Part load performance of single and two stage compressors – a comparative experimental study in a R410A chiller unit

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Abstract. The rising cooling needs for the residential and commercial air conditioning sectors and the requirement of higher SEER values increased the focus on energy efficient part load performance. Although several studies exist on the topic of capacity modulation, there is no comprehensive study that compares different modulation strategies in the same experimental facility. In a first step, this paper aims to experimentally compare two different capacity controls for scroll compressors (single speed vs two stage compressor) using the same R410A water chiller having a nominal cooling capacity of 8kW. The tests were conducted according to AHRI Standard 551/591 (2018). The Integrated Part Load Value (IPLV.SI) was used to fairly compare all the different capacity modulation strategies. Based on the experimental results, differences on the order of 8% in IPLV.SI are found.

1. Introduction
Almost all HVAC&R systems need to be designed to handle cooling loads below the target or design condition. These load variations are often caused by ambient weather variations, different levels of product loading, or the fact that systems are overdesigned to achieve quick pull-down. The range of load variations can be very substantial, in some cases even down to 30% or less of the design capacity. Different capacity control strategies employ different methods so that they can be implemented into the compressor or outside the compressor, in which case the system needs to provide adequate provisions. The modulation target is the same across the different techniques employed: the (average) refrigerant flow rate across the evaporator is reduced to adjust for the different levels of cooling capacity needed. There are many different methods of how to achieve this goal.

The main shortcoming of any literature-based comparison to the best of our knowledge is the fact that different data sets were obtained with different systems using different refrigerants, evaporation and condensation temperatures, secondary flow rates, and different levels of system controls. It is therefore nearly impossible to derive meaningful comparisons between the different studies when it comes to efficiency.

As a first step, this paper aims to experimentally compare two capacity modulation strategies, a single speed scroll compressor and a two stage scroll compressor using the same R410A WEG chiller system. These compressors when used in the experimental setup provide comparable nominal cooling capacity. The cooling capacity of a compressor is proportional to the mass flow rate provided by the compressor. The tests are conducted according to AHRI Standard 551/591 (2018) and the comparison is done using a single figure of merit called IPLV.SI (Integrated Part Load Value) [1].

2. Modulation strategies being considered
The following are the different compressor modulation strategies being considered as part of this study. As a first step, this paper compares a single speed and two stage compressor. These compressors are scroll and can provide similar cooling capacity and belong to the same generation.

2.1. Single speed compressor
A single speed compressor modulates capacity by cycling i.e. turning on and off. The on/off capacity control is the simplest method of adjusting the predetermined temperature (setpoint) using a thermostat. After the temperature reaches the setpoint, the thermostat stops the compressor and circulating refrigerant in the cycle. Since the secondary fluid continues to circulate, the temperature of the water or air gradually raises. When the thermostat detects, this rise turns the compressor on [2].

2.2. Two stage compressor
This is a commercially available two stage compressor, marketed as two-stage Copeland ultratech compressor. It is different from conventional two stage compressors that use a low-pressure stage and a high-pressure stage. In this compressor, two stage refers to its ability to operate at two stages (100% and 67%). Instead, it has two internal bypass ports which enable the system to run at 67%-part load capacity. If the compressor is operated at the same condenser and evaporator water temperature, then the capacity at the 67% stage is approximately 67% of the capacity at the 100% stage. This compressor can be operated either at a high stage (provides 100% capacity) or a low stage (provides 67% capacity). The capacity is reduced by bypassing a portion of the gas in the scroll back to suction. These bypass ports are opened or closed using a solenoid valve present in the scroll chamber [3].

3. Details of the experimental setup
Figure 1 shows the schematic of R410A Water Ethylene Glycol (WEG) chiller. 20% ethylene glycol in water (by volume) is used as the secondary fluid. Two closed WEG loops are connected to the evaporator and condenser. Variable speed pumps and electric heaters are used to control the WEG flow rate and the inlet temperature of the condenser and the outlet temperature of the evaporator. An additional heat exchanger with chilled water flowing through is included in the condenser WEG loop to reject the heat from the condenser WEG loop. All the heat exchangers used in the facility are brazed plate heat exchangers. A 0.9 L receiver is connected between the condenser and the subcooler. Superheat is controlled by an Electronic Expansion Valve (EEV) while the subcooling is a function of the charge. The geometric dimensions of the evaporator, condenser, and subcooler can be found in Table 1.

Type-T thermocouples, absolute and differential pressure transducers, and Coriolis-type mass flow meters are used to obtain the refrigerant side measurements while type-T thermocouples, differential pressure transducers, and Coriolis-type mass flow meters are used to obtain WEG measurements. Data is collected at steady-state conditions at 5s intervals for 20 consecutive minutes, and the data is averaged over the collection period.

The uncertainties of the calculated properties associated with the instrument uncertainties of measured properties are calculated by uncertainty propagation as shown in Equation (1):

$$U_Y = \sqrt{\sum_i \left( \frac{\partial Y}{\partial X_i} \right)^2 U_{X_i}^2}$$

where $U_Y$ is the uncertainty of calculated property and $U_{X_i}$ is the uncertainty of measured property. The uncertainty of the other sensors used in the experimental facility is presented in Table 2.
Figure 1: Schematic of the R410A experimental facility

Table 1: Dimensions of the brazed plate heat exchangers

| Heat exchanger | Length (mm) | Width (mm) | Number of plates |
|----------------|-------------|------------|------------------|
| Evaporator     | 311         | 111        | 28               |
| Condenser      | 311         | 111        | 14               |
| Subcooler      | 207         | 77         | 14               |

Table 2: Summary of measured and calculated property uncertainties

| Instrument      | Thermocouple (°C) | Mass flow meter (g/s) | Wattmeter (kW) | Capacity (kW) | COP (−)       |
|-----------------|--------------------|------------------------|----------------|---------------|---------------|
| Uncertainty     | ±0.1°C             | ±0.2%                  | ±0.5%          | ±2.8%         | ±2.9%         |

Capacity is calculated on the refrigerant side and WEG side. For the WEG side, mass flow rate, temperature, and specific heat are used to calculate capacity as shown in Equation (2). For the refrigerant side, temperature and pressure are used to calculate the enthalpy which is then used with the mass flow rate to calculate the capacity as shown in Equation (3). The capacity reported is the average of the refrigerant side and WEG side capacity given by Equation (4). The difference between the two capacities is indicated by the error given by Equation (5). AHRI standard requires only the capacity measured from
WEG side, however in the current calculations, capacity is average of the refrigerant and WEG side. Equation (5) is a way to indicate the difference between these two energy balances. This error is always less than 3% for the part load rating tests. Power consumed by the compressor is measured using a Wattmeter. The ratio of the average capacity and power consumed by the compressor is used to calculate the COP_{test} as shown in Equation (6).

\[
\dot{Q}_{ev,WEG} = \dot{m}_{WEG} C_p \Delta T
\]

\[
\dot{Q}_{ev,ref} = \dot{m}_{ref} \Delta h
\]

\[
\dot{Q}_{ev,avg} = \frac{(\dot{Q}_{ev,WEG} + \dot{Q}_{ev,ref})}{2}
\]

\[
\varepsilon_{Q_{ev}} = \left(\frac{\dot{Q}_{ev,avg} - \dot{Q}_{ev,WEG}}{\dot{Q}_{ev,avg}}\right) \cdot 100
\]

\[
\text{COP}_{test} = \frac{\dot{Q}_{ev,avg}}{\dot{W}_{cp}}
\]

4. AHRI 551/591 (2018) standard

AHRI 551/591 is used for the determination of the part-load performance of water chillers. The standard defines a single number part-load efficiency figure of merit called Integrated Part Load Value (IPLV.SI) calculated at part load rating conditions. These part load rating conditions are shown below in Table 3. IPLV.SI is the weighted average of the COP_R measured at these standard rating conditions as shown in Equation (7). These factors in Equation (7) are based on the weighted average of the most common building types and operations using average weather in 29 U.S. cities.

\[
\text{IPLV.SI} = 0.01 \cdot A + 0.42 \cdot B + 0.45 \cdot C + 0.12 \cdot D
\]

\[
A = \text{COP}_R \text{ at 100%}
\]

\[
B = \text{COP}_R \text{ at 75%}
\]

\[
C = \text{COP}_R \text{ at 50%}
\]

\[
D = \text{COP}_R \text{ at 25%}
\]

If a system with a compressor cannot be unloaded to 25%, 50%, or 75% load point, then the compressor is run at the minimum step of unloading at the condenser entering water shown in Table 3 for 25%, 50%, or 75% capacity points as required. Once the COP_{test} is calculated at these conditions using Equation (6), it is degraded to COP_R using the Equation (8), (9), (10), and (11). The standard does not explicitly state the effects included in the cycling losses (indicated by C_D). Refrigerant migration during OFF cycle, pressure equalization and thermal mass of the components are major causes of cycling losses [4].

\[
\text{COP}_R = \frac{\text{COP}_{test}}{C_D}
\]

\[
C_D = (-0.13 \cdot LF) + 1.13
\]

\[
LF = \frac{(\% \text{Load}) (\dot{Q}_{ev 100\%})}{\dot{Q}_{ev \text{min} \% \text{Load}}}
\]
\[ \% \text{Load} = \frac{\text{(Part load net capacity)}}{\text{(Full load rated net capacity)}} \]  

(11)

Table 3: AHRI 551/591 part load conditions for IPLV.SI

| Condition | Part load ratio (%) | Condenser Inlet/Outlet (°C) | Evaporator Inlet/Outlet (°C) |
|-----------|---------------------|-----------------------------|-----------------------------|
| A         | 100                 | 30/35                       | 12/7                        |
| B         | 75                  | 24.5/*                      | */7                         |
| C         | 50                  | 19/*                        | */7                         |
| D         | 25                  | 19/*                        | */7                         |

Table 3 shows the part load conditions. As seen in the table, condenser inlet, outlet, and evaporator inlet, outlet are mentioned for A condition while for the B, C, and D conditions, only the condenser inlet and evaporator outlet temperature are mentioned. Standard required that the condenser and evaporator WEG flow rate used for the A condition be used for the B, C, and D conditions.

5. Results

Before testing the compressor at the part load rating conditions, charge optimization was done at the A condition i.e. condenser WEG inlet temperature of 30°C for both the compressors to determine the optimum charge for the compressor. After that, the compressors are tested at the part load rating conditions at their respective optimum charge. The next section shows the charge optimization of a single speed compressor.

5.1. Charge optimization of a single speed compressor

Figure 2: Variation of \( W_{cp} \), superheat and subcooling with charge

Figure 2 shows the variation of the compressor work, superheat, and subcooling with the charge. As the charge increases, subcooling increases until it reaches a plateau. Superheat remains the same after the first few points because the EEV maintains the same superheat. The high superheat at the initial charge is because of the undercharged system. Figure 3 shows the variation of COP, average capacity, and the
compressor discharge pressure with the charge. There is a range of charge from 1300 g to 1800 g where the curves remain almost constant. This range exists because of the receiver present after the condenser. As the charge increases in this range, the receiver gets filled and the amount of charge present in the remaining system remains almost constant. If the charge is increased beyond this charge, the receiver can no longer hold the charge and liquid starts backing up in the condenser. This is seen by an increase in the compressor discharge pressure and increase in the subcooling. A charge of 1500 g is selected as the optimum charge.

Figure 3: Variation of COP<sub>test</sub>, Q<sub>ev,avg</sub>, and P<sub>cp,ro</sub> with charge

5.2. Single speed compressor part load performance
As explained before, as single speed compressor does not have any modulation, it can operate only at 60 Hz. The variation of the capacity with part load rating conditions is shown in Figure 4. The highest load (corresponding to A condition) is the maximum cooling capacity provided by the system with this compressor operating at A condition (which happens to be 7.3 kW). For the other conditions, this load is multiplied by the corresponding part load ratio from Table 3. Thus load is a linear function of the part load rating conditions.

Figure 4: Variation of capacity, power with percent load for single speed compressor
As seen in Figure 4, as the percent load changes from 100% to 25%, the standard requires a reduced condenser WEG inlet temperature, this reduces the condensing temperature which in turn causes the capacity to increase. The shaded region between the capacity curve and the load curve is an indication of how much the obtained capacity differs from the required load and is a representation of the cycling losses. The capacity remains constant for the C and D conditions because the condenser WEG entering temperature is maintained the same as required by Table 3, only the percent load changes. Since a single speed compressor only operates at one stage, and thus the performance of the compressor at these percent loads is the same. As expected, the compressor power reduces as the percent load reduces because of reduced condensing temperature and compressor discharge pressure.

5.3. Two stage compressor part load performance
A two stage compressor can be operated at either high stage (100%) or low stage (67%). The performance of this compressor at different part load conditions is shown in Figure 5.

For the A condition, the compressor is operated at a high stage. For the B, C, and D conditions, as the required load reduces, the compressor is operated at the low stage. The capacity is not a continuous line between A and B conditions because this compressor does not provide a continuous modulation, it provides a step modulation. At the B condition when the compressor is operated at a low stage, it provides a capacity which matches the required load. A low stage provides 67% cooling capacity compared to high stage when operated at the same conditions. For the B condition, the low stage happens to provide a capacity which is equal to the required load (75% of the load at A condition). However, at the C and D conditions, the low stage cannot match the capacity and it behaves similarly to a single speed compressor and hence there will be cycling losses. The shaded region is a representation of the cycling losses. By comparing Figures 4 and 5, you can see that the shaded region i.e. cycling losses are higher in a single speed compressor compared to a two stage compressor.

5.4. Comparison between single speed and two stage compressor
Figure 6 compares the capacities between a single speed and two stage compressor. At the 100% load, both the compressors provide a similar capacity which indicates that both the compressors are comparable. However, for the remaining conditions, the single speed compressor has higher capacities than a two stage compressor because the two stage compressor can operate at a low stage.
Figure 6 shows the $COP_{\text{test}}$ given by Equation (6). Both the compressors have the same $COP_{\text{test}}$ at 100% load. It is interesting to note that at the 75% load, though the two stage compressor is operating at the low stage and the single speed compressor is operating at its highest (only) stage, the $COP_{\text{test}}$ are still comparable. However, the $COP_{\text{test}}$ for the two stage is significantly higher at the lower percent loads because of reduced capacity and reduced compressor power. The $COP_{\text{test}}$ for each compressor remains the same for the 50% and 25% load because the part load operating conditions are the same for these points.

Figure 7: Comparison of $COP_{\text{test}}$ between single speed and two stage compressor

Figure 8 shows the $COP_R$ given by Equation (8). Except for the 100% load when both the compressors have similar performance, the two stage compressor has a higher $COP_R$ than a single speed compressor. This is because of its ability to operate at lower stage. This trend can also be graphically explained by the cycling losses represented in Figures 4 and 5. Though the $COP_{\text{test}}$ is the same for each compressor at the 25% and 50% load, $COP_R$ at the 25% load is lower than 50% load because higher cycling losses reduce the $COP_{\text{test}}$ by greater value as shown in Equation (9).
IPLV.SI is calculated using the $COP_R$ from Figure 8 and Equation (7). A single speed compressor has an IPLV.SI of 3.6 while the two stage compressor has an IPLV.SI of 3.9. This 7% difference can be explained by the higher $COP_R$ of the two stage compressor at 50% and 75% load which have the highest weighting factors in Equation (7). A two stage compressor does not have any degradation coefficient at the 75% load and though it does incur some cycling losses at the 50% load, it is still operating at the low stage and these losses are lower than that incurred by the single speed compressor.

6. Conclusions
This study experimentally compared a single speed and a two stage compressor in the same R410A WEG chiller. Both the compressors have similar capacity and they belong to the same generation. Charge optimization was done to determine the optimum charge for both the compressor and AHRI 551/591 was used to determine the IPLV.SI for both the compressors at their respective optimum charge. Both compressors showed comparable capacities and $COP_{test}$ at the higher modulation conditions. However, two stage compressor has a higher $COP_R$ at the lower condition because of its ability to operate at a lower stage and better meet the required load, this reduces the $C_D$. Single speed compressor has an IPLV.SI of 3.6 while the two stage compressor has an IPLV.SI of 3.9.

7. Future work
As explained this paper is a first step in a series of comparison studies. The following compressor modulation strategies will be investigated as part of the future work.

7.1. Variable speed compressor
A system with this compressor modulates its capacity by varying the speed of the compressor motor [5]. The focus will be on the inverter driver variable speed compressor with a brushless DC motor. The selected compressor can be operated between 20 Hz and 100 Hz. Variable speed compressor
This compressor modulates its capacity by varying the speed of the motor [4]. The focus will be on the inverter driver variable speed compressor with a brushless DC motor. The selected compressor can be operated between 20 Hz and 100 Hz.

7.2. Digital scroll compressor
In this commercially available compressor, the modulation is achieved through rapid engagement/disengagement of the upper scroll (using a piston and solenoid valve). The scrolls are separated in a periodic cycle to obtain a time-averaged capacity based on the ratio of loading and unloading times. This provides a modulation range from 10% to 100% [6].

7.3. Tandem compressors
In this configuration multiple compressors operate in parallel. The modulation is achieved by turning a different combination of compressors on and off [7]. This study will cover a tandem combination of two compressors. The study will include combinations of a single speed compressor with a variable speed, digital scroll, two stage, and another single speed compressor.

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Nomenclature

| Symbols | Description | Units |
|---------|-------------|-------|
| \( W \) | power | [kW] |
| \( \tilde{C}_d \) | degradation coefficient | [-] |
| \( \tilde{C}_{OP} \) | coefficient of performance | [-] |
| \( \tilde{C}_p \) | specific heat | [kJ/kg-K] |
| IPLV.SI | integrated part load value | [-] |
| \( h \) | enthalpy | [kJ/kg] |
| LF | load factor | [-] |
| \( m \) | mass flow rate | [kg/s] |
| \( Q \) | heat transfer rate | [kW] |
| SC | subcooling | [°C] |
| SH | superheat | [°C] |
| T | temperature | [°C] |
| WEG | water ethylene glycol | |
| \( \varepsilon \) | error | [-] |

Subscripts

| Subscripts | Description |
|-----------|-------------|
| avg | average |
| cp | compressor |
| ev | evaporator |
| ref | refrigerant |
| ro | refrigerant, outlet |

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