Performance Measurements and Mapping of a R-407C Vapor Injection Scroll Compressor

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Abstract. Environmental conditions significantly define the performance of HVAC&R systems. Vapor compression systems in hot climates tend to operate at higher pressure ratios, leading to increased discharge temperatures. Higher discharge temperatures can lead to higher irreversibilities in the compression process, lower specific enthalpies differences across the evaporator, and possibly a reduction in the compressor life due to the breakdown of the oil used for lubrication. To counter these effects, the use of economized, vapor injection compressors is proposed for vapor compression systems in high temperature climates. Such compressors are commercially available for refrigeration applications, in particular, supermarket refrigeration systems. However, compressor maps for vapor injection compressors are limited and none exist for R-407C. Through calorimeter testing, a compressor map for a single-port vapor injection compressor using R-407C was developed. A standard correlation for mapping single-port vapor injection compressors is proposed and validated using the compressor test results. The system and compressor performance with and without vapor injection was considered. As expected, with vapor injection there was a reduction in compressor discharge temperatures and an increase in the system coefficient of performance. The proposed dimensionless correlation is more accurate than the AHRI polynomial for mapping the injection ratio, discharge temperature, and compressor heat loss. The predicted volumetric efficiency values from the dimensionless correlation is within 1% of the measured values. Similarly, the predicted isentropic efficiency values are within 2% of the measured values.

1. Introduction
It is well documented that heat sink and source temperatures strongly influence vapor compression system performance. HVAC&R systems operating in extreme temperature climates are required to operate at higher pressure ratios. The system refrigeration capacity and efficiency decline with higher condensing temperatures and lower evaporating temperatures. Lower system performance is attributed to an increase in irreversibilities during compression, and higher compressor discharge temperatures. Economized, refrigerant vapor-injection (VI), vapor compression cycles have been considered promising solutions for reducing the compressor discharge temperature for systems with large compression ratios (Winandy et al., 2002; Wang et al., 2009). This is especially important because higher discharge temperatures can result in additional wear on the compressor, reducing the
compressor reliability and life. Vapor-injection during the compression process has also been proven to improve the refrigeration capacity and coefficient of performance (COP) (Wang et al., 2009).

Typically, the AHRI polynomial equations, Equation (1), capture the compressor behavior as a function of test pressure ratios specified by the manufacturer. These equations are then used to predict the compressor power input and suction mass flow rate at the operator’s desired operating conditions within the working envelope. The AHRI standard polynomial works best for single-stage compressor with no refrigerant injection. The AHRI polynomial does not capture the performance of a vapor-injection scroll compressor. This polynomial neglects the impact of the injection fluid on the compressor performance. As vapor-injection scroll compressors are commercially available and implemented in current systems, the proposed dimensionless correlation, Equation (2), is a method for mapping vapor-injection scroll compressor performance.

This work presents a new compressor map available for a R-407C vapor injection scroll compressor for use in an air-conditioning system. R-407C is an HFC refrigerant that has been used as a replacement for R-22 in residential and commercial air-conditioners and heat pumps and some existing medium-temperature applications since 2010. The vapor-injection compressor performance was compared with the baseline performance for the same compressor with the injection port closed. The coefficient of performance was defined using Equation (8).

Thus far, economized, vapor injection research has considered the economization method, the injected refrigerant phase, and the port location. The characteristics of a two-stage heat pump system in hot climates was investigated by Wang et al., 2009 using both flash tank (FTC) and internal heat exchanger (IHX) configurations for vapor-injected compression. Compared to a conventional system using a single-stage compressor with the same suction volume, there was a cooling capacity gain of 15% and a COP improvement of 2% at an ambient temperature of 46.1°C. It was concluded that the maximum cooling COP improvement ranged from 2-4% depending on the ambient conditions. Also, the vapor injection almost equally affected the capacity and power consumption. At an ambient temperature of -17.8°C the heating capacity improvement was 30% with a maximum COP improvement of 23% which was achieved by the FTC. Both the IHX and FTC configurations showed an improvement from the baseline. The maximum COP improvement from baseline was achieved by the FTC at the -17.8°C ambient. The IHX configuration enabled a wider operating range of the injection pressure than that of the FTC due to its freedom of setting the injected refrigerant superheat at the injection port. A TXV was suggested for simple and effective control of the injected refrigerant and the FTC required a larger TXV at the evaporator inlet.

An experimental performance analysis of different hermetic scroll compressors with vapor injection, liquid injection and no injection was conducted by Winandy et al., 2002. This work generated a vapor-injection and a liquid-injection compressor maps for the refrigerant R-22. A semi-empirical model that was able to compute variables of first importance, such as the mass flow rate, the electric power consumption and the discharge temperature, as well as secondary variables, such as suction heating-up, discharge cooling-down, and ambient losses, was validated using 45 experimental results.

Navarro et al., 2013 characterized a vapor-injection scroll compressor in a heat pump at low, intermediate, and high pressure and temperature conditions. A correlation for the injection mass flow rate as a function of external conditions and the intermediate pressure was developed. The VI compressor behavior was compared with the same compressor with the injection port closed and to a single-stage compressor of the same displacement. Compared to the single-stage compressor, the heating capacity and COP increased up to 20% and 10%, respectively. The primary performance differences were shown at higher pressure ratios. The VI compressor isentropic and volumetric efficiency were lower for low and medium pressure ratios and had negligible differences at higher pressure ratios. The discharge temperature decreased, especially at extreme conditions, with more than a 10 K difference. This enables a wider operating map and improved compressor life and reliability. Regarding the injection state, the injection superheat did not influence the system COP and the intermediate pressure improved the cycle performance until wet injection occurred.
Dimensionless PI-type mappings for vapor-injected compression for system simulation were introduced by Bach et al. 2015 and are the basis of the correlation presented here. In Bach et al.’s work, the differences of in-system compressor performance testing using a cold-climate heat pump with a dual-port vapor injected compressor were compared to compressor performance testing using a hot-gas bypass test stand and a calorimeter. The main differences between in-system testing and test stand testing can be found in suction and injection superheat values, discharge temperatures, and range of injection flow rates. More mapping points were created from in-system testing whereas the test stand data was limited to two discharge pressures. The Buckingham-PI theorem was used to create a dimensionless output group and an appropriate dimensionless input group. Mappings of volumetric and overall isentropic efficiencies, compressor heat loss, injection flow rates, and discharge pressure were developed as the output group. It was found that inclusion of injection superheats did not lead to significant improvement of accuracy for the mappings. In-system data alone did not always result in reasonable predictions of the mapped test stand data due to the large difference of the test matrices and operating conditions. The reverse was also true and therefore, combining all data sets for finding mapping coefficients lead to more generally applicable mappings.

2. Experimental Setup
Economized vapor injection systems are vapor compression systems modified to include a flash tank (open economizer) or intermediate heat exchanger (closed economizer) for system performance improvement. The closed economizer increases the subcooling of the refrigerant exiting the condenser outlet resulting in a specific enthalpy decrease at the evaporator inlet leading to increased cooling capacity. The refrigeration cycle is represented in Figure 1 and was the basis of the experimental setup in a calorimeter originally constructed by Moesch et al [3].

An existing calorimeter was modified to include an internal heat exchanger as shown in Figure 2 using a plate heat exchanger as the closed economizer and downstream extraction of the injected refrigerant. The modification was integrated after the filter drier in the liquid line of the calorimeter unit using existing oil sample ports. An electronic expansion valve (EXV) was used to control the injection flow as it extended the range of injection superheat control and enabled two-phase injection. A solenoid valve was installed before the EXV to protect the compressor from two-phase refrigerant entering the compressor chamber. Two service valves at the upper connections of the heat exchanger allowed a pressurization and evacuation of the modification section independent from the main system. The compressor vibration was reduced by a vibration absorber before the injection port to insulate the modification. All modification related components and pipes were placed inside the compressor chamber to reduce the overall volume added to the existing test setup, thus reducing the additional refrigerant charge. To reduce the impact of the chamber temperature on the liquid subcooling and the injection superheat, all modification related pipes and components within the compressor chamber were insulated. The instrument measurement accuracies, testing procedures and test results are compliant to the ANSI/ASHRAE Standard 23.1-2010 [1]. A schematic of the instrumentation and piping diagram indicating all significant features is shown in Figure 3.

The operating temperatures are measured with T-Type thermocouples, which were either fully immersed in the refrigerant flow (TC-1 to TC-8B) and water flow (TC-9, TC-10) or closely attached to the measured ambient temperature (TC-11, TC-12). Four temperature sensors (TC-13 to TC-16) were added for measurement of the economized, vapor-injected refrigerant.

The system pressures were measured using high-accuracy absolute capacitance pressure transducers with full scale ranges of 0 to 689 kPa for low-pressure measurements (PE-1 and PE-6), 0 to 3,447 kPa for the secondary calorimeter pressure measurement (PE7) and 0 to 6,895 kPa for high pressure measurements (PE-2 to PE-5). Three pressure sensors (PE-8, PE-9) were added for refrigerant pressure measurements of the economized, vapor-injected refrigerant. The pressure transducers provided an output signal of 4 to 20 mA, which was translated to a 1 to 5 V signal by using a 250 Ohm resistor parallel to each output.
The water flow through the condenser was measured using a turbine flow meter with a measurement range of 0 to 4.542 m³/h. The output signal is a frequency with a maximum of 1,800 Hz. The frequency signal is converted into a 4 to 20 mA signal by the flow computer. The process water control in the condenser was modified by installing an additional 15.875 mm ball valve to raise and maintain condensing temperatures in the air-conditioning temperature envelope.

The compressor chamber temperature was set to 35 °C, the fan was set to its maximum speed, the value of the suction superheat is adjusted to be 11.1 K, and the voltage for each phase of the compressor was set to 230V at 60 Hz.

![Figure 1. Schematic of an economized vapor injection vapor compression system with downstream extraction and the corresponding log(P)-h diagram](image1)

![Figure 2. Instrumentation and piping diagram of the vapor injection modification](image2)

The primary method for the refrigerant flow measurement was a Coriolis-effect mass flow meter. The flow meter was connected to a transmitter, which translates a mass flow rate of 0 to 544 kg/h to a frequency signal 0 to 2,000 Hz. A transmitter translated the frequency to a 4 to 20 mA signal. The refrigerant mass flow rate was confirmed with the secondary calorimeter method based on ANSI/ASHRAE Standard 41.9-2010.

The air flow in the compressor chamber (AV-1) was measured using an air velocity transmitter, which was a hot wire anemometer with a range of 0 to 5.08 m/s and an output signal of 4 to 20 mA.

The power consumption of the compressor was measured using a watt/watthour transducer with a full-scale range of 0 to 20 kW. The transducer uses two power meters which are set up as an Aron circuit and enable the measurement of the effective power consumption for the 3-phase power supply. The compressor power consumption is displayed in W and its total energy consumption in Wh.
compressor voltage and current were measured only for the first phase using a current transducer and a voltage transducer with full scale ranges of 0 to 20 A and 0 to 600 V. The power consumption of the secondary calorimeter heater is measured using a watt/watthour transducer with a full-scale range of 0 to 18 kW. The output of the power, voltage and current transducers is a 0 to 1 mA signal, which is transformed to 0 to 5 V using a 5,000 Ohm resistor.

Figure 3. Full piping and instrumentation diagram

3. Compressor Map and Performance Results
The R-407C vapor-injection compressor map is presented in Table 1. The injection ratio ranged from 0.09 to 0.28 (9). The compressor input power ranged from 3.2 to 5.6 kW. The isentropic efficiency, Equation (4), ranged from 64 to 67%. The volumetric efficiency, Equation (3) ranged from 88 to 93%.

| P_{EVAP} (kPa) | P_{COND} (kPa) | \dot{m}_{ref} (kg/s) | \dot{m}_{inj} / \dot{m}_{ref} | W (W) | T_{DIS} (K) | P_{INJ} (kPa) | T_{INJ} (K) | T_{SUC} (K) |
|----------------|----------------|----------------------|-----------------------------|-------|-------------|---------------|-------------|-------------|
| 445.4          | 2221           | 0.06426              | 0.2824                      | 4815  | 366.0       | 859.1         | 298.2       | 283.2       |
| 488.1          | 2221           | 0.07031              | 0.2688                      | 4870  | 363.5       | 813.6         | 296.2       | 286.2       |
| 591.6          | 2222           | 0.08618              | 0.2105                      | 5003  | 360.4       | 935.6         | 301         | 291.8       |
| 444.0          | 1690           | 0.06577              | 0.1839                      | 3812  | 351.2       | 661.9         | 289.8       | 283.3       |
| 538.5          | 1686           | 0.08013              | 0.1415                      | 3929  | 348.5       | 775.7         | 294.8       | 288.9       |
| 329.6          | 1460           | 0.04838              | 0.2031                      | 3239  | 350.8       | 584.7         | 286         | 275.2       |
| 536.4          | 1459           | 0.08013              | 0.09434                     | 3501  | 343.3       | 830.8         | 296.9       | 289         |
| 539.2          | 1934           | 0.07938              | 0.1714                      | 4369  | 355.6       | 915.6         | 300.2       | 289.2       |
| 403.3          | 1938           | 0.05897              | 0.2436                      | 4218  | 360.7       | 753.6         | 293.5       | 280.3       |
| 364.7          | 1938           | 0.05292              | 0.2714                      | 4179  | 363.5       | 704           | 291.7       | 277.9       |
| 593.6          | 2518           | 0.08543              | 0.2478                      | 5588  | 368.0       | 1095          | 306.4       | 292         |
The baseline compressor performance is compared to the vapor-injection compressor performance in Table 2 with the measured condensing pressures. The reduction in the discharge temperature ranged from 0 to 3.3 K. The highest reduction in discharge temperature occurred at the largest pressure ratio occurring at 322 K condensing and 267 K evaporating temperatures. The highest COP improvement of almost 9.9% occurred at this pressure ratio as well. Conversely, there was virtually no reduction in discharge temperature and no COP improvement at the lowest pressure ratio at 277.7 K evaporating and 311.4 K condensing temperatures. The largest reduction in isentropic efficiency was 2.86%. The volumetric efficiency reduction ranged from 0 to 1.56%. The reduction in isentropic efficiency is due to the additional power required for vapor-injected compression. The difference in volumetric efficiency was minimal but is likely due to rounding and measurement uncertainties.

Table 2: Closed-port (baseline) and open-injection port compressor performance comparison

| P_{EVAP} (kPa) | P_{COND} (kPa) | T_{DIS} (ΔK) | COP (Δ%) | η_{isen} (Δ%) | η_{vol}(Δ%) |
|----------------|----------------|--------------|----------|---------------|-------------|
| 536.4          | 1459           | 0.7          | 0.0      | -2.29         | 0.10        |
| 329.6          | 1460           | -2.6         | 7.2      | -0.67         | -0.07       |
| 444.0          | 1690           | -1.9         | 2.1      | -0.61         | 0.05        |
| 538.5          | 1686           | -1.0         | 0.9      | -1.86         | -0.17       |
| 539.2          | 1934           | -0.3         | 1.9      | -2.56         | -0.39       |
| 403.3          | 1938           | -3.0         | 8.0      | -1.42         | -0.12       |
| 364.7          | 1938           | -3.3         | 9.9      | -1.17         | -0.06       |
| 593.6          | 2518           | -2.2         | 6.9      | -2.50         | -0.86       |
| 445.4          | 2221           | -3.2         | 9.4      | -1.78         | -1.01       |
| 488.1          | 2221           | -3.0         | 4.7      | -1.10         | -1.56       |
| 591.6          | 2222           | -1.6         | 2.8      | -2.02         | -0.97       |

4. Mapping of Results
The AHRI polynomial as shown in Equation (1) is typically used to predict the power input, suction mass flow rate, and the refrigerating capacity for positive displacement compressors and compressor units. Its mapping coefficients C_1 through C_{10} are generated based on manufacturer data. The user can then predict their compressor behavior at specified operating conditions.

\[ W, \dot{Q}, \dot{m}_{\text{ref}} = C_1 + C_2 T_{\text{evap}} + C_3 T_{\text{cond}} + C_4 T_{\text{evap}}^2 + C_5 T_{\text{evap}} T_{\text{cond}} + C_6 T_{\text{cond}}^2 + C_7 T_{\text{evap}} + C_8 T_{\text{cond}} T_{\text{evap}} + C_9 T_{\text{evap}} T_{\text{cond}}^2 + C_{10} T_{\text{cond}}^3 \]  

(1)

The Buckingham-PI theorem as shown in Equation (2) was used to map the compressor performance for the single-speed R-407C scroll compressor. Appropriate dimensionless input groups were considered for each normalized output PI group listed in Equation (3). This correlation considers the compressor power supply frequency f_{\text{nominal}} as well as the suction state, injection mass flow rate and discharge pressure. Definitions for the abbreviations are documented in the nomenclature. The injection state was normalized by the suction state. The discharge and suction states were normalized by the appropriate refrigerant critical point values. The compressor chamber temperature was considered by normalizing the suction temperature to this temperature.

The enthalpy difference Δ\( h_{\text{suc,sh}} \) at the compressor inlet is characterized by the difference between the measured inlet enthalpy and the saturated liquid enthalpy at the same pressure. The definition of the volumetric efficiency was defined as the ratio of the actual compressor suction volume flow rate to the maximum theoretical compressor suction volume flow rate as shown in Equation (3). The theoretical suction volume flow rate was defined as the product of the maximum volume of the suction process \( V_{\text{disp}} \) and the compressor rotation rate \( N_c \). The isentropic efficiency was defined as the ratio of the actual compressor power consumption to the power consumption needed for an adiabatic and
reversible process from the suction port to the discharge port as shown in Equation (4). The isentropic injection specific enthalpy \( h_{\text{inj,isen}} \) was defined assuming isentropic compression from the suction state to the intermediate pressure. The specific enthalpy of the refrigerant entering the compressor defines \( h_{\text{suc}} \). The specific enthalpy of the refrigerant vapor after mixing between the intermediate pressure flow and the partially compressed flow at state \( h_{\text{post,inj}} \) documented in Equation (5). The specific enthalpy of the fluid at the compressor discharge pressure following an isentropic compression of the refrigerant vapor from state \( h_{\text{post,inj}} \) results in \( h_{\text{dis,isen}} \). The compressor heat loss is estimated from an energy balance assuming a control volume around the compressor surface as shown in Equation (6). The heat loss ratio \( f_q \), Equation (7), was then defined as the ratio of the compressor heat loss to the measured power input. The mapped coefficients for the discharge temperature at the compressor outlet, the injection mass flow rate, the overall (isentropic) efficiency, volumetric efficiency, and heat loss of the compressor are highlighted in Table 3. The coefficients were generated using the Variable Metric optimization method for minimization in Engineering Equation Solver (Klein, 2012).

\[
\eta_{\text{out}} = C_0 \frac{f_{\text{power}}}{f_{\text{nominal}}} C_1 \frac{m_{\text{inj}}}{m_{\text{ref}}} C_2 \frac{p_{\text{inj}}}{p_{\text{suc}}} C_3 \frac{T_{\text{suc}}}{T_{\text{amb}}} C_4 \frac{p_{\text{dis}}}{p_{\text{suc}}} C_5 \frac{\Delta h_{\text{suc,sh}}}{\Delta h_{\text{suc,fg}}} C_6 \frac{p_{\text{dis}}}{p_{\text{crit}}} C_7 \frac{p_{\text{suc}}}{p_{\text{crit}}} C_8 \frac{T_{\text{suc}}}{T_{\text{crit}}}
\]

(2)

\[
\eta_{\text{vol}} = \frac{m_{\text{ref}} h_{\text{suc}}}{V_{\text{disp}} N_C}
\]

(3)

\[
\eta_{\text{isen}} = \frac{m_{\text{ref}} (h_{\text{inj,isen}} - h_{\text{suc}}) + m_{\text{tot}} (h_{\text{dis,isen}} - h_{\text{post,inj}})}{W}
\]

(4)

\[
h_{\text{post,inj}} = \frac{m_{\text{ref}} h_{\text{inj,isen}} + m_{\text{inj}} h_{\text{inj}}}{m_{\text{tot}}}
\]

(5)

\[
m_{\text{ref}} h_{\text{suc}} + m_{\text{inj}} h_{\text{inj}} - m_{\text{tot}} h_{\text{dis}} + W - \dot{Q}_{\text{loss}} = 0
\]

(6)

\[
f_q = \frac{\dot{Q}_{\text{loss}}}{W}
\]

(7)

\[
\text{COP} = \frac{\dot{Q}}{W}
\]

(8)

Injection ratio = \( \frac{m_{\text{inj}}}{m_{\text{ref}}} \)

(9)

The mapped coefficients for the dimensionless correlation for the different aspects of the compressor performance are presented in Table 3 along with the \( R^2 \) value for comparison with the AHRI polynomial performance. The proposed dimensionless correlation is more accurate than the AHRI polynomial for mapping the injection ratio, discharge temperature, and compressor heat loss as shown in Figures 4, 5 and 8, respectively. The predicted volumetric efficiency and isentropic efficiency from the AHRI-polynomial is slightly more accurate. However, the predicted volumetric efficiency values from the dimensionless correlation is within 1% of the measured valued as shown in Figure 6. Similarly, the predicted isentropic efficiency values are within 2% of the measured values as shown in Figure 7.
Table 3: Mapping results for the compressor performance

| Coefficients | $\frac{m_{\text{inj}}}{m_{\text{ref}}}$ | $\frac{T_{\text{dis}}}{T_{\text{crit}}}$ | $\eta_{\text{isen}}$ | $\eta_{\text{vol}}$ | $f_q$ |
|--------------|-------------------------------------|-------------------------------------|---------------------|-------------------|------|
| $C_0$        | 0.1087                              | 0.7637                              | 1.237               | 1.217             | 0.9727 |
| $C_1$        | 1                                    | 1                                    | 1                   | 1                 | 1     |
| $C_2$        | 0                                    | -0.117                               | 0.4838              | -0.0004           | -0.9649 |
| $C_3$        | -0.7606                              | 0.0109                               | 0.03063             | -0.00329          | -1.694 |
| $C_4$        | 0.2705                               | -0.1385                              | 0.2827              | -0.1713           | 0.9395 |
| $C_5$        | 1.443                                | 0.714                                | -0.1018             | 0.2913            | 1.758  |
| $C_6$        | -0.6861                              | 0.7784                               | 1.033               | -0.4828           | 0.6728 |
| $C_7$        | 0.7626                               | 0.103                                | -0.7837             | -0.369            | 1.47   |
| $C_8$        | 0.316                                | 0.3898                               | 0.3174              | 0.3454            | 0.6815 |
| $C_9$        | -0.3924                              | 0.1167                               | -0.2301             | 0.1807            | -1.147 |

Dimensionless Correlation $R^2$ value: 99.5
AHRI-polynomial $R^2$ value: 99.29

Figures 4 and 5. Injection ratio parity plot (left) and the discharge temperature parity plot (right)
5. Conclusion
A new correlation for mapping single-port vapor injection compressors is proposed and was validated using compressor test results obtained with a standard calorimeter. The new correlation is expected to more accurately predict compressor performance than the AHRI-polynomial vapor-injection compressors including variable speed compressors than the AHRI-polynomial. An R-407C vapor-injection scroll compressor map was generated and a dimensionless correlation for mapping results was presented and compared to the standard AHRI-polynomial.

6. References
[1] Hall, R. L., Douglas, J. L., Irons, M. T., Leyderman, A. D., Scott, J. P., Slayton, C. R., … Peterson, J. C. (2010). ASHRAE STANDARD Methods of Testing for Rating the Performance of Positive Displacement Refrigerant Compressors and Condensing Units that Operate at Subcritical Temperatures of the Refrigerant. Ashrae Standard.
[2] Klein, S. A. (2012). Engineering equation solver, academic commercial version 9.434-3d. F-Chart Software, LLC. Madison, WI.
[3] Moesch T W, Bahman A M, Groll E A, 2016 Performance Testing of a Vapor Injection Scroll Compressor with R407C, Proc. Int. Comp. Eng. Conf. (Purdue)
[4] Navarro, E., Redón, A., González-Macia, J., Martínez-Galvan, I. O., & Corberán, J. M. (2013). Characterization of a vapor injection scroll compressor as a function of low, intermediate
and high pressures and temperature conditions. *International Journal of Refrigeration*, 36(7), 1821–1829. http://doi.org/10.1016/j.ijrefrig.2013.04.022

[5] Standard, A. R. I. (2004). Standard For Performance Rating Of Positive Displacement Refrigerant Compressors And Compressor Units, 540.

[6] Wang, B., Shi, W., Han, L., & Li, X. (2009). Optimization of refrigeration system with gas-injected scroll compressor. *International Journal of Refrigeration*, 32(7), 1544–1554. http://doi.org/10.1016/j.ijrefrig.2009.06.008

[7] Wang, X., Hwang, Y., & Radermacher, R. (2009). Two-stage heat pump system with vapor-injected scroll compressor using R410A as a refrigerant. *International Journal of Refrigeration*, 32(6), 1442–1451. http://doi.org/10.1016/j.ijrefrig.2009.03.004

[8] Winandy, E. L., & Lebrun, J. (2002). Scroll compressors using gas and liquid injection: Experimental analysis and modelling. *International Journal of Refrigeration*, 25(8), 1143–1156. http://doi.org/10.1016/S0140-7007(02)00003-8

### 7. Nomenclature

| Symbol | Definition | Unit | Subscripts |
|--------|------------|------|------------|
| H      | enthalpy   | (kJ/kg) | comp compressor |
| ṁ     | mass flow rate | (kg/s) | evap evaporator |
| S      | entropy    | (kJ/kg-K) | cond condenser |
| P      | pressure   | (kPa) | crit critical |
| W      | Power      | (W) | dis discharge |
| C      | mapping coefficients | (-) | disp displacement |
| T      | temperature | (°C) | loss |
| Q      | cooling capacity | (W) | inj injection |
| η      | efficiency | (–) | isen isentropic |
| V      | specific volume | (m³/kg) | post after injection |
| COP    | coefficient of performance | (–) | ref compressor inlet state |
| V      | volume     | (m³) | sh superheat |
| IHX    | intermediate heat exchanger | (–) | sub subcooled |
|        |            |      | suc suction |
|        |            |      | tot total, discharge mass flow rate |
|        |            |      | vol volumetric |