Influence of Suction Conditions and Refrigerant Fluid on Scroll Compressor Mass Flow Rate and Extrapolation Strategies

Javier MARCHANTE-AVELLANEDA[1]*, Rubén OSSORIO[1], Emilio NAVARRO-PERIS[1], Som S. SHRESTHA[2]

[1]Universitat Politècnica de València, Institute for Energy Engineering,
Valencia, Spain
Contact Information (jamarav@iie.upv.es)

[2]Buildings and Transportation Science Division, Oak Ridge National Laboratory,
Oak Ridge, TN 37831-6070, USA
Contact Information (shresthass@ornl.gov)

* Corresponding Author

ABSTRACT

This paper explores how suction conditions and refrigerant selection affect the mass flow rate of fixed-speed scroll compressors. The current compressor characterization standard only provides guidelines for correcting compressor performance with suction conditions, relying on the 1981 Dabiri correlation. This study seeks to analyze if the correction reported in the standard is adequate and also to supply an adequate correction when it is intended to extrapolate to another refrigerant. In order to conduct the following analysis, the calorimeter tests included in the AHRI-11, AHRI-21 and AHRI-33 reports have been used. These reports were published some years ago by the Air-Conditioning, Heating, and Refrigeration Institute (AHRI) within the Low-GWP Alternative Refrigerants Evaluation program. They include the analysis of two Copeland scroll compressors of different sizes (20 and 51 cm^3), tested under several suction conditions (SH = 11K, SH = 22K, T_i = 18ºC) and by using different refrigerant fluids (R134a, R32, R410A, R404A, …). Prior research has examined the response surfaces for the energy consumption and mass flow rate variables for these scroll compressors. This study intends to extend the analysis by first determining the optimal strategy to extrapolate the mass flow rate prediction — correlated at specific suction conditions — to other suction conditions. Furthermore, it aims to identify the most suitable approach for extrapolating when changing the refrigerant fluid.

1. INTRODUCTION

Nowadays, engineers and manufacturers increasingly rely on simulation and modeling tools to improve product quality. In the field of Heat Pumps (HP) and refrigeration equipment, the use of simulation tools helps manufacturers during the design stage. Some examples of these simulation tools are the ORNL Heat Pump Design Model (Fischer, S.K. Rice, 1983), the CYCLE_D-HX (Brown et al., 2021), the VapCyc (Richardson et al., 2002) and the IMST-ART (Thermal Area IUIIE, 2024) software. These tools enable researchers and manufacturers to analyze HP component substitutions more effectively, providing immediate feedback for new designs. In addition, by performing in-depth analysis through modeling and simulation, they can significantly reduce development costs by minimizing the need for costly experimental prototypes and expensive experimental campaigns.

A detailed description of each component is required when building a heat pump model with these types of tools. This usually involves obtaining performance data for each of its components from information published in the manufacturer's catalog to model them. In this sense, compressor modeling plays a key role in the design and optimization of these units because the performance and efficiency of this component largely determine the overall performance of these systems.

Compressor modeling has been a topic of special interest over the years, and many authors have reported several modeling approaches. These approaches can generally be categorized into theoretical, semi-empirical, and empirical models. Currently, the most prevalent approach is empirical modeling, specifically employing AHRI polynomials as outlined in the current characterization standard AHRI-540 (AHRI, 2020). The reason lies in a higher accuracy compared to theoretical or semi-empirical models if sufficient experimental information is available to empirically characterize the compressor (Cheung & Wang, 2018). Thus, AHRI polynomials are able to accurately predict the energy consumption and mass flow rate in fixed-speed compressors as a function of a third-degree polynomial depending on the evaporating and condensing temperatures.

However, a significant limitation of this map-based model approach is that the polynomial coefficients are calibrated based on experimental data at specific refrigerant and suction conditions, such as a particular level of suction superheat or fixed suction temperature. Although it has been shown that suction superheating has a negligible effect on energy consumption (Dabiri & Rice, 1981; Marchante-Avellaneda et al., 2023a), it has a greater influence on mass flow rate. Therefore, predicting the mass flow rate under different suction conditions would involve refitting the models with new experimental data or apply an adequate correction to the original model. The same applies in the case of extrapolating results using another refrigerant. Following the guidelines specified in the AHRI-540 standard, a simple correction can be applied based on the ratio of densities between the calibrated/mapped and new suction conditions and a single correction coefficient. This correction was originally published by Dabiri & Rice (1981), reporting a base value for the correction coefficient of 0.75 and based on the experimental results of Jacobs (1976). It is currently the most widespread way of applying an appropriate correction to the mass flow rate prediction, typically using the value reported by Dabiri for the correction coefficient (Shen et al., 2009). Unfortunately, no correction is provided to extrapolate with other refrigerants, where the most common approach is the use of dimensionless parameters such as volumetric efficiency.

Considering the large number of years since the original study was published, this work aims to analyze in detail whether this correction included in the standard is still adequate to extrapolate to other suction conditions. Furthermore, in an initial review of the original study published by Dabiri, we noticed some discrepancies between the original publication and the AHRI-540 standard. The main discrepancy lies in the fact that in the original study, the authors defined the correction based on a density ratio by considering the conditions at the internal suction port. To do this, they estimated the internal superheat that the mass flow rate suffers from the inlet shell pipe to the internal suction port. However, over the years, this internal superheat estimation has been omitted by using the same type of correction but directly considering the ratio of densities at inlet shell conditions.

Against, this background this work has conducted an in-depth analysis in order to evaluate the Dabiri correlation and other extrapolation strategies to accurately predict compressor mass flow rate from reference suction conditions to other suction conditions. Furthermore, the extrapolation from reference refrigerant fluids to other refrigerant is also evaluated in this study. The experimental data checked include three massive datasets from the test reports of the "Low-GWP Alternative Refrigerants Evaluation Program". These datasets include calorimeter data of two fixed-speed scroll compressor evaluated at two SH levels, a fixed suction temperature and different refrigerants. The experimental results from Winandy et al. (2002) and Jacobs (1976) are also analyzed in order to check the assumptions presented in Dabiri & Rice (1981).

2. METHODOLOGY

This work aims to evaluate how to accurately characterize the mass flow rate in fixed-speed scroll compressors, when extrapolations to other suction conditions or refrigerant are required. The data evaluated include three datasets extracted from the AHRI Low-GWP AREP project for two fixed-speed scroll compressors. The AHRI 21 report (Shrestha et al., 2013b) includes 866 calorimetric tests with a reference refrigerant (R404A) and four alternative refrigerants tested at three suction conditions (SH=11K, SH=22K and T_i=18ºC). AHRI 11 and AHRI 33 reports (Shrestha et al., 2013a, 2014) include a similar number of points for the same suction conditions and five refrigerants, with R410A as the reference refrigerant. The tests have been performed using the same original lubricant charged in both compressors. The datasets reported by Winandy et al. (2002) and Jacobs (1976) will also be analyzed to evaluate the assumptions considered in Dabiri's original work. Tables 1 and 2 summarize the compressors and the composition of the new refrigerants.

The study will be divided into three parts. First, we'll assess the original Dabiri correlation using data from AHRI 21, 11, and 33 reports, considering the ratio of densities at the inlet shell and port conditions. The correction coefficient will be obtained for each refrigerant, including all suction conditions for the regression adjustment. The objective will be to evaluate if the original value of 0.75 is still suitable. Subsequently, the three modeling approaches included in section 4 will be evaluated. First, the models will be adjusted using the data at SH=11K as a reference for each refrigerant and evaluating the extrapolation capabilities to other suction conditions. Finally, the same models will be reevaluated but only adjusted with the base refrigerant and the same superheat level, checking the extrapolation capabilities to other suction conditions and refrigerant.

R software (R Core Team, 2023) was employed for statistical analysis and Refprop database (Lemmon et al., 2018) for refrigerant properties. Error metrics utilized to evaluate model predictive power included Maximum Relative Error (MRE), the Root Mean Square Error (RMSE), and the Coefficient of Variation of Root Mean Square Error (CV_{RMSE}).

Table 1: Main compressor characteristics and tested refrigerants

Source	Type	Model (Manufacturer)	Disp(freq.) cm3(Hz)	Refrigerants tested	Test points	Conditions by refrigerant test
AHRI 21	Scroll	ZS21KAE-PFV (Copeland)	50.96 (60)	R404A/ARM31a/D2Y65/L40 /(R32+R134a)	191/186/183 /173/133	SH=11.11K
AHRI 11	Scroll	ZP21K5E-PFV (Copeland)	20.32 (60)	R410A/R32/DR5/L41a	196/166/189 /186	SH=22.22K T_i=18.33ºC
AHRI 33	Scroll	ZP21K5E-PFV (Copeland)	20.32 (60)	R410A/(R32+R134a)	196/168	
Winandy 2002	Scroll	-	174 (50)	R22	28	T_i=25ºC
Jacobs 1976	Reciproc.	-	-	R22	2	T_i=18.33ºC

Table 2: New refrigerant's composition (Mass%)

Source	Name	ASHRAE name	Composition
AHRI 21	ARM31a	-	R32/R134a/R1234yf (28/21/51)
	D2Y65	R454A	R32/R1234yf (35/65)
	L40	-	R32/R152a/R1234yf/R1234ze(E) (40/10/20/30)
	R-32/R134a	-	R32/R134a (50/50)
AHRI 11	DR5	≈R454B	R32/R1234yf (72.5/27.5)
	L41a	≈R459A	R32/R1234yf/R1234ze(E) (73/15/12)
AHRI 33	R32/R134a	-	R32/R134a (94/6)

3. ANALYSIS OF THE EFFECT OF SH AND REFRIGERANT ON MASS FLOW RATE

It is well known that the suction conditions, specifically the refrigerant density at the compressor suction, significantly affect the compressor mass flow rate. Considering a given operating point, a specific swept volume, motor speed and refrigerant, the mass flow rate is mainly fixed by the evaporating temperature and the superheat — fixing the suction density — and is also influenced by the volumetric efficiency depending on the pressure ratio (1).

$$\dot{m} = \rho V_s N \eta_v \tag{1}$$

As mentioned above, the most widespread way to accurately model compressors is using empirical map-based models, specifically the so-called AHRI polynomials. Recent research has evaluated this type of polynomials, introducing alternative polynomial expressions with several advantages over the conventional polynomials reported in the standard (Marchante-Avellaneda et al., 2023a, 2023b). Unfortunately, this type of empirical approach has the disadvantage of fitting to specific suction conditions. This does not imply a problem for the energy prediction, generally observed to remain constant with SH change, but predicting mass flow rate under varying suction conditions requires correcting it with Dabiri's correlation (2), also provided in the standard characterization.

$$\dot{m}_{new} = \dot{m}_{map} \cdot \left[1 + F \cdot \left(\frac{\rho_{new}}{\rho_{map}} - 1\right)\right] \tag{2}$$

The original value reported by the author for the F factor is 0.75, a value based on Jacobs' experimental results (Jacobs, 1976) in order to correct the variation of volumetric efficiency evaluated at "internal port conditions" with the internal SH. So, equation 2 was originally obtained considering the density ratio at internal port conditions instead of conventional inlet shell conditions, but the latter is the most widespread form in use today.

The main difference lies in the estimation of the internal superheat. It is well known that the refrigerant entering the inlet compressor shell at a certain SH level undergoes additional superheat before reaching the internal suction port.

The reason is that the compressor motor windings are exposed to contact with the refrigerant gas to cool down the electric motor. Furthermore, internal friction coupled with suction and discharge manifold heat transfer also promotes this internal superheat, decreasing refrigerant density compared to inlet shell conditions. As is reported in Hiller & Glicksman (1976), this internal SH can result in a temperature increase from 10ºC (large non/semi-hermetic compressors) to 30ºC (small hermetic compressors), which can be higher if the compressor design is intended to protect against liquid slugging.

As originally reported in Dabiri & Rice (1981), let's assume in a simple way that the refrigerant pressure remains the same from the inlet shell to the internal port conditions. Moreover, we can consider that the specific heat of the refrigerant at constant pressure also remains constant between inlet shell and internal port conditions ($c_p \approx c_{p,i}$). In this way, equation (3) determines the heat transfer to the suction:

$$\dot{Q}_{i,p} = \dot{m} c_p \left(T_p - T_i\right) = \dot{m} \Delta \mathrm{h}_{i,p} \tag{3}$$

So, if an estimate of $\Delta \mathrm{h}_{i,p}$ is available, the temperature at the internal port could be calculated with equation (4).

$$T_p = T_i + \frac{\Delta h_{i,p}}{c_p} = T_i + SH_{i,p} \tag{4}$$

From the experimental results of Jacobs, Dabiri & Rice (1981) considered as a simplifying hypothesis that $\Delta h_{i,p}$ can be considered as constant with a value of 21 kJ/kg. Although this value of $\Delta \mathrm{h}_{i,p}$ has been proven to be correct considering the data reported by Jacobs, this assumption may need to be revised, considering that it is only based on the measurement of 2 experimental points. Therefore, this assumption will be reevaluated in this work by analyzing the data reported in Winandy et al. (2002), where a total of 28 experimental points with internal port measurements of both temperature and pressure are included.

On the other hand, considering that currently this correction is applied directly estimating the ratio of density at inlet shell conditions, both approaches will be evaluated in this work. In order to obtain suitable expressions to introduce this analysis, considering the real gas equation and equation (4), it is possible to estimate the mass flow rate according to inlet shell conditions (5) or inlet port conditions (6) as follows:

$$\dot{m} = \frac{V_s N}{\frac{R_u}{M} Z_i} \cdot \frac{1}{T_i} \cdot P_e \cdot \eta_{v,i} \tag{5}$$

$$\dot{m} = \frac{V_s N}{\frac{R_u}{M} Z_p} \cdot \frac{1}{T_p} \cdot P_e \cdot \eta_{v,p} \tag{6}$$

Thus, two possible volumetric efficiencies can be defined, $\eta_{v,i}$ evaluated at inlet shell conditions and $\eta_{v,p}$ evaluated at internal port conditions. This gives us a suitable formulation that will also facilitate the analysis when also evaluating extrapolation to other refrigerants, as it includes their properties: molar mass (M), compressibility factor (Z_p and Z_i) and specific heat at constant pressure (c_p).

The next subsection will evaluate Winandy's experimental results to determine how $\Delta \mathrm{h}_{i,p}$ evolves at different operating points. The evolution of $\eta_{v,i}$, and $\eta_{v,p}$ will also be evaluated to understand how the internal superheat affects the volumetric efficiency and to determine if there is any advantage in approximating the density ratio at internal port conditions.

3.1 Internal superheat and volumetric efficiencies at inlet shell and internal port conditions

In order to evaluate the assumptions considered by Dabiri, the dataset reported in Winandy et al. (2002) is analyzed in this section. This dataset includes 27 measurements at constant suction temperature of 25ºC and an extra test at 18ºC in a hermetic scroll compressor equipped with internal sensors. These data allow us to determine the refrigerant conditions at both the internal suction port and the inlet shell, so $\Delta h_{i,p}$ can be obtained. In this dataset, $\Delta h_{i,p}$ ranges from 6.7 to 21.8 kJ/kg, with an average of 13.5 kJ/kg, based on inlet shell pressure ($\Delta P_{i,p}$close to 0).

A more detailed analysis shows a dependence of $\Delta h_{i,p}$ on the pressure ratio (in the original study, this fact was already reported for $\Delta T_{i,p}$). An even greater dependence on the enthalpy difference between discharge and suction ($\Delta h_{i,2}$) has also been observed. This makes sense since higher $\Delta h_{i,2}$ implies longer compressions and higher discharge temperature, leading to higher internal superheat. Furthermore, considering an enthalpy difference instead of a pressure ratio may be more appropriate to extrapolate to other refrigerants. The latter dependence is reflected in Figure 1, where it has been found that a good estimate of $\Delta h_{i,p}$ can be obtained as a constant percentage of $\Delta h_{i,2}$.

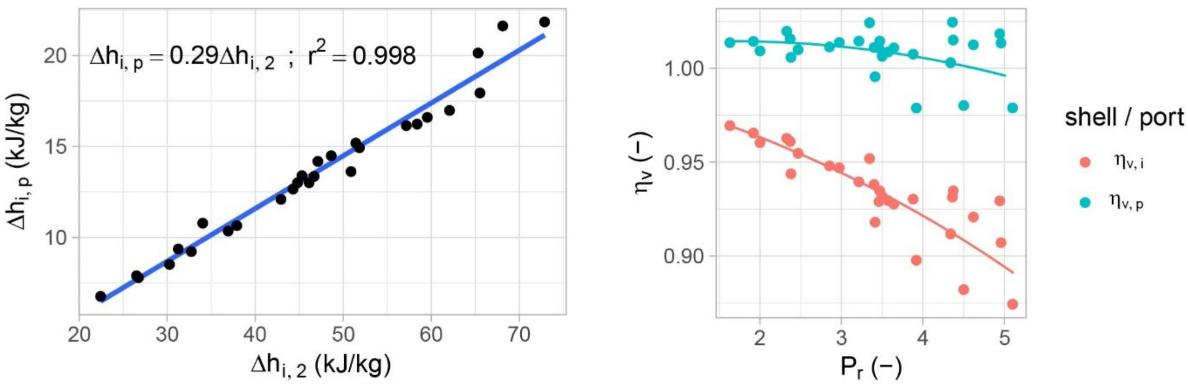

Figure 1: $\Delta h_{i,p}$ as a function of $\Delta h_{1,2}$ (left-hand) and η_v at shell inlet and internal port conditions (right-hand)

This holds true even when using $\Delta h_{i,2s}$ instead of $\Delta h_{i,2}$, although with slightly more spread in the data. In this case the regression adjustment obtains a higher value for the adjustment factor (0.4) and a slightly lower r^2 (0.994).

The Winandy dataset also reports an additional point with 18.3ºC of suction temperature and evaporating and condensation temperatures of 7.4 and 54.5 ºC, the same conditions and refrigerant as the point evaluated by Jacobs, but with a $\Delta h_{i,p}$ value of 12.6 kJ/kg. Therefore, we can assume that this value will depend on the size and design of the compressor being characterized and probably the refrigerant used, with higher values expected for those refrigerants with higher discharge temperatures. So, it is not appropriate to consider a constant value of $\Delta h_{i,p}$, being more appropriate to estimate it as a given percentage of $\Delta h_{i,2s}$.

Finally, the volumetric efficiency at the inlet shell and inlet port conditions has been plotted in Figure 1-right.

On the one hand, the volumetric efficiency calculated at inlet shell conditions shows a practically linear trend with the pressure ratio (P_r). This dependence is well known and is a common approach characterizing mass flow rate through volumetric efficiency and P_r. On the other hand, at the inlet port, efficiency still changes with pressure ratio, but less drastically. It's almost constant at moderate pressure ratios and drops slightly at higher ones. The efficiency values are slightly higher than 1. However, this may be due to the value considered for the displacement, being more difficult to determine the internal volume in scroll compressors.

Considering the above, correlating the volumetric efficiency at inlet shell conditions ($\eta_{v,i}$) with pressure ratio includes the effect of internal superheat, making the relationship nearly linear and more dependent on pressure ratio. Efficiency at inlet port shows less change with pressure ratio, being more appropriate to consider a constant value or a quadratic dependence to model a constant $\eta_{v,p}$ at moderate pressure ratios and with a slight decrease at higher pressure ratio. These assumptions should be considered only for scroll compressors and may not be true for other technologies.

4. EXTRAPOLATION STRATEGIES EVALUATED

Based on the findings from the previous section, three cases have been formulated to assess the extrapolation capabilities of mass flow rate to different suction conditions and refrigerants, using as database the experimental tests of AHRI 21, 11 and 33 reports.

First of all, a mean value for the F factor from Dabiri's correlation (8) will be fixed. This involves calculating the mass flow rate ratio, density ratio and make a regression adjustment of coefficient F by using all suction conditions for each refrigerant. The ratios will be calculated considering 11K of superheat as map conditions. The aim is to compare the mean value obtained with the original value reported in Dabiri & Rice (1981). The ratio of densities will be considered at the inlet shell and internal port conditions. A constant $\Delta h_{i,p}$ of 21kJ/kg, will be assumed for inlet port conditions like Dabiri & Rice (1981). After analyzing an appropriate value for F, three proposed approaches will be analyzed.

Case A includes a model homologous to the AHRI polynomial (7) and reported in Marchante-Avellaneda et al. (2023a). Equation (7) will be adjusted for each refrigerant at SH=11K — "map" conditions — and equation (8)

provides the correction to other suction conditions. The extrapolation to other refrigerants will also be evaluated fitting equation (7) and (8) to the base refrigerant at SH=11K data. In this case, the correction is first applied to other suction conditions using equation (8), considering the base refrigerant. Subsequently, an additional density ratio between the new refrigerant and the base refrigerant is applied as an additional correction factor.
Case B, equations (9) and (10), include the traditional approach considering volumetric efficiency as a polynomial depending on pressure ratio. The properties of the refrigerant are evaluated at inlet shell conditions.
Case C, equations (11) and (12) are employed, similar to Case B but evaluating refrigerant conditions at the internal port. A two-step adjustment methodology is proposed based on the observed trends from the previous section: first, iteratively adjust the k-factor (equation 11) while considering a constant volumetric efficiency at inlet port conditions. Then, with the estimated k-factor, we can calculate port conditions and adjust equation (12) accordingly. Equation (11) with real discharge conditions will also be considered to quantify the precision loss when estimating $\Delta h_{i,p}$ under isentropic discharge conditions.
Like Case A, extrapolation capabilities to other suction conditions and refrigerants will be evaluated in case B and C.

Case A:
$$\dot{m}_{map} = k_0 + k_1 P_e + k_2 P_c + k_3 P_e P_c \tag{7}$$
$$\dot{m}_{new} = \dot{m}_{map} \cdot \left[1 + F \cdot \left(\frac{\rho_{new}}{\rho_{map}} - 1\right)\right] \tag{8}$$

Case B:
$$\dot{m} = \frac{V_s N}{\frac{R_u}{M} Z_i} \cdot \frac{1}{T_i} \cdot P_e \cdot \eta_{v,i} \tag{9}$$
$$\eta_{v,i} = k_0 + k_1 P_r + k_2 P_r^2 \tag{10}$$

Case C:
$$\dot{m} = \frac{V_s N}{\frac{R_u}{M} Z_p} \cdot \frac{1}{T_i + k \cdot \frac{(h_{2s} - h_i)}{c_{p,i}}} \cdot P_e \cdot \eta_{v,p} \tag{11}$$
$$\eta_{v,p} = k_0 + k_1 P_r + k_2 P_r^2 \tag{12}$$

5. ANALYSIS OF RESULTS

The results of the three modeling approaches introduced in the previous section are reported below. Due to the extension limit of this paper, only the prediction errors obtained will be reported in two summary tables, including the adjustment values for the correction factors F and k from equations (8) and (11) (Tables 3 and 4). A series of correlation graphs will also be reported with the extrapolation capabilities of the models adjusted to the base refrigerant and 11K of superheat to the rest of the suction conditions and refrigerants.
First of all, we will review the results to determine an appropriate mean value for the F factor of the Dabiri's correlation. These results have been included in Table 3. As mentioned in the previous section, the density ratio was considered at inlet shell conditions by adjusting the F coefficient and considering all available suction conditions for each refrigerant and 11K of superheat as map conditions. F-fitting values have also been obtained considering the density ratio at the internal port, estimating the internal superheat with a constant $\Delta h_{i,p}$ value of 21kJ/kg, as described in the original work.
The first thing we can notice is that we obtain practically the same prediction errors independently of where the density ratio is estimated (at inlet shell or inlet port conditions). These errors range between 0.4-1.3% of CV_{RMSE} (1-4% of MRE). The maximum error is observed for the L41a refrigerant in the AHRI 11 report. Reviewing the values fitted for the F coefficient, we can see that they are higher when selecting the ratio of densities at internal port conditions and lower when using inlet shell conditions. It can be noticed that these values are not constant even considering the same report, oscillating F from its average value between 0.2 to 17% for the evaluated refrigerants. The average value of F at inlet shell conditions is 0.736 and 0.833 for internal port conditions. Therefore, we can conclude that the original value of 0.75 is adequate, considering the density ratio at shell inlet instead of the internal port conditions. On the other hand, the internal superheat estimation considering a constant $\Delta h_{i,p}$ does not seem to improve results, resulting only in a higher average value for F of 0.833. Based on these results, the original value of 0.75 will be selected as the F coefficient for case A, and the ratio of densities will be estimated at inlet shell conditions for its greater simplicity.

Table 3: Adjustment of coefficient F using all suction conditions by refrigerant. Density ratio calculated at inlet shell and inlet port conditions. SH=11K selected as map conditions

ρ at inlet shell			ρ at inlet port: $\Delta h_{i,p} = 21$kJ/kg				
F	CV_{RMSE} (%) (RMSE (kg/h))	MRE (%)	F	CV_{RMSE} (%) (RMSE (kg/h))	MRE (%)	Fluid	Report
0.863	0.431 (0.489)	1.349	0.975	0.435 (0.494)	1.351	R410A	AHRI 11
0.789	1.015 (0.825)	3.259	0.862	1.010 (0.821)	3.227	R32	
0.657	1.100 (0.974)	3.974	0.735	1.099 (0.972)	3.977	DR5	
0.429	1.339 (1.077)	3.475	0.480	1.338 (1.075)	3.468	L41a	
0.862	0.286 (0.554)	0.931	1.013	0.295 (0.570)	1.009	R404A	AHRI 21
0.734	0.222 (0.277)	0.856	0.850	0.222 (0.277)	0.858	ARM31a	
0.750	0.360 (0.495)	1.640	0.865	0.359 (0.493)	1.634	D2Y65	
0.755	0.511 (0.569)	1.662	0.863	0.509 (0.567)	1.638	L40	
0.759	0.502 (0.593)	1.846	0.861	0.496 (0.586)	1.826	R32+R134a	
0.772	0.485 (0.566)	1.477	0.872	0.487 (0.568)	1.491	R410A	AHRI 33
0.722	0.365 (0.300)	1.621	0.791	0.361 (0.296)	1.617	R32+R134a	

After establishing a suitable mean value for the F coefficient at 0.75, we evaluated approaches A, B, and C. The extrapolation capability was first considered to extrapolate to other suction conditions. Table 4 presents the prediction errors for cases A, B, and C, with models fitted at 11K superheat for each refrigerant. In case C, we have additionally reported the results considering the real discharge conditions in equation (11). Although the real discharge conditions cannot be considered as a variable of the model, this case is reported to compare them with the prediction results of case C considering the isentropic discharge conditions.

Table 4: Summary of errors in cases A, B, and C: SH=11K used for adjustment and extrapolation across all suction conditions for each refrigerant.

	Case A	Case B	Case C ($\Delta h_{i,2s}$)		Case C ($\Delta h_{i,2}$)			
F	CV_{RMSE} (%) MRE (%)	CV_{RMSE} (%) MRE (%)	k_{2s}	CV_{RMSE} (%) MRE (%)	k_2	CV_{RMSE} (%) MRE (%)	Fluid	Report
0.75	0.86 (1.99)	1.02 (2.25)	1.19	0.83 (2.13)	0.74	0.41 (0.99)	R410A	AHRI 11
0.75	1.06 (2.80)	1.44 (2.86)	1.01	1.37 (3.80)	0.73	0.69 (1.67)	R32	
0.75	1.11 (4.30)	1.94 (6.13)	1.05	1.65 (6.64)	0.71	1.03 (4.49)	DR5	
0.75	1.53 (3.93)	2.77 (5.92)	1.03	2.25 (5.58)	0.71	1.41 (3.01)	L41a	
0.75	0.77 (1.94)	0.90 (2.54)	0.71	0.61 (1.78)	0.36	0.45 (1.20)	R404A	AHRI 21
0.75	0.26 (1.04)	1.67 (3.28)	0.63	1.14 (2.68)	0.36	0.65 (1.22)	ARM31a	
0.75	0.29 (1.27)	1.57 (3.10)	0.56	1.17 (2.53)	0.31	0.81 (1.62)	D2Y65	
0.75	0.51 (1.48)	1.52 (2.83)	0.55	1.01 (2.34)	0.35	0.56 (1.37)	L40	
0.75	0.61 (1.78)	1.36 (3.29)	0.72	1.01 (2.78)	0.43	0.60 (2.01)	R32+R134a	
0.75	0.68 (1.58)	1.15 (2.72)	1.09	0.90 (3.13)	0.70	0.33 (1.06)	R410A	AHRI 33
0.75	0.57 (1.61)	1.35 (3.09)	0.89	1.27 (3.41)	0.67	0.72 (2.08)	R32+R134a	

Comparing cases A, B, and C first, considering the real discharge conditions in C, we can observe that generally lower prediction errors are obtained in the case of C. Something interesting to emphasize is the values obtained for the k coefficient. These values were obtained using the 2-step fit described in the previous section, using only the data at superheat 11 for each refrigerant. We can observe that the values of k are fairly constant depending on the selected compressor, with a value of 0.7 for the 20 cm^3 compressor of the AHRI 11 and 33 reports and 0.35 for the 50 cm^3 compressor of the AHRI 21 report. Despite being unable to contrast the internal superheat estimate in the absence of

experimental measurements at the internal port, the low prediction errors seem to indicate a good estimate of the internal superheat. A higher value of k for the smaller compressor is also understandable, as it has more heat transmitted to the suction port. It has to be highlighted that, in the case of not knowing the actual discharge conditions (which is typically the case), a model based on isentropic discharge conditions has to be used instead [Case C ($\Delta h_{i,2s}$)]. When using isentropic conditions instead of the actual ones a higher value of k tends to be fitted — as $\Delta h_{i,2s}$ is always lower than $\Delta h_{i,2}$ — and higher prediction errors occur which are closer to the ones obtained by Cases A and B. So, no improvement is achieved by considering an estimate of internal superheat based on $\Delta h_{i,2s}$.
Finally, Figure 2 shows the results of cases A, B, and C considering for the adjustment the reference refrigerant and a superheat of 11K and evaluating the extrapolation to all suction conditions and refrigerants. The L41a refrigerant has been excluded due to consistently higher prediction errors in all cases evaluated. Upon close examination of the data, suspicions arise about possible problems in these tests. Both Cases A and B show higher prediction errors when extrapolating refrigerant. On the other hand, both Cases C tend to perform better, showing average errors close to 1 kg/h (approximately one-third of the obtained errors in Cases A and B). Regarding Case C using isentropic discharge conditions, it was proven to have solid predictions when extrapolating to different refrigerants resulting in promising new usable model for making predictions at different refrigerants.

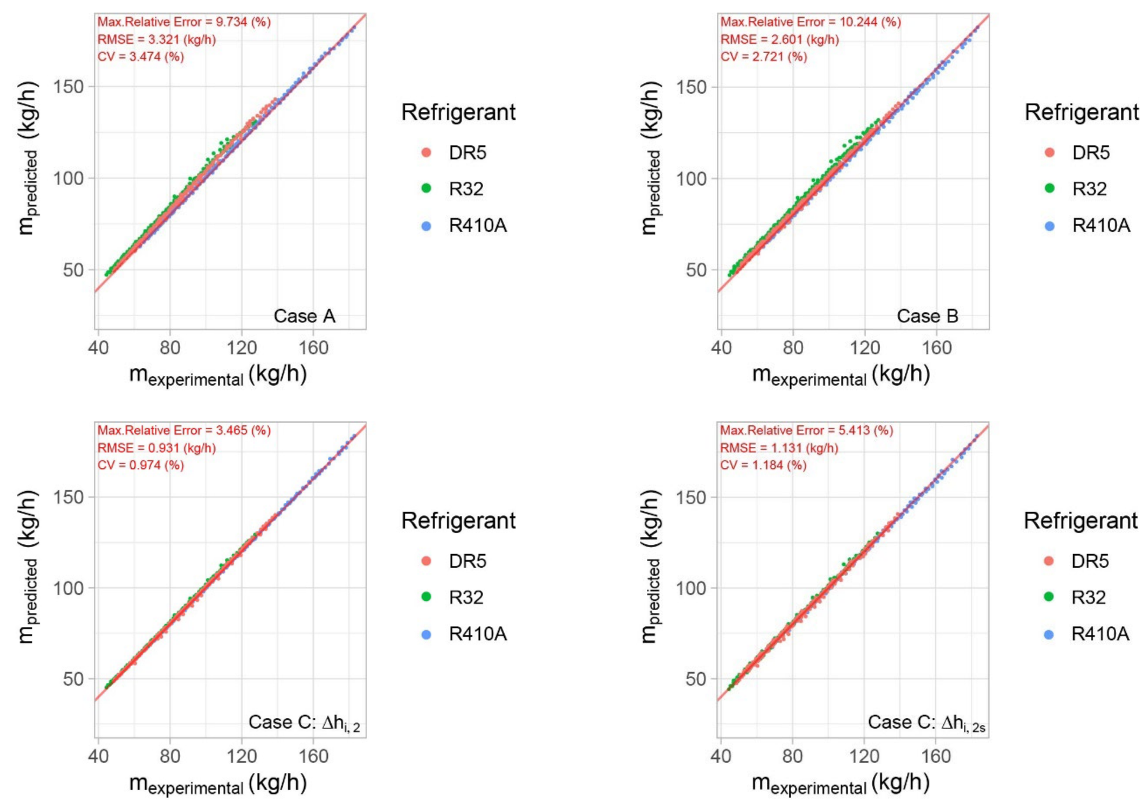

Figure 2: AHRI 11: Adjustment to reference refrigerant (R410A) and SH=11K. Extrapolation results to other refrigerants and suction conditions in case A, B and C.

6. CONCLUSIONS

An exhaustive analysis has been carried out in order to establish how to accurately extrapolate predictions for the mass flow rate to other suction conditions and refrigerants in scroll compressors. The following main conclusions can be drawn from the study:

- Map-based modeling methodologies accurately predict energy consumption and mass flow rates in the compressors field. This approach has the disadvantage of fitting the models empirically to specific suction conditions. However, energy consumption is usually observed independently of the level of superheat fixed. In the case of mass flow rate, a correction is required to extrapolate to other suction conditions.

- The most widespread approach to extrapolate models based on mass flow rate maps to other suction conditions is by using the Dabiri correlation. This correction consists of a linear polynomial depending on the ratio of densities between the map conditions, where the model is fitted, and the new suction conditions, including an adjustment factor.
- The adjustment factor originally reported is 0.75 based on Jacobs' experimental results and considering a constant $\Delta h_{i,p}$ of 21kJ/kg in order to estimate internal port conditions. However, the AHRI-540 standards report the same correction but considering the density ratio at inlet shell conditions.
- Winandy's experimental data for a hermetic scroll compressor, which includes internal temperature and pressure port measurements, have been analyzed. From this analysis, it is concluded that it is not appropriate to consider a constant value of $\Delta h_{i,p}$, being a value dependent on the size, refrigerant, and compressor design. Moreover, it has been found that the volumetric efficiency estimated at internal suction port remains almost constant at moderate pressure ratios, decreasing slightly at higher values of pressure ratio.
- Three datasets with many experimental tests on two fixed-speed scroll compressors have been analyzed. The data include three suction conditions and several refrigerants. It has been possible to establish that the original value of the F-factor from the Dabiri correlation is adequate, considering the ratio of densities at inlet shell conditions. Assuming a constant $\Delta h_{i,p}$ does not seem to increase the accuracy of the extrapolations.
- Three possible approaches to model the mass flow rate at given suction and refrigerant conditions have been checked. Considering extrapolation to other suction conditions and the same refrigerant, approach A using the Dabiri correction obtains the best results. Approaches B and C based on inlet shell and inlet port efficiencies are also adequate, with a slight increase in the prediction error.
- On the other hand, considering the extrapolation to other refrigerants, approaches A and B obtain higher prediction errors. However, approach C obtains the best results based on the estimation of internal superheat and volumetric efficiency at inlet port conditions. This leads us to consider that the estimation of the internal superheat of approach C is adequate. We can also state that for the scroll compressors analyzed, the volumetric efficiency estimated at inlet port conditions remains constant between refrigerants.

NOMENCLATURE

c_p	specific heat at constant pressure (J/kgK)	SH	superheat (K)
h	enthalpy (J/kg)	T	Temperature (K)
$\dot{m}$	mass flow rate (kg/s)	V_s	swept volume (m^3/rev)
M	refrigerant molar mass (kg/mol)	Z	compressibility factor (-)
N	compressor speed (rps)	η_v	volumetric efficiency (-)
P	pressure (Pa)	ρ	density (kg/m^3)
P_r	pressure ratio (-)	MRE	Maximum Relative Error (%)
$\dot{Q}$	heat flow rate (W)	RMSE	Root Mean Square Error (kg/h)
R_u	ideal gas constant 8.314 (J/molK)	CV_{RMSE}	Coefficient of Variance of RMSE (%)
Subscript			
c	condenser	new	new suction conditions
e	evaporator	p	internal suction port conditions
i	inlet shell conditions	2	discharge conditions
map	reference suction conditions	2s	discharge conditions (isentropic compression)

REFERENCES

AHRI. (2020). *AHRI-540. Performance Rating of Positive Displacement Refrigerant Compressors*.

Brown, J., Brignoli, R., Domanski, P., & Yoon, Y. (2021). *CYCLE_D-HX: NIST vapor compression cycle model accounting for refrigerant thermodynamic and transport properties; version 2, user's guide*. Technical Note (NIST TN), National Institute of Standards and Technology, Gaithersburg, MD. doi:10.6028/NIST.TN.2134

Cheung, H., & Wang, S. (2018). A comparison of the effect of empirical and physical modeling approaches to extrapolation capability of compressor models by uncertainty analysis: A case study with common semi-empirical compressor mass flow rate models. *International Journal of Refrigeration*, *86*, 331–343. doi:10.1016/j.ijrefrig.2017.11.020

Dabiri, A. E., & Rice, C. K. (1981). *Compressor-simulation model with corrections for the level of suction gas superheat* (CONF-810657-2). Oak Ridge National Lab., TN (USA); Science Applications, Inc., La Jolla, CA (USA). https://www.osti.gov/biblio/6345140

Fischer, S.K. Rice, C. K. (1983). *ORNL heat pump model: A steady-state computer design model of air-to-air heat pumps* [Manual]. Oak Ridge National Laboratory (ORNL). doi:10.2172/814817

Hiller, C. C., & Glicksman, L. R. (1976). *Improving heat pump performance via compressor capacity control: Analysis and test* [Technical Report]. MIT Energy Lab. https://dspace.mit.edu/handle/1721.1/27257

Jacobs, J. (1976). Analytic and Experimental Techniques for Evaluating Compressor Performance Losses. *International Compressor Engineering Conference*. https://docs.lib.purdue.edu/icec/179

Lemmon, E. W., Bell, I. H., Huber, M. L., & McLinden, M. O. (2018). *NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties-REFPROP, Version 10.0, National Institute of Standards and Technology* [Computer software]. https://www.nist.gov/srd/refprop

Marchante-Avellaneda, J., Corberan, J. M., Navarro-Peris, E., & Shrestha, S. S. (2023a). A critical analysis of the AHRI polynomials for scroll compressor characterization. *Applied Thermal Engineering*, *219*, 119432. doi:10.1016/j.applthermaleng.2022.119432

Marchante-Avellaneda, J., Navarro-Peris, E., Corberan, J. M., & Shrestha, S. S. (2023b). Analysis of map-based models for reciprocating compressors and optimum selection of rating points. *International Journal of Refrigeration*, *153*, 168–183. doi:10.1016/j.ijrefrig.2023.06.002

R Core Team. (2023). *R: A Language and Environment for Statistical Computing* (4.3.0) [Computer software]. R Foundation for Statistical Computing. https://www.R-project.org/

Richardson, D. H., Jiang, H., Lindsay, D., & Radermacher, R. (2002). Optimization of vapor compression systems via simulation. *International Refrigeration and Air Conditioning Conference. Paper 529*. http://docs.lib.purdue.edu/iracc/529

Shen, B., Braun, J. E., & Groll, E. A. (2009). Improved methodologies for simulating unitary air conditioners at off-design conditions. *International Journal of Refrigeration*, *32*(7), 1837–1849. doi:10.1016/j.ijrefrig.2009.06.009

Shrestha, S., Mahderekal, I., Sharma, V., & Abdelaziz, O. (2013a). *TEST REPORT #11. Compressor calorimeter test of R-410A alternatives R-32, DR-5, and L-41a* (11; Low-GWP Alternative Refrigerants Evaluation Program (Low-GWP AREP)). Oak Ridge National Laboratory. https://www.ahrinet.org/App_Content/ahri/files/RESEARCH/AREP_Final_Reports/AHRI%20Low-GWP%20AREP-Rpt-011.pdf

Shrestha, S., Sharma, V., & Abdelaziz, O. (2013b). *TEST REPORT #21. Compressor calorimeter test of R-404A alternatives ARM-31a, D2Y-65, L-40, and R-32/R-134a (50/50)* (21; Low-GWP Alternative Refrigerants Evaluation Program (Low-GWP AREP)). Oak Ridge National Laboratory. https://www.ahrinet.org/App_Content/ahri/files/RESEARCH/AREP_Final_Reports/AHRI%20Low-GWP%20AREP-Rpt-021.pdf

Shrestha, S., Sharma, V., & Abdelaziz, O. (2014). *TEST REPORT #33. Compressor calorimeter test of R-410A alternative: R-32/R-134a mixture using a scroll compressor* (33; Low-GWP Alternative Refrigerants Evaluation Program (Low-GWP AREP)). Oak Ridge National Laboratory. https://www.ahrinet.org/App_Content/ahri/files/RESEARCH/AREP_Final_Reports/AHRI-Low-GWP-AREP-Rpt-033.pdf

Thermal Area IUIIE. (2024). *IMST-ART: A simulation tool to assist the selection, design and optimization of refrigeration equipments and components* [Manual]. IMST-GROUP. http://www.imst-art.com/

Winandy, E., O, C. S., & Lebrun, J. (2002). Experimental analysis and simplified modelling of a hermetic scroll refrigeration compressor. *Applied Thermal Engineering*, *22*(2), 107–120. doi:10.1016/S1359-4311(01)00083-7

ACKNOWLEDGEMENT

The present work has been supported by the project “Descarbonización de edificios e industrias con sistemas híbridos de bomba de calor”, funded by the “Ministerio de Ciencia e Innovación”, MCIN, Spain, with code number: PID2020-115665RB-I00 and by the postdoctoral fellowship funded by the “Universitat Politècnica de València” inside the program “Ayudas para Contratos de acceso de personal investigador doctor (PAID-10-23)”. This research used resources at the Building Technologies Research and Integration Center, a DOE Office of Science User Facility operated by the Oak Ridge National Laboratory of USA.Many thanks as well to the late Dr. José Miguel Corberán, without whom this work would never have been possible. Sadly, Dr. José Miguel Corberán passed away in July of 2022. I wish to give my wholehearted support to José Miguel’s family. I hope we did you proud.