Dual port vapor injected compression: in-system testing versus test stand testing, and mapping of results.

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This paper introduces the differences of in-system testing of a dual-port vapor injected compressor as compared to test stand testing and presents the performance results. Single-port vapor injected compression has been studied previously, typically using scroll compressors. Single-port vapor injection employs one port with a single intermediate injection pressure, while dual-port vapor injection uses two ports at different angular locations, leading to two intermediate pressure levels for the injected vapor. The main differences between in-system testing and test stand testing can be found in suction and injection superheat values, discharge temperature, and range of injection flow rates. Dimensionless PI-type mappings are introduced that can be used for system simulation.

Keywords: vapor-injected compression, mapping, compressor, vapor injection, cold climate heat pump

INTRODUCTION

Air source heat pumps are known to provide up to three times more heat energy to a home than the electricity they consume (DOE, 2015). In addition, the electricity used to operate air source heat pumps could come from renewable sources. The use of renewables for heating using other technologies is more challenging as pointed out by Hutzler and Groll (2013). However, Borre (2009) points out that it is much more difficult to apply heat pumps in extreme cold conditions because of diminishing heating capacity and efficiency. In fact, a significant amount of auxiliary heat is needed in cold climates, which reduces the overall system efficiency. As a result, the European Union considers heat pumps to be renewable if they produce more heat than they consume in auxiliary energy (EN Directive 2009/28/EC, point 31). The work described in this paper was carried out as part of a project to develop high performance heat pumps for cold climates and focuses on characterizing the performance of a specialized compressor for this purpose.

Compressors are the main power consumer in air source heat pumps and therefore are of primary interest for reducing power consumption and increasing efficiency. The overall isentropic efficiency of the compression process generally decreases with increasing compression ratio. This decreases the coefficient of performance in applications with high pressure ratio, such as cold climate heat pumps (CCHPs). One way to overcome this problem is to use a staged compression process with multiple compressors, implemented in form of a cascade, intercooler, or economizer cycle as shown in Bertsch and Groll (2005).

Caskey et al. (2012) introduced a cold climate heat pump with economizer cycle and noted that an oil management system is needed to compensate for oil migration between compressors. In the final field test results (Hutzel and Groll, 2013), it is noted that compressor failure occurred due to oil migration between the compressors. Hutzel and Groll (2013) were able to overcome the issue of oil migration by using better controls. However, compressor failure is preprogrammed in case of a failure of these controls. Vapor injected compression circumvents any such oil migration issues by employing only a single compressor. Additionally, the reduced count of components and controls complexity could reduce cost.

Single-port vapor injected compression has been investigated by a number of different researchers (e.g. Wang et al. 2009, Navarro, et al. 2013, and Xu et al. 2013). Xu et al. (2009) provides an extensive literature review, which includes potential applications for vapor injected technology. The common conclusion of different researchers is that single-port vapor injection at low suction pressures leads to an increase in COP, an increase in low temperature heating capacity, and a decrease in discharge temperature. For high suction pressures (e.g. CCHP at high ambient temperature), the COP can be lower than the one for the baseline system due to re-expansion losses at the injection ports.

Note that there are different ways to integrate vapor injected compression within the system. Heo et al. (2011) presented flash tank and subcooler cycles, which are the two most common approaches, together with two more complicated system layouts. They experimentally investigated the different cycles and found that the flash tank cycle leads to the highest capacity while the COPs of the different configurations were similar.

Mathison (2011) and Mathison et al. (2010, 2011, and 2014) conducted simulation studies to investigate the effect of increasing the number of injection ports as well as using two-phase refrigerant rather than saturated vapor for the injection. Mathison et al. (2014) showed that two injection ports lead to a 28% improvement in COP if saturated vapor injection is used, with the injection of two phase mixture only leading

to an additional 1% improvement. Further increasing the number of ports was shown to lead to additional benefits, with the per-port improvement of performance decreasing with each additional port. From a cost perspective, two ports might be an economic optimum between initial cost and energy savings during system lifetime.

Since two-phase injection is complicated to achieve, experimental work carried out by the same group was limited to using saturated or slightly superheated vapor (Ramaraj, 2012, Song, 2013, Song et al. 2014, Bach, 2014). The main result was an increase in heating COP and capacity. This increase generally was larger for lower suction pressures, while a decrease in COP occurred for higher suction pressures.

Song et al. (2014) introduced a mapping for suction, as well as injection flow-rates, power consumption, and discharge temperature. Their mapping was based on test stand data and was limited to a single value of discharge pressure, compressor speed, and suction superheat and did not show a satisfactory performance for the power consumption ($R^2 < 84\%$).

The purpose of this paper is to obtain a more accurate mapping, which includes different compressor speeds, discharge pressures and a range of suction superheat values, and furthermore compare test stand data against in-system data. Accurate mappings will be useful for system simulation, while a general understanding of the difference between test stand and in-system data will be useful to understand the limitations of the resulting data.

Scroll compressor with dual port vapor injection

Figure 1 shows a cross-section of the scroll compression mechanism of the scroll compressor used to obtain data for this paper. It is composed of an orbiting scroll and a stationary scroll. The stationary scroll has two injection ports, which are connected to external ports at the compressor shell. The low pressure vapor injection port is located at an angular position that opens up directly after the outside suction pocket is closed. The high pressure injection port is located further away from the point at which the inside suction pocket is closed. The angular position between seal-off point and injection port determines the optimum pressure for the injection ports. The injection ports were not equipped with check-valves, but the passage to the port can be plugged inside of the stationary scroll for single-stage operating mode. The displacement of the compressor is 4 cubic inches (65.5 cm³), with an inbuilt volume ratio of 3.29. More details on this compressor can be found in Song (2013).

Figure 1: Employed dual port VI scroll, Song (2013), modified.

Test setups and test plans

The data used for the development of the correlation was obtained using three different test setups:

1. A modified calorimeter test stand (“calorimeter”) that allows for adjustment of the injection flow rates.
2. A hot-gas bypass test stand (“HGB test stand”) with two brazed plate type heat exchangers used for the generation of vapor for injection.
3. A cold climate vapor injected heat pump (CCHP, “in-system testing”) modified for vapor injection by using two vapor separators.

Each of the setups led to a different range of injection flow rates, condensing and evaporating pressures and compressor ambient temperature. While the compressor was directly exposed to the ambient air in the test stands, it was insulated for the in-system testing.

Modified Calorimeter. A calorimeter was modified by the compressor manufacturer to allow for vapor injected compression as shown in [Figure 2](#). Part of the refrigerant exiting the compressor is cooled by a desuperheater, while the remaining refrigerant enters the calorimeter. The refrigerant exiting the desuperheater is expanded using two expansion valves and injected at a high and low pressure level into the injection ports of the compressor. Temperature and pressure is measured at both injection ports as well as at

the suction and discharge of the compressor. Compressor motor power consumption as well as internal motor shaft speed were measured. The environment of the compressor was maintained at 95°F (35°C), and the compressor was not insulated.

Figure 2: Calorimeter modified for vapor injection

Modified Hot-Gas Bypass Test Stand. An existing hot-gas bypass test stand was modified for vapor injection by implementing two desuperheaters with control valves for both branches. Pressures and temperatures of the refrigerant entering and leaving the compressor as well as compressor power were measured. The setup was operated in an air conditioned laboratory environment with an air temperature of $\approx 70^\circ\text{F}$ (21°C). The compressor was not insulated. A more detailed schematic of this setup, including a listing of all instrumentation, can be found in Song (2013), while details on the controls of the original hot gas bypass test stand can be found in Bell (2011).

Figure 3: Simplified schematic of modified hot gas bypass test stand

Cold Climate Heat Pump (CCHP, In-System Testing). The cold climate heat pump uses two vapor separators to split up the expansion process to generate high and low pressure saturated vapor for injection. Injection temperatures were measured using surface mount thermocouples on the insulated ($\frac{1}{2}$ inch (1.3 cm) thick closed cell foam) injection lines ≈ 10 cm from the compressor's shell, while the injection pressures were measured ≈ 5 cm from the compressor shell via capillary tubes. The measurement on the lines as well as live video from sight glasses installed ≈ 25 cm upstream of the injection ports confirmed that the condition was indeed saturated and not superheated or containing large amounts of suspended liquid droplets.

Compressor inlet and outlet pressures and temperatures as well as compressor power consumption were measured. The compressor was insulated with 1 inch (2.5 cm) closed cell insulation foam on all sides except for the bottom surface of the shell which was not accessible after mounting it inside of the outdoor unit.

The layout of the test stand, [Figure 4](#), allows reconfiguring the system for different operating modes: single stage, evaporator flash-gas bypass, and vapor injection mode. In single-stage and flash-gas bypass mode, internal plugs are used within the stationary scroll to reduce recompression losses. In vapor injected operating mode, these plugs are removed. All components, with exception of the indoor unit and its instrumentation, as well as the refrigerant mass flow meter, are located in the outdoor environment. More details on this setup can be found in Bach (2014).

Figure 4: Simplified schematic of cold climate heat pump, grey components inactive for VI operating mode.

Test Plans. The test plans for the calorimeter (~~Table 1~~~~Table 1~~) as well as the hot-gas bypass (HGB) test stand (~~Table 2~~~~Table 2~~) employ one single condensing temperature with fixed nominal suction superheat. The calorimeter test plan uses four distinct evaporation temperatures, which are each assigned to a set of injection pressures. The HGB test plan uses three distinct evaporation temperatures, which do not cover the higher end of the pressures from the calorimeter. A range of combinations of higher- and lower injection pressures is given in the test plans. For the HGB test stand, not all injection pressure combinations were possible to achieve during actual testing, reducing the number of available data points. The injection superheats for the HGB test stand were kept as low as possible; for the calorimeter, higher injection superheat was fixed to 5.6 K with lower injection superheat being a result of the common gas temperature from the desuperheater and the expansion process. Compressor speed for both test plans was 60 Hz, referring to the power supply frequency rather than the actual compressor speed. Calorimeter and in-system testing was performed with the same prototype compressor, while the HGB testing was performed with an identically manufactured compressor.

Table 1: Test plan for calorimeter

Parameter	Unit	Nominal Values			
Condensing Temperature	F (K)	110 (316.5)			
Evaporating Temperature	F (K)	-22 (243)	-4 (253)	5 (258)	14 (263)
Compressor Suction Superheat	F (K)	20 (11.1)			
Higher/Lower Vapor Injection Pressure	kPa	970/500	1225/760	1300/850	1465/1020
Higher/Lower Vapor Injection Superheat ¹	F(K)	10 (5.6)/floating: 9...16.7 (16...30)			

Table 2: Test plan for hot gas bypass test stand

Parameter	Unit	Nominal Values		
Condensing Temperature	°C (K)	43.3 (316.6)		
Evaporating Temperature	°C (K)	-30 (243)	-20 (253)	-10 (263)
Compressor Suction Superheat	°C or K	11.1		
Higher/Lower Vapor Injection Pressure	kPa	1100/900, 1300/900, 1300/1100, 1500/1300, 1500/1100, 1500/900, 1700/1500, 1700/1300		
Higher/Lower Vapor Injection Superheat	°C or K	minimum stable, actual range: 1..7/4...13		

¹ Lower pressure injection superheat is floating, see system schematic of modified calorimeter.

Actual number of data points ²	[-]	7	10	7
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The test plan for in-system testing with the CCHP, [Table 3Table 3](#), differs from the previously shown test plans in the following aspects:

- Compressor speed is increased toward lower ambient temperatures, e.g. from 55 Hz at 8.3°C ambient temperature to 70 Hz for ambient temperatures of -8.3°C and lower to increase heating capacity. While this makes sense from a heat pump application point of view, it also leads to statistical correlation between compressor speed and ambient temperature in the data. Additional data points were therefore added to avoid potential misfitting of the data by the chosen map.
- Saturated compressor discharge and suction temperature are not defined explicitly but rather are a result of system operating condition, evaporator configuration and blockage, and compressor speed. The range of saturated discharge temperatures was 307 to 328 K. The minimum observed evaporation temperature at the suction pressure was 238 K, while the maximum was 274 K, which is a larger range than for the test stand testing.
- The nominal evaporator superheat was kept at the minimum possible value or 5 K, whichever is smaller. Compressor suction superheat is therefore a result of the operating conditions and pressure drop/heat-gain between evaporator and compressor and varies between 5.5 K and 19 K.
- Vapor injection pressures are a result of the mass balance between vapor generation at the vapor separators and flash gas intake at the compressor.
- Saturated vapor rather than superheated vapor is injected.

A total of 70 CCHP data points were recorded, which includes points where compressor frequency is varied to match the capacity between different system configurations (e.g. evaporator with and without hybrid flow control), and additional mapping test points at conditions different from the ones outlined in [Table 3Table 3](#).

² A larger range of lower injection pressures for the high pressure injection pressure of 1500 kPa than indicated in the test-plan was recorded.

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Table 3: Test plan for in-system testing with cold climate heat pump.

Test description	Air Entering Indoor Unit Temperature [°C]			Air Entering Outdoor Unit Temperature [°C]			Compressor Speed	Airside blockage
	Dry Bulb	Wet Bulb	Dew Point	Dry Bulb	Wet Bulb	Dew Point	[Hz]	-/light/severe
H1 - low - clean	21.1	≤15.6	≤12.06	8.33	6.11	3.74	40	-
H1 - low - light block	21.1	≤15.6	≤12.06	8.33	6.11	3.74	40	light
H1 - low - severe block	21.1	≤15.6	≤12.06	8.33	6.11	3.74	40	severe
H2 - mid - clean	21.1	≤15.6	≤12.06	1.67	min	min	55	-
H2 - mid - light block	21.1	≤15.6	≤12.06	1.67	min	min	55	light
H2 - mid - severe block	21.1	≤15.6	≤12.06	1.67	min	min	55	severe
H3 - full - clean	21.1	≤15.6	≤12.06	-8.33	min	min	70	-
H3 - full - light block	21.1	≤15.6	≤12.06	-8.33	min	min	70	light
H3 - full - severe block	21.1	≤15.6	≤12.06	-8.33	min	min	70	severe
HX - full - clean	21.1	≤15.6	≤12.06	-17.78	min	min	70	-
HX - full - light block	21.1	≤15.6	≤12.06	-17.78	min	min	70	light
HX - full - severe block	21.1	≤15.6	≤12.06	-17.78	min	min	70	severe

Notes:

> Tests to be repeated with each different system configuration.

> H1 test condition adopted from AHRI 210/240; H2, and H3 modified humidity compared to AHRI 210/240 (2008)

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183 DATA SETS AND COMPARISON OF TESTING RESULTS.

184 Each of the three data sets has its own limitations, due to the differences in the setups and the limitations
 185 of the test plans. This section points out these differences in graphical form. In the mapping section it will
 186 become apparent that none of the data sets by itself is sufficient for obtaining a mapping that can be used to
 187 accurately predict the results of a different setup.

188 **Pressure Ratio.** Figure 5 shows that the pressure ratios increase with decreasing suction pressure for the
 189 test stand data, since the discharge pressure is held constant. The HGB test data has its data points clustered
 190 together at three distinct pressure ratios, while the four data points of the calorimeter contain one additional
 191 pressure ratio. Figure 6 shows that most of the data from the in-system testing shows a similar behavior:
 192 pressure ratio increases with decreasing suction pressure while the discharge pressure generally decreases in
 193 the same direction. Additionally, compressor speed (marker size) increases with decreasing suction pressure.
 194 A number of mapping points were added, which do not follow that general trend. The range of discharge
 195 pressures, compressor speed, and suction pressures of the in-system data is larger than for the test stand
 196 testing.

Figure 5: Pressure ratio for test stand data sets (60 Hz only, 2606 to 2631 kPa discharge pressure).

Figure 6: Pressure ratio for in-system data set with variable speed (mapping points data marked).

Normalized Injection Flow Rates. The injection mass flow rates are normalized by the suction mass flow rate. This normalization makes the flow rates independent from the compressor speed. An excessively large injection flow rate is not beneficial for heat pump applications, since it reduces the flow rate at the evaporator, reducing the amount of heat that can be absorbed from the ambient.

Figure 7 shows the normalized high-pressure (HP) injection flow rates as a function of the low-pressure (LP) mass flow rates for the test stand testing. The HP injection mass flow rate varies between 17% and 65% of the suction mass flow rate, while the LP injection mass flow rate varies between 4% and 54% of the suction mass flow rate. In the most extreme case, 118% of the suction mass flow rate is injected at the 2 injection ports. The general trend is an increase of both injection flow rates with increasing pressure ratio.

Figure 8 shows the normalized HP injection mass flow rates as a function of the LP mass flow rate for the in-system testing. The maximum and minimum HP injection mass flow rates are 4% and 30%, smaller than from the test stand testing datasets. HP injection mass flow rate increases with pressure ratio due to more vapor being available from the first expansion process. The range of LP injection mass flow rates (8%-13%) is much narrower than the range of the test stand data.

Figure 7: Injection flow rates/suction flow rates for test stand data sets (60 Hz only, calorimeter data marked).

Figure 8: Injection flow rates/suction flow rates for in-system data set with variable speed (mapping points data marked).

Injection Pressures. The injection pressures were normalized by suction pressure to be able to directly compare it to the overall pressure ratio of the compressor. Figure 9 shows the normalized HP injection pressure as a function of the normalized LP injection pressure from the test stand testing. The maximum injection pressure ratio for the three pressure ratios is approximately 50% of the overall compressor pressure ratio. The calorimeter used smaller injection pressures, with the LP injection pressures being the smallest for each pressure ratio in the test stand data.

Figure 10 shows the data from the in-system testing; note that a different axis scaling was used. Normalized injection pressures are smaller than for the test stand testing. The injection pressure ratios increase with pressure ratio, due to a larger amount of flash gas being available for higher pressure ratios.

Figure 9: Normalized injection pressures for test stand data sets (60 Hz only, calorimeter data marked).

Figure 10: Normalized injection pressures for in-system data set with variable speed (mapping points data marked).

Suction Superheat. The suction superheat for the data of the test stand testing was within a range of 10.4 to 12.5 K, with exception of the lowest evaporation temperature test of the calorimeter test stand.

Figure 11 shows the suction superheat of the in-system tests as a function of the suction pressure. The suction superheat of the in-system testing covers a much wider range of 5.5 to 19.2 K than that of the test stand testing. Note that five of the ten mapping points were added to reduce the covariance between suction pressure and suction superheat.

Injection Superheats. Figure 12 shows the superheat at the high-pressure vapor injection port as a function of the superheat at the low-pressure vapor injection (VI) port for the test stand data. For the calorimeter, the high-pressure VI superheat is, with 5.4 to 5.9 K, close to the test plan value, while the low-pressure VI superheat varies between 8.1 and 16.6 K. The injection superheats for the high pressure vapor injection of the HGB test stand have a wider spread than the ones of the calorimeter; the dataset contains three data points below a superheat value of 1.5 K, which might indicate that there was no superheat and two-phase flow entered the injection port.

Figure 11: Suction superheat for in-system data set with variable speed (mapping points data marked).

Figure 12: Injection superheat, for test stand data sets (60 Hz only, calorimeter data marked).

Volumetric Efficiency. The volumetric efficiency is defined in terms of the actual volumetric flow rate at the suction port ($\dot{v}_{suc}$) and the mechanical displacement at the suction ($\dot{v}_{disp}$). $\dot{v}_{disp}$ was approximated as the product of displacement per revolution (v_{disp}) and the compressor speed, which was approximated by the actual frequency of the power supply to the compressor (f_{actual}), resulting in

$$\eta_{vol} = \frac{\dot{v}_{suc}}{\dot{v}_{disp}} \approx \frac{\dot{v}_{suc}}{v_{disp} \cdot f_{actual}}. \quad (1)$$

Based on the 60 Hz data from the calorimeter, an error of $\approx 2\%$ is introduced by assuming the rotor frequency being equal to the power supply frequency.

Figure 13 shows the volumetric efficiency for the test stand data as a function of pressure ratio and speed. The volumetric efficiency decreases with increasing pressure ratio, caused by increasing leakage losses and motor slip. Note that the vapor injection flow rates and pressures for the low pressure injection increase towards higher pressure ratios (as shown in Figures 7 and 9) – the higher overall pressure ratio is therefore not the only driver of internal leakage.

Figure 14 shows the volumetric efficiency for the in-system testing as a function of pressure ratio and speed. The volumetric efficiency tends to decrease with decreasing speed. The outliers at low compressor speed are mainly caused by a higher suction superheat and, for the 2 points of lowest volumetric efficiency, by a higher discharge pressure – increasing the influence of rotor slip and internal leakage.

Figure 13: Volumetric efficiency for test stand data sets (60 Hz only, calorimeter data marked).

Figure 14: In-system data set with variable speed (mapping points data marked).

Heat Loss. Heat loss is defined using an energy balance as the difference between compressor power

consumption $\dot{W}_{comp,el}$ and refrigerant inlet and outlet mass flow rates and enthalpies:

$$\dot{Q}_{loss} = \dot{W}_{comp,el} + \dot{m}_{suc} \cdot h_{suc} + \dot{m}_{vi,hp} \cdot h_{vi,hp} + \dot{m}_{vi,lp} \cdot h_{vi,lp} - \dot{m}_{tot} \cdot h_{dis}. \quad (2)$$

Normalized heat loss is defined as

$$\dot{Q}_{loss,norm} = \frac{\dot{Q}_{loss}}{\dot{W}_{comp,el}}. \quad (3)$$

Figure 15 shows the normalized heat loss for the test stand data as a function of discharge temperature and saturated suction temperature, which is within a range of 0 to 6%. Discharge temperature tends to correlate with an increase in heat loss.

Figure 16 shows the heat loss for the in-system testing as a function of discharge temperature and saturated suction temperature. 69% of the data is within a range of 4 to 8% of normalized heat loss. The discharge temperature tends to decrease with increasing saturated suction temperature (= increasing pressure), while the normalized heat losses stay approximately constant due to the simultaneous reduction in compressor speed and mass flow rate. Some points have a higher heat-loss, four of these points are mapping points with a higher suction superheat, while the two points with the highest heat loss are points with reduced compressor speed.

Figure 15: Normalized heat loss for test stand data sets (60 Hz only, calorimeter data marked).

Figure 16: Normalized heat loss for in-system testing (mapping points data marked).

Figure 17 shows that there is a trend of decreasing heat loss with increasing ambient temperature of the compressor environment. Moreover, this trend is approximately consistent across all test data and a linear correlation over the data sets will result in most points being within 1.5% of the measured heat loss. Scatter in the y-direction is caused by different saturated suction pressures as well as by different injection mass flow rates and discharge pressures.

Figure 17: Heat loss as function of ambient and suction temperature.

Over- and Under Compression. Due to the built in volume ratio of scroll compressors, over- and under compression can occur during off-design operating conditions. Over-compression occurs for scroll compressors without vapor injection if the ratio between actual suction and discharge vapor volume is smaller

than the inbuilt volume ratio. In that case, the pressure in the discharge pocket is larger than the discharge pressure that is actually needed, which leads to expansion losses once the compression pocket opens to discharge pressure. Under-compression is the opposite case, the expansion losses are however limited due to less time being available for pressure equalization caused by backflow into the discharge pocket. Figure 18 shows overall isentropic efficiency data taken with the vapor injected compressor with plugged injection ports. The overall isentropic efficiency $\eta_{oi,comp}$ is defined as

$$\eta_{oi,comp} = \frac{\dot{W}_{ise}}{\dot{W}_{comp,el}}, \quad (4)$$

where $\dot{W}_{ise}$ is the isentropic power consumption. For the single stage compressor, $\dot{W}_{ise}$ is based on the suction conditions and becomes

$$\dot{W}_{ise,suc} = \dot{m}_{suc} \cdot (h_{dis,ise} - h_{suc}). \quad (5)$$

Figure 18 shows the overall isentropic efficiency as a function of volume ratio and pressure ratio. The compressor has a built in volume ratio of 3.29. It can be seen that over-compression is much more detrimental to the overall isentropic efficiency than under-compression.

Figure 18: Dependency of overall isentropic efficiency on volume ratio, injection ports plugged, calorimeter and in-system test data

For the vapor injected compressor, the suction to discharge volume ratio cannot be directly used to detect over- or under compression, since refrigerant enters through the injection ports. However, over-compression

can be detected using the discharge volumetric flow rate per revolution as well as the discharge pocket volume. This was defined as the discharge pocket to refrigerant discharge volume ratio, e.g.

$$\frac{v_{dis}}{v_{dis,pocket}} = \frac{\dot{v}_{dis}}{\dot{v}_{mech}} = \frac{\dot{m}_{dis}/\rho_{dis}}{\dot{v}_{suc}/VR} \approx \frac{\dot{m}_{dis}/\rho_{dis}}{v_{dis} \cdot f_{actual}/VR} \quad (6)$$

Where VR is the inbuilt volume ratio of the compressor (3.29), and ρ_{dis} is the density at discharge pressure and temperature measured at the discharge of the compressor. A discharge volume ratio of greater than one indicates that more mass flow rate is discharged than physically possible based on the volumetric displacement of the discharge pocket at the given measured discharge pressure and temperature. Therefore discharge pocket refrigerant density needs to be larger than measured at the discharge conditions when the pocket opens to the discharge pressure, which means that over-compression occurred!

Figure 19 shows that over-compression occurs for the test stand data if the pressure ratio is below 5, e.g. discharge pocket volume ratio increases above 100%, leading to throttling losses once the pocket opens to the high pressure side of the compressor. If a high total injection mass flow rate (normalized by the discharge flow rate) is chosen, than over compression can occur for a pressure ratio of up to 6.5.

Figure 20 shows that, for the in-system testing, over compression (discharge pocket volume ratio > 100%) only occurs below pressure ratios of 4.5. Note that significant under-compression occurs towards higher pressure ratios.

Figure 19: Discharge pocket volume ratio for test stand data sets (60 Hz only, calorimeter data marked).

Figure 20: Discharge pocket volume ratio for in-system testing (mapping points data marked).

Overall Isentropic Efficiency. Overall isentropic efficiency for the vapor injected compressor is defined in the same way as for the single stage compressor. The isentropic power consumption, however, needs to be updated to include the injected flow rates $\dot{m}_{vi,lp}$ at lower injection pressure and $\dot{m}_{vi,hp}$ at higher injection pressure as well as their specific isentropic compression power consumption, e.g.

$$\dot{W}_{ise,vi} = \dot{W}_{ise,suc} + \dot{m}_{vi,lp} \cdot (h_{dis,ise(vi,lp)} - h_{vi,lp}) + \dot{m}_{vi,hp} \cdot (h_{dis,ise(vi,hp)} - h_{vi,hp}). \quad (7)$$

Figure 21 shows that overall isentropic efficiency decreases with increasing total injection flow rate, independently from the pressure ratio. Larger mass flow rates through the injection lines generally correspond to larger injection pressures at the external port, and will cause higher internal injection port throttling losses, reducing overall isentropic efficiency.

Figure 22 shows the overall isentropic efficiency for the in-system testing. When excluding the mapping points, the best overall isentropic efficiency is achieved at a pressure ratio of 5 to 7.5. Above a pressure ratio of 4.5, the decrease in overall isentropic efficiency is smaller than for the test stand data – most likely caused by smaller throttling losses for the vapor injection since the injection mass flow rates are smaller. Note that the maximum overall isentropic efficiency for the in-system testing with vapor injection is approximately 6% points lower than for the data without vapor injection, as can be seen by comparing Figures 18 and 22. This lower overall isentropic efficiency does not necessarily result in a lower system COP, since part of the flash gas of the expansion process now does not need to be compressed from suction pressure.

Figure 21: Overall isentropic efficiency for test stand data sets (60 Hz only, calorimeter data marked).

Figure 22: Overall isentropic efficiency for in-system testing (mapping points data marked).

Mapping of Results

Mapping of the results is initially conducted for all data of the different systems. For some parameters, an additional mapping was based on the test stand or in system data only. The Buckingham PI-theorem (Buckingham, 1914) was employed for all shown mappings, therefore, suitable dimensionless groups were employed for all inputs and each normalized output PI group (Π_{out}), resulting in the following general form:

$$\Pi_{out} = c_0 \cdot \left(\frac{f_{actual}}{f_{nominal}} \right)^{c_1} \cdot \left(\frac{\dot{m}_{vi,lp}}{\dot{m}_{suc}} \right)^{c_2} \cdot \left(\frac{\dot{m}_{vi,hp}}{\dot{m}_{suc}} \right)^{c_3} \cdot \left(\frac{p_{vi,lp}}{p_{suc}} \right)^{c_4} \cdot \left(\frac{p_{vi,hp}}{p_{suc}} \right)^{c_5} \cdot \left(\frac{\Delta h_{SH,vi,lp}}{\Delta h_{fg,lp}} \right)^{c_6} \cdot \left(\frac{\Delta h_{SH,vi,hp}}{\Delta h_{fg,hp}} \right)^{c_7} \cdot \left(\frac{p_{dis}}{p_{suc}} \right)^{c_8} \cdot \left(\frac{\Delta h_{SH,suc}}{\Delta h_{fg,suc}} \right)^{c_9} \cdot \left(\frac{p_{dis}}{p_{crit}} \right)^{c_{10}} \cdot \left(\frac{T_{suc}}{T_{crit}} \right)^{c_{11}} \cdot \left(\frac{T_{amb}}{T_{crit}} \right)^{c_{12}} \cdot \left(\frac{p_{suc}}{p_{crit}} \right)^{c_{13}}, \quad (8)$$

where $f_{nominal}$ is the nominal compressor power supply frequency. The groups with the coefficients c_6 , c_7 , and c_9 capture the effect of superheat at the low and high injection pressure, and at the main suction port onto the output group. The enthalpy difference on the top of each group is enthalpy at the actual inlet condition minus enthalpy of saturated liquid, at the same pressure. The different variables and indices are explained in the nomenclature.

The main issue with Equation 8 seems to be that the injection mass flow rates cannot be known a priori, since they depend on how much vapor can be taken into the injection ports - additionally the normalization depends on the unknown suction mass flow rate. Therefore, these variables need to be determined using a mapping of the test results. It was found that the groups can be dropped, since their mapping depends on the

remaining groups only. The mapping results are shown, together with the mapped output PI groups, in Tables 4 and 5. The subsequent paragraphs show the results together with physical explanation of the coefficients.

Injection Mass Flow Rates. The main parameter influencing the amount of injected vapor is the injection pressure ratio. The injection superheat will influence the density, while the compressor speed and suction temperature will influence the average pressure ratio of the pocket when open to the injection port. Using the pressure ratio for the injection pressure alone lead to an R^2 of 94% for both high pressure and low pressure injection port mass flow rate. While adding additional dimensionless groups leads to an up to 1.6% improvement in R^2 , it seems not wise to do so due to the correlation between different variables in the test plans. The sign for the coefficient of the injection pressure groups is positive and the value of the coefficient is larger for the low pressure injection port due to the smaller pressure of that compression pocket when it opens up to the injection line.

Figure 23 shows the parity plot for the high pressure injection mass flow rate. All data points, except for 5 points below 10% normalized injection mass flow rate, are predicted within $\pm 5\%$ of the measured injection mass flow rate. Figure 24 shows similar results for the low pressure injection mass flow rates, 87% of the points are predicted within 5% of the measured value.

Figure 23: Parity plot –high pressure injection flow rate normalized by suction flow rate

Figure 24: Parity plot –low pressure injection flow rate normalized by suction flow rate

Overall Isentropic Efficiency. Using dimensionless groups for the pressure ratios of discharge to suction and injection ports to suction as well as compressor frequency lead to an R^2 of 86%. The addition of further dimensionless groups lead to an increase in the accuracy of the mapping. The addition of the injection superheat groups leads to opposite exponents for the high and low pressure injection, potentially

indicating overfitting. The overall isentropic efficiency mapping D is therefore not recommended, while mappings A to C lead to qualitatively reasonable results for the coefficients: an increase in suction temperature leads to an increase of compressor heat losses (if all other conditions are the same), reducing the temperature difference between measured suction temperature and inlet temperature of the (suction gas cooled) scroll compressor within the compressor shell. This means that the calculated isentropic work based on externally measured conditions increases by a larger amount than if the temperature was directly measured at the inlet to the scroll. The suction gas superheat group served as a correction for this, with the coefficient c_6 having a negative sign for both the type C and type D mappings.

Overall isentropic efficiency increases with compressor frequency ($c_1 > 0$) due to reduced time for heat transfer and reduced leakage losses. The sign for the injection pressure groups is negative, since larger injection pressures lead to increased throttling losses at the injection ports. Finally, increased overall pressure ratio decreases overall isentropic efficiency ($c_8 < 0$), since it leads to increased internal leakage, larger heat transfer, and increased mechanical losses.

Figure 25 shows the performance of the type C mapping, all points are predicted within $\pm 5\%$ of the measured value. Figure 26 shows the results when relying solely on the in-system data for training the mapping. While the in-system data is still predicted within $\pm 5\%$ of the measured values, this is not the case for all of the HGB test stand data. The most extreme deviation between measurement and prediction is 20%, and 11 points deviate more than 10% from the measured value. This shows that in-system testing data is insufficient if it becomes necessary to extrapolate to larger injection mass flow rates – all data points outside of the 5% limit have injection mass flow rates greater than available in the in-system (= training) data.

Figure 25: Parity plot overall isentropic efficiency - all data used for determination of coefficients of type C mapping

Figure 26: Parity plot overall isentropic efficiency - only in-system data used for mapping of type C mapping

Normalized Heat Loss. The normalized heat loss, $\dot{Q}_{\text{loss,norm}}$, was chosen as output for equation (8). Using only the dimensionless groups of pressure ratio, suction temperature, and ambient temperature resulted in an R^2 of 75%. The coefficients show that the normalized heat loss increases with suction temperature, and pressure ratio and decreases with increasing ambient temperature and compressor speed. The decrease of normalized heat loss with compressor speed is caused by the resulting larger mass flow rate, that increases the power input that is used for normalizing the group – while the compressor shell surface area remains constant.

Figure 27 shows that most (94%) of the data points are predicted within $\pm 2\%$ of the measured value. Two of the three data sets had compressors that were not insulated but operated without forced convection across the shell. These two data sets had heat losses of 6% or less. The third data set (CCHP) had an insulated compressor compressor with forced convection over the outside of the insulation. The results suggest that the lack of insulation (HGB, calorimeter) was partially compensated for by the lack of forced convection.

Figure 27: Parity plot - normalized heat loss

Volumetric Efficiency. The following variables affect the volumetric efficiency of the compressor:

- Pressure ratio, suction and discharge pressure, injection pressures, and compressor speed – affecting leakage through scroll sealing surfaces, wall heat transfer and throttling losses at suction and injection ports.
- Suction superheat and ambient temperature/heat losses– affecting the density at the suction pocket.
- Injection temperature and pressure should only have minor effects onto volumetric efficiency, since the suction pockets closes and turn into a compression pockets before they opens up to the injection ports. Their influence is therefore not considered in the mapping.
- Suction density will be affected by type and thickness of insulation (not considered).

The mapping for volumetric efficiency is therefore based on the same groups as in the heat loss as well as on the groups for suction pressure and suction superheat to capture the effects of suction density. This lead to an R^2 of 79% for all data. Using only the data of the in-system testing increased the R^2 for that data by less than 1%. The coefficient c_6 for the suction superheat is positive to compensate for the decrease in temperature difference between externally measured suction temperature and actual scroll inlet temperature with increase of suction superheat described in the previous paragraphs. Increasing pressure ratios decrease the volumetric efficiency ($c_8 < 0$). The influence of normalized suction temperature and pressure seems to be less clear, with a sign inversion for the suction pressure if only test stand data is used. Most likely the

signs of these groups are a result of achieving a better fit to the data in conjunction with the other simultaneously used dimensionless groups.

Figure 28 shows the mapping results if all data was used for the coefficients, volumetric efficiency is predicted better than $\pm 4\%$ for all but 2% of the data points. A very similar result is obtained if only in-system data is used for obtaining the coefficients. However, if only test stand data is used, then most of the predicted points for the in-system data are very inaccurate while only a small improvement in accuracy was achieved for the test stand data.

Figure 28: Volumetric efficiency, mapping based on all data

Discharge Temperature. The output PI-group for the discharge temperature is discharge temperature normalized by the critical pressure. The effect of vapor injection flow rate absorbed in compressor speed and vapor injection pressures. Inclusion of the groups for the injection superheats did not lead to a significant improvement in accuracy. The coefficients of the mappings show that discharge temperature increase with pressure ratio, LP injection pressure ratio and suction pressure. HP injection pressure increases or decreases discharge temperature depending on the chosen data for finding the coefficients. However, the absolute value of the coefficient for the HP injection pressure is smaller than for the LP injection pressure, suggesting that its influence on the final result is smaller. Larger ambient temperatures lead to larger discharge temperatures, due to reduced compressor heat losses ($c_{12} > 0$ for type B mappings).

Figure 29 shows a parity plot of the type A mapping. The discharge temperatures for all but 4 points of the HGB test stand data are predicted within 5 K of the measured value.

Figure 29: Discharge temperature type A mapping, based on all data for coefficients

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Table 4: Mapping results for overall isentropic efficiency and volumetric efficiency

		Mapped Dimensionless Group							
Π_{out}		$\eta_{oi,comp}$	$\eta_{oi,comp}$	$\eta_{oi,comp}$	$\eta_{oi,comp}$	$\eta_{oi,comp}$	η_{vol}	η_{vol}	η_{vol}
Mapping type		A	B	C	C	D	NA	NA	NA
Data for coefficients		all data	all data	all data	(in-system) in-system data	all data	all data	in-system data	test stand data
Coefficient, Group corresponding to coefficient		Coefficients for Each Mapping							
c_0	NA	0.7878	0.7736	0.7917	1.387	0.7893	0.7446	0.7206	0.003189
c_1	$\left(\frac{f_{actual}}{f_{nominal}}\right)$	0.3877	0.3957	0.4000	0.5074	0.3973	0.2721	0.2431	0 (NA)
c_2	$\left(\frac{\dot{m}_{vi,lp}}{\dot{m}_{suc}}\right)$	0	0	0	0	0	0	0	0
c_3	$\left(\frac{\dot{m}_{vi,hp}}{\dot{m}_{suc}}\right)$	0	0	0	0	0	0	0	0
c_4	$\left(\frac{p_{vi,lp}}{p_{suc}}\right)$	-0.2563	-0.2609	-0.2693	-0.7846	-0.2802	0	0	0
c_5	$\left(\frac{p_{vi,hp}}{p_{suc}}\right)$	-0.1356	-0.1133	-0.1024	-0.6106	-0.09574	0	0	0
c_6	$\left(\frac{\Delta h_{SH,vi,lp}}{\Delta h_{f,g,lp}}\right)$	0	0	0	0	-0.002209	0	0	0
c_7	$\left(\frac{\Delta h_{SH,vi,hp}}{\Delta h_{f,g,hp}}\right)$	0	0	0	0	+0.1612	0	0	0
c_8	$\left(\frac{p_{dis}}{p_{suc}}\right)$	-0.01586	-0.03183	-0.03736	0.1775	-0.03722	-0.3575	-0.3256	-5.119
c_9	$\left(\frac{\Delta h_{SH,suc}}{\Delta h_{f,g,suc}}\right)$	0	0	-0.5503	-0.2387	-0.5602	2.7944	2.837	33.59
c_{10}	$\left(\frac{p_{dis}}{p_{crit}}\right)$	0	-0.03943	-0.04822	0.1238	-0.05181	0	0	0
c_{11}	$\left(\frac{T_{suc}}{T_{crit}}\right)$	0	0	0	0	0	-3.508	-3.707	-29.42
c_{12}	$\left(\frac{T_{amb}}{T_{crit}}\right)$	0	0	0	0	0	0.1789	0.2997	0.08537
c_{13}	$\left(\frac{p_{suc}}{p_{crit}}\right)$	0	0	0	0	0	0.07417	0.09378	-2.414
Data set		Coefficient of Correlation - R^2							
All data		85.6%	85.6%	86.0%	74.3%	86.0%	79.2%	74.1%	2.0%
Test stand data		85.2%	85.0%	85.1%	82.2%	85.4%	68.4%	64.5%	79.0%
Calorimeter data		64.1%	65.2%	69.5%	46.4%	69.3%	98.7%	99.6%	99.8%
HGB data		82.4%	82.1%	81.9%	77.6%	82.0%	70.7%	70.7%	77.0%
In-system data		77.2%	77.6%	78.3%	80.3%	78.3%	86.2%	86.8%	8.4%

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Table 5: Mapping results for discharge temperature, heat loss, and injection flow rates

Π_{out}		Mapped Dimensionless Group							
		$\frac{T_{dis}}{T_{crit}}$	$\frac{T_{dis}}{T_{crit}}$	$\frac{T_{dis}}{T_{crit}}$	$\frac{T_{dis}}{T_{crit}}$	$\frac{\dot{Q}_{loss}}{\dot{W}_{comp,el}}$	$\frac{\dot{Q}_{loss}}{\dot{W}_{comp,el}}$	$\frac{\dot{m}_{vi,lp}}{\dot{m}_{suc}}$	$\frac{\dot{m}_{vi,hp}}{\dot{m}_{suc}}$
Mapping type		A	A	B	B	NA	NA	NA	NA
Data for coefficients		<i>all data</i>	<i>in-system data</i>	<i>all data</i>	<i>in-system data</i>	<i>all data</i>	<i>in-system data</i>	<i>all data</i>	<i>all data</i>
Coefficient, Group corresponding to coefficient		Coefficients for Each Mapping							
c_0	NA	0.9395	0.8661	0.9492	0.8652	0.008704	0.009111	0.03247	0.03847
c_1	$\left(\frac{f_{actual}}{f_{nominal}}\right)$	-0.05321	-0.07314	-0.05591	-0.07524	-1.245	-1.207	0	0
c_2	$\left(\frac{\dot{m}_{vi,lp}}{\dot{m}_{suc}}\right)$	0	0	0	0	0	0	0	0
c_3	$\left(\frac{\dot{m}_{vi,hp}}{\dot{m}_{suc}}\right)$	0	0	0	0	0	0	0	0
c_4	$\left(\frac{p_{vi,lp}}{p_{suc}}\right)$	0.02939	0.09460	0.02135	0.1098	0	0	1.690	0
c_5	$\left(\frac{p_{vi,hp}}{p_{suc}}\right)$	-0.008084	0.05148	-0.01695	0.09491	0	0	0	1.672
c_6	$\left(\frac{\Delta h_{SH,vi,lp}}{\Delta h_{fg,lp}}\right)$	0	0	0	0	0	0	0	0
c_7	$\left(\frac{\Delta h_{SH,vi,hp}}{\Delta h_{fg,hp}}\right)$	0	0	0	0	0	0	0	0
c_8	$\left(\frac{p_{dis}}{p_{suc}}\right)$	0.1976	0.1455	0.2020	0.1249	0.8485	0.8053	0	0
c_9	$\left(\frac{\Delta h_{SH,suc}}{\Delta h_{fg,suc}}\right)$	0.7739	0.7322	0.6984	0.4946	0	0	0	0
c_{10}	$\left(\frac{p_{dis}}{p_{crit}}\right)$	0	0	0	0	0	0	0	0
c_{11}	$\left(\frac{T_{suc}}{T_{crit}}\right)$	0	0	0	0	4.645	4.369	0	0
c_{12}	$\left(\frac{T_{amb}}{T_{crit}}\right)$	0	0	0.07698	0.2181	-6.130	-5.997	0	0
c_{13}	$\left(\frac{p_{suc}}{p_{crit}}\right)$	0.10178	0.0722	0.09457	0.05174	0	0	0	0
Data set		Coefficient of Correlation - R^2							
<i>All data</i>		94.9%	80.8%	95.6%	71.7%	74.8%	74.8%	93.5%	94.2%
<i>Test stand data</i>		93.2%	84.1%	93.0%	84.5%	8.5%	8.3%	93.8%	99.1%
<i>Calorimeter data</i>		98.1%	95.8%	98.0%	94.5%	87.8%	87.6%	83.0%	99.2%
<i>HGB data</i>		93.3%	91.5%	93.5%	91.8%	10.9%	10.8%	98.5%	99.4%
<i>In-system data</i>		94.9%	95.5%	95.5%	96.2%	65.4%	65.4%	66.1%	93.4%

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Summary and Conclusions

This paper introduced mappings for a dual port vapor injected scroll compressor trained with data from 3 different test setups. Experimental data was obtained from a dedicated hot gas bypass (HGB) teststand, a modified calorimeter, and a cold climate heat pump (CCHP, “in-system testing”).

Test stand data was limited to two discharge pressures, while in-system data had a range of discharge pressures which was additionally expanded by adding dedicated “mapping” points. The test stand data shows generally much larger normalized injection mass flow rates and pressures for both injection ports, in sum up to 54% of the discharge mass flow rate. The low pressure injection mass flow rate shows very little variation for the in-system testing (6%-10%), while the high pressure injection flow rate varied from 4% to 21% of the discharge mass flow rate. The data of the test stands showed suction superheat values between 10 and 13 K, while the in-system testing data spanned a wider range of 6 to 19 K.

Mappings of volumetric and overall isentropic efficiency, compressor heat loss, injection flow rates, and discharge pressure were developed. These mappings were based on the Buckingham-PI theorem, where a dimensionless (output) group is predicted by suitable dimensionless (input) groups. It was found that inclusion of injection superheats did not lead to significant improvement of accuracy for the mappings.

Relying solely on the in-system data did not always lead to reasonable predictions for the test stand data for some of the mapped values, due to the large difference of the test matrices and operating conditions. The same was true when relying only on test stand data and predicting in-system data. Combining all data sets for finding mapping coefficients lead to more generally applicable mappings.

Future work.

Additional data analysis should include the calculation of uncertainty for the measured as well as for the predicted values. Compressor modifications should include check valves in direct vicinity to the injection ports to avoid back-flow and the resulting re-expansion losses. Furthermore, the actual injection temperature at the injection port within the scroll should be measured or simulated to determine whether it is necessary to insulate the injection lines within the compressor. The optimization of the test plans could lead to smaller test plans with specifically selected points to reduce the prediction uncertainty of the mappings, especially at points other than the training data.

477 NOMENCLATURE

478 VARIABLES

c_i	Mapping coefficient [-]
COP	Coefficient of performance [kW/kW]
VR	Inbuilt volume ratio of compressor [m^3/m^3]
Δh	Enthalpy difference [kJ/kg]
η	Efficiency [kW/kW or m^3/m^3]
f	Power supply frequency [1/s]
h	Enthalpy [kJ/kg]
$h_{a,ise(b)}$	Enthalpy at point a based on isentropic compression from point b [kJ/kg]
$\dot{m}$	Mass flow rate [kg/s]
P	Pressure [kPa]
$\dot{Q}$	Heat transfer [kW]
ρ	Density [kg/m ³]
T	Temperature [K]
v	Volume [m^3/kg]
$\dot{v}$	Volumetric flow rate [m^3/s]
$\dot{W}$	Power consumption [kW]

479 SUBSCRIPTS

actual	Actual value
amb	Ambient
comp	Compressor
crit	Critical
dis	Discharge
disp	Displacement
el	Electric

fg	Fluid (liquid) to gas
hp	High pressure injection
i	Index
ise	Isentropic
loss	Heat loss
lp	Low pressure injection
mech	Mechanical
nominal	Nominal value
norm	Normalized
oi	Overall isentropic
pocket	Volume enclosed by scroll
SH	Superheat
suc	Suction
tot	Total
vi	Vapor injection
vol	Volumetric

480 **ACRONYMS AND ABBREVIATIONS**

CCHP	Cold Climate Heat Pump
Comp	Compressor
HGB	Hot gas bypass
HP	High pressure
LP	Low pressure
Map.	Mapping
Pts.	Points
Sys.	System
VI	Vapor injection

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