Advance thermodynamic analysis of a CO_2 compression system using heat-pump for CCS

Hafiz Ali Muhammad[a], Beomjoon Lee[b], Haider Sultan[c], Muhammad Imran[d] and Young-Jin Baik[e]

[a] Renewable Energy Engineering, University of Science and Technology, Daejeon 305-350, Republic of Korea, hafiz@kaist.ac.kr
[b] Thermal Energy Systems Laboratory, Korea Institute of Energy Research, Daejeon 305-343, Republic of Korea, beomjoon.lee@kier.re.kr
[c] Renewable Energy Engineering, University of Science and Technology, Daejeon 305-350, Republic of Korea, haidersultan@ust.ac.kr
[d] Mechanical Engineering and Design, School of Engineering and Applied Sciences, Aston University, B4 7ET, Birmingham, UK
[e] Thermal Energy Systems Laboratory, Korea Institute of Energy Research, Daejeon 305-343, Republic of Korea, twinjin@kier.re.kr, CA

Abstract:

CO_2 pressurization represents a significant portion of energy penalty resulting from CO_2 capture in carbon capture and storage process (CCS). Effective measures to improve the pressurization scheme directly translates into an improvement of plant economics. In this paper we aim to reduce the energy expenditure of CO_2 pressurization step in the CCS process by assisting the CO_2 multi-stage compressors with a heat-pump system. The CO_2 compressors raised CO_2 pressure to an intermediate liquefaction pressure. Afterwards, the CO_2 is liquefied in the heat-pump and subsequently pumped to the high pressure instead of being compressed. In addition to the conventional energetic analysis, the advance exergy analysis of the heat-pump assisted pressurization scheme is also presented. The advance exergy analysis of the proposed strategy reveals the optimum operating parameters and design of the heat pump system. Through the implementation of results obtained by the advance exergetic analysis, it was found out the proposed system can achieve 6.21% saving in electric power.

Keywords:

Carbon capture and storage, CO_2 pressurization, Supercritical CO_2 power cycle, Advance exergetic analysis

1. Introduction

Carbon capture and storage (CCS) is a widely acknowledged technique to mitigate the anthropogenic CO_2 emissions to the environment and consequentially global warming [1, 2]. CCS also has a strategic importance since it enables the continuous use of fossil fuels such as coal in the power sector by sequestering the produced CO_2. Despite the broad consensus on CCS as a leading technology to decarbonize the power and industrial sector, the significant energy penalty associated with CCS process has vastly deterred the implementation of CCS to the power and industrial sector [3, 4].

The CCS process framework involves separation of carbon content of either fuel (pre-combustion) or exhaust (post-combustion), CO_2 transportation and subsequent storage to the underground repositories. The preferred mode of CO_2 transportation from capture to storage sites, which are normally separated by a distance of 1000 kms, is pipelines [5, 6]. The CO_2 pressure required for pipeline transportation ranges from 150—200 bar, while, the captured CO_2 using state of the art capturing techniques is at near atmospheric pressures (1.2—3.5 bar) [7]. Therefore, in the first stage of transportation, the pressure of captured CO_2 is boosted to the pipeline pressures. The major portion of CCS process penalty, indeed comes from the CO_2 capture or separation process. However, the pressurization of large volumes of CO_2 over such high pressure ratios can account for an efficiency penalty of as high as 12% of the total loss of power plant efficiency [8].

The literature is rich in performance assessment and process intensification measures of the CCS process as indicated in [3, 4, and 9]. Due to the sizable contribution of CO_2 pressurization to the overall process penalty, some researchers have exclusively focussed on improving CO_2 compression, transportation and storage chain. Zhang and Huisingh [10] reviewed the CO_2 storage schemes and conduct a techno-economic assessment of various CO_2 storage methods and repositories. Their research concluded the high cost and significant energy penalty are still the main barriers to the deployment of CO_2 storage schemes. Witkowski et al. [5] investigated CO_2 compression and transportation from the viewpoint of safety issues and environmental concerns in case of CO_2 leakage from the pipeline. They examined the effects of ambient temperature and thickness of thermal insulation layer on the flow of CO_2 in the pipelines and determined the locations of subsequent CO_2 booster stations.

The design of CO_2 compressors capable of handling high flowrates such in CCS process and assessing the improvement potential of CO_2 compression chain and has also been examined. Moore et al. [11] designed and fabricated a centrifugal compressor with internal intercooling of CO_2. They optimized the design of cooling jacket embedded within the centrifugal compressors to enhance the heat transfer without inducing additional pressure drop. The same group in another research activity [12] optimized the thermodynamic path of CO_2 compression for an Integrated Gasification Combined cycle (IGCC) and concluded a 35% reduction in power compared to a benchmark case. Their optimum path involves compressing CO_2 to an intermediate liquefaction pressure using multi-stage CO_2 compressors, then liquefying it and achieving the reaming pressure rise up to the pipeline pressure using CO_2 pump. The optimum thermodynamic path of CO_2 compression depends on the plant type and separation technique [13], and due to the CO_2 separation at multiple pressure levels and the availability of low temperature N_2, the IGCC plant differs significantly from natural gas combined cycle (NGCC) or coal fired power plant. Thereby, Botero et al. [14] focussed on natural gas combined cycle (NGCC) and conducted a thermodynamic and preliminary economic feasibility analysis of CO_2 compression strategies for a NGCC. They also concluded that the complementing CO_2 multi-stage compressor with liquefaction and pumping successfully reduces the required electric power. Assisting the CO_2 compressors with liquefaction and pumping is thus an established path to reduce the required power. However, the CO_2 requires sub-zero condensing temperatures to liquefy and thus refrigeration cycles are required for CO_2 liquefaction. Therefore, Alabdulkarem et al. [15] designed and optimized vapour compression cycle (VCC) for CO_2 liquefaction and achieved 5.1% reduction in power using NH_3 as the refrigerant.

During the CO_2 compression, a considerable amount of low grade heat during intercooling is available, and therefore, any optimum design targets the reusing of that heat to produce power. Romeo et al. [16] integrated the intercooling heat to the low pressure steam and optimized the intercooling compression from the energetic and economic view point. Organic Rankine Cycle has proven to be a promising cycle to recover the low to mid temperature heat source [17]. Therefore, some research activities focussed on recovering the intercooling heat by integrating the CO_2 compression chain with ORC [17, 18]. Recently, a new concept of integrating the CO_2 compression with liquefaction and pumping with supercritical CO_2 power cycle was also investigated and showed a reduction in 13.88% saving in power using the proposed design [19].

In the light of previous discussion this study is set out to examine and optimize a novel CO_2 compression strategy. The strategy involves multi-stage compressors and a heat pump (HP) system for CO_2 liquefaction. Distinct from the previous work, the advance exergy analysis of the HP system for CO_2 liquefaction in CCS process is carried out. The advance exergy analysis is a strong tool to study the interdependencies between the system components, identify the location of thermodynamic inefficiencies, and the potential measure that can be taken to improve the system [20, 21]. The potential measures are not only identified using the advance exergy analysis but are implemented on the proposed HP system and the resulting improvement in performance of the system is shown.

Therefore, the novelty of this study includes designing and optimizing a new CO_2 pressurization strategy by the aid of advanced exergy analysis.

2. System description

2.1 Initial data and benchmark case

A NGCC was selected for the analysis of CO_2 pressurization strategy and the boundary conditions taken throughout the analysis are summarized in Table 1. For the analysis presented here it was assumed the system is in steady state and the pressure losses across the pipes and the HX's are negligible.

Table 1. Boundary conditions for CO_2 pressurization.

Parameters	*Values*
Plant type	NGCC ~400 MW
Compressor & pump isentropic efficiency	0.80& 0.85
CO_2 mass flow rate (m_{CO2})	37.5 kg/s
Pinch point temperature (PP)	4 K
CO_2 captured pressure (P_{In})	1.9 bar
CO_2 target pressure (P_{Out})	160 bar
Cooling water temperature in ($T_{CW,In}$)	293 K
Cooling water temperature out ($T_{CW,Out}$)	298 K
Reference state temperature (T_o) and pressure (P_o)	288 K & 101.325 kPa

The conventional mean of multistage compression was established as a benchmark case, and the performance improvement of the new design with respect to the benchmark case was quantified and reported. For the benchmark case 4 stages of compression were considered as shown in Fig. 1. The pressure ratio (P_R) across each stage is evenly distributed to achieve the desired pressure rise from the captured pressure (P_{In}) to the desired pressure (P_{Out}).

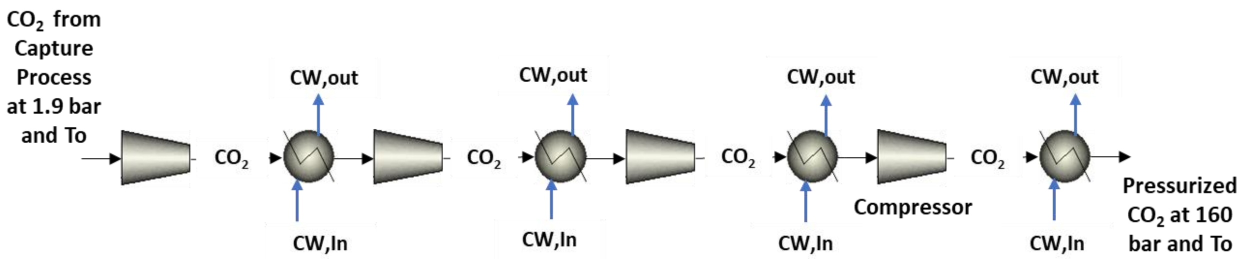

Fig. 1. Benchmark case for CO_2 pressurization.

2.2 Proposed design

In the proposed design the multi-stage compressors are assisted with the HP system to liquefy CO_2 as shown in Fig. 2. The incoming CO_2 from the capture unit is at 1.9 bar, while the triple point pressure of CO_2 is 5.17 bar [19]. Therefore, the initial two stages of compressor are necessary to raise the CO_2 pressure greater than the triple point pressure before it can be liquefied. After the initial two stages, the remaining stages are replaced by a HP system and a pump in the proposed design.

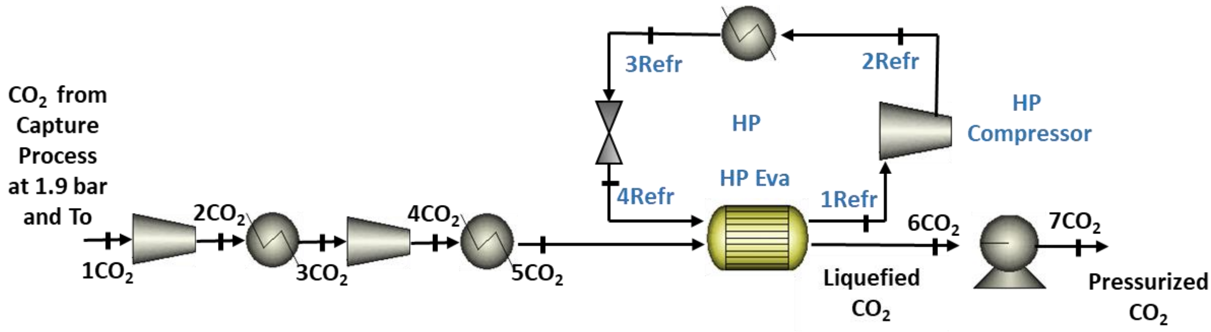

Fig. 2. Proposed design for CO_2 pressurization.

3. Modelling and analysis

3.1. Energetic or first law analysis

In the benchmark case the net power consumed for CO_2 pressurization (W_{BC}) is the sum of individual compressors, $W_{BC} = \Sigma W_{Comp}$, while the intercooling is achieved using the cooling water in the intercoolers. For the proposed case shown in Fig. 2, the net power consumed ($W_{Net,PC}$) is the sum of initial two multi-stage compressors ($W_{CO2,Comp}$), the HP's compressor work ($W_{Refr,Comp}$), and the CO_2 pump ($W_{CO2,Pump}$), and is:

$$W_{Net,PC} = W_{CO2,Comp} + W_{CO2,Pump} + W_{Refr,Comp} \quad (1)$$

The initial two stages of compressors raised CO_2 pressure to the liquefaction pressure (P_{5CO2}) at State 5CO$_2$. After the compression until P_{5CO2}, the CO_2 is cooled down using the ambient intercooler before it is fed into the HP's evaporator. In the evaporator of the HP the heat (Q_{Eva}) is transferred from CO_2 to the refrigerant. For the HP design, it was assumed CO_2 is saturated liquid at State 6CO$_2$, therefore the saturation temperature of CO_2 in the evaporator (T_{6CO2}), State 6CO$_2$ and Q_{Eva} are fixed. The saturation temperature of HP's refrigerant in the evaporator is consequentially determined by the T_{6CO2} and the pinch point (PP) limit. After identifying the saturation state of the refrigerant in the evaporator, the State 1Refr is determined by assuming a degree of SH at the compressor's inlet. The refrigerant is designed to reject its heat to the ambient, therefore, the saturation state of the refrigerant in the condenser is determined by the cooling water temperature. With the known saturation pressure in the condenser and isentropic efficiency of the compressor (η_{Comp}), State 2Refr is solved. The State 3Refr and 4Refr are determined by assuming the saturated liquid state at 3Refr and isenthalpic expansion. With the known state properties and Q_{Eva}, the required mass flow rate of the refrigerant (m_{Refr}) is calculated. The methodology to solve the proposed design is given in Table 2.

Table 2. Calculation methodology.

Step	*State*	*Temperature*	*Pressure*	*Enthalpy*	*Comment*
1	1Refr	$T_{1Refr} = f(T_{Sat,Eva}, SH)$ $T_{Sat,Eva} = f(T_{6CO2}, PP)$	$P_{1Refr} = P_{Sat,Eva}$	$f(P_{1Refr}, T_{1Refr})$	SH is assumed.
2	2Refr	$T_{2Refr} = f(T_{Sat,Cond}, \eta_{Comp})$ $T_{Sat,Cond} = f(T_{CW,Out}, PP)$	$P_{2Refr} = P_{Sat,Cond}$	$f(P_{2Refr}, T_{2Refr})$	Compressor isentropic model.
3	3Refr	$T_{3Refr} = T_{Sat,Cond} = T_{CW,Out} + PP$	$P_{3Refr} = P_{Sat,Cond}$	$f(P_{3Refr}, X=0)$	Where quality (X) is zero.
4	4Refr	$T_{4Refr} = T_{Sat,Eva}$	$P_{4Refr} = P_{Sat,Eva}$	$f(P_{4Refr}, h=h_{3Refr})$	Isenthalpic expansion.

The discretization scheme is used for the modelling of heat exchangers whose detail can be found in [22] and the thermodynamic properties of CO_2 and R290 were obtained from the REFPROP 8.0 database [23].

3.2. Conventional exergetic analysis

An exergy based analysis requires defining the product (Ex_P) and fuel (Ex_F) exergy. The product and fuel exergy are defined based on the purpose of the component. The exergetic balance for a kth component is:

$$Ex_F = Ex_P + Ex_D \quad (2a)$$

Where the measure of irreversiblities in a component are reflected in destruction of exergy (Ex_D). The exergetic efficiency (η_{Ex}) for the kth component is:

$$\eta_{Ex} = Ex_P / Ex_F \quad (2b)$$

While for the overall system:

$$Ex_{F,Total} = Ex_{P,Total} + \sum Ex_D + Ex_{L,Total} \quad (2c)$$

Here the exergy loss (Ex_L) is the exergy that is not used within the system and is transferred to the environment. For the benchmark case, the performance results of the first law and the exergetic analysis are summarized in Table 3.

Table 3. Energetic and exergetic balance of the benchmark case.

Parameter	*Value*	*Comments*
$Ex_{F,Total}$	11613.42 kW	The sum of the power consumed by all the compressors in Fig. 1 is the Ex_F for the benchmark case. Thus, $Ex_{F,Total} = W_{BC}$
$Ex_{P,Total}$	6530.94 kW	The increase in exergy of the CO_2 is the desired output of the compressors and, thus, $Ex_{P,Total} = Ex_{CO2,In} – Ex_{CO2,Out}$
$Ex_{D,Total}$	1721.20 kW	The exergy is destroyed during the compression, and the total sum is therefore $Ex_{D,Total}$.
Ex_L	3361.28 kW	The transfer of exergy in the intercoolers is transferred to the surroundings, and not used within the system, therefore it is the Ex_L.
$\eta_{Ex,BC}$	56.24%	System exergetic efficiency.

For the new design given in Fig.2, in addition to the same CO_2 compressors and intercoolers as in the benchmark case, there is a HP system for CO_2 liquefaction. Table 4 summarizes the thermodynamic states of CO_2 and the refrigerant in the new design. For the new design, as can be seen from the Table 4, the reference temperature is crossed in all the components of the HP except condenser.

For a component operating below the reference temperature or if the temperature is crossed, the exergetic balance will take into account the fact at these conditions, the exergy of the stream will be higher if its temperature is lower. Therefore, at these states the exergy of the stream is split into thermal (Ex^T) and mechanical (Ex_M) parts [21]. The definition of Ex_F and Ex_P according to the SPECO approach [25] for the new design is given in Table 5.

Table 4. Thermodynamic state of the CO_2 and the refrigerant in the new design.

State	*m [kg/s]*	*P [kPa]*	*T [K]*	*h [kJ/kg]*	*s [kJ/kg-K]*	*Ex^T [kW]*	*Ex^M [kW]*	*Ex^{PH} [kW]*
$1CO_2$	37.5	190	298	504.88	2.61	NC*	NC	1278.63
$2CO_2$	37.5	575.56	397.18	592.19	2.66	NC	NC	4066.15
$3CO_2$	37.5	575.56	297	500.32	2.39	NC	NC	3496.04
$4CO_2$	37.5	1743.56	396.54	585.49	2.44	NC	NC	6215.30
$5CO_2$	37.5	1743.56	298	489.17	2.16	NC	NC	5621.13
$6CO_2$	37.5	1743.56	249.26	146.14	0.80	1824.08	5614.97	7439.06
$7CO_2$	37.5	16000	261.61	161.94	0.81	135.84	7795.88	7931.72
1Refr	6.51	132.20	245.26	1569.60	6.50	95.20	435.40	530.61
2Refr	6.51	1127.80	432.96	1970.52	6.69	1454.07	3152.40	4606.48
3Refr	6.51	1127.80	302.00	479.35	1.94	181.53	3152.40	3170.56
4Refr	6.51	132.20	245.26	479.35	2.05	2345.31	435.40	2780.70

Table 5. Product and fuel exergy balance of the proposed design.

Component	*Ex_F*	*Ex_P*
Evaporator [a]	$Ex^T_{4Refr} - Ex^T_{1Refr}$ [b]	$Ex^T_{6CO2} + Ex^T_{7CO2}$ [b]
Compressor [a]	$Ex^T_{1Refr} + W_{Refr,Comp}$	$Ex^T_{2Refr} + (Ex^M_{2Refr} - Ex^M_{1Refr})$
Condenser [c]	$Ex_{2Refr} + Ex_{3refr}$	$Ex_{CW,out} + Ex_{CW,In}$
Expansion valve [a]	$Ex^T_{3Refr} + (Ex^M_{3Refr} - Ex^M_{4Refr})$	Ex^T_{4Refr}
Overall system	$Ex_{F,Total} = W_{Refr,Comp}$	$Ex_{P,Total} = Ex^T_{6CO2} + Ex^T_{7CO2}$

[a] The reference temperature is crossed during the operation of this component.

[b] Since there is no pressure drop across the HX, Ex^M will remain same.

[c] Operates entirely above the reference temperature. Also the condenser is the dissipative component, however, the Ex_F and Ex_P for condenser are define only to implement the advance exergy analysis.

3.3. Advance exergetic analysis

Advance exergetic analysis is a strong tool to identify the improvement measures within a system. In this study we applied the advance exergetic analysis to the HP system shown in Fig. 2. The product exergy of the system remain constant as shown in the Table 3. In the advance exergy analysis, the Ex_D in each component is split into unavoidable ($Ex_{D,k}^{UN}$), avoidable ($Ex_{D,k}^{AV}$), endogenous ($Ex_{D,k}^{EN}$), and exogenous ($Ex_{D,k}^{EX}$) exergy destruction. With the aid of advantage exergy analysis the avoidable exergy destruction in a component is separated from the unavoidable part. Also, whether the exergy destruction is due to the component irreversibilities ($Ex_{D,k}^{EN}$) or is due to the system irreversiblities ($Ex_{D,k}^{EX}$) is determined using the advance exergy analysis. The equations required for the advance exergy analysis are summarized in the Table 6. With the help of results obtained through advance exergy analysis, the designer and the analyst can identify whether the improvement in system performance requires the improvement in the design of the kth-component ($Ex_{D,k}^{AV,EN}$) or the improvement of the other components ($Ex_{D,k}^{AV,EX}$). The detail implementation procedure of the

* NC: Not computed: For temperature greater than reference temperature (T_o), the Ex^T and Ex^M aren't computed.

advance exergetic analysis and the further insights about the significance of each exergy destruction category can be found in [24].

Table 6. Implementation of advance exergy analysis.

Equation	*Comment*
$Ex_{D,k}^{UN} = Ex_{P,k}^{Real}\left(\frac{Ex_{D,k}}{Ex_{P,k}}\right)^{UN}$	Where, the $(Ex_{D,k}/Ex_{P,k})^{UN}$ is calculated by simulating the system in which only unavoidable exergy destruction occurs.
$Ex_{D,k}^{UN} = Ex_{P,k}^{Real}\left(\frac{Ex_{D,k}}{Ex_{P,k}}\right)^{UN}$	
$Ex_{D,k}^{UN,EN} = Ex_{P,k}^{EN}\left(\frac{Ex_{D,k}}{Ex_{P,k}}\right)^{UN}$	To calculate the endogenous exergy destruction, a 'theoretical cycle' is simulated in which all the components, except the k-th component operates ideally. The number of theoretical cycles thus is equal to the number of components in the system.
$Ex_{D,k}^{UN,EX} = Ex_{D,k}^{UN} - Ex_{D,k}^{UN,EN}$	In the similar way, the $Ex_{D,k}^{AV,EN}$ and $Ex_{D,k}^{AV,EX}$ are solved.

Table 7 summarized the real (actual operating conditions used for the conventional analysis), unavoidable (with extremely high efficiency) and theoretical operating conditions (the theoretically maximum efficiency used to simulate the theoretical cycle) of all the components in the HP system [24].

Table 7. Values for the real, theoretical and unavoidable operating conditions.

Component	Real [24]	Theoretical [24]	Unavoidable [24]
Compressor isentropic efficiency	0.80	1	0.98
PP temperature difference during heat transfer	4	0	1
Expansion valve	Isenthalpic	Isentropic	Isenthalpic

4. Results and discussions

4.1. Selection of HP refrigerant

Propane (R290) and Ammonia (NH_3) are studied as the HP refrigerant, as both are considered as environmental friendly and have been investigated for the liquefaction of CO_2 in CCS process [15, 19]. The exergy destruction was split in unavoidable and avoidable part for the HP system for both the refrigerants and is given in Fig. 3. From the Fig. 3, we can see the major contribution to the exergy destruction for R290 is unavoidable, while for NH_3 the Ex_D^{AV} and Ex_D^{UN} are comparable. Thus, from the figure, we can deduce, the HP working with NH_3 has more potential for improvement and is thereby shortlisted for further investigation in this study. The same results were reported by [15, 19], after conducting the first law analysis of the HP system.

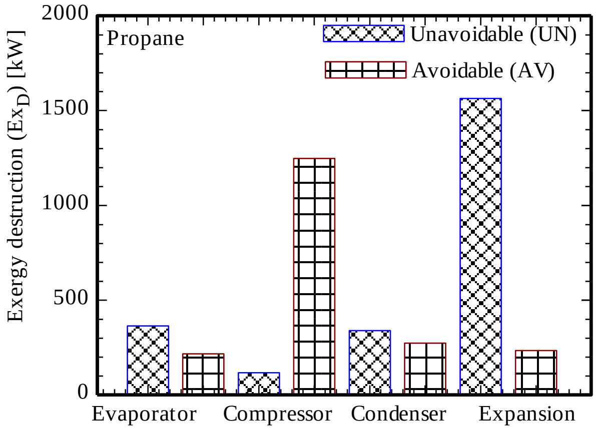

Fig. 3(a). Avoidable and unavoidable exergy destruction in all components for propane.

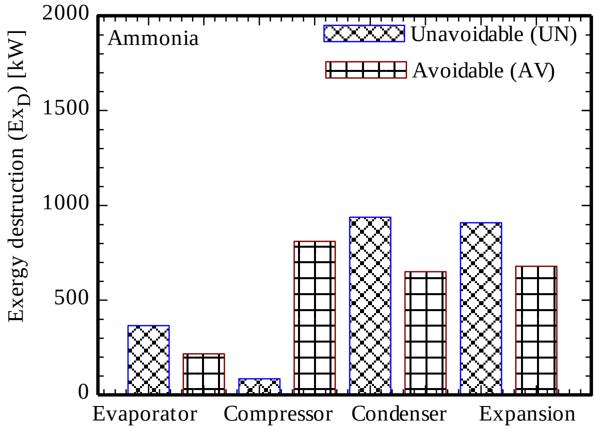

Fig. 3(b). Avoidable and unavoidable exergy destruction in all components for ammonia.

4.2. Implications of the advance exergy analysis

Fig. 4 shows the results of advance exergy analysis for the HP with NH_3 as the refrigerant. In the figure we present only the avoidable part of exergy destruction i.e. $Ex_{D,k}^{AV,EN}$ and $Ex_{D,k}^{AV,EX}$. Now we will discuss the implication of advance exergy analysis for the individual component.

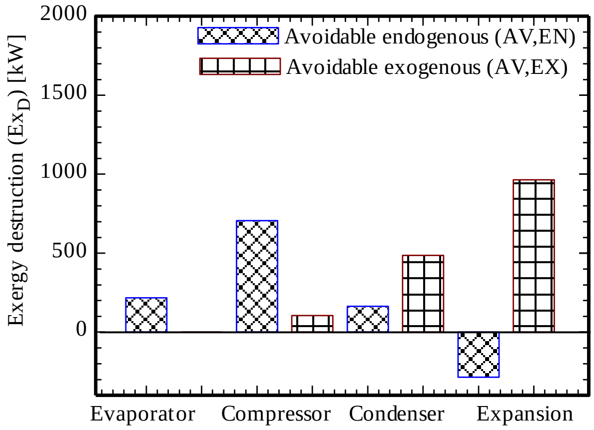

Fig. 4. Advance exergy analysis for NH_3 in all components.

For the evaporator the exergy destruction is endogenous, which is true for all the refrigeration machines [21]. The results imply any performance measure for the evaporator will require the improvement in the design of the evaporator, and is irrespective of the design of other components. The $Ex_{D,Eva}^{AV,EN}$ can be reduced if we increase the SH at the compressor inlet. Increasing the SH, will increase the specific refrigeration capacity of the refrigerant in the evaporator and thereby, decrease the m_{Refr}. However, when the SH is increased, the compressor outlet temperature would also increase which consequentially would increase the $Ex_{D,Cond}^{EX}$. Fig. 5 shows the impact of SH on the complete system, and since the increase in $Ex_{D,Cond}$ is more profound than the change in Ex_D for other components, therefore, the SH is undesirable for the HP system with NH_3.

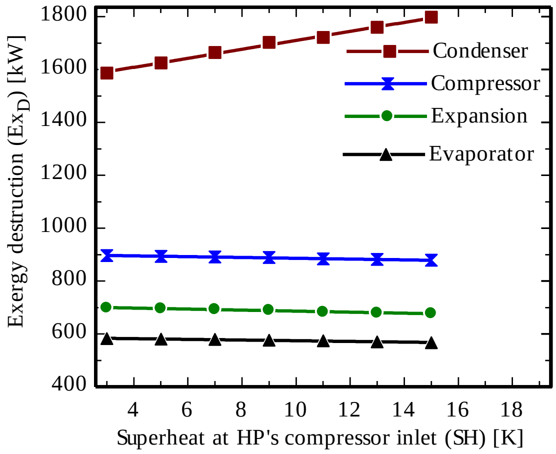

Fig. 5. Effect of SH on the HP system.

In case of compressor the major portion is $Ex_{D,Comp}^{AV,EN}$, which in turn implies increasing the isentropic efficiency of the compressor is the most desired performance improvement measure while any other measure wouldn't have any significant effect. For the condenser and the expansion valve the considerable portion of avoidable exergy destruction is exogenous. Therefore, the performance of these components can be significantly improved by taking measures that can improve the system's design rather improving the components themselves. One potential measure to improve the system is sub-cooling the refrigerant, but since the refrigerant is already at ambient state at 4Refr, therefore, the ambient cooling isn't viable for sub-cooling. However, if we look at the CO_2 path, the temperature of CO_2 at State $7CO_2$ is only 261.61 K, and therefore, it can be applied either to sub-cool the refrigerant or cool the incoming CO_2 at State $7CO_2$ before it enters the evaporator. The latter measure would have more profound impact on the system performance, since, it reduces the $Ex_{D,Eva}^{AV,EN}$ as well as $Ex_{D,AV}^{EX}$. Therefore, by the aid of the advance exergy results and their implications, we devised an improved strategy for the HP system and is presented in Fig. 6.

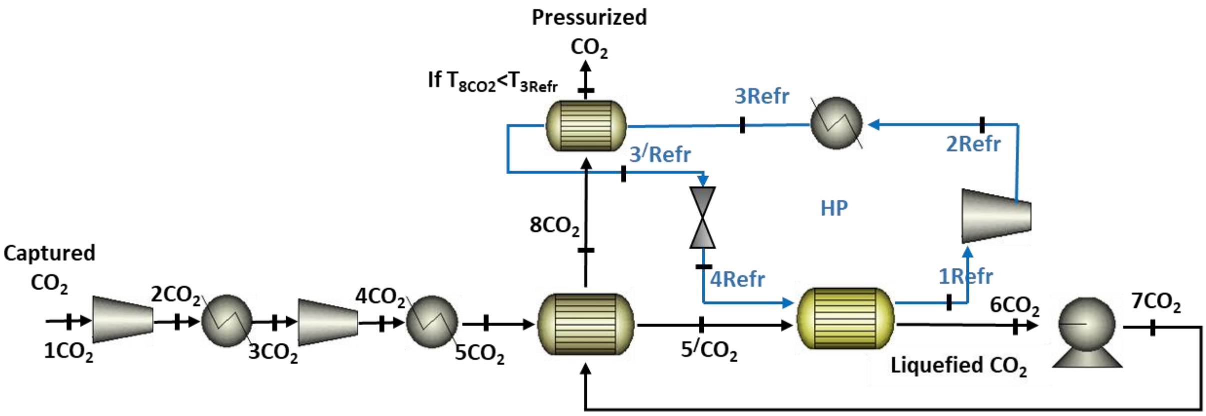

Fig. 6. Improved HP design for CO_2 liquefaction.

The improved design consumes 10892.37 kW electric power compared to the 11613.41 kW for the benchmark case and offers an exergetic efficiency of 59.83% compared to 56.24% for the benchmark case. The net effect of the proposed design, therefore, comes out to be 6.21% saving in the electric power.

Conclusions

The energy expenditure of CO_2 compression in CCS process is reduced by exploring and optimizing a new design for CO_2 pressurization using advance exergy analysis. The new design scheme involves CO_2 compressors which compresses CO_2 to an intermediate liquefaction pressure. The CO_2 is then liquefied in the evaporator of the heat pump and subsequently pumped to the target pressure. To optimize the performance of the heat-pump, the avoidable, unavoidable, endogenous and exogenous exergy destruction within each component of the heat-pump system is quantified. With the help of insights and results obtained through the advance exergy analysis: the performance parameters including the selection of the heat pump refrigerant, the superheat degree at the compressor inlet, and the design of the condenser is optimized in this study. This study by far demonstrates the strength of advance exergy analysis and through the information obtained through this analysis, the new design is optimized to achieve 6.21% saving in the electric power for CO_2 pressurization in CCS process. Furthermore, the advance exergy analysis couldn't indicate the measure to reduce the exergy losses. However, in the proposed design in this study a substantial portion of exergy is lost to the environment. Therefore, to improvise and investigate a new strategy that can utilize the exergy loss is currently underway.

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