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

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ABSTRACT

A method for experimental data adjustment consisting of correction equations for the performance parameters of the refrigeration/heat pump vapor compression cycle, when operation conditions depart from those established in testing standards, is here presented. A basic thermodynamic model allowed for a methodology to be developed so as to correct vapor compression cycle performance to a desirable operating condition. Correction factor equations are proposed for refrigerant mass flow rate, compressor specific enthalpy gain and evaporator refrigeration effect, for situations when condensing pressure has not followed standards conditions or has not been properly controlled during experiments. The method was verified against experimental data from a vapor compression water-to-water heat pump with controlled condensing temperatures of 30°C, 40°C and 50°C. In spite of the purposely excessive correction, ±10°C, a relatively good smoothness, as well as a good agreement among all conversions, was obtained with the standardized points. The model was also applied to a refrigeration system running with water-SWCNT nanofluid (single walled carbon nanotube with water as the base fluid) as the secondary fluid. It contributed to a better discernment of the actual influence of the nanofluid in the system performance.

NOMENCLATURE

c_p	specific heat at constant pressure, kJ/kgK
c_v	specific heat at constant volume, kJ/kgK
f_i	pressure ratio parameter, equation (18)
h	specific enthalpy, kJ/kg
$\hat{h}$	function for specific enthalpy, kJ/kg
$\hat{h}_L$	function for liquid specific enthalpy, kJ/kg
$\dot{m}$	mass flow rate, kg/s
n	polytropic exponent, -
N	compressor rotational speed, Hz
P	pressure, kPa
P_{cd}^*	normalized condensing pressure, kPa
q	heat transfer per unit mass of refrigerant, kJ/kg
$\dot{Q}$	rate of heat transfer, kW
r	clearance ratio, -
R^2	coefficient of determination, -
s	specific entropy, kJ/kgK
T	temperature, ^{o}C
v	specific volume, kJ/kg
V_c	compressor displaced volume, m^3
$\dot{W}_{cp}$	compressor shaft power, kW

Greek symbols

γ	specific heat ratio, -
ΔT_{sc}	degree of subcooling, K
ΔT_{sh}	degree of superheat, K
ξ_m	correction factor for mass flow rate, -
ξ_n	correction factor for compressor specific enthalpy gain, -
ξ_q	correction factor for refrigeration effect, -
η_{cp}	compressor overall efficiency, -
θ	pressure ratio parameter, equation (16), -

Subscripts

1	evaporator outlet, compressor inlet
2	compressor discharge, condenser inlet
2s	isentropic compression discharge
3	condenser outlet, expansion device inlet
4	evaporator inlet, expansion device outlet
cd	condenser

ev   evaporator
sc   subcooling
sh   superheat

Subscripts

*   refer to standardized operating conditions

INTRODUCTION

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

The present paper presents mathematical expressions, based on a simple thermodynamic refrigeration model, that correct calculated (or processed) variables, derived from measured (raw) 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 a long time, with the testing of internal combustion engines (Giacosa, 2000; Taylor, 1937). As 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 the utilization of correction factor formulae, particularly, for shaft power.

An overview in the literature reveals no refrigeration or heat pump standard (e.g., AHRI, 2007; European Standard, 2007) envisaging performance correction towards standard operating conditions. On the other hand, Vance Payne and Domanski (2006) worked with a Linear-Fit-Based Rating Procedure which has the inherent potential of providing more accurate ratings on heat pumps and air conditioners.

THERMODYNAMIC 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 compressor (1-2), condenser (2-3), expansion device (3-4) and evaporator (4-1) 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 thoughout 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:

$$q_{ev} = \left(h_1 - h_4 \right) \qquad (1)$$

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

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

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

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, $\left(h_2 - h_1 \right)$, 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.

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

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

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,

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

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.

CORRECTION FACTORS

According to the thmodynamic 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.

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

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

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

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

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

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.

$$\xi_m = \frac{\dot{m}^*}{\dot{m}} = \frac{\left(1 - r\theta^*\right)}{\left(1 - r\theta\right)} \quad (16)$$

where

$$\theta = \left[\left(\frac{P_{cd}}{P_{ev}}\right)^{\frac{1}{n}} - 1\right] \text{ and } \theta^* = \left[\left(\frac{P_{cd}^*}{P_{ev}}\right)^{\frac{1}{n}} - 1\right]$$

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

$$\xi_n = \frac{(\Delta h)_{cp}^*}{(\Delta h)_{cp}} = \frac{\left(h_{2s}^* - h_1\right)}{\left(h_{2s} - h_1\right)}\frac{f_n^*}{f_\gamma^*}\frac{f_\gamma}{f_n} \quad (18)$$

where

$$f_n^* = \left[\left(\frac{P_{cd}^*}{P_{ev}} \right)^{\frac{n-1}{n}} - 1 \right]; \quad f_\gamma^* = \left[\left(\frac{P_{cd}^*}{P_{ev}} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right];$$

$$f_\gamma = \left[\left(\frac{P_{cd}}{P_{ev}} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right]; \quad f_n = \left[\left(\frac{P_{cd}}{P_{ev}} \right)^{\frac{n-1}{n}} - 1 \right].$$

VALIDATION OF THE METHOD

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 ½ 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, was set as the standard condition. Then, experimental data from all runs were converted to 40°C condition, following 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 parameter correction methodology 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 (Di Bucchianico, 2008), 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 whole amount of points, seen as a single group, Fig. 2, corrected to a reference condensing temperature of 40°C, 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 lattitude of the conversions ($\pm$ 10°C).

Figure 3 shows the variation of the product of mass flow rate and refrigeration effect, wich 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.
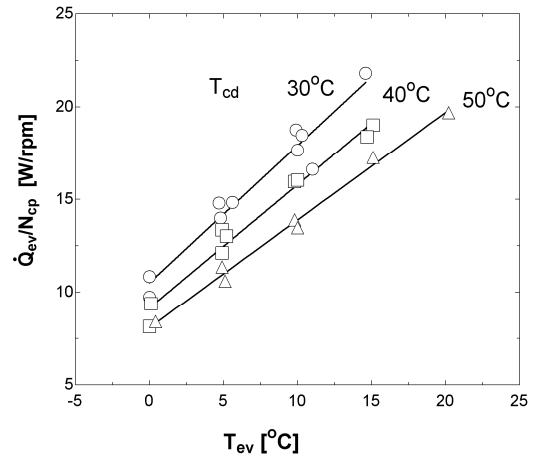

Figure 1. Refrigeration capacity per compressor speed as a function of evaporating and condensing temperatures (Parise and Cartwright, 1988).
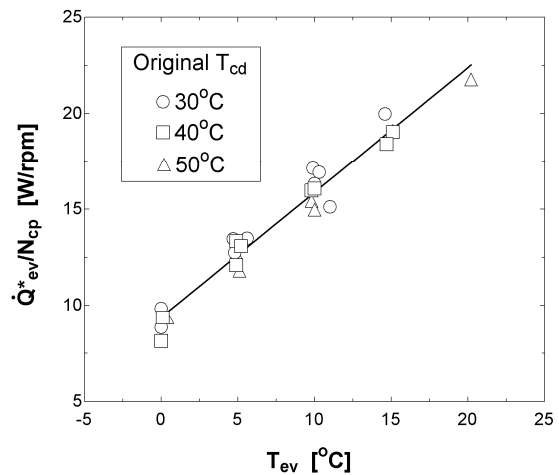

Figure 2. Refrigeration capacity per compressor speed as a function of evaporating, for an standardized condensing temperature of 40°C.
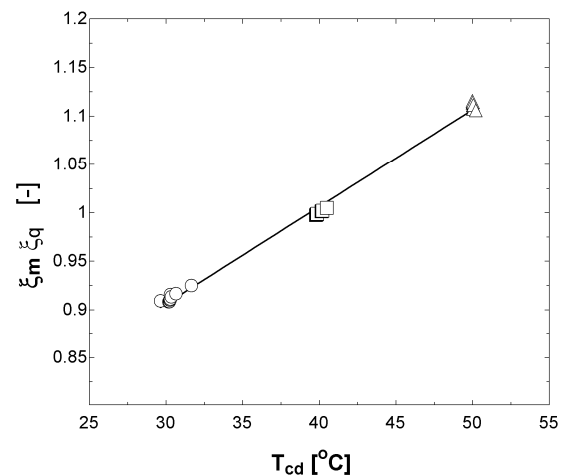

Figure 3. Variation of the refrigeraton capacity correction factor with the condensing temperature. Legend as in Figure 2.
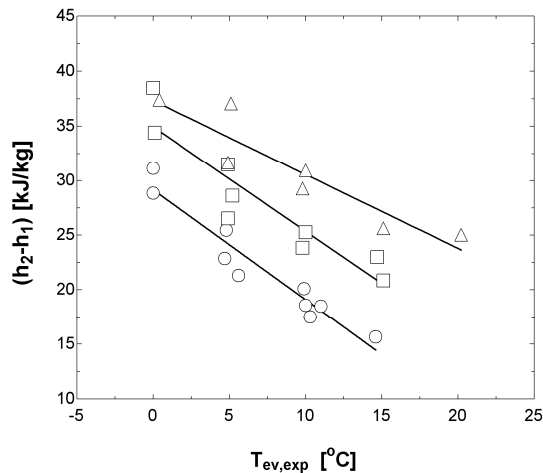
Figure 4. Compressor specific enthalpy gain as a function of evaporating and condensing temperatures (Parise and Cartwright, 1988). Legend as in Figure 2.
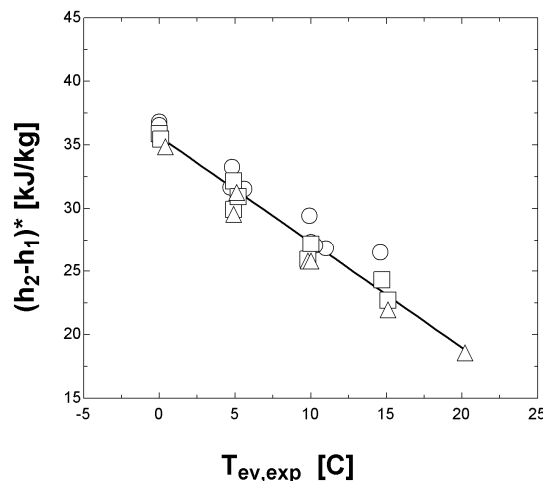

Figure 5. Compressor specific enthalpy gain as a function of evaporating, for an standardized condensing temperature of 40ºC. Legend as in Figure 2.

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 the 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.
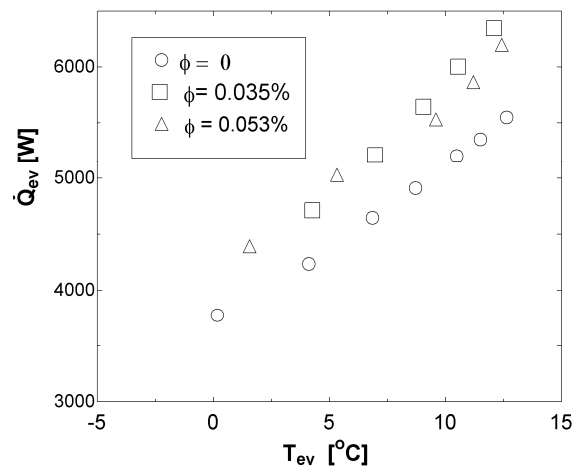

Figure 6. Variation of refrigeration capacity with evaporating temperature, for pure water and water-SWCNT nanofluid at two fractions.

CONCLUSIONS

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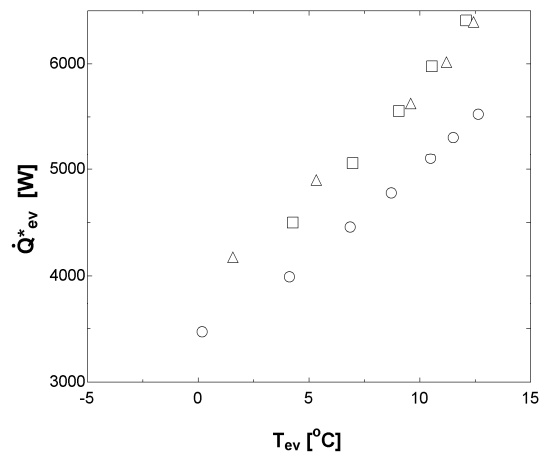
Figure 7. Variation of corrected refrigeration capacity with evaporating temperature, for pure water and water-SWCNT nanofluid at two fractions. Legend as in Figure 6.

ACKNOWLEDGEMENTS

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