N \left\{ 1 - r \left[\left(\frac{P_{cd}^*}{P_{ev}} \right)^{\frac{1}{n}} - 1 \right] \right\} C_v \quad (12)$$

$$h_3^* = \hat{h}_L(T_{cd}^* - \Delta T_{sc}) \quad (13)$$

Consequently, new values for refrigeration capacity and compressor power consumption, are calculated as follows:

$$\dot{Q}_{ev}^* = \dot{m}^* q_{ev}^* = \dot{m}^* (h_1 - h_3^*) \quad (14)$$

$$\dot{W}_{cp}^* = \frac{\dot{m}^* (h_2^* - h_1)}{\eta_{cp}} \quad (15)$$

Correction factors for refrigerant mass flow rate, refrigeration effect and compressor specific enthalpy gain, ξ_m , ξ_q and ξ_n , respectively, can be derived from the model, Eqs. (1) to (11), and Eqs. (12) to (15), above.

$$\xi_m = \frac{\dot{m}^*}{\dot{m}} = \left\{ 1 - r \left[\left(\frac{P_{cd}^*}{P_{ev}} \right)^{\frac{1}{n}} - 1 \right] \right\} \left\{ 1 - r \left[\left(\frac{P_{cd}}{P_{ev}} \right)^{\frac{1}{n}} - 1 \right] \right\}^{-1} \quad (16)$$

$$\xi_q = \frac{q_{ev}^*}{q_{ev}} = \frac{[h_1 - \hat{h}_L(T_{cd}^* - \Delta T_{sc})]}{[h_1 - \hat{h}_L(T_{cd} - \Delta T_{sc})]} \quad (17)$$

$$\xi_n = \frac{(\Delta h)_{cp}^*}{(\Delta h)_{cp}} = \frac{(h_{2s}^* - h_1)}{(h_{2s} - h_1)} \frac{\left[\left(\frac{P_{cd}^*}{P_{ev}} \right)^{\frac{n-1}{n}} - 1 \right]}{\left[\left(\frac{P_{cd}}{P_{ev}} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right]} \frac{\left[\left(\frac{P_{cd}}{P_{ev}} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right]}{\left[\left(\frac{P_{cd}}{P_{ev}} \right)^{\frac{n-1}{n}} - 1 \right]} \quad (18)$$

4. VALIDATION OF THE METHODOLOGY

The methodology was tested against experimental data where condensing and evaporating pressures were the controlled variables (Parise and Cartwright, 1988). A reciprocating, twin-cylinder, air-cooled, 2 5/8 in. x 2 1/2 in., open compressor was driven by a diesel engine. Condenser was water-cooled, of the shell-and-coil type and evaporator, dry-expansion, shell-and-coil type. The expansion device was a thermostatic expansion valve and refrigerant, CFC12. Condensing temperatures were nominally set at 30°C, 40°C and 50°C. The temperature of 40°C, and its corresponding saturation pressure, were set as the standard condition, and experimental data from all runs were converted according to the method outlined above. Figure 1 shows the variation of the refrigeration capacity, Eq. (2), from the original data, showing distinct curves for each one of the condensing temperatures, 30°C, 40°C and 50°C. Refrigeration capacity was divided by compressor speed, as this parameter was also made to vary in the original data. Application of the correction methodology outlined above resulted in a single group of points, Fig. 2. The straight lines in Figs. 1 and 2 are the linear fits of the correspondent data. The coefficients of determination, r^2 , for the 30°C, 40°C and 50°C curves in Fig. 1 were, respectively, 0.9449, 0.9764 and 0.9917, whereas for the single group of points, Fig. 2, corrected to a reference condensing temperature of 40°C, it resulted in 0.9711. It can be seen that, by grouping the experimental data under a single standard condensing pressure, did not affect adversely the scattering of the points, in spite of the relatively high latitude of the conversions ($\pm 10^\circ\text{C}$).

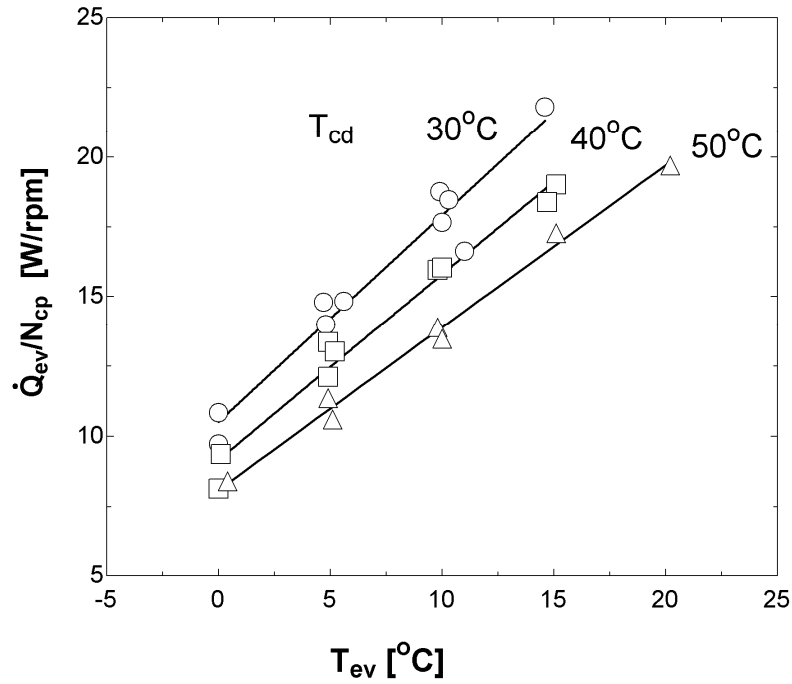


Figure 1. Refrigeration capacity per compressor speed as a function of evaporating and condensing temperatures (Parise and Cartwright, 1988).

Figure 3 shows the variation of the product of mass flow rate and refrigeration effect, which provides the correction for the refrigeration capacity. It can be seen that corrections ranged within $\pm 10\%$. Results are also shown for the correction of the enthalpy gain in the compressor, Figs. 4 and 5. The coefficients of determination, r^2 , for the 30°C, 40°C and 50°C curves in Fig. 4 were, respectively, 0.9263, 0.8592 and 0.8479, and, with all cases corrected to a condensing temperature of 40°C, 0.9355.

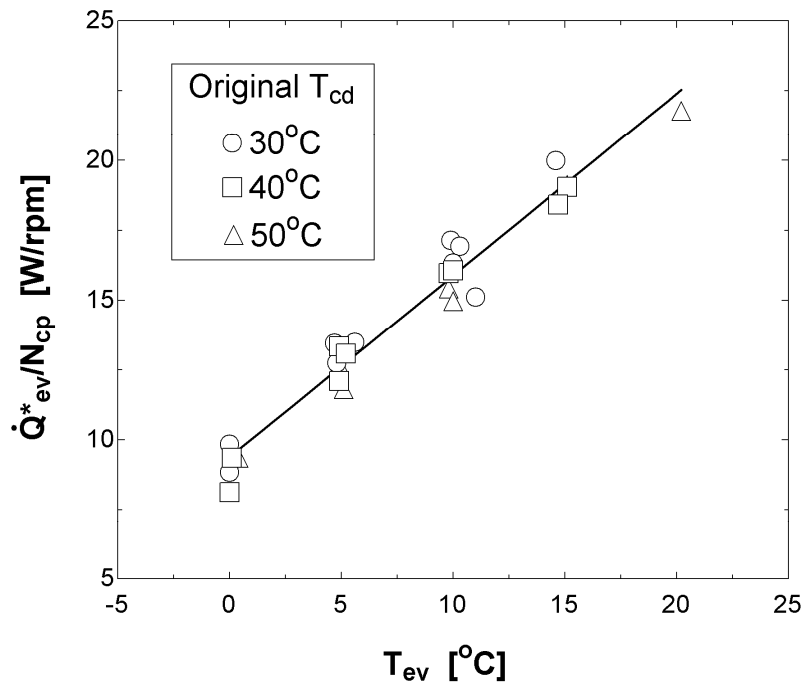


Figure 2 - Refrigeration capacity per compressor speed as a function of evaporating, for an standardized condensing temperature of 40°C.

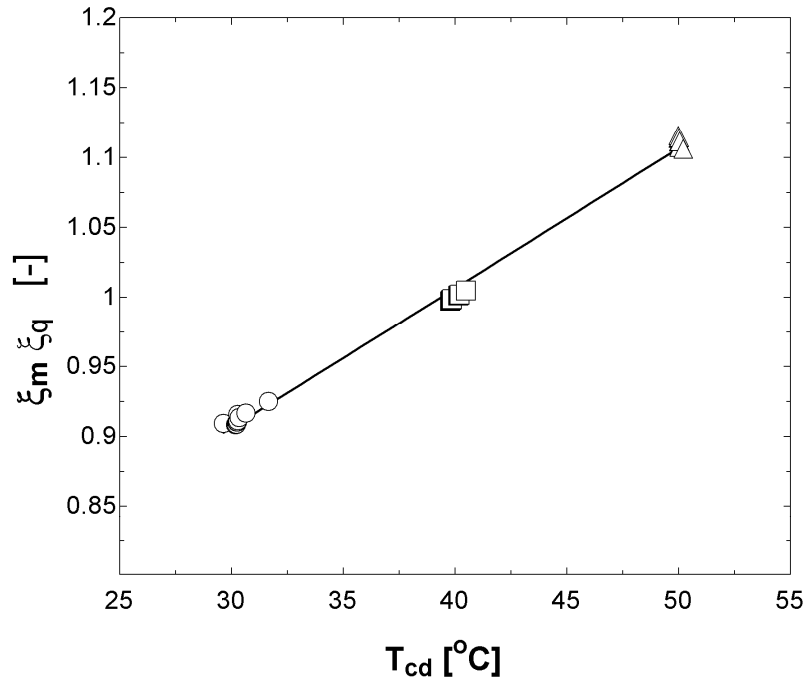


Figure 3 – Variation of the refrigeraton capacity correction factor with the condensing temperature. Legend as in Figure 2.

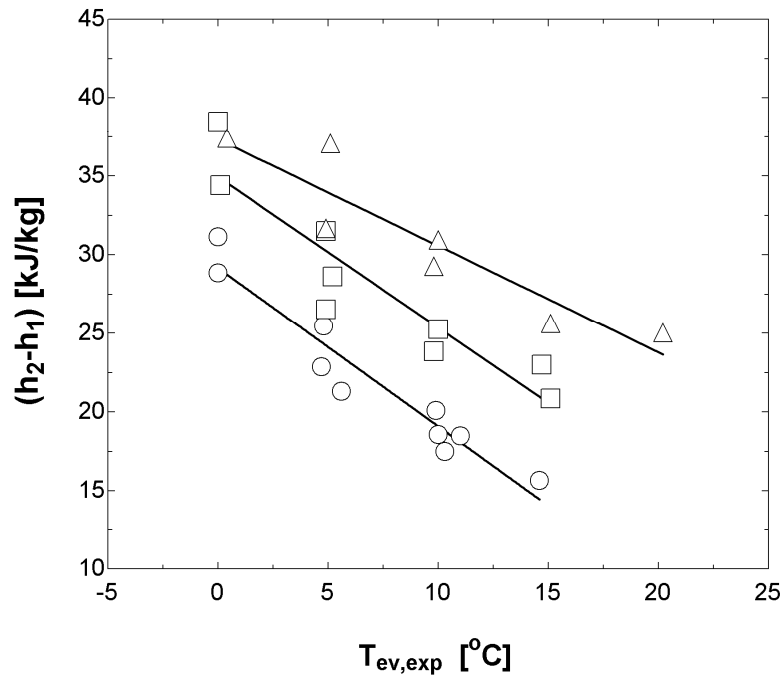


Figure 4 – Compressor specific enthalpy gain as a function of evaporating and condensing temperatures (Parise and Cartwright, 1988). Legend as in Figure 2.

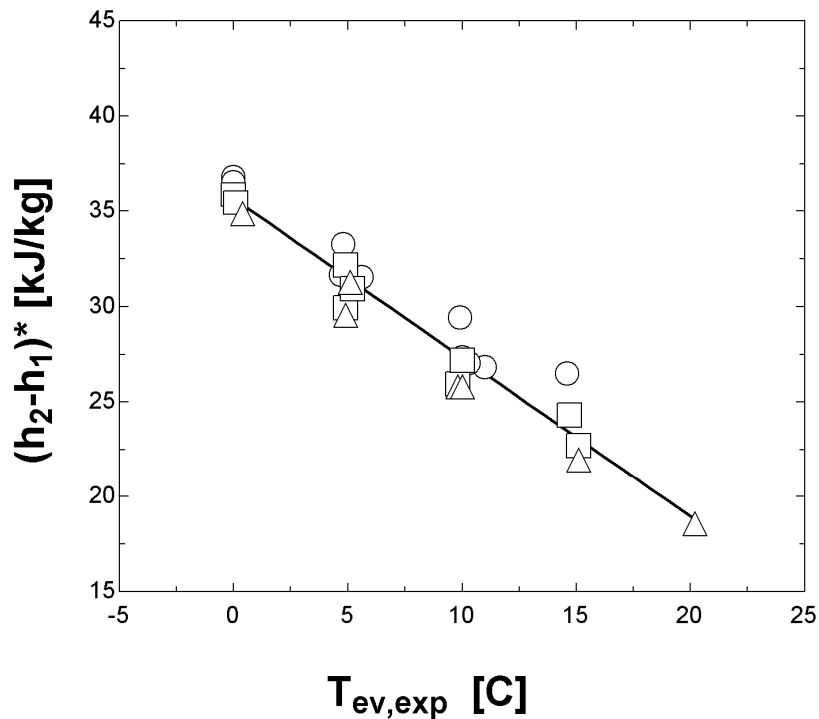


Figure 5 – Compressor specific enthalpy gain as a function of evaporating, for an standardized condensing temperature of 40°C. Legend as in Figure 2.

5. APPLICATION OF THE METHODOLOGY

The method was applied to a set of experimental data of a vapor-compression HCFC22 refrigeration system. Water-SWCNT (single walled carbon nano tube) nanofluid was tested as a secondary fluid for the 4-10 kW indirect vapour compression refrigeration system. The evaporator, with boiling refrigerant exchanging heat with the nanofluid, was of the brazed plate counter-flow type. A semi-hermetic compressor, an electronic expansion valve (EEV) and an air-cooled condenser were also the main components of the refrigeration cycle. Figure 6 shows the refrigeration capacity of a system with the secondary fluid (that flows through the evaporator), versus the evaporating temperature. Incidentally, one observes the superior capacity of the system operating with nanofluid, as a results of their enhanced thermal conductivity. Figure 7 depicts the same data with the refrigeration capacity now corrected by the method outlined in the present work. Comparison between graphs in Figs. 6 and 7 show subtle differences between original and corrected data points. This is expected, as the variation of the measured condensing pressure data points is not excessive, allowing for a choice of the reference condensing pressure relatively close to most of the runs). Nevertheless, Fig. 7, with corrected points of the refrigeration capacity versus secondary fluid, depicts a smoother distribution of the points.

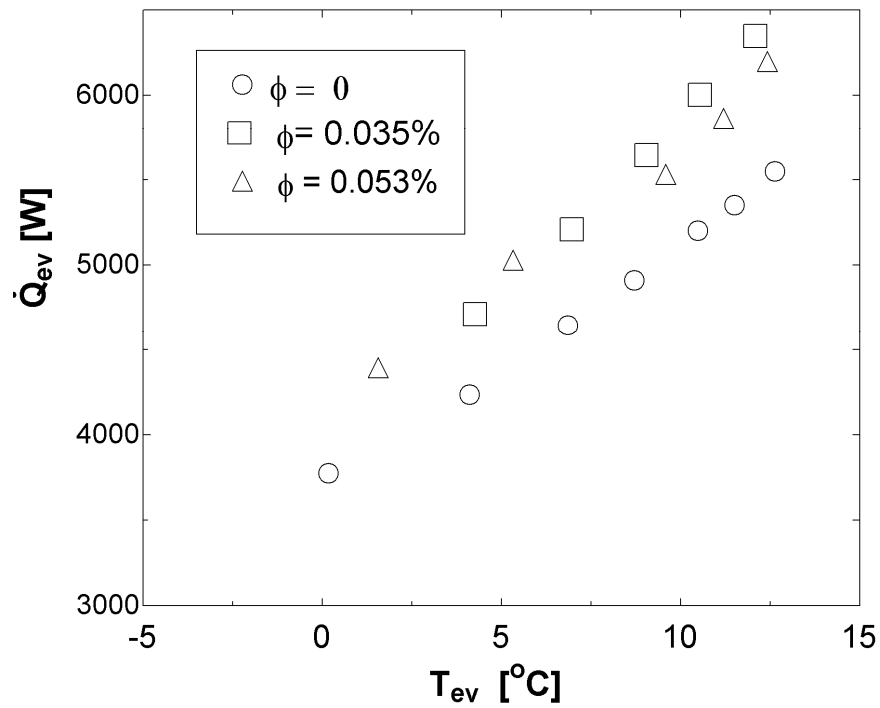


Figure 6 – Variation of refrigeration capacity with evaporating temperature, for pure water and water-SWCNT nanofluid at two fractions.

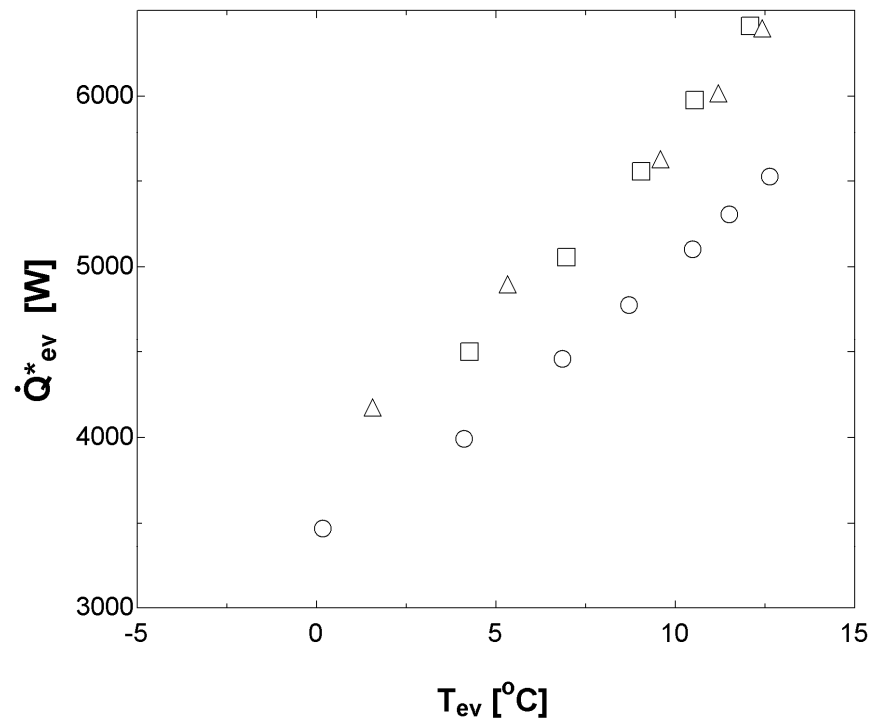


Figure 7 – Variation of corrected refrigeration capacity with evaporating temperature, for pure water and water-SWCNT nanofluid at two fractions. Legend as in Figure 6.

6. CONCLUDING REMARKS

A method was devised for the correction of refrigeration cycle performance results in experiments where condenser parameters are not controlled. The method is based on a simple thermodynamic model of the vapour compression cycle. It was verified against data from a set of CFC12 experiments where evaporating and condensing temperatures were the main controlling parameters. In spite of the large differences between standard and actual condensing pressures, results were encouraging, with a single curve compentently representing all points avialable in the set. The method was, then, applied to a set of experimental data of a vapor-compression HCFC22 refrigeration system with water or water-SWCNT nanofluid acting as the secondary fluid. It is felt that the smoothness of the data points improved, in comparison with original data.

The method herewith outlined may prove helpful to the researcher or engineer involved with experimental with uncontrolled parameters. This method could, of course, be extrapolated to other components of the vapour compression cycle, for both refrigeration and air conditioning or heat pump systems.

Acknowledgements

Authors are indebted to CAPES, CNPq, FAPERJ and FAPEMIG for the finantial support provided.

7. REFERENCES

- ANSI/AHRI, 2012, Performance Rating of Unitary Air-Conditioning & Air-Source Heat Pump Equipment, Standard 210/240-2008 WITH ADDENDUM 2, March.
- COMITÉ EUROPÉEN DE NORMALISATION (CEN), 2007, EN 14511-3, Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling - Part 3: Test methods, November.
- Giacosa, D. 2000. Motori Endotermici, quindicesima edizione, Editore HOEPLI.
- Klein, B., 2012. Independent testing of heat pumps is needed for reliable COP, REHVA Journal, October, pp. 15-18.
- Parise, J.A.R., Cartwright, W.G., 1988. Experimental analysis of a diesel engine driven water-to-water heat pump, Heat Recovery Systems & CIIP Vol. 8, No. 2, pp. 75-85.
- Vasconcelos, A.A., Cárdenas Gómez, A.O., Bandarra Filho, E. P., Parise, J.A.R., 2015. Experimental Evaluation of Water-SWCNT Nanofluid as a Secondary Fluid in a Refrigeration System, Joint Conference of 5th UK-China and 13th UK Particle Technology Forum, Leeds, 12-15 July.