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Characterization of the performance for water-to-water and air-to-water heat pumps: Development of simple and compact polynomial correlations

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ABSTRACT

Nowadays, the development of models is one of the most common problems in many research studies. In the field of refrigeration and heating, researchers increasingly require compact models with a reduced computational time. These models must be able to predict the performance of heat pumps and chillers with reasonable accuracy.

This paper describe the development of simple and compact polynomial models for the performance characterization in these machines. To this end, this study consists of two main parts: Obtaining the polynomial models and then its final adjustment with experimental data.

The unit selected for this study, is an innovative prototype of reversible dual source heat pump, with aerothermal and geothermal heat sources working with R32 as refrigerant. This has enabled us to analyze the response surface for the performance of the two main heat pump technologies: water-to-water and air-to-water heat pumps.

Keywords: *Refrigeration; Heat pumps; Empirical models; Regression analysis*

1. Introduction

Nowadays, the use of empirical polynomial models is widely extended in industry and research. The objective of these simple models is to characterize, with enough accuracy, the behavior of systems or their components. This allow us to improve the design and efficiency of the equipment using these type of models instead of testing with the corresponding development cost savings. In the field of refrigeration systems, the AHRI polynomials are a clear example. These functionals predicts the compressor performance for their entire work map, adjusting 10 coefficients for the bivariate cubic polynomials provided by the AHRI 540 standard[1].

Alternatively, another possible approach is the development of more complex models, including the main physical processes that have an influence on the system. However, the disadvantages of these methodologies are the increase of computational time and number of inputs.

Considering the empirical models as a basis for work, this study shows the approach used by the authors to propose robust and high-quality empirical models for the performance prediction on Heat Pumps (HPs). The two main disadvantages of the empirical models is the need of extended experimental data for the polynomials adjustment and the inability to extrapolate results out of the experimental range.

According to this, the document is structured in two main parts. The first part describes the process to get the functional that characterizes the Response Surface (RS) of HP's performance with a maximum relative error less than 10%. The parameters that define the HP's performance are the condenser capacity (Qc) and the compressor power input (Wc).

Finally, the second part shows the adjustment of the final correlations developed in the first part to the experimental data for the analyzed operating modes of the unit (water-to-brine and air-to-water HP).

The entire experimental database used in this study originates from the Geot€ch project[2], a 4 years' duration project (2015-2018) funding by the European Commission. One of the main research lines is dedicated to the development of an innovative prototype of Dual Source Heat Pump (DSHP). This DSHP combines the aerothermal and geothermal technology in a single equipment allowing it to work in one mode or the other one. Specifically, this study uses the experimental data obtained for the Ground Source Heat Pump (GSHP) mode, a water-to-brine HP and for the Aerothermal Source Heat Pump (ASHP), an air-to-water HP.

2. Test campaign

A total of 227 test points were used as experimental database in this study. These experimental data were collected from the experimental campaign performed on a previous study, Marchante et al.(2018)[3]. Table 1 shows a summary of the experimental data collected:

Table 1. DSHP experimental database

Test	***M1 Summer Air***	***M2 Summer Ground***	***M3 DHW User***	***M4 Winter Air***	***M5 Winter Ground***	***M6 DHW Air***	***M7 DHW Ground***
Nominal conditions	1	1	1	2	2	2	2
Variation fc (User 40/45)	-	-	-	4	4	-	-
Variation fc (User 30/35)	-	-	-	4	4	-	-
Variation fan freq. (User 40/45)	-	-	-	3	-	-	-
Frost	-	-	-	4	-	-	-
Test Matrix	30	20	20	30	30	20	20
Winter Ground as DHW	-	-	-	-	3	-	-
Double evaporator (WA/WG)	-	-	-	4	-	-	-
Extra test	1	1	2	-	6	3	3
Total test points: 227							

The HP analysed is a reversible DSHP with 7 operating modes for supply hot water, cold water and domestic hot water, using two possible sinks/sources (ground or air). Further details of the design can be found in Marchante et al.(2018)[4].

Based on [3], this paper shows a more extended analysis of the collected data. We have expanded the initial parametric study carried out in [3] from 640 to 3125 points. The following section describe the use of it and how it is carried out.

The final correlations are obtained and adjusted only for the operating mode as geothermal HP (Winter Ground) and aerothermal HP (Winter Air) with the experimental data. However, the methodology presented can be used for the other operating modes. The rest of the experimental database is used in order to update the submodel of the compressor in the IMST-ART model (see section 3 and 3.1).

3. Development of the empirical correlations

The development process of new empirical correlations is one of the most common problems faced by industry or researchers. The building of models is often linked to the search for functions able to describing the RS of the equipment. We can define the RS as the surface of performance measure or quality characteristic of the analyzed process. We know as response this performance measure typically measured on a continuous scale. Furthermore, we will control the independent variables during tests and they will define the response value.

Identifying the variables involved in our process, Figure 1 shows a schematic diagram of the operating modes analyzed. The first one, Winter Ground, works as a GSHP with 2 Brazed Plate Heat Exchangers (BPHEs). One of them connects the HP with the user loop (condenser) and the other one links with the ground source (evaporator). The other one, Winter Air, works as ASHP using the same BPHE for the user loop and a Round Tube Plate Fin Heat Exchanger (RTPFHx) as evaporator in the Air source. The compressor installed in this prototype is an inverter driven compressor. As we can see below, Figure 1 identifies the following independent and response variables in each working mode:

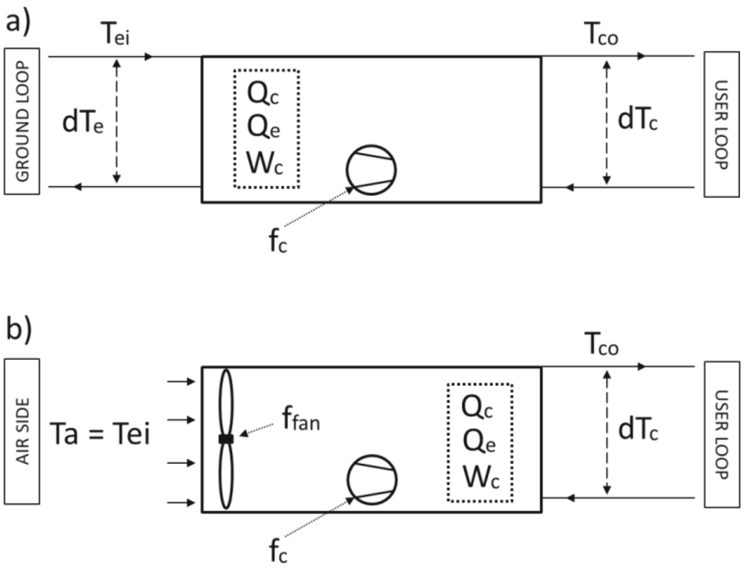

Figure 1: Response and independent variables for Winter Ground (a) and Winter Air (b)

On the one hand, the secondary fluid outlet temperature, Tco (User loop in this case), and the secondary fluid temperature difference across the condenser, dTc, set the conditions in it for both operating modes. In practice, the mass flow rate is what is modified in the test bench but the temperature difference is the controlled variable.

On the other hand, the secondary fluid inlet temperature to the evaporator, Tei (brine in this case), and the secondary fluid temperature difference across the evaporator, dTe, are the independent variables in the evaporator side for Winter Ground mode. By contrast the Winter Air mode sets the conditions in the evaporator with Tei (air inlet temperature to the RTPFHx) and ffan (fan frequency).

The last independent variable, in both modes, to take into account is the compressor frequency, fc. Finally, condenser capacity (Qc) and compressor power input (Wc) are the response variables which we want to correlate.

As can be observed, we select the control variables over the secondary loops as independent variables. Other possible approach would be to select the independent variables referred to primary loop. This option was rejected because the controlled variables in the test bench are the independent variables listed above. This corresponds to the normal operation of these systems. The source conditions and user demand, together with the compressor frequency, set the condensation and evaporation temperature in the refrigerant (Tc, Te).

According to this selection of independent variables, Allen and Hamilton (1983)[5] and Jin and Spitler (2002)[6] already presented the use of second order degree polynomials to predict the full load performance of chillers. This applies in the performance characterization of units with constant speed compressors, correcting for partial load operation. However, new refrigeration/heat pump units are currently incorporating variable speed compressors and also variable speed fans and water pumps.

Once we have identified all the independent variables, the second step is to look for the function able to predict the response variables with highest precision. In general, the true response function is unknown and perhaps very complicated but we can to obtain a good representation using regression analysis. There are many statistical methods but at the end, a simple division of all of these tools allows us to select parametric or non-parametric regression methods. The first one gives us a clear function for the response easy to program and it will be the selected branch, specifically the linear regression for their extensive applications and its relative ease of use.

Selected the analysis tools, the third step is to raise the methodology of study. In linear regression, we need a lot of data to study the better functional and adjusting its coefficients. The main problem is the limited number of experimental points that we can to do. This total number of test will depend on the levels selected for each independent variables:

- 3 levels for each variable → 35 = 243 test points
- 4 levels for each variable → 45 = 1024 test points
- 5 levels for each variable → 55 = 3125 test points

With the objective of detect any non-linear variation in the response, we selected 5 levels for each independent variable with a total of 3125 test points. This approach allow us to obtain a high-resolution mesh grid of the response surface. Of course, the huge quantity of test does not allow getting them by experimentation. In order to resolve that problem, the approach selected was to make the parametric study using an accuracy simulation tool, IMST-ART[7], allowing to obtain the full factorial fixed.

This model in IMST-ART was built and validated in [3]. The HP analyzed works with R32 as refrigerant. However, the submodel compressor in IMST-ART of [3] used the AHRI polynomial working with R410A because manufacturer did not have available data for R32. Consequently, we decide to update the compressor submodel using the experimental database available in the present study.

3.1 Compressor submodel and parametric study with IMST-ART

Many studies have been carried out on the modeling method of constant speed compressor. Nowadays, the AHRI polynomials are the most extended method used by manufacturers, defining the compressor performance as function of the condenser temperature (Tc) and the evaporator temperature (Te). Nevertheless, for the inverter compressor, the frequency is an additional parameter.

Another possible approach is the characterization of its efficiencies, equations 1, 2. This is a simpler model with enough accuracy and dimensionless results that it will enable us to analyze the effect of the frequency taking it into account.

$$\eta_c = \frac{\dot{m}_{ref}\Delta h_{is}}{W_c} \quad (1) \qquad\qquad \eta_v = \frac{\dot{m}_{ref}}{\rho_{suction}\dot{V}} \quad (2)$$

In order to update the compressor submodel in IMST-ART, we start to study the 227 experimental points and to calculate the efficiencies. The compressor power input is measured in the test bench and the refrigerant mass flow rate is calculated by heat balance of the secondary loop (the variables measured are the secondary mass flow rate and secondary inlet and outlet temperatures).

Concerning the compressor efficiency correlation, it is characterized by equation 3, which is composed by terms that depend on the pressure ratio (from c_1 to c_4) and by a variable intercept according to the compressor frequency (fc). Therefore, for each fc, the coefficients from k_0 to k_2 are fixed and the compressor efficiency curve is displaced with a different K_0 intercept (see Figure 2a).

$$\eta_c = (k_0 + k_1 \cdot f_c + k_2 \cdot {f_c}^2) + c_1 \cdot r_p + c_2 \cdot {r_p}^2 + c_3 \cdot {r_p}^3 + c_4 \cdot {r_p}^4 = K_0 + c_1 \cdot r_p + c_2 \cdot {r_p}^2 + c_3 \cdot {r_p}^3 + c_4 \cdot {r_p}^4 \quad (3)$$

This correlation has a maximum deviation of ±7.20 %, compared to the experimental points, and the difference for some of the compressor frequencies can be seen in Figure 2b. Furthermore, Figure 2a shows only the calculated values taking into account the equation 3 for each compressor frequency with an optimal frequency between 50 and 70 Hz.

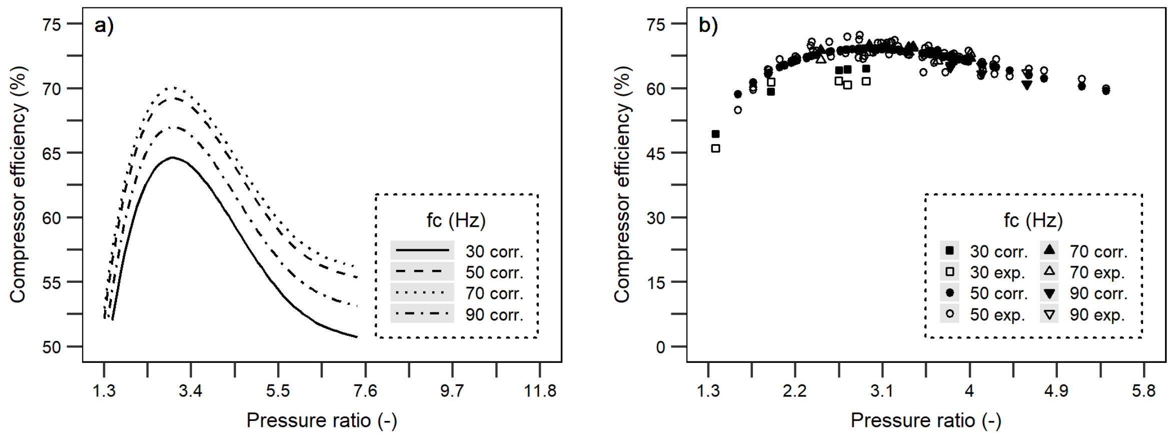

Figure 2. Compressor efficiency correlation dependent on pressure ratio (a) and comparison of the experimental and correlated points at ns = 30, 50 and 70 Hz (b).

Regarding the volumetric efficiency correlation, it has a similar expression with less terms (the volumetric efficiency has less curvature with the pressure ratio). It will also be corrected for each frequency with the intercept and its maximum deviation is 7.22%.

$$\eta_v = \left(k_0 + k_1 \cdot f_c + k_2 \cdot {f_c}^2\right) + c_1 \cdot r_p + c_2 \cdot {r_p}^2 + c_3 \cdot {r_p}^4 = K_0 + c_1 \cdot r_p + c_2 \cdot {r_p}^2 + c_3 \cdot {r_p}^4 \quad (4)$$

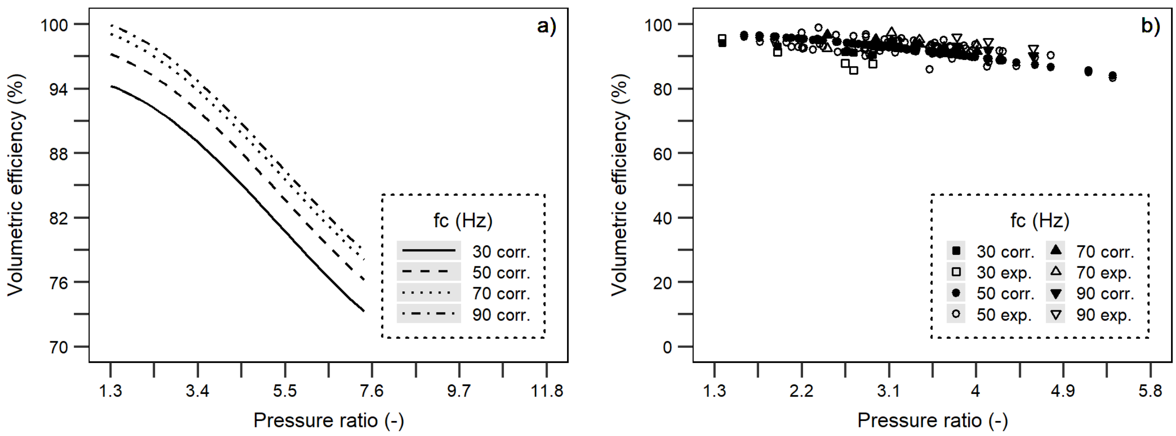

Figure 3 Volumetric efficiency correlation dependent on pressure ratio (a) and comparison of the experimental and correlated points at n_s = 30, 50 and 70 Hz (b).

Once the sub-model of the compressor has been updated with equations 3 and 4, the 3125 points for each operating mode are simulated in IMST-ART. Tables 2 (Winter Ground mode) and 3 (Winter Air mode) shows the levels considered for each independent variable.

Table 2. Independent variables levels for Winter Ground Mode.

Independent variables	Range values	Independent variables	Range values
fc	[30, 40, 50, 60, 70] Hz	Tei	[-5, 0, 5, 10, 15] ºC
Tco	[35, 40, 45, 50, 55] ºC	dTe	[2, 3, 5, 7, 9] ºC
dTc	[2, 3, 5, 7, 9] K		

Table 3. Independent variables levels for Winter Air Mode.

Independent variables	Range values	Independent variables	Range values
fc	[30, 40, 50, 60, 70] Hz	Tei	[4, 7, 11, 15, 19] ºC
Tco	[35, 40, 45, 50, 55] ºC	fan	[20, 35, 50, 65, 80] %
dTc	[2, 3, 5, 7, 9] K		

3.2 Modelling HPs performance

With a large number of points (3125) and an efficient suite of regression tools, R Core Team (2018)[8], we are able to study the better polynomial for Qc, and Wc. In order to obtain them, we need to develop a polynomial including the best selection of terms able to describe the physical effects of the process. Based on second-order polynomial models with interactions, we can identify 3 possible type of terms.

- Simple linear terms: They are used to model the main effects of the process.
- Squared terms: They are used to model curvature in the response surface.
- Cross-product terms: They are used to model interaction between the independent variables

The first problem is the large number of variables involved in the process. Nowadays, the HP manufacturers provides their units with compressors and circulation pumps with variable speed. This has the effect of increase the number of independent variables to 5 defining a 5D domain for the response variables. Therefore, the first step will be to simplify the problem.

As figures 4a and 4b shows, compressor power input region changes depending on the mode: for the Winter Ground mode (brine-to-water heat pump) it moves from pressure ratio 2 to 7.5 approximately, and for the higher values of r_p a curvature in the input power distribution can be seen. On the other hand, Winter Air mode (air-to-water heat pump) is located in lower pressure ratio values (from 1.7 to 5) and only for the lowest values of r_p a small deflexion in the input power distribution can be found.

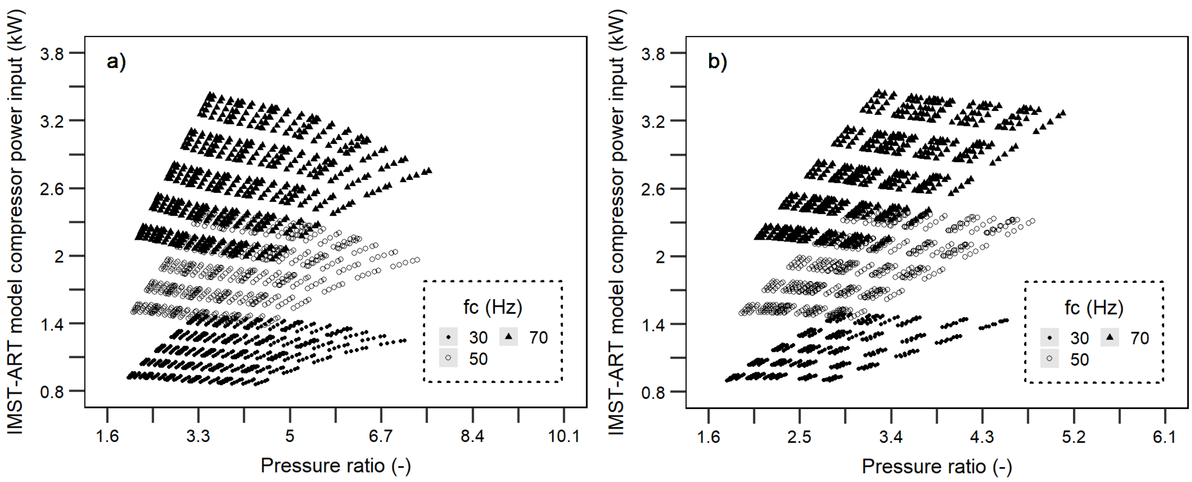

Figure 4. Compressor power input dependent on the pressure ratio for a) Winter Ground and b) Winter Air Mode.

In order to improve the correlation fitting of the unit power input and reduce the effects of the compressor frequency as there is an evident dependence between them, power input is divided by f_c, so the major effects that can be seen in figures 4a and 4b are mostly removed (Figures 5a and 5b) and it only remains some secondary influence with the frequency that is corrected by adding terms with this response variable (f_c).

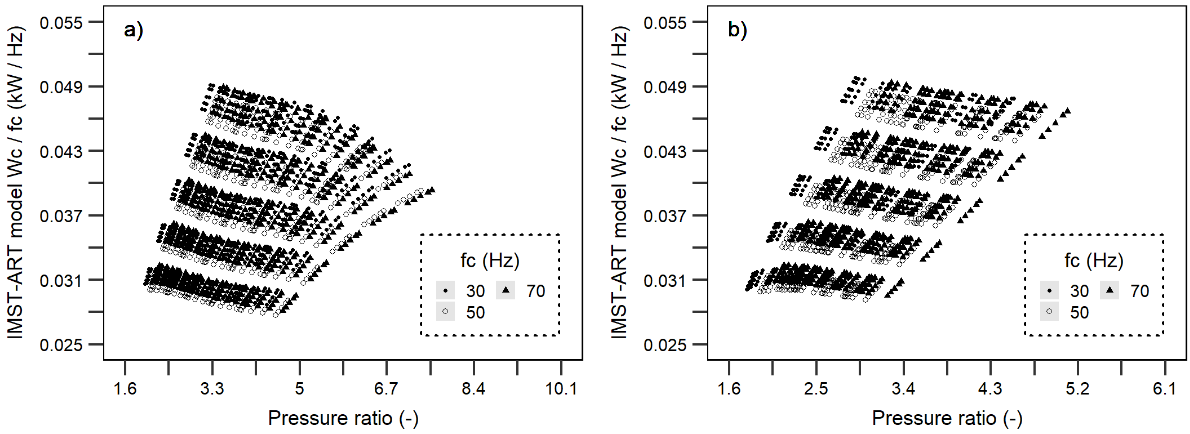

Figure 5. Compressor power input divided by the frequency dependent on the pressure ratio for a) Winter Ground and b) Winter Air Mode.

The methodology used to obtain the correlations for the characterization of the response variables is summarized in Figure 6. It consists on starting with a second-degree polynomial model with interactions, which means a large number of terms (N coefficients). Equation 5 shows an example with two independent variables.

$$y = \beta_0 + \beta_1 \cdot x_1 + \beta_2 \cdot x_2 + \beta_{11} \cdot x_1^2 + \beta_{22} \cdot x_2^2 + \beta_{12} \cdot x_1 \cdot x_2 \tag{5}$$

This first polynomial is reduced for obtaining a more compact expression by means of an initial reduction of coefficients with stepwise regression with Akaike criterion (AIC)[9]. Specifically, bidirectional regression has been used, which combines the forward and backward strategies, meaning that response variables are added or deleted following the AIC criterion. This step will result into eliminating non-significant terms (M coefficients), leaving after this stage a (N-M) coefficients polynomial.

Afterwards, in order to obtain more compact expressions, we apply a reduction of terms with an iterative process using Pearson's partial correlation coefficient as indicator, removing in each iteration the coefficient that has the lowest value of this indicator. The result is a polynomial with (N-M-i) coefficients and i iterations to obtain a final polynomial with a reasonable number of terms (10-15 terms) and enough accuracy (error < 5%).

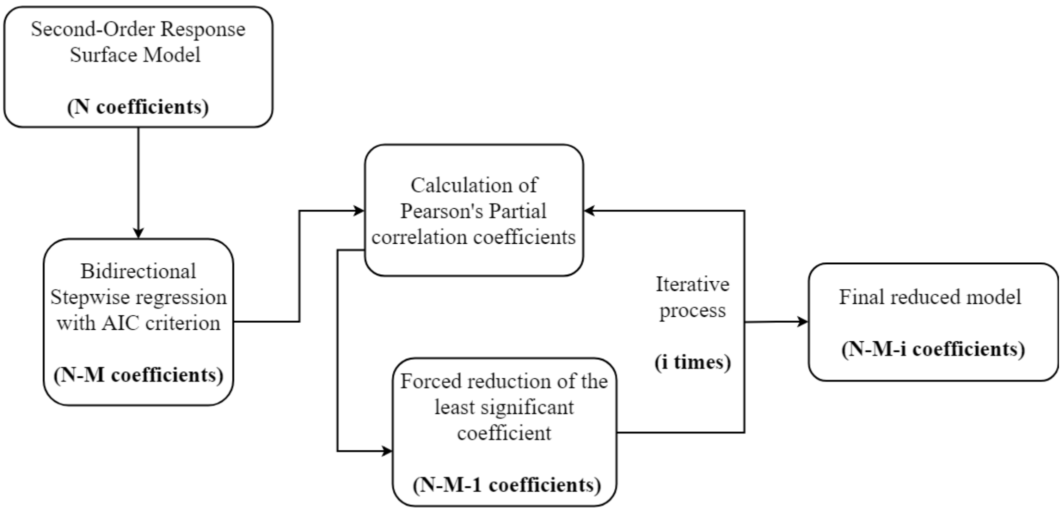

Figure 6. Applied methodology for obtaining the final compact models.

The results of implementing this methodology for the compressor power input are shown in figures 7a for Winter Ground and 7b for Winter Air mode.

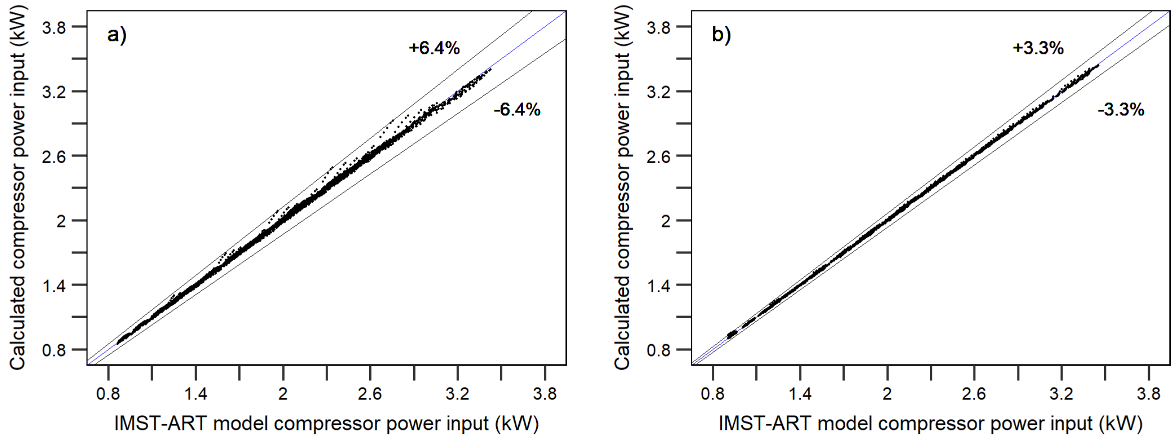

Figure 7. Compressor power input calculated for Winter Ground a) and Winter Air Mode b).

Figure 7b a good prediction for Winter Air mode with a maximum deviation of 3.3 % and the final polynomial is shown in equation 7.

$$\frac{W_c}{f_c} = k_0 + k_1 \cdot T_{co} + k_2 \cdot dT_c + k_3 \cdot T_{co}^2 + k_4 \cdot T_{eo}^2 + k_5 \cdot f_c^2 + k_6 \cdot T_{co} \cdot f_c + k_7 \cdot T_{co} \cdot T_{eo} + k_8 \cdot T_{co} \cdot dT_c \qquad (6)$$

$$\frac{W_c}{f_c} = k_0 + k_1 \cdot T_{co} + k_2 \cdot dT_c + k_3 \cdot T_{co}^2 + k_4 \cdot T_{ei}^2 + k_5 \cdot f_c^2 + k_6 \cdot Tco \cdot fc + k_7 \cdot f_c \cdot fan + k_8 \cdot T_{co} \cdot T_{ei} + k_9 \cdot T_{co} \cdot dT_c \qquad (7)$$

Nevertheless, for Winter Ground mode (Figure 7a) some families of points do not correlate properly and thus this leads to a higher deviation. This fact is explained due to the curvature that is shown in Figure 4a for high values of r_p. Therefore, it will be necessary to correct equation 6 and to take into account the effect of the pressure ratio.

According to this, an adaptation of August law (which is a simplified form of Antoine equation) is used. This law estimates the pressure ratio as a function of the condenser and evaporator temperatures of refrigerant. However, as we do not directly know T_c and T_e, August law is modified with an expression depending on T_{co} and T_{eo}. This is feasible considering that heat exchangers are counter-current and the secondary outlet temperatures are a representation of the condenser and evaporator temperatures. Furthermore, an intercept is used in the expression (equation 8) in order to improve the prediction of pressure ratio (see Figure 8a).

$$\log(r_p) = k_0 + k_1 \cdot \left(\frac{1}{T_{co}} \cdot \left(1 - \frac{T_{co}}{T_{eo}}\right)\right) \qquad (8)$$

The final polynomial for the compressor power input in Winter Ground mode is shown in equation 9, where two terms with the pressure ratio have been added (r_{pp}). It is important to note that the term r_{pp} is a calculated value of the pressure ratio applying equation 8. Figure 8b shows the maximum deviation for the equation 9 (maximum error: 3.3%).

$$\frac{W_c}{f_c} = k_0 + k_1 \cdot T_{co} + k_2 \cdot dT_c + k_3 \cdot T_{co}^2 + k_4 \cdot T_{eo}^2 + k_5 \cdot f_c^2 + k_6 \cdot T_{co} \cdot f_c + k_7 \cdot T_{co} \cdot T_{eo} + k_8 \cdot T_{co} \cdot dT_c + k_9 \cdot r_{pp} + k_{10} \cdot r_{pp}^2 \qquad (9)$$

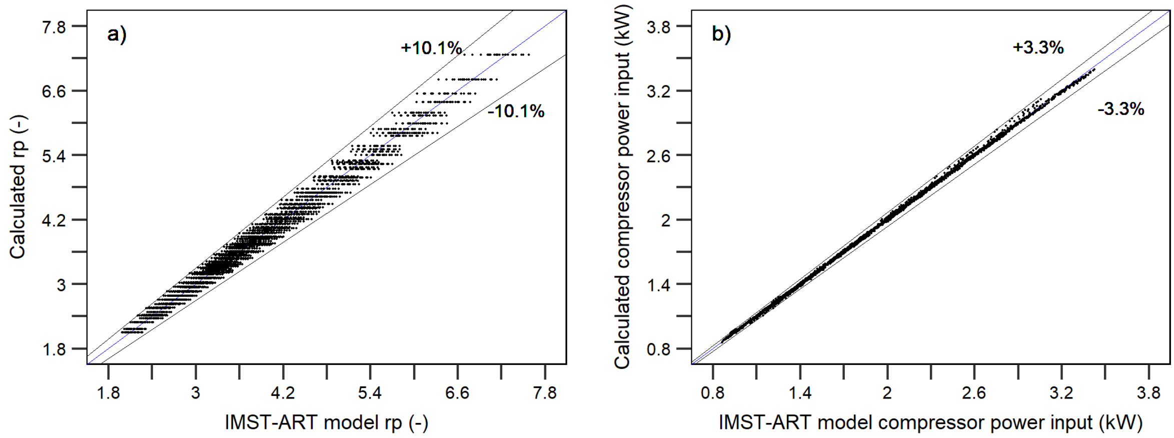

Figure 8. Result of pressure ratio correlation in Winter Ground Mode (a) and Compressor power input corrected with the pressure ratio calculated with Eq 8 (b).

Regarding the condenser capacity correlation, same methodology as described before has been followed. It is important to note that in the condenser capacity correlation for Winter Air mode has been necessary to add an extra independent variable, the relative humidity (RH). This is necessary due to the difficulty of adjustment only taking into account the dry bulb inlet temperature of the evaporator and the frequency of its fan, giving better fitting results introducing this additional variable.

Finally, the final polynomials for the condenser capacity prediction are the equation 10 for Winter Ground mode and 11 for Winter Air mode, giving both a maximum deviation lower than 3 % (Figures 9 a and b).

$$\frac{Q_c}{f_c} = k_0 + k_1 \cdot T_{co} + k_2 \cdot T_{eo} + k_3 \cdot T_{eo}^2 + k_4 \cdot dT_e^2 + k_5 \cdot T_{eo} \cdot f_c + k_6 \cdot T_{co} \cdot T_{eo} + k_7 \cdot T_{co} \cdot dT_c + k_8 \cdot T_{eo} \cdot dT_e \quad (10)$$

$$\frac{Q_c}{f_c} = k_0 + k_1 \cdot T_{ei}^2 + k_2 \cdot fan^2 + k_3 \cdot f_c \cdot T_{ei} + k_4 \cdot f_c \cdot fan + k_5 \cdot f_c \cdot RH + k_6 \cdot T_{ei} \cdot T_{co} + k_7 \cdot T_{co} \cdot dT_c + k_8 \cdot fan \cdot T_{co} + k_9 \cdot T_{ei} \cdot fan + k_{10} \cdot fan \cdot RH \quad (11)$$

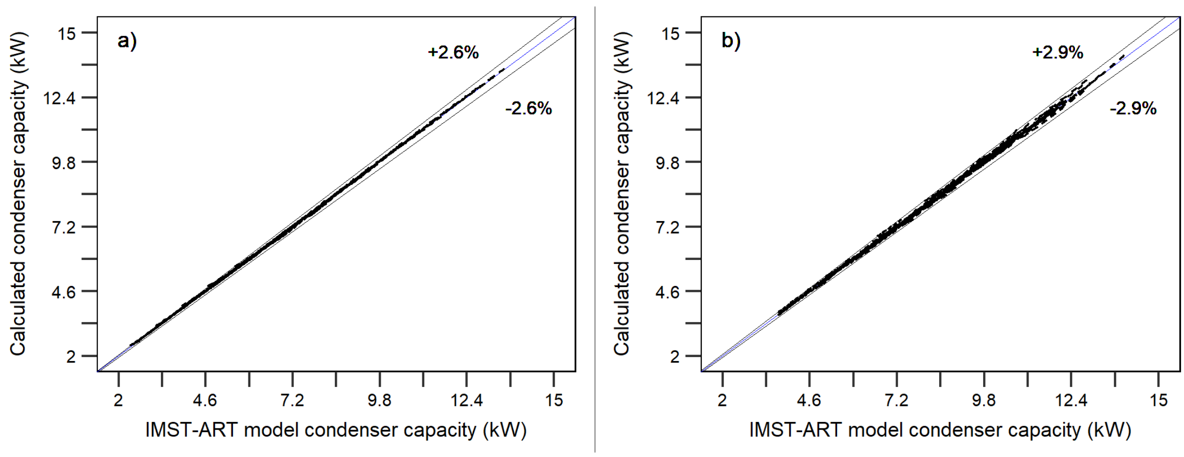

Figure 9. Condenser capacity for Winter Ground (a) Winter Air Mode (b).

3.3 Correlations adjustment with experimental data

This final sections shows the process of adjustment to the experimental data the polynomials for the prediction of the HP's performance. For that purpose, the coefficients obtained with the equations 7 and 9 for the compressor power input and the equations 10 and 11 for the condenser capacity have been fixed, and Qc and Wc have been calculated for the experimental available.

The reason for fixing the coefficients of the correlations is to avoid its mismatch. This mismatch would be given by the change of the coefficients signs or the order of magnitude. In this way, the deviations obtained will be due to possible imbalances between the IMST-ART model and the experimental data.

Applying the approach above we could observe a good prediction for the compressor power input correlations in both working modes (Figure 11). However, the prediction for the condenser capacity presented more than 10% of maximum error. Figure 10 shows the results for the condenser capacity correlations, 10a for Winter Ground mode and 10 b for Winter Air mode.

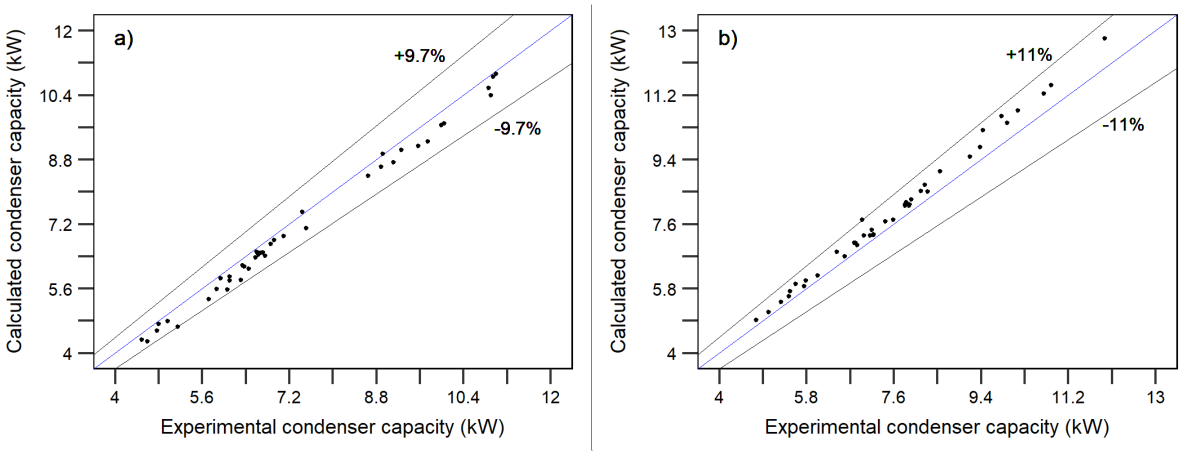

Figure 10. Experimental condenser capacity for Winter Ground mode (a) and Winter Air (b) without intercept adjustment.

We can observe in the figures above that the correlations for the Winter Ground mode (Figure 10 a)) obtains lower capacity values compared to the experimental results and in the case of Winter Air mode the opposite (Figure 10 b)).

Finally, in order to eliminate this mismatch between the IMST-ART model and the experimental results, the correlations for the condenser capacity have been readjusted. This adjustment consisted in fixing all the coefficients of the correlation with a new adjustment of the intercept (k_0 coefficient). Figure 11 (Winter Ground) and Figure 12 (Winter Air) shows the results for the final correlations without any adjustment of the coefficients in the compressor power input polynomials, and the readjustment only of the intercept for the condenser capacity polynomials.

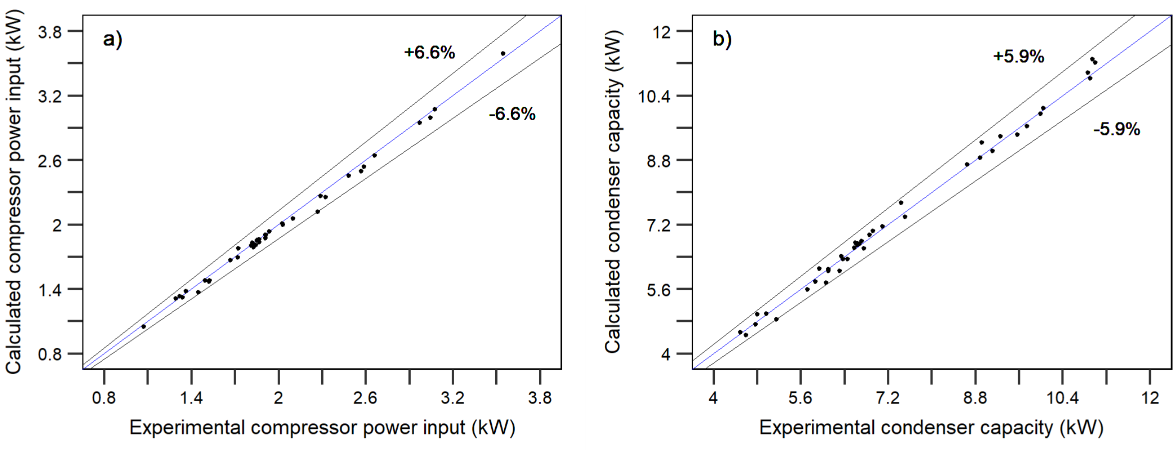

Figure 11. Final correlations for compressor power input (a) and condenser capacity (b) in Winter Ground mode

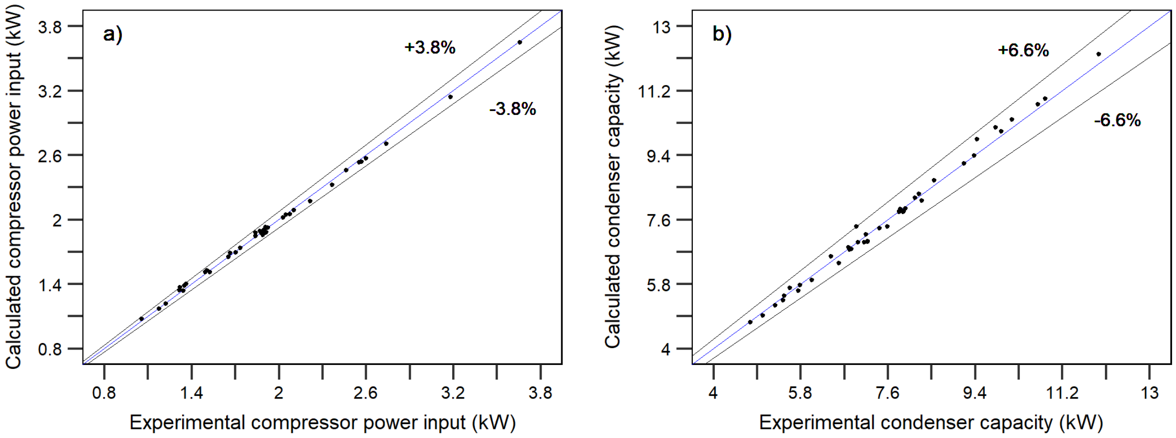

Figure 12. Final correlations for compressor power input (a) and condenser capacity (b) in Winter Air mode.

4. Conclusions

The fundamental purpose of this study was the development of simple and compact polynomial correlations, and after conducting it the main conclusions are:

- Following the methodology of stepwise regression and an iteratively elimination of terms with the lowest Person's partial correlation coefficient allows us to obtain compact polynomials with maximum deviations, compared to the IMST-ART model, lower than 3.5% for the 4 polynomials defined to characterize the compressor power input and the condenser capacity.

- Defining the compressor submodel in IMST-ART as compressor efficiency correlations has the disadvantages of to introduce a dependence with the pressure ratio in the simulating results of the parametric studies. This results in a final dependence with the pressure ratio in the final polynomials that it is necessary to correct in the Winter Ground mode.

- It has been found a greater difficulty to correlate the condenser capacity for the Winter Air Mode than for the Winter Ground Mode, due to the fact that for BPHEs, Tco and Teo represent better the condenser and evaporator temperatures of the refrigerant. By contrast, the independent variable for Winter Air mode is the dry bulb temperature at the inlet of the RTPFHx instead of the outlet temperature and with the additional difficulty of the dehumidification in this Hx.

- There is an evident dependence between the compressor consumption and the condenser capacity with the compressor frequency. Dividing by f_c these variables the main effect that this parameter has on the HP's performance can be reduced.

- By adjusting the intercept of the polynomials obtained with the experimental data and fixing the coefficients that had been calculated with the wide grid of the 3125 points simulated in each mode, it is observed that a good fitting is obtained for the two operating modes of the HP with maximum deviations lower than 6.6%.

- The final correlations calculates the performance (Wc, Qc) of the HP prototype analysed, as a function of the external variables (fc, Tco, dTc, Teo, dTe). However the methodology applied in this paper can be used to develop new polynomials for other units adjusting their coefficients to experimental data. The final result is a powerful compact model of the unit behaviour.

5. Acknowledgements

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