Advanced exergy analysis of transcritical CO₂ heat pump system based on experimental data

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Abstract. Conventional exergy analysis identifies component exergy destruction, however, does not give clear picture whether occurs because of component itself or due to remaining components and then the advanced exergy analysis is emphasized. The main purpose of this study is to apply advanced exergy analysis to the transcritical CO₂ vapor compression heat pump system for simultaneous water cooling and heating, along with the conventional exergy analysis. The novelty of the present study is the advanced exergy analysis (with modified methodology) of transcritical superheat cycle based on experimental data. Endogenous, exogenous, avoidable and unavoidable exergy destructions are determined for each component of this system. Results show that compressor is having highest exergy destruction among the CO₂ system components and also its major part is avoidable endogenous exergy destruction. According to conventional exergy analysis, the sequence for improvement seems to be compressor, gas cooler, expansion valve and evaporator; however, the in-depth analysis (advanced exergy analysis) shows compressor is the first component followed by evaporator, gas cooler and expansion valve to be considered for improvisation. Thus it is concluded that compressor provides highest opportunity for improvement and new design of the same will improve overall transcritical CO₂ system performance.

Keywords. Transcritical heat pump; carbon dioxide; exergy analysis; experimentation; irreversibility; system improvement.

1. Introduction

The exergy analysis is useful in identifying the causes, locations and magnitudes of process inefficiencies. Although the common perception is that the exergy analysis is application of second law of thermodynamics, its uses and principles are applicable to a wide variety of engineering problems. Exergy analysis allows for improvement not necessarily attainable via energy methods, like increased efficiency, reduced fuel use and environmental emissions, and cost savings. Thus, the results of exergy analysis can be used to determine the total exergy destruction within a component. However, the total exergy destruction occurring in a component is not only due exclusively to the component but is also caused by the inefficiencies within the remaining system components. The splitting of exergy destruction into endogenous and exogenous exergy destruction parts is nothing but advanced exergetic analysis. Endogenous exergy destruction of a particular component of a system is caused because of component's own irreversibilities. Exogenous part of exergy destruction is caused because of irreversibilities in

other components of the system that cause performance degradation in the component under consideration. Alternatively, splitting the exergy destruction into unavoidable and avoidable parts in a component provides a realistic measure of the potential for improving the thermodynamic efficiency of that component. With the combination of 'avoidable and unavoidable exergy destruction' and 'endogenous and exogenous exergy destruction', engineers should be able to consider interactions among system components and improvement possibilities of each component and then modify or design the system accordingly. They will be able to attack the component for improvement with enhanced conviction.

Morosuk and Tsatsaronis [1] proposed the cycle (thermodynamic) method of splitting the exergy destruction into endogenous and exogenous parts—applicable to refrigeration machines. They applied same method of advanced exergy analysis to the absorption refrigeration machine [2]. Refrigeration system utilizing R22 was analyzed with thermodynamic and engineering approaches by Kelly *et al* [3]. Morosuk and Tsatsaronis [4] discussed advanced exergetic evaluation of refrigeration machines using different working fluids and applied to vapor-compression refrigeration machines using different one-component

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working fluids (R125, R134a, R22 and R717) as well as azeotropic (R500) and zeotropic (R407c) mixtures. Erbay and Hepbasli [5] performed advanced exergy analysis of heat pump dryer for food drying. Mosaffa *et al* [6] applied thermodynamic approach to air-conditioning system incorporating thermal energy storage. Gong and Boulama [7] performed advanced exergy analysis of water–lithium bromide absorption refrigeration system. Mehrpooya and Ansarinassab [8] applied advanced exergoeconomic analysis on three multi stage mixed refrigerant liquefaction processes.

In view of environmental concern, the transcritical CO₂ vapor compression heat pump cycle has been emerged as valuable alternative for various cooling and heating applications [9]. However, studies on experimental exergy analysis of transcritical CO₂ vapor compression system are very limited. Tao *et al* [10] performed exergy analysis of transcritical CO₂ residential air-conditioning system with internal heat exchanger based on experimental data. Aprea *et al* [11] conducted exergy analysis of CO₂ refrigeration plant based on experimental data. Ma *et al* [12] experimentally investigated the effects of electronic expansion valve (EEV) opening on the exergy loss of different components in the transcritical CO₂ system. But, none of the research works above has discussed splitting of exergy destruction into endogenous or exogenous and avoidable or unavoidable parts for transcritical CO₂ heat pump system.

In this paper, effort is made towards analyzing the CO₂ transcritical vapor compression heat pump system with the modified thermodynamic approach of advanced exergy analysis. The experimentation was conducted on transcritical CO₂ heat pump prototype for simultaneous water heating and cooling and the splitting of exergy destruction into endogenous or exogenous and avoidable or unavoidable parts has been conducted based on experimental data.

2. Methodology

2.1 Basics of conventional and advanced exergy analyses

All real processes are irreversible due to effects such as chemical reaction, heat transfer through a finite temperature difference, mixing of matter at different compositions or states, unrestrained expansion, and friction. Hence, all components of a heat engine or heat pump system are suffered by exergy destruction or irreversibility and the irreversibility is responsible for the degraded performance of system. The exergy balance for k th component of a system is given by

$$\dot{E}_{f,k} = \dot{E}_{d,k} + \dot{E}_{p,k} \quad (1)$$

and for the overall system,

$$\dot{E}_{f,tot} = \dot{E}_{p,tot} + \sum_k \dot{E}_{d,k} + \dot{E}_{l,tot} \quad (2)$$

where the exergy destruction of k th component is given by

$$\dot{E}_{d,k} = T_0 \dot{m}_k s_{gen}. \quad (3)$$

Now, the exergy efficiency of k th component is given by

$$\varepsilon_k = \frac{\dot{E}_{p,k}}{\dot{E}_{f,k}} = 1 - \frac{\dot{E}_{d,k}}{\dot{E}_{f,k}}. \quad (4)$$

The ratio of exergy destruction to fuel exergy is also called exergy destruction ratio of a component [13]. Now, according to advanced exergy analysis, the exergy destruction of a component can be splitted into endogenous and exogenous parts. The endogenous exergy destruction is associated with the inefficiencies that occurs within the component and the exogenous exergy destruction is associated with the inefficiencies of the component which occurs due to rest of the components of a system [3], i.e.,

$$\dot{E}_{d,k} = \dot{E}_{d,k}^{EN} + \dot{E}_{d,k}^{EX}. \quad (5)$$

Thermodynamic method of advanced exergy analysis was used to determine endogenous and exogenous exergy destruction of each component [3]. When all the components of the system are considered to follow ideal process while the part under consideration is considered to follow real process, the exergy destruction of the component under consideration is called endogenous exergy destruction. Part of exergy destruction other than endogenous is nothing but exogenous exergy destruction. Endogenous part of exergy destruction appears because of irreversibilities in that component while the exogenous part appears because of irreversibilities in other components. The endogenous part of the exergy destruction in the k th component is calculated through analysis of the hybrid cycle. The hybrid cycle represents the theoretical cycle with irreversibilities in the k th component only. The number of the hybrid cycles that should be created for the analysis is equal to the number of the components in the overall system. Another outcome of advanced exergy analysis is to splitting the exergy destruction into avoidable and unavoidable parts. The exergy destruction, which can be reduced by design improvement up to technical limitation, is avoidable part and rest, which cannot be reduced due to technical limitation, is called unavoidable part.

2.2 Conventional exergy analysis of CO₂ heat pump system

The transcritical CO₂ refrigeration or heat pump cycle for simultaneous water cooling and heating is shown in figure 1. Various processes are as follows: 1–2: real compression, 2–3: cooling in gas cooler, 3–4: isenthalpic expansion, 4–1: evaporation in evaporator, 6–7: water heated in the gas cooler, 8–9: water cooled in the evaporator.

Figure 1. CO₂ heat pump system: schematic and T-s diagrams.

Conventional exergy analysis of transcritical CO₂ system is similar to subcritical heat pump system except the fact that condenser is replaced by gas cooler. Equations for calculation of exergy destruction in each component are shown below (based on Eq. 1). Exergy destruction is nothing but the difference between the values of exergy of fuel and exergy of product. For example, exergy of fuel for the compressor is work while exergy of product is $(\dot{E}_2 - \dot{E}_1)$.

$$\text{Compressor: } \dot{E}_{d,CM} = \dot{W} - (\dot{E}_2 - \dot{E}_1) \quad (6)$$

$$\text{Gas cooler: } \dot{E}_{d,GC} = (\dot{E}_2 - \dot{E}_3) - (\dot{E}_7 - \dot{E}_6) \quad (7)$$

$$\text{Evaporator: } \dot{E}_{d,EV} = (\dot{E}_4 - \dot{E}_1) - (\dot{E}_9 - \dot{E}_8) \quad (8)$$

$$\begin{aligned} \text{Throttling valve: } \dot{E}_{d,TV} &= (\dot{E}_3^M - \dot{E}_4^M) - (\dot{E}_4^T - \dot{E}_3^T) \\ &= \dot{E}_3 - \dot{E}_4 \end{aligned} \quad (9)$$

where M: Mechanical part and T: Thermal part.

Neglecting kinetic and potential energies, the physical exergy at any state i is given by

$$\dot{E}_i = \dot{m}_i e_i = \dot{m}_i [(h_i - h_0) - T_0(s_i - s_0)]. \quad (10)$$

2.3 Advanced exergy analysis of CO₂ heat pump system

Thermodynamic or cycle method has been used to splitting the exergy destruction for transcritical CO₂ vapor compression system. The same method already has been used for subcritical vapor compression system [3]. However, two major modifications have been done for transcritical CO₂ system:

- (i) In the earlier study, saturation cycle (evaporator exit is saturated) was used, leading to a single pinch point in evaporator (inlet of refrigerant). In this study,

Figure 2. Actual and theoretical cycles for CO₂ heat pump system.

superheated cycle (realistic cycle) is used and hence the pinch point of heat transfer in evaporator may occur either at refrigerant inlet or exit.

- (ii) In the earlier study, due to subcritical nature of heat rejection in condenser, pinch point was occurred at saturated vapor point of refrigerant. Whereas due to supercritical nature of heat rejection in gas cooler, it is very difficult to locate the pinch point by graphical method and hence a novel numerical method has been used for pinch point selection. Whole temperature zone of gas cooler is discretised into large number of sub-zones by using enthalpy rate equation with the help of an iterative technique and then the pinch point (of minimum temperature difference between CO₂ and water) has been detected.

Actual cycle and theoretical cycle have been constructed on T-s plot of CO₂, generated by NIST REFPROP. Actual cycle has been constructed by using experimental data. The corresponding theoretical cycle has been constructed in figure 2 by procedure as follows. Temperature in the evaporator is assumed with $\Delta T_{\min} = 0$ to determine theoretical evaporation. Keeping degree of superheat same in the evaporator, point 1 is obtained. $\Delta T_{\min} = 0$ for gas cooler is assumed. As the theoretical process for compression is isentropic, isentropic compression is considered from point 1. Pressure drops in the heat exchangers are assumed to be negligible. While calculating the value of endogenous exergy destruction in a component, the effect that the small exergy destruction within some theoretical heat exchanger has on these values is neglected [3]. Point 2 is obtained where compression process cuts the ideal gas cooling process. A throttling process is always an irreversible one and never can be described as an ideal or theoretical process. For creating the theoretical cycle of a vapor-compression heat pump machine, the throttling process is replaced by an ideal expansion process, which cuts ideal evaporation at point 4.

Figure 3. Hybrid cycle for compressor: CO₂ heat pump system.

To determine the endogenous exergy destruction of compressor, compression process must be kept real while other processes are assumed to be ideal. Compression process, keeping isentropic efficiency constant, is plotted from point 1, while all the other processes will be theoretical. The hybrid cycle for compression is shown in figure 3. Similarly, hybrid cycles are created for other components of the system. In figure 4, the cycle 1R-2R-3R-4R is the real cycle for CO₂ refrigeration system and the theoretical cycle 1T-2T-3T-4T is shown by dashed line while construction for hybrid cycle is shown by dotted line. As there are four components in the system, four hybrid cycles have been plotted: cycle 1T-2H-3T-4T is the hybrid cycle for compressor, cycle 1T-2H*-3R-4H is hybrid cycle for gas-cooler, cycle 1T-2T-3T-4H* is hybrid cycle for throttling valve, cycle 1R-2H*-3T-4H** is hybrid

Figure 4. Real, hybrid and theoretical cycles: CO₂ heat pump system.

cycle for evaporator. Exergy destruction of compressor in the hybrid cycle of compressor will be its endogenous exergy destruction. Similarly endogenous exergy destruction of each component is obtained. Avoidable and unavoidable exergy destructions in components have been also evaluated by cycle method [4] in modified form as discussed above. For this, compressor is assumed to perform with unavoidable thermodynamic efficiency while all the other components operate in an ideal way. Endogenous unavoidable exergy destruction of k th component has been evaluated as the exergy destruction occurring within the k th component when that component operates with its maximum attainable efficiency (here, isentropic efficiency for compressor and pinch point temperature difference for evaporator and gas cooler, which strongly influences value of unavoidable part) due to technical limitations and all the other components operate in ideal way. After finding endogenous unavoidable exergy destruction, other parts have been found by

$$\dot{E}_{d,k}^{EX,UN} = \dot{E}_{d,k}^{UN} - \dot{E}_{d,k}^{EN,UN} \quad (11)$$

$$\dot{E}_{d,k}^{EN,AV} = \dot{E}_{d,k}^{EN} - \dot{E}_{d,k}^{EN,UN} \quad (12)$$

$$\dot{E}_{d,k}^{EX,AV} = \dot{E}_{d,k}^{AV} - \dot{E}_{d,k}^{EN,AV} \quad (13)$$

2.4 Verification of the model

To verify the methodology, example of NH₃ vapor compression refrigeration system given by Tsatsaronis and

Table 1. Advanced exergy analysis of each component of NH₃ refrigeration system.

	Component	Reference value	Calculated value	% Deviation
$\dot{E}_f$	CM	34.9	34.866	-0.0986
	CN	8.92	8.917	-0.0307
	TV	21.82	21.801	-0.0890
	EV	18.14	18.186	0.2537
$\dot{E}_p$	CM	29.92	29.979	0.1966
	CN	1.14	1.138	-0.1852
	TV	18.95	18.925	-0.1316
	EV	11.41	11.419	0.0770
$\dot{E}_d$	CM	4.98	4.887	-1.8719
	CN	7.78	7.779	-0.0080
	TV	2.87	2.876	0.1924
	EV	6.73	6.767	0.5534
$\dot{E}_d^{EN}$	CM	3.48	3.761	8.0747
	CN	4.59	4.510	-1.7429
	TV	1.35	1.560	15.5556
	EV	6.72	6.767	0.7030
$\dot{E}_d^{UN}$	CM	0.97	1.470	51.5812
	CN	2.74	2.762	0.8034
	TV	2.17	2.108	-2.8734
	EV	2.363	2.328	-1.4874

Figure 5. Experimental setup transcritical CO₂ heat pump prototype.

Morosuk [13] is studied and results are obtained with the modified thermodynamic approach. The results of advanced exergy analysis are compared in table 1. It can be said that deviation is almost negligible except in the case of E_d^{UN} of compressor, which is 51%. This may be due to the assumptions required to determine the unavoidable part of E_d are not mentioned in Tsatsaronis and Morosuk [13]. In this study, practice of taking 95% isentropic efficiency of compressor and 1°C pinch point temperature difference are considered to determine E_d^{UN} . Method used in Tsatsaronis and Morosuk [13] is based on approach of analytical equations, while in this study, thermodynamic approach based on cycle method is used.

3. Results and discussion

The advanced exergy analysis has been applied to the transcritical CO₂ heat pump prototype, developed for simultaneous water cooling and heating. The layout of the experimental setup with full instrumentation is shown in figure 5. The details of prototype development and

Table 3. Throttling process.

Property	3	4
h_0	517.21	517.21
s_0	2.7800	2.780
T	315.5	285.2
P	10.40	3.80
h	320.16	327.44
s	1.380	1.440
h_x	299.94	480.34
s_x	1.310	2.000
Mech. exergy	238.69	204.05
Thermal exergy	0.140	21.11

description, test procedure, instrument accuracies and measurement uncertainty analysis have been published earlier [9]. Prior to the analysis, the experimentation has been conducted and properties of all the state points have been measured at the steady state condition. Uncertainties of enthalpy and entropy were estimated to be 2.6% and 2.8, respectively, and uncertainty of exergy was estimated as 4.7%.

Tables 2 and 3 show the calculation of physical exergy at each points of the cycle. Exergy destruction (irreversibility) of the components, calculated based on conventional exergy analysis is shown in table 4. As shown, the exergy destruction is highest in compressor and lowest in evaporator. This may be due to the range of pressure over which compressor is operating is quite large causing considerable pressure drop at inlet and outlet, which leads to lower isentropic efficiency (isentropic efficiency of compressor obtained with experiment is 66.7%). According to conventional exergy analysis, improving the compressor will improve the system performance significantly. However, it does not discuss what portion of exergy destruction is due to the irreversibilities of compressor itself and what portion of exergy destruction is due to irreversibilities of other components. Thus advanced exergy analysis has to be performed.

Table 2. Exergy analysis: transcritical CO₂ heat pump system.

Point	T (K)	p (MPa)	Quality	h (kJ/kg)	s (kJ/kg K)	$\dot{m}$ (kg/s)	e_i (kJ/kg)
Ambient (CO ₂)	311.4	0.1	—	517.210	2.777	—	—
1	302.8	3.8	1	469.473	1.968	0.012	204.20
2'	429.7	10.4	1	578.189	2.109	0.012	268.77
3	315.5	10.4	0	320.160	1.377	0.012	238.83
4	285.2	3.8	0.508	320.160	1.444	0.012	217.88
Ambient (water)	311.4	0.1	—	160.301	0.549	—	—
6	311.4	0.1	—	160.301	0.549	0.017	0.00
7	347.4	0.1	—	310.935	1.007	0.017	8.09
8	305.8	0.1	—	136.897	0.473	0.025	0.21
9	292.6	0.1	—	81.705	0.289	0.025	2.47

Table 4. Exergy analysis of each component.

Component	$\dot{E}_f$ (kW)	$\dot{E}_d$ (kW)	$\dot{E}_p$ (kW)	ε
Compressor	1.321	0.5365	0.784	0.594
Gas cooler	0.364	0.2288	0.135	0.371
Throttling valve	0.421	0.1661	0.255	0.605
Evaporator	0.166	0.1097	0.057	0.340
For cycle	1.321	1.0411	0.057	0.212

The endogenous and exogenous exergy destructions of each component of the system are shown in table 5 and figure 6. Endogenous exergy destruction of compressor is more which means that compressor has more irreversibilities and reducing them will improve the performance of compressor and hence the system. Throttling process is always irreversible, but a proper throttling valve may improve the performance. Theoretically throttling valve must be replaced by turbine for isentropic expansion. Exogenous exergy destruction of gas cooler is more which means that exergy destruction in gas cooler is more due to irreversibilities of other components rather than gas cooler itself. Hence less attention is advisable to the improvement of gas cooler. Evaporator has almost all its exergy destruction because of its own irreversibilities (keeping in mind that evaporator operates at a temperature lower than atmospheric). Hence, although total exergy destruction of evaporator is less as compared to the other components, endogenous part is more compared to that of gas cooler as shown in figure 7.

Results show that avoidable endogenous exergy destruction of compressor is maximum (30%) and that of expansion valve is minimum ($\sim 0\%$). Hence, the compressor is the first and expansion valve is the last component to be considered for improvisation. Some part of improvement can be observed from improvement in other components, since avoidable exogenous exergy destruction in compressor is significant, 16% of system exergy destruction (figure 8). Although gas cooler has second highest exergy loss, scope for improvement is less due to significantly lower avoidable endogenous part. Hence, the advanced exergy analysis yields in-depth information for system improvement option. Based on the present data, improvement option for compressor is 91% followed by 89, 26 and 18% for gas cooler, expansion valve and evaporator, respectively.

Table 5. Splitting into $\dot{E}_d^{EN}$ and $\dot{E}_d^{EX}$ for CO₂ heat pump system.

Component	$\dot{E}_d$ (kW)	$\dot{E}_d^{EN}$ (kW)	$\dot{E}_d^{EX}$ (kW)	% of $\dot{E}_d^{EN}$ in $\dot{E}_d$
Compressor	0.536	0.3678	0.1682	68.6194
Gas cooler	0.229	0.03011	0.19889	13.14847
Throttling valve	0.166	0.1088	0.0572	65.54217
Evaporator	0.1096	0.1096	0	100

Figure 6. Pie chart showing $\dot{E}_d^{EN}$ and $\dot{E}_d^{EX}$ in each component of CO₂ heat pump system. **Figure 7.** $\dot{E}_d^{EN}$ and $\dot{E}_d^{EX}$ (% of total exergy destruction for CO₂ system).

Figure 8. Various exergy destruction for CO₂ system.

4. Conclusions

Conventional and advanced exergy analyses of transcritical CO₂ heat pump system have been performed based on experimental data. From the results and discussion, the following conclusion can be made:

- Exergy destruction of compressor is highest while that of evaporator is lowest. Thus according to conventional exergy analysis, modification or new design of the compressor is required to improve overall performance of system.
- Endogenous exergy destruction of compressor is 68%, this means that it is required to improve the design of compressor itself rather than other components.
- Exogenous exergy destruction in gas cooler is more as compared to endogenous exergy destruction which means that contribution of other components on irreversibility of gas cooler may be more than gas cooler itself.
- Endogenous exergy destruction in throttling valve is more and hence proper throttling device can be selected.
- Endogenous exergy destruction of evaporator is 100% which means that exergy destruction is solely due to irreversibilities in the evaporator itself.
- As avoidable endogenous exergy destruction is maximum for compressor and minimum for expansion valve, the compressor is the first and expansion valve is the last component to be considered for improvisation.

Nomenclatures

p	pressure (MPa)
T	temperature (K)

$\dot{E}$	exergy rate (kW)
e	specific exergy (kJ/kg)
$\dot{m}$	mass flow rate (kg/s)
h	specific enthalpy (kJ/kg)
s	specific entropy (kJ/kg K)
s_{gen}	specific entropy generation (kJ/kg K)
ε	exergetic efficiency
$\dot{W}$	power (kW)
ΔT_{min}	pinch point temperature difference (K)

Subscripts

CM	compressor
GC	gas cooler
TV	throttling valve
EV	evaporator
k	kth component
0	property at dead state
f	fuel
d	destruction
p	product
tot	overall system
l	losses
x	property at (T ₀ , p)

Superscripts

AV	avoidable
EN	endogenous
EX	exogenous
UN	unavoidable

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