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Comparative Experimental Analysis Of Different Compressor Capacity Modulation Strategies In R410A Chiller With Focus On Seasonal Performance

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ABSTRACT

Rising cooling needs for the residential and commercial air conditioning sectors and the demand for higher SEER require HVAC&R to handle cooling loads below the design condition. Though several studies exist focusing on a particular capacity modulation strategy, there is no extensive study that explores all these available modulation strategies in the same experimental facility. This paper is the next step to a previous paper that compared two capacity controls for scroll compressors (single speed and two-stage compressors). The current paper provides insights on additional capacity modulation strategies, including tandem combinations of two single-speed compressors, a single-speed and two-stage compressor, and a variable-speed compressor. These different modulation strategies were tested according to AHRI Standard 551/591 (2020), and their seasonal performance is given by a figure of merit, Integrated Part Load Value (IPLV.SI). All the experimental tests were done using the same R410A water chiller having a nominal cooling capacity of 8 kW. Based on the experimental results, the variable speed compressor has the best IPLV.SI, followed by the tandem combinations. Interestingly, both the tandem combinations (single speed + single speed and two-stage + single speed) have comparable seasonal performance despite having a different number of control states. The estimation of cycling losses using different standards also had an observable effect on the seasonal performance of the tandem combination.

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 (Ekren, 2017).

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.

This paper is the next step to a previous paper that compared two capacity controls for scroll compressors (single speed and two-stage compressors) (Inampudi *et al.*, 2021). The current paper provides insights on additional capacity modulation strategies, including tandem combinations (Cecchinato, 2010) of two single-speed compressors, a single-speed and two-stage compressor (Wang *et al.*, 2012), and a variable-speed compressor (Qureshi and Tassou, 1996) using the same R410A WEG chiller system. These compressors have the comparable nominal cooling capacity. The tests are conducted according to AHRI Standard 551/591 (2020), and the comparison is done using a single figure of merit called IPLV.SI (Integrated Part Load Value).

2. DETAILS OF THE EXPERIMENTAL SETUP

Figure 1 shows the schematic of R410A Water Ethylene Glycol (WEG) chiller. A mixture of 20 % water and ethylene glycol 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. All the compressors used are scroll compressors with a nominal speed of 1800 min^{-1}.

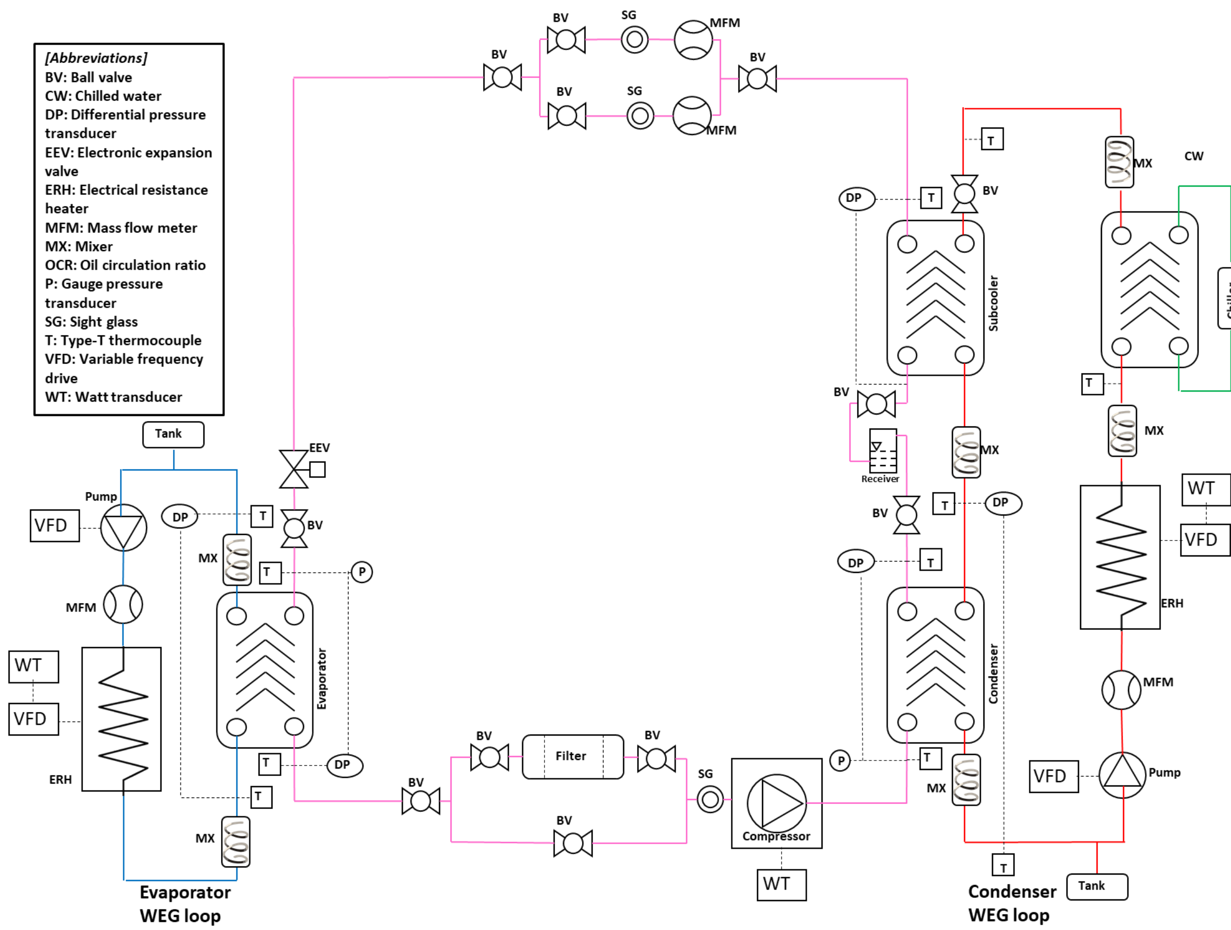

Figure 1: Schematic of the R410A experimental facility

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. REFPROP 10.0 was used to calculate the WEG and R410A properties (Lemmon *et al.*, 2012).

The uncertainty of the sensors used in the experimental facility is presented in Table 2. This uncertainty estimation does not include the uncertainty in thermophysical properties. However, the uncertainty in enthalpy difference can be approximated as the uncertainty in specific heat which is around ±0.5% (Lemmon *et al.*, 2012).

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)	Pressure transducer (kPa)	Mass flow meter (g/s)	Wattmeter (kW)	Capacity (kW)	COP (-)
Uncertainty	±0.1	±0.2%	±0.2%	±0.5%	±1.5%	±1.6%

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 (1). 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 (2). The capacity reported is the average of the refrigerant side and WEG side capacity given by Equation (3). The difference between the two capacities is indicated by the error given by Equation (4). 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 (5).

$$\dot{Q}_{ev,WEG} = \dot{m}_{WEG} C_p \Delta T \tag{1}$$

$$\dot{Q}_{ev,ref} = \dot{m}_{ref} \Delta h \tag{2}$$

$$\dot{Q}_{ev,avg} = \frac{(\dot{Q}_{ev,WEG} + \dot{Q}_{ev,ref})}{2} \tag{3}$$

$$\varepsilon_{Qev} = \frac{(\dot{Q}_{ev,avg} - \dot{Q}_{ev,WEG}) \cdot 100}{\dot{Q}_{ev,avg}} \tag{4}$$

$$COP_{test} = \dot{Q}_{ev,avg} \big/ \dot{W}_{cp} \tag{5}$$

3. AHRI 551/591 (2020) 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 (6). These factors in Equation (6) are based on the weighted average of the most common building types and operations using average weather in 29 U.S cities.

$$IPLV.SI = 0.01 \cdot A + 0.42 \cdot B + 0.45 \cdot C + 0.12 \cdot D \tag{6}$$

$$\begin{aligned} A &= COP_R \ at \ 100\% \\ B &= COP_R \ at \ 75\% \\ C &= COP_R \ at \ 50\% \\ D &= COP_R \ at \ 25\% \end{aligned}$$

If 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 (5), it is degraded to COP_R using the Equation (7), (8), (9), and (10).

$$COP_R = {COP_{test}}/{C_D} \tag{7}$$

$$C_D = (-0.13 \cdot LF) + 1.13 \tag{8}$$

$$LF = \frac{(\%Load)(Q_{ev\ 100\%})}{(Q_{ev\ min\%Load})} \tag{9}$$

$$\%Load = {(Part\ load\ net\ capacity)}/{(Full\ load\ rated\ net\ capacity)} \tag{10}$$

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.

4. RESULTS

4.1 Comparison between single speed and two stage compressor

The seasonal performance was compared between a single speed and two stage compressor (Inampudi *et al.*, 2021). The single speed compressor performance is shown in Figure 2 and two stage compressor performance is shown in Figure 3.

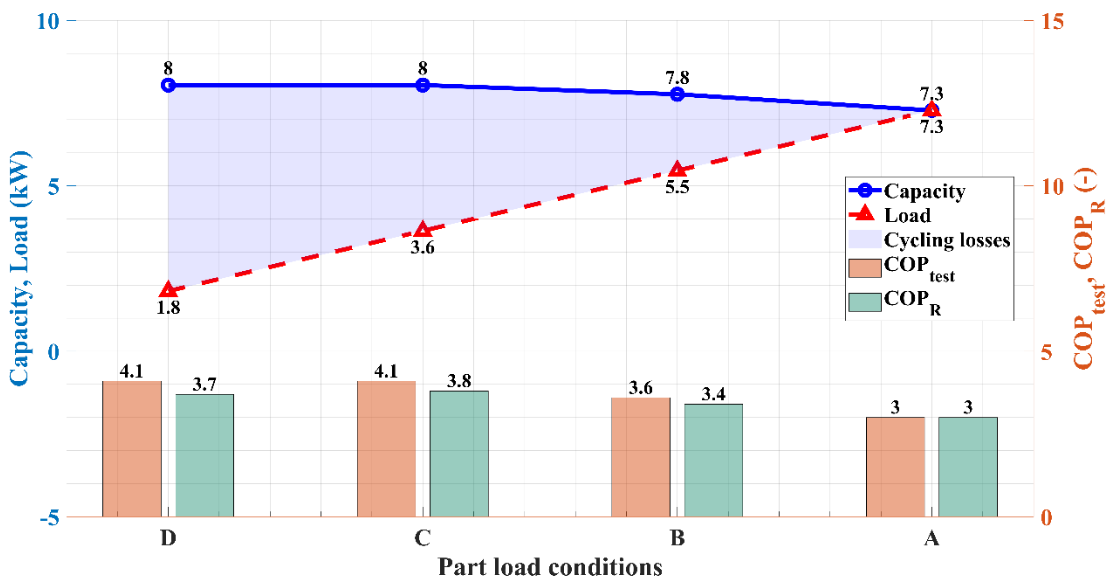

Figure 2: Variation of capacity, COP with percent load for single speed compressor

Figure 2 also shows COP_{test} and COP_R for the single speed compressor. The difference between these two COP is an indication of the cycling losses. The shaded region on the graph also depicts the extent of cycling losses. Except for the A condition, the single speed compressor encounters cycling losses for B, C and D conditions. These cycling losses are because the single speed compressor does not have any modulation and can operate at 60 Hz only.

Figure 3 shows the seasonal performance of the two stage compressor. This compressor can operate at either at a high stage (100% capacity) or a low stage (67% capacity). 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. 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 2 and 3, you can see that the shaded region i.e., cycling losses are higher in a single speed compressor compared to a two stage compressor.

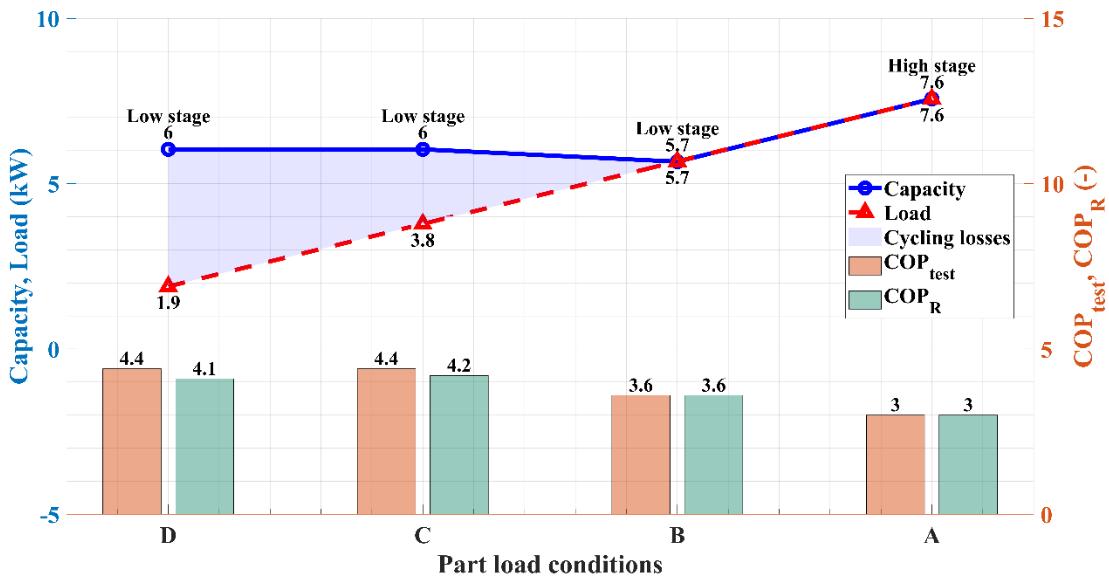

Figure 3: Variation of capacity, COP with percent load for two stage compressor

It can be observed that the COP_{test} at B condition are comparable for the single speed and two stage compressor. This is interesting because the single speed compressor is operating at 60 Hz while the two stage compressor is operating at low stage. The two stage compressor is expected to be significantly better than the single speed compressor. This comparable COP_{test} can be explained by observing the isentropic compressor efficiency, LMTD and pressure drop of the condenser and evaporator shown in Table 4. A higher isentropic compressor efficiency, lower LMTD and pressure drop will cause a higher COP_{test}. From the Table 4, the single speed compressor has the higher isentropic compressor efficiency than the two stage compressor (operating at low stage). If only the compressor efficiency were to be considered, the single speed compressor would have the higher COP_{test}. However, the two stage compressor has the lower LMTD, and pressure drop due to lower mass flow rate. Due to these opposing trends, the two compressors have comparable COP_{test}.

Table 4: Single speed and two stage compressor at performance at B condition

Compressor	Compressor Isentropic Efficiency (%)	Evaporator LMTD (°C)	Condenser LMTD (°C)	Evaporator pressure drop (kPa)	Condenser pressure drop (kPa)
Single speed	62	8.8	14.8	7.8	3.4
Two stage (low stage)	55	7.1	12.1	6.5	3.2

4.2 Effect of compressor motor

A single speed compressor with a later generation motor was also tested. This is the only compressor with a different motor generation, all the other compressors have the same generation motor. This compressor has the comparable cooling capacity as the single speed compressor discussed in Section 5.1. This can be seen by comparing the Figure 2 and 4, these two compressors have comparable cooling capacity at all the test conditions. However, the COP is higher for the compressor with the later generation motor. Since the cooling capacity is comparable for both these compressors and the only difference is the motor generation, this higher COP can be explained by the higher isentropic efficiency of the single speed compressor with new motor as shown in Table 5.

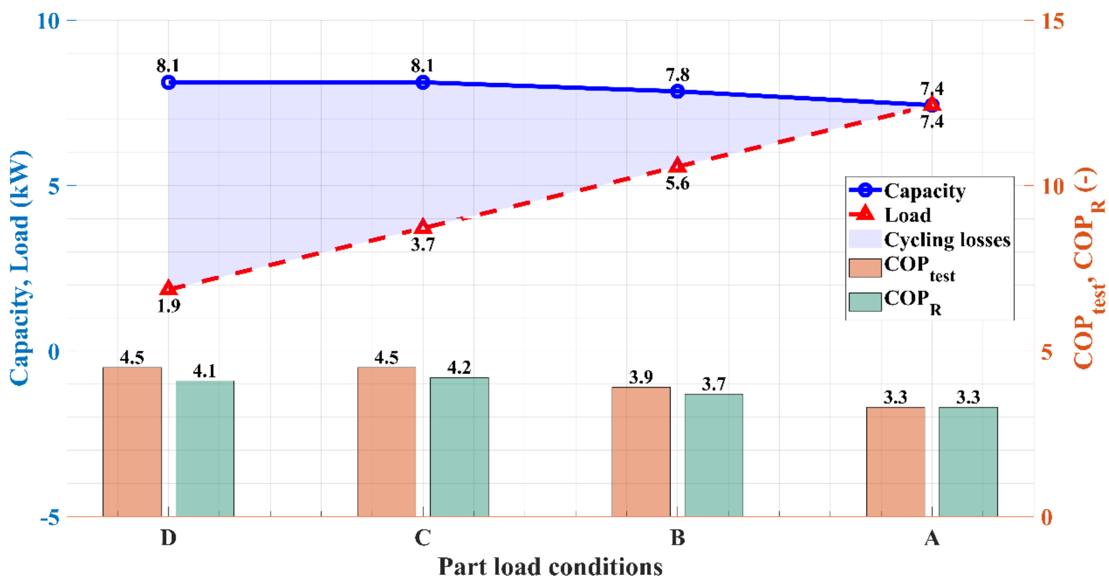

Figure 4: Variation of capacity, COP with percent load for single speed compressor (with new motor)

Table 5: Isentropic efficiency of single speed compressor with old and new motor at different test conditions

Compressor	A condition (%)	B condition (%)	C condition (%)	D condition (%)
Single speed	59	62	63	63
Single speed (new motor)	66	68	68	68

4.3 Tandem combination 1: Two single speed compressors

A tandem combination with two single speed compressors was tested. These single speed compressors are identical, and their cooling capacity is lower than the single speed compressor discussed in Section 5.1. The lower capacity is because of the unavailability of compressor combinations in the market that have the same cooling capacity as the single speed compressor (Section 5.1). This combination can operate in three stages shown in Table 6. The tandem combination operates at Stage 1 for A condition. For the B condition, the capacity is in between when two compressors are ON and when only one compressor is ON. The AHRI standard recommends the COP to be an interpolation between the case with two compressors ON and one compressor ON. Though the second compressor undergoes cycling, the standard does not recommend any cycling losses for the case of interpolation. For C and D conditions, only one compressor is ON, and it undergoes cycling losses. These cycling losses are indicated by the shaded region and the difference between COP_{test} and COP_R as shown in Figure 5.

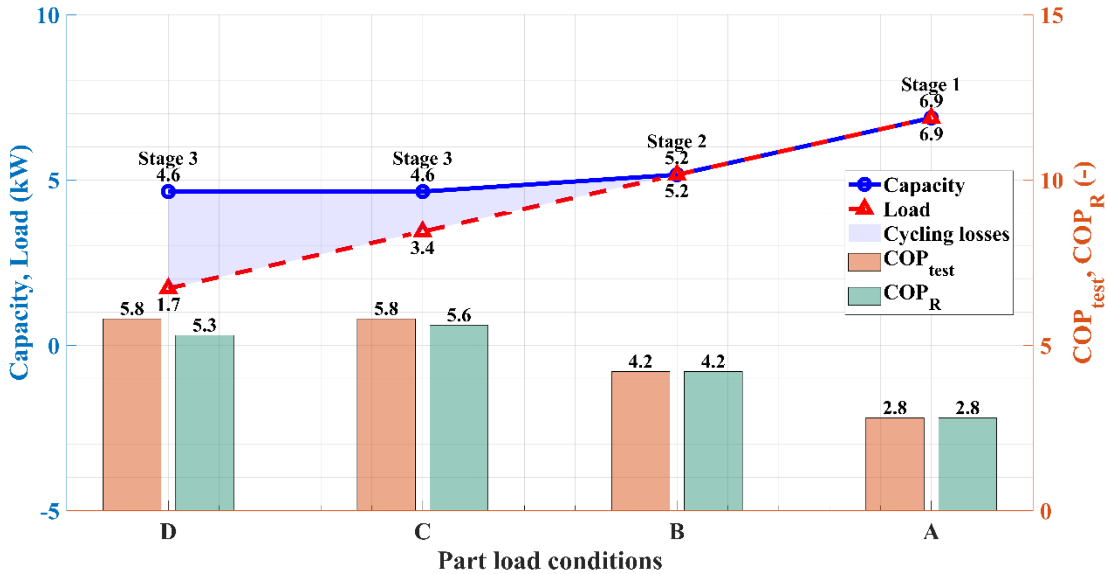

Figure 5: Variation of capacity, COP with percent load for tandem combination-1

Table 6: Different stages for tandem combination-1

Different Stages	**Description**
Stage 1	Both compressors are ON
Stage 2	Only one compressor is ON. The other compressor undergoes cycling losses
Stage 3	One compressor is ON and undergoes cycling losses. The other compressor is OFF

4.4 Tandem combination 2: Single speed and two stage compressor

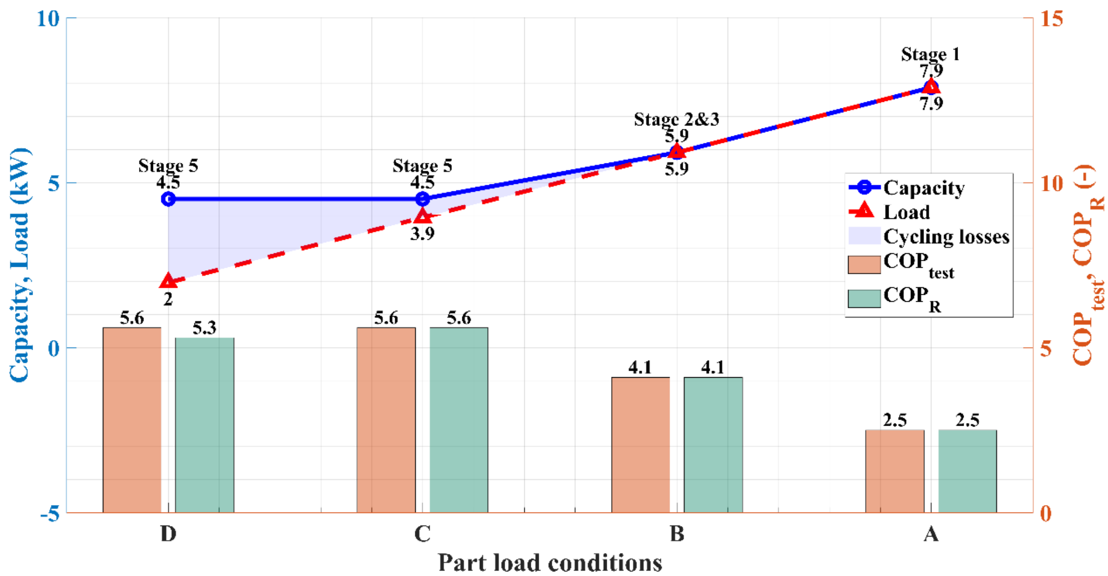

Figure 6: Variation of capacity, COP with percent load for tandem combination-2

A second tandem combination with a single speed and two stage compressor was tested. The single speed compressor has the same cooling capacity as the one in Section 5.3 and the two stage compressor is smaller than the one discussed in Section 5.1. This is the only possible combination in the market which can provide comparable cooling capacity as the other tested compressors. This tandem combination can operate in five stages as shown in Table 7. In an ideal scenario when multiple stages are available, it is expected that you would utilize all the available stages in a tandem combination. The ideal scenario where all the available stages are used is shown in Table 8. However, due to the size of the compressors used and the similar capacities provided by a couple of available stages, only a few stages are used as shown in Figure 6 and Table 8. At the A condition, the tandem combination operates at Stage 1 and at B condition, it operates between Stages 2 and 3. This is like the interpolation done for the tandem combination 1 at the B condition. At C and D condition, the tandem combination operates at Stage 5 but does undergo cycling losses. Stages 4 and 5 have the same cooling capacity but the Stage 5 has better COP_{test}, so Stage 5 was used for both C and D conditions.

Table 7: Different stages for tandem combination-2

Different Stages	**Description**
Stage 1	Single speed + Two stage (High stage)
Stage 2	Single speed + Two stage (Low stage)
Stage 3	Two stage (High stage)
Stage 4	Single speed
Stage 5	Two stage (Low stage)

Table 8: Ideal and actual scenario

Scenario	**A condition**	**B condition**	**C condition**	**D condition**
Ideal scenario	Stage 1	Stage 2 + Stage 3	Stage 3 + Stage 4	Stage 5
Actual scenario	Stage 1	Stage 2 + Stage 3	Stage 5 (with cycling losses)	Stage 5 (with cycling losses)

4.5 Variable speed compressor

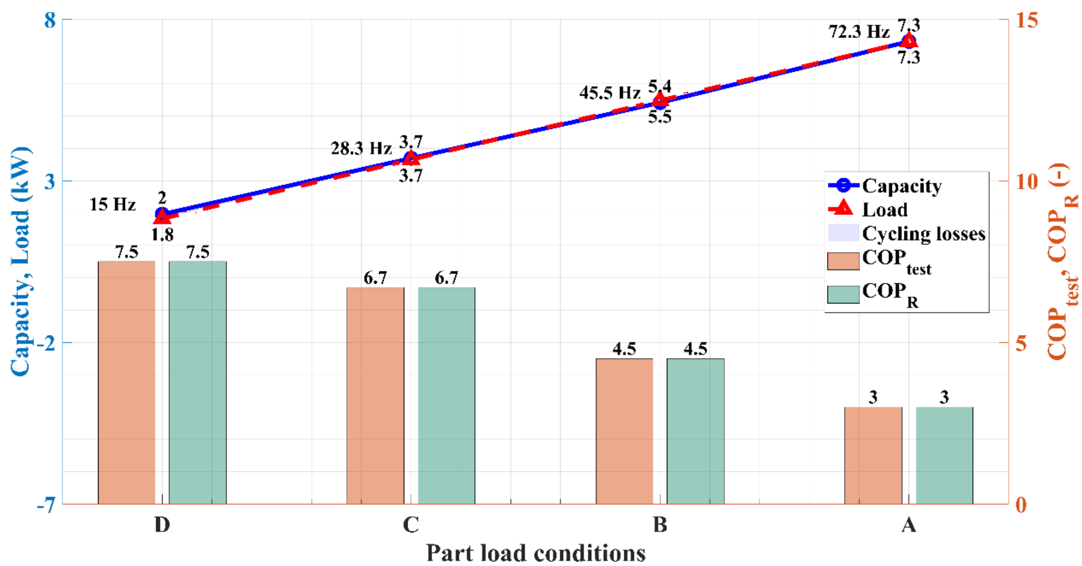

Figure 6: Variation of capacity, COP with percent load for variable speed compressor

A variable speed compressor is tested. This variable speed compressor can operate between 100 and 15 Hz. The operating frequency of this compressor at different test points is shown in Figure 6. This compressor can match the cooling capacity with the required cooling load at all the points and thus it does not undergo any cycling losses. Additionally, there is no difference between COP_{test} and COP_R. This higher COP_{test} is due to lower LMTD on both heat exchangers and comparable compressor isentropic efficiency as the other compressors.

4.6 Seasonal performance of the different compressors

The COP_{test} and COP_R for different compressors tested is shown in Table 9 and the IPLV.SI is shown in Table 10. According to Equation (7), the COP_R for B and C have highest weight (87%) in the IPLV.SI. Any compressor that operates efficiently at these points will have the best seasonal performance (i.e., IPLV.SI). The variable speed compressor has the highest COP_R at all the tested points. This is because of its ability to operate at low compressor speeds and match the cooling capacity with the required load. Additionally, the lower refrigerant mass flow rate reduces the LMTD in the condenser and evaporator, thus reducing the pressure ratio. Interestingly the two tandem combinations have the same IPLV.SI though the tandem combination-2 has the higher number of stages than tandem combination-1. This proves that just because a tandem combination has higher stages does not mean it would have more seasonal performance. Though tandem combination-2 has no significant cycling losses at B and C condition compared to tandem combination-1, it has a lower COP_{test}. This lower COP_{test} is because of the lower isentropic efficiency of the two stage compressor when operating at the low stage.

Table 9: COP_{test} and COP_R for different compressors tested

Compressor	**A condition**		**B condition**		**C condition**		**D condition**	
	COP_{test}	COP_R	COP_{test}	COP_R	COP_{test}	COP_R	COP_{test}	COP_R
Single speed	3.0	3.0	3.6	3.4	4.1	3.8	4.1	3.7
Two stage	3.0	3.0	3.6	3.6	4.4	4.2	4.4	4.1
Single speed (new motor)	3.3	3.3	3.9	3.7	4.5	4.2	4.5	4.1
Tandem combination-1	2.8	2.8	4.2	4.2	5.8	5.6	5.8	5.3
Tandem combination-2	2.5	2.5	4.1	4.1	5.6	5.6	5.6	5.3
Variable speed	3.0	3.0	4.5	4.5	6.7	6.7	7.5	7.5

Table 10: IPLV.SI for the different compressors tested

Compressor	**IPLV.SI**
Single speed	3.6
Two stage	3.9
Single speed (new motor)	4.0
Tandem combination-1	4.9
Tandem combination-2	4.9
Variable speed	5.8

5. CONCLUSIONS

This study experimentally compared a single speed, a two stage compressor, tandem combinations of two single-speed compressors, a single-speed and two-stage compressor, and a variable-speed compressor using the same R410A WEG chiller system. These compressors have the comparable nominal cooling capacity. When comparing the two stage and single speed compressor performance, it was seen that heat exchanger pay an important role in addition to the compressor modulation strategy. It is also observed that compressors should be of same motor generation to ensure a fair comparison. The tandem combination with higher number of stages was seen to have a same seasonal performance as that of a tandem combination with lower number of stages. Thus, not all the stages in a tandem combination might be useful. Variable speed compressor had the best IPLV.SI followed by the two tandem combinations.

NOMENCLATURE

C_d	degradation coefficient [-]	m	mass flow rate [kg/s]
COP	coefficient of performance [-]	Q	heat transfer rate [kW]
C_p	specific heat [kJ/kg-K]	SC	subcooling [°C]
ε	error [-]	SH	superheat [°C]
IPLV.SI	integrated part load value [-]	T	temperature [°C]
h	enthalpy [kJ/kg]	W	power [kW]
LF	load factor [-]	WEG	water ethylene glycol
LMTD	log mean temperature difference [°C]		

Subscript

avg	average
cp	compressor
ev	evaporator
ref	refrigerant
ro	refrigerant, outlet
weg	water ethylene glycol mixture

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