Comparative study of Exergetic and Economic analysis of Multi-evaporator NH3 and NH3-CO2 CRS for a Seafood Processing Plant

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ABSTRACT

In a seafood processing plant, refrigeration is a vital and energy intensive process. In India, NH_3 is predominately used to cater to such refrigeration demands. These industrial refrigeration systems have the multi-evaporator configuration with high pressure ratios necessitating adoption of multistage compression. In recent years, owing to excellent and favorable thermo-physical properties of CO_2 in low temperature, adoption of NH_3-CO_2 cascade refrigeration system (CRS) with NH_3 in high temperature cycle is gaining worldwide acceptance. In the present work, performance of a conventional NH_3 multistage industrial refrigeration system is compared with a proposed NH_3-CO_2 CRS system in terms of energy, exergy and economic perspectives. Investigation is focused to thermodynamically evaluate and optimize the performance of the proposed cascade configuration with respect to operating parameters. A sensitivity analysis has been performed as well to study the effects of climatic conditions on system performance. The proposed NH_3-CO_2 CRS showed a maximum benefit of 13.3% in COP, 14% in exergy efficiency and 9% less TEWI compared to the baseline system. The estimated payback period is 25 months.
Keywords: *Industrial refrigeration, Multi-evaporator, NH_3, NH_3-CO_2 cascade, Exergy, Seafood*

1. INTRODUCTION

Seafood processing industries have cooling demands at various temperatures during processing, storage and transit. For example, storage of raw material at 0 °C, chilled water for washing at around 5 °C, quick freezing of product at -35 °C and storage of processed product at -20 °C. In general, about 20 kWh energy per ton of fish is spent for fish processing, in which freezing and cold storage consume 50-70% depending on the variety of fish and processing (FAO, 2015). NH_3 vapor compression refrigeration systems are widely used in India in industrial refrigeration applications due to its decent heat transfer properties, wide range of application temperature and ready availability of trained manpower for operation and maintenance. Further, NH_3 has zero ozone depleting potential (ODP) and global warming potential (GWP). However, toxicity, flammability and issues with material compatibility are some of the challenges for NH_3 system apart from restriction in installation of NH_3 plant near populated areas.

In recent years CO_2 has emerged as one of the top choices among the natural refrigerants and is viewed as a long term alternative. It has favorable thermo-physical properties like high specific heat, non-toxicity, and non-flammability along with zero ODP and unit GWP values. Besides, it has higher refrigerating capacity at low temperature application such as in freezing and cold storages applications compared to NH_3. But, the low critical point and higher gas cooler pressure of CO_2 are challenges which lowers its performance when heat rejection is at high ambient (Gupta and Dasgupta, 2015). Bhattacharyya et al. (2005) recommended the use of cascade refrigeration system (CRS) in high ambient condition to achieve better performance. Cascade refrigeration systems are multi-stage refrigeration cycles which are coupled through a cascade heat exchanger. This cascade heat exchanger operates as a condenser for the low temperature circuit LTC and an evaporator for the high temperature circuit, HTC.

NH_3-CO_2 refrigerant pair in CRS has been explored by many researchers as natural refrigerant pair with CO_2 in LTC and NH_3 in HTC. Bellos and Tzivanidis (2019) reported an analytical study on 18 different refrigerants in HTC to find out the refrigerant which are most suitable to be used along with CO_2 in an LTC evaporator temperature range of -35 ºC to -5 ºC. They compared performances based on the energy efficiency and TEWI and concluded that NH_3, R290, R600, R600a and R1270 are the most promising options. Sun et al., (2019) reported a study on a wide range of refrigerants in a three-stage cascade system and converged to six different refrigerant pairs. They found NH_3 as a good choice for HTC in high ambient conditions for large capacity refrigeration systems. Zhang et al., (2020) studied potential replacement of R1270 in a CRS pair of R1270-CO_2. They found NH_3-CO_2 pair as one of the most suitable

alternatives. Aktemur et al. (2020) reviewed a large number of studies on CRS systems and observed good attention on the NH_3-CO_2 pair.
Exergetic, economic and environmental analysis are the key parameters to evaluate feasibility of a refrigeration system for an application. Exergoeconomic term is used in many studies which includes the exergetic and economic analysis of the system and is also used to optimize the system design. Rezayan and Behbahaninia (2011) presented a thermoeconomic optimization and exergy analysis of a NH_3-CO_2 CRS considering cooling capacity for constant source and sink temperatures of both circuits. Aminyavari et al. (2014) presented a study on NH_3-CO_2 CRS based on exergetic, economic and environmental parameters using a genetic algorithm to optimize the multi-objective function for optimal design of the system. They considered exergetic efficiency and total cost of the system which includes capital, operational and maintenance cost along with social cost due to CO_2 emissions, as the objective function. Mosaffa et al. (2016) presented a comparative exergoeconomic and environmental study of two NH_3-CO_2 CRS, one equipped with two flash tanks and the other with a flash tank and a flash intercooler. They concluded that a system with two flash tanks performs better. They also optimized the operating parameters of the systems for maximizing COP as well as exergy efficiency and for minimizing the total annual cost. Patel et al. (2019) analyzed and compared two CRSs having NH_3-CO_2 and C_3H_8-CO_2 as refrigerant pairs, on the basis of total annual cost and exergy efficiency and concluded that a NH_3-CO_2 CRS offer 6.42% higher exergy efficiency but with 5.33% higher annual cost. Aktemur et al. (2020) compared CRSs with various refrigerant pairs for ultra-low temperature application based on the energetic and exergetic performance. Jin et al. (2020) reported that the otherwise complex multi-evaporator refrigeration systems are gaining acceptance in industries due to their higher operating efficiencies, design and installation flexibility, lower initial cost and compactness. Refrigerant pair NH_3-CO_2 is therefore selected for this study in multi-evaporator configuration for the seafood application considering performance, feasibility, high ambient operation and in view of confidence on NH_3 system in this business sector.
The scope of this study is to evaluate advantage of NH_3-CO_2 cascade systems with respect to NH_3 systems. The various refrigeration loads and other operating conditions are adopted from a surimi (seafood) processing plant located in Mumbai, India. In the economic analysis, initial capital costs, maintenance costs, operating costs and life cycle costs are calculated and compared while for environmental analysis, total equivalent warming impact (TEWI) are calculated and compared.

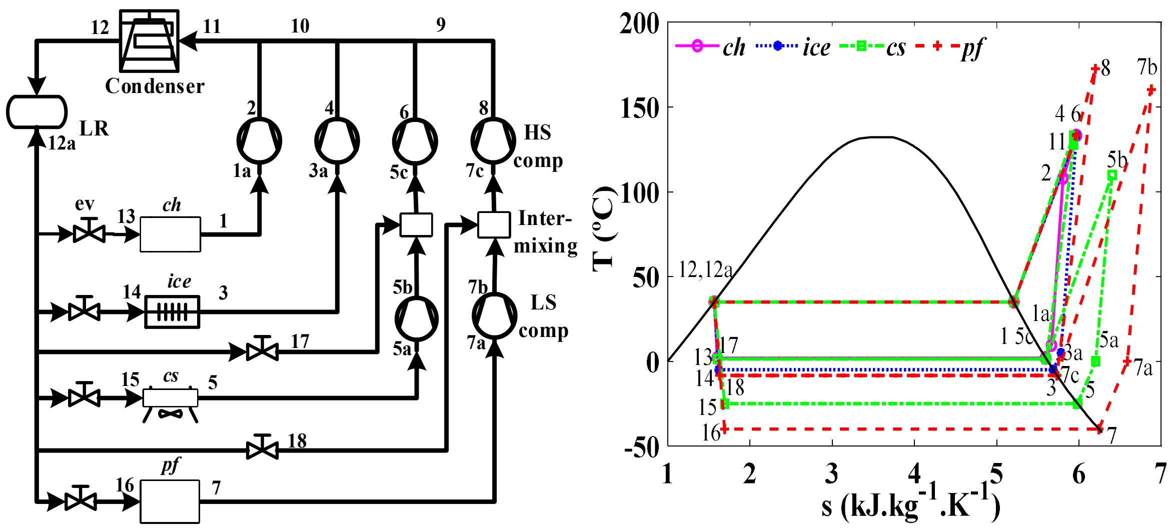
2. SYSTEM DESCRIPTION

Piping diagram and P-h chart of the 'baseline' NH_3 system and the proposed NH_3-CO_2 CRS are shown in Figure 1 (a) and (b). This study considered the refrigeration demands of a Mumbai (India) based surimi processing plant as discussed in Saini et al. (2020). The system has four evaporators, termed *ch*, *ice*, *cs*, and *pf* for chilled water, ice, cold storage and plate freezing applications respectively, they have evaporator outlet state points at 1, 3, 5, and 7 with reference to Figure 1. All the evaporators are flooded type and have separate individual expansion valves that have refrigerant inlet from the liquid receivers (LR). The compression ratios in *cs* and *pf* refrigerant lines are high due to large temperature difference between evaporator and condenser, thus double stage compressors with intercooling are employed in these refrigerant lines in the baseline system.
For intercooling, refrigerant is expanded from receiver up to the intermediate pressure and mixed with compressed refrigerant of first stage compressor as shown in piping diagram. In the proposed CRS, *ch* and *ice* evaporators have the same arrangement as baseline while the *cs* and *pf* evaporators receive CO_2 as refrigerant and the lines have single stage compressors. The HTC and LTC interact at cascade condenser *cc*, which functions as a condenser for LTC and as an evaporator for HTC. The compressor discharge for all the lines are at the same pressure and heat rejection takes place in a water-cooled evaporative condenser. The condensed refrigerant is collected in a liquid receiver.

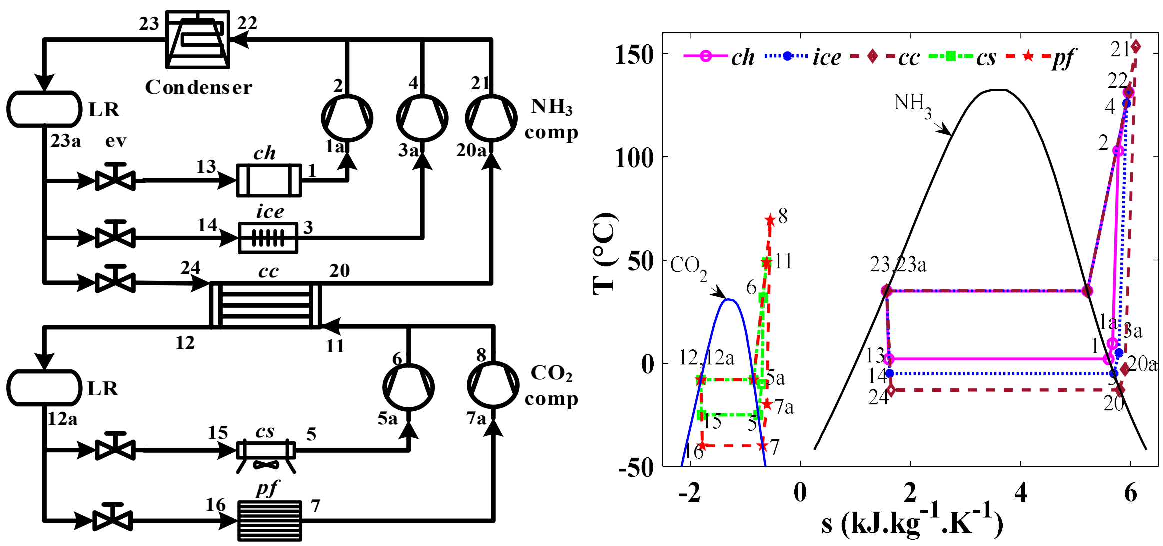
3. MODELLING AND BOUNDARY CONDITIONS

3.1 Model for Thermodynamic Analysis

Mathematical models are developed for both the configurations assuming steady flow steady-state condition across each component, while neglecting pressure drops in pipes and all heat exchangers. Isenthalpic expansion is assumed in all expansion valves. Refrigerant state is assumed as saturated liquid at the outlet for all evaporators and condensers. Energy balance and exergy destruction equations used are tabulated in Table 1. Using equation (1), mass flow rate through all the evaporators is calculated. Work consumption rate of all the compressors is calculated using equation (2). The compressor isentropic efficiency component η_s is a function of compressor pressure ratios (R) and are given by equations (3) and (4) for NH_3 and CO_2 compressors respectively as per literature (Patel et al., 2019). Exergy destruction in all the system components is calculated using equations (9)-(14). COP of both the systems are calculated using equation (22) and the second law efficiency (ϵ_{ex}) is calculated using equation (25). For the proposed system, the optimum cascade temperature was determined for maximizing COP through simulation.



(a) Baseline system



(b) Proposed NH_3-CO_2 CRS

Figure 1: Piping and P-h diagrams of (a) baseline system, (b) CRS system

Approach temperature in all heat exchangers including condensers and evaporators are considered to be 5 ºC. The suction superheat assumed is 5 ºC in *ch* and *ice*, 10 ºC in *cc* and 15 ºC in *cs* and *pf*. Evaporation temperature of *ch*, *ice*, *cs* and *pf* are 2 ºC, -5 ºC, -25 ºC and -40 ºC respectively while refrigeration demands (in kW) in these evaporators are 115, 55, 60 and 70 correspondingly. In this study, system performance is compared in terms of COP and ϵ_{ex} for a range of condensing temperature 25 °C to 45 °C considering ambient conditions of Mumbai. Thermodynamic state points for each operating condition are obtained iteratively utilizing the EES (Klein, 2018) platform.

Table 1: Abstract of equation used in the system modelling

Component/ Parameter	Energy balance	Exergy balance
Evaporator	$\dot{m}_{ref} = \frac{\dot{Q}_{evp}}{(h_{out}-h_{in})_{evp}}$ (1)	$\dot{Ex}_{d,evp} = T_{amb}[\dot{m}_{ref}.(s_{out} - s_{in})_{evp} - \frac{\dot{Q}_{evp}}{T_{evp}}]$ (9)
Compressor	$\dot{W}_{comp} = \frac{\dot{m}_{ref}.(h_{out,s}-h_{in})_{comp}}{\eta_s}$ (2) $\eta_{s,NH3} = -0.00097R^2 - 0.01026R + 0.83955$ (3) $\eta_{s,CO2} = 0.00476R^2 - 0009238R + 0.89810$ (4)	$\dot{Ex}_{d,comp} = T_{amb}.\dot{m}_{ref}.(s_{out} - s_{in})_{comp}$ (10)
Condenser	$\dot{Q}_{cond} = \dot{m}_{ref}.(h_{out} - h_{in})_{cond}$ (5)	$\dot{Ex}_{d,cond} = T_{amb}[\dot{m}_{ref}.(s_{out} - s_{in})_{cond} + \frac{\dot{Q}_{cond}}{T_{cond}}]$ (11)
Expansion valve	$h_{in,ev} = h_{out,ev}$ (6)	$\dot{Ex}_{d,ev} = T_{amb}.\dot{m}_{ref}.(s_{out} - s_{in})_{ev}$ (12)
Mixing of refrigerant	$\dot{m}_{ref,in,1}h_{in,1} + \dot{m}_{ref,in,2}h_{in,2} = \dot{m}_{ref,out}h_{out}$ (7)	$\dot{Ex}_{d,mix} = \dot{m}_{ref,in,1}[(h_{in,1} - h_{out}) - T_{amb}.(s_{in,1} - s_{out})] + \dot{m}_{ref,in,2}[(h_{in,2} - h_{out}) - T_{amb}.(s_{in,2} - s_{out})]$ (13)
Cascade condenser	$[\dot{m}_{ref}.(h_{in} - h_{out})_{cc}]_{LTC} = [\dot{m}_{ref}.(h_{out} - h_{in})_{cc}]_{HTC}$ (8)	$\dot{Ex}_{d,cc} = T_{amb}.[\left(\dot{m}_{ref}.(s_{out} - s_{in})\right)_{LTC} - \left(\dot{m}_{ref}.(s_{out} - s_{in})\right)_{HTC}]$ (14)

3.2 Model for Economic Analysis

An economic model consisting of initial capital cost (ICC), annual operating cost (AOC), maintenance cost (MC) and payback period is constructed. The ICC comprises component cost, installation cost and additional cost and is calculated using equation (27). Following the studies by Dai et al. (2019) and Cui et al. (2020), component costs are calculated using the cost function given in Table 2. Component cost for all the heat exchangers is calculated based on heat exchanger area (A), the compressor cost is calculated based on power consumption ($\dot{W}$) while expansion valves and liquid receiver cost are calculated based on the refrigerant mass flow rate ($\dot{m}_{ref}$).Refrigeration systems with CO_2 are still in the developing phase, and hence system capital costs are high compared to the mature NH_3 refrigeration systems components.

Table 2: Component cost function of the system

Components	Cost function	
Evaporator/condenser	$331.7.A^{0.939}$	(15)
CO_2 compressor	$\dot{W}_{CO2} = 17547.\dot{W}^{0.4488}$	(16)
NH_3 compressor	$\dot{W}_{NH3} = 758.15.\dot{W}^{0.8768}$	(17)
Cascade condenser	$1874.4.A^{0.9835}$	(18)
Expansion valve	$817.\dot{m}_{ref}$	(19)
Liquid receiver	$2000.\dot{m}_{ref}^{0.67}$	(20)

The heat transfer area is calculated using equation (21) utilizing logarithmic mean temperature difference (*LMTD*) and the heat transfer coefficients of heat exchangers *U* in (W.m^{-2}.K^{-1}).

$$A = \frac{Q}{U.LMTD} \quad (21)$$

Values of U for evaporators, condensers and cascade condensers are considered as 30, 40 and 1000 $W.m^{-2}.K^{-1}$ respectively as per literature (Mosaffa et al., 2016). The installation and additional costs (C_{add}) are assumed as 15% of the total component cost. Annual operating cost (AOC) is calculated using equation (28), where annual energy consumption (AEC) is calculated for the temperature bin hour of Mumbai and local commercial electricity price is considered as 0.1$ per kWh (Pandey, 2019). Life cycle cost (LCC) is calculated using equation (30), where the system service lifetime (n) is considered as 15 years. Payback period of the proposed CRS system relative to the baseline system is calculated using equation 31.

3.3 Model for Environmental Impact Analysis

Total equivalent warming impact (TEWI) is a significant parameter which considers sum of direct ($TEWI_{direct}$) and indirect ($TEWI_{direct}$) CO_2 emissions from a refrigeration system. The TEWI is calculated using the equations (32)-(34). Total refrigerant leakage ($M_{leakage}$) is computed from total refrigerant charge and refrigerant leakage rate. In this study, refrigerant charge is considered as 3kg of refrigerant per kW of refrigeration load and annual refrigerant leakage is taken as 15% as per literature (Cui et al., 2020). Recycling factor (α) is taken 95% while GWP values for NH_3 and CO_2 are 0 and 1 respectively. The Electricity regional conversion factor (β) is taken as 0.9 (kg CO_2 per kWh), which depends on the regional energy mix (Lata and Gupta, 2020). Various parameters compared for both the refrigeration systems are tabulated in Table 3.

Table 3: Parameters evaluated for the analysis

Parameter	Equations	
Thermodynamic model	$COP_{baseline} = \frac{\dot{Q}_{ch}+\dot{Q}_{ice}+\dot{Q}_{cs}+\dot{Q}_{pf}}{\dot{W}_{net,baseline}}$, $COP_{CRS} = \frac{\dot{Q}_{ch}+\dot{Q}_{ice}+\dot{Q}_{cs}+\dot{Q}_{pf}}{\dot{W}_{net,CRS}}$	(22)
	$\dot{W}_{net,baseline} = (\dot{W}_{ch} + \dot{W}_{ice} + (\dot{W}_{LS} + \dot{W}_{HS})_{cs} + (\dot{W}_{LS} + \dot{W}_{HS})_{pf})$	(23)
	$\dot{W}_{net,CRS} = (\dot{W}_{ch} + \dot{W}_{ice} + \dot{W}_{cc} + \dot{W}_{cs} + \dot{W}_{pf})$	(24)
	$\dot{Ex}_{d,total} = \dot{Ex}_{d,evp} + \dot{Ex}_{d,comp} + \dot{Ex}_{d,cond} + \dot{Ex}_{d,ev} + \dot{Ex}_{d,mix} + \dot{Ex}_{d,cc}$	(25)
	$\epsilon_{ex} = (1 - \frac{\dot{Ex}_{d,total}}{\dot{W}_{net}})$	(26)
Economic model	Initial capital cost; $ICC = \sum_i^n C_{component,i} + C_{add}$	(27)
	Annual operating cost; AOC =(AEC) x electricity price per unit	(28)
	Maintenance cost; MC= 0.01. ICC	(29)
	Life cycle cost; $LCC = ICC + (AOC + MC).n$ n is system service lifetime in years	(30)
	Payback period= $\frac{(ICC_{CRS}-ICC_{baseline})}{(AOC+MC)_{baseline}-(AOC+MC)_{CRS}}$	(31)
Environmental model	Total equivalent warming impact; $TEWI = TEWI_{direct} + TEWI_{indirect}$	(32)
	$TEWI_{direct} = \left(M_{leakage}.n + M_{charge}.(1-\alpha)\right).GWP$	(33)
	$TEWI_{indirect} = \beta.AEC.n$	(34)

4. RESULTS AND DISCUSSION

COP of both the systems are calculated using equation (22) and presented for a range of condenser temperature from 25 °C to 45 °C in Figure 2. It is observed that the COP of the CRS varies with LTC condenser temperature (T_{MC}) as well as the HTC condenser temperature, therefore, optimum T_{MC} is calculated at all condenser temperatures for maximizing COP, keeping other operating parameters constant. Referring to figure 2, the optimum values of T_{MC} are presented on the secondary vertical axis.

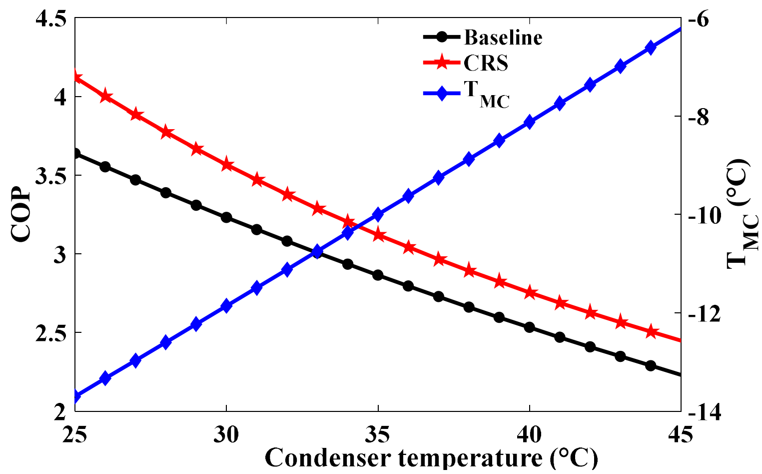


Figure 2: COP variation with condenser temperature

Optimum T_{MC} varies linearly from -14 °C to -4 °C with condenser temperature varying from 25 °C to 45 °C. As expected, an increase in the condenser temperature leads to increase in total power consumption of the systems due to increase in compression ratios. However, COP of the CRS is always higher than the baseline system and the difference is more prominent at lower condenser temperature (lower ambient). The maximum possible COP advantage in CRS over baseline system is 13.3% at 25 °C. Using the temperature bin hour of Mumbai, the annual energy consumption computed for baseline and CRS system are 983.18 MWh and 901.82 MWh respectively.
Total exergy destructions and second law efficiency variation with condenser temperature is presented in Figures 3 and 4. The total exergy destruction in the CRS is lower than the baseline for full range of operation. For example, at condenser temperature of 25 °C (ambient temperature 20 °C), the total exergy destruction in the CRS is 41.15 kW which is 14.5% lower than the baseline system. CRS shows higher exergy efficiency due to lower power consumption by the system. The higher exergy destruction and lower exergy efficiency in the baseline system indicates greater potential of improvement at component level.

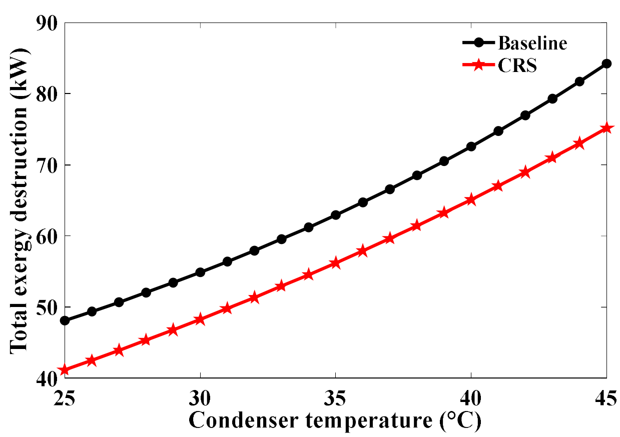


Figure 3: Total exergy destruction

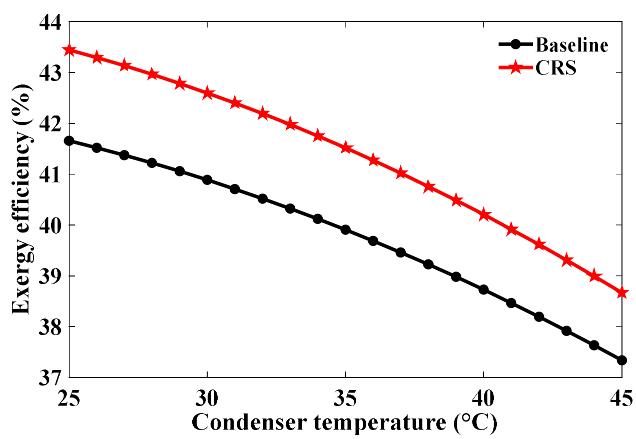


Figure 4: Exergy efficiency

Component wise exergy destruction is computed to pinpoint the components for improvement. Percentage exergy destruction in major components of both the systems are presented in Figure 5. It shows that compressors and condensers are the components where exergy destruction is maximum, while it is minimum at evaporators and suction pipes. In the CRS, exergy destruction share of the cascade condenser is 12.9%. While in the baseline system, exergy destruction occurring at the mixing of the refrigerant in intercooling and at the inlet of the condenser is about 16% which is rather high compared to 2% in CRS.

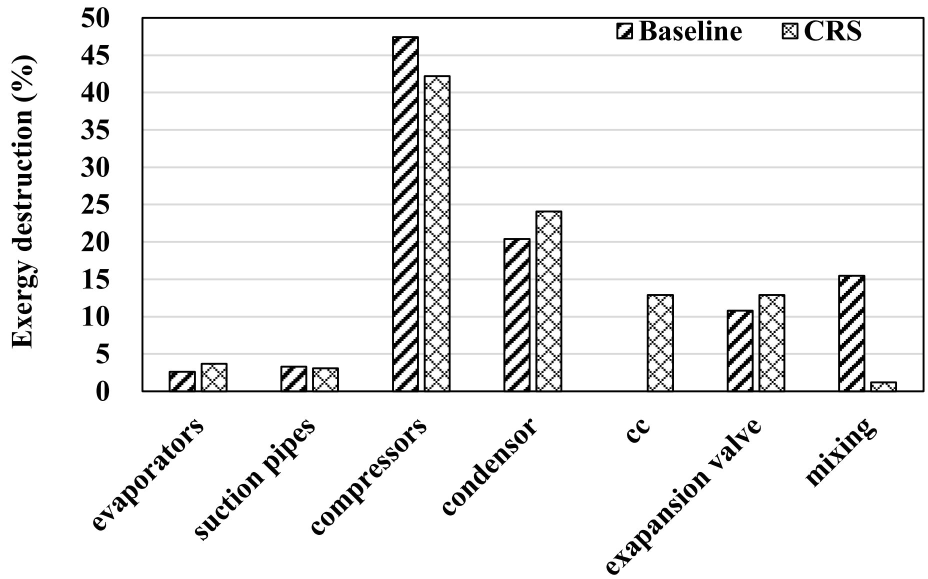


Figure 5: Component wise percentage exergy destruction

Based on the economic model, ICC of baseline and CRS are computed as \$702,800 and \$720,800 while AOC of baseline and CRS are \$108,149 and \$99,200. The maintenance cost of baseline and CRS are computed as \$7,028 and \$7,208 while the LCC of baseline and CRS are \$2,430,460 & \$2,316,925. The payback period estimated using the model for CRS relative to baseline refrigeration system is 25 months. This short extension of payback period offers potential cost benefits for adoption of NH_3-CO_2 CRS compared to NH_3 system for new installation.
Due to the zero GWP value of NH_3, the baseline system has zero direct TEWI; however, the higher annual energy consumption leads to higher indirect TEWI compared to CRS. Total TEWI calculated for baseline system is 13273 tons of CO_2 which is 1098 tons of CO_2 more than the CRS. These results shows that proposed CRS system has lower emissions compared to baseline system thus being more environmental friendly.

5. CONCLUSION

This study presents thermodynamic, economic and environmental benefits of using a proposed multi-evaporator NH_3-CO_2 CRS over a conventional NH_3 system for surimi (seafood) processing industry for ambient of Mumbai, India. Based on the analytical study, following conclusion are drawn:

- For the range of condenser temperature analysed, which is typical of tropical warm climate, the COP of the CRS is found higher than the baseline system. The maximum benefit in COP using CRS is about 13.3% at 25 °C condenser temperature.
- With increase in condenser temperature, the total exergy destruction of both the systems increase while exergy efficiency decreases. The CRS shows about 14% less exergy destruction compared to baseline system.
- In both systems, the compressors contribute to the largest exergy destruction, with an average value of 45%. While the evaporators contribute the least exergy destruction, with a modest average value of 4%.
- CRS has higher ICC but lower AOC over baseline. The payback period calculated relative to baseline system is 25 months.
- The TEWI of CRS is about 9% lower than the TEWI of baseline system.

This study has shown that it would be better to have a cascade refrigeration system than a single stage ammonia refrigeration system in a surimi plant, producing cold for processing and storage, since it has higher COP and a moderate payback time.

NOMENCLATURE

The nomenclature should be located at the end of the text using the following format:

$\dot{m}$	Mass flow rate	($kg.s^{-1}$)
$\dot{W}$	Power consumption by compressors	(kW)
$\dot{Q}$	Refrigeration load	(kW)
COP	Coefficient of performance	
Ex	Exergy	(kW)
h	Specific enthalpy	($kJ.kg^{-1}$)
s	Specific enthalpy	($kJ.kg^{-1}.K^{-1}$)
T	Temperature	(ºC)
Greek symbols:		
η	Efficiency	(%)
ϵ	Exergy efficiency	(%)

Subscript

1, 1a	State points
d	destruction
in	Inlet
out	Outlet
s	Isentropic

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