Development of accurate and widely applicable compressor performance maps

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Abstract. The compressor performance map provides critical information, including mass flow rate, power consumption, capacities, and operating current. The map can be applied to an alternative efficiency determination method (AEDM) for HVAC&R products. However, most of the existing maps have a low prediction accuracy under low ambient temperature heating modes, as such conditions were not thoroughly considered when maps were generated. This paper provides a comprehensive study of a fixed speed compressor performance under a wide range of operating conditions, including the AHRI 210/240 standard conditions. The test matrix was developed based on realistic heat pump system operating conditions. We then quantified the existing performance map's prediction inaccuracy, particularly under the low ambient temperature heating modes. Driven by the significant deviation at lower ambient temperature conditions, a comprehensive performance map was generated based on the test results. We also found that one polynomial equation could simultaneously fulfill the needs for predicting the compressor performance under both the cooling and heating modes. Furthermore, the accuracy has improved significantly from 63% to 2% regarding the relative maximum prediction errors over existing performance maps. It suggests that to increase prediction accuracy when using the AEDM, one must provide a suitable superheat model in the capacity simulation of the heat pump systems with the fixed speed compressor.

Keywords: alternative efficiency determination method, compressor performance map, fixed speed compressor

1. Introduction

The Code of Federal Regulations part 429.70 (DOE, 2014) illustrates how we can apply alternative methods to determine energy efficiency and energy use instead of sending the product to a qualified psychrometric test room to conduct experimental tests. VapCyc (Huang et al., 2021) provides an excellent vapor compression cycle's steady-state performance simulation capability. It can be a handy tool along with a heat
exchanger's steady-state performance simulation tool, CoilDesigner (Jiang et al., 2006). Utilizing these two tools after validating their accuracy with the experimental laboratory test results can serve as an alternative method to evaluate the product's energy efficiency and energy use. Industry can save lots of time and cost for experimental works by adopting the alternative efficiency determination method (AEDM). However, some inconsistencies are found when the experimental results of the heat pump systems are compared with the simulation results. When it comes to low ambient temperature heating conditions, the deviation starts to increase. Therefore, the AEDM accuracy becomes questionable. In the previous studies, Kegel (2017) and Le Lostec (2014) pointed out that the heat pump performance only relies on the manufacturer's performance curve and sometimes not reliable, typically under low ambient temperature. Hence, this study investigated the effect of ambient temperature on the condensing temperature, evaporating temperature, and suction superheat. A new comprehensive performance map is proposed to improve the model accuracy quantitatively in terms of mass flow rate, power consumption, and volumetric efficiency. Then the improved compressor performance map is applied to the AEDM to see the accuracy improvement after the model enhancement.

2. Test facility

2.1 Test facility design

The compressor test stand was developed based on a hot gas bypass test stand similar to Vetsch et al.'s work (2016). The main specifications are as follows. The capacity of compressor testing ranges from 0.58 kW to 7.3 kW. The condensing temperature ranges from 25 to 80 °C, and the evaporating temperature ranges from -20 to 15 °C. The isentropic and volumetric efficiencies are measured with an uncertainty below 3%. Figure 1 shows the schematic of the compressor performance test stand. The mass flow meter measuring the total refrigerant flow was placed right after the discharge port. After the mass flow rate meter, the electronic expansion valve (EEV), together with parallel needle valves, control the high side condensing pressure. To achieve more precise control of high side pressure, 0.37 cv needle valves and 0.09 cv needle valves were used parallel to the target pressure value. The electronic expansion valve (EEV) was used for fine-tuning by the PID control. After the first expansion section, the refrigerant flow splits into two paths. One directly bypasses the mass flow meter and enters the second expansion section, and the other passes through the heat exchanger and a receiver, which collects the excess refrigerant in the system, and a sub-cooler to ensure it is in the liquid phase. The heat exchanger provides the cooling capacity by condensing the vapor refrigerant and stabilizes the intermittent pressure. Two RTDs and pressure transducers were placed before the heat exchanger and after the sub-cooler to monitor the pressure difference and temperature change. To better understand the mass flow split percentage between two paths, the mass flow meter before the EEV2 section plays a role in providing the system's refrigerant flow information. EEV2 section and EEV3 section serve jointly for a low side pressure control in the system. The PID control of EEV2 aims at controlling the suction pressure, and the EEV3 controls the suction superheat. Some auxiliary components help to improve the smooth operating of the system. The sight glass helps to check if moisture was in the system. Two shredder valves help vacuum the system and charge or discharge the refrigerant. Figure 2 shows the picture of the entire setup after the construction. The entire test facility was installed inside the environmental chamber, providing temperatures from 10 °C ~ 40 °C. To expand the testing temperature range, we installed a local cooling device to supply a stable air temperature. The target airflow rate was the same as the measured one in the actual air conditioning system. The compressor was installed in a 0.3 m x 0.3 m x 0.3 m casing and connected with the 0.3 m diameter insulated flexible duct to minimize the heat loss. An R-134a vapor compression system controlled the air inlet temperature with a 1,000 W cooling capacity. A 600W heater assisted in stabilizing the air temperature through the heater's PID control. The air inlet temperature was maintained within 0.3 °C variations. Two thermocouple sensors monitored the compressor's temperature. It should be noted that the air velocity was lower than 0.2 m/s in this study.
2.2 Instruments and uncertainties

Table 1 lists the instrument used in our compressor test facility. The compressor discharge and suction temperatures were measured by using the RTD sensor. In addition, T-type thermocouple sensors were placed over the entire system to monitor the system's overall system behavior, including the intermittent system temperature, compressor surface temperature, and local environment temperature. The pressure transducers were in 0 ~ 3,447 kPa measurement range, except the discharge one was 0 ~ 6,894 kPa for the sake of safety. There were two mass flow meters used. One was for the total mass flow rate, located right after the compressor discharge site, and the other was for measuring the hot gas bypass mass percentage. Both of the mass flow meters had 0.11% reading accuracy. For compressor's power consumption, the wattmeter with 0.2% reading accuracy was adopted. All the test instruments were calibrated before installation to the test facility. The uncertainty analysis includes the systematic error as shown in Table 1 and the random error. Before acquiring the test data, the system was operated for one or two hours to reach the steady state and then data was recorded every second for at least half an hour. The random error was then evaluated to calculate the standard deviation of the sample mean. The absolute uncertainty was then calculated by the root sum of squares (RSS) of the systematic error and random error (Mandel, 2012).
Table 1: Instruments and Uncertainties

| Instrument          | Manufacturer / Model | Range                  | Systematic Error                      |
|---------------------|----------------------|------------------------|---------------------------------------|
| RTD                 | Omega 1/10 DIN       | -100 °C ~ 400 °C       | ± 1/10 * (0.3 + 0.005 * t) °C         |
| Thermocouple        | Omega T Type         | -200 °C ~ 200 °C       | ± 0.5 °C                              |
| Pressure transducer | Setra 280E           | 0 ~ 3,447 kPa          | ± 0.11% FS                            |
| Pressure transducer | Setra 280E           | 0 ~ 6,895 kPa          | ± 0.11% FS                            |
| Mass flow meter     | Micro Motion 2700+CMFS025M | 0 ~ 310 g/s             | ± 0.35% of reading                     |
| Watt meter          | Ohio Semitronics GH-020D | 0 ~ 4,000 W             | ± 0.2% of reading                      |
| Power supply        | Preen AFV-P-2500     | Voltage 0~310 V         | ± 0.07% FS                            |
|                     |                      | Frequency 50/60         |                                       |

3. Test matrix

3.1 Heating mode

3.1.1 Heating mode based on experimental data

The condensing temperature, evaporating temperature, ambient dry-bulb, wet-bulb temperature, and suction superheat were used as inputs to the VapCyc to calculate the high-side and low-side pressures of the heat pump system operating under realistic conditions. From the AHRI 210/240 standard (2017), we know the humidity ratio of the ambient air for four test conditions. Then we took a second-order polynomial curve fitting to predict each condition’s wet-bulb temperature, as shown in Figure 4. A correction factor 1.1 was necessary for the VapCyc results to match the experimental laboratory data (Alabdulkarem, 2013). Based on the VapCyc simulation results, the relation between ambient temperature and the evaporating temperature was depicted as a yellow line in Figure 4. Then the red points are chosen as the test matrix for the experiment. The entire test matrix is shown in Table 2 with varied superheat for each test condition.

3.1.2 Heating mode based on fixed superheat

In the previous section, the heat pump unit’s superheat values were assumed to be the same as the previous experimental work by Alabdulkarem et al. (2013). However, we do not know the exact superheat value of the current system. Therefore, the 11K superheat was used in the following test matrix to compare with the original compressor performance map, which complies with the AHRI standard 540. The entire test matrix for the fixed superheat is shown in Table 3.
Figure 4: Temperature difference between ambient and evaporating temperature and ambient humidity ratio

Table 2: Heating mode test conditions with experimental superheat

| Test# | Evaporating Temperature (°C) | Condensing Temperature (°C) | Superheat (K) | Outdoor Air Temperature(°C) |
|-------|-----------------------------|-----------------------------|---------------|-----------------------------|
| 1     | 6.4                         | 36.2                        | 10.2          | 16.7                        |
| 2     | 4.9                         | 35.9                        | 8.2           | 14.0                        |
| 3     | 3.0                         | 35.0                        | 6.3           | 11.0                        |
| 4     | 0.8                         | 34.5                        | 4.7           | 8.3                         |
| 5     | -1.5                        | 33.5                        | 3.3           | 5.0                         |
| 6     | -4.0                        | 33.5                        | 2.1           | 1.7                         |
| 7     | -6.5                        | 32.0                        | 1.0           | -1.0                        |
| 8     | -9.2                        | 31.1                        | 0.5           | -4.0                        |
| 9     | -11.8                       | 30.2                        | 0.5           | -7.0                        |
| 10    | -12.6                       | 30.0                        | 0.5           | -8.3                        |
| 11    | -14.1                       | 29.4                        | 0.5           | -10.0                       |
| 12    | -15.7                       | 28.8                        | 0.5           | -12.0                       |

Table 3: Heating mode test conditions at fixed superheat

| Test# | Evaporating Temperature (°C) | Condensing Temperature (°C) | Superheat (K) | Outdoor Air Temperature(°C) |
|-------|-----------------------------|-----------------------------|---------------|-----------------------------|
| 1     | 5.6                         | 36.8                        | 11.0          | 16.7                        |
| 2     | 2.8                         | 35.5                        | 11.0          | 14.0                        |
| 3     | -0.1                        | 34.6                        | 11.0          | 11.0                        |
| 4     | -3.1                        | 33.6                        | 11.0          | 8.3                         |
| 5     | -6.0                        | 32.6                        | 11.0          | 5.0                         |
| 6     | -9.0                        | 31.6                        | 11.0          | 1.7                         |
| 7     | -12.0                       | 30.5                        | 11.0          | -1.0                        |
| 8     | -15.0                       | 29.4                        | 11.0          | -4.0                        |
| 9     | -18.0                       | 28.1                        | 11.0          | -7.0                        |
| 10    | -18.5                       | 28.1                        | 11.0          | -8.3                        |
3.2 Cooling mode

3.2.1 Cooling mode based on experimental data

For the cooling mode conditions, the superheat from experiment data by Alabdulkarem et al. (2013) was considered again here, and the correction factor 1.05 was applied to fit the experimental data. The yellow line in Figure 5 represents the VapCyc simulation results. Also, the AHRI standard 210/240 provides us information on the ambient air humidity ratio. Therefore, the red circle points in Figure 6 were chosen as test points. Figure 5 shows that the simulation results deviate from the experiment data by Alabdulkarem et al. (2013) because the experimental data was based on a split unitary system while the current study was for the packaged terminal heat pump unit. Otherwise, the trends of both systems are similar. Furthermore, the superheat values were consistent at 2 K while the ambient temperature varied.

![Figure 5: Temperature difference between the condensing and ambient temperature, and ambient humidity ratio](image)

### Table 4: Cooling mode test conditions with experimental and fixed superheat

| Test # | Evaporating Temperature (°C) | Condensing Temperature (°C) | Superheat (Experimental / Fixed) (K) | Outdoor Air Temperature(°C) |
|--------|-----------------------------|-----------------------------|-------------------------------------|-----------------------------|
| 1      | 13.3                        | 58.1                        | 2 / 11                              | 46.0                        |
| 2      | 13.1                        | 55.5                        | 2 / 11                              | 43.0                        |
| 3      | 12.8                        | 52.9                        | 2 / 11                              | 40.0                        |
| 4      | 12.6                        | 50.2                        | 2 / 11                              | 37.0                        |
| 5      | 12.3                        | 48.4                        | 2 / 11                              | 35.0                        |
| 6      | 12.2                        | 46.6                        | 2 / 11                              | 33.0                        |
| 7      | 12.0                        | 44.8                        | 2 / 11                              | 30.6                        |
| 8      | 11.8                        | 42.0                        | 2 / 11                              | 27.8                        |
| 9      | 11.6                        | 40.2                        | 2 / 11                              | 26.0                        |

3.2.2 Cooling mode with fixed superheat

In the cooling mode, we considered the superheat as an additional test variable, while the original performance map defined the 11 K of superheat by default. Table 4 denotes the cooling mode test conditions with two superheat values (2 K and 11 K). The test matrix for experimental data superheat was combined with the fixed superheat conditions in Table 4.
4. Test results

4.1 Heating mode results

4.1.1 Heating mode results with experimental superheat data

In Table 2, we selected 12 data from the previous VapCyc simulation results. Figure 6a shows the comparison of experiment mass flow rate results and compressor performance map output. As the ambient temperature decreases, the relative deviation starts to increase from 8.5% to 40%, suggesting that the performance prediction against lower ambient temperature underestimates the actual heating capacity. From Figure 6b, the power consumption's relative deviation increases from 5% to 9.8%, and it overestimated the actual power consumption under low ambient temperature conditions. Figure 7 shows that both isentropic and volumetric efficiencies decrease as the ambient temperature decreases, the isentropic efficiency dropped from 65.7% to 58.2%, and the volumetric efficiency from 90.3% to 85.1%. The larger temperature difference between indoor and ambient temperature causes a larger pressure ratio between the evaporator and the condenser. Therefore, the isentropic efficiency decreases when it needs more power to overcome the larger pressure ratio, and the volumetric efficiency decreases due to increased internal leakage and a lower suction density.

![Figure 6: Parity plot of mass flow rate and power consumption for cooling mode and heating mode](image1)

![Figure 7: Isentropic efficiency and volumetric efficiency for cooling mode and heating mode](image2)
4.1.2 Heating mode results with fixed superheat

The relative deviation in fixed superheat testing results shows an even more significant value than experimental superheat data, 62.5% in mass flow rate and 11% in power consumption. The results reconfirm that the current compressor performance map did not cover the low ambient temperature conditions adequately. Moreover, most of the existing maps have a low prediction accuracy under low ambient temperature heating mode because the training data points did not include the low ambient temperature conditions. This study shows that if extrapolating the original map, the deviation could be tremendously significant and not appropriate to predict the performance in such region.

4.2 Cooling mode results

The mass flow rate with the experimental superheat is more consistent toward the performance map, which is within 2.2%. On the other hand, the fixed superheat case is within 5%, and both the isentropic and volumetric efficiencies are higher in the fixed superheat case. The power consumption has a similar accuracy, in the experimental superheat case was within 4.2%, and in the fixed superheat case was within 3.2%. Either the mass flow rate or power consumption is following the performance map values closely, indicating that the current performance map is suitable for the cooling mode conditions but not for heating mode at the low ambient temperature. The isentropic and volumetric efficiencies have a similar trend as the heat pumping mode. The temperature difference between the indoor and ambient temperature decreases means a reduced pressure ratio. As the pressure ratio decreases, the isentropic and volumetric efficiency increases by around 3% and 8%, respectively.

4.3 Repeatability tests results

So far, the characteristics of the compressor have been tested and described in the previous section. Repeatability tests were conducted with two more compressors if there is any difference between the same model products. Three test conditions were selected from Table 2. Results show that the mass flow rate deviation was within 2%, and the power consumption deviation was within 1%. Therefore, the first compressor results were taken to develop the new map in the next section.

5. Generated 10-coefficient model and validation

5.1 Combined performance map

With the fixed superheat in cooling mode and heating mode, the 19 points data can generate a high-quality, comprehensive performance map applicable to the VapCyc software. These 19 data points, including mass flow rate, power consumption, current, and capacity, were used to generate the third-order polynomial equation by Matlab regression tool. The experiment-based performance map was then applied to the VapCyc to see the deviation between simulation results and experiment results. The comparisons are shown in Figure 6a and Figure 6b. As shown, the relative deviations of the new model were improved from 62.5% to 0.7% for the mass flow rate and 13.1% to 0.4% for the power consumption in heating mode. The mass flow rate in cooling mode was improved from 1.2% to 1% and 3.2% to 0.2% in power consumption. The newly generated map can accurately predict the compressor performance under wide ambient temperature, -20°C to 50°C. The third-order polynomial equation, which comes from standard AHRI 540, is described below, and the coefficient of the manufacturer's map and newly generated map are list in Table 5.

\[ X = C_1 + C_2 \cdot t_s + C_3 \cdot t_D + C_4 \cdot t_s^2 + C_5 \cdot (t_s \cdot t_D) + C_6 \cdot t_D^2 + C_7 \cdot t_s^3 + C_8 \cdot (t_s^2 \cdot t_D) + C_9 \cdot (t_s \cdot t_D^2) + C_{10} \cdot t_D^3 \]

Where:
- \( X \) = Power Input, or Refrigerant Mass Flow Rate
- \( t_D \) = Discharge Dew Point Temperature, °C
- \( t_s \) = Suction Dew Point Temperature, °C

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Table 5: Coefficients of Manufacturer's Map and Newly Generated Map

| Coefficients | Manufacturer's Power Map (W) | Manufacturer's Mass Flow Rate Map (g/s) | New Generated Power Map (W) | New Generated Mass Flow Rate Map (g/s) |
|--------------|------------------------------|----------------------------------------|-----------------------------|----------------------------------------|
| C₁           | 5.2021E+02                   | 1.8187E+01                            | 5.2020E+02                  | 1.4850E+02                            |
| C₂           | 5.3703E-01                   | 2.5048E-01                            | -1.1830E+01                 | 3.5180E+00                            |
| C₃           | -3.9258E+00                  | -3.2905E-01                           | -3.5960E+00                 | -5.6360E+00                           |
| C₄           | 1.2128E-01                   | -1.8153E-03                           | 2.1350E-02                  | 1.6610E-01                            |
| C₅           | -1.2666E-01                  | 3.9346E-04                            | 2.0930E-01                  | -1.2530E-01                           |
| C₆           | 7.2080E-02                   | 2.4886E-03                            | 2.7810E-02                  | 8.6030E-02                            |
| C₇           | -3.5722E-04                  | 3.1050E-05                            | 7.1760E-05                  | 6.4650E-04                            |
| C₈           | -9.4307E-04                  | 2.9808E-06                            | -7.4150E-04                 | -2.1050E-03                           |
| C₉           | 1.1574E-03                   | -4.7693E-06                           | -6.7700E-04                 | 1.1140E-03                            |
| C₁₀          | -2.6292E-04                  | -6.3590E-06                           | 1.2330E-04                  | -3.8550E-04                           |

5.2 System performance comparison using VapCyc

The new performance map has been generated and compared with the old one, as shown in Table 6. Under the same superheat, the new performance map could approach the experimental data closer without using any correction factor. The correction factors of 10% and 5% were needed when using the old map for heating and cooling capacity. However, the superheat value was found by a try-and-error approach to match the capacity near the experimental value. Therefore, knowing the right superheat value for the system would be critical for the AEDM. In this study, the compressor performance map generated encompassing the low ambient temperature conditions would deliver much better prediction accuracy in application.

Table 6: Results for performance prediction

| Mode    | Experimental Capacity [W] | Superheat [K] | Capacity, MFR, and Power by Old Map [W] / [g/s] / [W] | Capacity, MFR, and Power by Old Map with Correction Factor [W] / [g/s] / [W] | Capacity, MFR, and Power by New Map [W] / [g/s] / [W] |
|---------|---------------------------|---------------|------------------------------------------------------|-----------------------------------------------------------------------------|---------------------------------------------------|
| Heating | 2,468                     | 12.3          | 2,324 / 9.7 / 548                                     | 2,484 (1.1) / 10.5 / 554                                                   | 2,476 / 10.5 / 551                                 |
| Cooling | 2,852                     | 0.1           | 2,761 / 18.5 / 701                                    | 2,855 (1.05) / 19.2 / 711                                                  | 2,819 / 18.7 / 700                                 |

6. Conclusions

The heat pump system has become the most critical heating method for the world to achieve the carbon-neutrality. However, the critical component, the compressor, has an unsatisfied prediction capability under low ambient temperature when the conventional compressor performance map approach is used. To better understand the compressor's realistic performance, a compressor test stand has been built, validated, and assembled with the local cooling device. The test capacity ranges from 580 W to 7,300 W. The compressor ambient temperature ranges from -20 °C to 60 °C. From the test results, we confirmed that the original compressor performance map has inaccurate results as compared to the experimental data when the ambient compressor temperature is low. In other words, the prediction of the heating performance using the system...
simulation tool with conventional compressor performance map would only generate a biased result as compared to the system test data. This study has constructed a new comprehensive performance model covering a wide range of operating conditions. The newly generated performance map improves the relative error of mass flow rate prediction compared to the experimental data, from 62.5% to 0.7%. On the other hand, the superheat in the system would be critical to accurately predicting the capacity in the system model simulation. Moreover, the difference in mass flow rate between experimental superheat and fixed superheat can vary from 3.3% to 6%.

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