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Report

D1.3: Report on energy efficient refrigeration systems

Surimi case

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ABSTRACT

This report is a part of WP1 in the Revalue project and it presents the findings from research on energy-efficient and environment-friendly refrigeration system for Surimi processing and storage in warm ambient temperatures. During 2017-18, India exported about $17.6 million worth of surimi and it has good potential for further growth. The majority of surimi processing industries in India are located along the western coast and for this study, operational data from one such industry located in Mumbai has been used in this study. A surimi supply chain has substantial cooling demand at various temperatures, ranging from -40 to 8 ºC, from harvesting to the final product. Conventionally, a single NH_3 refrigeration system with multiple evaporators is employed to meet the cooling demands in a processing plant, including the cold storage. However, due to several challenges faced by the surimi industry in India (Dasgupta et al, 2019), the plants operate in part-load conditions for a major part of the year and the overall efficiency is rather low. To improve energy efficiency, a smaller capacity plant is recommended and a CO_2-NH_3 cascaded refrigeration system (CRS) is proposed. In the cascade system, CO_2 is used in low temperature circuit and NH_3 in high temperature circuit, which also reduces the contamination hazard of food from NH_3. Modelling and analysis of various CO_2-NH_3 configurations were conducted, and a CRS system having a COP of 6.2% higher than the conventional NH_3 system was identified.

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1 Introduction

Surimi is processed fish meat that mainly serves as intermediate product used in the preparation of a variety of ready to eat seafood products. It is rich in protein and omega-3's. The global production of surimi is about 820,000 MT with a global market of $4.06 billion (Seaman, 2018). Tropical countries produce about 60% of the surimi, in which India contributes with about 11%. During 2017-18, India exported about $17.6 million worth of surimi and it has good potential for growth. Price of surimi in international market depend upon quality and therefore, appropriate refrigeration and handling is important at every stage of surimi production, storage and transport.

Surimi contains stabilized myofibrillar proteins from fish that are obtained through mechanical mincing and washing of fish meat. Generally, it is fish species which are underutilized, contain low fat and have otherwise lower commercial value, that are used for surimi production (Venugopal and Shahidi, 1998). The most favourable species in India is Threadfin bream or Pink perch as it has low fat content, white meat and good gelation properties. Other species utilized are croaker, lizard fish, goat fish, ribbon fish, sardine, big eye snapper etc. Western coast of India has large fishing activities and is a prominent provider of job and sustenance for a large number of marginal fishermen; most surimi processing plants are also located there. In order to meet all its in-plant refrigeration needs, the surimi processing plants employs NH_3 based vapour compression refrigeration system. Data from one such surimi plant in Mumbai has been utilised in this study.

Significant amount of energy and fuel is consumed in the fisheries sector. Its vulnerability to changing energy supplies and prices highlight the need to review the sector's energy and fuel efficiency level in conjunction with future trends. Cooling technology used is of great economic and energetic importance for surimi production and various other stages of its supply chain. During plant visits we observed cooling systems used are not specifically designed for the application and therefore, are not operating at optimum efficiency level. An energy efficient operation of vapour compression cycles not only help to save electrical energy but today this is also being increasingly associated with saving fossil fuels and reduction of overall carbon foot print.

Some of the well-recognised strategies to improve the energy efficiency of refrigeration system like the recommendations from UN Environment report (Briefing Note B, 2018) are explored in this study:

i. Minimising the cooling load.

ii. Minimising the temperature lift.

iii. Accounting for variable operating conditions.

iv. Selecting the most efficient refrigeration cycle and components.

v. Design of effective control systems.

vi. Checking operating performance and correcting any faults of existing RACHP systems

Additionally there are challenges in warm weather. The refrigeration systems used in very hot countries generally use more energy than equivalent systems in cooler countries, because:

i. The cooling load is higher for a given mass of content (surimi)
ii. The temperature lift is bigger, because the "hot end" of the plant is rejecting heat at a much higher ambient with air or circulating water[1]. (1-degree C extra temperature lift add about 2% to 4% to the energy used by a plant)

[1] Most ammonia systems in India are water cooled. Water sprinkled down with air moving up. The incoming water is also at ambient temperature. Sea coast like Mumbai have additional challenge of low rate of evaporation due to high humidity

These two factors lead to considerably higher energy consumption by refrigeration system in warm ambient temperatures. However, there is significant potential to improve the efficiency of refrigeration equipment when new equipment is designed and fabricated specifically for the desired load condition.

The environmental implications for the choice of new equipment and choice of refrigerant are likely to be increasingly open to review, particularly with respect to GHG emissions associated with the sourcing and use location. India as member of Group 2 countries as per Kigali amendments, have a longer time frame for reduction of use of HFCs than European countries which are in Group 1. The de-accelerated phasedown presents significant advantages for the India to leapfrog to the best alternatives later. India has fulfilled the Montreal Protocol targets ahead of schedule and should aspire to maintain the legacy. The choices made in near future on refrigerant in surimi processing plant could increasingly be influenced based on long- and short-term policy conditions resulting into market incentives or penalties. In this study we have explored gainful utilisation of a biosphere gas CO_2 as refrigerant in a sub-critical system, which has very low direct environmental impact. However, the secondary impact of enhanced carbon emission from possible low efficiency of such system can easily tide over the benefit. Use of CO_2 at high ambient is known to have detrimental effect on its efficiency unless some modifications are done. Researchers have reported advantage through cycle modification as well as component modification. Deployment of two-phase ejector, use of multi-stage compression with intercooler, use of internal heat exchanger (IHX), use of parallel compression etc. are some of the strategies under cycle modification, while use of work recovery expander, simultaneous heating-cooling, use of VFD compressor etc., are categorised as component improvement. We designed an innovative multi-evaporator NH_3 – CO_2 cascade system having evaporators above and below the cascade temperature with the CO_2 utilised at the lower temperatures. A few variants of the configuration designed for enhancing overall system COP were explored. The performance was compared with conventional NH_3 system and advantages of proposed configurations were identified.

This report is one of the deliverables under the project ReValue – *Innovative technologies for improving resource utilization in the Indo-European fish value chains* funded through the INNO-INDIGO Joint Call on Bio-economy. This work is a part of WP1, deliverable 1.3: Report on energy efficient refrigeration systems. The structure of the remainder of this report is as follows: Section 2 states the problem statement and scope of the work, Section 3 provides main objectives of this study. Section 4 presents the methodology adopted to achieve the stated objectives and other related processes. Section 5 presents the results obtained from the study. Discussion based on the results is given in section 6 while section 7 presents the conclusions from this study.

2 Problem statement

An energy efficient refrigeration system is essential in the overall supply chain of surimi that can enable Indian firms to maintain quality and attain competitive advantage in the global seafood market. The scope of this study is the exploration of efficient refrigeration systems for the specific needs of a surimi processing plant in Indian perspective. Use of environmentally benign, low flammability, low toxicity refrigerant and efficient refrigeration cycle are essential.

3 Objectives

The objective of this study was to perform an extensive analysis of the cooling demands in a surimi processing plant and then put forward energy efficient and environment friendly refrigeration options dedicated for the same, suitable for Indian surimi industry, which has a good growth potential both in terms of market volume and market spread.

Performance study of a few state-of-the-art CO_2-NH_3 CRS configurations were made and compared with conventional multi-evaporator NH_3 system. For better performance, the cascade condensing temperature of the CRS was optimized. In the analysis, the refrigeration demands at various evaporating temperatures were estimated for a typical 10 MT/day surimi production plant.

4 Methodology

The refrigerant NH_3 that is employed in refrigeration systems across surimi processing plants is environment friendly and such systems can have high energy efficiency. However, due to uncertainties in supply of raw material due to absence of supply chain integration, the plant was observed to operate in part load condition for major part of the year. Further, the refrigeration systems were not designed for the specific load conditions of today and substantial retrofittings have been carried out to accommodate various loads. These reduces the operating efficiency of refrigeration systems.

The various cooling demands in the plant are for chilling of water, production of ice, plate freezing of product and maintaining the cold storage temperature. The amount (weight) of chilled water required is estimated to be about 10 times that of surimi produced or 4 times that of raw material input (Park, 2013). Ice is required for occasional holding of raw material and pre-chilling before production and is observed to be about 1:1 in ratio of product. Surimi is packed into 10 kg blocks and they are frozen in plate freezers down to -35 ºC surface temperature to ensure a core temperature below -20 ºC. The cold storage is maintained at -20 ºC and has capacity to hold the total production of 5 months of the plant. Due to part-load operation, the overall plant efficiency is found to be rather low. Further, the NH_3 based refrigeration system utilised at the plant was not optimally designed for the prevailing cooling load condition. These motivated us to explore designing a lower capacity refrigeration system to support 10 MT/day production. The design will be modular in the sense that a higher overall capacity plant can be built by suitably scaling or deploying multiple cooling systems, such that part load operations are handled better. The typical cooling demand at various low temperatures in other seafood processing plants have similarity with that of demand in a surimi plant and a similar cooling system architecture can be useful there too, although cooling loads must be specifically computed.

The system models were developed using EES. First, we computed the various cooling loads for a typical 10 MT/day production capacity plan with optimum layout. Corresponding load calculations for various evaporators is provided in Appendix A.1 with a summary of cooling demands at various evaporators in Table 1. Then we modelled a conventional NH_3 system for the same loads, refer Appendix A.2 for details. Efficiency of such system will be higher than those employed currently in the field. This is not only due to obvious thermodynamic benefits of an idealized system but also due to non-optimal operations of existing systems. Further we conceptualised CO_2 - NH_3 cascade systems to exploit benefit of superior thermal properties of CO_2 at low temperature application. Novelty of the conceptualised system is evaporators placed both above and below the cascade temperature. CO_2 being a biosphere gas has low primary impact on environment, however, to be useful such system must have higher efficiency compared to the simulated NH_3 system discussed above.

4.1 Assumptions for simulation-based study

For simplification of thermodynamic modelling, the following assumptions were made:

i. Steady state flow process.
ii. Pressures drops and heat losses in pipes other than suction pipes neglected.
iii. Refrigerant at the outlet of evaporator is saturated vapour.
iv. Refrigerant at the outlet of condenser and cascade condenser is saturated liquid.
v. Isenthalpic operation of throttle valves.
vi. Compression is adiabatic with isentropic efficiency varying with pressure ratio.
vii. Refrigerant is superheated vapour at inlet of compressor.

viii. Power consumption of fan and water circulation pump is negligible.

The overall mass and energy balance equations for modelling are expressed in Eqs. (1) and (2), while component-wise expressions are presented in Table 2.

$$\sum_{in} \dot{m} = \sum_{out} \dot{m} \quad (1)$$

$$\dot{Q} + \sum_{in} \dot{m}\, h = \sum_{out} \dot{m}\, h + \dot{W} \quad (2)$$

4.2 NH_3 system design for the specific cooling demand in surimi process plant

A pipe diagram and a P-h diagram of the baseline conventional NH_3 vapour compression refrigeration system is shown in Fig. 4.1. The system has four evaporators, termed ch, ice, cs, & pf for chilled water, ice, cold storage, and plate freezing applications, respectively. Outlet of these evaporators are at state points 1, 3, 5 and 7, respectively. All the evaporators are flooded type and have separate individual expansion valves, which have refrigerant inlet from the receiver. Due to the low pressure-ratio between condenser and ch as well as ice evaporators, a single stage compressor is used. In cold storage and plate freezer applications, the pressure ratios are higher; hence, a two-stage compressor with intercooling is used in the modelling, ditto as in existing plant.

For intercooling, refrigerant is expanded from receiver up to the intermediate pressure and mixed with compressed refrigerant of first stage compressor. The compressor discharge for all the lines are at the same pressure and heat rejection takes place in a water-cooled evaporative condenser. Condensed NH_3 is collected in a receiver.

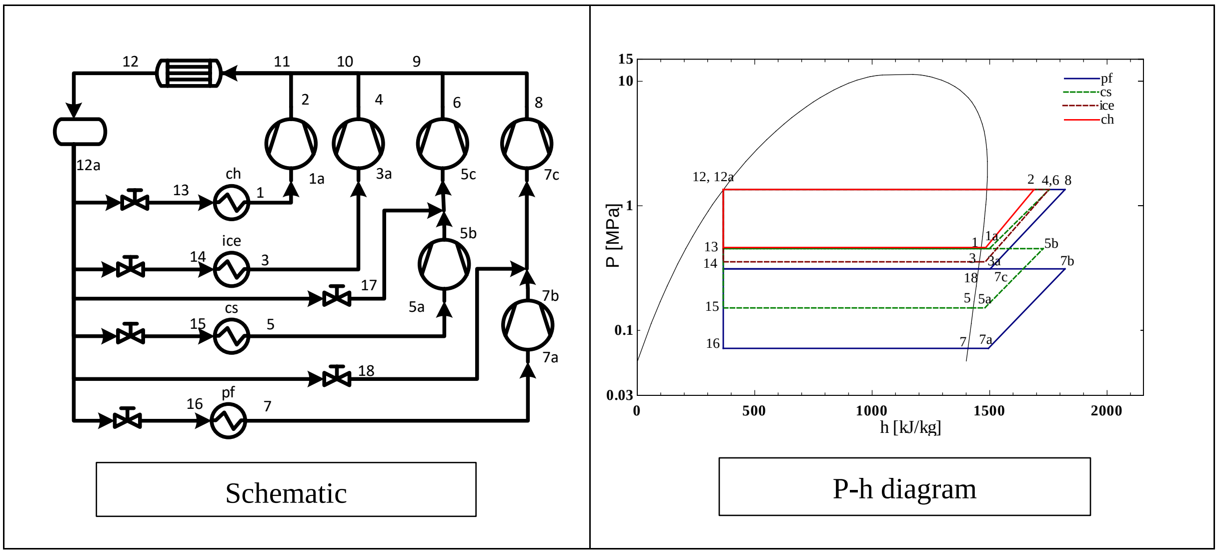


Figure 4.1: Conventional NH3 refrigeration system

4.3 CO_2-NH_3 cascade system for the specific cooling demands in a surimi processing plant

Analysing the NH_3 system we observed that about 50% of energy is consumed in the compressor system associated with the plate freezer that is for the lowest temperature refrigeration, while the cooling load is only 20%. This can be observed in Fig. 4.7 and Table A1.1. At the same time the thermo-physical properties of CO_2 as refrigerant is known to be more pronouncedly superior to NH_3 at lower temperature. These triggered us to explore CO_2-NH_3 in CRS system with CO_2 at lower end of temperature. Many researchers have earlier analysed the refrigerant pair of CO_2-NH_3 in CRS, using CO_2 in the low temperature circuit (LTC) and NH_3 in high temperature circuit (HTC) Lee et al. (2006), Belozerov et al. (2007), Dopazo et al. (2009) etc. Dopazo et al. (2011) later presented an experimental evaluation of the same. Lee et al. (2006) and Dopazo et al. (2011) also discussed trend of optimum cascade temperature with variation in evaporator temperatures. Mosaffa et al. (2016) presented a comparative study of CO_2-NH_3 CRS having various configurations equipped with flash tank on the basis of exergy, economy and environmental parameters. Patel et al. (2019) presented a comparison of economic aspects of performance of CO_2-C_3H_8 and CO_2-NH_3 CRS system. At the same time here are literature on the possible use of CO_2 as a refrigerants for secondary systems for example Winkler and Quack (2007), Kumar (2017) etc. Secondary refrigeration systems can operate either in pumped circulation or in natural circulation mode. Due to the volatile nature of CO_2, it does not remain a liquid and is partially evaporated, it therefore, has a significantly greater cooling capacity than other secondary fluids. Both these concepts have been utilised by us in designing and simulating the cascade configurations.

Pipe diagram of three most promising designs of proposed cascade systems are shown in Fig 4.2 – 4.4 along with p-h chart. The technical details, modelling, equations used and analysis are provided in Appendix A.3. Comparative study of performance of the various refrigeration systems was carried out for various ambient conditions.

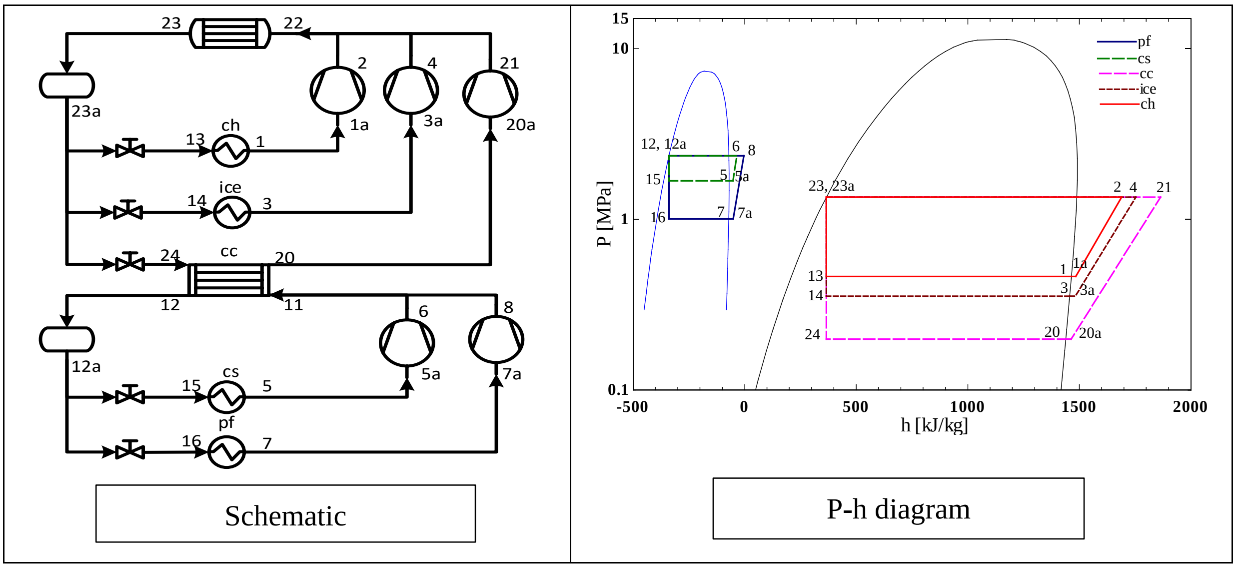


Figure 4.2: Proposed CRS1

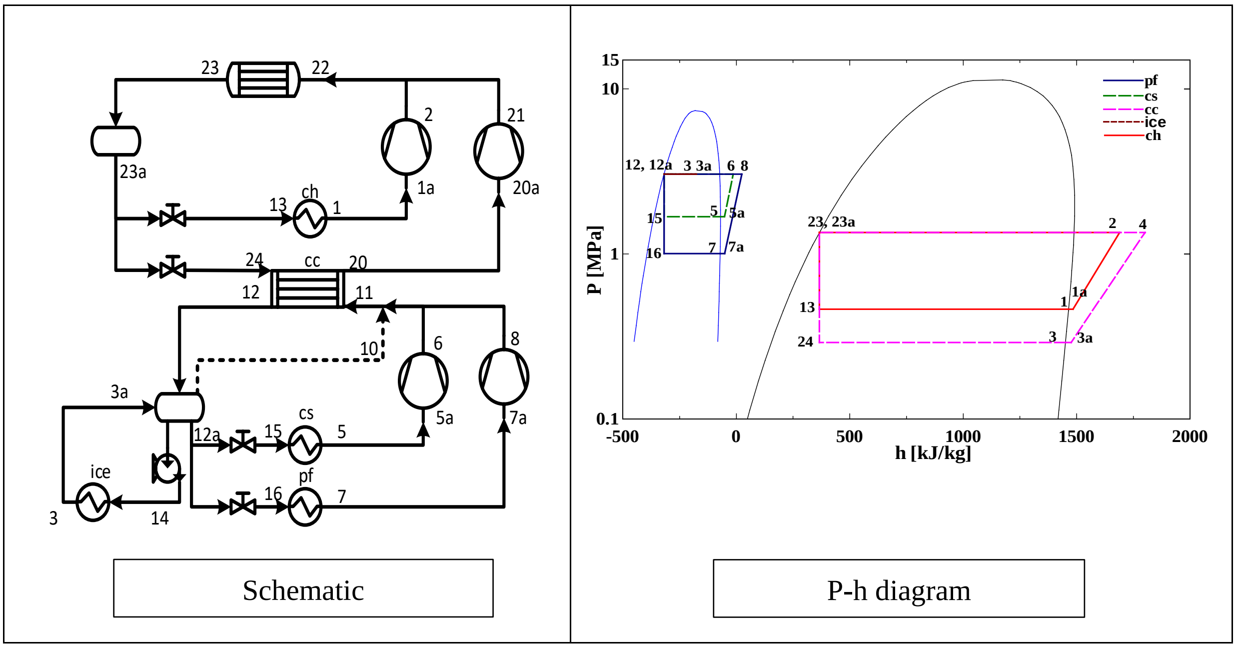


Figure 4.3: Proposed CRS2

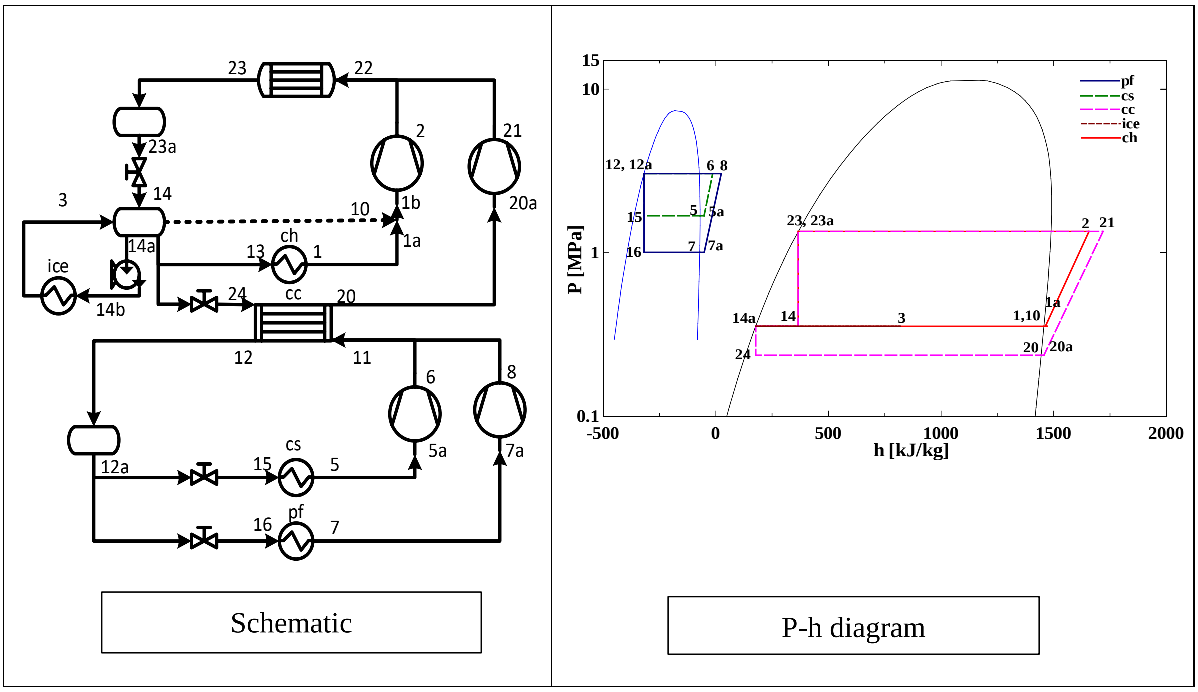


Figure 4.4: Proposed CRS3

5 Results

5.1 Energy consumption in various compressors of NH_3 system

Models developed in EES are used for the simulation based study. In the performance analysis of the baseline NH_3 system, the compressor work of all the evaporators was calculated at design conditions and it is presented in Fig. 4.5. Overall COP of the baseline system was found to be 2.53.

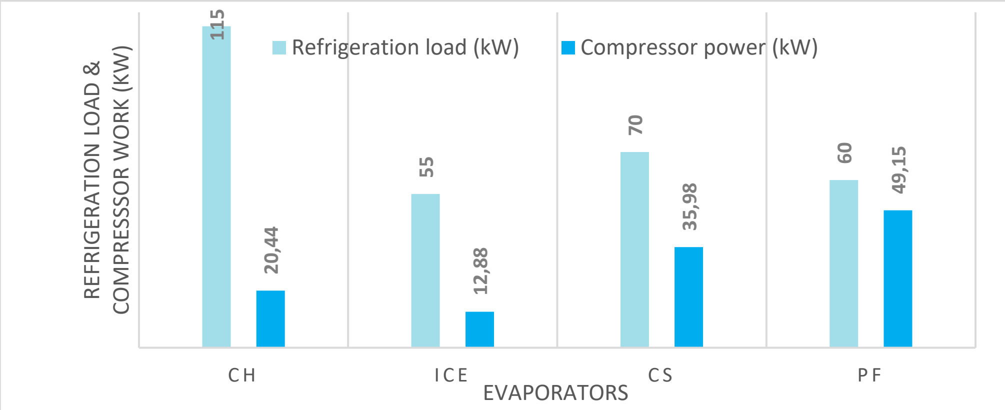
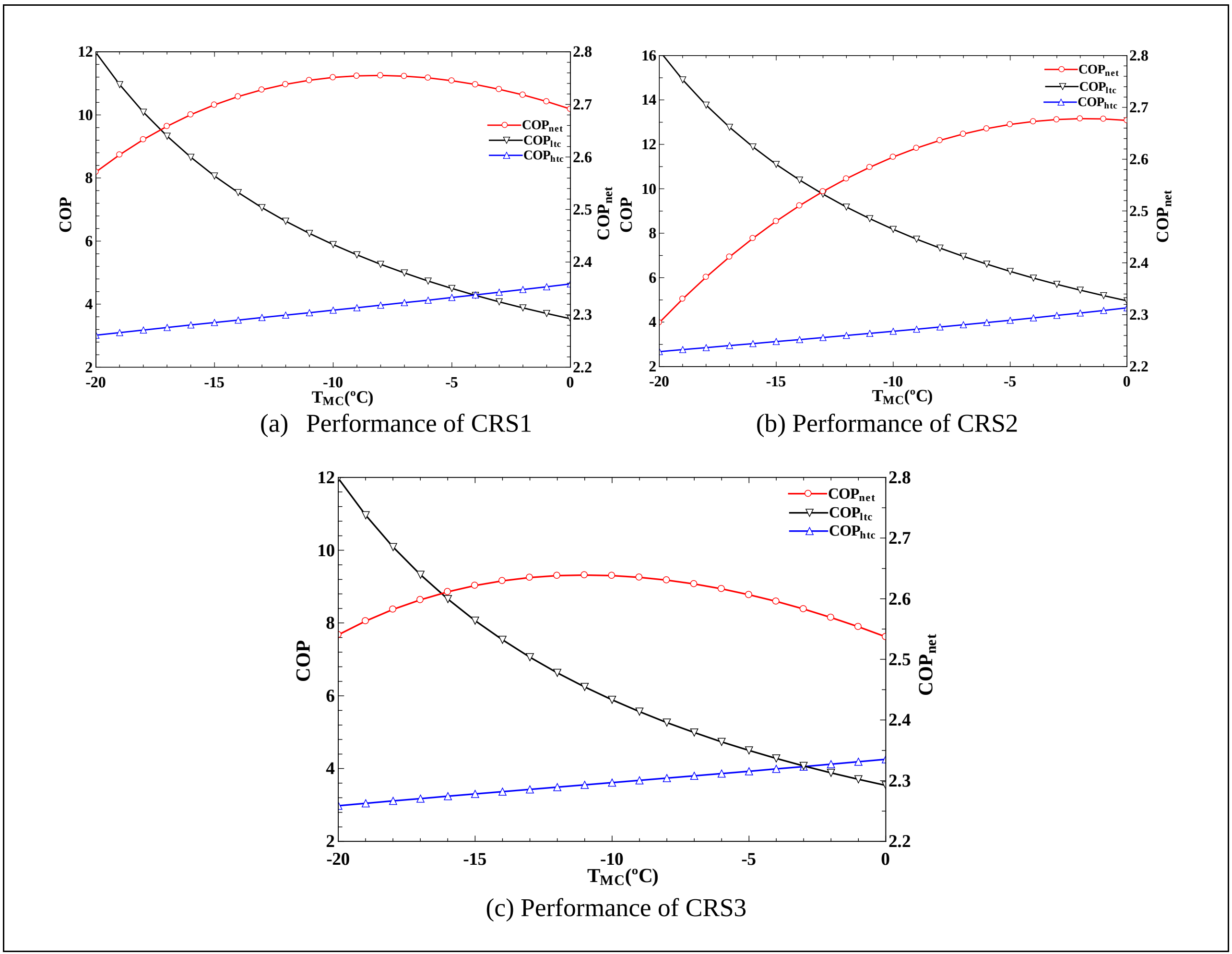


Figure 4.5: Refrigeration load and compressor work of NH_3 system

The relatively higher compressor work demand in the lower temperature refrigeration (ref. Fig. 4.5, cs & pf) and favourable thermophysical properties of CO_2 led us to explore CO_2 in low temperature evaporators in cascade systems. Comparative study of performance of the various refrigeration systems was carried out for a range of ambient conditions. While for the overall cascade system, the term COPnet is used to describe efficiency (coefficient of performance), terms COP_L and COP_H describe coefficient of performance of individual low and high temperature refrigeration circuits.

5.2 Optimum cascade temperature vs variation of COP of cascade systems

COP_{net}, COP_L and COP_H are plotted for CRS1, CRS2 & CRS3 for various cascade temperatures (T_{MC}) as shown in Figs. 4.6 a, 4.6 b & 4.6 c, at fixed refrigeration demands and for initial operating conditions. As T_{MC} increases, we infer from the definition that the pressure ratio of HTC compressors decreases while the pressure ratio of LTC compressors increases. Increase in pressure ratio of LTC results in increased compressor work and hence decrease in COP_L. Decrease in pressure ratio of HTC leads to lower compressor work contributing to higher COP_H. The combined effect is an increase in the COP_{net} to 2.76, 2.68 & 2.69 for CRS1, CRS2 & CRS3 respectively at their corresponding optimum value of TMC, which are found to be around -9 ºC, -3 ºC and -11 ºC for the three configurations (highest point of COP_{net}) . Further increase in T_{MC} is observed to decreases the COP_{net} for all the three systems.



(a) Performance of CRS1 (b) Performance of CRS2

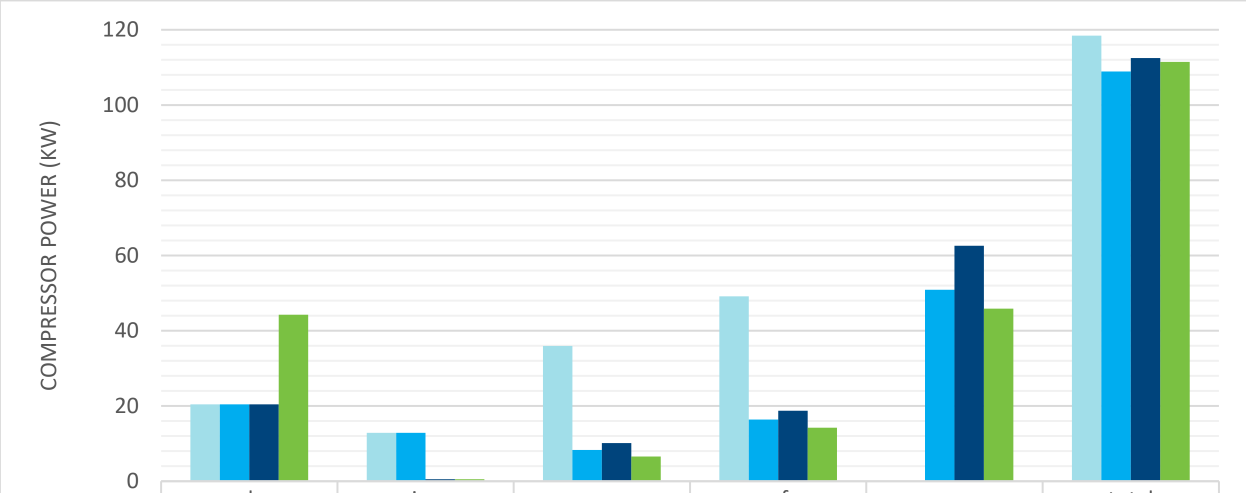
(c) Performance of CRS3

Fig. 4.6 Performance variation with cascade temperature T_{MC}

In this research work we have analysed a cycle having four evaporators distributed above and below TMC, which is a novel arrangement. However, the variation of optimum TMC has similarity with what is available in literature for simple CRS having one evaporator above and below the cascade temperature.

Power consumption of compressors for individual evaporator lines for all the four investigated refrigeration systems are compared in Fig. 4.7 with that of baseline system. It can be seen that the total power consumed in CRS1 configuration is the minimum while the same is the maximum for the baseline system. The compressor power saving is about 8.1% for CRS1, compared to the NH_3 system.

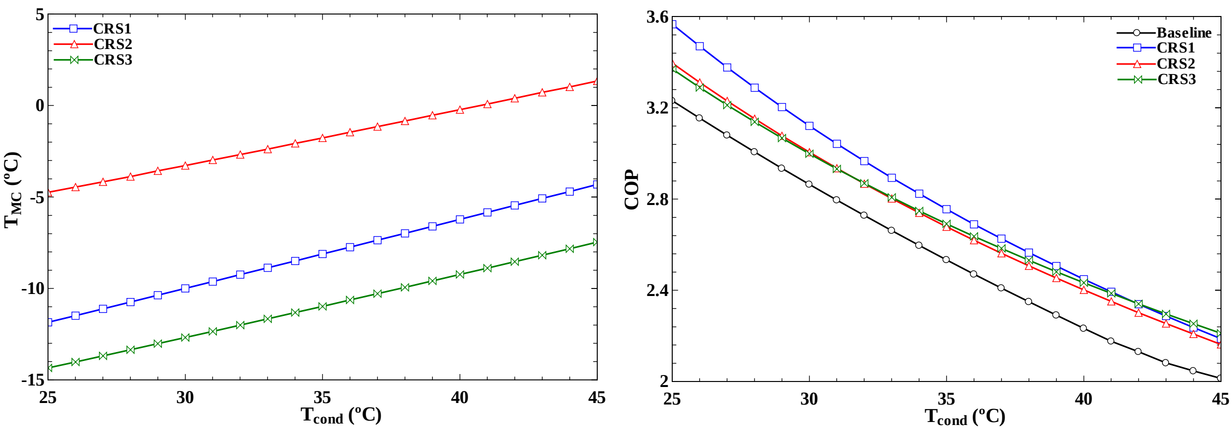
For a constant condensing temperature (T_{cond}) suitable for Mumbai, performance of CRS1, CRS2 and CRS3 are found to be better than the baseline system by 8.1%, 5% & 6.2% respectively in terms of COP_{net}.



	ch	ice	cs	pf	cc	total
Baseline	20,44	12,88	35,98	49,15	0	118,45
CRS1	20,44	12,88	8,301	16,39	50,89	108,901
CRS2	20,44	0,55	10,16	18,73	62,6	112,48
CRS3	44,24	0,55	6,583	14,24	45,86	111,473

Fig. 4.7: Comparison of compressor work in baseline CRS1, CRS2 & CRS3

Apart from Mumbai, there are other surimi processing plants along the west coast of India where the design conditions may vary. Tropical countries such as Thailand, Vietnam, India, Indonesia, Malaysia, Myanmar, Pakistan etc. contribute to about 60% of the total global surimi production (Seaman, 2018). To evaluate the suitability of the proposed refrigeration systems for various other ambient conditions, we plotted the variation of weekly averaged data of ambient dry bulb temperature (DBT) for Indian west coastal cities of Veraval, Mumbai, Ratnagiri, Mangalore, and Vishakhapatnam where there are surimi industries, along with prominent tropical cities internationally having large finishing activities such like Hanoi (Vietnam), Jakarta (Indonesia), Bangkok (Thailand) and Kuala Lumpur (Malaysia). The data of variation of dry bulb temperature at various places is provided in Appendix A.4. The result of performance analysis within the ambient temperature range (25-45 ºC) covering design conditions for these places is presented in Fig. 4.8.



(a) Variation of optimum T_{MC} (b) Performance variation at corresponding T_{MC}

Fig. 4.8: Performance comparison at various ambient temperature

Fig. 4.8 (a) shows the variation in optimum T_{MC} at various T_{cond} and Fig. 4.8 (b) shows the variation of COP_{net} for all four investigated system configurations. The trend of COP_{net}, for all the investigated systems, shows a similar decreasing behaviour with increase in T_{cond}. At a T_{cond} of 25 ºC, the COP_{net} value of CRS1, CRS2 & CRS3 are found higher than the baseline system by 9.4%, 6.9% & 8.9% respectively while at a T_{cond} of 45 ºC, the corresponding improvements observed are 7.9%, 6.9% & 8.9% respectively. It is also observed that CRS1 performs the best for a wide range of operating temperature T_{cond} 25 to 41 ºC among all the systems; however, for very high ambient, T_{cond}> 41 ºC, performance of CRS3 is found marginally better than that of CRS1.

6 Discussion

All the cascade systems performed better than NH_3 system as low temperature cooling load is shifted to LTC having CO_2 as refrigerant, exploiting its higher volumetric efficiency at low temperature compared to NH_3. In all the cascade systems, the *ch* load is in HTC and handled by NH_3 as refrigerant. In CRS2, adding the ice load in LTC boosts the TMC higher which is expected to decrease the power consumption of the *cc* compressor due to lower pressure lift required. Further, the secondary loop circulation introduced for *ice* evaporator is also expected to reduce power consumption by reducing one compressor from the overall circuit. However in CRS2, the performance of the LTC decreases due to elevation of T_{MC} and corresponding pressure which increases load in cascade condenser, resulting in higher refrigerant flow rate in cc compressor and ultimately increasing its power consumption beyond the benefits derived as stated earlier. The power consumption of various compressors and pumps of proposed CRS systems at 30 ºC ambient (Fig. 4.9) explains the same. This effect is consistent across ambient temperature range investigated.

CRS3 introduces pumped circulation in *ice* evaporator in HTC which reduces power consumption as one compressor is eliminated over CRS1 configuration. However, to introduce pumped circulation, a larger volume of refrigerant, feeding both *ch* and *ice* evaporator, is expanded to a lower pressure which results in high power consumption in *ch* compressor and this pulls down the overall COP of the system. With increase in ambient temperature, the benefit from removing a compressor from HTC become more prominent, therefore, we observe COP of CRS3 system marginally overshoot that of CRS1 at higher ambient.

Overall benefits of the proposed cascade systems are reduction in number of compressors, reduction in pressure ratio in compression, reduction in total NH_3 charge in system, isolation of food from in proximity of NH_3 in cold storage & plate freezer as well as reduction in energy consumption. The system is also specifically designed keeping in view of the cooling loads at various temperature. These, along with low GWP and ODP of the refrigerants used are arguments in favour of the proposed system presenting it as a suitable option for new surimi plants in future.

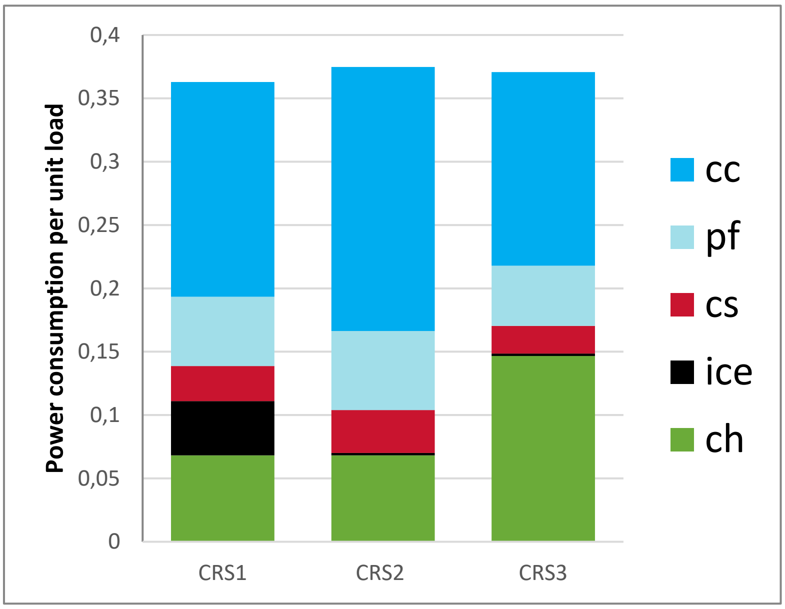


Fig. 4.9: Performance comparison at various ambient temperature

7 Implementation in industry

Based on our discussion with a limited number of seafood processing industries in India, it is found that the industries are not averse to adopt a cascade system with NH_3 and is forthcoming about new technology for their future installations, once they have access to a demonstrable benefit based on a pilot plant data in a warm climate. However, the industries are unlikely to come forward to invest in pilot plant project. As per local policy, there is neither any immediate pressure nor tangible benefit for industries from shifting over to other refrigerants from NH_3, therefore, the motivation of industries to invest in R&D is low.

8 Conclusions

Three novel CO_2-NH_3 cascade refrigeration systems (CRS1, CRS2 & CRS3) having evaporators above and below the cascade temperature were analysed. The simulation results were compared with that of a conventional NH_3 designed to cater to typical cooling demands in a surimi processing plant and associated frozen storage. For specific system and operating condition, cascade condensing temperature T_{MC} was optimized to maximize the COP. The value of optimum T_{MC} was found to vary with T_{cond} as well as refrigeration load ratio as expected. CRS1 configuration exhibited superior performance in a surimi processing plant for a wide range of condensing temperature for tropical regions. CRS3 configuration performs marginally better when the ambient temperature is above 41 ^{o}C. This study suggests possible gainful implementation of CO_2-NH_3 cascade system in meeting refrigeration demands at surimi processing plant for tropical region.

9 Further work

Before implementation, however, an economic study based on component fixed cost, variable cost of year round operation, influence of local policy of market incentive or penalty and long-time policy outlook needs to be examined to ascertain superiority of a new scheme compared to conventional NH_3 system.

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APPENDIX A.1

Refrigeration load calculation for a 10MT/day surimi production plant

For a 10 MT per day capacity plant, 1000 blocks of surimi of 10 kg each can be produced per day. These should be frozen within 24 h, including loading and unloading time. Observed freezing time for a batch is 2 h, additionally loading and unloading time is 0.5 h, therefore, for minimum load operation, we require operating the plate freezer in 9 batches every day, each batch for freezing 112 surimi blocks. In this process, 3425 kJ energy must be removed from one surimi block to cool from 10 ºC to -35 ºC (Park et al., 2013). Refrigeration load of a plate freezer to freeze 112 blocks in 2 h is, hence, 53 kW. Assuming an additional 10% cooling load for heat ingress and other losses in the plate freezer, the total freezer load is 60 kW. Frozen surimi blocks after packaging is shifted to cold storage maintained at -20 ºC. The capacity of the cold storage to be designed is obtained assuming capability to store 5 months of production volume which is about 1500 MT of surimi having a volume 1752 m3. A cold storage having a gross volume of 5256 m3 is considered suitable for the application with 200% extra space for circulation and approach. Accordingly, a single storey 7 m tall square cross-section storage space having length and width of 28 m is considered for estimating the cold storage refrigeration demand. The refrigeration load is estimated as per NHB standard 01:2010 (NHB, 2010). Assuming heat transfer coefficient for walls, roof, and floor as 0.58, 0.24 and 0.29 Wm-2K-1 respectively, total transmission load is about 48 kW. Assuming 2 air changes per day due to openings of gate during loading, unloading, and various leakages, approximately 10 kW is added. For the circulation of cold air, six blowers each of 1 kW rating operating at an average 20 h/day contribute 6 kW of additional load. 10 Wm-2 of lighting for an average 6 h/day leads to about 1.96 kW of cooling load. For loading and unloading, assuming that five persons are working inside the cold room for an average 2 h/day with each adding 250 W, the additional occupancy load is 0.11 kW. Thus, the total estimated refrigeration load in the cold storage due to transmission, infiltration, air distribution, lighting and occupancy is 70 kW.

Computation of cooling load for chilled water and ice requirement is shown in Table 1, taking a peak ground water temperature of Mumbai as 30 ºC. The design refrigeration load is ~300 kW.

Table A1.1: Cooling demands computed for various temperature applications

Parameter	Evaporators	Amount	Product Temperature (ºC)	Evaporation Temperature (ºC)	Refrigeration load (kW)	Load %
Chilled water	ch	100 MT/day	7	2	115	38.4
Ice	ice	10 MT/day	0	-5	55	18.3
Freezing	pf	10 MT/day	-35	-40	60	20.0
Cold storage capacity	cs	1500 MT	-20	-25	70	23.3

APPENDIX A.2

NH_3 system modeling

A pipe diagram and a P-h plot of the baseline conventional NH_3 based vapour compression refrigeration system is shown in Fig. 4.1. The system has four evaporators, termed *ch*, *ice*, *cs*, and *pf* for chilled water, ice, cold storage and plate freezing applications, respectively. Outlet of these evaporators are at state points 1, 3, 5 and 7 respectively. All the evaporators are flooded type and have separate individual expansion valves, which have refrigerant inlet from the receiver. Due to the low pressure-ratio between condenser and *ch* as well as *ice* evaporators, a single stage compressor is used. In cold storage and plate freezer applications, the pressure ratios are higher; hence, a two-stage compressor with intercooling is used. For intercooling, refrigerant is expanded from receiver up to the intermediate pressure and mixed with compressed refrigerant of first stage compressor. The compressor discharge for all the lines are at the same pressure and heat rejection takes place in a water-cooled evaporative condenser. Condensed NH_3 is collected in a receiver. Table 2: Mass and energy balance equations for system components

Table A2.2: Mass and energy balance equations for system components

Cooling Load	Component	Mass Balance	Energy Balance
Chilled water	Expansion Valve	$\dot{m}_{12'} = \dot{m}_{13}$	$h_{12'} = h_{13}$
	Evaporator	$\dot{m}_1 = \dot{m}_{13}$	$\dot{m}_1 h_1 = \dot{Q}_{ch} + \dot{m}_{13} h_{13}$
	Compressor	$\dot{m}_1 = \dot{m}_{1a} = \dot{m}_2$	$\dot{m}_2 h_2 = \dot{W}_{ch} + \dot{m}_{1a} h_{1a}$
Ice production	Expansion Valve	$\dot{m}_{12'} = \dot{m}_{14}$	$h_{12'} = h_{14}$
	Evaporator	$\dot{m}_{14} = \dot{m}_3$	$\dot{m}_3 h_3 = \dot{Q}_{ice} + \dot{m}_{14} h_{14}$
	Compressor	$\dot{m}_3 = \dot{m}_{3a} = \dot{m}_4$	$\dot{m}_4 h_4 = \dot{W}_{ice} + \dot{m}_{3a} h_{3a}$
Cold storage	Expansion Valve	$\dot{m}_{12'} = \dot{m}_{15}$	$h_{12'} = h_{15}$
	Evaporator	$\dot{m}_{15} = \dot{m}_5$	$\dot{m}_5 h_5 = \dot{Q}_{cs} + \dot{m}_{15} h_{15}$
	1st stage Compressor	$\dot{m}_5 = \dot{m}_{5a} = \dot{m}_{5b}$	$\dot{m}_{5b} h_{5b} = \dot{W}_{cs1} + \dot{m}_{5a} h_{5a}$
	Intercooling mixing	$\dot{m}_{5b} + \dot{m}_{15'} = \dot{m}_{5c}$	$\dot{m}_{5c} h_{5c} = \dot{m}_{5b} h_{5b} + \dot{m}_{15'} h_{15'}$
	2nd stage compressor	$\dot{m}_{5c} = \dot{m}_6$	$\dot{m}_6 h_6 = \dot{W}_{cs2} + \dot{m}_{5c} h_{5c}$
Plate freezer	Expansion Valve	$\dot{m}_{12'} = \dot{m}_{16}$	$h_{12'} = h_{16}$
	Evaporator	$\dot{m}_{16} = \dot{m}_7$	$\dot{m}_7 h_7 = \dot{Q}_{pf} + \dot{m}_{16} h_{16}$
	1st stage Compressor	$\dot{m}_7 = \dot{m}_{7a} = \dot{m}_{7b}$	$\dot{m}_{7b} h_{7b} = \dot{W}_{pf1} + \dot{m}_{7a} h_{7a}$
	Intercooling mixing	$\dot{m}_{7b} + \dot{m}_{16'} = \dot{m}_{7c}$	$\dot{m}_{7c} h_{7c} = \dot{m}_{7b} h_{7b} + \dot{m}_{16'} h_{16'}$
	2nd stage compressor	$\dot{m}_{7c} = \dot{m}_8$	$\dot{m}_8 h_8 = \dot{W}_{pf2} + \dot{m}_{7c} h_{7c}$
	Mixing in discharge line	$\dot{m}_{11} = \dot{m}_2 + \dot{m}_4 + \dot{m}_6 + \dot{m}_8$	$\dot{m}_{11} h_{11} = \dot{m}_2 h_2 + \dot{m}_4 h_4 + \dot{m}_6 h_6 + \dot{m}_8 h_8$
	Condenser	$\dot{m}_{11} = \dot{m}_{12}$	$\dot{m}_{12} h_{12} = \dot{m}_{11} h_{11} - \dot{Q}_{cond}$

Overall coefficient of performance (COP), one of the metrics used for performance analysis, is given by:

$$COP = \frac{\dot{Q}_{ch}+\dot{Q}_{ice}+\dot{Q}_{cs}+\dot{Q}_{pf}}{\dot{W}_{ch}+\dot{W}_{ice}+\dot{W}_{cs1}+\dot{W}_{cs2}+\dot{W}_{pf1}+\dot{W}_{pf2}} \quad (3)$$

Compressor work ($\dot{W}$) is estimated as given below:

$$\dot{W} = \frac{(h_{out,s} - h_{in})}{\eta_s} \tag{4}$$

Compressor isentropic efficiency (η_s) depends on the compressor pressure ratios. Based on the manufacturer data of refrigeration capacity, power consumption and mass flow rate for designated load, expressions for isentropic efficiency with respect to pressure ratios are developed using polynomial fitting through regression analysis and are presented in Table 3.

Table A2.3: Relation between compressor isentropic efficiency and pressure ratio

Compressor	Pressure ratio (R)	Isentropic efficiency
pf	$R_{pf} = P_{cond}/P_{7b} = P_{7b}/P_7$	$\eta_{s,pf} = 0.0107{R_{pf}}^3 - 0.1801{R_{pf}}^2 + 0.981R_{pf} - 1.030$
cs	$R_{cs} = P_{cond}/P_{5b} = P_{5b}/P_5$	$\eta_{s,cs} = 0.0596{R_{cs}}^3 - 0.6135{R_{cs}}^2 + 2.0485R_{cs} - 1.493$
ice	$R_{ice} = P_{cond}/P_3$	$\eta_{s,ice} = 0.0032{R_{ice}}^3 - 0.032{R_{ice}}^2 + 0.139R_{ice} + 0.549$
ch	$R_{ch} = P_{cond}/P_1$	$\eta_{s,ch} = 0.0032{R_{ch}}^3 - 0.032{R_{ch}}^2 + 0.1392R_{ch} + 0.549$

Initial values of thermodynamic parameters considered in the simulation are listed in Table 4.

Table A2.4: Thermodynamic parameters considered for simulation

Parameters	Value (ºC)
Ambient temperature (T_{amb})	30
Approach temperature of NH_3 condenser (ΔT_{app})	5
Condensing temperature of NH_3 (T_{cond})	35
Degree of suction superheat in ch evaporator ($T_{suction\_sh,ch}$)	7.5
Degree of suction superheat in ice evaporator ($T_{suction\_sh,ice}$)	10
Degree of suction superheat in cs evaporator ($T_{suction\_sh,cs}$)	15
Degree of suction superheat in pf evaporator ($T_{suction\_sh,pf}$)	20
Degree of superheat at evaporator outlets (T_{sup})	0
Degree of sub cooling after condenser (T_{sub})	0
Temperature difference between product temperature and refrigerant evaporation temperature ($\Delta T_{product}$)	5
Temperature difference between condenser temperature and evaporator temperature of cascade condenser (ΔT_{cc})	5

The system model is developed using EES (Klein, 2018) for the simulation based study. Simulation results are discussed in main article, results section.

APPENDIX A.3

CRS1, CRS2 & CRS-3 system and modelling

The relatively higher compressor work required in the lower temperature refrigeration (Fig. 4.5) and favourable thermos-physical properties of CO_2 at lower temperature is the trigger to incorporate CO_2 in low temperature evaporators *cs* and *pf*. CRS1 is a cascade and CRS2 and CRS3 are cascade with secondary loop refrigeration systems. In CRS1, CO_2 is used as refrigerant in the two LT evaporators (*cs* and *pf*) while NH_3 is used in the two medium temperature evaporators (*ch* and *ice*). In CRS2 and CRS3, the *ice* evaporator is integrated with pumped circulation of CO_2 and NH_3 respectively to investigate the suitability of the secondary refrigeration system. In CRS2 system, the evaporation temperature of ice evaporator is set equal to condensing temperature of LTC. Flash gas generated in the receiver due to heat addition in ice evaporator is removed from the receiver and mixed with compressed CO_2 of *pf* and *cs* compressor. In CRS3, liquid NH_3 is circulated in *ice*, ch and *cc* evaporator while the flash gas is separated from receiver and mixed with *ch* evaporator outlet before compressor suction in HTC. The pump circulation ratio, the ratio of total mass of liquid vapour mixture to mass of vapour phase refrigerant, is considered as 1.5 in the analysis for effective heat transfer.

The assumptions made for modelling of the various cascade systems are the same as in the case of baseline system. Mass and energy balance used for each component is the same except the cascade condenser and pump power. The expression for heat balance for the cascade heat exchanger is given below.

$$\dot{Q}_{cc} = \dot{m}_{11}(h_{11} - h_{12}) = \dot{m}_{20}(h_{20} - h_{24}) \quad (5)$$

Required mass flow rate in the secondary pumped loop through *ice* evaporator is expressed as:

$$\dot{m}_3 = \frac{\dot{W}_p}{(h_{14} - h_{12})} \quad (6)$$

where $\dot{W}_p$ is pump power required to circulate the refrigerant which is assumed to be 1% of the total refrigeration load of the ice evaporator. The compressor used in *ch* and *ice* in all the three systems is the same as in the baseline system. COP_H, COP_L, and COP_{net} are coefficient of performance for the LTC, HTC and for net cycle respectively and are calculated using Eq. (7) for CRS1 and CRS3 and Eq. (8) for CRS2.

$$COP_L = \frac{\dot{Q}_{cs}+\dot{Q}_{pf}}{\dot{W}_{cs}+\dot{W}_{pf}}; \quad COP_H = \frac{\dot{Q}_{ch}+\dot{Q}_{ice}+\dot{Q}_{cc}}{\dot{W}_{ch}+\dot{W}_{ice}+\dot{W}_{cc}}; \quad COP_{net} = \frac{\dot{Q}_{ch}+\dot{Q}_{ice}+\dot{Q}_{cc}+\dot{Q}_{cs}+\dot{Q}_{pf}}{\dot{W}_{ch}+\dot{W}_{ice}+\dot{W}_{cs}+\dot{W}_{pf}+\dot{W}_{cc}} \quad (7)$$

$$COP_L = \frac{\dot{Q}_{ice}+\dot{Q}_{cs}+\dot{Q}_{pf}}{\dot{W}_{p}+\dot{W}_{cs}+\dot{W}_{pf}}; \quad COP_H = \frac{\dot{Q}_{ch}+\dot{Q}_{cc}}{\dot{W}_{ch}+\dot{W}_{cc}}; \quad COP_{net} = \frac{\dot{Q}_{ch}+\dot{Q}_{ice}+\dot{Q}_{cc}+\dot{Q}_{cs}+\dot{Q}_{pf}}{\dot{W}_{ch}+\dot{W}_{p}+\dot{W}_{cs}+\dot{W}_{pf}+\dot{W}_{cc}} \quad (8)$$

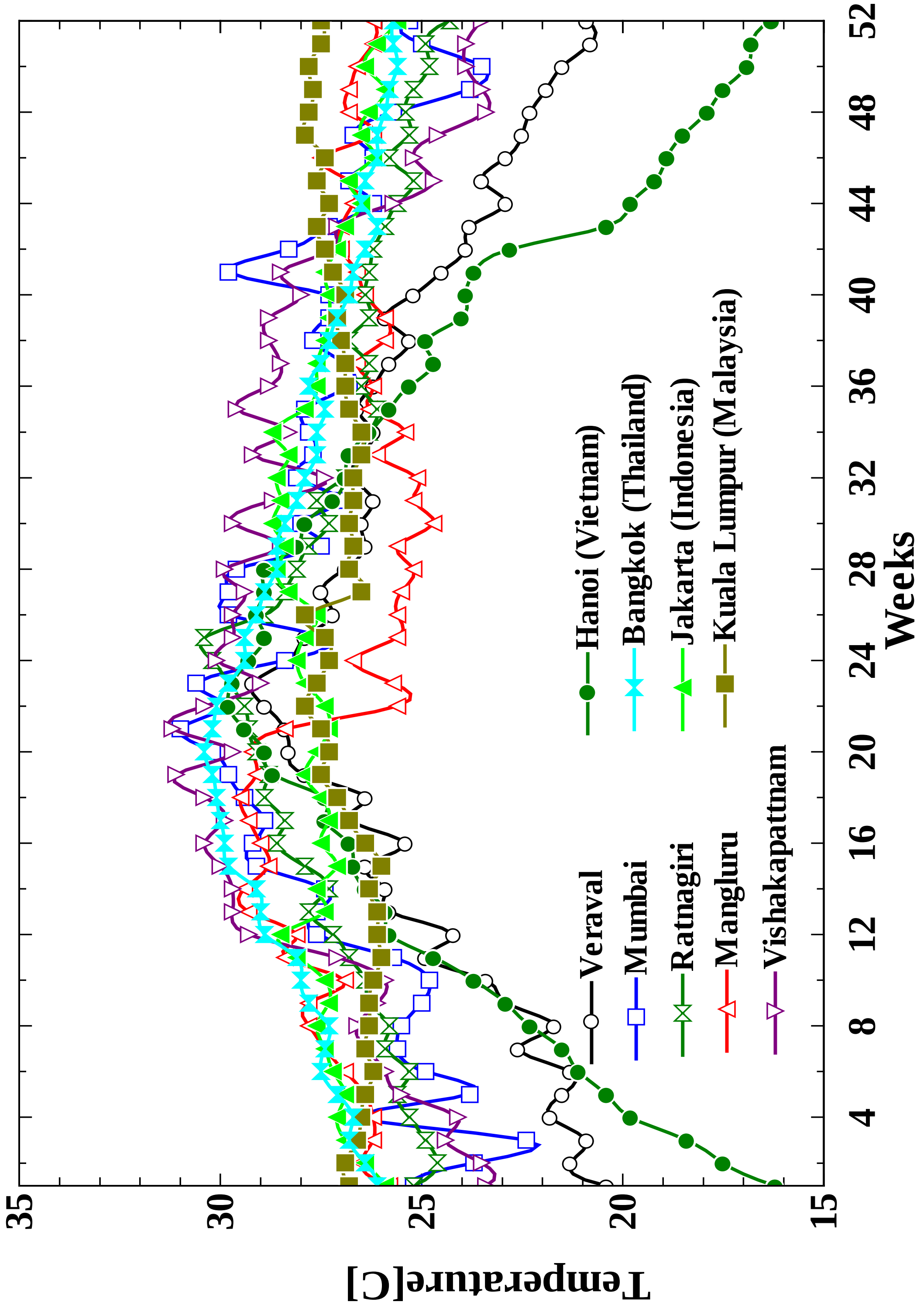
In the analysis, we used the same relation between compressor isentropic efficiency and compression ratio for both *pf* and *cs.* For CO_2 compressors in LTC, expression reported in Patel et al. (2019) is employed and is tabulated in Table 5. Initial thermodynamic parameter values used are the same as in Table 4.

Table A3.5: Relation between compressor isentropic efficiency and pressure ratio

Compressor	Pressure ratio (R)	Isentropic efficiency
pf	$R_{pf} = P_{MC}/P_7$	$\eta_{s,pf} = 0.00476{R_{pf}}^2 - 0.09238R_{pf} + 0.89810$
cs	$R_{cs} = P_{MC}/P_5$	$\eta_{s,cs} = 0.00476{R_{cs}}^2 - 0.09238R_{cs} + 0.89810$
cc	$R_{cc} = P_{cond}/P_{ME}$	$\eta_{s,cc} = -0.00097{R_{cc}}^2 - 0.01026R_{cc} + 0.83955$

APPENDIX A.4

Weekly averaged dry bulb temperature data



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