Experimental Investigation and Advanced Exergy Analysis of Different Factors That Can Affect Seasonal Performance in an R410A Chiller

Sugun Tej Inampudi and Stefan Elbel

Abstract Due to the new energy regulations, there is an increased demand for higher efficiency HVAC&R equipment. Improvements in compressors and heat exchangers (BPHX) can lead to a higher system COP. Compressors are compared using efficiency and BPHX are evaluated based on their LMTDs and pressure drops. It is difficult to compare the contribution of these two on the system COP using traditional first law analysis. Advanced exergy analysis (AEA) can help with this estimation by using exergy destruction as a metric. In the current paper, AEA framework is used to analyze the different factors that can impact the system performance and seasonal performance. The study is done on an 8-kW R410A water ethylene glycol chiller (WEG) and the results from seasonal performance of three different scroll compressors are used for this analysis. Three factors included are compressor motor efficiency, condenser size and compressor capacity modulation strategy. AEA shows that optimizing the compressor followed by condenser has the highest potential of improving system performance. $\dot{E}_D^{EX,total}$ drops by 18% with upgraded compressor while it drops by 28% with oversized condenser showing that that the condenser has higher impact on the exergy destruction. AEA comparison of a two-stage compressor and a variable speed compressor show that $\dot{E}_D^{EN}$ of the two-stage compressor is 39% higher indicating that variable speed compressor is more efficient at matching the cooling load. Based on the relative magnitude of $\dot{E}_D^{AV,EN}$, compressor upgrade and condenser upgrade would yield higher system performance in two-stage and variable speed compressor systems respectively.

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1 Introduction

There is an increased demand for higher efficiency HVAC&R components to meet the new energy regulations. The performance of a system can be increased by improving the compressor and the heat exchangers. The compressors are compared using isentropic efficiency while the heat exchangers are assessed based on the temperature difference between the two working fluids and pressure drops. It is difficult to compare the contribution of a compressor and heat exchanger on the system COP using traditional first law and energy balance. An exergy analysis which calculates the exergy destruction within the compressor and heat exchangers can help with this comparison.

In Conventional Exergy Analysis (CEA), the exergy is split among the different components, however, the exergy destruction in the components may neither be caused by their imperfections and nor be reduced by the optimization of that component. Splitting the exergy destruction into parts (avoidable and unavoidable, or endogenous and exogenous) can improve the quality of the conclusions from an exergy analysis and this type of analysis is called Advanced Exergy Analysis (AEA).

Splitting the actual exergy destruction of the kth component ($\dot{E}_{D,k}^{real}$) into endogenous ($\dot{E}_{D,k}^{EN}$) and exogenous ($\dot{E}_{D,k}^{EX} = \dot{E}_{D,k}^{real} - \dot{E}_{D,k}^{EN}$) makes it possible to separately estimate the exergy destruction in a component caused by the component itself and by the remaining components. $\dot{E}_{D,k}^{EN}$ part of the exergy destruction is associated with only the irreversibilities occurring in the kth component when all the other components operate in an ideal way and the component being considered operates with its current efficiency. $\dot{E}_{D,k}^{EX}$ includes the exergy destruction caused by the irreversibilities that occur in the remaining components. $\dot{E}_{D,k}^{EN}$ is used as an indicator of the theoretical maximum potential of the system by optimizing the concerned component. $\dot{E}_{D,k}^{EN}$ and $\dot{E}_{D,k}^{EX}$ are calculated using the hybrid cycles [1, 2] and is further discussed in Sect. 4.

Splitting the actual exergy destruction of the kth component ($\dot{E}_{D,k}^{real}$) into unavoidable ($\dot{E}_{D,k}^{UN}$) and avoidable ($\dot{E}_{D,k}^{AV} = \dot{E}_{D,k}^{real} - \dot{E}_{D,k}^{UN}$) provides a realistic measure of the potential for improving the thermodynamic efficiency of a component [1, 2]. The exergy destruction rate that cannot be reduced due to technological limitations such as availability and cost of materials and manufacturing methods is the $\dot{E}_{D,k}^{UN}$ [3]. $\dot{E}_{D,k}^{AV}$ is the remaining part of the exergy destruction. $\dot{E}_{D,k}^{UN}$ will always be there if this component is being used in the system [3].

The two approaches of splitting the exergy destruction can be combined to calculate the part of the total exergy destruction that depends on the inefficiencies within the k^{th} component and that cannot be reduced because of technical limitations for the kth component, i.e., endogenous unavoidable ($\dot{E}_{D,k}^{EN,UN}$) part of the exergy destruction. Endogenous avoidable ($\dot{E}_{D,k}^{EN,AV}$) part of the exergy can be reduced through changes in the component being considered whereas the exogenous avoidable ($\dot{E}_{D,k}^{EX,AV}$) part of the exergy destruction describes the part that can be reduced by improving the efficiency of the remaining components or by improving the structure of the overall system [1, 2]. Decreasing $\dot{E}_{D,k}^{EN,AV}$ of the kth component, in general will lead to a decrease of $\dot{E}_{D}^{EX,AV}$ of the other components. $\dot{E}_{D,k}^{EN,AV}$ indicates the real maximum improvement potential at the current technical level [3].

In the current paper, AEA framework is used to analyze the different factors that can impact the system performance and seasonal performance. The results from seasonal performance of three different compressors single speed, two-stage and variable speed compressor are used for this analysis. The study is done on an 8-kW R410A water ethylene glycol chiller (WEG) and seasonal performance (IPLV.SI) is measured using AHRI 551/591. The three factors included in this analysis that impact the system performance are compressor motor efficiency, heat exchanger size and compressor capacity modulation strategy. The improvement in system COP due to improved compressor motor and increased condenser size are analyzed using the single speed and the two-stage compressor in Sects. 5.1 and 5.2. While the effect of different ways to modulate capacity is studied using the two-stage and variable speed compressor in Sect. 5.3.

2 Experimental Facility Description

An 8-kW R410A WEG chiller with a mixture of 20% water and ethylene glycol as the secondary fluid is used for this study. 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 temperature. All the heat exchangers used in the facility are brazed plate heat exchangers (BPHX). A 0.9 L receiver is connected between the condenser and the subcooler. Superheat is controlled by an electronic expansion valve (EXV). The system is charged with enough refrigerant to fill the receiver halfway and thus subcooling is a function of the charge. The geometric dimensions of the evaporator, different condensers, and subcooler can be found in Table 1. Compressors used are scroll compressors with a nominal speed of 1800 min^{-1}. Additional details and the schematic of the experimental facility, instrumentation used can be found in [4]. Uncertainty of the sensors used in the experimental facility is presented in Table 2.

Table 1 Dimensions of the BPHX

Heat exchanger	Length (mm)	Width (mm)	Number of plates
Evaporator	311	111	28
Subcooler	207	77	14
Condenser (C-14)	311	111	14
Oversized condenser (C-24)	311	111	24

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	± 0.2	± 0.2	± 0.2	± 0.5	± 3.0	± 3.2

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

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

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

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

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

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

3 Standard Used for Testing

AHRI 551/591 is used for the determination of the part-load performance of water chillers [5]. 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. 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. The standard required that the condenser and evaporator WEG flow rate used for the A condition be used for the B, C, and D conditions. IPLV.SI is the weighted average of the COP_R measured at these standard rating

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
*Same condenser WEG mass flow rate as A condition			
**Same evaporator WEG mass flow rate as A condition			

conditions as shown in Eq. (6). These factors in Eq. (6) are based on the weighted average of the most common building types and operations using average weather in 29 U.S cities. Additional details of how cycling losses are estimated using this standard are available in [4, 6].

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

$$A = COP_R \, at \, 100\%$$

$$B = COP_R \, at \, 75\%$$

$$C = COP_R \, at \, 50\%$$

$$D = COP_R \, at \, 25\%$$

4 Advanced Exergy Analysis (AEA)

The exergy destruction of each component is split into endogenous/exogenous and unavoidable/avoidable exergy destruction using the three different cycles shown in Fig. 1. The theoretical cycle includes the assumption that the exergy destruction within the component, $\dot{E}_D^{th}$ is minimum or zero. The $\dot{E}_D^{th}$ of the EXV and the compressor are set to be zero by considering an isentropic expansion and compression. $\dot{E}_D^{th}$ of the condenser (and evaporator) are set to the minimum possible by considering the minimum temperature difference between the secondary fluid and refrigerant ΔT_{CO}^{th} (and ΔT_{EV}^{th}) to be zero [1–3].

To split the exergy destruction into unavoidable and avoidable parts, separate cycle with only unavoidable exergy destructions occurring in each component should be considered. The unavoidable exergy cycle is built using the theoretical cycle but also includes the irreversibilities caused by unavoidable temperature difference in the

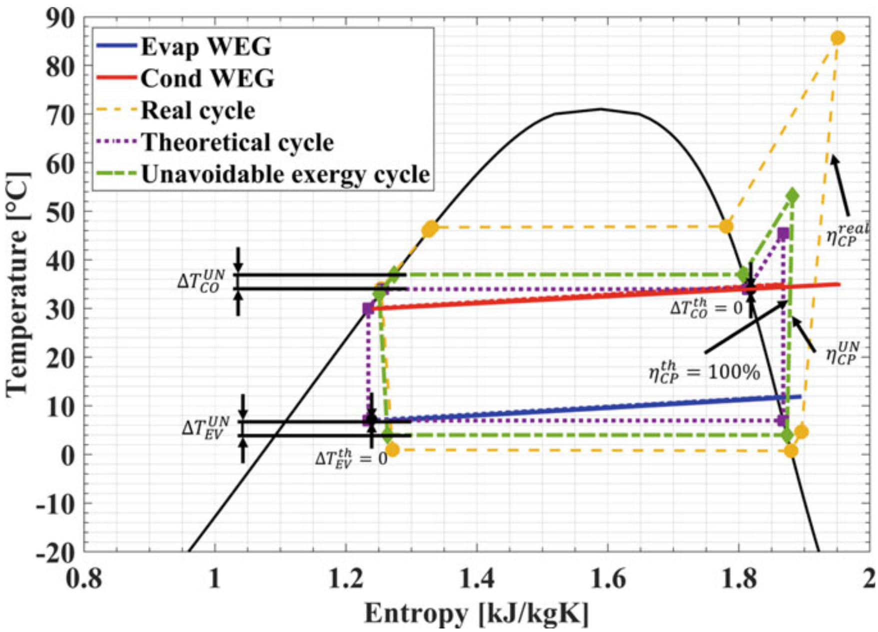

Fig. 1 Cycles used to calculate the different parts of exergy destruction in a component

heat exchangers (ΔT_{CO}^{UN}, ΔT_{EV}^{UN}), compressor efficiency (η_{CP}^{UN}) and throttling in the EXV. For the current analysis, the following parameter values for the unavoidable efficiencies were assumed: $\Delta T_{CO}^{UN} = 0.5K$, $\Delta T_{EV}^{UN} = 0.5K$ and $\eta_{CP}^{UN} = 95\%$. These values represent the technological limitations [1–3].

To calculate the endogenous part of the exergy destruction within each component, the following hybrid cycles should be analyzed. In each of these hybrid cycles, there is only one irreversible component. These cycles are shown in Fig. 2.

- Compressor (Fig. 2a): $1_T - 2_H - 3_T - 4_T$ (η_{CP}^{real} from experiment)
- Condenser (Fig. 2b): $1_T - 2_{H^*} - 3_R - 4_H$ (ΔT_{CO}^{real} from experiment)
- Evaporator (Fig. 2c): $1_T - 2_{H^{**}} - 3_T - 4_{H^{**}}$ (ΔT_{EV}^{real} from experiment)
- EXV (Fig. 2d): $1_T - 2_T - 3_T - 4_{H^*}$ (Isenthalpic process from experiment)

To calculate the unavoidable endogenous part of the exergy destruction within each component, the approach for calculating the endogenous part of the exergy destruction is used with the efficiency of each component being equal to the efficiency used to calculate its unavoidable exergy destruction [1–3].

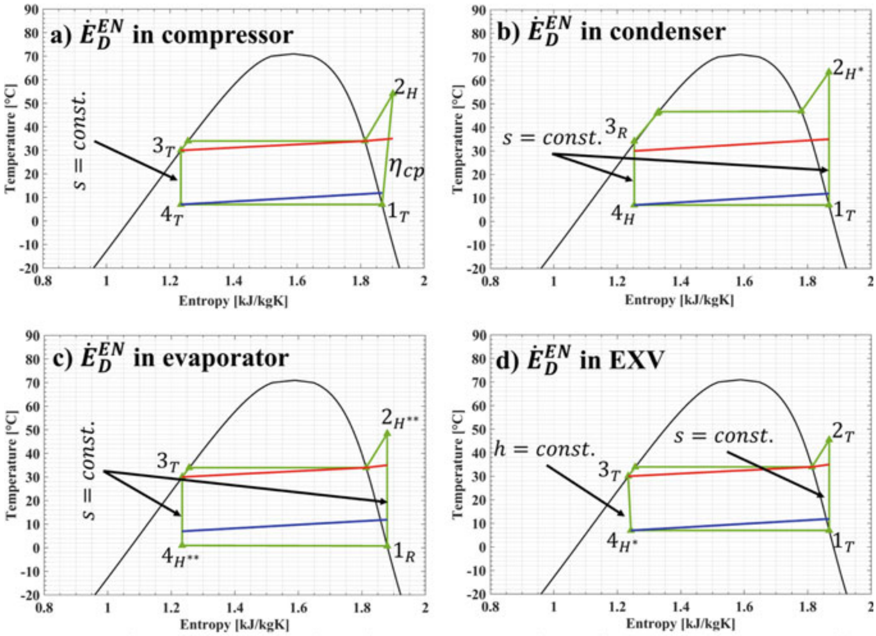

Fig. 2 Cycle used to calculate the endogenous exergy destruction within **a** compressor, **b** condenser, **c** evaporator and **d** EXV

5 Results and Discussion

5.1 *Effect of Improved Compressor Motor (Higher Isentropic Efficiency)*

A single speed compressor does not have any capacity modulation. It can operate at its maximum possible speed at all the conditions and whenever the load is less than the provided capacity, it undergoes cycling losses. A single speed compressor with a baseline compressor motor and a similar capacity single speed compressor with an improved motor were tested according to AHRI 551/591. Improved compressor motor has higher isentropic efficiency at all the tested conditions [4, 6]. Entropy destruction of different components at full load conditions when operating with these two different compressors are shown in Tables 4 and 5. Since, the load matches the capacity, the single speed compressor does not encounter any cycling losses at this A condition.

Comparing the $\dot{E}_D^{AV,EN}$, compressor (0.37 kW) followed closely by condenser (0.30 kW) have the highest potential to improve the performance of the system. Though a similar trend is observed by comparing the $\dot{E}_D^{real}$, the relative magnitudes of the $\dot{E}_D^{real}$ might lead to a conclusion that upgrading the compressor (0.82 kW) will

Table 4 Exergy destruction when using a single speed compressor at A condition ($COP_{test} = 3.0$; $IPLV.SI = 3.6$)

Component	Exergy destruction (kW)					
	$\dot{E}_D^{real}$	$\dot{E}_D^{EN}$	$\dot{E}_D^{EX}$	$\dot{E}_D^{UN}$	$\dot{E}_D^{AV}$	$\dot{E}_D^{AV,EN}$
Compressor	0.82	0.51	0.31	0.14	0.68	0.37
Condenser	0.43	0.36	0.09	0.06	0.39	0.30
Evaporator	0.23	0.23	0.00	0.08	0.15	0.15
EXV	0.21	0.11	0.10	0.11	0.09	0.00
Total	1.69		0.51			

Table 5 Exergy destruction when using a single speed compressor with an improved motor at A condition ($COP_{test} = 3.3$; $IPLV.SI = 4.0$)

Component	Exergy destruction (kW)					
	$\dot{E}_D^{real}$	$\dot{E}_D^{EN}$	$\dot{E}_D^{EX}$	$\dot{E}_D^{UN}$	$\dot{E}_D^{AV}$	$\dot{E}_D^{AV,EN}$
Compressor	0.65	0.40	0.25	0.14	0.50	0.26
Condenser	0.41	0.36	0.05	0.06	0.35	0.30
Evaporator	0.24	0.23	0.01	0.08	0.16	0.15
EXV	0.21	0.11	0.11	0.11	0.10	0.00
Total	1.51		0.42			

yield the maximum improvement and the $\dot{E}_D^{real}$ of the compressor is almost twice the $\dot{E}_D^{real}$ of condenser. This shows the potential insights that can be gained through AEA which are not evident through the traditional exergy analysis.

Note that the $\dot{E}_D^{EN}$, $\dot{E}_D^{AV,EN}$ remain constant between Tables 4 and 5 for the condenser, evaporator and EXV. The $\dot{E}_D^{UN}$ is constant for the components.

The $\dot{E}_D^{real}$ of the compressor drops by 21% while $\dot{E}_D^{real}$ of the other components remain the same when the compressor motor is improved. This causes the total exergy destruction ($\dot{E}_D^{real,total}$) drops by 11%. The $\dot{E}_D^{AV,EN}$ of the compressor drops by 30%. Due to the improved compressor motor, the total $\dot{E}_D^{EX}$ which accounts for the interdependence between the components drops by 18%. Interestingly, only the $\dot{E}_D^{EX}$ of the compressor reduced significantly from 0.31 to 0.25 kW. The compressor improvement did not cause the exergy destruction to drop in the other components showing that the interdependence of compressor on the other components is not significant.

5.2 Effect of Upgrading the Condenser (Oversized Condenser)

A two-stage compressor can operate at either 100% capacity or 67% capacity depending on the opening or closing of the internal bypass ports [4, 6]. The two-stage compressor has similar compressor motor as the single speed compressor shown in Table 4 and the cooling capacity provided at the full load conditions are also comparable. Experiments were conducted with this two-stage compressor operating at high stage at A condition with a 14-plate condenser and another 24-plate condenser with similar plate geometry. The exergy destruction for these two cases is shown in Tables 6 and 7. By analyzing the results from Sect. 5.1 and 5.2, it is possible to compare the effect of upgrading a compressor motor and increasing the size of condenser.

From Tables 6 and 7, it is seen that the $\dot{E}_D^{real,total}$ drops by 13% when the size of the condenser increases by 70%. This 13% drop in $\dot{E}_D^{real,total}$ with condenser upgrade is like the 11% drop with upgraded compressor motor. $\dot{E}_D^{real}$ of the condenser drops by 31% which is significantly higher than the 21% drop in $\dot{E}_D^{real}$ of the upgraded compressor. $\dot{E}_D^{AV,EN}$ of the condenser drops by 27% which is comparable to the $\dot{E}_D^{AV,EN}$ drop of the compressor. Interestingly, upgrading the condenser drops the $\dot{E}_D^{EX,total}$ from 0.53 to 0.38 kW (28% drop) while this drop was only

Table 6 Exergy destruction when using a two-stage compressor at A condition ($COP_{test} = 3.0$; $IPLV.SI = 3.9$)

Component	Exergy destruction (kW)					
	$\dot{E}_D^{real}$	$\dot{E}_D^{EN}$	$\dot{E}_D^{EX}$	$\dot{E}_D^{UN}$	$\dot{E}_D^{AV}$	$\dot{E}_D^{AV,EN}$
Compressor	0.84	0.51	0.33	0.15	0.69	0.37
Condenser	0.45	0.39	0.09	0.06	0.41	0.33
Evaporator	0.25	0.25	0.00	0.09	0.16	0.16
EXV	0.22	0.11	0.11	0.12	0.10	0.00
Total	1.76		0.53			

Table 7 Exergy destruction when using a two-stage compressor with an oversized condenser at A condition ($COP_{test} = 3.3$; $IPLV.SI = 4.6$)

Component	Exergy destruction (kW)					
	$\dot{E}_D^{real}$	$\dot{E}_D^{EN}$	$\dot{E}_D^{EX}$	$\dot{E}_D^{UN}$	$\dot{E}_D^{AV}$	$\dot{E}_D^{AV,EN}$
Compressor	0.79	0.51	0.28	0.15	0.64	0.37
Condenser	0.31	0.30	0.01	0.06	0.25	0.24
Evaporator	0.24	0.24	0.00	0.09	0.16	0.16
EXV	0.20	0.11	0.09	0.12	0.08	0.00
Total	1.54		0.38			

18% with the upgraded compressor. This difference in the drop of $\dot{E}_D^{EX,total}$ shows that the condenser has higher impact on the exergy destruction than upgrading the compressor. This impact of oversized condenser can be different at part load conditions with a compressor which can match the required cooling load by reducing the refrigerant mass flow rate.

5.3 Comparison of Two Compressor Modulation Strategies

A variable speed compressor can match the required cooling load by reducing their refrigerant mass flow rates at the 75% part load condition (B condition). The variable speed compressor reduces its operating speed from 72.5 to 45.5 Hz while the two-stage compressor operates at low stage (the bypass ports are opened) to reduce the refrigerant mass flow rate. Both the compressors do not undergo the cycling losses and since the refrigerant mass flow rate across the heat exchangers are same, the only contributing factor to the difference in the system performance must be the compressor [6].

The exergy destruction with a two-stage compressor and a variable speed compressor when operating at B condition is shown in Tables 8 and 9 respectively. The $\dot{E}_D^{real}$ of the two-stage compressor is 40% higher than that of the variable speed compressor. Interestingly, $\dot{E}_D^{EN}$ of the two-stage compressor and variable speed compressor is 60% of the $\dot{E}_D^{real}$ indicating that improving the compressor efficiency is the best way to reduce the compressor exergy destruction. $\dot{E}_D^{EN}$ of the two-stage compressor is 39% higher and the $\dot{E}_D^{AV,EN}$ is twice that of the variable speed compressor. This shows that variable speed compressor has lower overall and endogenous exergy destruction and thus is more efficient at matching the required cooling load than the two-stage compressor. Additionally, the higher $\dot{E}_D^{AV,EN}$ of the two-stage compressor indicates that in a system with this compressor, improving the efficiency of this compressor will yield higher performance. On the other hand, the higher $\dot{E}_D^{AV,EN}$ for the condenser in the case of variable speed compressor shows that priority should be given to condenser upgrade. The AEA shows that different components should be prioritized for upgrading depending on the compressor and capacity modulation strategy used.

6 Conclusions

In the current paper, AEA framework is used to analyze the different factors that can impact the system performance and seasonal performance. Comparing the avoidable endogenous exergy destruction ($\dot{E}_D^{AV,EN}$) of the single speed compressor, optimizing the compressor followed closely by condenser has the highest potential of improving system performance. With an upgraded compressor motor $\dot{E}_D^{real}$ of

Table 8. Exergy destruction when using a two-stage compressor at B condition ($COP_{test} = 3.6$; $IPLV.SI = 4.0$)

Component	Exergy destruction (kW)					
	$\dot{E}_D^{real}$	$\dot{E}_D^{EN}$	$\dot{E}_D^{EX}$	$\dot{E}_D^{UN}$	$\dot{E}_D^{AV}$	$\dot{E}_D^{AV,EN}$
Compressor	0.60	0.36	0.24	0.08	0.52	0.28
Condenser	0.29	0.23	0.06	0.04	0.25	0.19
Evaporator	0.15	0.12	0.03	0.05	0.10	0.08
EXV	0.09	0.05	0.05	0.05	0.04	0.00
Total	1.13		0.37			

Table 9 Exergy destruction when using a variable speed compressor at B condition ($COP_{test} = 4.5$; $IPLV.SI = 5.8$)

Component	Exergy destruction (kW)					
	$\dot{E}_D^{real}$	$\dot{E}_D^{EN}$	$\dot{E}_D^{EX}$	$\dot{E}_D^{UN}$	$\dot{E}_D^{AV}$	$\dot{E}_D^{AV,EN}$
Compressor	0.36	0.22	0.15	0.08	0.28	0.14
Condenser	0.23	0.23	0.01	0.04	0.20	0.19
Evaporator	0.12	0.12	0.00	0.05	0.08	0.08
EXV	0.09	0.05	0.04	0.05	0.04	0.00
Total	0.81		0.20			

the single speed compressor reduced by 21% while the $\dot{E}_D^{real,total}$ drops by 11%. When the condenser is oversized by 70%, $\dot{E}_D^{real,total}$ with a similar sized two-stage compressor drops by 13%. $\dot{E}_D^{real}$ of the condenser drops by 31% which is significantly higher than the 21% drop in $\dot{E}_D^{real}$ of the upgraded compressor. Interestingly, only the $\dot{E}_D^{EX}$ (which accounts for the interdependence between the components) of the compressor reduced significantly while it remained constant for other components with the compressor upgrade causing the $\dot{E}_D^{EX,total}$ to drop by only 18%. On the other hand, upgrading the condenser dropped the $\dot{E}_D^{EX,total}$ by 28%. This difference in the drop of $\dot{E}_D^{EX,total}$ shows that the upgrading/oversizing condenser has higher impact on the exergy destruction than upgrading the compressor and that interdependence of compressor on the other components is not as significant. The exergy destruction with a two-stage compressor and a variable speed compressor are compared at 75% load condition. $\dot{E}_D^{EN}$ of the two-stage compressor is 39% higher and the $\dot{E}_D^{AV,EN}$ is twice that of the variable speed compressor. This shows that variable speed compressor is more efficient at matching the required cooling load than the two-stage compressor. Based on the relative magnitude of $\dot{E}_D^{AV,EN}$ compressor upgrade and condenser upgrade would yield higher system performance in two-stage and variable speed compressor system respectively.

References

1. T. Morosuk, G. Tsatsaronis, Advanced exergetic evaluation of refrigeration machines using different working fluids. Energy **34**(12), 2248–2258 (2009)
2. S. Kelly, G. Tsatsaronis, T. Morosuk, Advanced exergetic analysis: approaches for splitting the exergy destruction into endogenous and exogenous parts. Energy **34**(3), 384–391 (2009)
3. T. Morosuk, G. Tsatsaronis, in *ASME International Mechanical Engineering Congress and Exposition*, vol. 54907. Advanced Exergoeconomic Analysis of a Refrigeration Machine: Part 1—Methodology and First Evaluation (2011), pp. 47–56
4. S. Inampudi, F. Botticella, S. Elbel, in *IOP Conference Series: Materials Science and Engineering*, vol. 1180, no. 1. Part Load Performance of Single and Two Stage Compressors—A Comparative Experimental Study in a R410A Chiller Unit (IOP Publishing, 2021), p. 012050
5. AHRI, *AHRI Standard 551/591 (SI): 2020 Standard for Performance Rating of Water-Chilling and Heat Pump Water-Heating Packages Using the Vapor Compression Cycle* (Air-Conditioning, Heating, Refrigeration Institute, Arlington, VA, USA, 2020).
6. S.T. Inampudi, F. Botticella, S. Elbel, in *19th International Refrigeration and Air Conditioning Conference at Purdue.* Comparative Experimental Analysis of Different Compressor Capacity Modulation Strategies in R410A Chiller with Focus on Seasonal Performance (West Lafayette, IN, 2022) (Paper 1198)