Baseline testing of a variable-speed water-cooling Chiller according AHRI standard 550/590

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Abstract. A new experimental test facility was designed to test vapor compression chillers of up to 145.9 refrigeration tons (approximately 513 kW). To implement such a large-capacity test facility in a university laboratory research environment, the system design is based on regenerative shell and tube heat exchangers. The capacity load on the chiller can be controlled via a bypass valve that allows a flow change over the internal heat exchangers. In addition to the internal sensors of the chiller, a variety of sensors in the system ensures a detailed recording of data as well as an automation of the test procedure.

The test facility is constructed and commissioned to execute the performance according the AHRI Standard 550/590. The chiller has been tested in full load and part load rating conditions to achieve a characteristic performance line of the machine. Recorded data is averaged and evaluated according to the requirements of the AHRI standard. The results of the baseline testing and the data processing are presented in this paper. A detailed analysis of the outcomes allows a comparison with the manufacturer rated efficiency values. Finally, the potential instabilities of the test systems that caused problems during the test runs and their reasons are discussed, as well as possibilities to overcome them. The given ability of the test facility for a rapid change between different load conditions allows the arrangement of test cycles to investigate the performance degradation of the twin-screw chiller over time.

1. Introduction
Over the lifespan of a chiller, performance degradation and various faults can occur. Automated fault detection and diagnosis (FDD) approaches (either model-based or data-driven) can help monitoring the health state of the chillers and predict or detect anomalies in the chiller operation. Data-driven modelling relies on historical data of the equipment and can be quite powerful in identifying anomalies in the operation patterns of chillers. In the literature, only a limited number of studies deals with time-dependent performance analysis of chillers and prediction of performance degradation. For instance, Ding and Fu [1] analysed the impact of the ambient temperature on the capacity of the screw chiller as well as the different ways of adjusting the control system to maintain optimal operating conditions. For this a chiller with constant water inlet temperatures on the evaporator inlet was operated at different...
ambient temperatures between -10 °C and 40 °C. Based on the collected data, a model-predictive control strategy was developed to run the chiller over a wide ambient temperature range. Another modeling approach was developed by Tiang et al. [2] to provide a transient coefficient of performance prediction model. More specifically, an artificial neural network model using Levenberg-Marquardt-algorithm was developed and validated with daily operating data from a 1043.7 kW screw compressor chiller. The model is used to further analyze the chiller performance under various operating conditions. Sala-Cardoso et al. [3] described an approach to predict load ranges and usual capacity trends based on previous chiller performance maps and data-driven multi-criteria orchestration strategies with the aim to operate the chiller based on load predictions in an energy effective mode by using a multi-layer pre-training strategy, a load forecasting model for the refrigeration systems. The chillers for the validation had a thermal capacity of between 130 and 430 kW. Recently, Bao et al. [4] adopted the integrated part load value (IPLV) for chillers operating in the Hong Kong area in terms of climate and temperature conditions. Data from 20 office buildings was used to develop a uniform building cooling load profile using the Monte Carlo analysis and the ASHRAE bin method. With this profile the weighting factors were optimized to the local conditions of Hong Kong and validated with the building data.

A performance mapping and related model of a screw compressor chiller is not available yet in a way that it can be used for life-cycle control and maintenance predictions. To be able to develop an accelerated life cycle testing of chillers, an accurate dataset at rated conditions is needed as a reference. As next step, a holistic system monitoring with multiple sensors is needed to map the operating conditions of the test facility and the equipment. It is therefore even more important that the measurements are carried out professionally and properly according to the specifications published in the AHRI 550/590 standard [5].

In this paper, the new experimental test facility is employed to obtain the baseline performance data of a 145.9 refrigeration tons variable-speed screw chiller, and the challenges encountered during the experimental work are discussed in detail.

2. System setup and commissioning

Before the system can be tested it needs to be set up and commissioned. The existing system was based on a 316.5 kW-chiller unit with turbo compressor. The following paragraphs include a description of the new chiller unit as well as the system set up and its control possibilities.

2.1. Chiller

The observed test unit is a water-cooled chiller with a R-134a vapor compression cycle. Core component of the chiller is a semi-hermetic screw compressor that is controlled by a variable speed drive. Due to its internal control logic, two operation modes with different maximum capacities can be chosen. The chilled water mode enables a maximum cooling capacity of 513.1 kW according to the specification at a supply temperature of 12.2 °C and a return temperature of 6.7 °C. In the second mode, the so-called ice making mode, 341 kW capacity can be achieved at a temperature level of 0 °C for the supply and -3.9 °C for the return.

The variable speed drive does not allow an external control of the compressor speed. That is why the capacity control during the baseline tests must be realized by a flow and temperature adjustment of the test system. Another factor that must be considered during the test process are the internal functions and thresholds of the chiller system that can affect the test feasibility and the behavior of the system in steady state operation. The speed control unit does not just control the compressor speed dependent on the capacity demand, but also logs 78 important system data and sensor recorded vales time stamped on a memory card. Both heat exchangers of the chiller are refrigerant to water shell-and-tube heat exchangers.
2.2. System and sensors
The test rig was set up to allow a capacity control of the chiller by adjusting the coolant temperatures and the flow over the chiller heat exchangers. To solve this task within an area and with a low need of external heat dissipation and heat supply, internal heat exchangers were used to recover the heat. The following cycle description refers to the PI-diagram in Figure 1. In the flow diagram, all components and sensors of the test stand are displayed. For further simplification, the system is divided into two main cycles. On the left side, the condenser water cycle in red and on the right side the evaporator water cycle in blue.

The actual load control during the test runs is implemented by the condenser water cycle. After the cooling water is brought to the aimed mass flow by the condenser pump unit, the condenser cooling water passes the two serial heat exchangers HX1 and HX2 in this shell-and-tube heat exchangers the main heat load of the condenser water cycle is dissipated and at the same time it is recovered as load for the evaporator water cycle. A bypass line to the heat exchangers in combination with a three-way valve allows to control the flow over the internal recovery heat exchangers, with this function, the load on the evaporator cycle which is also the load on the chiller can be controlled dependent on the aimed capacity for the test run. After the fluid is cooled down in the internal heat exchangers, it flows back to the chiller at aimed entering temperature conditions. Once the cooling medium gets heated up in the chiller condenser it passes the shell-and-tube heat exchanger HX3. This heat exchanger is connected to the building chilled water cycle displayed in green. HX3 helps to discharge the heat that is brought into the system by the compressor and that cannot be recovered in HX1 and HX2. The capacity of HX3 can be adjusted by a motor valve in the chilled water line that regulates the flow. After the surplus heat is dissipated, the cycle begins new. The condenser water cycle is equipped with five temperature sensors that give feedback of the important temperatures in the systems. A flow meter in the main flow, in combination with a flow regulation valve allows to regulate the mass flow over the heat exchangers, especially the condenser. A separate mass flow meter in the bypass line enables the detailed observation and control of the flow over the internal recovery heat exchangers.

In the evaporator water cycle, the water is cooled down in the chiller first. After leaving the chiller, the coolant flows into the two internal shell-and-tube heat exchangers HX1 and HX2 which are connected in series. In that heat exchanger set, heat is recovered from the higher temperature level in the

Figure 1. PI-diagram of the test stand.
condenser water cycle to simulate load on the evaporator water cycle. Once the heat of the condenser cycle is absorbed by the evaporator cycle the water passes the pump unit which maintains the necessary mass flow. Before entering the chiller again, the evaporator cooling water passes the shell-and-tube heat exchanger HX5. This heat exchanger is connected to an auxiliary cycle (drawn in black) includes a pump and a flow regulation valve and that again is connected to the building heating steam network displayed in gray. This indirect coupling to the steam heating cycle is not necessary for the load simulation on the evaporator but it provides an additional heat source during the start-up procedure of the system and for faster load changes during the test implementation. Furthermore, the evaporator water cycle is equipped with a flow regulation valve that allows to adjust the flow over the heat evaporator based on the measured data of the flow meter. Also, five temperature sensors provide real-time data of the temperature conditions in the cycle.

3. Baseline testing

Purpose of the baseline testing is to gather reference data for future performance degradation tests. This reference includes the capacities of the chiller and the related efficiency during operation. In the baseline test, the characteristic values such as efficiency, capacity, current draw, and the overall system behavior are tested and tracked for different load conditions. With this information, a compressor mapping can be conducted. Base for this mapping is for example the values of temperature, pressure, mass flow that are logged either by the chiller system itself or by the sensors in the test stand.

3.1. Test implementation

The implementation of the baseline test targets test runs at the three part-load steps 25%, 50%, and 75% of the maximum chiller cooling capacity as well as in full load operation. If the three part-load steps cannot be achieved exactly, AHRI 550/590 allows interpolation and extrapolation as explained in Section 3.2. This is specifically important because the speed control of the chiller has an automatic shut down for cooling loads of lower than 30% of maximum that cannot be set down further which makes it impossible to collect test data at 25% load.

A valid test run must have a minimum length of 15 minutes in steady state where the mass flow and the temperatures remain constant in the system. During this time, all operational data points are logged with a related time stamp from the test software on a test protocol or from the chiller on an internal memory card of the variable speed drive. The data must be brought together in one file manually for the test evaluation after the runs. All part-load test runs are executed with a fixed setpoint of 6.7 °C for the evaporator water leaving temperature. The evaporator entering water temperature is load dependent and calculated for a pump volume flow of 15.8 kg/s. This flow rate is given by the given fix speed water pump for the evaporator cycle. Since this flow rate is not sufficient to reach 100% load at the test conditions, the full load operation is aimed at lower temperatures. This is not standard conform but the measured data should at least give an idea of the full load efficiency in comparison to the technical data in the specification for this lower temperature range. The condenser entering water temperature is staggered in equal steps so that a sufficient heat dissipation is ensured. Table 1 shows the aimed temperature dependent on the target loads for the part load and the full load testing. All temperatures must be controlled within an allowed deviation of ±1.1 K.

| Target load (%) | Target capacity (kW) | Setpoint temperature (°C) | Evaporator inlet temperature (°C) | Condenser inlet temperatures (°C) |
|-----------------|----------------------|----------------------------|----------------------------------|----------------------------------|
| 30              | 127                  | 6.7                        | 8.3                              | 18.3                             |
| 50              | 257                  | 6.7                        | 10.6                             | 18.3                             |
| 75              | 383                  | 6.7                        | 12.6                             | 23.9                             |
| 100             | 440                  | 2.2                        | 8.8                              | 26.7                             |

* calculated based on a flow rate of 13.8 kg/s
3.2. Data analysis

An important task related to the baseline testing is the data analysis. A basic data set is evaluated in the same way for all test runs in steady state. As a preparation for this analysis, the time-dependent data that was recorded during the test run must be converted into a csv file and gets reviewed. This time related data includes time-stamped values of all important temperatures, mass flows, pressures, and the power consumption during the test runs. Time periods of this raw data that indicate a steady-state behavior and fulfill the standard requirements get highlighted and if they are at least 15 minutes long, they get extracted into separate files. Then, average values are calculated over the steady state time which are used for the actual evaluation and further computation. It is the aim for a proper analysis according AHRI 550/590 to cover not just the full load behavior during the test runs but also part loads. For the part load conditions the target part loads are 25%, 50%, and 75% of the full load.

At first, the cooling capacity is calculated as a core value of the whole analysis. In equation (1) it is shown how the cooling capacity $Q_E$ is a function of the mass flow over the evaporator $m_E$, the specific heat $c_p$ of the coolant, and the temperature difference between inlet $T_{E,i}$ and outlet $T_{E,o}$ of the evaporator. The cooling capacity allows the exact classification of the test section and the validation of the relative chiller load which is calculated in equation (2) as the ratio of the calculated cooling capacity over the maximum cooling capacity $Q_{E,max}$ that is specified as 513.1 kW for the chiller, times the factor of 100.

$$Q_E = m_E \cdot c_p \cdot (T_{E,i} - T_{E,o})$$ (1)

$$\text{Load} = \frac{Q_E}{Q_{E,max}} \cdot 100\%$$ (2)

Based on the capacity provided from equation (2) and the power consumption of the chiller that is measured by a power meter, the efficiency of the tested device can be calculated in two ways. The more common efficiency metric is the cooling coefficient of performance $\text{COP}_E$. As equation (3) displays, the $\text{COP}_E$ is defined as the ratio of the cooling capacity in kW over the electrical power draw of the chiller $W_{el}$ in kW.

$$\text{COP} = \frac{Q_E}{W_{el}}$$ (3)

The AHRI 550/590 standard defines an additional efficiency metric. This efficiency for is calculated as the ratio of the power consumption of the chiller in kW over the cooling capacity $Q_{ev}$ in TR. Therefore, it is not the inverse value of the $\text{COP}_E$ as it may be assumed from simply looking at the equation (4).

$$\text{Efficiency} = \frac{P_{el}}{Q_{ev}}$$ (4)

By using the efficiency according to the AHRI 550/590 standard [5], an integrated part load value $\text{IPLV.IP}$ can be computed. The IPLV.IP is a value to express the part load efficiency in all conditions by combining the measured efficiencies as in equation (5), where A is the efficiency value for 100% load, B for 75%, C for 50% and D for 25%.

$$\text{IPLV.IP} = \frac{1}{A^{0.01} + 0.99 \cdot B^{0.02} + 0.98 \cdot C^{0.05} + 0.12 \cdot D}$$ (5)

For the case, that the target part load steps cannot be achieved exactly in a steady state condition it is possible to interpolate linear between the lowest and the highest measured point. If the 25% load cannot be achieved, linear extrapolation is not allowed but the AHRI standard provides a correlation. The efficiency value calculated from the test data following equation (4) therefore gets corrected by the
degradation correction factor $C_D$ as in equation (6). The degradation correction factor (equation 7) again needs a load factor $LF$. The load factor calculated in equation (8) is the ratio of the target load ($Load\%$ times the maximum capacity $Q_{\text{max}}$) over the load measured in the test run $Q_{\text{Test}}$.

Efficiency$_{CD} = \text{Efficiency}_\text{Test} \cdot C_D$ \hspace{1cm} (6)

$CD = (-0.13 \cdot LF) + 1.13$ \hspace{1cm} (7)

$LF = \frac{Load\% \cdot Q_{\text{max}}}{Q_{\text{Test}}}$ \hspace{1cm} (8)

3.3. Baseline test results

The baseline test results are a mix of directly measured values and calculated values from the data analyses. All results give a feedback of the operating conditions and characteristics during the test and allow a comparison with specified values from the manufacturer. During the baseline tests, seven test runs were performed within a load range of 34% to 90%. The tests were conducted during steady state. The detailed test conditions and results are displayed in Table 2. Based on these values, the efficiency values for 50% and 75% can be interpolated while the value for 25% can be extrapolated with the method described in paragraph 3.2. A 100% load operation could not be achieved. Potential reasons for this are mentioned in paragraph 3.4.

| Load (%) | Evap. mass flow (kg/s) | Evap. EWT (°C) | Evap. LWT (°C) | Cond. mass flow (kg/s) | Cond. EWT (°C) | Cooling capacity (RT) | Elec. power draw (kW) | COP | Efficiency (kW/RT) |
|----------|------------------------|----------------|----------------|------------------------|----------------|------------------------|------------------------|-----|-------------------|
| 34       | 15.40                  | 9.73           | 7.04           | 18.71                  | 15.95          | 49.70                  | 14.15                  | 12.36 | 0.2846           |
| 42       | 15.41                  | 10.00          | 6.66           | 18.17                  | 18.06          | 61.46                  | 21.31                  | 10.14 | 0.3467           |
| 48       | 15.42                  | 10.56          | 6.72           | 17.68                  | 17.94          | 70.73                  | 26.45                  | 9.40  | 0.3740           |
| 56       | 15.43                  | 11.00          | 6.58           | 17.25                  | 18.41          | 81.54                  | 32.27                  | 8.89  | 0.3958           |
| 74       | 15.45                  | 12.48          | 6.62           | 17.31                  | 23.83          | 108.25                 | 58.74                  | 6.48  | 0.5426           |
| 79       | 15.44                  | 12.74          | 6.52           | 17.00                  | 23.74          | 114.73                 | 63.87                  | 6.32  | 0.5567           |
| 90       | 15.46                  | 14.23          | 7.14           | 18.42                  | 26.87          | 130.94                 | 86.07                  | 5.35  | 0.6573           |

Figure 2 shows a diagram with all efficiency values of the chiller. The small green dots are the values that were analyzed from the measurements conducted in the test runs while the bigger red dots are the interpolated or extrapolated values. Since a 100% measurement could not be implemented, the value for this operation mode was linear extrapolated (gray dot). An extrapolation for full load is not intended in the standard but it was implemented to allow a calculation of the IPLV.IP value. The yellow dot is the efficiency for full load mentioned in the specification. The value for this is much higher and probably more realistic but this could just be proven by measurement. The IPLV.IP calculated from the values is 0.4191 which is 5.3% higher than in the specification. It is important to mention that this does not mean that the specification is wrong but more that the linear extrapolation for the 100% point is not suitable. A 100% measurement must be implemented to proof this. The 5.3% deviation from the IPLV.IP value in the specification shows that the efficiency measurements are overall pretty close to the specified values.
3.4. Difficulties during test implementation

As already stated in Section 3.1, there were some difficulties during the test implementation due to the large capacity of the chiller. The test stand was initially built and used in previous tests for chillers with a maximum cooling capacity of 317 kW. For those test devices, a flow rate of nearly 14 kg/s was always sufficient. However, for the current chiller was not sufficient. According to the AHRI 550/590 standard conditions a flow rate of 22.1 kg/s is needed. The currently installed pump was not able to provide this mass flow. Therefore, higher load conditions could not be achieved under standard rated conditions. Another harmful factor that appeared was measurable at a part-load condition between about 70 and 80% load. In this load range, the variable speed drive adjusted the compressor speed whenever a steady state came up for a few seconds. Due to this control behavior, the VSD experienced a hunting situation where the temperatures developed a fluctuation of ±1 K. An example of this control instability behavior is displayed in Figure 3(a) where the condenser inlet temperature (red), the evaporator inlet temperature (black), and the evaporator outlet temperature (blue) show a consistent rise and fall of the values.
The third and last problem is the heat dissipation over the campus chilled water line. The heat exchanger HX3 (Figure 1) has enough capacity in theory but the effectiveness went down drastically in the practical use. For this, not enough heat could be conducted and therefore the temperatures in the system rose during higher load conditions what did not allow the system to come to a steady state. Figure 3b shows a test run where the temperatures were higher than aimed and could not be decreased. Also, a slight increase of the temperature can be seen in this diagram.

The test system has two internal heat exchangers that recover heat from the system. Since more heat is added to the system than can be recovered an additional heat dissipation is necessary. HX3 dissipates heat from the system via the building chilled water loop. The effectiveness of the heat exchanger HX3 is generally low but decreases sharply at higher load conditions as table 3 shows. Due to this, all temperatures in the system start to raise (Figure 3) so that the target temperatures cannot be achieved anymore. One option to solve this problem would be the installation of an additional heat exchanger in the condenser loop.

| Load (%) | HX3 effectiveness (%) | $\dot{Q}_h,\text{max}$ (kW) | $\dot{Q}_c,\text{max}$ (kW) | $Q$ (kW) |
|----------|-----------------------|------------------|------------------|-------|
| 34       | 36.89                 | 862.01           | 60.74            | 22.41 |
| 42       | 37.54                 | 1062.61          | 69.11            | 25.94 |
| 48       | 30.62                 | 1108.41          | 105.50           | 32.31 |
| 56       | 24.39                 | 1082.34          | 149.50           | 36.46 |
| 74       | 19.54                 | 1755.57          | 335.32           | 65.52 |
| 79       | 17.03                 | 1779.06          | 414.02           | 70.52 |
| 90       | 11.80                 | 2062.72          | 724.45           | 85.50 |

4. Conclusion and further steps

The seven implemented tests in which a steady state measurement was conducted show that the general quality of the measurements is sufficient and that the values are close to the stated values in the specification. Nonetheless, the problems that appeared during the test implementation must be solved to achieve a reliable data base for further tests of the chiller unit. A test stand redesign is necessary for this. Such a rework of the test stand has to include an improvement of the heat dissipation over the campus chilled water system in the first place. This can be realized over an additional shell-and-tube heat exchanger to support the existing heat exchanger HX3 (Figure 1). The test data from the implemented measurements can help to calculate the design data for the new heat exchanger and to size it properly. Furthermore, the improvement of the pump system to achieve the AHRI 550/590 flow rate of 0.043 kg/s per kW chiller capacity can be carried out during the redesign of the rig. For this both, the pump in the evaporator loop and in the condenser loop should be replaced by pumps with higher performance. A higher flow rate could allow a test implementation of all load conditions according AHRI 550/590 standard and with a constant evaporator water outlet temperature.

Once the redesign is done, the baseline test can be reimplemented in all load areas. Based on this data, following test can build up.
5. Nomenclature

| Symbol | Description                                      | Unit          |
|--------|--------------------------------------------------|---------------|
| A      | area                                             | m²            |
| AHRI   | air-conditioning, heating, and refrigeration institute |               |
| C_d    | degradation correction factor                    |               |
| C      | condenser                                        |               |
| COP    | coefficient of performance                       |               |
| E      | entering water temperature                       | °C            |
| E      | evaporator                                       |               |
| HX     | heat exchanger                                   |               |
| IPLV   | integrated part load value                       |               |
| k      | thermal conductivity                             | W/m²*K        |
| LF     | load factor                                      |               |
| LWT    | leaving water temperature                        | °C            |
| MV     | motor valve                                      |               |
| ṁ      | mass flow rate                                   | kg/s          |
| P      | power                                            | kW            |
| Q      | cooling capacity                                 | kW            |
| Q_E    | cooling capacity                                 | RT            |
| RTD    | resistance temperature detector                  |               |
| T      | temperature                                      | °C            |
| TR     | tons of refrigeration                            |               |
| TC     | thermo couple                                    |               |
| TWV    | three-way valve                                  |               |
| VSD    | variable speed drive                             |               |
| ε      | effectiveness                                    | %             |
| c      | cool side                                        |               |
| h      | hot side                                         |               |
| i      | inlet                                            |               |
| max    | maximum                                          |               |
| min    | minimum                                          |               |
| o      | outlet                                           |               |
| test   | evaluated in test conditions                     |               |
| E      | evaporator                                       |               |
| el     | electric                                         |               |

6. References

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