Identification and Evaluation of Energy Efficient and Environmentally Benign Refrigeration Technologies for Seafood Cold Chains in India

THESIS

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by

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Certificate

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Abstract

Fish and majority of seafood are highly perishable in nature and the quality starts deteriorating immediately after harvest due to various pathogenic, biological and chemical reactions resulting in quality degradation and large post-harvest losses. Preservation at low temperature using refrigeration is of paramount importance to maintain its quality. The world is conscious about contribution to global warming from refrigeration and air-conditioning sector. As per an assessment, refrigeration sector alone is responsible for about 7.8% of global GHG emissions due to its direct and indirect emissions. The pathway to implement the alternative refrigeration technologies is to reduce dependency on high GWP refrigerants and adoption of low GWP alternatives with energy efficient technologies. India has fulfilled the Montreal Protocol targets by phasing out CFC and HCFC fluids ahead of schedule and ratified the Kigali Amendment in September 2021 to oversee phase down of HFCs. The choices made in the near future on refrigerants in seafood cold chains could increasingly be influenced based on long and short-term policy conditions incorporating market incentives or penalties.

Further, there is a continuous quest for improved technology to reduce energy consumption of refrigeration systems to reduce operating cost as well as to reduce indirect emissions. The environmental implication about the choice of new equipment, systems and choice of refrigerant are likely to be increasingly open to review, particularly with respect to GHG emissions associated with the sourcing and location of use. A search is on for energy-efficient refrigeration systems and environmentally friendly refrigerants. There are significant environmental and possible economic benefits from switching over to low GWP alternatives and to explore possible design changes to meet specific cooling demands. Natural refrigerants such as ammonia (NH_3), carbon dioxide (CO_2), hydrocarbons etc. are gaining importance due to their competitive thermophysical properties and environmentally friendly nature.

Literature related to various methods of preserving fish immediately after the harvest in sea are reviewed. Based on techno-economic scenario of fishermen in India, a novel compensatory refrigeration system is proposed. It is designed to compensate for the heat ingress into the insulated storage compartments and ensure storage near 0 °C. Heat from diesel engine exhaust of the fishing boat is used as heat source for the absorption cooling device and sea water is

used as heat sink for the vapour absorption refrigeration (VAR) system. The payback period of the VAR is found to be about 1 year 4 months.

This study is also intended to comprehensively explore environmentally friendly refrigeration systems with improved energy efficiency that are suitable for specific seafood sector such as surimi. An assessment of the refrigeration needs across the seafood cold chain in India was carried out based on site visit and literature survey, identifying potential temperature breaches, energy inefficiency in the refrigeration systems and use of refrigerants at various stages. There are various low temperature demands in seafood processing and preservation industry where a cascade refrigeration system (CRS) can be gainfully employed. Extensive mathematical modeling and simulation of state-of-the-art refrigeration systems are carried out to derive the recommended systems. The study has also embarked on to review, evaluate and identify potential low GWP refrigerants that can be paired for maximum benefits in a CRS.

This study proposed three novel CO_2-NH_3 multi-evaporator CRS and evaluated the same for application in seafood processing for high ambient temperature of tropical region. These systems essentially exploit the comparatively superior thermo-physical properties of CO_2 in the low temperature evaporators while NH_3 in the high temperature condenser keep the overall operating pressure low. The CRSs improve the overall efficiency, reduce ammonia charge and reduce food contamination hazards from NH_3. Suitability of the proposed CRSs are established for a wide range of tropical ambient temperature and also for cooling demands in other prominent seafood processing and storage applications such as fish fillets and shrimp/prawn.

An all-natural multi-evaporator CO_2-NH_3 CRS is analyzed and compared with a conventional system having R22, R404A, and NH_3. Energy, environmental and economic parameters are used for the comparison. The CRS also exhibited better performance in the economic front in terms of annual cost rate and total life cycle cost estimate. Further, to enhance the performance of the CRS, integration of an internal heat exchanger (IHX) is proposed. The IHX in the sub-critical CO_2 refrigeration system produces required sub-cooling at the exit of the condenser, increasing the refrigeration effect, and thus the overall performance improves.

This study also investigated various potential environmentally friendly refrigerants suitable for seafood processing and storage application for multi-target temperature in hot climates. Performance is investigated and compared based on energetic, environmental and a few important design parameters. An investigation of energetic and environmental benefits from adoption of suitable natural refrigerants at various stages of the entire surimi and shrimp cold chains under various scenario was also undertaken. Additionally, an attempt is made to identify various technological, economic and policy related barriers towards speedy adoption of environmentally friendly refrigeration technologies in India.

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Nomenclature

Abbreviations

A	Heat exchanger area (m^2)
ACR	Annual cost rate (US \$ $year^{-1}$)
AEC	Annual energy consumption (kWh)
AOC	Annual operating cost (US \$)
cc	Cascade condenser refrigerant line
CFC	Chlorofluoro carbon
ch	Chilled water refrigerant line
COP	Coefficient of performance
c_p	Specific heat
CRS	Cascade refrigeration system
cs	Cold storage refrigerant line
CSW	Chilled Sea Water
d_i	Refrigerant gas density (kg m^{-1})
$\dot{E}x_d$	Exergy destruction (kW)
FAO	Food and agricultural organization
FY	Financial year
GHG	Greenhouse gas
GWP	Global warming potential
h	Specific enthalpy (kJ kg^{-1})
HC	Hydrocarbons
HCFC	Hydro chlorofluoro carbon
HFC	Hydro fluoro carbon
HFO	Hydro fluoro olefins
hp	Horse power
HTC	High temperature circuit
ICAP	India Cooling Action Plan
ICC	Initial capital cost (US \$)

ice	Ice refrigerant line
IHX	Internal heat exchanger
IIR	International Institute of Refrigeration
L	Latent heat (kJ kg^{-1} K^{-1})
LCC	Life cycle cost (US $)
LMTD	Log mean temperature difference (K)
LR	Liquid receiver
LTC	Low temperature circuit
$\dot{m}$	Mass flow rate (kg s^{-1})
MC	Maintenance cost (US $)
m_{charge}	Refrigerant charge (kg)
MPEDA	Marine products export development authority
N	Number of operational years
n, m	Number of refrigerants
NBP	Normal boiling temperature (ºC)
ODP	Ozone depleting potential
P	Pressure (MPa)
pf	Plate freezer refrigerant line
PMMSY	Pradhan Mantri Matsya Sampada Yojana
$\dot{Q}$	Refrigeration load in evaporators (kW)
R	Ratio of compressor discharge to inlet pressure
RRM	Rest raw material
RSW	Refrigerated sea water
s	Entropy
SDG	Sustainable Development Goals
SHX	Solution heat exchanger
T	Temperature (ºC)
TEWI	Total equivalent warming impact
T_{MC}	Cascade condenser temperature (ºC)
TPD	Tonnes per day
U	Overall heat transfer coefficient (W m^{-2} K^{-1})

UNEP	United Nations environment programme
V	Volume of refrigerated compartment
VAR	Vapour absorption refrigeration/system
VCR	Vapour compression refrigeration
ΔT	Temperature difference (ºC)
$\dot{W}$	Power consumption in compressor (kW)

Greek Letters

α	Recycling factor
β	Electricity regional conversion
ε	Exergy efficiency
ϵ	Heat exchanger effectiveness
η	Efficiency (%)
λ	Solution circulation ratio
ρ	Density of cold air

Subscripts

1, 1a	State points
abs	Absorber
amb	Ambient
cond	Condenser
crit	Critical
evap	Evaporator
f	Fish
GC	Gascooler
gen	Generator
H	HTC
I	Ice
infr	Infiltration
L	LTC
p	Pump
rect	Rectifier
s	Isentropic

CHAPTER 1 Introduction

1. Introduction

The growing population and resulting growth in demand for food and nutrition puts prime importance on enhancing food production and reduction of food loss during harvesting, storage and distribution. Uninterrupted cold chain logistics play pivotal role in maintaining product quality and optimum shelf-life along the entire food value chain. A cold chain comprises a sequence of logistical activities with controllable variables like temperature, humidity etc. between production to consumption stage in a perishable supply chain (Mercier and Uysal, 2018; Sarr *et al.*, 2021). The necessity of a cold chain gets amplified in a country like India, where it serves a dual purpose of satisfying the in-house requirements for its large population and in maintaining its position as a leading global exporter (Setia, 2019).

It is estimated that out of the global total food lost or wasted i.e. 1.3 billion tons per year, about 14% of food is lost at post-harvest stages and 17% is wasted at consumer end every year, in monetary terms, a loss of about $936 billion a year to the global economy (UNEP, 2021 and FAO, 2022). The global food loss, a total of 526 million tonnes of food loss or waste (enough to feed 1 billion people) is due to the lack of appropriate cold chain coverages (UNEP and FAO, 2022). Globally, food cold chains are responsible for an estimated emission of about 1.01 billion tonnes of CO_2 equivalent that is 4% of total GHG emissions, including emission due to cold chains and methane emissions caused due to food loss and waste. Majority of the food waste occurs at the consumption stage in high income regions, however, in India most of the losses occur at the post-harvest stage due to improper handling and limitation of cold chain infrastructure (Thakur *et al.*, 2021). About 55% of the current global food loss can be avoided which is also responsible for about 47% of the total CO_2 emission from food cold chains (Sarr *et al.*, 2021) by developing suitable cold chains in the low income regions.

In India, with an estimated food waste or loss of 40% in the food sector summing to $5.4 billion, the requirement of an appropriate and connected cold chain is highly essential (NAAS, 2019). Added derivatives from the report (Emerson Report, 2015) clearly indicate a trivial in-country storage capacity of 11% for its self-produced products, obliging the need of a $6-$10 billion investment in the cold chain sector (Gonsalves, 2017). Cold chains require robustness in terms of continuity when fresh products with short shelf-life such as seafood are handled. India's share of seafood discards sums up to an annual loss of $1.85 billion, many factors attribute to the losses, but inadequate cooling and interrupted cold chains contribute the most (Banu and Lunghar, 2019). Food processing is recognized as a priority sector by the Government of India (Rais, 2013), aiming technological advancements in handling and quality assurance throughout the entire supply chain. The food supply chains have a huge potential of improvements in India with appropriate cold chain integration. Fish and seafood is one such sector which is given importance under the national scheme of Pradhan Mantri Matsya Sampada Yojana (PMMSY, 2020). Significant amount of energy and fuel is consumed in the fisheries sector. Its vulnerability to the changing mix of energy supplies and prices highlight the need to review the sector's energy and fuel efficiency level in conjunction with the increasing expectations.

To study various aspects of Indian seafood cold chain such as supply chain management, thermal management, biotechnology of seafood etc., and to identify several solutions to enhance the overall resource utilization, a three-year (2018–21) project "ReValue: Innovative technologies for improving resource utilization in the Indo-European fish value chains" was launched among collaborating institutes from India, Norway and Spain. The project was jointly funded by the partner countries; in India Department of Biotechnology (DBT) funded the project in a joint call for bio-economy. This thesis is a part of the knowledge and information gained from the project in the field of thermal management of the seafood cold chain. In the project, first hand data was collected through direct observations of activities and processes at fish landing sites, pre-processing facilities, seafood processing and storage plants. Formal and informal interactions were made with the fishers, landing site managers, processing plant owners and technicians etc. followed by a detailed literature survey to collect the secondary data.

1.1 Seafood Cold Chain in India

Seafood is one of the most widely consumed foods worldwide and its popularity is growing over time. Globally, total fish captured in 2020 was about 175 million tonnes and is projected to increase by 15% by 2030 (FAO, 2020). India has large coastline over 7500 km long, along with inland water resources of 7 million hectares, which naturally makes it a major seafood and aquaculture producer and exporter. During the FY22, an estimated 14 million tonnes of seafood was produced in India, out of which 1.37 million metric ton (9.8%) was exported which made a earning of $7.8 billion (Ministry of Commerce & Industry, GOI, 2022).

Fish and majority of seafood are highly perishable in nature and the quality starts deteriorating immediately after harvest due to various pathogenic, biological and chemical reactions resulting in quality degradation and large post-harvest losses. Preservation at low temperature is by far the most acceptable method of lowering the rate of quality degradation and is of paramount importance to maintain quality in order to obtain good price in the market. Measurement and control of temperature of fish from the point of harvest until the fish reaches the consumer needs constant vigilance to ensure food quality, nutritional content and also provide traceability during the journey.

Depending on the end product and processing, the cooling demand in seafood supply chain varies. The frozen seafood supply chain, which includes shrimp/prawns, whole fish, filleted or minced fish meat, have prominent pre-processing requirements like heading, gutting, skinning, salting etc. mostly requiring human intervention before mechanized processing and deep freezing. Seafood has an acute requirement of maintaining low temperatures throughout the supply chain in order to retain its consistency, color, gel forming ability etc. that are important metrics for determining its commercial value. The temperature requirements are varied. Various stages in a typical seafood cold chain in India for processed seafood are presented in Figure 1.1.

1.1.1. **Fish harvesting**

A large number of marginal fishers are generally engaged in the coastal areas of India, who are major players in harvesting from sea. They venture into the sea in small motorized boats, mostly using the *purse seines gear* for fishing and carry crushed ice within compartments of the boats for preservation of fish during the trip, duration of which is can be between 5 to 10 days. Immediately after a catch, the same is sorted in the deck, the fish having commercial value are segregated in terms of type and size and are stored in different insulated chambers within the boat within layers of crushed ice and the rest is discarded in sea. The melting ice wets the fish and is an effective medium to carry away field heat and preserve the fish near 0 ºC. Ice consumed is in a ratio of 1:1 to 2:1 depends upon number of days and harvesting season (Shawyer *et al.*, 2003). In contrast, in a high income country like Norway, where seafood is the second largest contributor in its economy, the practices are quite different (Myrseth, 2022). Fishers in Norway predominantly use large vessels or trawlers having on-board processing and freezing plants which reduces the time between catch and freezing and also the storage temperature is lower thereby ensuring better preservation (Söylemez *et al.*, 2022).

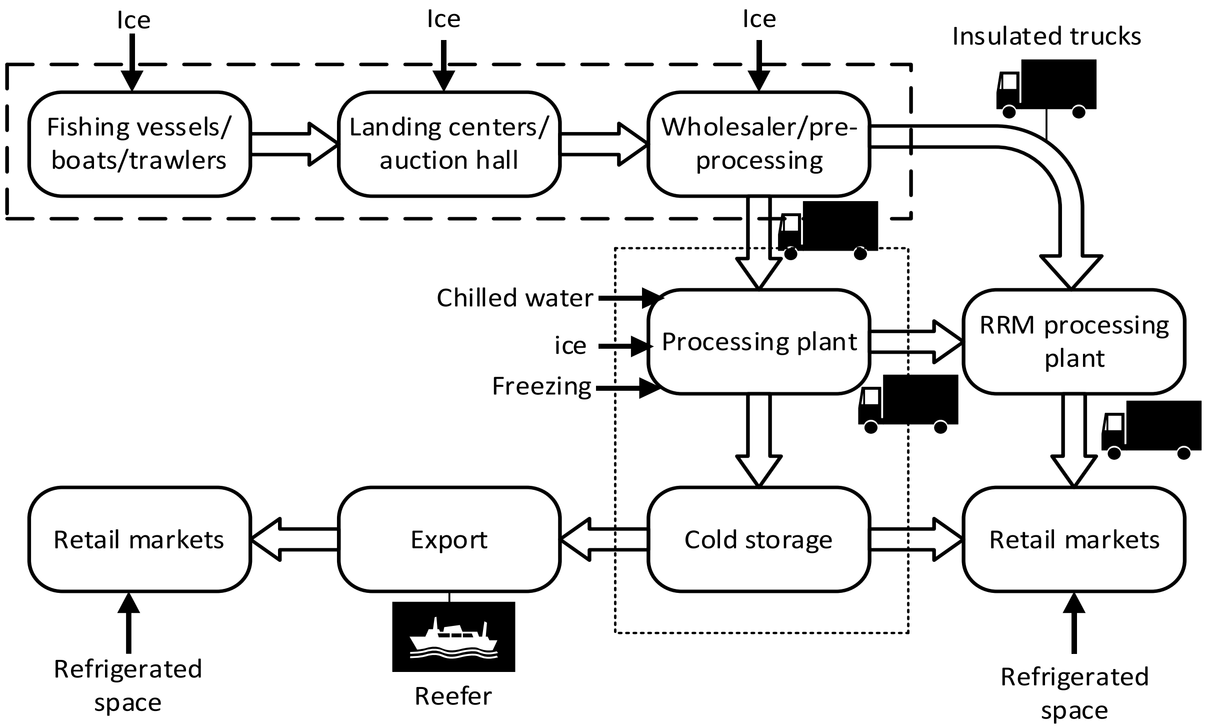


Figure 1.1: Flow of seafood through various stages in processed seafood cold chain

1.1.2. **Fish handling at landing centers/ preprocessing plants**

In India, the fish handling practice in the boat as well as at the fishing harbour are mostly manual. Apart from temperature breach, the quality may also be affected due to inappropriate handling, the presence of many manual touch points and improper storage or stacking conditions. Due to the fragmented nature of the supply chain prevalent in India, the catch changes hand at landing site requiring de-icing and weighing to settle payments. This practice leads to exposure of the fish to ambient temperature and is identified as one of the most prominent temperature breaches and also a source of pilferage and quality loss in the overall cold chain (Dasgupta *et al.*, 2019). Figure 1.2 shows pictures of activities carried out at various stages of a seafood cold chain during our survey. The general practice of unloading from the boat compartments is sequential depending on the commercial value of fish. It was also observed that a boat is sometimes needed to dock at multiple landing centers to unload different variety of catch. Landing at multiple sites also extends the time from catch to processing, which affect the quality of the seafood. In general, manual sorting and preprocessing in open space without any cooling arrangement at the landing sites is observed. After weighing and sorting, the pre-processed fish is then re-iced at landing centers. From here, most of the fishes are transported to local markets for retail sale while fish intended for processing are transported to the nearby processing plants in insulated trucks. Rest raw material (RRM) or low quality fishes are transported to RRM processing plants (Sultan *et al.*, 2021).

1.1.3. Transportation to processing plant

Insulated trucks, pick-ups or even open trucks are used to carry plastic crates filled with iced fish from landing sites to the processing plants. Additional quantity of crushed ice is used for transportation and the quantity of ice deployed varies with travel time and ambient temperature.

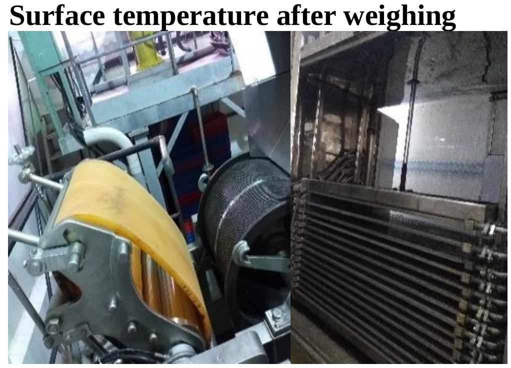
Common fishing boats in use

Manual fish unloading from boat

Fish for weighing

Surface temperature after weighing

Manual heading and gutting



Processing and freezing product

Processed seafood in cold storage

Loading product in insulated truck

Figure 1.2: Practice and processes at a fish landing centre

1.1.4. **Processing plants**

The processing plants either receive whole fish or pre-processed (headed and gutted) fish. As the supply side is dependent on many marginal fishers and is non-integrated, the processing plants faces challenges in terms of uncertainty of continuous supply (Dasgupta *et al.*, 2019). This necessitates intermediate storage facility for arriving fish before processing in the plant, in order to ensure adequate batch size for the processing. Additional ice is used during the waiting time to maintain low temperature. Based on the particular processing operations and type of seafood, the processing plants have large refrigeration demands to maintain controlled temperature conditions which is generally maintained by circulation of chilled wash water at 4 °C to 8 °C. Processed product requires quick freezing using plate/ blast/ spiral etc. freezers at temperatures about −40 °C and thereafter the packed product is stored in cold storages at about −18 °C for long time storage (Saini *et al.*, 2021).

1.1.5. **Cold storages**

Depending on the temperature and stage of cold chain, size of cold storages vary such as large cold storages near the processing plants to store bulk amount of seafood for long time, and small cold storages for distribution nearer the retailers (Gonçalves and Blaha, 2010). After deep freezing in plant, the processed and frozen product is stored in cold storage at temperature range about −18 ºC for long term storage. Suitable monitoring and recording of temperature data is necessary at a cold storage in order to ensure product quality and traceability.

1.2 Surimi as a Case Study

1.2.1 Introduction

Surimi is deboned and minced fish meat washed with water in multiple cycles and blended with cryoprotectants to avoid protein denaturation and to extend its frozen shelf life. Surimi mostly serves as intermediate product for making ornamental food like pseudo crab meat and many other ready to eat seafood products. It contains stabilized and concentrated myofibrillar proteins from fish (Park and Morrissey, 2000). Surimi derived products include both traditional Japanese fish cakes and imitation seafood products (e.g. ‘crab sticks’) which have great demand in Japan, USA and Europe. Japan is by far the largest importer of Indian surimi. Considering the large surimi demand in Japan, USA, and various European countries, there is also a huge potential for growth in Indian surimi export (MPEDA, 2021).With almost negligible domestic consumption in India, most of the frozen surimi from India is exported to USA, Japan and EU countries (MPEDA, 2021). Fish species such as Alaska Pollock, Pacific whiting, Threadfin bream, Bigeye snapper, Goat fish, Lizard fish and Sardine are mostly used for surimi production globally (Xiong, 2018). In India generally, fish species which are underutilized as whole fish, containing low fat white meat and have otherwise lower commercial value, are used for surimi production (Venugopal and Shahidi, 1998). The most favorable species in India is Threadfin bream or Pink perch as it also has good gelation properties suitable for surimi.

1.2.2 Surimi production in India

The global production of surimi is about 820,000 tonnes with a global market of over $4 billion in 2018. Tropical countries produce about 60% of the surimi and India alone contributes with about 11% of total global production (Seaman, 2018). Among the processed seafood exported from India, surimi is an important product. During 2017-18, India exported about $17.6 million worth of surimi which had a value share of 2.5% of total export earnings from seafood. The 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 therefore, located along the western coast in Gujarat, Maharashtra, Karnataka and Kerala. In India, the marginal fishermen carry out significant portion of fishing and a part of their catch which have lower commercial value is generally diverted to processing plants located nearby for surimi production. The frozen surimi is transported according to the demand in market mostly to overseas locations. The storage time of surimi in cold stores is governed by its demand in international markets, which generally peaks in December or January. However, the surimi production generally peaks after the monsoon usually around August, as fishing is restricted for about two months during monsoon (breeding season) in the western coast of India. Demand and price fluctuate in the market during a year, and raw material availability fluctuates as well. The time difference between the demand and production causes a long-time storage of surimi in cold storages.

1.2.3 Refrigeration in surimi cold chain

With increase in population and affordability, the refrigeration and cold chain demands are increasing across the globe but are more prominent in low income countries such as India where the infrastructure in development stage at present. Seafood and aquaculture supply chains have large refrigeration demands at various stages from harvesting to consumption leading to a large amount of energy consumption. Depending on the end-product and processing, the cooling demand in seafood supply chain varies. Cold chain requirements increase when processed frozen seafood products such as surimi are involved. The frozen seafood supply chain, which includes surimi and other minced meat, shrimp/prawns, and fish fillets, have significant refrigeration requirements during pre-processing/processing like heading, gutting, skinning etc. and during cold storages.

Surimi supply chain includes all the activities from "Sea" to "Plate" such as inbound logistics, warehousing practices, process sequence, management of RRM and finally the outbound logistics of surimi paste and various surimi products (Sultan *et al.*, 2021). In seafood cold chains, iced storage or chilled water is predominantly adopted to maintain low temperature at various stages including harvesting, activities at landing centers, transportation to processing plants during processing etc. The refrigeration systems to produce ice or chilled water mostly employ ammonia (NH_3), R404A or R22 refrigeration systems. Many such refrigeration plants near coastal areas are certified by MPEDA to supply of un-contaminated ice (Department of Fisheries, 2020).

Similarly, the processing plants and storage facilities have large refrigeration demands including ice at 0 ºC, chilled water at 4 to 8 ºC for washing during the processing, quick and deep freezing of product at about −40 ºC and long-term storage at about −20 ºC. The large quantity of chilled water circulation in the work place also contributes to maintaining a comfortable ambience for the operators. Majority of the plants are based on refrigerants such as NH_3, R404A, and R22. It is estimated that about 20 kWh of energy per ton of fish is consumed in the frozen seafood cold chains for fish processing, in which deep freezing and cold storage consume 50–70% of energy depending on the fish variety and process (Muir, 2015).

Depending on the cooling loads and large temperature difference between the refrigerated space and ambient, multi-evaporator, multi-stage compression refrigeration systems are common in the surimi processing plants. In these plants, due to the lower temperature lift, single stage compression refrigeration system are used for the chilled water and ice production, whereas, two-stage compression refrigeration system are deployed to achieve low temperature for cold storage and deep freezing application. Cooling technology used is of great economic and energetic importance for surimi production and various other stages of the supply chain. An energy efficient operation of vapour compression cycles not only help to save electrical energy but also is associated with saving of fossil fuel and greening the cold chain. Replacing use of synthetic refrigerants having high ozone depleting potential (ODP) and global warming potential (GWP) in the refrigeration process with natural refrigerants having zero ODP and negligible GWP are also important from sustainable development point of view.

1.3 Refrigeration and Its Environmental Impacts

Refrigeration is defined as a process of removing heat from a space or a body maintained at desired low temperature compared to the surrounding. Over the years, advancements in the design of refrigeration systems and components have led to enhancement in their efficiency, safety and durability. Vapour Compression Refrigeration (VCR) systems have dominated the refrigeration industry since 20th century. In a cyclic VCR process, a fluid called refrigerant vaporizes, compresses, condenses and expands repetitively at different temperatures and pressures. In this process, the refrigeration systems consume significant amount of electrical energy that indirectly leads to release of GHG to the atmosphere from burning of fuel. Some of the refrigerants in use also have unacceptable levels of ODP and GWP.

The refrigerants with significant ODP release chlorine when exposed to intense ultra violet light in the stratosphere. This released chlorine acts as a catalyst and chemically reacts with the oxygen molecules of ozone leading to depletion of the ozone layer in stratosphere. The ODP is a relative amount of degradation to the ozone layer the refrigerant can cause compared to R11 or CFC11. The GWP is a quantified cumulative measure of a gas which articulates the absorption capacity by unit mass of the gas of infrared radiation over a given time horizon or atmospheric lifetime of the gas compared to carbon dioxide (CO_2). A high GWP value of a gas means that it has capacity to absorb more radiative energy for long time leading to greater global warming. Refrigerants released to atmosphere due to leakages from the system during operation & maintenance or post disposal at the end of life, directly contributes in GHG emissions ultimately resulting to increase in global warming.

Most of the chlorofluorocarbons (CFCs) refrigerants having high ODP value are either already phased out (e.g. R11, R12) or is in the process of being phased out (e.g. R22) under the Montreal Protocol 1987 (UNEP, 2018a). Hydrochlorofluorocarbons (HCFCs) which have lower ODP compared to the CFCs were allowed to be used during the phaseout of CFCs because of their lower ODP . HCFCs were further regulated in revisions of Montreal Protocol due to their hazardous effect on the ozone layer and hydrofluorocarbons (HFCs) continue to be used. However, use of refrigerants with higher GWP (most of the HFCs) were regulated and being phased out globally under the Kigali Amendment to the Montreal Protocol (Heath, 2017). The Kigali amendment to the Montreal Protocol is regarding regulating the use of high GWP refrigerants worldwide and it aims to reduce the production and consumption of the same by 80% over the next 30 years (Kigali Amendment, 2016).

As per an assessment by the International Institute of Refrigeration (IIR), refrigeration sector alone is responsible for about 7.8% of global GHG emissions due to its direct and indirect emissions (Dupont *et al.*, 2019). The pathway to implement this regulation is to reduce dependency on high GWP refrigerants and adoption of low GWP alternatives with energy efficient technologies. Further, there is continuous quest for improved technology to reduce energy consumption of refrigeration systems to reduce operating cost as well as to reduce indirect emissions.

The environmental implication about the choice of new equipment and/ or systems and choice of refrigerant are likely to be increasingly open to review, particularly with respect to GHG emissions associated with the sourcing and location of use. India has fulfilled the Montreal Protocol targets by phasing out CFC and HCFC fluids ahead of schedule and will aspire to maintain the legacy. The choices made in the near future on refrigerants in seafood cold chains could increasingly be influenced based on long and short-term policy conditions resulting into market incentives or penalties.

India demonstrated leadership in 2019, by being one of the first countries to launch the India Cooling Action Plan (ICAP) to regulate refrigerant use and enhance cooling efficiency. In

September 2021, India ratified the Kigali Amendment and is working towards development of a national strategy in the next year to oversee phase down of HFCs in consultation with the stakeholders (Chatterjee, 2022). India as member of Group 2 countries as per the Kigali amendments, have a longer time frame for reduction of use of HFCs than European countries which are in Group 1 (Heath, 2017). The de-accelerated phasedown presents significant advantages for the India to leapfrog to the best alternatives later.

1.4 Motivation of the Study

In order to support the ever-increasing cold chain requirements, the global research community is working to find environmentally effectual, sustainable and affordable refrigeration systems that can ensure safety of the perishable products at various stages of the cold chains. Research and analysis to develop energy efficient, and environmentally benign refrigeration system is an imperative.

A full assessment of the refrigeration needs across the surimi cold chain in India, identifying potential temperature breaches, energy inefficiency in the refrigeration systems and use of refrigerants at various stages is the first step towards delivering sustainable cold chain in association with the climate goals. Some of the well-recognised strategies to improve the energy efficiency of refrigeration system as recommended by UN Environment Program (UNEP, 2018b) are explored in this study such as minimising the cooling load, selecting the most efficient refrigeration cycle and components, design of effective control systems, examining operation performance and correcting any faults of existing systems.

Additionally, there are challenges in warm climates. The refrigeration systems used in high ambient temperature climates generally use more energy than equivalent systems in cooler climates, because there is higher cooling load for a given mass of content (surimi) for higher condensing temperature or higher temperature lift; e.g. 1 ºC extra temperature lift causes about 2-4% additional energy consumption (UNEP, 2018b). However, there is significant potential to improve the efficiency of refrigeration equipment when new equipment and/or system is designed and fabricated specifically for the desired load condition.

Due to the high energy consumption and high refrigeration demands in the surimi processing and storage plant, this study aims to better understand those needs. As mentioned in *Section 1.2.4.*, refrigeration consumes a significant part of energy in surimi processing and storage plant; multi-evaporator multi-stage refrigeration systems are commonly used to meet the refrigeration demands. Most of the seafood processing plants are located in the coastal region of India that have a warm or hot climatic condition throughout the year. The temperature difference between ambient and some of the evaporators are higher than 70 ºC for most part of a year. Therefore, two-stage compression is used along with intermediate cooling. Studies shows that when the temperature difference between the product and ambient is high, performance of a cascade refrigeration system (CRS) is comparatively superior (Bhattacharyya *et al.*, 2005; Dopazo and Fernández-Seara, 2011). This study intends to identify the applicability of a CRS to the surimi processing and storage plants. Along with the system efficiency, use of synthetic refrigerants with high GWP have alarming impact on environment. There are significant environmental and possible economic benefits from switching over to low GWP alternatives and to explore possible design changes to meet the specific cooling demand. Natural refrigerants such as ammonia (NH_3), carbon dioxide (CO_2), hydrocarbons etc. are gaining importance due to their competitive thermo-physical properties and performance and environmentally friendly nature. Refrigerants that are commonly employed in seafood cold chains in India include R22, R404A, R134a, R410A, R407C and NH_3 (Kumar *et al.*, 2018), some of which have high GWP and leading to high greenhouse gas emissions directly on release to the atmosphere.

The United Nations Sustainable Development Goals (SDGs) indicator 12.3 on *Global Food Loss and Waste* purposes to halve the per capita global food waste at the upstream side and reduce the global food loss at the downstream side of the food supply chains by 2030 (United Nations, 2015). The present study targets food loss reduction by evaluating and applying enhanced refrigeration technologies for temperature control along the supply chain of selected seafood and RRM processing sector and will help in reaching SDG2 target of zero hunger. The study also analysed various methods of reducing energy use and environmental emission in refrigeration at various stages of the cold chain by improving energy efficiency of the refrigeration systems and adopting low GWP refrigerants which meets the targets of SDG7 of efficient energy use and SDG12 of climate action (United Nations, 2015).

To conclude, this study is intended to comprehensively explore refrigeration systems with improved energy efficiency and suitable for specific seafood sector like surimi. Furthermore, the study has emphasized to review, evaluate and identify potential low GWP refrigerants that can be paired for applications in CRS to obtain maximum benefits.

1.5 Organization of the Thesis

The thesis is structured in six chapters. Overall motive of each chapter is briefly stated below:

Chapter 1 presents a brief background of the seafood cold chains and the research context. It discusses how refrigeration is an important weapon to reduce food loss and waste. The state of art typical Indian seafood cold chain is briefly discussed as well. Further, a brief introduction about surimi, which is taken as a case study in the thesis, its production in India, process involved in the surimi production and refrigeration requirements at various stages are explained. Besides, the environmental impacts related to the refrigeration systems are deliberated, followed by a discussion on the motivation of the study. In the end, overall structure of the thesis is stated.

Chapter 2 presents a comprehensive review of literature supported by first hand data collected from site visits and stakeholder interactions. The literature survey conducted includes various low temperature seafood preservation methods, and refrigeration systems on-board. The Literature review also presents studies reported on refrigeration demands in a surimi processing and storage plant, refrigeration systems, and limitation of a two-stage refrigeration system and importance of CRS for such applications. This chapter also includes the studies that reported history of refrigerants, various regulations responsible for the refrigerant transition and potential low GWP alternatives for the future. At the end of each section of this chapter, research gaps in the particular research domain are highlighted.

Chapter 3 explains the research questions or objectives addressed in this thesis work.

Chapter 4 presents an analysis of a compensatory absorption refrigeration system driven from waste heat of a boat engine for deployment on-board a small fishing boat. In this section, thermal and economic analysis of installing such a system is discussed in detail.

Chapter 5 discusses conceptualization of various cascade refrigeration systems based on the refrigeration demands in surimi processing and storage plant. In this study, three CRSs were emphasized and their performance is compared with a conventional system in terms of the energetic performance. Further, a sensitivity analysis was also carried on these system in order to check their applicability in other seafood processing plants having cooling demands at similar temperature but in different ratios. Additionally, a comparison of component wise exergy destruction and exergy efficiency between the multi-stage system and one of the proposed CRS is also presented in this chapter.

Chapter 6 presents a comparative study between CO_2-NH_3 CRS and one conventional system working with R22, R404A and NH_3 based on energetic, environmental and economic analysis. The study also discusses further improvement of energetic performance based on the use of an internal heat exchanger along with a gascooler with LTC integrated with the CRS.

Chapter 7 illustrates comparative investigation of low GWP pure fluids as potential refrigerant options for a CRS in seafood applications. This study investigates various environmentally friendly refrigerants for seafood processing and storage application having multi-target temperature in a warm climate. Fourteen potential low GWP pure fluids and one popularly used blend are identified for the study as refrigerants based on literature survey. Additionally, it also presents an overall assessment of benefits of employing natural refrigerants in seafood cold chains in India. This section mainly gives an overview of three different scenarios of refrigeration technology and refrigerant adoption at various stages of seafood cold chains in India. This includes cooling demands at stages during fish harvesting, transport of fresh catch, sorting and processing, and long-time storage. The results highlight the potential benefits from the use of natural refrigerants in seafood sector. Further, the various barriers related to adoption, technology, economy and policy are highlighted.

Chapter 8 summarizes the work presented in this thesis and outlines areas of future research on the topic.

CHAPTER 2 Literature Survey and Site Visits

2.1 Abstract

This chapter attempts a comprehensive review of the state-of-the-art in refrigeration technologies used in seafood cold chain globally with specific focus on India. The literature survey brings out the best practises and technologies used globally as well as their adoptability in Indian surimi seafood cold chain. Based on the literature survey, critical areas are identified where improvements in refrigeration systems, refrigerant usages and other methods of cooling or maintaining a low temperature for the fish. Distinct areas are identified throughout the complete cold chain with a special emphasis on preservation in fishing boats or trawlers, temperature control within processing plants, quick freezing of product at very low temperature and long-time storage in cold storages. Considering the high energy consumption in maintaining low temperatures at various stages and its consequent impact on the environment, the study is deemed important from various aspects of the SDGs. The literature is reviewed across various domains such as food loss prevention, quality assurance, system and component modifications for energy efficiency, and adoption of improved technologies for overall environmental and economic benefits.

The survey is carried out covering peer reviewed and reputed publications spanning 1990 to 2022. Journals and relevant conference proceedings are chosen based on their coverage of the topics related to seafood cold chains, refrigeration, and processes in fisheries sector. For this study work first hand data is also gathered from multiple site visits that also supplemented the low number of available published literature on India specific studies in international journals. The site visits and survey essentially covered state of the art in fishing boats, landing sites, pre-processing and processing plants for surimi and shrimp, cold storages, RRM processing plants. The sites visited are majority from the western coastal areas of India such as Mumbai (Maharashtra), Ratnagiri and Mangalore (Karnataka) and Kochi (Kerala), while Kolkata (West Bengal) comprised the eastern coastal area. Furthermore, many formal and informal engagements were made with various stakeholders of the cold chain like fishermen, landing site mangers, factory owners, etc. Enriching discussions and interactions at several national and international workshops and symposia assed some exclusive elements of knowledge and wisdom of this body of work. The significant research gaps are identified in terms of refrigeration systems for on-board cooling, temperature control in

processing plants, quick freezing and cold storages. Scope of various environmentally friendly refrigerants are also highlighted along with their major advantages and disadvantages.

2.2 Review Objectives and Methodology

The objective of this chapter is to analyse the conventional refrigeration or cooling methods along with refrigeration technology and refrigerants utilized for the following three upstream stages of the surimi cold chain: on-board cooling in fishing vessels/ boats, surimi processing plants and cold storages. The specific objectives of this chapter are to review and analyse cooling or refrigeration systems or methods with a target of identifying implementable system in India for an energy efficient and environmentally friendly surimi cold chain. Firstly, literature related to various methods of preserving fish just after the harvest in sea are reviewed. Secondly, refrigeration systems and refrigerants which are suitable for industrial refrigeration applications in surimi processing industry in high ambient temperature are considered. Finally, in the analysis, recent technological advancement in the refrigeration systems especially CRS are tracked, due to the large temperature difference encountered in the application. Use of low GWP refrigerants are prioritized in the screening process.

2.3 Fish Harvesting and On-board Preservation

The keywords used for the search in this topic include “fishing boats”, “fishing trawlers”, “fishing vessels”, “refrigeration”, “ice”, “freezing”, “chilling” etc. With a primary search with these keywords a few more keywords appeared in the related articles such as “refrigerated sea water (RSW)”, “cooling”, “refrigeration system” etc. which are also added to deploy a composite search string leading to a total count of 310 articles. To maintain the breadth of the study, papers were also screened individually to identify any grey literature. In order to identify the studies about India specific applications on refrigeration on-board and other relevant details, a few more reports and articles were included in the review process that are not Scopus indexed. The net number of articles after all these inclusions and exclusion was 246. The methodology followed is represented in Figure 2.1.

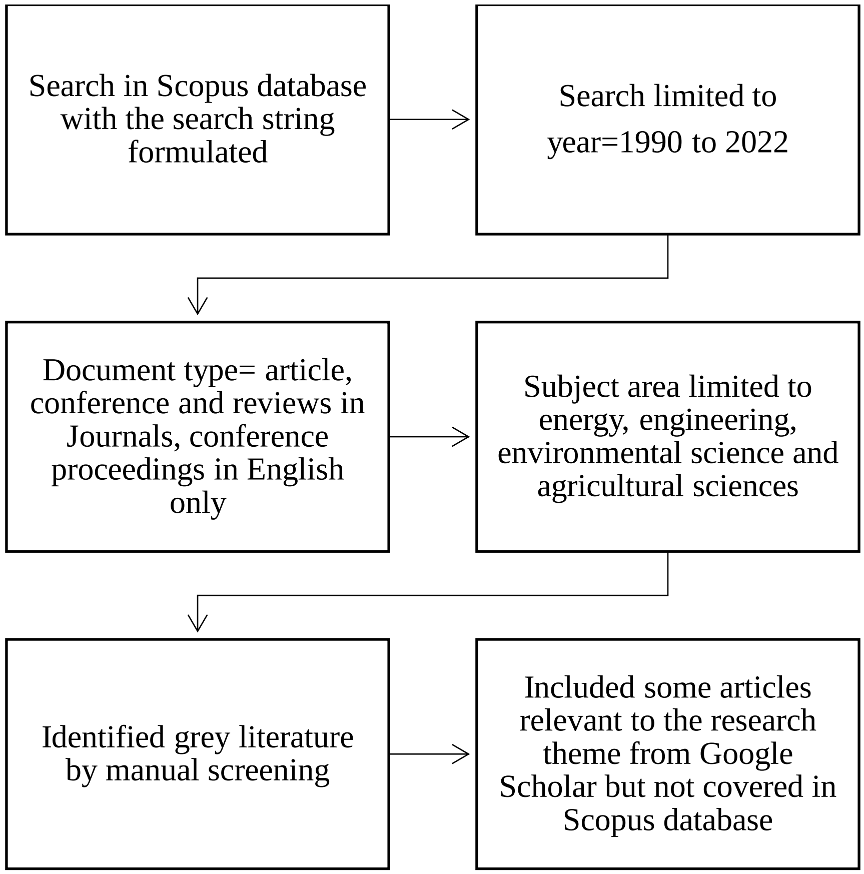


Figure 2.1: Overview of methodology adopted for literature review

The research trend is first captured through a year-wise analysis of the published articles focusing on the various preservation methods of fish on-board and is presented in Figure 2.2. A steady increase is observed in the number of publications, demonstrating a growing interest among academics in the same. The rising number of articles may reflect growing community interest in the preservation methods of fish, public awareness about quality and safety, energy and environmental concerns (FAO, 2022). Despite the increase in research trends over the recent years, the overall number of articles published yearly is relatively low.

In order to know the location and origin of the published studies, affiliations of the authors are tracked. The results of the analysis for the country wise distribution of articles are presented in Figure 2.3. The analysis reveals that majority authors are from UK (12.2%), followed by Norway (11.5%), USA (11.2%), China (10%), Canada (9.7%) and Spain (9.7%). Further, the study locale of vast majority of the articles are for cold climates zones. Despite being a leading producer and exporter of seafood, only 3 publications focused on India.

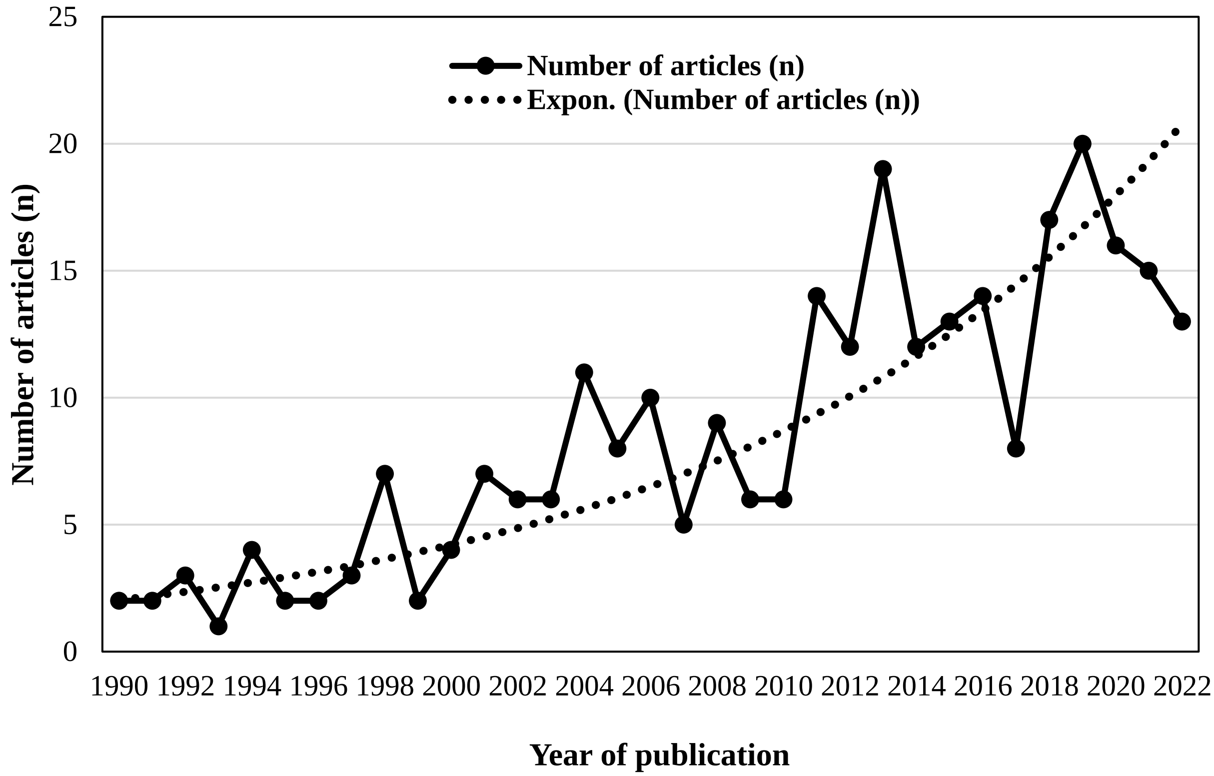


Figure 2.2: Research trend on on-board refrigeration in fishing boats

The research institutes contributing to the topic are Shanghai Jiao Tong University, China (n=17), Norwegian University of Science and Technology, Norway (n=11), Technical University of Denmark (n=11) and SINTEF Ocean, Norway (n=9). R. Z. Wang (n=13) co-authored with L.W. Wang (n=9) and J.Y. Wu (n= 8) from Institute of Refrigeration and Cryogenic, Shanghai Jiao Tong University China is the top author contributed on the topic. All their research articles are published in between 2004 to 2008 and mainly focused on heat pipe adsorption refrigeration on-board to produce ice by utilizing the boat engine waste heat. Their research was mainly published in five international journals namely International Journal of Refrigeration, Energy Conversion and Management, Renewable Energy, Energy and Applied Thermal Engineering. K.N. Widell (n=7), SINTEF Ocean and A. Hafner (n=5), Department of Energy and Process Engineering, NTNU Norway are the other top researchers who majorly contributed to this research topic. They mainly focused on active refrigeration on-board using natural refrigerants especially carbon dioxide (CO_2 designated as R744).

A detailed study was planned following these results from the preliminary literature review. This will focus on important issues discussed in various preservation methods, active refrigeration technologies, refrigeration system types and energy consumption, their environmental impacts and possible mitigations etc.

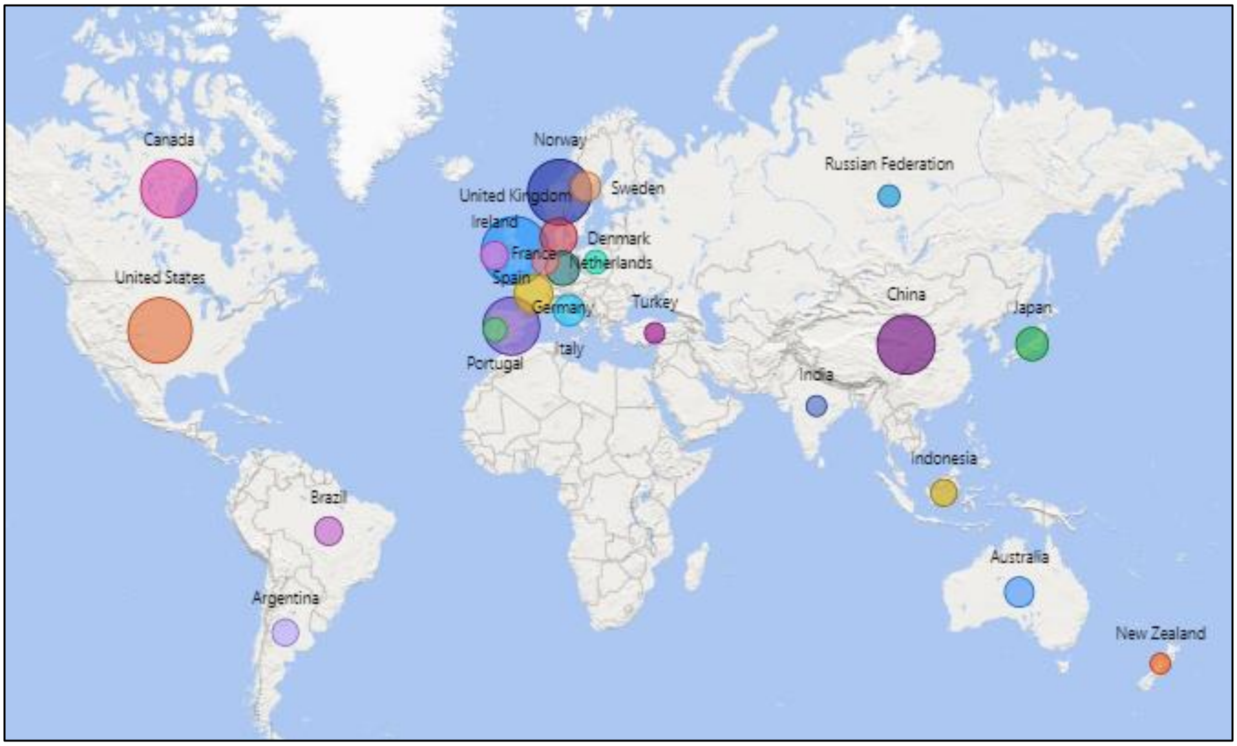


Figure 2.3: Country wise distribution of published research article

2.3.1 Preservation methods

Fishing practice in seafood supply chains are found to vary depending on factors such as targeted variety of catch, type of species, final product, fishing region, type of boat and processing etc. Subject to the end product which may be chilled fresh, fillet or minced frozen, smoke dried etc., pre-processing and preservation during the catch also differs. The catch is generally preserved at low temperature just after harvest which is further segregated based on the variety of fish and frozen seafood which includes whole fish and size, fillets or minced meat etc.

Hedges (2002) discussed three types of supply chain for frozen seafood namely sea based, land based and hybrid supply chain for processed seafood. In sea-based supply chain all the processes are performed at sea including the freezing of the same. In a typical land-based supply chain, after sorting of the catch it is stored within crushed ice within insulated compartments of the boats followed primary processing like sorting, heading and gutting of fish at land-based pre-processing and processing at land-based processing plant and quick deep freezing. Hybrid supply chain includes gutting and sometimes heading too at sea-based plant, freezing at intermediate temperature at sea. The frozen block packed in cartons are transported to land based plant where the frozen blocks are thawed, skinned, minced/filleted and re-frozen. In India, mostly land based supply chain is practised in which the catch is preserved in crushed ice carried from the land based ice plant after initial sorting and all other processes are performed in land based plants. Denham *et al.* (2015) reviewed seafood cold chain management including various stages like capture and aquaculture, transport, processing, storage, retail and identified various redundant handling and energy usages in cold chain.

On-board preservation is important because once the freshness and nutritional value of fish is lost, it cannot be recovered during the processing stages. Margeirsson *et al.* (2010) emphasized the importance of minimizing the holding time between harvests and chilling the catch to maintain the product quality. After the pre-processing, it should be precooled down to a temperature as close to the intended storage temperature as possible. Gang *et al.* (2014) investigated the effect of ambient temperature and holding time between harvest and chilling the catch on the quality of fish. They found that higher ambient temperature and longer holding time without icing results in low quality and lessens the shelf life of the fish for example two hours of holding before chilling can cause loss of about 22% (3 days) shelf life compared to immediately chilled product. After the precooling of the catch it should be stored in the insulated holds sandwiched in layers of evenly spaced crushed ice. There are many chilling techniques in practice all over the world like use of ice (block ice, crush ice, flake ice, slurry ice), Chilled Sea Water (CSW), Refrigerated Sea Water (RSW), super chilling etc. Water used for production of ice must be contamination free as it comes in direct contact with fish. Generally, chlorinated water is recommended for abatement of bacterial contamination. Depending upon the type of catch, its temperature compatibility, intended shelf life, fishing trip duration, boat size, ambient conditions etc., and the chilling methods to be chosen. These methods have specific characteristics and are discussed in following subsections.

a) Ice chilling

Use of ice to provide cooling and preservation of fish is in practice before the invention of mechanical refrigeration systems (Wang and Wang, 2005). Melting of ice at 0 °C was an appropriate handling practices since long as it is a cheap and reasonably efficient method of cooling. Graham *et al.* (1993) described various aspects of using ice for the chilling and storage of the catch. They reviewed the production method and properties of ice and storage equipment from both technical and economical point of view. Huss *et al.* (1998) reviewed fish handling and chilling methods in practise to maintain the low temperature and effect of melting of crushed ice and its impurities on chilled fish. Shawyer *et al.* (2003) described advantages and drawbacks of various types of ice used for the preservation of fish and discussed methods to produce ice and handling of ice and fish. They also described the physical changes that occurs when fish is exposed to heat and how the icing delays the processes. Design of the insulated fish storage containers and material used for insulation are also discussed.

Generally ice is used in a ratio of 2:3 to 1:1 of volume catch and it is replenished at intervals to maintain the low temperature (Jeyasekaran *et al.*, 2006; Shawyer *et al.*, 2003). In order to keep their catch fresh for as long as possible, traditional fishing boats normally make use of crushed ice. Block ice or tube ice is hard, and has rough surfaces which can produce bruise on fish, which can lead to downgrading their commercial value. Melinder and Ignatowicz (2015) suggested that for instant and effective chilling slurry ice is better due to high energy density associated with the latent heat and large surface area per unit mass which enables rapid removal of heat. Baheramsyah *et al.* (2017) reported various types of slurry ice producing machines. Kauffeld and Gund (2019) described thermal and rheological properties of slurry ice with its characteristics, energy efficient production techniques, and its possible future developments. It has been shown that slurry ice performs better than ice cubes or crushed ice at maintaining fish quality and hence it is increasingly being used. Because there is no air pocket formed between the fish and the ice slurry, the wetting of fish is better and cooling is quicker because of the large contact surface area. In addition, ice slurry can reduce the temperatures by up to 0 °C so it can chill the fish faster and maintain a lower temperature. Use of ice slurry in food industry in general and various systems suitable for ice slurry production on-board in boats are reviewed by (Lyu *et al.*, 2022).

b) Superchilling

Seawater contain 3-4% salt and hence the freezing point of seawater is around −2 °C, which can help in rapid cooling of fish (Kolbe, 1990). Superchilling or deep-chilling or partial freezing are the techniques in which temperature is kept between chilling and freezing. It has many advantages over the ice chilling method such as maintaining high food freshness, retaining high food quality and suppressing growth of microbes in food. The surface temperature of seafood deeps below 0 °C that absorbs the internal heat of the fish faster. In superchilling, low temperature up to −3 °C is achieved on the surface but core temperature of the fish is around 0 °C which results in 5 to 30%

of water inside the product converted into ice (Beaufort *et al.*, 2009). Kaale *et al.* (2011) presented a review on superchilling technologies and compared performance of the same with mechanical freezers, cryogenic freezers and impingement freezers towards maximizing product quality, operating flexibility and return on investment while minimizing the waste. Biohaz *et al.* (2021); Hoang *et al.* (2016) and Stevik and Claussen (2011) reviewed various equipment and methods of superchilling and compared the practical aspects and advantages of using superchilling method over the conventional ice chilling method. Bantle *et al.* (2016) pointed out the challenges with superchilling such as growth of ice crystals and resulting damage of cells and the texture. Eliasson *et al.*, (2019) reported an experimental study which revealed that by superchilling of whole gutted fish on-board a fish trawler, its shelf life can be increased up to 2-3 days. These studies concluded that superchilling showed great potential in preservation of seafood.

By using CSW/RSW fish can be stored at temperature near freezing point of seawater and thereby enhancing its shelf life. Two options to use the seawater are a) RSW, using mechanical refrigeration system on board and b) CSW, using ice brought from shore to lower the sea water and fish temperature. Chilling of the catch can be carried out in water fluidization using cooled seawater by immersion, sprinkling, and immersion with bubbling through. Chilling of fish using CSW/RSW is advantageous over direct ice such as rapid cooling, lesser bruise and pressure, lower holding temperature, quicker handling of large quantity and extended storage time (Graham *et al.*, 1993). Sometimes ice slurry is also used instead of CSW. Thorsteinsson *et al.* (2003) presented a case study which simulated and analysed a combined RSW/CSW system for the chilling of the catch and operational cost.

Shawyer *et al.* (2003) recommended CSW for small fishing vessels while RSW for large fishing vessels due to the size constraint to carry refrigeration system on board. RSW is being used in fishing trawlers and big vessels in many high-income regions. Some disadvantages of using RSW on product are excessive uptake of salts, loss of protein, problems with anaerobic spoilage and modification of characteristics of fish quality indicators (Tuckey *et al.*, 2012). Knotek *et al.* (2015) discussed developments of a low cost refrigerated seawater system named ‘flow-through environment’, which ensures required oxygen content in water thereby reducing anaerobic spoilage. Bodys *et al.* (2018) presented design and simulations of a compact RSW unit using CO_2 as refrigerant for the warm water of Mediterranean and East-Asian area. The study claimed that the system operation is feasible in warmer climates without the need for an additional compressor unit, thus maintaining the compactness of the system while also maintain the designed cooling capacity.

c) Freezing

Freezing of fish or fully processed seafood is a common technique to extend its shelf life and preserve fish quality, which has long been established and used for storage of seafood (Guo *et al.*, 2014). Quick freezing is the best way to preserving fish in a natural and safe manner for periods of many months or even years (Fellows, 2009). Gonçalves and Blaha (2010) reported that in quick freezing, temperature of the fish or product is reduced from 0 °C to –40 °C within 2 hours using either plate freezer, air blast freezers, cryogenic tunnel freezers, immersion freezers, fluidized-bed freezers etc. Verpe *et al.* (2018) experimentally investigated the freezing time for various systems at different temperatures and reported that reduced refrigeration temperature can lower the time required to freeze the product. The quality of the frozen seafood depends upon three main factors which are handling & pre-freezing, rate of freezing and frozen storage. Shelf life of the product increases with decrease in storage temperature, the storage time can be drastically increased for storage around or below –80 °C, but is high energy consuming and expensive (Burgaard and Jørgensen, 2011; Indergård *et al.*, 2014).

The cold water fish species are more prone to deterioration when they are subjected to ambient temperature after catch than tropical fish species (Hunt and Park, 2014). Tolstorebrov *et al.* (2014) reported the phase transition in muscle and in oil of a few commercial fish species from the North-Atlantic region and found out that the best stability of a fish quality can be achieved at temperature as low as –86 °C. Lee *et al.* (2016) reported a study on effects of pre-freezing treatments on the quality of Alaska Pollock fillets subjected to freezing/ thawing and showed that postharvest processing and refrigeration conditions and subsequent freezing affect the quality of frozen fish fillets. Söylemez *et al.*, (2022) presented various refrigeration system development and status of refrigeration systems on-board fishing boats. They also reviewed a large number of refrigeration techniques which are suitable to be deployed on fishing vessels.

2.3.2 Refrigeration systems on-board

Use of on-board refrigeration system on bait boats employed in recreational fishing and commercial vessels has been in vogue since the 1960s. The literature mainly discusses chilling of tuna seiners in cold countries (Wang and Wang, 2005). These refrigeration systems consume a significant amount of fuel and contribute to about 30-50% of operating expenses. Moreover, the use of fossil fuel is linked to massive level of GHG emissions. Bodys *et al.* (2018) presented merits of using on-board refrigeration systems such as enhanced product quality on landing and the increase in catch quantity due to longer duration of fishing. They also highlighted the limitation or challenges of refrigeration system operation when it comes to high ambient temperature conditions such as off the southern Mediterranean coast. The study also reported modifications required in a R744 refrigeration systems for such high ambient temperature conditions.

Hafner *et al.* (2019) presented a survey on various refrigerants, regulations, substitute refrigerants and example of refrigeration systems applicable for various type of vessels according to the size and application. Söylemez *et al.* (2022) discussed various methods of achieving cooling on-board for preservation of fish. They also reported developments on on-board refrigeration system and their energy efficiency enhancement by integrating various components like ejectors, internal heat exchangers, heat recovery and thermal storage options etc. The study concluded that use of natural refrigerants like R744 is helpful in lowering the GHG emissions and improving system energy efficiency.

However, integration of refrigeration systems is not suitable for the small fishing boats or vessels which have a capacity below 100 tonnes storage due to challenges of power supply to the compressor-icemaker unit on-board from lower horsepower of boat engine (Wang and Wang, 2005). There are various articles on possible utilization of waste heat of the diesel engines to run sorption-based refrigeration systems. The diesel engines have a maximum efficiency of about 40% and the rest of the waste energy is rejected to the atmosphere in the form of hot exhaust gases. As reported in (Fernández-Seara *et al.*, 1998; Wang *et al.*, 2006; Xu *et al.*, 2017), a typical distribution of energy in a typical boat diesel engines is shown as a Sankey diagram in Figure 2.4.

Sorption based techniques, absorption and adsorption refrigeration, are capable of utilizing heat as input to produce the refrigeration effect. Wang and Wang (2005) discussed feasibility of on-board absorption and adsorption refrigeration systems powered by engine exhaust waste heat. Shu *et al.* (2013) reviewed various waste heat recovery technologies from engine, applicable to on-board ships. They found that the exhaust heat temperature of a typical boat diesel engine is in the range 250-500 °C for exhaust gases, 130-150 °C for scavenge air and 70-120 °C for engine jacket cooling fluid, which are available to power the vapour absorption refrigeration (VAR) systems.

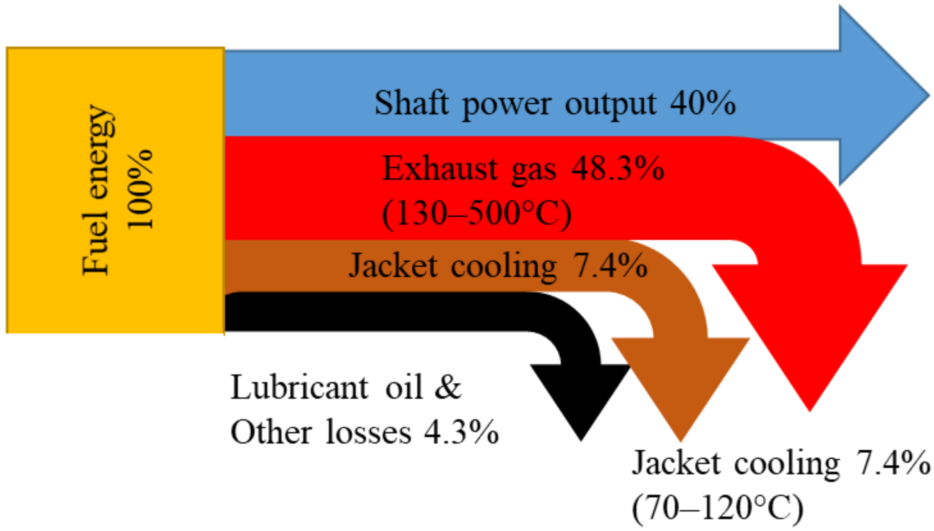


Figure 2.4: Energy balance of a boat diesel engine

Fernández-Seara *et al.* (1998) analysed a gas to thermal fluid heat recovery system to power an ammonia VAR system for fishing trawlers. They proposed the design of an economizer and generator heat exchanger to recover the waste heat from exhaust gas and concluded that the recovered waste heat is adequate. Táboas *et al.* (2014) analysed the use of waste heat energy of jacket cooling fluid in diesel engines of fishing ships to drive a VAR system. They also analysed the performance of VAR system using three different working fluids $NH_3/(LiNO_3+H_2O)$, $NH_3/LiNO_3$ and NH_3/H_2O. Performance of the VAR system depends on the sink temperature, where absorber and condenser heat are rejected. As the average seawater temperature remains within 20-30 °C most of the time in the tropical regions, a stainless-steel pipeline running along the starboard immersed in sea can serve as a heat sink.

As reported in the literature, various theoretical as well as experimental studies have successfully established the heat recovery potential and its utilization to drive adsorption, absorption or heat pipes-based refrigeration systems; a survey is listed in Table 2.1.

Table 2.1: Literature studies highlighting utilization of waste heat in refrigeration on-board

Application field	Technology	Refrigerants	Type of study	Reference
Heat recovery	Absorption	NH_3-H_2O	Theoretical	Fernández-Seara *et al.* (1998)

Space cooling	Absorption	NH_3-H_2O	Theoretical	Ouadha and El-Gotni (2013)
Space cooling	Absorption	H_2O-LiBr	Theoretical	Cao *et al.* (2015)
Space heating and cooling	Absorption	H_2O-LiBr	Theoretical	Ezgi (2014)
Ice making	Adsorption	Methanol and activate carbon	Experimental	Wang *et al.* (2005)
Ice making	Adsorption	NH_3 and compound adsorbent	Experimental	Shi *et al.* (2016)
Ice making	Adsorption	NH_3 and compound adsorbent with $CaCl_2$	Experimental	Wang *et al.* (2004)
Refrigeration	Absorption	NA	Theoretical	Sapienza *et al.* (2016)
Refrigeration	Adsorption and absorption	NH_3 and compound adsorbent, NH_3-H_2O	Experimental	Lu and Wang (2016)
Refrigeration	Absorption	NH_3-H_2O	Theoretical	Saini *et al.* (2019)

Srikhirin *et al.* (2001) reviewed various vapour absorption refrigeration (VAR) systems and also processes in an absorption system. VAR system employ a binary solution of refrigerant and absorbent as working fluid such as NH_3-H_2O, LiBr-H_2O. The input energy in a VAR system is heat (low grade energy) so it is also known a thermal energy driven system and is only preferred when waste heat or solar thermal energy is available. Schematic diagram and major components of a VAR systems are shown in Figure 2.5.

In VAR system refrigerant vapour at low temperature and low pressure from the evaporator is absorbed by weak liquid absorbent solution in absorber. Rich solution of absorbent and refrigerant is pumped to generator where heat is added from an external source and refrigerant converts into vapour. High pressure refrigerant vapour passes through condenser and get condensed by rejecting heat to the ambient. On the other hand, absorbent from generator returns back to absorber. After rejecting heat in condenser, high pressure the refrigerant expands to low pressure and enters into evaporator where it absorbs heat from the refrigerated space.

Garimella *et al.* (2011) conceptualized and analysed a waste heat driven LiBr-H_2O absorption/subcritical CO_2 vapour compression cascade refrigeration system to provide cooling and heating at various temperatures. The proposed system claimed to be 31% more energy efficient compared to a two stage VCR system. Zhu and Jiang (2012) proposed a refrigeration system integrated with ejector cooling cycle driven by waste heat from condenser which increase the cooling capacity and performance of the system.

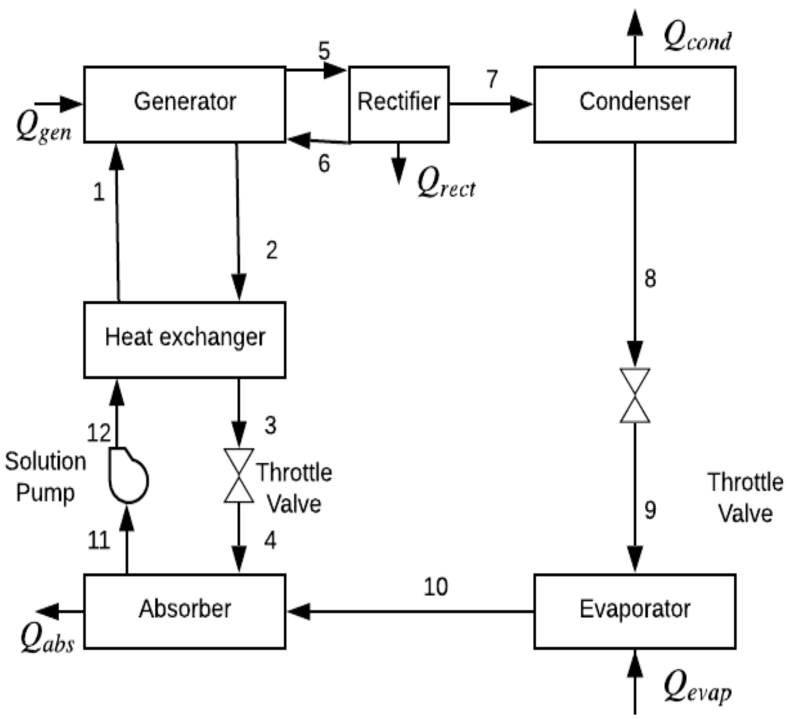


Figure 2.5: Schematic diagram of vapour absorption refrigeration system

As discussed above, the sorption-based refrigeration technologies have applicability which utilizes waste engine heat for space conditioning, ice making as well for refrigeration application. These technologies may also be suitable for small fishing boats to provide refrigeration for the catch during long fishing periods. In many low income countries like China, India, Indonesia, Thailand etc. over 85% of their total fishing fleet are small wooden fishing boats, which are not suitable for installing a conventional refrigeration system (Budiyanto *et al.*, 2022; Desai *et al.*, 2016; Nakhawa *et al.*, 2018; Xu *et al.*, 2017). General design of a small fishing boat can be found in Budiyanto *et al.* (2022) reproduced in Figure 2.6. These motorized boats are powered by diesel engines of capacity 40-175 hp. Amin et al. (2013) discussed the type of boats in India, their development over time, modification, mechanisation and also progress in various countries and organization in the development of small boats, deep sea fishing boats, trawlers etc. over time.

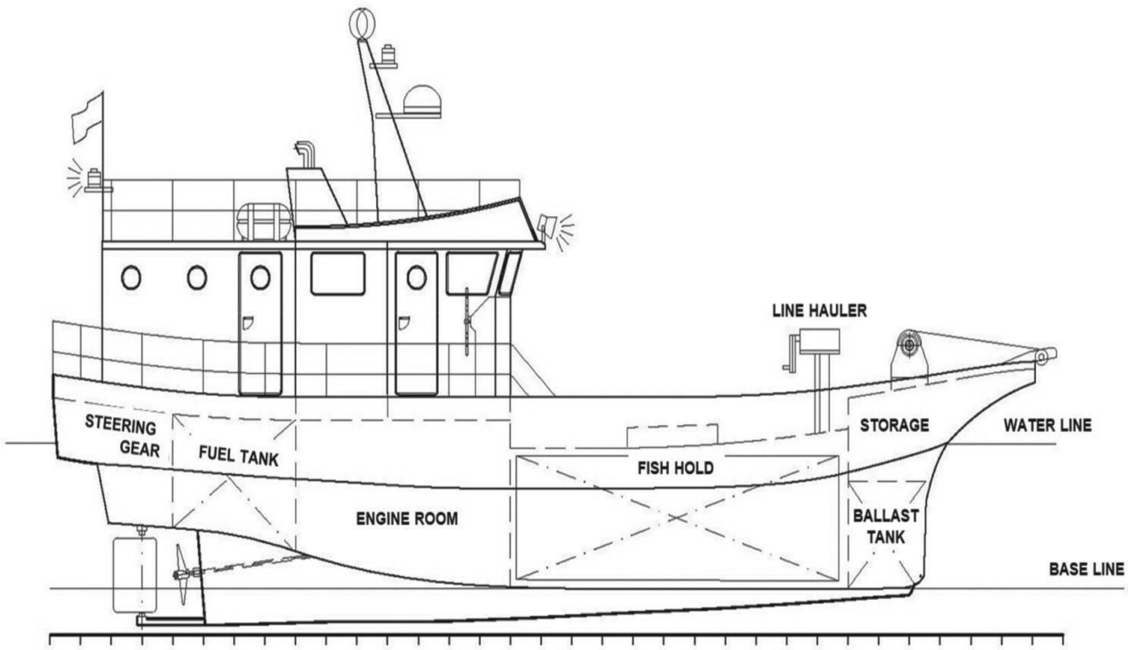


Figure 2.6: Illustrative presentation of fishing vessels

A study by Raghuram and Asopa (2008) pointed out the absence of requisite cold chain infrastructure in the supply side in Indian seafood industry as one of the main barriers for growth. Desai *et al.* (2016) reported the design and technical specification of steel *purse seiners* operating along Ratnagiri coast of Maharashtra. During site visits to such locations, design and technical specification were collected which are fairly similar to that reported by Desai *et al.* (2016). These fishing vessels/ boats generally have 6-10 compartments in deck serving as 'fish holds' with square cross section and of about 2 to 3 m depth with a square lid on top.

From the literature review and site visits it is found that appropriate temperature management should be given prime importance in the entire seafood cold chain starting from the harvesting in sea to landing. A variety of reasons, such as longer than planned fishing duration, improper estimate of ice quantity, excessive heat ingress, impaired insulation and accumulation of water inside insulated compartments, lead to temperature of the catch to rise during storage in turn resulting in fish quality deterioration. According to an estimate by TNAU Agritech (2014), about 20% of the fish harvested is lost due to improper storage. A potential solution to maintain low temperature is to improve insulation on the walls of the iced compartments and provide a low capacity refrigeration system to compensate for the heat ingress.

Thus, there is a need to explore compact and low power refrigeration units suitable for use in small fishing vessels in India. Considering the small size and lack of mechanization of Indian fishing boats further exploration of the estimation of cooling requirements on-board and waste heat driven compensatory refrigeration technologies is needed.

2.4 Surimi Processing and Cold Storage

In this section, various refrigeration demands in processing of surimi and subsequent long-term storage in cold storage are reviewed followed by an examination of refrigeration systems and refrigerants used. The review is extended in order to capture the potential refrigeration technologies and refrigerants which are also compliant with contemporary environmental regulations and that demonstrate superior energetic performance. This survey aims to address the following: a) to formulate a structured understanding of the refrigeration demands in various stages of surimi processing and long-term storage, b) to identify commonly used refrigerants and refrigeration systems in seafood processing and other industries with similar refrigeration demands, c) to explore state of the art in environmentally friendly refrigerants accompanied by improved efficiency of such systems for ambient conditions in India.

The keywords employed in the initial search are “refrigeration system”, “surimi”, “seafood”, “shrimp”, “prawn”, “freezing”, “refrigeration”, “chilling”, “cooling”, “ice”, “cold storage” etc.; “shrimp” and “prawn” were used as keywords due to their similarity with surimi in refrigeration demands during processing. A keyword search string was formulated incorporating all these keywords and a Scopus search result yielded 213 articles. It was found that majority of the articles are related to study of the changes in properties of the seafood, their biochemistry, microbiology, toxicology etc. during the low temperature refrigeration.

Thus, in order to screen these articles and align the search focusing to refrigeration systems and refrigeration demands, subject area of the search was limited to energy, engineering and environmental science only, yielding 53 items till November 2022. To maintain the breadth of the review, the articles were screened manually to identify grey literature. Incorporating studies on India specific work on refrigeration in processing plants and cold storages, a few reports and articles outside the search, the number of articles expanded to 57. Data recorded during plant visits also supplemented the study particularly on refrigeration practices in India. Based on the objectives, the literature reviewed is classified into three parts namely refrigeration demands, refrigeration systems and refrigerants as presented below.

2.4.1 Refrigeration demands in surimi processing and storage

In India, pre-processing such as sorting, heading/gutting are generally carried out either at fish landing sites or at processing plant. Before feeding to the plant, headed and gutted fish are washed with chlorinated chilled water at 4-8 °C (Dasgupta *et al.*, 2019). Figure 2.7 depicts the typical sequence of processes carried out in a semi-automated surimi processing plant as the product temperature is maintained.

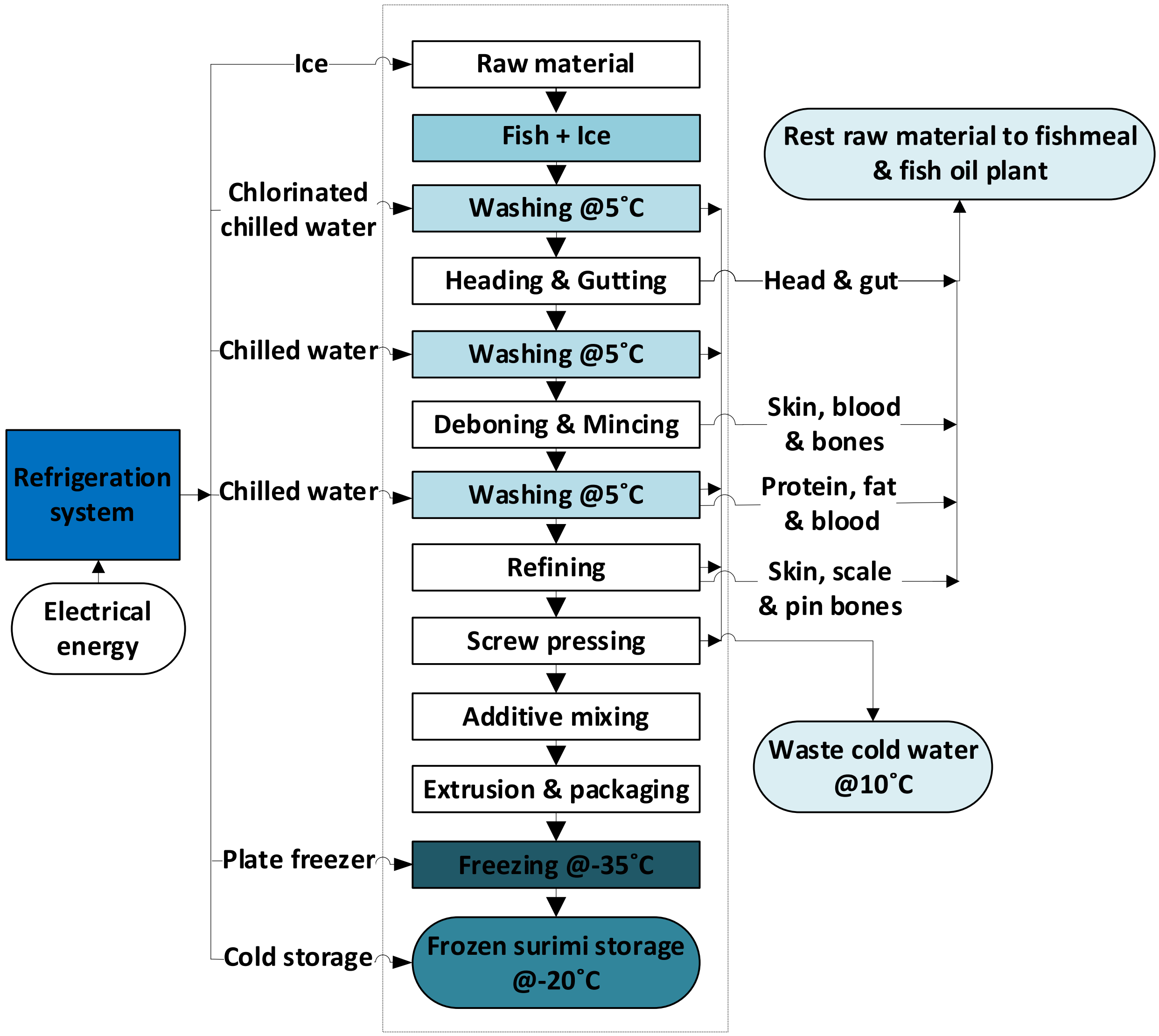


Figure 2.7: Surimi production process flow chart

For the subsequent process of mincing and deboning, generally a meat separator removes skin, bones and other impurities by pressing the fish. Mincing of fish is followed by a sequence of washing with chilled water to remove water soluble proteins, fat, nitrogenous compounds, blood, pigments, odorous compounds and other impurities (Hall and Ahmad, 1997). Removal of water-soluble proteins from mince concentrates the remaining myofibrillar protein leading to enhancement of the gelling ability of surimi. A large amount of RRM is produced during the processing. The pre-processing of fish for surimi production also generates head and viscera termed as RRM usually discarded as waste or processed by specific secondary industries. Fish processing industries generate substantial RRM which has the potential yield value added products such as fishmeal and fish oil, gelatine etc. (Hilmarsdóttir *et al.*, 2022).

Depending upon input species, two to four washing cycles are required to ensure the right whiteness and gelling ability which is manually adjudicated. The large quantity of chilled water circulation also contributes to maintaining the composition temperature low as well as work space temperature comfortable for operators. Various kinds of processing of seafood, its temperature requirement during processing, use of chilled water, type of freezing after processing and for long time storage are discussed by various authors (Lin and Park, 1997; Park *et al.*, 1997).

After washing cycles, surimi is dewatered in rotary drum. Lee *et al.*, (2016) reported the effects of various pre-treatment methods and effect of blending of refined & dewatered surimi is with cryoprotectants, and various other additives to maintain required functional properties of it. Thereafter, surimi is extruded in the form of standard slabs. Based on the observations at the plants, it is found that surimi blocks are cut into 10 kg packs of size 360 mm x 55 mm in cross-section and 590 mm in length and sealed in plastic bags. The surimi in the form of blocks still contains about 82-84% water. Total yield of surimi blocks from the raw material (headed and gutted fish) is about 22-25% (Kim and Park, 2007).

The packed surimi slabs are rapidly frozen using plate freezers to about –35 °C to –40 °C within 2 hours (Park, 2015) with block core temperature reaching –25 ºC. The process has three distinct phases depending on removal of sensible and latent heat from the product - pre-chilling from 10 ºC to –1.5 ºC, removal of latent heat of fusion from –1.5 ºC to about –7 ºC, and from –7 ºC to final product temperature of –18 ºC to –30 ºC (Park, 2015). The freezing rate plays an important role in attaining better product quality. Higher the rate, lower is the size of ice crystal formation and better is the quality (Xie *et al.*, 2020). Finally, the surimi blocks are packed in cardboard boxes and stored for long time at −18 ºC to −30 ºC.

Quality of the surimi mixed with cryoprotectants and stored at low temperature also deteriorates with time but the deterioration rate can be controlled by maintaining the cold storage temperature (Scott *et al.*, 1988). In an experimental study on the surimi produced from threadfin bream, bigeye snapper, lizardfish and croaker for 6 months of storage at storage temperature −18 ºC, Benjakul *et al.* (2005) showed the deterioration in functional properties of surimi. Kaba (2006) analyzed the surimi quality produced from anchovy and concluded that the reduction in quality of surimi was negligible during a 5 month storage at −29 ºC. Dey and Dora (2011) conducted an experimental study on Croaker surimi with chitosan cryoprotectant and found that quality of surimi is good till 6 months of storage if stored at −20 ºC. These studies show that surimi mixed with suitable cryoprotectants can be stored for even over 6 months at a temperature lower than −20 ºC. The frozen surimi is transported according to the demand in market; it is essentially an export market for the Indian surimi. Demand and price fluctuate in the market during a year, and raw material availability also fluctuates, ultimately affecting the time of storage and energy consumption by the refrigeration systems.

In order to improve the quality of frozen seafood during the storage and transportation stages various studies explored different ways of packaging during the processing. Bono *et al.* (2016) studied the combined effect of freezing and modified atmospheric packaging (MAP) on shrimp and analysed the quality characteristics during frozen storage. Dehghani *et al.* (2018) reviewed the use of edible films and coatings in seafood during storage, its effect on extending the shelf life of product. Jin *et al.* (2018) investigated the effect of deep-freeze temperature on storage and quality of shrimp. Wang *et al.* (2019) studied the influence of heating rate on surimi texture before packing. In this study microwave assisted pasteurization systems (MAPS) were used to evaluate the effect of various heating rates on textural properties; it was observed that surimi that pasteurized in MAPS has much better texture than that pasteurized in traditional thermal process as water bath. Rusanova *et al.* (2022) studied various effects of using different packaging methods on the nutrients on deep water rose shrimps.

2.4.2 Refrigeration systems

Since the inception of refrigeration technologies, food processing industry heavily relies on VCR systems for food preservation and processing (Tassou *et al.*, 2010). VCR systems dominated in various existing and emerging refrigeration technologies like absorption, adsorption refrigeration (Gado *et al.*, 2021; Li *et al.*, 2004; Wang *et al.*, 2003), ejector refrigeration system (Chunnanond and Aphornratana, 2004; Yu *et al.*, 2006), air cycle refrigeration (Kikuchi *et al.*, 2005; Williamson and Bansal, 2003), thermoelectric refrigeration (Min and Rowe, 2006; Riffat and Ma, 2003, 2004; Söylemez *et al.*, 2018), thermoacoustic refrigeration (Gardner and Swift, 2003), magnetic refrigeration (Gschneidner Jr. *et al.*, 1999; Saito *et al.*, 2016; Xie *et al.*, 2021), and VCR systems (Ahamed *et al.*, 2011; Barbosa *et al.*, 2012; McLinden *et al.*, 2020). VCR system has priority because of various reasons such as ease of operation, higher performance, availability, technological advancements, maturity and financial viability. Tassou *et al.* (2010) reviewed many states of the art technologies having potential of being adopted in food refrigeration with their related drivers and barriers in the adoption and technological advancement over the years followed by the research and development needs.

Main components of a simplest VCR system are compressor, condenser, expansion device and evaporator as shown in Figure 2.8a. Refrigerant enters the compressor in vapour phase and get compressed up to a high pressure and is passed through condenser where it loses the heat and is liquefied. Liquid refrigerant further flow through expansion valve where it expands from high pressure to low pressure and this pressure drop causes the evaporating temperature of the refrigerant to fall below that of the evaporator. After expansion refrigerant enters the evaporator and absorbs the heat from the products by consuming the latent heat of vaporization.

In order to provide refrigeration at various temperatures with large heat load capacity and large temperature differences between evaporators and condensers, multi-evaporator and multi-stage compressors refrigeration systems are used in seafood applications (Kairouani *et al.*, 2009; Llopis *et al.*, 2010; Nikolaidis and Probert, 1998). In seafood processing applications two or more temperature are generally needed to be maintained for various purposes, like in a surimi processing plant four different temperatures are generally need to be maintained. These are chilled water at 4-8 ºC, ice production at −20 to 0 ºC and deep freezing at −55 ºC to −35 ºC and long-term storages at −35 ºC to −20 ºC are required. These evaporators can be maintained either using individual refrigeration system or by combining the compressor for evaporators having smaller temperature differences known as multi-evaporator systems.

Multi-stage compression system are generally deployed where the pressure ratio or the temperature difference is higher due to low temperature refrigeration requirements and high ambient temperature conditions (Arora, 2012). In seafood freezing and cold storage applications, the evaporating temperature of the refrigeration system must be between –20 ºC to –50 ºC; thus, it is difficult to achieve very low temperature conditions with a single-stage VCR; instead, two-stage or cascade refrigeration systems are mostly deployed to cater the cooling demands matching the high compression ratio with acceptable performance and operational economy for these applications (Messineo, 2012). Thus, while selecting a refrigeration system for a specific application in a VCR system, it is necessary to select the number of compression stages which is further dependent on refrigerant type, specific volume, suction and discharge temperature and pressures. A two-stage refrigeration system uses the same refrigerants for both the high- and low-pressure sides, while a cascade system generally uses different refrigerant for the high- and low-pressure sides. Two stage compressors are commonly used with intermediate cooling (Eini *et al.*, 2016). Intermediate cooling or removal of flash gas in between the multi-stage compression increases the refrigeration effect and also minimizes the work required in compression and thus, performance of the refrigeration systems improves. Intermediate cooling or removal of flash gas in between the two stages reduces the discharge temperature and also reduces heat load on condenser due to the lower discharge temperature (Hundy *et al.*, 2008). Figure 2.8b shows a schematic diagram of a two-stage compression VCR system with intermediate cooling.

Two-stage compression refrigeration systems are the preferred choice for applications that demand evaporator temperature in the range –20 ºC to –60 ºC and are suitable for application such as ice-cream manufacture, food freezing or frozen food storages (Arora, 2012). Widell and Eikevik (2010) discussed multi compressor industrial refrigeration system and various refrigerants used in freezing tunnels for freezing of seafood. The concluded that there is substantial improvement in energy efficiency in freezing tunnels for an optimally operated compressor and fan system using variable frequency drives.

In high ambient temperature conditions and lower temperature of refrigeration, a single or two-stage refrigeration system perform poorly (Bhattacharyya *et al.*, 2005). Roy and Mandal (2020) also discussed that using a single-stage or two-stage refrigeration system in case of large temperature lift results in decrease in cooling effect, and increase in compressor power consumption leading to an enhanced wear and tear of the compressor. Use of multi-stage refrigeration systems are limited by the normal boiling point (NBP) temperature of the refrigerants, below the NBP systems start working in vacuum resulting in increased chance of ingress of air and moisture. At lower temperatures, mass flow rate reduces due to the lower specific volumes of gas which may lead to ineffective cooling of motors at lower temperatures, motors heat up and sometimes results in burnout (Arora, 2012). It also increases the large mass flow rates of refrigerants leading large swept volume rate and thus size of the compressors becomes very large at lower temperatures.

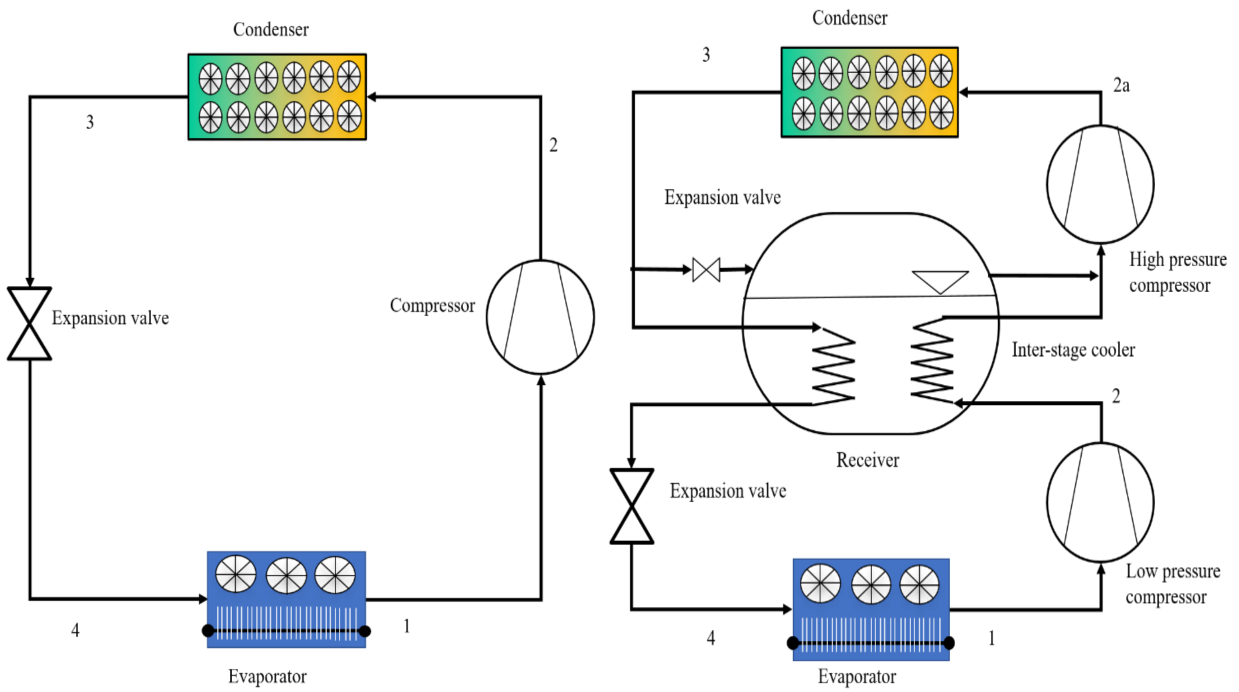


Figure 2.8: Schematic diagrams single-stage VCR system; Two-stage VCR system

Multi-stage systems also face the problem of *oil wandering* as the lubricant dissolves with the pressurized refrigerants and accumulate in high pressure compressor, resulting in high wear and tear of the reciprocating compressor components. This problem is avoided by using multi cylinder compressor in lower and higher-pressure sides. Generally, multi-stage compression systems has higher capital costs than the single stage compression system (Arora, 2012; Hundy *et al.*, 2008). Multi-stage compression systems also suffer from low efficiency due to the use of a single refrigerant at a wide temperature and pressure range. For example, a refrigerant with lower NBP can yield good performance for lower temperature applications but it is not expected to perform equally well in high condenser temperatures or pressures. As per Guldberg number, the NBP of refrigerants are 2/3rd of their critical temperatures on the absolute scale (Bowden, 1954), implying that refrigerants with low NBP may also have critical temperature lower than ambient temperature, resulting in transcritical heat rejection in gas coolers; this makes the heat rejection process more complex and less efficient. Similarly, on the high pressures side, refrigerants with higher NBP can be used but they may not be a good option for the low temperature applications.

A CRS on the other hand overcomes some of the above mentioned challenges and can provide improved performance based on optimal refrigerant selection (Bhattacharyya *et al.*, 2005; Eini *et al.*, 2016; Jankovich and Osman, 2015). Based on the above discussion, the study domain was converged to explore CRS to meet the processing plant and cold storage cooling demands. CRSs are widely used in the industrial refrigeration to achieve low or ultra-low temperature in high ambient temperature with their ability to provide cooling at two or more different temperatures with high performance (Kasi and Cheralathan, 2021). A two-stage CRS consists of two vapour compression refrigeration circuits that can be identified as: low-temperature circuit (LTC) and high-temperature circuit (HTC), which are connected thermally with a heat exchanger known as a cascade heat exchanger (Figure 2.9). For the CRS, selection of refrigerant for a specific circuit is very important considering all the required properties as well as performance and safety issues. The refrigerants can be independently selected for LTC and HTC, thereby taking advantage of distinct thermo-physical properties at different temperature and pressure ranges. A suitable refrigerant pair can provide a larger temperature lift with high system performance.

In order to get more details about the gradual progress in research and application of CRS, a bibliometric analysis was carried out using a keyword search string that included “cascade refrigeration” OR “cascade refrigeration system”. The search was limited to journal articles published in English during 1990-2022. The search yielded 276 articles. The research trend shows almost an exponential rise in number of articles published (Figure 2.10).

This implies growing interest among the academics on the topic. The increase in the number of publications can also be related to enactment of various regulations on synthetic refrigerants like European F-gas regulation, Kigali amendments etc. which introduced restrictions on production and use of HFCs. Figure 2.11 presents country wise distribution of the published articles. The analysis shows that a majority share, about 34% is from China, followed by Iran (13%), India (11.3%), Turkey (8.7%) and Spain (6.9). The top contributing countries, except the first are from warm or high ambient temperature countries, which are seeking viable solution in CRS configuration.

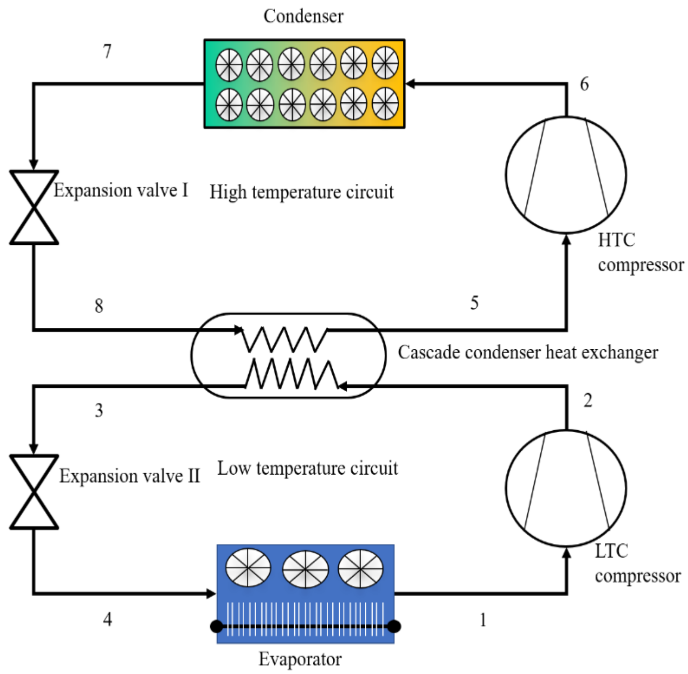


Figure 2.9: Schematic diagram of a cascade refrigeration system

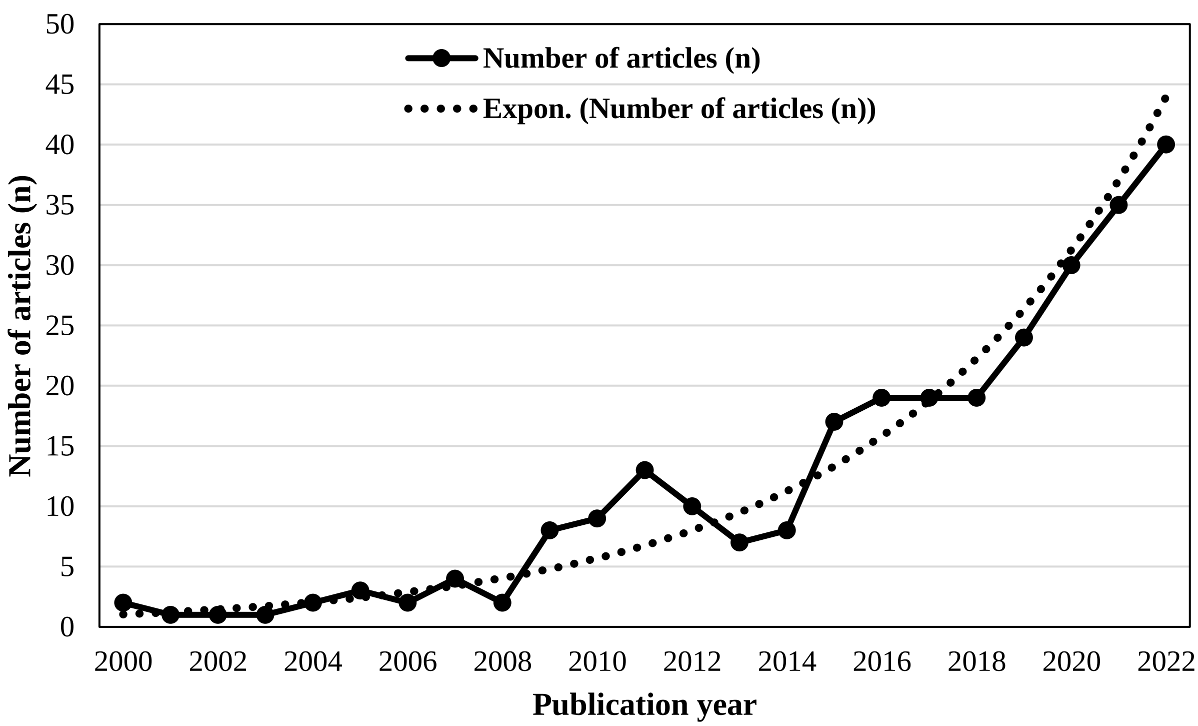


Figure 2.10: Research trend on cascade refrigeration system

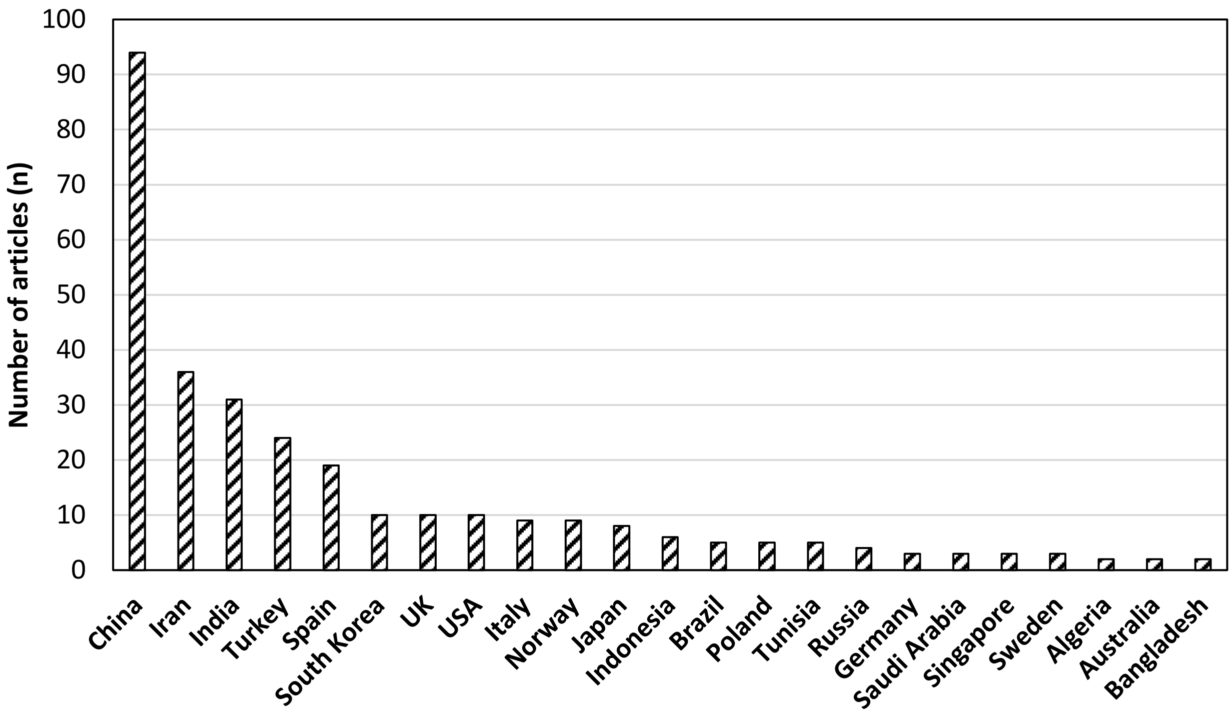


Figure 2.11: Country wise distribution of published research article

Saini *et al.* (2021) reported a study on three CRS configurations and compared the energetic performance with a conventional NH_3 (R717) multistage refrigeration system for various seafood applications. It was concluded that the CRSs demonstrated 11-20% higher coefficient of performance (COP) compared to the multistage refrigeration system. Among various options, the CRS using NH_3 in HTC and CO_2 (R744) in LTC for refrigerant has attracted greater attention in recent years due to their favorable thermophysical and environmentally friendly properties (Eini *et al.*, 2016; Messineo, 2012; Patel *et al.*, 2019). Bingming *et al.* (2009) reported a comparative experimental study between a single-stage NH_3 VCR system and a CO_2-NH_3 CRS with the CRS exhibiting distinct advantage when the evaporator temperature was below −40 ºC.

Dopazo and Fernández-Seara (2011) developed a prototype of CO_2-NH_3 CRS for deep freezing application and conducted experimental evaluation of the performance with variation in operating temperature of evaporator and condenser of the LTC. They also compared the performance of the CO_2-NH_3 CRS with two conventional two-stage NH_3 VCR systems and reported that the CRS exhibits higher COP by 19.4% compared to conventional double-stage VCR systems when the evaporator temperature is below −40 °C. They reported that the volumetric flow in LTC can be reduced up to 840% at −50 °C when CO_2 replaces NH_3. Messineo (2012) showed that CO_2-NH_3

CRS has strong competitiveness with a two-stage R404A VCR system equipped with an ejector when the evaporator temperature is in the range of −50 to −30 ºC. Jankovich and Osman (2015) reported a comparative thermodynamic study between a two-stage NH_3 VCR system and a CO_2-NH_3 CRS and concluded that it is an economically feasible solution to replace VCR system with CRS. Eini *et al.* (2016) stated that using CO_2 and NH_3 not only improves the performance of the system but also improves the operational economy of the system. Further, through an advance exergy analysis, Gholamian *et al.* (2018) reported that CO_2-NH_3 CRS showed better exergy efficiency compared to conventional systems.

Effect of cascade condenser temperature (T_{MC}) on the overall performance to improve COP and to minimize exergy destruction is also studied for CO_2-NH_3 CRS (Getu and Bansal, 2008; Lee *et al.*, 2006). The effect of evaporation temperature of CO_2 on the performance of a CRS with a CO_2-NH_3 system for cold storage and freezing application was studied by Belozerov *et al.* (2007). Dopazo *et al.* (2009) reported analytical study on CO_2-NH_3 CRS to optimize various design and operating parameters. Dopazo and Fernández-Seara (2011) later presented an experimental evaluation of the same system and discussed effect of optimization of some of the operating parameters. They also compared the results with that of a two stage NH_3 refrigeration system.

Jain *et al.* (2015) developed thermodynamic model of a vapour compression-absorption cascaded refrigeration system and analysed its performance using various environmentally friendly refrigerants which can substitute R22. This study considered various parameters and demonstrated that the proposed system can save up to 61% power consumption. Mosaffa *et al.* (2016) investigated two CO_2-NH_3 cascaded refrigeration systems integrated with a flash tank and a flash intercooler with indirect subcooler, they also optimized parameters to improve its COP. This study also assessed the exergoeconomic and environmental aspects considering various components of the proposed system. Patel *et al.* (2019) conducted a comparative analysis of two CRSs using CO_2-NH_3 and CO_2-C_3H_8 refrigerant pair in term of their total annual cost and exergy destruction. The authors compared the total annual cost and exergetic efficiency of CO_2-NH_3 CRS with CO_2-C_3H_8 CRS and concluded that the CO_2-NH_3 CRS showed 6.42% lower exergy destruction.

Tsamos *et al.* (2019) reported comparative study of three commercial refrigeration system configurations CO_2 booster system, CO_2 booster system with parallel compressor & CO_2-NH_3 CRS having natural refrigerants for various climatic conditions. The study showed that for low temperature ambient conditions, CO_2 booster system with parallel compression arrangement gave

8.4% higher efficiency than CO_2 booster system and 8.6% higher than CO_2-NH_3 CRS respectively. However, in warm climatic conditions CO_2-NH_3 CRS was found to have lowest energy consumptions due to avoidance of high pressure supercritical operation of CO_2 in CRS. Sánchez *et al.* (2019) reported an experimental study to determine the energy and environmental impacts of replacing R134a in a CO_2-R134a CRS with potential low GWP refrigerants like R152a, R1234ze (E), R290 and R1270 in the HTC. They compared the performance of CRS equipped with indirect expansion for a range of evaporation and condensing temperatures. The study concluded that use of indirect expansion CRS with R152a allows drastic (68.5%) reduction in refrigerant mass which helps to reduce the total energy consumption. Bellos and Tzivanidis (2019) reported an analytical study on 18 different refrigerants in HTC to identify the refrigerant which can be used in HTC along with CO_2 in LTC at four evaporating temperatures i.e. −35 ºC, −25 ºC, −15 ºC, and −5 ºC. They compared the performances based on energy efficiency and total equivalent warming impact (TEWI) for year round operation and concluded that NH_3, R290, R600, R600a and R1270 are the most promising options for such CRS configuration. Sun *et al.* (2019) reported a study to identify the best suitable refrigerant pairs with low GWP for a three-stage CRS and converged to six refrigerant pairs. They found NH_3 as a good choice for HTC in high ambient temperature conditions for large capacity refrigeration systems. Zhang *et al.* (2020) reported an experimental study on the performance of a CRS having refrigerant pair CO_2-R1270. The study focused on the performance assessment of the refrigerant pair in CRS with R1270 in HTC and to check the potential of R1270 to replace with either of NH_3, R22, R134a and R290. The study reported that R1270 showed better performance and can replace R22 and R134a.

Therefore, recent developments in CRS are promising, due to its enhanced efficiency, ease of refrigerant selection for individual circuit, positive pressure throughout whole cycle etc. The literature study noticeably shows that for single evaporator system, it is promising to use CRS, however, there is limited study that reported the benefits of using CRS in applications having multi-evaporator cooling needs. A seafood processing unit require simultaneous operation of multi-target-temperature evaporators maintained at three to four different evaporation temperatures (Saini *et al.*, 2020). This application is thus unique, and there are no established results on the relative performance comparison between multi-evaporator conventional system and a CRS for such applications. No study presented so far has discussed energetic, environmental and economic performance comparison between a multi-evaporator CRS and a conventional multi-evaporator two-stage VCR system.

2.4.3 Refrigeration system modifications for performance enhancement

Researcher are continuously exploring modifications and innovations in refrigeration systems and components to enhance their energy efficiency, safety and reliability. There are significant environmental and economic benefits from using low GWP alternative refrigerants and to explore possible design changes to meet the specific cooling demand. To increase the system energetic efficiency, use of parallel compression, ejectors expansion, work recovery expander, as well as use of internal heat exchanger, flooded evaporators, secondary refrigeration, twin-staging, sub-cooling, economizer, etc. have been endorsed in literature (Nguyen and Le, 2020; Purohit, Gupta, *et al.*, 2018; Rodríguez-Jara *et al.*, 2022; Singh and Dasgupta, 2016). An improvement in the performance of refrigeration cycle results in energy saving of the overall system. Utilization of additional heat exchanger for subcooling and superheating is a common practice in performance improvisation such a system. A few recent publications reported such modification and integration of the mentioned components are discussed briefly in the following paragraphs.

a) Internal heat exchanger

Integrating VCR system with internal heat exchanger (IHX) has become common practice in order to gainfully utilize the cooling effect of low temperature refrigerant leaving the evaporator to decrease the temperature or sub-cool the refrigerant leaving the condenser below ambient temperature (Chen and Gu, 2005). There are, however, two contradicting effects attributable to the introduction of such an IHX. It increases the specific refrigeration capacity in the evaporator, but on the other hand, it increases the compressor work due to increased specific suction volume of refrigerant (Aprea and Maiorino, 2008). Many researchers have demonstrated that the IHX improves overall performance when the working fluid has high heat capacity and is within a range of evaporating and condensing temperatures (Domanski *et al.*, 1994). Klein *et al.* (2000) concluded that installing IHX in the NH_3 refrigeration system reduces the overall system performance in any operating condition. Zhang *et al.* (2011) theoretically analyzed the effect of IHX integration with CO_2 system both in subcritical and trans-critical operations. Llopis *et al.* (2015) experimentally validated the results obtained by Zhang *et al.* (2011) for the subcritical CO_2 cycle and concluded that refrigeration capacity and COP were reduced by 3.5% and 3.29%, respectively, at −25 °C evaporation temperature. However, for a lower temperature of −40 °C in the evaporator, the system COP increased by a mere 0.45%. Llopis *et al.* (2016) reported a simulation-based comparative study of an R143-CO_2 CRS with and without IHX and observed that an overall increase in COP of 3.7%. Purohit *et al.* (2018) experimentally investigated the energetic and exergetic performance of CO_2 chiller system with IHX for extreme warm climatic condition around 45 °C. The authors reported that integration of the IHX improves energetic and exergetic efficiency by 5.71% and 5.05% in a high ambient temperature of 45 °C.

Liu *et al.* (2021) presented a study with three configurations using IHX at different locations in a refrigeration system with CO_2. By analyzing the thermodynamic model, they concluded that system performance can be increased up to 6.3% by optimally placing the IHX which further resulted in improved annual performance in subtropical as well as tropical climates. It is therefore concluded that there is merit in using the IHX with CO_2 subcritical cycles to enhance performance and there is a possibility of improving the performance by incorporating IHX and gas-cooler in CRS.

b) Secondary or indirect refrigeration system

In VCR systems direct refrigeration is quite common in which refrigerant in evaporators comes into direct contact with the objects or fluid which needs cooling. Due to diffusion and leakage, direct interaction between food and refrigerants is possible in such cases and trace amounts of refrigerant can contaminant the fluid and make it un-safe for consumption. In food refrigeration application, therefore, it is important to avoid refrigerants with toxicity and employ a benign secondary fluid or refrigerants for the heat transfer. Secondary refrigeration system have an increasing popularity in food processing industries (Kumar, 2017).

Secondary refrigeration system are available as either in natural gravity circulation or forced circulation using pumps according to the application (Kumar and Gopal, 2009; Yadav *et al.*, 2012). Secondary refrigeration systems have been shown to use up to 15% less energy than their direct expansion counterparts while still providing the same level of cooling performance (Wang *et al.*, 2010). Apart from the performance enhancement and reduced chance of contamination, secondary system also has the potential to address the defrosting challenges which are common in direct refrigeration systems and also provide design flexibility and better load adjustment flexibility (Liu *et al.*, 2021).

Use of various fluids such as ethylene, propylene glycol, ethyl alcohol, CO_2, brines etc. has been explored as secondary refrigerant in the literature (Wang *et al.*, 2010). Kumar (2017) reported that both NH_3 and CO_2 are suitable candidates as a secondary fluid through a comparative study among various refrigerants. Due to the volatile nature of CO_2, it does not remain as a liquid and is partially evaporated, the phase of CO_2 depends upon the design and operating parameters and it therefore, provide a significantly greater cooling capacity compared to other secondary fluids. This reduces the temperature difference required at the heat exchanger and power required to pump CO_2 (Wang *et al.*, 2010).

c) Flooded evaporators

In order to ensure the optimal use of heat transfer area of the evaporators, flooded evaporators are advised to integrate, which reduce the surface area and improves the evaporator performance. Due to the large amount of refrigerant vapour formation in conventional evaporators, most of the part of evaporator is covered with the vapour which reduce the heat transfer rate compared to the liquid phase (Browne and Bansal, 1999). In flooded evaporators, only liquid phase of the refrigerant is ensured using an electronic expansion valves which increase the wet surface for heat transfer resulting into improved evaporator performance. In this arrangement, vapour refrigerant is removed from the liquid receiver and thus there is no superheating inside the evaporator with enhanced heat transfer and allows the system to operate at some elevated temperatures with the same cooling effect (Okereke *et al.*, 2019).

d) Ejector expansion

An ejector is a work recovery device which helps in reducing the exergy loss by reducing the pressure expansion losses. Common physical configuration of an ejector is an arrangement of nozzle, suction chamber and diffuser. In order to utilize the high pressure energy of refrigerant stream 'motive', it accelerates the motive by passing through a nozzle and converting pressure energy into velocity. It creates low pressure space in suction chamber and utilises the same to suck refrigerant from evaporator. The refrigerant mixture after entrainment with the motive passes through a diffuser that reconverts the velocity into pressure. This process provides a pressure lift to the low pressure refrigerant from evaporator and reduces the compressor work.

Sun (1996) discussed various geometries of ejector along with design details such as mass flow rates, entrainment ratios, geometries etc. A comparative study of performance from using ejector configuration with eleven refrigerants including water has been published by the author in a different study (Sun, 1999). Chen *et al.* (2013) and Besagni *et al.* (2016) presented literature reviews on the recent developments in ejector integrated refrigeration technologies, applications of ejector system, and reported system performance improvements. They also reviewed ejector usages with various refrigerants, their geometric optimizations and parameters affecting their performance. Zheng and Deng (2017) reported an experimental study on ejector integrated refrigeration system with transcritical CO_2 in two-stage evaporation. The study claimed that adoption of an ejector improved performance by 21.7% and cooling capacity by 27.2% of the refrigeration system.

In order to improve the performance of conventional refrigeration systems, any or a combination of modifications discussed above can be explored, especially in the system dealing with low temperature applications. Several studies reported performance improvements with conventional refrigeration systems, however, integration of the modification strategies with CRS are limited and thus further exploration is needed.

2.4.4 Refrigerants: history, regulations and transition

In a VCR system, rejection of heat from a refrigerated body or space to the high temperature sink is carried out using a working fluid termed as refrigerant which alternatively vaporizes as it collects heat from the evaporator and condenses by rejecting heat at the condenser. Calm (2008) reviewed the progression of refrigerants from early use to the present-day usages, and also explored possible future refrigerants. In another review, Abas *et al.* (2018) reported use of various natural fluids as refrigerants in transferring heat, along with their year of introduction. Use of other fluids such as water and hydrocarbons (HCs) were also reported; all these were natural refrigerants. Many of these have detrimental properties like toxicity, flammability, high chemically reactivity, and are thus hazardous. Later, synthetic chemicals like Freon gases, CFCs such as R11 & R12 emerged as an alternative which were efficient over the existing refrigerants (Benhadid-Dib and Benzaoui, 2012). Due to the favourable thermophysical properties, superior thermal performance, better safety due to non-toxic, non-flammable nature of CFCs, and their promotion by the manufacturers for economic benefits caused their widespread use and natural refrigerants started phasing out. By the early 1970s, CFCs were in widespread use in refrigeration industry with approximately one million tons per year production. CFC gases are chemically inert with an atmospheric life in the range of 40-150 years. However, photo-dissociation of CFCs in the stratosphere causes production of Chlorine which further reacts with Oxygen molecules of ozone layers and result in its destruction (Molina and Rowland, 1974). In 1987, 56 countries agreed under Montreal Protocol to cut down production and use of CFC by 50%. In consequent years, the protocol was strengthened and all the ozone depleting substances including CFCs were phased out worldwide.

Meanwhile the use of CFCs, HCFCs which have very low ODP but very high GWP, continued but regulated under the Montreal Protocol. In 2007, in an agreement, all parties agreed to the scheduled phase-out of HCFCs. Developed countries phased-out HCFCs during 2007-20, whereas, developing countries started the stepwise phase-out in 2013 and will complete it until 2030 (UNEP, 2018a). Along with HCFCs, HFCs which have zero ODP with very high GWP were also introduced. After realizing high accumulation of greenhouse gases and their effect on climate change, international community agreed on a treaty in 2016 known as the Kigali amendment to the Montreal Protocol in order to gradually reduce the production and consumption of substance with high GWP (UNEP, 2018a). The Kigali amendment proposes to the production and consumption of the same by 80% over the next 30 years. HFCs are being used as an interim solution but they also possess high GWP and they needs to be phased out before 2040 worldwide according to the nationally declared contributions (NDCs) norms for individual country (Abas *et al.*, 2018).

The concerns regarding high GWP from various synthetic refrigerants are proposed to be mitigated by regulating their usage in a time bounded way under the Montreal and Kyoto Protocols and encourage use of environmentally friendly refrigerants. The pathway to implement this regulation is to reduce dependency on high GWP refrigerants and adoption of low GWP alternatives with energy efficient technologies (Wu *et al.*, 2021). In search of better refrigerants, natural refrigerants like NH_3, CO_2, and a few hydrocarbons (HCs) are reintroduced whereas some low GWP synthetic refrigerants like HFOs are also being used in the industry. The leading candidates, as long-term alternative appears to be natural refrigerants like NH_3, CO_2, HCs and a few HFOs.

McLinden *et al.* (2017) conducted a screening of the PubChem database with more than sixty million chemicals. They reported that only twenty-seven low GWP fluids possess the required combination of chemical, thermodynamic and safety properties suitable for practical application as refrigeration and air-conditioning needs. However, these options are required to be extensively verified for specific applications and ambient conditions to establish their suitability and overall impact on environment. Abas *et al.* (2018) reviewed various natural and synthetic refrigerant addressing the global warming issue and developed a refrigerant qualitative parametric (RQP) quantification model to assist the process of refrigerant selection. It analysed a set of 16 refrigerants including both natural and synthetic for VCR cycle and reported that natural NH_3, CO_2, HCs and a few synthetic refrigerants like R152a, R1234yf are optimal choices. Several studies have been reported about use of low GWP refrigerants for specific applications, some of them also highlighted concerns and various limitations of these small set of refrigerants and also about the technology readiness level and safety issues (Ciconkov, 2018; Nair, 2021; Wu *et al.*, 2021). Various biosphere gases are important candidates for use as natural refrigerants as they are environmentally friendly. However, their applicability is limited by the temperature range of application or because in existing technological state the efficiency is very low, they contribute to indirect source of global warming.

NH_3 is one of the natural refrigerant which is most abundantly used as refrigerants in industrial refrigeration due to its favorable thermos-physical properties such as high latent heat of vaporization, high critical temperature and pressure, high thermal performance (Widell and Eikevik, 2010). It is a preferred choice in large refrigeration plants due to the availability of skilled operators, local component manufacturers, easy leakage detection, etc. As per ASHRAE Standard 34, NH_3 comes in B2L safety group due to its flammability and toxicity. Thus, there are local regulations stipulating special safety measures or restricting installation near populated areas because of its foul smell, toxicity and flammability in the event of a leakage. The toxicity of NH_3 also adds to the restrictions of its usage in proximity with food. Since the normal boiling point of NH_3 is −33.4 °C, therefore, the saturation pressure of NH_3 is lower than atmospheric pressure when used as refrigerant below this temperature. There is a challenge in maintaining the low-temperature evaporator below −35 ºC as the evaporating pressure is lower than atmospheric pressure, introducing the possibility of intake of non-condensable fluids such as air, water vapour etc. into the system. Seafood industries, for example have refrigeration demand below −40 °C (Pearson, 2008). Further, the volumetric refrigeration capacity, defined as the ratio of refrigeration effect to the volume of refrigerant, decreases substantially for NH_3 from a value of 4797 kJ m^{-3}at 0 ºC, to 1034 kJ m^{-3} at −35 ºC (Khudhur *et al.*, 2022). It has a very high latent heat of vaporization about 6 to 8 times that of other common refrigerants. NH_3 system also require of higher compression ratios and increased volumetric displacement particularly when operated at high ambient.

Therefore, a larger compressor displacement is required for maintaining a lower temperature evaporator increasing energy consumption.

More recently, CO_2 has re-emerged as a preferred choice owing to its favorable thermophysical properties, especially at the lower temperature range applications (Hafner, 2016). The non-toxic, non-flammable nature of CO_2 makes it safer than other refrigerants in food processing applications such as plate freezers, cold storages, etc. CO_2 as a refrigerant has favorable thermo-physical properties such as high liquid and vapour density, high volumetric refrigeration capacity, low compression ratio, lower normal boiling temperature, higher safety group A1 rating. Vapour density and volumetric refrigeration capacity of CO_2 at 0 ºC is 97.6 kJ m^{-3} and 22,545 kJ m^{-3} respectively which is almost 28 and 5 times higher than NH_3 (Saini *et al.*, 2021). The higher value of vapour density and volumetric refrigeration capacity of CO_2 leads to more compact components (Purohit *et al.*, 2018). The difference in vapour density is more pronounced for CO_2 at lower temperature. Low boiling temperature and high pressure of CO_2 ensures a smaller compressor displacement at low temperature ensuring higher performance. Therefore, it is more likely to be gainfully utilized for low temperature applications. Figure 2.12, shows the comparison of various properties of NH_3 and CO_2.

In heat exchanger design, refrigerant properties such as liquid to vapour density ratio, surface tension, temperature change per unit pressure drop, etc. play an important role. A lower liquid to vapour density ratio ensures a more homogenous two-phase flow. CO_2 has the lowest density ratio among common refrigerants i.e. 9.5 at 0 ºC, while the same are 182.4, 66.8, and 106 for NH_3, R1234yf, and R1234ze, respectively (Saini *et al.*, 2021). In evaporators, heat transfer is dominated by nucleate boiling which is affected by its surface tension, and lower the value of surface tension, lower is the requirement of superheat for nucleation and vapour bubble growth, ultimately affecting the heat transfer rate in the evaporator. CO_2 offers one of the lowest surface tensions among commonly compared refrigerants which is 4.54 kN m^{-1} at 0 ºC. Temperature change per unit drop in pressure is about 0.011 K kPa^{-1} for CO_2 which is 6-9 times lower than other refrigerants at a given evaporation temperature.

However, the low critical point and high gas cooler pressure of CO_2 are factors that are detrimental to its performance in high ambient temperature operations. It is believed that, these challenges can be mitigated to a good extent by adopting various circuit modification strategies. Further, adopting a CRS in which CO_2 operates in a LTC only can also mitigate some of the challenges (Purohit *et* *al.*, 2018). Further, thermally efficient copper pipes can be used in heat exchangers for CO_2 and is a distinct advantage compared to NH_3.

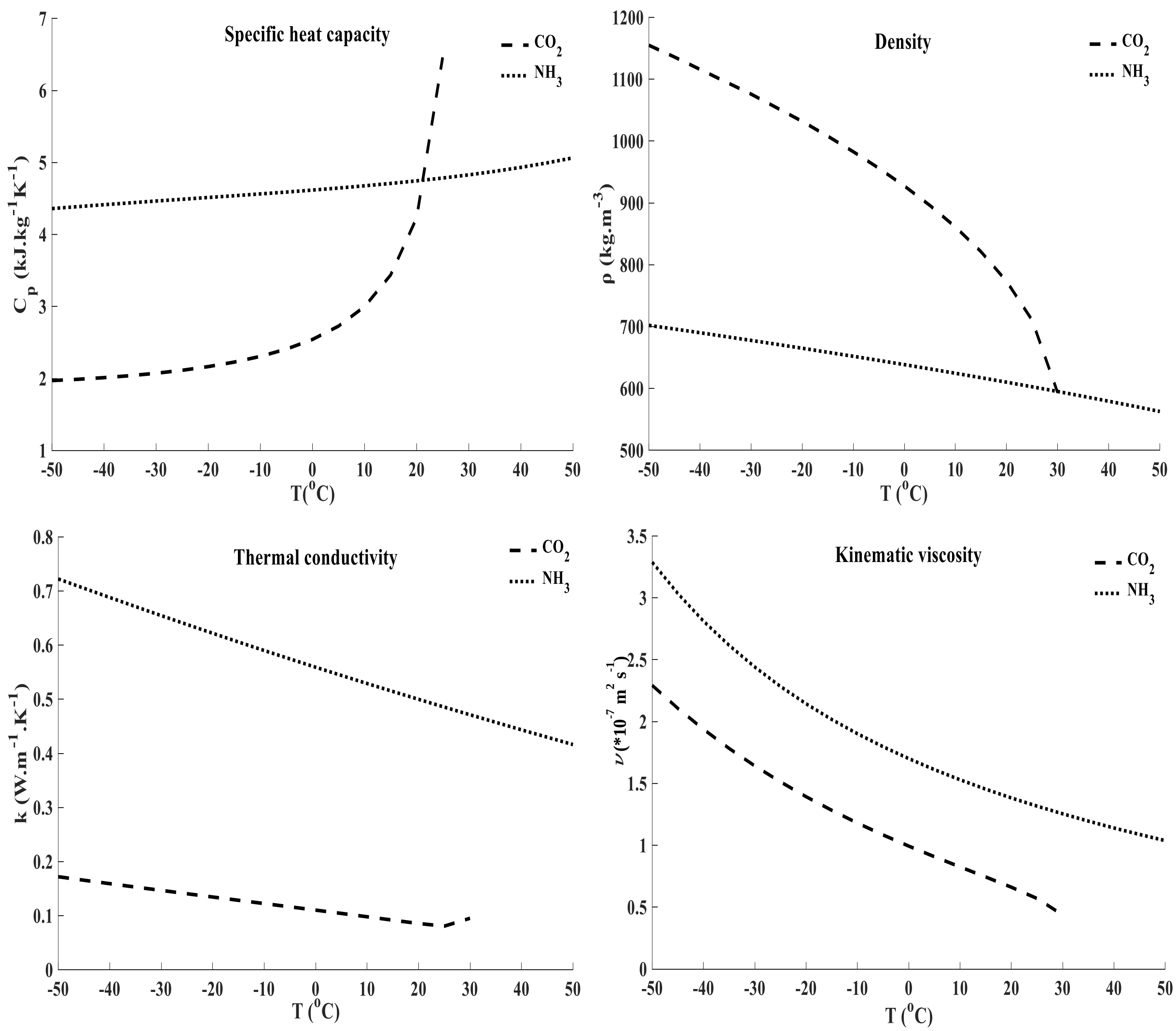


Figure 2.12: Variation of liquid NH_3 and CO_2 thermal properties with temperature

Based on a survey, Kumar *et al.* (2018) reported that R22, R134a, R404A and NH_3 are the most commonly used refrigerants by Indian refrigeration industry, and they also captured the percentage share of these refrigerants across various application. Among these refrigerants, R22, R134a, and R404A have high GWP values of 1810, 1430, and 3943, respectively, and thus are under regulation to be phased out by 2028 in India (Chatterjee, 2022). Although, there are many refrigerants like R32, R1270, R1141 and R1123 which have better thermo-physical properties and show comparatively high energy performances, but the high flammability and various other operational challenges are barriers to their popularity in industrial application (McLinden *et al.*, 2017).

A number of HFC/HFO mixtures have also been identified and studied as potential replacement for R404A with various system modifications (Heredia-Aricapa *et al.*, 2020). However, most of these refrigerants also have relatively high GWP and thus will be eventually regulated, so the search for refrigerants having optimum balance of emissions from direct and indirect sources is still ongoing. As an immediate replacement of R404A, Yang *et al.* (2021) reviewed a few possible alternatives such as R454A, R455A, R457A, R459B, R468A and L40. These refrigerants have almost similar thermophysical properties, and retrofitting may be possible but are not regarded as a permanent substitution solution due to their higher GWP value.

As one of the first nations to introduce the India Cooling Action Plan (ICAP) to control use of refrigerants and improve cooling efficiency, India displayed leadership in 2019. India ratified the Kigali Amendment in September 2021 and is working to build a national strategy to oversee the phasedown of HFCs in the following year in conjunction with the stakeholders (Chatterjee, 2022). It increases the need for the development of low GWP refrigerants that are safer, more adaptable for use in a variety of applications, and perform better.

In order to find out various refrigerant pairs suitable for use in CRS, a focused literature review is conducted. A number of studies are found to have been published in recent years that focused on the performance analysis of various refrigerant pairs in CRS configuration. Table 2.2 gives a few relevant studies focused on various refrigerants pairs and conclusions drawn from their studies about the performance of specific refrigerant pairs in a CRS. Table 2.3 shows the important properties of the commonly studied low GWP refrigerants in the literature (Klein, 2018; McLinden *et al.*, 2017).

Following the selection of a suitable refrigeration system and appropriate refrigerant for particular application, performance and feasibility of the refrigeration systems for any application is generally analyzed based on parameters namely energy, exergy, economy and environmental emissions. Exergoeconomic term is also 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 CRS considering cooling capacity for constant source and sink temperatures of both circuits. Aminyavari *et al.* (2014) presented a study on 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 CRSs, 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. Jin *et al.* (2020) reported that the otherwise complex multi-evaporator refrigeration systems are gaining acceptance in industries due to their higher operational efficiencies, design and installation flexibility, smaller additional costs and compactness. In the economic analysis, generally the initial capital costs, maintenance costs, operating costs and life cycle costs compared while for environmental analysis, TEWI are computed and compared.

Literature review noticeably shows that most of the studies explored only a few refrigerants like NH_3, CO_2, R290, R1270, R41, R161 etc. and left other low GWP refrigerants like RE170, R170, N_2O, and R152a from comparison in a CRS. Moreover, majority of the studies considered only CO_2 in LTC and other refrigerants are not explored much. The reported studies also mostly looked at parameters such as COP, energy consumption, exergy destruction, and emissions, leaving out parameters such as refrigerant cost and availability, safety, system compactness, discharge temperature, etc., which also plays an important role in system selection. The current research work attempts to bridge these research gaps.

Table 2.2: Recent studies on low GWP refrigerants in CRS

Reference study (study type)	Refrigerants pairs		Parameters	Evap / cond temp (ºC)	Remarks
	LTC	HTC			
Chen *et al.* (2022) (Theoretical)	CO_2	NH_3	COP, energetic and exergetic efficiency	–60 to –40/ 40	This study proposed integration of an auxiliary refrigeration loop in the HTC to increase subcooling in the LTC and compared with conventional CRS. An increase in subcooling of LTC by 15 ºC, resulted in increased COP by 6%, and exergetic efficiency by 4.4%. Discharge temperature of NH_3 also reduced by ~10 ºC in the proposed system.
Liu *et al.* (2022) (Theoretical)	R170	R290	COP, exergetic efficiency	–55/ 32	An ejector-enhanced auto-cascade cycle is proposed, and compared with two baseline cycles. To recover expansion work, an ejector was installed in place of the expansion valve, improving COP by up to 42.85%.
Adebayo *et al.* (2021) (Theoretical)	CO_2	HFE7000, HFE7100, NH_3, R134a	COP, TEWI, compressor work, exergetic efficiency	–50/ 35	In terms of energy and exergy, NH_3 is observed to perform best among the four tested fluids in HTC to have the highest performance. As a promising substitute for R134a, a new refrigerant HFE7000 is suggested.
Ko *et al.* (2021) (Theoretical)	CO_2	CO_2	COP, the performance of phase	–35/20 to 45	This study established a dynamic model to examine a CO_2-CO_2 CRS integrated with thermal energy storage for freezing application. Implementing the on/off control,

			change material		compressor running time and rate of evaporator temperature change are compared with and without the heat storage. The study concludes that thermal storage integrated with low temperature refrigeration can result in reduction of power demand.
Aktemur *et al.* (2021) (Theoretical)	R41	R1243zf, R423A, R601, R601a, R1233zd (E), RE170	COP, exergetic efficiency, compressor discharge temperature	–70 to –30/ 40 to 60	Among the compared refrigerant pairs, R41-R423A performed the worst, while R41-RE170 performed the best across all criteria. In comparison to earlier similar studies, COP improvement of 13.05% is reported.
Roy and Mandal (2020) (Theoretical)	R41, R170	R404A, R161	COP, exergetic efficiency, economic and TEWI	–55 to –15/ 30 to 60	The authors reported that R41-R161 has the maximum COP and exergetic efficiency, followed by R170-R161. The plant cost rate using these refrigerant pairs is lower compared with R41-R404A.
Bellos and Tzivanidis (2019) (Theoretical)	CO_2	CO_2, NH_3, R290, R600, R600a, R1270, R152a, R1234ze (E), R1234yf, R32, R448A, R513a, R450A, R134a,	COP, TEWI	–35 to –5/ 10 to 45	This study analysed and compared performance of potential low GWP refrigerants. Several natural refrigerants like NH_3, R290, R600, R600a, and R1270 are found appropriate for high COP and low TEWI. R152a is a good choice for high performance and lower TEWI.

		R407C, R227ea, R404A, R507A			
Sánchez *et al.* (2019) (Experimental)	CO_2	R134a, R152a, R1234ze (E), R290, R1270	COP, total energy consumption, mass charge reduction, TEWI	–20 to 2/23 to 46	Among the compared refrigerant pairs, this study found that R152a is the most effective substitute of R134a. This increases energy consumption by 3.4% while reducing mass by 62% and TEWI by 30%.
Patel *et al.* (2019) (Theoretical)	CO_2	NH_3, C_3H_8	The annual cost, exergy destruction	–40/25	This study reported that among the two compared refrigerant pairs, CO_2-C_3H_8 pair offered 5.33% lower cost and 6.42% higher exergy destruction.
Purohit *et al.* (2018) (Theoretical)	CO_2	NH_3	COP, LCCP	–35 to 5/ 7 to 45	The authors presented a performance comparison of CO_2-NH_3 CRS with multi-stage R404A and CO_2 refrigeration systems. The study concluded that CO_2-NH_3 CRS offers 12.23% higher COP and 11.20% less emissions.
Megdouli *et al.* (2017) (Theoretical)	N_2O	CO_2	COP, exergetic efficiency	–45/ 40	In order to enhance the thermodynamic performances of the CRS, this study theoretically analysed the possibilities of integrating CRSs with ejector and concluded that CRS with ejector in HTC yields 9.16% higher COP.
Cabello *et al.* (2017) (Experimental)	CO_2	R152a, R134a	COP, compressor power and	–30 to –40/30 to 50	The authors conducted experimental analysis to compare the performance of two refrigerant pairs R152a-CO_2 and R134a-CO_2 CRS. The study reported that R152a-CO_2

	discharge temperature	showed better performance at higher temperature difference between evaporator and condenser.

Table 2.3: Thermo-physical properties of selected refrigerants

Refrigerant	Class	GWP	Safety group	Mol. mass (kg mol^{-1})	NBP (ºC)	Critical temp. (ºC)	Critical pressure (MPa)	Refrigeration capacity (kJ m^{-1}) at 0 ºC
R41	HFC	116	A2	34.02	-78.3	44.13	5.90	14264
R152a	HFC	138	A2	66.05	-24.0	113.2	4.52	2569
R161	HFC	4	A3	48.06	-37.6	102.1	5.01	3880
R170	HC	6	A3	30.07	-88.6	32.17	4.87	13947
RE170	HC	1	A3	46.07	-24.92	124.2	5.37	2532
R290	HC	4	A3	44.1	-42.1	96.7	4.25	3882
R1225ye(Z)	HFO	1	A1	132	-19.5	106.9	3.53	2197
R1234yf	HFO	4	A2L	114	-29.5	94.7	3.38	2888
R1234ze(E)	HFO	6	A2L	114	-19.3	109.4	3.64	2195
R1243zf	HFO	1	A3	96.05	-24.2	103.8	3.52	2445
R1270	HC	2	A3	42.1	-47.7	92.4	4.67	4666
R404A	Blend	3943	A1	97.6	-46.3	72.2	3.74	5053
CO_2	Natural	1	A1	44.01	-78.4	30.9	7.38	22545
NH_3	Natural	0	B2L	17.03	-33.4	132.3	11.34	4365
N_2O	Natural	265	A1	44.01	-88.5	36.4	7.25	19820

CHAPTER 3 Thesis Objectives

The cold chain logistics play a significant role towards preservation of food and its quality and nutritional values. Right from the point of harvest in the sea, until it is consumed, it is essential to maintain low temperature for highly perishable products such as seafood. The importance of a cold chain is amplified in a tropical country like India where the ambient temperature is generally high. Apart from assuring domestic food security, a large quantity of seafood is also exported from India thereby earning valuable foreign currency. Through literature survey and site visits, it is observed that the cold chain infrastructure in India is fragmented in nature and is inadequate, the same was highlighted in the first two Chapters of this dissertation - Introduction and Literature Survey. Maintaining a cold chain also has large energy demand, which directly and indirectly contributes to GHG emissions. Therefore, this study is aligned towards identifying and evaluating suitable refrigeration technologies and refrigerants that will ensure better energy efficiency in high ambient temperature surroundings as well as are environmentally friendly solutions.

With support of funding received from DBT, Government of India under 'ReValue' project under technology collaboration with SINTEF Ocean, Norway, this study is focused on analyzing surimi seafood cold chain in India and identify the potential point of interventions necessary in cold chain at various stages. Based on the literature survey and site visits, a few research gaps were identified and are highlighted. These are mainly at fishing stage, surimi processing stage and cold storage stage. This dissertation is an attempt to bridge the highlighted research gaps with the following research objectives:

1. To study existing techniques for maintaining low temperature for on-board preservation after fish harvesting. Further, to explore effective refrigeration system suitable to adopt in small fishing boats in India.
2. To identify refrigeration technologies suitable to deploy in seafood processing plants to meet the various refrigeration demands like supply of chilled water, ice, quick deep freezing and maintaining low temperature in cold storage. Further, to provide a scientific study of comparative performance evaluation based on energetic, environmental emission performance as well as provide an economic feasibility analysis. Further, to explore the

possibility of improving performance of the proposed CRS system by integration of internal heat exchanger and gascooler.

3. To identify potential low GWP and better performing refrigerant alternatives for high temperature ambient.
4. To assess the overall energetic and environmental benefits that the entire Indian surimi supply chain may gain based on various scenarios of implementation of proposed refrigeration technologies and refrigerants.

The objectives of thesis are summarized in Figure 3.1.

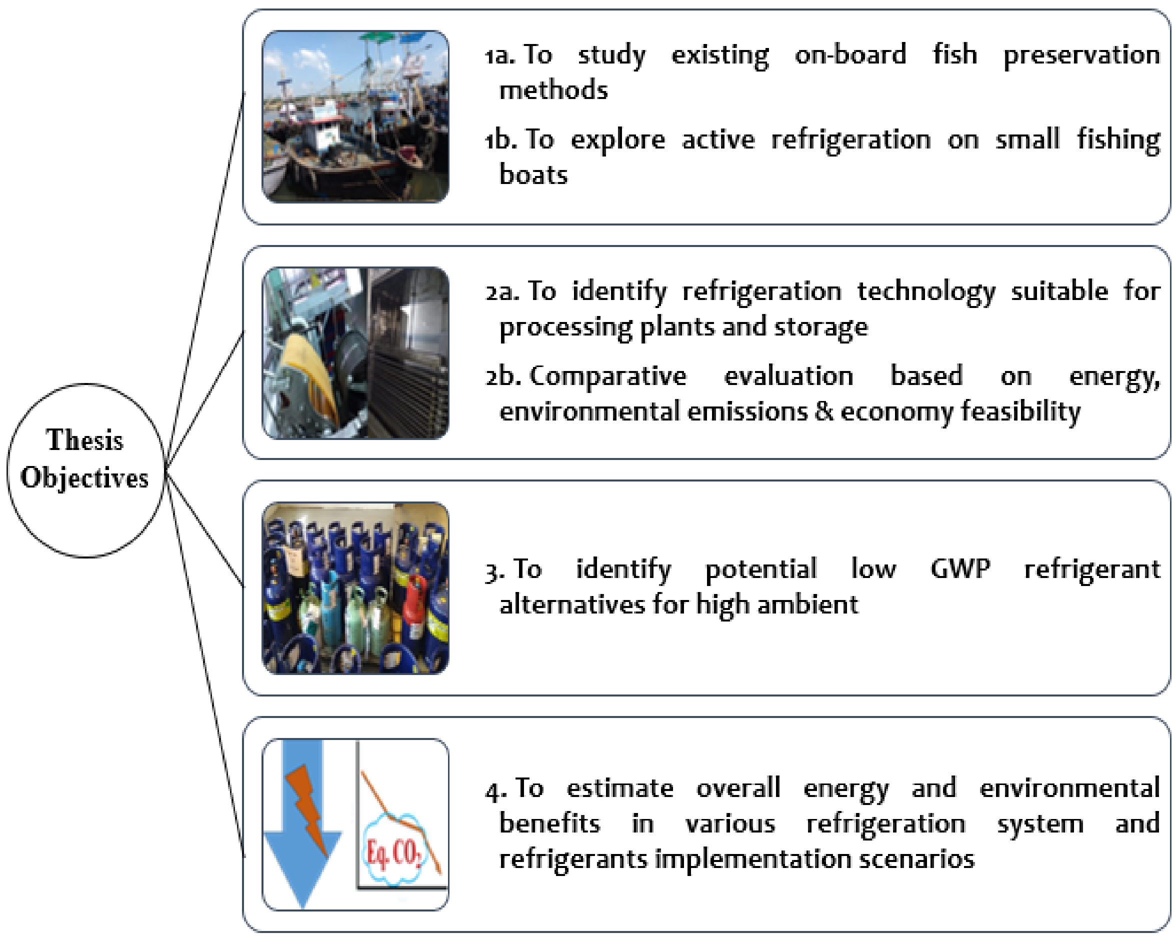


Figure 3.1: Pictorial representation of thesis objectives

CHAPTER 4 On-board Compensatory Refrigeration System

4.1 Abstract

In this Chapter, the feasibility of a novel low capacity on-board compensatory refrigeration system is proposed for small motorized boats that are used by fisherman in the western coast of India. The proposed refrigeration unit will counteract the heat ingress into the iced compartments in the boat and ensure low temperature for preservation of fish quality for the average fishing trip of about 10 days. A heat transfer model of the storage compartment is developed considering parameters such as ambient condition, compartment openings, product removal/loading, container wall insulation, water drainage etc. Refrigeration load required for the proposed retrofitted compensatory refrigeration system is estimated. Feasibility of a vapour absorption refrigeration (VAR) system powered by exhaust heat from the diesel engine is analyzed. Market price of such a system is also estimated and payback period is computed.

4.2 Introduction

Fish and fish products serve as a major source of nutrients and also contribute to substantial export earning for India. The coastline of India spans more than 8100 km; there are about 7 million hectares of inland water bodies which produces nearly 14 million tons of fish annually and this is a major contributor to the food security of the nation. Fish is a perishable product and quality deterioration due to pathogenic, biological and chemical reactions starts immediately after the harvest. Preservation at low temperature is by far the most acceptable method of arresting the rate of quality degradation and is of paramount importance to maintain quality to obtain good price in the market. Measurement and control of temperature of fish from the point of harvest until the fish reaches the consumer needs constant vigilance to ensure food quality and nutritional content.

A large number of marginal fishers are engaged along the coastal areas of India who are major players in supplying fish. They venture into the sea in small motorized boats carrying crushed ice within insulated compartments of the boats for preservation of fish during the trip, average duration of which is about 10 days. Fishing boats are observed to carry crushed ice in multiple insulated compartments built into the boat deck. The harvested fish is sorted, cleaned and stored within layers of crushed ice in the compartments of the boat. Recommended ratio of fish to ice by weight, for preservation in boats is 1:1.5 (Shawyer *et al.*, 2003). Melting ice with adequate water drainage facility in the compartments can ensure storage of catch at 0°C. Fishermen presently use their experience to estimate the quantity of ice to be used. During a survey carried out at Sassoon Dock, a fish landing site near Mumbai (India), it was observed that in many cases, the quantity of ice carried by fishing boat for the typical duration of a fishing trip, was either inadequate or in excess. Less than adequate quantity of ice lead to fish quality deterioration causing loss to the fishermen while excess of ice is an unnecessary cost burden already incurred. There is uncertainty about the duration of a fishing trip depending upon the rate of harvest, changing weather conditions etc. which are beyond the control of the fishermen.

Due to a variety of reasons such as longer than planned fishing duration, improper estimate of ice quantity, excessive heat ingress due to impaired insulation and accumulation of water inside insulated compartments can cause temperature of the catch to rise during storage. This lead to fish quality deterioration resulting in momentary loss to fishers. According to an estimate by TNAU Agritech (2014), about 20% of fish is lost due to improper storage. A potential solution to maintain low temperature is to improve insulation on the walls of the iced compartments and provide a low capacity refrigeration system to compensate for the heat ingress. On-board refrigeration in large fishing boats or trawlers is a common practice elsewhere but due to low technology penetration and presence of marginal players in fishing, the same is not considered a viable solution for adoption in India in the immediate future.

This study explores designing of a system to ensure appropriate fish storage temperature on board the fishing boats. Commonly, the size of compartments in the boats are found to be of square cross section 4 ft x 4 ft (1.22 m x 1.22 m) having a depth of 6.2 ft (1.9 m) with a square lid on top with 2.5 ft (0.76 m) sides. Considering the major modes of heat ingress, total heat transfer rate from surrounding is calculated which constitutes the heat load for the compensatory refrigeration system. Desai *et al.* (2016) stated that majority of Indian fishermen use motorized boats powered by diesel engine of capacity 48-175 hp (35-130 kW). Shu *et al.* (2013) observed that these engines have maximum efficiency of around 40% and rest of the energy is wasted as exhaust heat taken away by exhaust gases, jacket water and cooling air.

We propose to explore utilization of exhaust heat from the diesel engine to power the compensatory ammonia-water VAR system. The working fluids chosen for this refrigeration system are environmentally friendly.

4.3 Theoretical Background

In a VAR system, a binary solution of refrigerant and absorbent is used as working fluid. Input energy in VAR system is heat (low-grade energy). Major components of a VAR system are evaporator, absorber, generator, pump, condenser and expansion valve. Low temperature and low-pressure refrigerant vapour from evaporator enter the absorber, gets absorbed by weak solution and rejects heat to the external sink. The solution is pumped to generator where heat is added from an external source and refrigerant converts into vapour. High-pressure refrigerant vapour passes through condenser and condenses by rejecting heat. Following the condenser, the vapour is throttled in the expansion device to evaporator pressure and the cycle is completed. NH_3-H_2O is a common pair of refrigerants–absorbent used in VAR systems and is suitable for low temperature applications. Here, the temperature of heat source of the generator for efficient working of the system ranges from 50-150 °C. Wang and Wang (2005) discussed feasibility of three different on-board refrigeration systems powered by engine exhaust waste heat - vapour absorption, adsorption and chemical reaction. Shu *et al.* (2013) reviewed various waste heat recovery technologies from engine, available on-board ships and discussed their working principle and feasibility. They found that exhaust heat temperature of a typical diesel engine, that is in the range 250-500 °C for exhaust gas, 130-150 °C for scavenge air and 70-120 °C for engine jacket cooling fluid are sufficient to power the VAR systems. Fernández-Seara *et al.* (1998) analyzed a gas to thermal fluid heat recovery system to power an ammonia VAR system for fishing trawlers. They designed an economizer and generator heat exchanger to recover the waste heat from exhaust gas and concluded that the recovered waste heat is adequate. Táboas *et al.* (2014) analyzed the use of waste heat energy of jacket cooling fluid in diesel engines of fishing ships to drive a VAR system. They also analyzed the performance of VAR system using three different working fluids NH_3/($LiNO_3$+H_2O), NH_3/$LiNO_3$ and NH_3/H_2O.

This study proposes to utilize heat from the exhaust gases from the engine to drive the VAR system. Performance of the VAR system also depends on the sink temperature, where absorber and condenser heat are rejected. As the average seawater temperature remains almost constant at around 28 °C, a stainless-steel pipeline running along the starboard immersed in sea can serve as sink.

The cooling requirements for the fish storage can be divided into three stages: precooling of the crushed ice containers, chilling of fresh catch from sea water temperature to 0 °C and the storage at 0 °C. The VAR system will have a refrigeration capacity covering the heat load/ingress from the surroundings for all stages, as a result the required quantity of crushed ice initially to be carried will reduce substantially. It will be utilized for initial fish chilling predominantly. The cooling requirement rate during fish chilling period is higher compared to other stages and the rate of heat removal from fresh catch is more effective with chilling water compared to ice, due to higher wetting of the surface. The proposed refrigeration system is designed to maintain the temperature at 0 °C, suggesting melting of ice for chilling.

4.4 Refrigeration Capacity

There are many variations in dimensions of fish storage compartments, their shapes and arrangements in a boat, type of insulation material used etc. The preferred insulation materials are polyurethane, fiberglass, cork and polystyrene due to their high strength and low moisture permeability. The present analysis is carried out for a small fishing boat that has 10 compartments of dimensions 1.22 m x 1.22 m x 1.9 m each. The compartments are insulated with 50 mm thick polyurethane foam that is sandwiched between fiberglass-reinforced plastic of 5 mm thickness in each side. These fish compartments are built into the boat deck and are assumed to be exposed to still air except the upper side of the compartment, which is exposed to flowing air above the deck surface. Poku and Ogbonnaya (2018) proposed a heat load and refrigeration capacity calculation for insulated compartments of a fishing boat. Generally, fish up to a certain level fill the compartments and a layer of stagnant air remains in the top part of the compartments. This layer act as additional insulation for heat ingress through the top of the compartments. For cooling load calculation, 1/3rd of the compartment height is assumed to be filled with air. Maximum water temperature in the Arabian Sea is found to fluctuate from 26.5 °C in January to 31.5 °C in June (Sea temperature, 2019). For the analysis of heat load, temperature of fresh catch of fish and ambient air are considered fixed as 30 °C and 35 °C respectively. The fish body temperature is assumed to be equal to the water temperature.

Cooling requirement ($\dot{Q}_c$) of chilling of fish is given by,

$$\dot{Q}_c = \dot{m}_f C_p (T_f - T_0) \quad (4.1)$$

Amount of ice required to melt to remove the heat of catch from its initial temperature T_f to (0 °C) is estimated as,

$$\dot{m}_i L = \dot{Q}_c \quad (4.2)$$

The values of specific heat of fish is within the range of (c_p) 3.13 to 3.38 $kJ\ kg^{-1} K^{-1}$ and latent heat of ice L= 334 $kJ\ kg^{-1} K^{-1}$. Assuming no additional heat load, the amount of ice required for chilling of fresh catch can be estimated as,

$$\dot{m}_i = \frac{1}{3}\dot{m}_f \quad (4.3)$$

Heat ingress into the fish compartments from surrounding takes place due to conduction and convection through the walls, roof and floor of the insulated compartments. The heat transfer equations for conduction and convection at surfaces is given by given by,

$$\dot{Q}_{trans} = UA(T_{amb} - T_0) \quad (4.4)$$

Overall heat transfer coefficient (U) is estimated as,

$$U = \frac{1}{\frac{1}{h_i} + \sum\frac{x_w}{k_w} + \frac{1}{h_o}} \quad (4.5)$$

Air leakages occur from surrounding to storage space due to opening during loading and unloading of fish and through the cracks leading to heat gain. Heat gain through infiltration ($\dot{Q}_{infr}$) is estimated as,

$$\dot{Q}_{infr} = n\dot{V}\rho\Delta h_a \quad (4.6)$$

Heat ingress from direct solar radiation is neglected due to the presence of thick opaque top cover of the compartments in the boat deck. The heat transfer properties used in computation are listed in Table 4.1(Shawyer and Pizzali, 2003) and estimated heat transfer through each component is shown in Table 4.2 along with the total heat ingress for a compartment. Total heat ingress through infiltration and transmission in 10 compartments is found to be 2873 W. For the boat, accordingly the capacity of the refrigeration system considered is 3.5 kW.

Table 4.1: Conductivity and heat transfer coefficients

Factors	Value	Units
Fiber glass	0.036	W m^{-1} K^{-1}
Polyurethane foam	0.048	W m^{-1} K^{-1}
Air	0.024	W m^{-1} K^{-1}
Heat transfer coefficient of air flowing over deck	34.05	W m^{-2} K^{-1}
Heat transfer coefficient of still air near side walls and floor	9.3	W m^{-2} K^{-1}
Heat transfer coefficient of ice-water slurry	598.5	W m^{-2} K^{-1}

Table 4.2: **Heat ingress in refrigerated compartments**

Surface	Surface area (m^2)	U value (W m^{-2} K^{-1})	Heat load (W)
Side walls	9.272	0.7	227.16
Roof	1.488	0.0363	1.89
Floor	1.488	0.7	36.45
Infiltration			21.8
Total ingress in 1 compartment			287.3

4.5 Modelling and Simulation

A schematic of the VAR system is shown in Figure 4.1. A mathematical model is developed on the *Engineering Equation Solver* (EES) platform to analyze the performance of a single effect VAR system using NH_3-H_2O as working fluid. The mathematical model is based on mass and energy balance for each component of the system, while making the following assumptions for simplification.

- Pressure changes in all pipes and components are neglected except pump and expansion valve.
- Steady state condition assumed at all stages.
- Ammonia is considered as saturated vapour at the outlets of the generator and evaporator and as saturated liquid at the outlet of condenser.
- No heat loss occurs to the environment.

- Isenthalpic expansion valve.

Based on the assumptions, the governing equations for mass and energy conservation are listed in Table 4.3. Performance of the VAR system is measured by coefficient of performance (COP) computed using equation 4.7.

$$\mathrm{COP} = \frac{\dot{Q}_{evap}}{(\dot{Q}_{gen}+\dot{W}_p)} \tag{4.7}$$

For the VAR system, solution circulation ratio (λ) is defined as the ratio of mass flow rate of solution entering generator to the mass flow rate of refrigerant.

$$\lambda = \frac{\dot{m}_1}{\dot{m}_7} \tag{4.8}$$

This ratio is used to analyze the performance and feasibility of the system. High circulation ratio means that the solution passing through pump is more compared to refrigerant mass flow rate implying higher work done by pump for less refrigeration effect. The nominal operating condition considered for the simulation are given in Table 4.4.

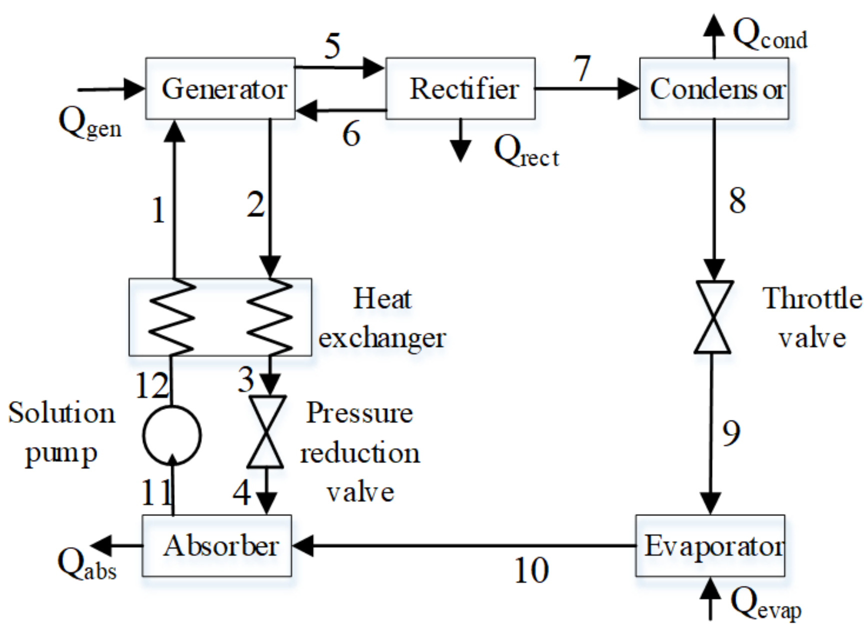


Figure 4.1: Schematic diagram of VAR system

Table 4.3: Governing equations of VAR system

Components	Energy equations
Mass balance	$\sum \dot{m}_i = \sum \dot{m}_o$
Ammonia fraction balance	$\sum \dot{m}_i x_i = \sum \dot{m}_o x_o$

Generator	$\dot{m}_1 h_1 + \dot{m}_6 h_6 + \dot{Q}_{gen} = \dot{m}_2 h_2 + \dot{m}_5 h_5$
Rectifier	$\dot{m}_5 h_5 = \dot{m}_6 h_6 + \dot{m}_7 h_7 + \dot{Q}_{rect}$
Condenser	$\dot{m}_7 h_7 = \dot{Q}_{cond} + \dot{m}_8 h_8$
Evaporator	$\dot{m}_9 h_9 + \dot{Q}_{evap} = \dot{m}_{10} h_{10}$
Absorber	$\dot{m}_4 h_4 + \dot{m}_{10} h_{10} = \dot{Q}_{abs} + \dot{m}_{11} h_{11}$
Solution heat exchanger	$\epsilon_{shx} = \frac{\dot{m}_1 h_1 - \dot{m}_{12} h_{12}}{\dot{m}_2 h_2 - \dot{m}_{12} h_{12}}$
Pump	$W_p = \frac{\dot{m}_1 v_{11} (P_{high} - P_{low})}{\epsilon_p}$

Table 4.4: Operating conditions considered in simulation

Parameters	Values
Evaporator temperature T_{evap}(°C)	–15 to –5
Condenser outlet temperature T_{cond}(°C)	25 to 40
Solution heat exchanger effectiveness ε_{shx}	0.8
Pump isentropic efficiency	0.5
Evaporative heat rejection Q_{evap} (kW)	3.5

4.6 Results and Discussion

The refrigeration system exchanges heat with the seawater that has seasonal variation and a fixed target evaporator temperature. Figure 4.2 shows variation in the generator temperature (T_{gen}) requirement with increase in condenser temperature for evaporator temperature (T_{evap} at –5 °C) and glide temperature in evaporator 4 °C. Maximum water temperature of the Arabian Sea fluctuates annually within the range of 26-32 °C as mentioned earlier. Assuming 5 °C approach temperature, the condenser outlet temperature will be below 40 °C year-round. In this condition, the temperature requirement of the generator is found to be ~118 °C from Figure 4.2. This can be met with available engine exhaust gases heat. The maximum COP of this system is 0.65 for 25 °C condenser temperature in winter. Increase in the condenser temperature in summer will decrease the performance of the VAR system. Figure 4.3 depicts the variation in generator temperature requirement to run the system for various target evaporator temperatures.

The dip in refrigerant temperature in the evaporator side will dictate the rate of heat removal in the evaporator. However, for summer operation the required generator temperature approaches the available heat source temperature (120 °C) and therefore, the rate of heat transfer to the generator will reduce. This will limit the cooling rate achievable with this system in summer. The VAR system configuration can be modified to improve its performance; for example, by inserting a sub-cooler in the configuration as shown in Figure 4.4. This will precool the refrigerant before entering the throttle valve by exchanging heat with the refrigerant after evaporator. Assuming 50% effectiveness of the sub-cooler, a 3.9% enhancement in the system performance in winter and 4.6% improvement in summer operation is observed as shown in Figure 4.5.

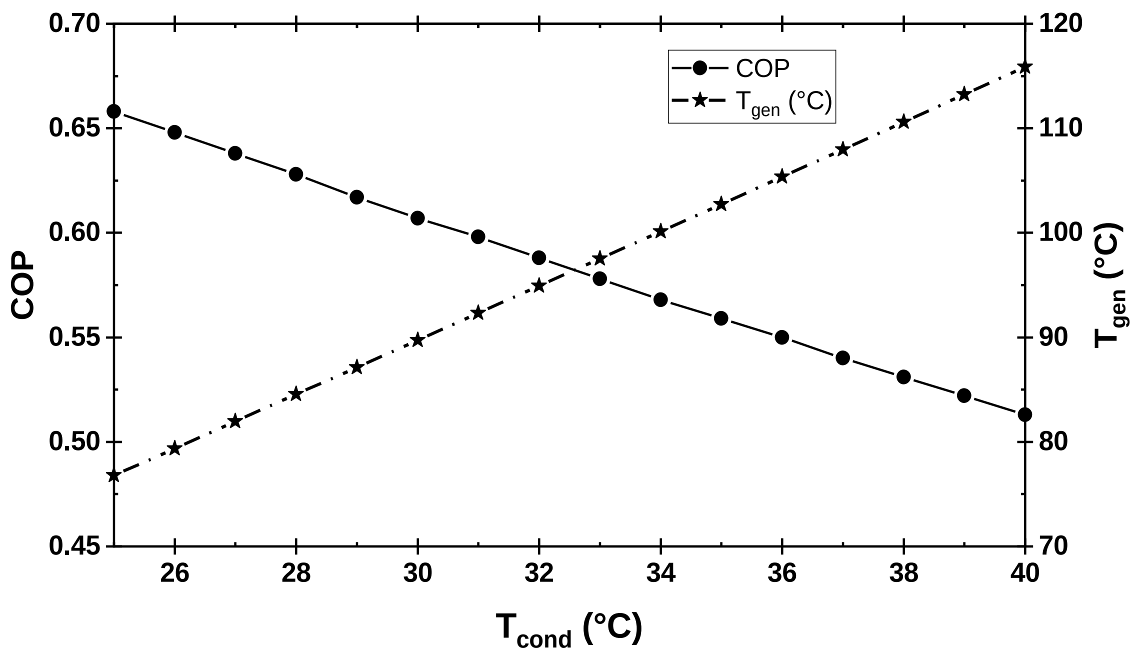


Figure 4.2: Effect of condenser temperature on COP and generator temperature

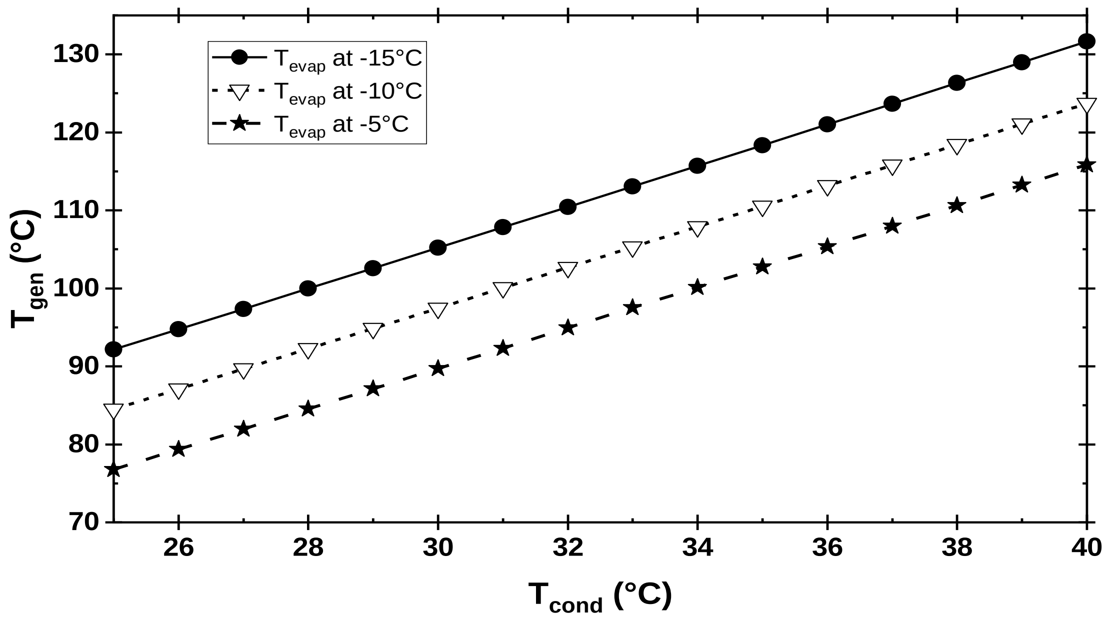


Figure 4.3: Effect of condenser temperature on generator and evaporator temperature

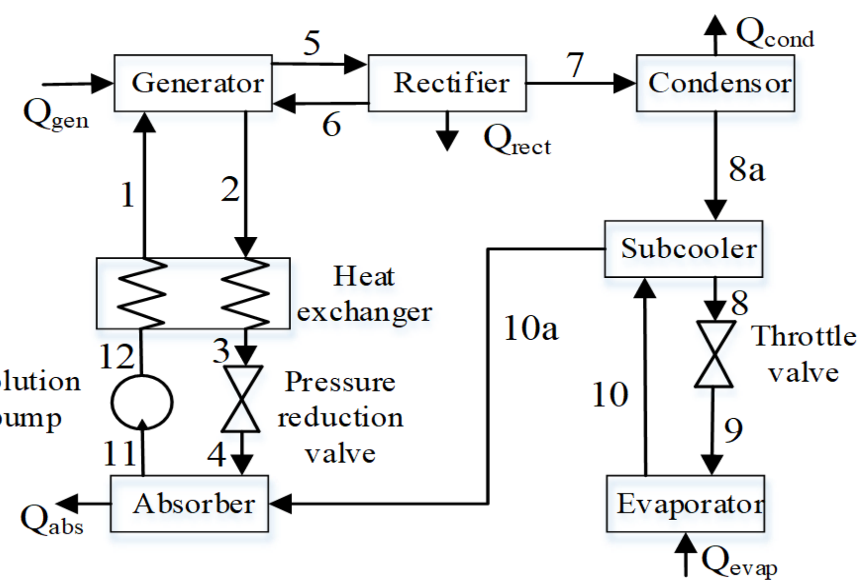


Figure 4.4: Schematic of VAR system with sub-cooler

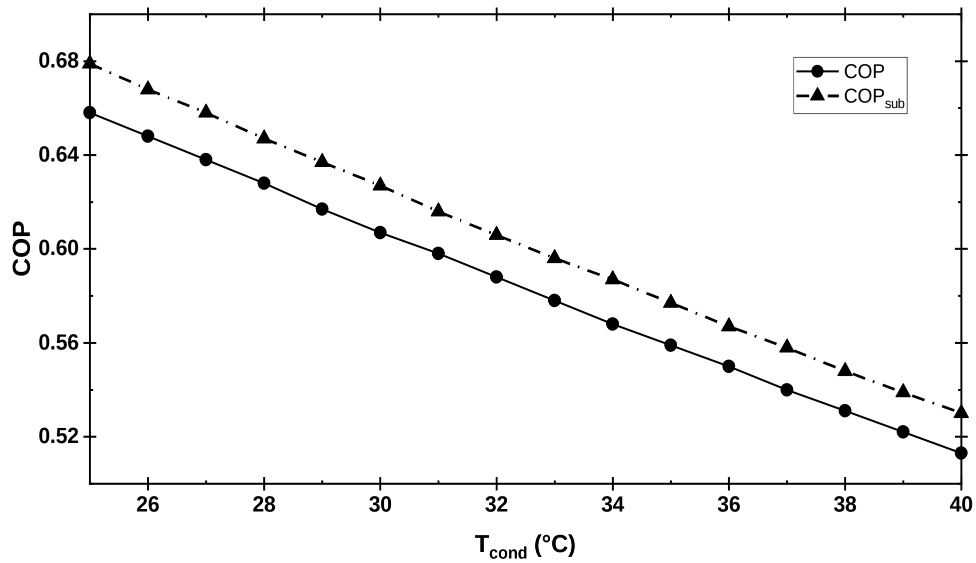


Figure 4.5: Comparison of system performance with and without sub-cooler

4.6.1 Payback period

Limited scale economic analysis is carried out to ascertain financial viability of the proposed VAR system installation on the boat. Initial capital cost of the VAR system is computed considering cost of all the components and their installation cost. Cost of heat exchangers is calculated based on heat transfer area and considering the required material (Purohit *et al.*, 2017). Jiang *et al.* (2002) presented a thermo-economic model of VAR system using waste heat as heat source. Simple payback period for the system is given by:

$$Payback\ Period = \frac{Initial\ capital\ cost}{Net\ annual\ saving\ by\ Installing\ VARS} \tag{4.9}$$

Initial capital cost of a VAR system having of 3.5 kW capacity is found to be ₹3.1 Lakh. Additionally, considering 50% expenses for piping and installation, the total investment is estimated as ₹4.65 Lakh (~$ 6000) per boat. Operational cost and interest rate on the capital cost are neglected in this analysis. The additional weight due to the installation of the VAR system is found to be about 1.2% of fish carrying capacity of the boat. At the same time the compensatory refrigeration system will reduce the amount of crushed ice to be carried by the boat. The combination of these two factors are expected to have nominal impact on operating cost of the boat engine.

During a survey at fish landing site we found that the number of trips of a fishing boat in a year varies from 10 to 35 with varying catch size. The analysis assumes a median 15 trips by boat in a year, average catch size of 8 tonnes per trip, and ₹30 per kg wholesale price of fish. Installation of the refrigeration system is expected to improve shelf life of fish and reduce spoilage level to 10% of the catch from initial 20% leading to increased profit. So the net annual savings per boat is estimated as ₹3.6 Lakh ($ 5160). Payback period of installing this absorption refrigeration system is thus assessed to be 1 year 4 months.

4.7 Conclusions

This study presents a novel compensatory refrigeration system for a small motorized fishing boat to improve the profit of fishermen by reducing deterioration of fish quality. The refrigeration system is designed to compensate for the heat ingress into the insulated storage compartments and ensure storage near 0 °C. This could also lead to better predictability of the appropriate amount of ice to carry. The system also has the potential to reduce quantity of ice usages for cooling. A single stage NH_3-H_2O VAR system is found to be suitable for the application, which derives heat from the engine jacket exhaust and rejects heat to the seawater. The payback period of the VAR system is found to be around 1 year 4 months.

CHAPTER 5 Cascade Refrigeration System for Seafood Processing and Storage

5.1 Abstract

Three novel CO_2-NH_3 multi-evaporator cascade refrigeration systems (CRS) are proposed and evaluated for application in seafood processing for high ambient temperature conditions prevailing in tropical regions. The CRSs have evaporators both above and below the cascade temperature and are intended to replace a multi evaporator NH_3 system. These systems essentially exploit the comparatively superior thermo-physical properties of CO_2 in the low temperature evaporators while NH_3 in the high temperature condenser keep the overall operating pressure low. Thermodynamic models of the CRSs for a single evaporator are validated against published literature. The first proposed system CRS1 has an individual compressor for each evaporator whereas the other two systems CRS2 and CRS3 incorporate a pumped circulation system in CO_2 and NH_3 circuits respectively. The CRSs improve the overall efficiency, reduce ammonia charge and reduce food contamination hazards from NH_3. Suitability of the proposed CRSs are established for a wide range of tropical ambient temperature and also for cooling demands in other prominent seafood processing and storage applications like fish fillets and shrimp/prawn. CRS1 is found to yield the highest COP advantage of about 11.5% in surimi processing where the foremost cooling load is at a high temperature evaporator. While CRS3 is found rewarding for fish fillet & shrimp/prawn industry where the major cooling load is at low temperature evaporator. Compared to the conventional baseline system, COP advantage of 16.5% and 20.3% respectively are attained for fish fillet & shrimp/prawn processing. Dealing with surimi in the climate condition of Mumbai (India), CRS1, showed a maximum of 8.3% reduction in annual average energy consumption, while for fish fillets and shrimp/prawn applications CRS3 showed the maximum annual average energy reduction of 7.5% and 13.2% respectively.

5.2 Introduction

The surimi supply chain has a large refrigeration requirement. Generally, iced storage is adopted in a boat for preservation while sophisticated refrigeration systems are implemented during processing, storage, and export. In general, about 20 kWh of energy per tonne of fish is consumed for fish processing, of which freezing and cold storage consume 50-70% depending on the fish variety and process (FAO, 2015). Therefore, an energy-efficient refrigeration system is essential in the overall supply chain of surimi that can enable Indian firms to maintain quality and attain a competitive advantage in the global seafood market.

A surimi processing plant located in Mumbai and having a design surimi production capacity of 27 tonnes per day (TPD) had partnered with us for this study and data of this plant has been utilized in this study. All the refrigeration demands in the plant are met with a single NH_3 vapour compression system. However, due to uncertainty in the supply of raw material, the plant was observed to operate in part load condition for a major part of the year. Consequently, the need was felt for a lower capacity plant, and accordingly, a refrigeration system to support 10 TPD production was designed. The design is modular in the sense that a higher overall capacity plant should be built by suitably scaling or deploying a cooling system for multiple of 10 TPD production plant, such that part-load operations are handled better.

In this study, first, the various cooling loads for a typical plant having production capacity of 10 TPD are estimated. A conventional multi evaporator NH_3 system for the same load is also modelled, adopted from the existing plant. The computed efficiency of such a system is expected to be more than those actually deployed currently in the field. Three novel CO_2-NH_3 multi-evaporator cascade refrigeration systems (CRSs) are proposed and evaluated having evaporators both above and below the cascade temperature and are intended to replace the multi evaporator NH_3 system. Superior thermal properties of CO_2 at low temperature application is exploited in these CRSs. The developed system models are validated against published theoretical and experimental studies. One of the novelties of the conceptualized system is that the evaporators are placed both above and below the cascade temperature, analytical study of such configuration is not available in the literature. The suitability of CO_2-NH_3 CRSs for other seafood processing plants that have similar cooling demands is also explored in this study.

5.2.1 Cooling load in surimi processing

The throughput time of surimi processing in a semi-automatic plant is about 125 minutes and throughout, the temperature of fish meat is maintained below 10 ºC. The various cooling demands in the processing plant are: chilling of a large quantity of wash water, production of ice, freezing of product, and maintaining the cold storage temperature. The amount of chilled water required is about 10 times that of surimi produced or 4 times the raw material (Park, 2013). Ice is required for pre-chilling before production and is observed to be about 1:1 in ratio of the product. Surimi produced is packed into 10 kg blocks of size 360 mm x 590 mm x 55 mm and are frozen using plate freezer up to –25 ºC.

For a 10 TPD capacity plant, 1000 blocks of surimi will be produced. These needs to be frozen in 24 hours including loading and unloading time. Observed freezing time for a batch is 2 hours, and additionally loading and unloading time is 0.5 hour; therefore, for minimum load operation, we require to operate the plate freezer in 9 batches every day, each batch for freezing to handle 112 Surimi blocks. During freezing, 3425 kJ heat must be removed from a surimi block to cool from 10 ºC to –35 ºC. To freeze 112 blocks in 2 hours, refrigeration load of a plate freezer is 53 kW. With 10% additional cooling load to compensate for heat ingress and other losses in the plate freezer, the estimated freezer load is 60 kW. The associated cold storage with the plant is designed to have capacity to store the production of at least 5 months i.e. capacity of the storage to be designed in our case is 1500 tonnes which has a product volume of 1460 m^3 assuming product density of 1029 kg/m^3 (Park et al., 2013) Frozen surimi blocks after packaging is shifted to cold storage maintained at –20 ºC. A cold storage having a gross volume of 4375 m^3 is found to be suitable for the application with 200% extra space for circulation and approach. Accordingly, a single-story square cross-section storage space having a length of 25 m and a height of 7 m is assumed for estimating the cold storage refrigeration demand. Considering various losses and heat ingress, the total estimated refrigeration load is 70 kW. Cooling demands estimated for various evaporators are presented in Table 5.1, for a groundwater temperature of 30 ºC, the design value for Mumbai (Saini et al., 2021).

5.2.2 Choice of refrigerants and refrigeration systems

NH_3 and CO_2 have negligible GWP and ODP values which place them in the highest rank among the environmentally friendly refrigerants. NH_3 has a very high latent heat of vaporization i.e. 6-8 times than other common refrigerants at 0 ºC. NH_3 is also a well-established refrigerant for application in seafood processing. It is a preferred choice in India for large refrigeration plants due to the availability of local component manufacturers, skilled operators, easy leakage detection, etc. However, the pungent smell in the event of a large leakage, toxicity, and flammability are some of the common criticisms and as a result, there are regulatory restrictions to deploy large capacity NH_3 systems near highly populated areas. Further, there is food contamination potential due to leaked NH_3. The high flammability and toxicity of NH_3 put it in the B2L safety group. The seafood industry has cooling demand at a temperature lower than –40 °C, and due to high boiling point, NH_3 operates in vacuum pressure at that temperature having possibility of air and moisture ingress into the system.

Table 5.1: Cooling demands in a 10 TPD per day surimi production plant in Mumbai

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

CO_2 as a refrigerant has favorable properties such as high liquid and vapour density, high volumetric refrigeration capacity, low compression ratio, lower normal boiling temperature, A1 safety group rating, and it is a biosphere gas as well. Vapour density and volumetric refrigeration capacity of CO_2 at 0 ºC is 97.6 kg m^{-3} and 22,545 kJ m^{-3} respectively which is almost 28 and 5 times higher than NH_3. The higher value of vapour density and volumetric refrigeration capacity of CO_2 helps to maintain components at a compact size. The difference in vapour density is more pronounced for CO_2 at a lower temperature. Low boiling temperature and high pressure of CO_2 ensure a smaller compressor displacement at a low temperature ensuring higher performance. Therefore, it is more likely to be gainfully utilized for low temperature applications. Further, thermally efficient copper pipes can be used in heat exchangers for CO_2 and is a distinct advantage compared to NH_3. The non-toxic, non-flammable nature of CO_2 also makes it safer than other refrigerants to be used in food processing applications such as in plate freezers, cold storages, etc. where there is a greater chance of food coming in direct contact with the refrigerant in the event of a leakage. The low critical temperature of CO_2 and corresponding higher operating pressure at high temperatures are major disadvantages of the CO_2 system.

In evaporator design, properties such as liquid to vapour density ratio, surface tension, temperature change per unit pressure drop, etc. also play an important role. A lower liquid to vapour density ratio may give a more homogenous two-phase flow. CO_2 has the lowest density ratio among common refrigerants i.e. 9.5 at 0 ºC. The density ratio is 182.4 for NH_3 and for HFOs R1234yf, and R1234ze, it is 66.8, and 106, respectively. In evaporators, heat transfer is dominated by nucleate boiling which is affected by its surface tension, lower the value of surface tension lower is the requirement of superheat for nucleation and vapour bubble growth, ultimately affecting the heat transfer rate in the evaporator. CO_2 offers one of the lowest surface tensions among all compared refrigerants which is 4.54 kN m^{-1} at 0 ºC. Temperature change per unit drop in pressure is about 0.011 for CO_2. It is 6 to 9 times lower than other refrigerants at a given evaporation temperature. Considering all the properties, in this study CO_2 and NH_3 have been used as refrigerants for low temperature and medium temperature evaporators respectively.

The use of both CO_2 and NH_3 has also been explored as a secondary refrigerant in the literature. Kumar (2017) reported that both NH_3 and CO_2 are suitable candidates as secondary fluids through a comparative study among various refrigerants. Due to the volatile nature of CO_2, it does not remain a liquid and is partially evaporated, the phase of CO_2 depends upon the design and operating parameters; it, therefore, has a significantly greater cooling capacity compared to other secondary fluids. This reduces the temperature difference required at the heat exchanger and the power required to pump CO_2. Based on the reported literature, CRS refrigeration systems with CO_2-NH_3 refrigerant pair is selected for this study considering their performance, feasibility, high ambient temperature conditions, and popularity of the NH_3 system in the industry. Three CRS systems are conceptualized and out of these system configurations, two systems CRS2 and CRS3 uses pumped circulation of CO_2 and NH_3 respectively to utilize the possible gain in system performance.

5.2.3 Baseline two-stage refrigeration system

The piping diagram and P-h plot of a conventional multi-evaporator NH_3 vapour compression refrigeration system are shown in Figure 5.1 which is used as a baseline system. It has four evaporators, termed *ch*, *ice*, *cs*, and *pf* for chilled water, ice, cold storage, and plate freezing applications, respectively. All the evaporators are flooded type and have individual expansion valves, which receive refrigerant from a receiver. In *cs* and *pf* applications, the pressure ratios are high; hence, two-stage compressors with inter-stage cooling are used. For inter-stage cooling, refrigerant is expanded from liquid receiver (LR) up to the intermediate pressure and is mixed with compressed refrigerant coming out from low 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. The condensed NH_3 is collected in the LR and recirculated through evaporators depending on the refrigeration demands.

5.2.4 Proposed cascade refrigeration systems

Piping diagrams of the three most promising designs of proposed CRSs along with respective p-h charts are shown in Figure 5.2-Figure 5.4. CRS1 is with individual compressors for each evaporator while and CRS2 and CRS3 are cascaded along with secondary refrigeration systems for *ice* in CO_2 and NH_3 circuits respectively. In CRS1, CO_2 is used as a refrigerant in the two lower temperature 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 a pumped circulation of and NH_3 respectively to investigate the suitability of the secondary refrigeration system in this application.

In the CRS2 system, the evaporation temperature of the *ice* evaporator is set equal to the condensing temperature of LTC. Flash gas generated in the receiver due to heat addition in the *ice* evaporator is removed from the receiver and mixed with compressed CO_2 of the *pf* and *cs* compressors. In CRS3, liquid NH_3 is circulated in the *ice*, *ch,* and *cc* evaporator while the flash gas is separated from the receiver and mixed with the *ch* evaporator outlet before compressor suction in HTC. The pump circulation ratio, defined as the ratio of the total mass of liquid-vapour mixture to the mass of vapour phase refrigerant, is taken as 1.5 (Sharma *et al*., 2014).

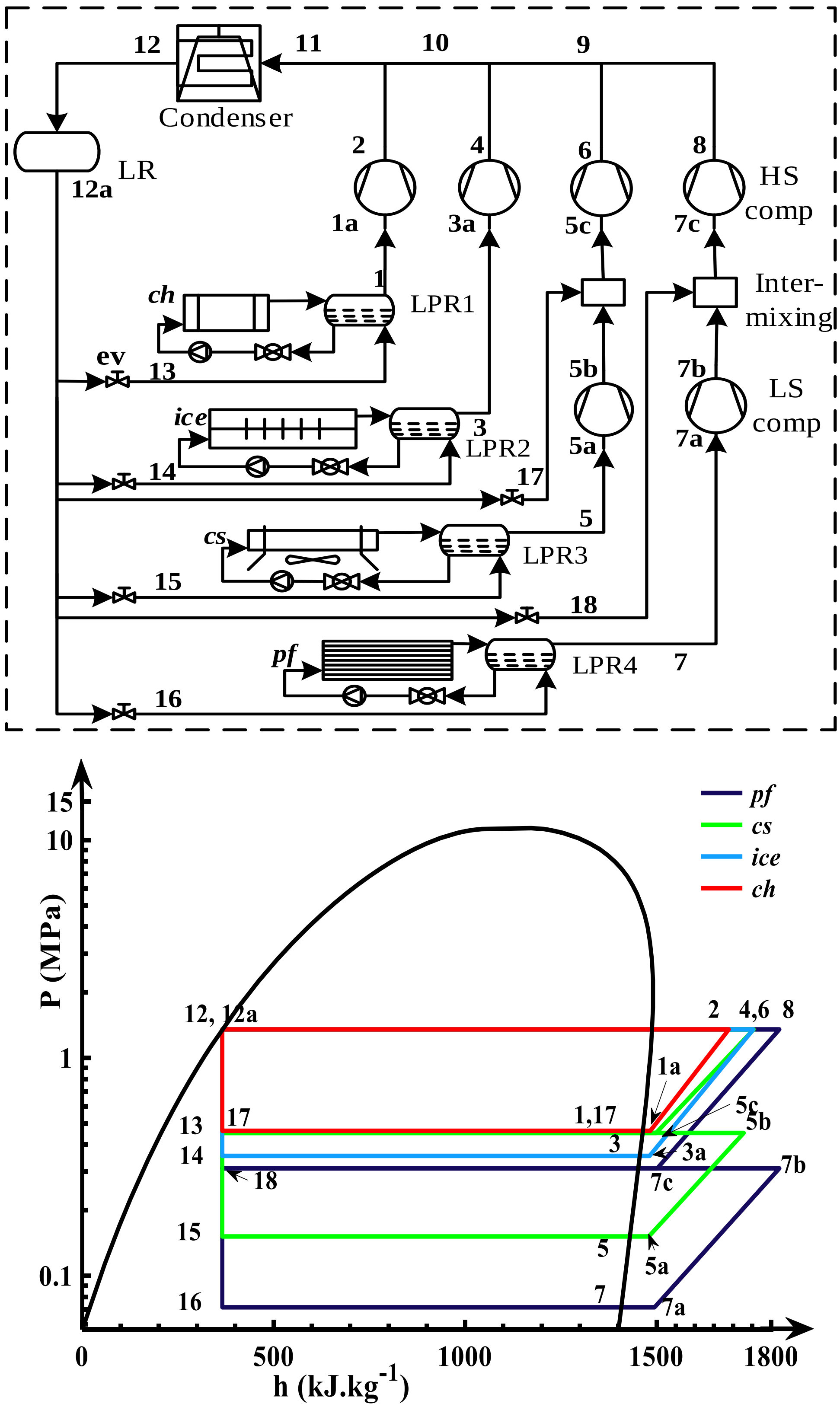


Figure 5.1: Piping and P-h diagram of baseline NH_3 system

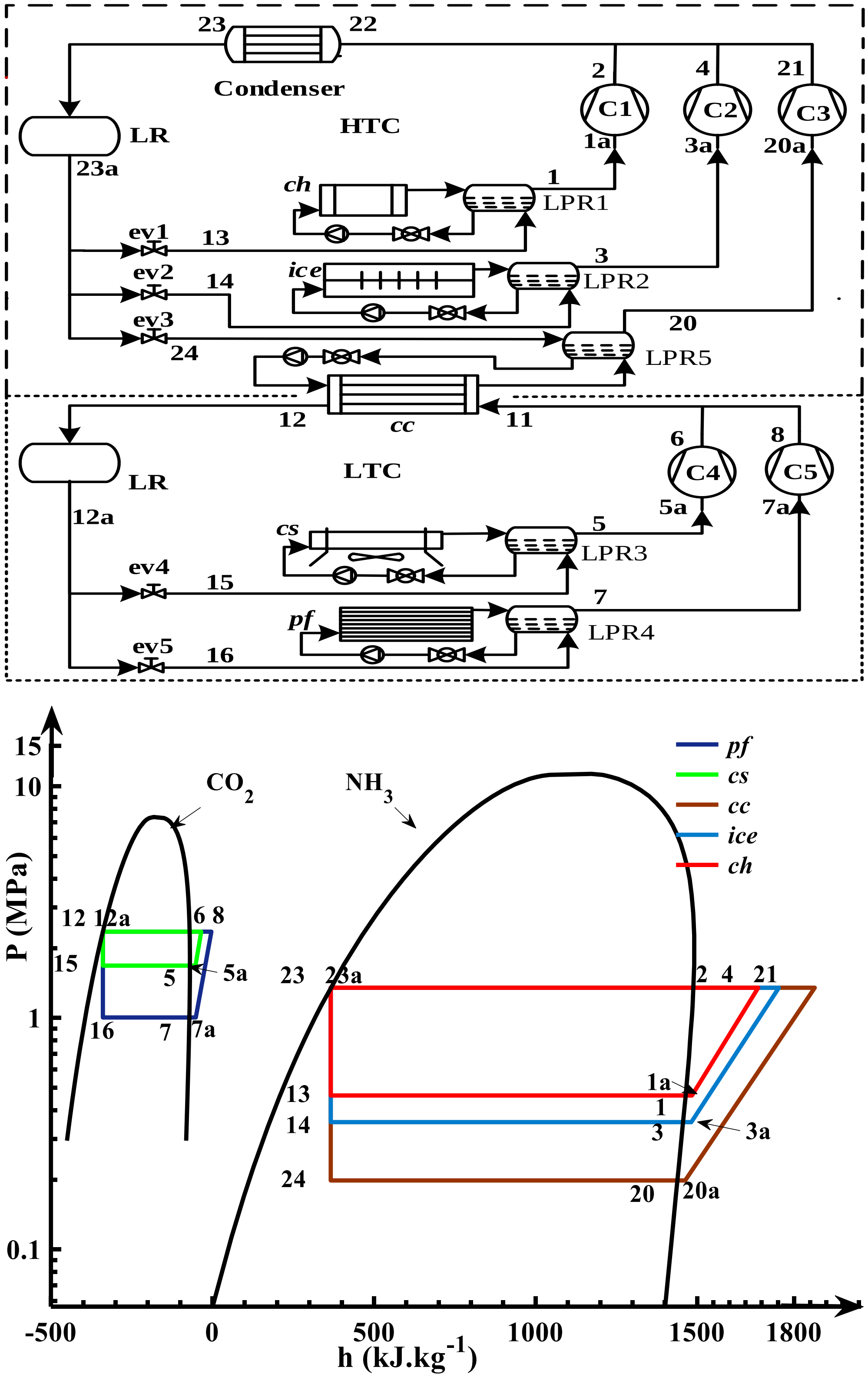


Figure 5.2: Piping and P-h diagram of proposed CRS1

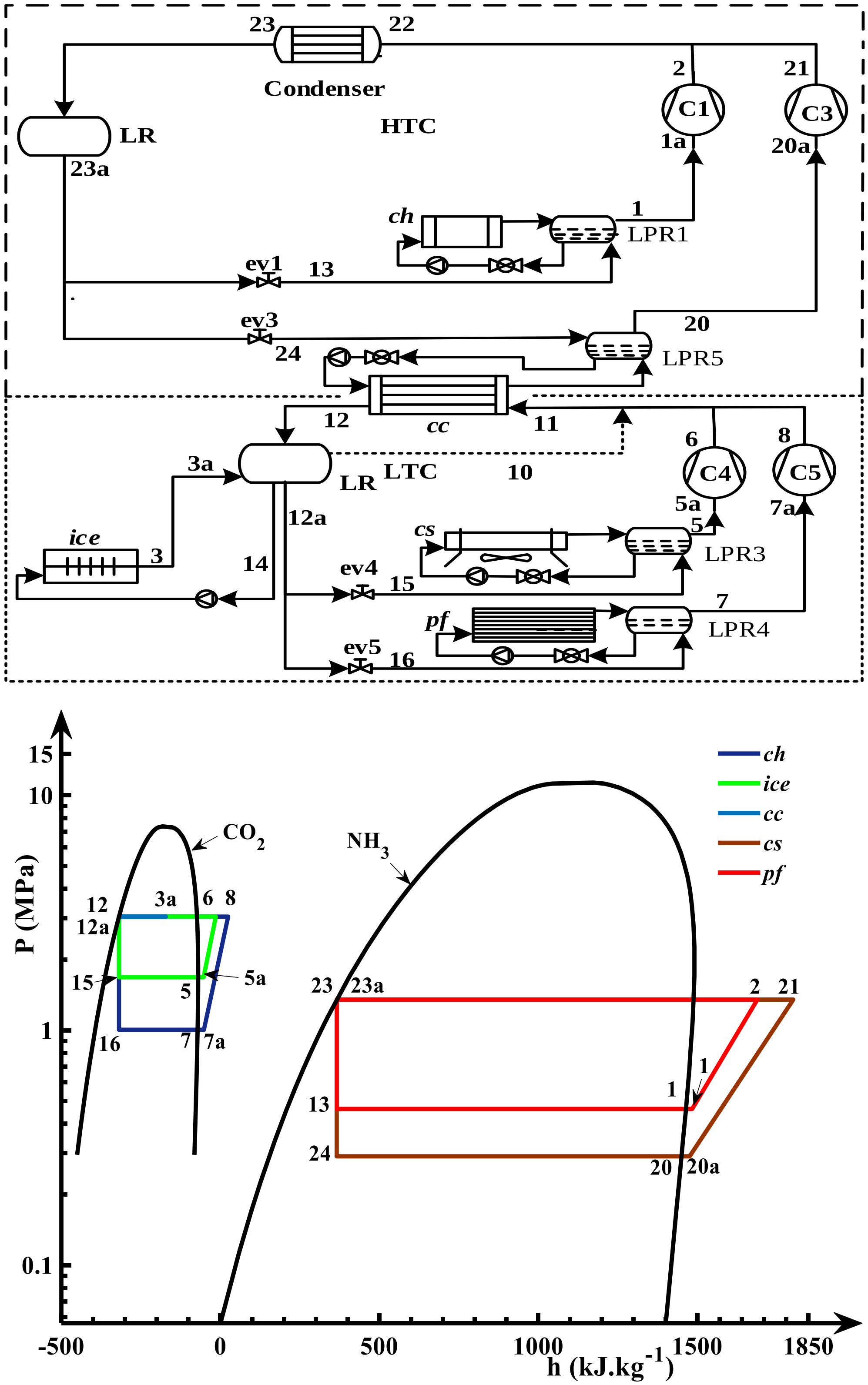


Figure 5.3: Piping and P-h diagram of proposed CRS2

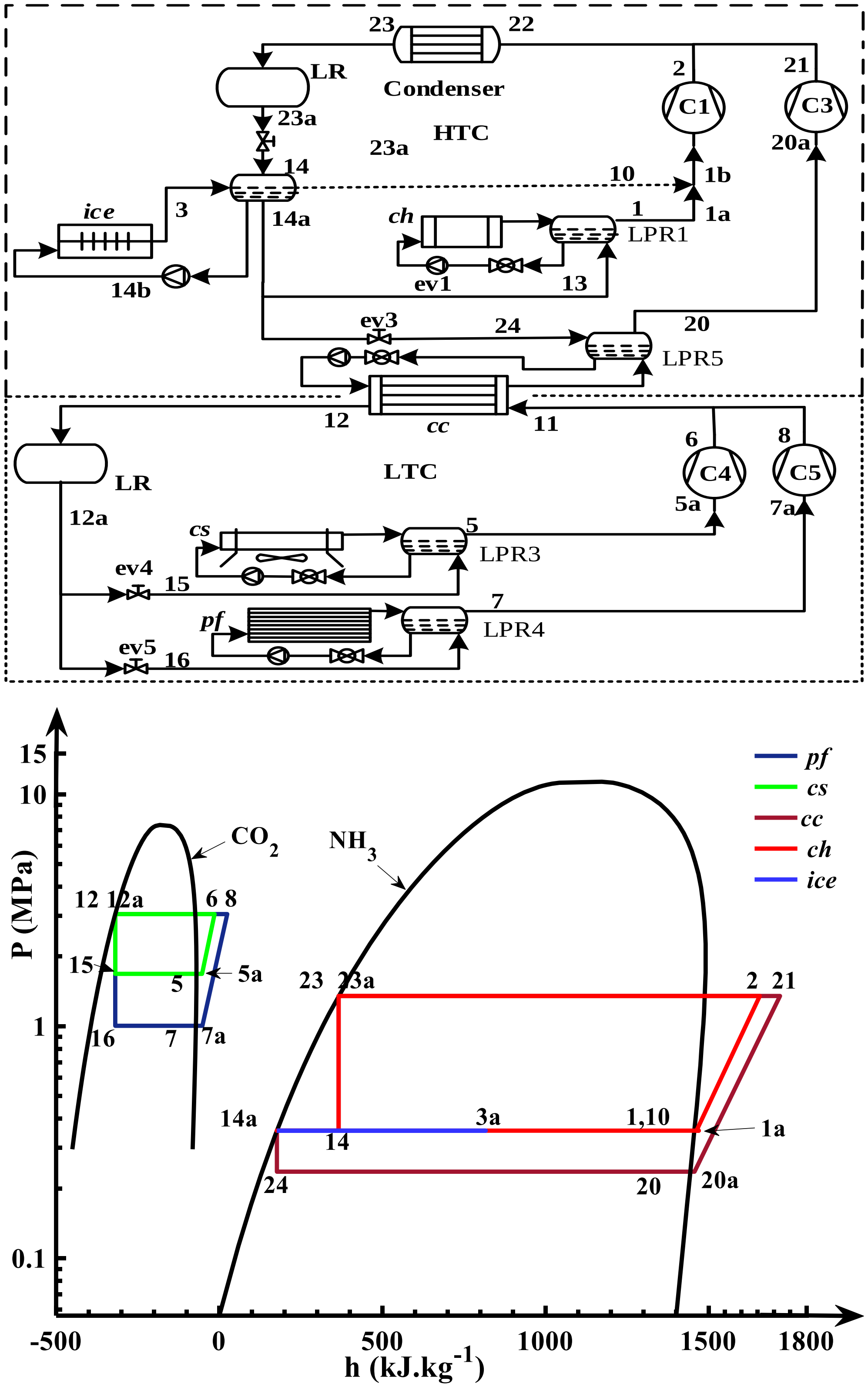


Figure 5.4: Piping and P-h diagram of proposed CRS3

5.3 Thermodynamic Model and Assumptions

The scope of this analytical study is to compare the performance of three major CO_2-NH_3 CRS systems conceptualized, with that of conventional NH_3 systems for refrigeration demands in a seafood processing plant in warm ambient.

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

- Steady-state steady flow process.
- Pressure drops and heat losses in pipes other than suction pipes are neglected.
- Refrigerant at the outlet of the evaporator is dry saturated vapour.
- Refrigerant at the outlet of the condenser and cascade condenser are saturated liquid.
- Isenthalpic operation of throttle valves.
- Compression is adiabatic with isentropic efficiency varying with pressure ratio.
- Refrigerant is superheated vapour at the inlet of the compressor.
- Power consumption of fan and water circulation pump is negligible.

Equations 5.1 and 5.2 present the overall mass and energy balance while component-wise equations are given in Table 5.2.

$$\sum_{in} \dot{m} = \sum_{out} \dot{m} \tag{5.1}$$

$$\dot{Q} + \sum_{in} \dot{m}\, h = \sum_{out} \dot{m}\, h + \dot{W} \tag{5.2}$$

The overall coefficient of performance (COP) is one of the metrics used for performance analysis, and is given by equation 5.3.

$$COP_{baseline} = \frac{\dot{Q}_{ch}+\dot{Q}_{ice}+\dot{Q}_{cs}+\dot{Q}_{pf}}{\dot{W}_{net,baseline}} \tag{5.3}$$

Individual compressor work ($\dot{W}$) is estimated using equation 5.4. $\dot{Q}_{ch}$, $\dot{Q}_{ice}$, $\dot{Q}_{cs}$, and $\dot{Q}_{pf}$ are cooling loads in kW on *ch*, *ice*, *cs*, and *pf* evaporators respectively while $\dot{W}_{net}$ is the sum of all compressor work as estimated using equation 5.5.

$$\dot{W} = \frac{\dot{m}_r(h_{out,s}-h_{in})}{\eta_s} \tag{5.4}$$

$$\dot{W}_{net,baseline} = \dot{W}_{ch} + \dot{W}_{ice} + \left(\dot{W}_{LS} + \dot{W}_{HS}\right)_{cs} + \left(\dot{W}_{LS} + \dot{W}_{HS}\right)_{pf} \tag{5.5}$$

Table 5.2: Mass and energy balance equations for system components

Cooling load	Component	Mass balance	Energy balance
Chilled water	Expansion Valve	$\dot{m}_{13} = \dot{m}_{12a} - (\dot{m}_{14} + \dot{m}_{15} + \dot{m}_{16} + \dot{m}_{17} + \dot{m}_{18})$	$h_{12a} = h_{13}$
	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	$\dot{m}_{14} = \dot{m}_{12a} - (\dot{m}_{13} + \dot{m}_{15} + \dot{m}_{16} + \dot{m}_{17} + \dot{m}_{18})$	$h_{12a} = h_{14}$
	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	$\dot{m}_{15} = \dot{m}_{12a} - (\dot{m}_{13} + \dot{m}_{14} + \dot{m}_{16} + \dot{m}_{17} + \dot{m}_{18})$	$h_{12a} = h_{15}$
	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	$\dot{m}_{5b} + \dot{m}_{17} = \dot{m}_{5c}$	$\dot{m}_{5c} h_{5c} = \dot{m}_{5b} h_{5b} + \dot{m}_{17} h_{17}$
	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	$\dot{m}_{16} = \dot{m}_{12a} - (\dot{m}_{13} + \dot{m}_{14} + \dot{m}_{15} + \dot{m}_{17} + \dot{m}_{18})$	$h_{12a} = h_{16}$
	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	$\dot{m}_{7b} + \dot{m}_{18} = \dot{m}_{7c}$	$\dot{m}_{7c} h_{7c} = \dot{m}_{7b} h_{7b} + \dot{m}_{18} h_{18}$
	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	$\dot{m}_{11} = \dot{m}_{12}$	$\dot{m}_{12} h_{12} = \dot{m}_{11} h_{11} - \dot{Q}_{cond}$

The isentropic efficiency of compressors (η_s), depends on the compressor pressure ratios (R) and is given by equation 5.6 for all compressors handling NH_3 (Patel et. al., 2019).

$$\eta_s = -0.00097R^2 - 0.01026R + 0.83955 \tag{5.6}$$

The assumptions made for modelling the CRSs, mass and energy balance equations are similar to the case of the baseline system. There are additional components like the cascade condenser and pumped circulation. Equation 5.7 expresses heat balance for the cascade heat exchanger and equation 5.8 gives the mass flow rate in the secondary pumped loop through the *ice* evaporator.

$$\dot{Q}_{cc} = \dot{m}_{11}(h_{11} - h_{12}) = \dot{m}_{20}(h_{20} - h_{24}) \tag{5.7}$$

$$\dot{m}_3 = \frac{\dot{W}_p}{(h_{14} - h_{12})} \tag{5.8}$$

$\dot{W}_p$ is pump power in kW required to circulate the refrigerant in secondary circuit and is assumed to be 1% of the total refrigeration load of the ice evaporator. The compressors used in *ch* and *ice* in all three systems are the same as in the baseline system. In calculation of $\dot{W}_{net}$ in CRS2 and CRS3, $\dot{W}_p$ is considered as $\dot{W}_{ice}$ in equation 5.9. Net COP of all CRS systems are calculated using equation 5.10.

$$\dot{W}_{net,CRS} = \dot{W}_{ch} + \dot{W}_{ice} + \dot{W}_{cc} + \dot{W}_{cs} + \dot{W}_{pf} \tag{5.9}$$

$$COP_{CRS} = \frac{\dot{Q}_{ch} + \dot{Q}_{ice} + \dot{Q}_{cs} + \dot{Q}_{pf}}{\dot{W}_{net,CRS}} \tag{5.10}$$

COP_L, and COP_H are coefficients of performance for the LTC and HTC cycle, calculated to measure the performance of individual circuit for the LTC and HTC. Equation 5.11, 5.12 and 5.13 are used to calculate COP_{LTC}, and COP_{HTC} for CRS1, CRS2 and CRS3 respectively.

$$COP_L = \frac{\dot{Q}_{cs} + \dot{Q}_{pf}}{\dot{W}_{cs} + \dot{W}_{pf}}; \qquad COP_H = \frac{\dot{Q}_{ch} + \dot{Q}_{ice} + \dot{Q}_{cc}}{\dot{W}_{ch} + \dot{W}_{ice} + \dot{W}_{cc}} \tag{5.11}$$

$$COP_L = \frac{\dot{Q}_{cs} + \dot{Q}_{pf} + \dot{Q}_{ice}}{\dot{W}_{cs} + \dot{W}_{pf} + \dot{W}_p}; \qquad COP_H = \frac{\dot{Q}_{ch} + \dot{Q}_{cc}}{\dot{W}_{ch} + \dot{W}_{cc}} \tag{5.12}$$

$$COP_L = \frac{\dot{Q}_{cs} + \dot{Q}_{pf}}{\dot{W}_{cs} + \dot{W}_{pf}}; \qquad COP_H = \frac{\dot{Q}_{ch} + \dot{Q}_{ice} + \dot{Q}_{cc}}{\dot{W}_{ch} + \dot{W}_p + \dot{W}_{cc}} \tag{5.13}$$

In the analysis, equation 5.6 is used for isentropic efficiency of all the NH_3 compressors while for CO_2 compressors in LTC, an equation reported by Patel et al. (2019) is employed equation 5.14.

$$\eta_s = 0.00476R^2 - 0009238R + 0.89810 \tag{5.14}$$

AEC (kWh) for all the refrigerants with their respective system performance are calculated using equation 5.15 assuming year-round operation, where H is the number of hours for particular ambient temperature T.

$$AEC = \sum_{i=T_{min}}^{T_{max}} \frac{\dot{Q}_{ch}+\dot{Q}_{ice}+\dot{Q}_{cs}+\dot{Q}_{pf}}{COP_T} * H_T \tag{5.15}$$

The analysis considered an annual ambient temperature variation of Mumbai (11 to 38 ºC), and corresponding T_{cond} (15 to 45 ºC) as in (Mathur *et al.*, 2017).

In refrigeration, heat transfer occurs between the systems and surrounding at a finite temperature difference leading to process irreversibility. Cycle irreversibility causes degradation in system performance and thus these losses need to be evaluated. Although the performance of a refrigeration system can be evaluated based on the 1st law of thermodynamics but it does not give the information about the losses occurring during the processes. The 2nd law of thermodynamics provides the necessary details about the energy losses in the various processes at different components by exergy analysis. Exergy analysis helps to improve, optimize and evaluate the refrigeration systems by the well-established calculation methods. Component wise exergy destruction can help in analyzing complex systems and identify major exergy destruction components where improvements can result in enhanced system performance. Exergy destruction in all the system components are calculated using equations as given in Table 5.3; total exergy destruction is given by equation 5.16, while the second law efficiency (ϵ_{ex}) is calculated using equation 5.17.

Table 5.3: Abstract of equation used in the system modelling

Component/ Parameter	Exergy balance
Evaporator	$\dot{Ex}_{d,evp} = T_{amb}\left[\dot{m}_{ref}.(s_{out}-s_{in})_{evp} - \frac{\dot{Q}_{evp}}{T_{evp}}\right]$
Compressor	$\dot{Ex}_{d,comp} = T_{amb}.\dot{m}_{ref}.(s_{out}-s_{in})_{comp}$
Condenser	$\dot{Ex}_{d,cond} = T_{amb}\left[\dot{m}_{ref}.(s_{out}-s_{in})_{cond} + \frac{\dot{Q}_{cond}}{T_{cond}}\right]$
Expansion valve	$\dot{Ex}_{d,ev} = T_{amb}.\dot{m}_{ref}.(s_{out}-s_{in})_{ev}$
Refrigerant mixing	$\dot{Ex}_{d,mix} = \dot{m}_{ref,in,1}\left[(h_{in,1}-h_{out}) - T_{amb}.(s_{in,1}-s_{out})\right]$

	$+\dot{m}_{ref,in,2}[(h_{in,2} - h_{out}) - T_{amb}.(s_{in,2} - s_{out})]$
Cascade condenser	$\dot{Ex}_{d,cc} = T_{amb}.\left[\left(\dot{m}_{ref}.(s_{out} - s_{in})\right)_{LTC} - \left(\dot{m}_{ref}.(s_{out} - s_{in})\right)_{HTC}\right]$

$$\dot{Ex}_{d,total} = \dot{Ex}_{d,evp} + \dot{Ex}_{d,comp} + \dot{Ex}_{d,cond} + \dot{Ex}_{d,ev} + \dot{Ex}_{d,mix} + \dot{Ex}_{d,cc} \tag{5.16}$$

$$\epsilon_{ex} = (1 - \frac{\dot{Ex}_{d,total}}{\dot{W}_{net}}) \tag{5.17}$$

The system thermodynamic models are developed using *Engineering Equation Solver* (Klein, 2018). Initial values of thermodynamic parameters considered for simulation are listed in Table 5.4.

Table 5.4: Thermodynamic parameters considered for simulation

Parameters	Value (K)
Approach temperature of NH_3 condenser	5
Degree of suction superheat in *ch* evaporator	5
Degree of suction superheat in *ice* evaporator	5
Degree of suction superheat in *cs* evaporator	10
Degree of suction superheat in *pf* evaporator	15
Degree of superheat at evaporator outlets	0
Degree of subcooling after the condenser	0
The temperature difference between product temperature and refrigerant evaporation temperature	5
The temperature difference between condenser temperature and evaporator temperature of cascade condenser	5

5.4 Model Validation

Cascade condenser temperature (T_{MC}) optimization for maximum COP (Figure 5.5) and compared with the published study by Lee *et al.* (2006). For the comparison only one evaporator in LTC working at –50 ºC is considerd and condensing temperature of HTC is considered to be 35 ºC in absence of experimental data on multi-evaporator system with CO_2. Primary vertical axis presents COP_H and COP_L, while secondary vertical axis represents net COP of the system at various values of T_{MC}. The results exhibit maximum percentage deviation in COP_H, COP_L and net COP as 3.11%, 7.98% and 3.91% respectively. For the validation, the same operating parameters were considered in the current study as in the reference study.

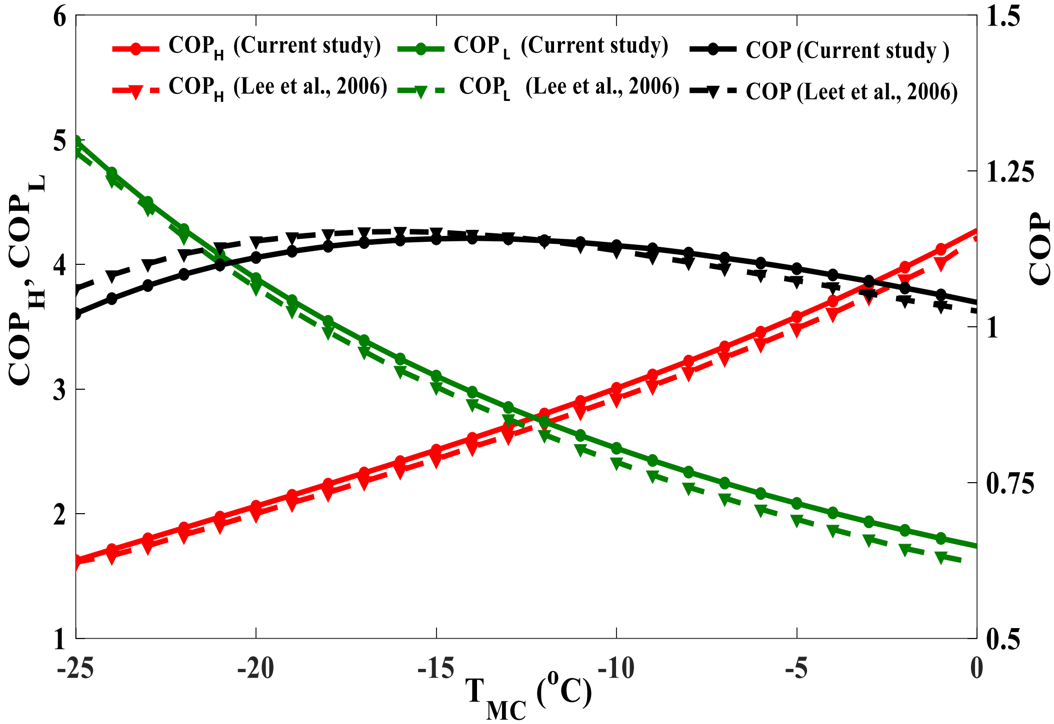


Figure 5.5: Validation of optimum T_{MC}

The thermodynamic models developed for the baseline system and all CRSs are validated using the published literature. Since the developed models are application specific and similar multi-evaporator CRSs have not been reported in open literature, a partial validation using the same operating conditions with single or double evaporators in CRS are compared. For the validation purpose, power consumption and COP of the baseline system is compared with the COP of actual system employed in the surimi processing plant located in Mumbai which showed maximum percentage deviation about 10%. Results from comparison of current study along with published theoritical and experimental studies are tabulated in Table 5.5. The maximum percentage deviation of cascade condenser temperature (T_{MC}) compared to theoritical and experimental results were found to be 1.6% and 0.7% respectively, while the maximum percentage deviation in COP were 1.64% and 16.6% respectively which exhibits good agreement between developed models and published data. Thus, the developed models may be considered as valid.

Table 5.5: Result comparison between current and published studies

Present study			Lee *et al.* (2006)				Dopazo and Fernández-Seara (2011)			
$T_{e,\ CO2}$ (ºC)	T_{MC} (ºC)	COP	T_{MC} (ºC)	Dev. (%)	COP	Dev. (%)	T_{MC} (ºC)	Dev. (%)	COP	Dev. (%)
–35	–8.6	1.63	–12.4	1.4	1.64	0.6	–9.3	0.3	1.58	3.0
–40	–10.5	1.45	–14.5	1.5	1.48	2.0	–10.3	0.0	1.42	2.0
–45	–12.4	1.29	–16.3	1.5	1.32	2.3	–13.3	0.4	1.15	10.8
–50	–14.1	1.14	–18.4	1.6	1.16	1.7	–15.8	0.7	0.95	16.6

5.5 Results and Discussion

5.5.1 Cascade condenser temperature optimization

In the design phase of a CRS, cascade condenser temperature T_{MC} plays an important role to achieve the maximum system COP. The optimum value of T_{MC} corresponding to maximum COP depends on the LTC evaporator temperature, HTC condenser temperature, the difference between cascade condenser and evaporator temperature, and refrigerant mass flow rates in LTC and HTC (Lee *et al.*, 2006). In this study all the evaporator temperatures and difference between cascade condenser and evaporator temperatures are constant, thus, optimum T_{MC} depends only on the HTC condenser temperature for a typical cooling load distribution. For a constant HTC condenser temperature, as T_{MC} increases, it reduces the pressure ratio of the HTC compressor and consequently its power consumption, but at the cost of an increase in pressure ratio of the LTC compressor and accordingly its power consumption. For a single evaporator CRS, Lee *et al.* (2006) and Dopazo and Fernández-Seara (2011) reported a trend of T_{MC} variation along with power consumption in the LTC and HTC. Figure 5.6 and Figure 5.7 show optimum T_{MC} for CRS1 and CRS3 for the cooling load as shown in Table 5.1 and operating conditions as given in Table 5.4. In CRS2, the *ice* evaporator is integrated with the cascade condenser, thus, the T_{MC} value of CRS2 will be the same as the evaporation temperature of the *ice* evaporator and T_{MC} optimization is not required for the same.

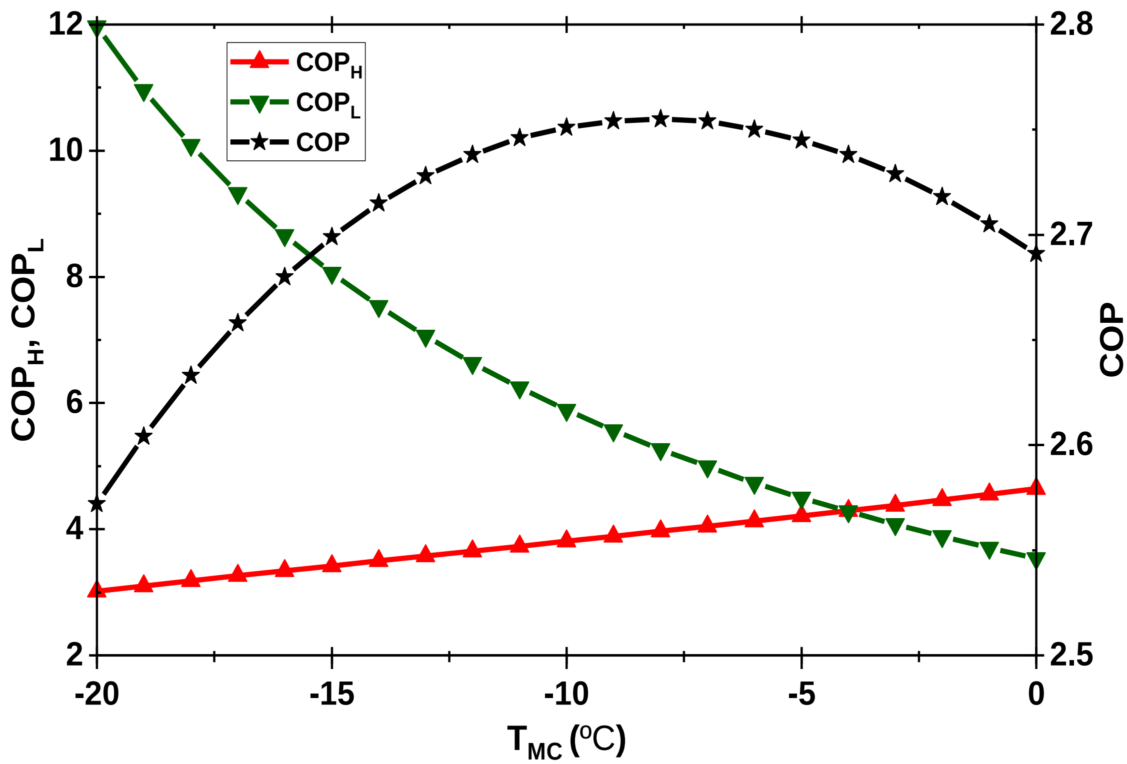


Figure 5.6: Performance variation with T_{MC} for CRS1

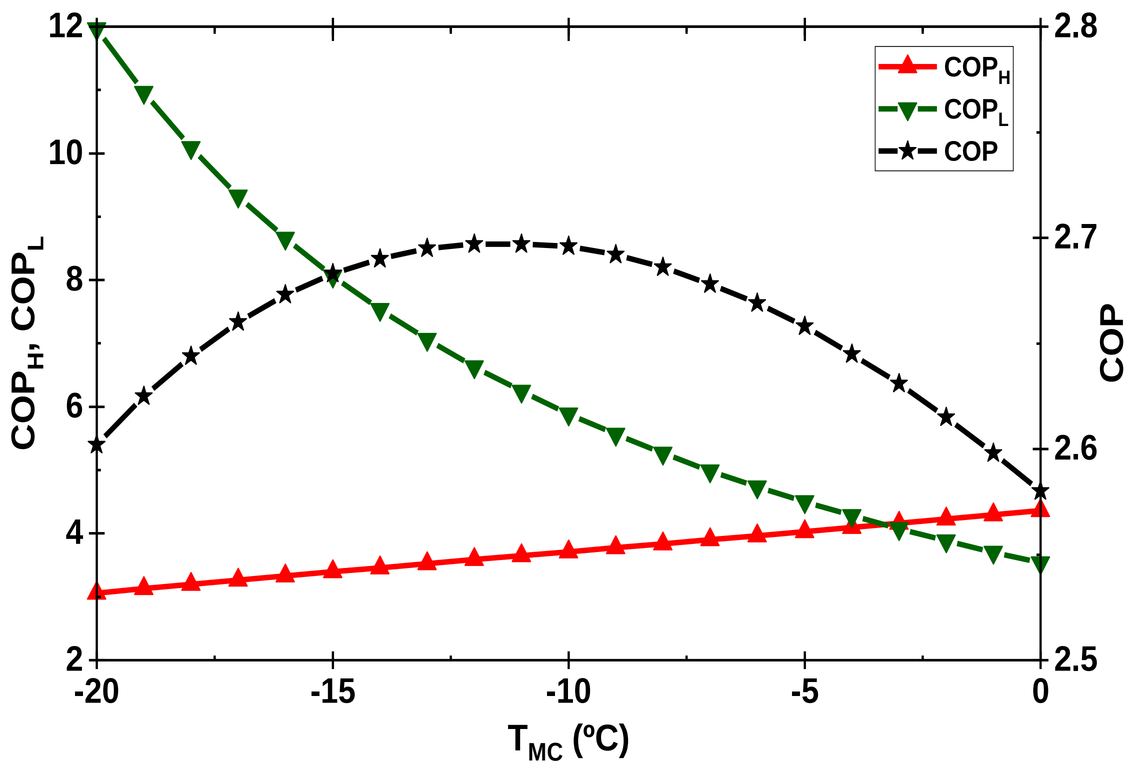
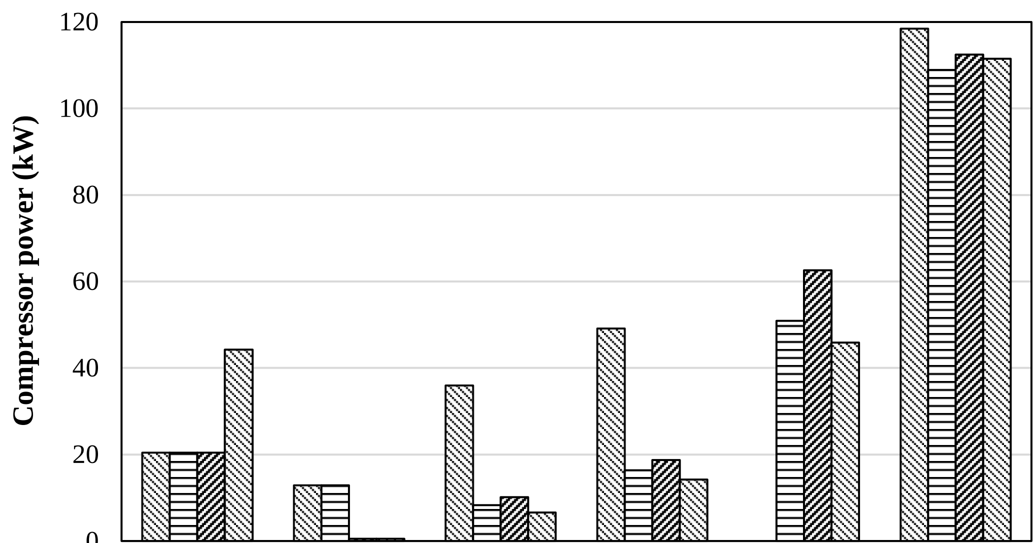


Figure 5.7: Performance variation with T_{MC} for CRS3

At optimum T_{MC}, CRS1 and CRS3 are found to have COP_{net} of 2.76, and 2.69 respectively and their performance is superior to the baseline NH_3 system. Optimum T_{MC} is different for CRS1 and CRS3 due to different performance behaviors of LTC and HTC (COP_L & COP_H) in both configurations.

5.5.2 Performance comparison with the baseline system

As a preliminary analysis, the compressor works are compared with the cooling loads of respective evaporators. It is observed that although the cooling load in the plate freezer is only 20% while it consumes about 42% of total power. Overall COP of the baseline system is computed to be 2.53. Due to part-load operation and other losses, the overall plant efficiency in the actual plant was much lower. Power consumption by the compressor of individual evaporator lines for all the three investigated refrigeration systems are compared in Figure 5.8 with that of the baseline system.



	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

Figure 5.8: Comparison of compressor work in baseline and proposed CRSs

It is observed that the total power consumed in CRS1 is the minimum while the same is the maximum for the baseline system. The decrease in compressor power consumption is mainly due to a reduction in power consumption in the CO_2 compressors in LTC. In CRS1, the combined power consumption in *pf*, *cs*, and *cc* compressors are less than in the baseline system for *pf* and *cs*, and the reduction achieved is about 8%. In CRS2 and CRS3, power consumed by the *ice* compressor is eliminated by using a pumped circulation in a secondary refrigeration loop. The secondary loop pump in both CRS2 and CRS3 are assumed to have a power consumption of about 1% of the refrigeration load in the evaporator. This, however, did not bring the intended benefit and the same was investigated. The reasons are found to be different for CRS2 and CRS3. In CRS2 the heat extracted from the *ice* evaporator is delivered to the *cc* evaporator, which increased the total load of the *cc* and the power consumed by the corresponding compressor. While in CRS3, the extracted NH_3 vapour from the receiver is mixed at the suction of the *ch* compressor leading to a higher mass flow rate and higher power consumption by the *ch* compressor for the same load combination.

Apart from Mumbai, there are surimi processing plants along the western coast of India where the ambient conditions vary. Other tropical coastal regions of Thailand, Vietnam, Indonesia, Malaysia, Myanmar, Pakistan, etc. have a prominent presence of surimi production. In order to check suitability of the proposed systems, performance of these systems are analyzed for the ambient conditions of these cities. The weekly averaged data of ambient dry bulb temperature (DBT), over a year, for Indian coastal cities of Veraval, Mumbai, Ratnagiri, Mangalore, and Vishakhapatnam having surimi processing plants, along with some tropical cities internationally having significant fishing activities such as Hanoi (Vietnam), Jakarta (Indonesia), Bangkok (Thailand) and Kuala Lumpur (Malaysia) are studied; the ambient temperature range is 20-40 ºC.

Performance of the three CRSs is studied within the ambient temperature range of 20 to 40 ºC implying the range of T_{cond} as 25 to 45 ºC. Figure 5.9 shows the variation in optimum T_{MC} with an increase in T_{cond} and Figure 5.10 shows the net COP variation of all four investigated systems corresponding to their T_{cond}. The trend of COPs, for all the investigated systems, shows a steady decrease with an increase in T_{cond}. For CRS1, and CRS3, the optimum temperature T_{MC} varies within range –12 ºC to –4 ºC, and –14.5 ºC to –8 ºC respectively while it remains constant for CRS2 at –5 ºC. The minimum percentage improvement in COP for CRS1, CRS2 & CRS3 compared to the baseline system are 8.7%, 4.8% and 6.3% respectively while the maximum percentage improvement for the same are 11.5%, 8.5% and 9.2%. It is also observed that CRS1 performs the best for the full range of T_{cond}. It is therefore concluded that CO_2-NH_3 CRS particularly CRS1 configuration, can be gainfully employed in the surimi processing industry across the tropical warm climate.

In general, the prominent seafood processing industries such as shrimp/prawn, fish fillets, and whole frozen fish have refrigeration needs for chilled water, ice, blast freezer, and refrigerated storage, therefore, the evaporator temperatures are similar to that of surimi processing although the refrigeration load ratios are quite different. Two shrimp processing plants located near Kolkata (eastern coast of India) and Kochi (Southwestern coast of India) were surveyed and the same data is obtained from published literature (Ates *et al.*, 2017) to conduct the following study. Table 5.6 presents the percentage fraction of refrigeration demands at various evaporation temperatures. Performance of the three CRS systems are compared for applications in shrimp/prawn and fish fillet processing plants and are presented in Figure 5.11and Figure 5.12, respectively. For both the applications, the CRS3 yielded maximum COP for the entire range of T_{cond} subjected to the corresponding refrigeration load and optimum value of $T_{MC.}$ The percentage improvement in COP with CRS3 in fish fillet processing is found to vary from a minimum of 9.6% to a maximum of 16.5% over the baseline NH_3 system, while the percentage improvement in shrimp/prawn processing is observed as a minimum of 17.6% to maximum 20.3%.

Table 5.6: Percentage cooling demands for various seafood applications

Evaporators	Evaporation temp (°C)	Surimi	Prawns/shrimps	Fillets
ch	2	38.4 %	10.0 %	10.0 %
ice	–5	18.3 %	35.0 %	12.3 %
pf	–40	20.0 %	43.0 %	10.0 %
cs	–25	23.3 %	12.0 %	66.7 %

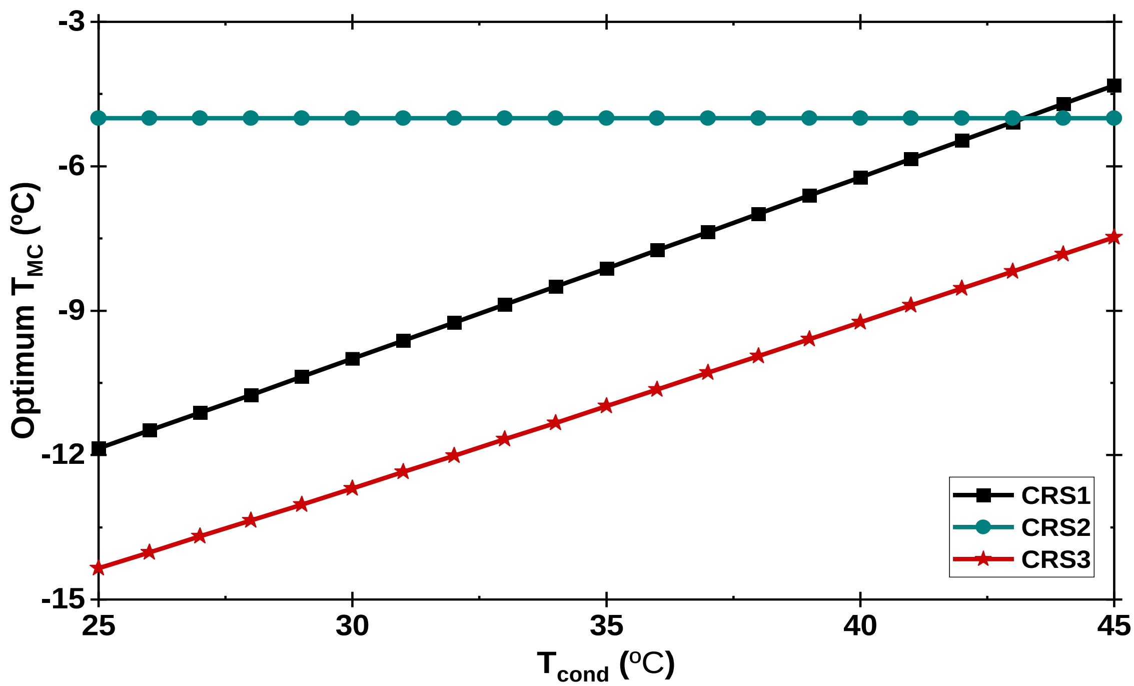


Figure 5.9: T_{MC} optimization

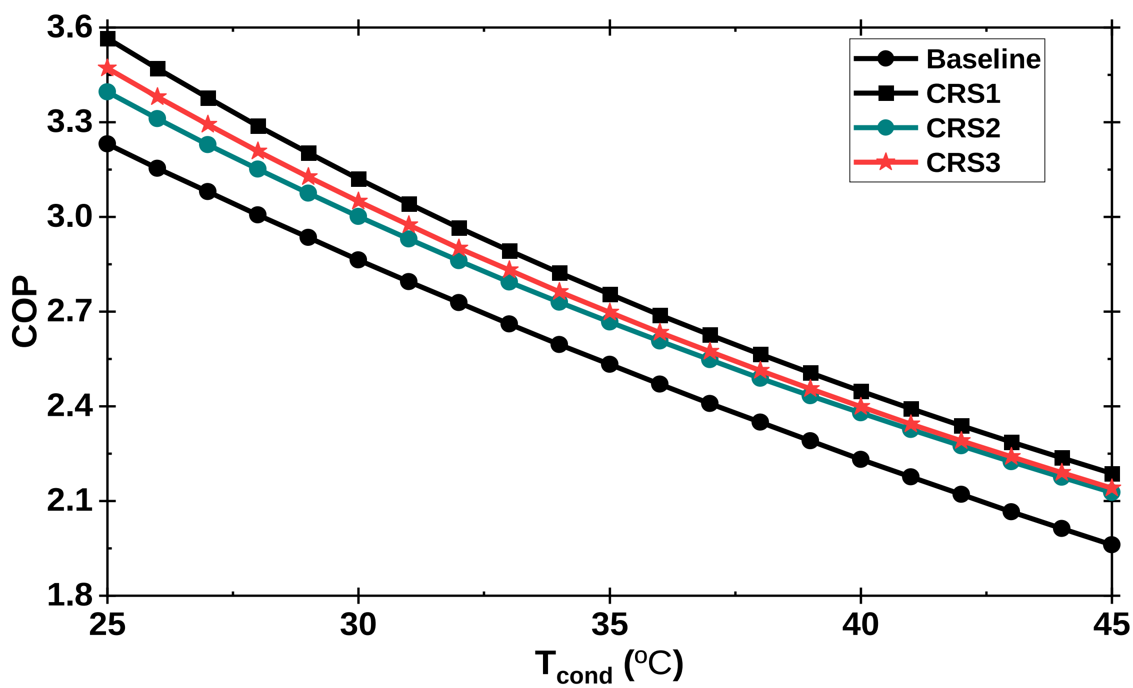


Figure 5.10: Performance comparison for surimi plant

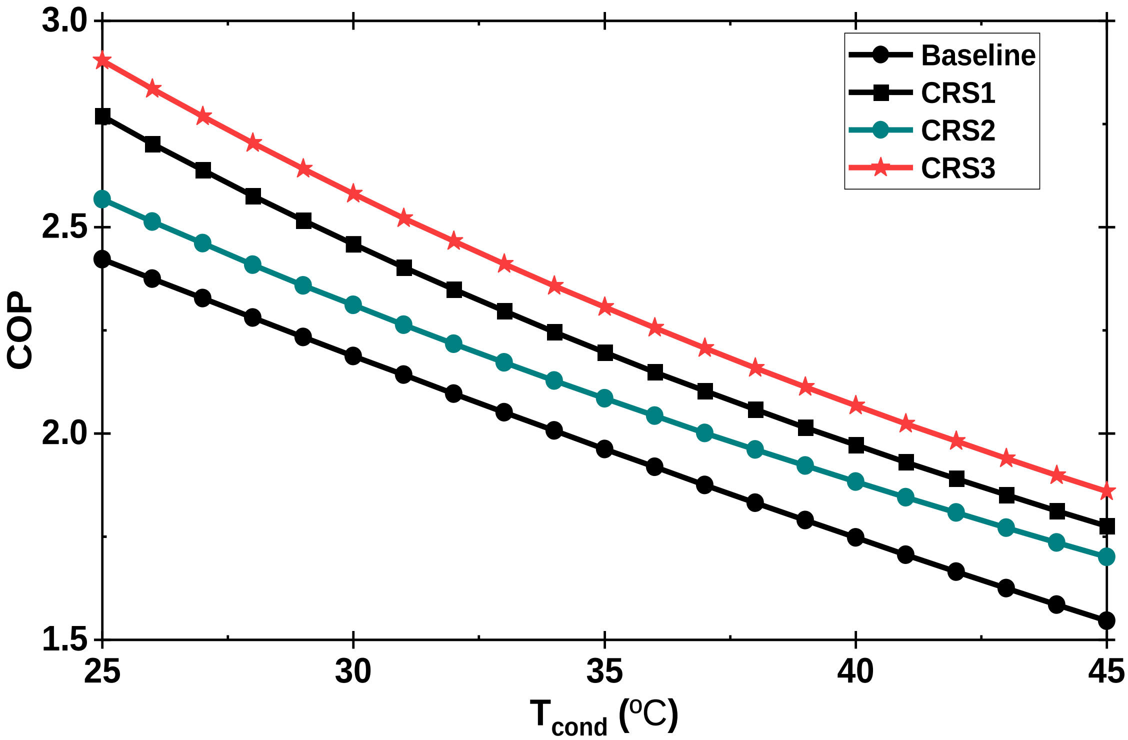


Figure 5.11: Performance comparison for shrimp/prawn plant

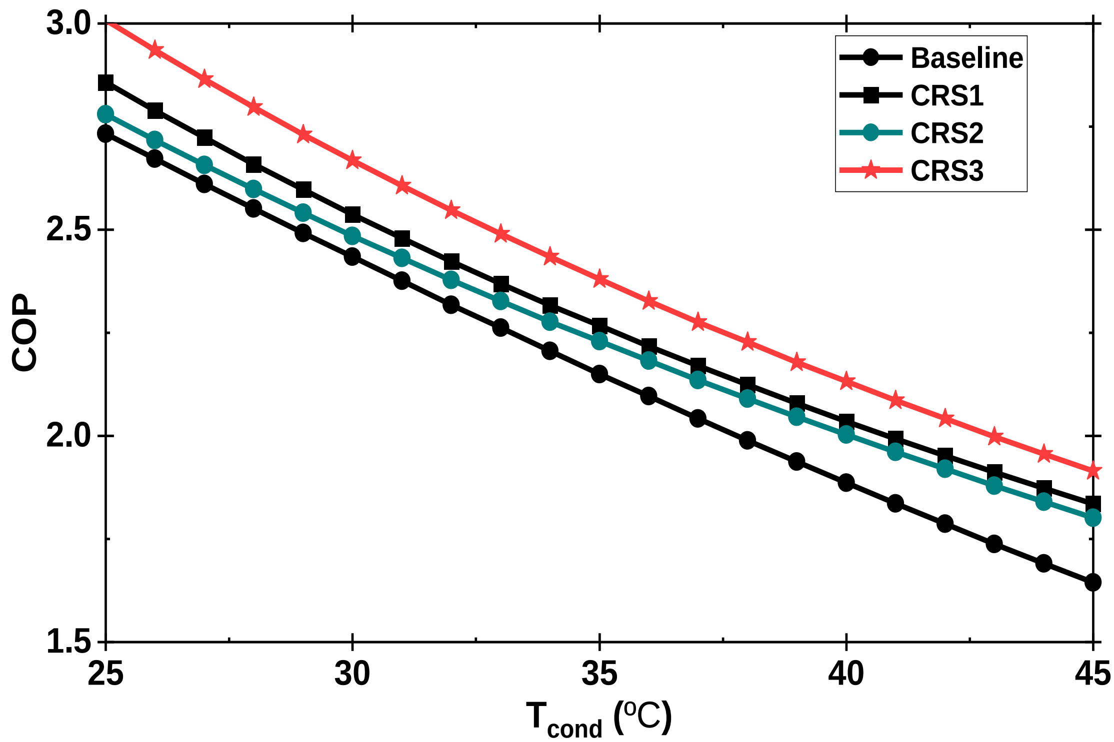


Figure 5.12: Performance comparison for fish fillet plant

5.5.3 Annual energy consumption comparison

The annual energy consumption (AEC) of the refrigeration system is computed by including the weather bin data instead of using the annual mean temperature as suggested by Yau *et al.* (2020). Temperature frequency distribution of surimi-producing coastal cities in the tropical region shows that the most frequent temperature bin varies from 12-40 ºC. Since the temperature bin variation of these cities is almost the same thus, the percentage reduction in annual energy consumption is also expected to be similar. For the surimi production plant, the percentage reduction in annual energy consumption by utilizing the CRS1, CRS2, and CRS3 compared to the baseline system are found to be about 8.3%, 5.3%, and 6.2% respectively for Mumbai (India) climatic conditions as shown in Figure 5.13. For the fish fillets and shrimp/prawns production plant, percentage reduction in annual energy consumption by the CRS1, CRS2 and CRS3 are 4.9%, 3.4% & 7.5% and 11.1%, 6.3% & 13.2% respectively as shown in Figure 5.13.

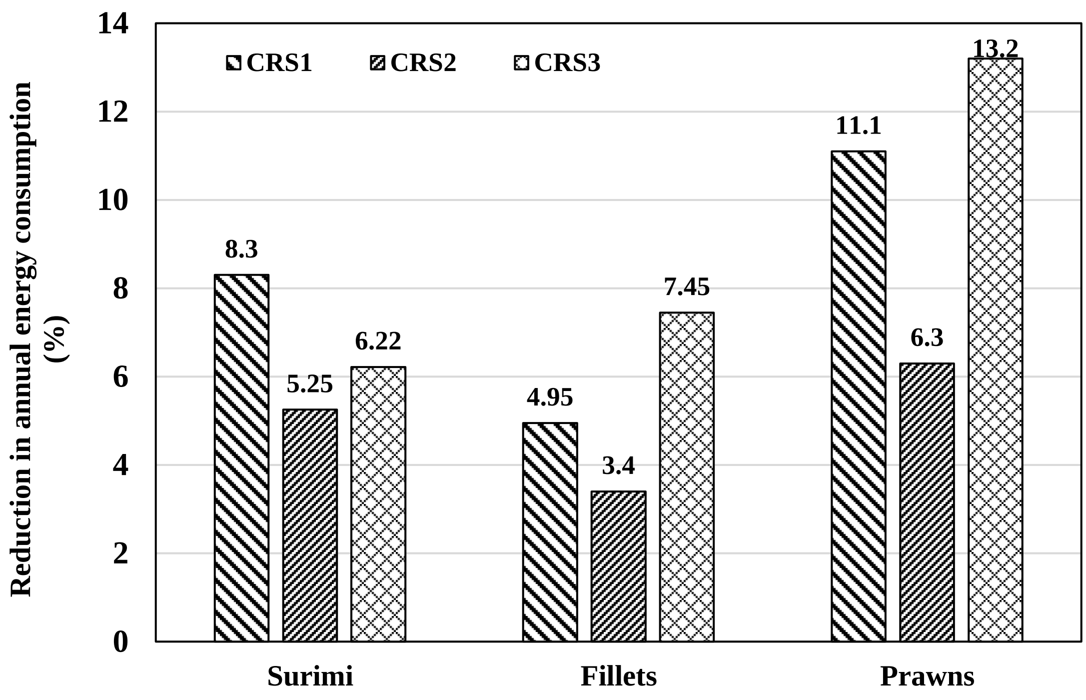


Figure 5.13: Reduction in AEC for different seafood plants

5.5.4 Compression ratio and compressor discharge temperature

Compression ratio and discharge temperature of compressors are two prominent parameters to decide feasibility of the system for an application. High value of compression ratio leads to lower isentropic efficiency of the compressor thus lowering the actual COP from the theoritically calculated value. While high discharge temperatures are known to enhance wear of rings, piston, valve, cylinders etc. and also cause degradation of lubrication oil (Stewart, 2018). Generally the permissible discharge temperature is below 135 ºC. Figure 5.14 presents compression ratio of compressors associated with *ch*, *ice*, *cc*, *cs*, & *pf* evaporators for all the investigated configurations while Figure 5.15 shows the compressors discharge temperature for the same at condenser temperature 35 ºC. Highest compression ratio is observed for *pf* compressor in the baseline system and *cc* compressor for all the CRSs. The discharge temperature is also observed higher for all the compressors with the baseline system and have lower values in all the CRSs.

It is, therefore, concluded that all the three proposed CO_2-NH_3 CRSs perform better than an NH_3 baseline system in meeting refrigeration demands of three common seafood processing industries. The CRS1 performs better when the largest cooling demand is at higher evaporation temperatures (0 °C), but the CRS3 works better when the largest cooling demand is at low evaporation temperatures (below −25 °C). The decision for the selection of a particular CRS configuration may also depend upon fixed cost. A more in-depth study of the effect of various factors on the system performance is required considering performance stability, fixed and variable cost, the influence of local policy that incentivizes or penalize use of various refrigerant, etc.

5.5.5 Exergetic performance comparison

After performance comparison, the CRS1 is found to be more efficient for surimi processing industry and thus exergy analysis is only compared between CRS and baseline system for surimi industry. The total exergy destruction and second law efficiency variation with condenser temperature are presented in Figure 5.16 and Figure 5.17. The total exergy destruction in the CRS1 is lower than the baseline for the entire range of operation. For example, at condenser temperature of 25 °C (ambient temperature 20 °C), the total exergy destruction in the CRS1 is 41.15 kW which is 14.5% lower than the baseline system. The CRS1 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.

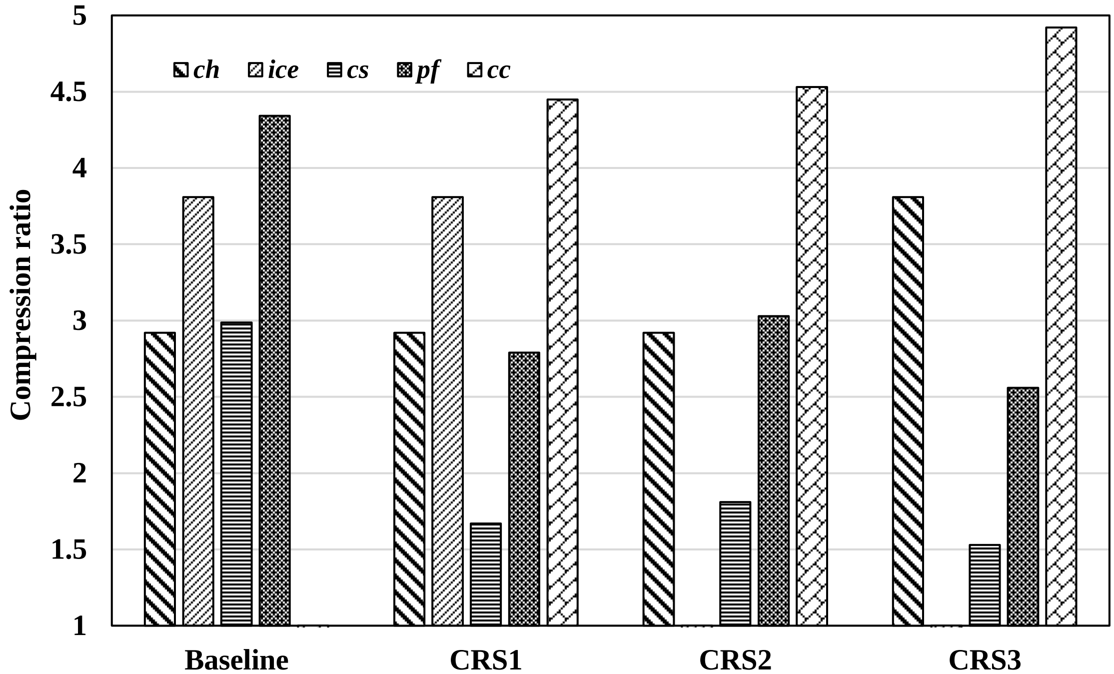


Figure 5.14: Variation of compression ratio

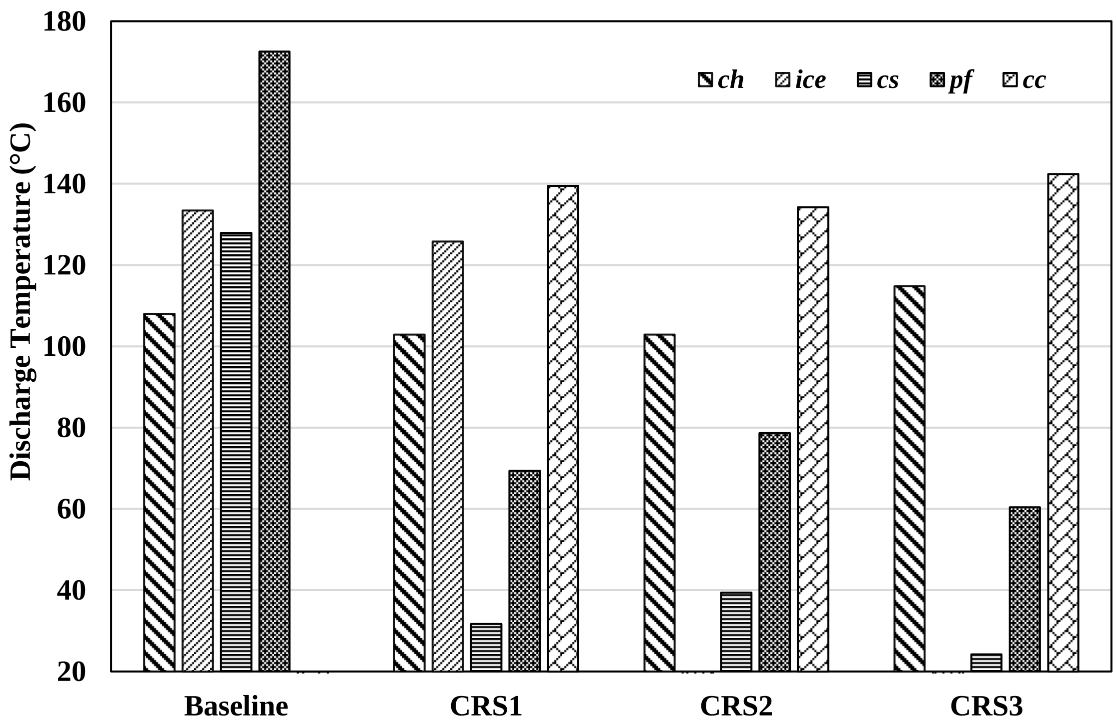


Figure 5.15: Variation of compressor discharge temperature

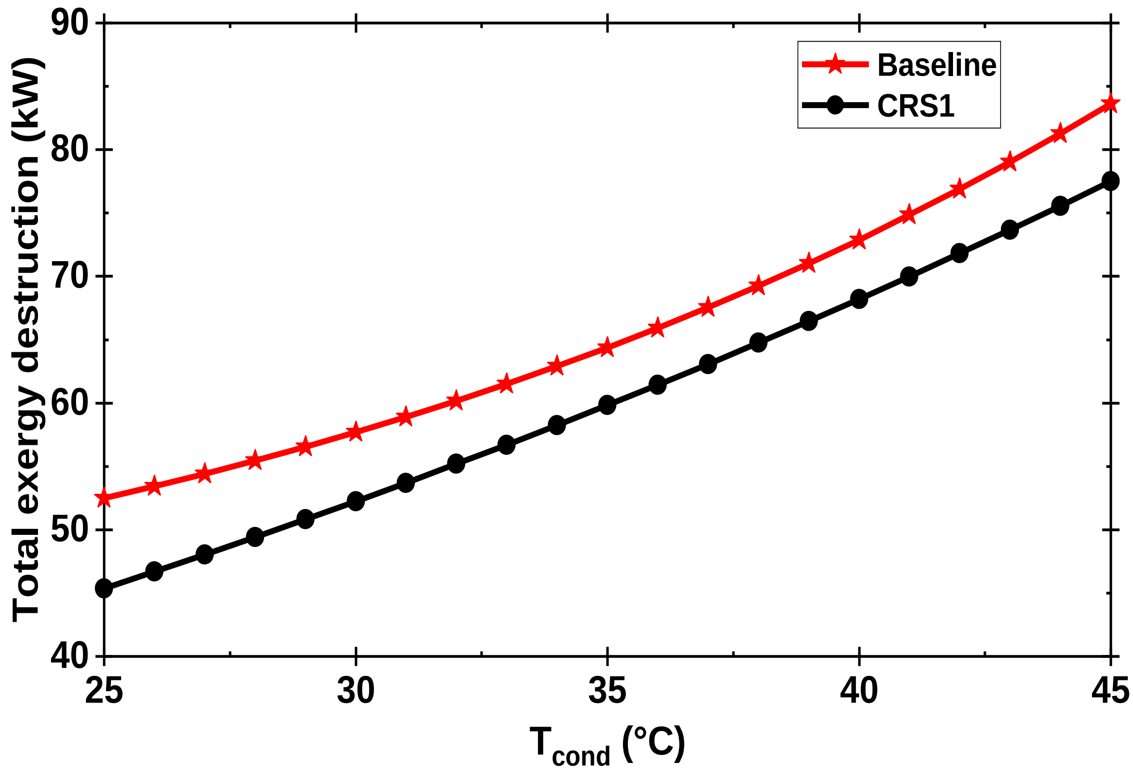


Figure 5.16: Total exergy destruction

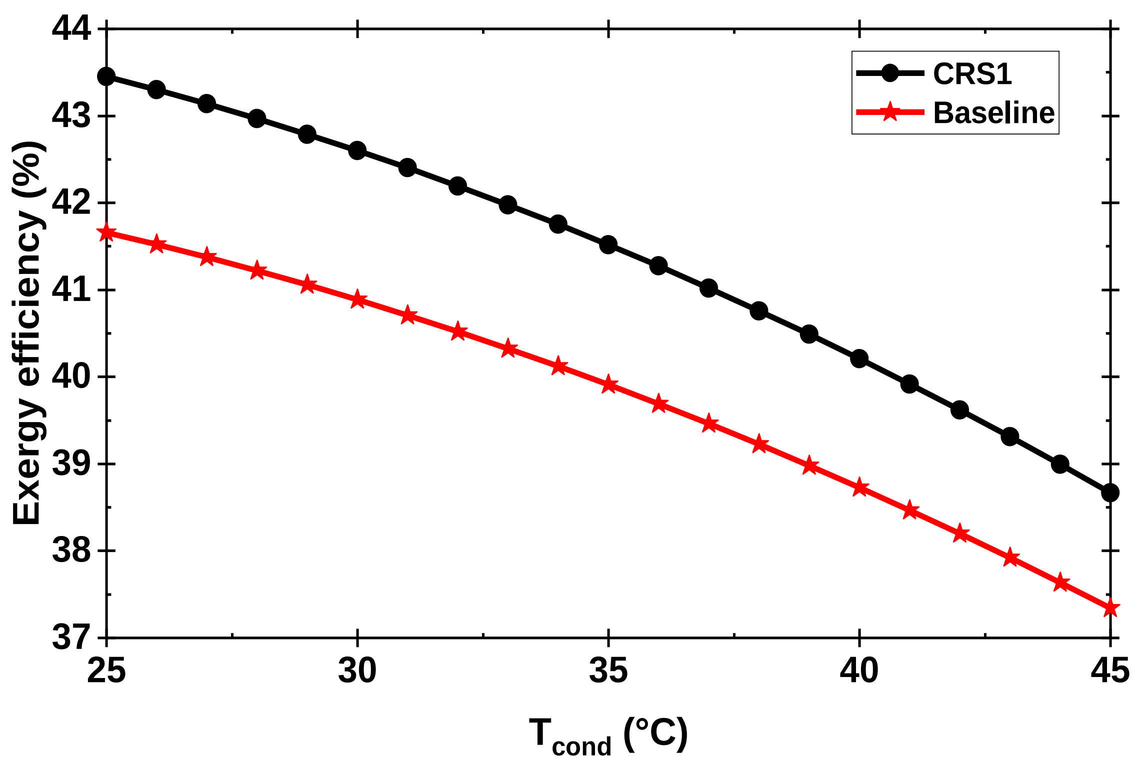


Figure 5.17: Exergy efficiency comparison

Component wise exergy destruction is computed to identify the components for improvement. Percentage exergy destruction in major components of both the systems are presented in Figure 5.18. 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 CRS1, 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 CRS1.

In summary, the benefits in the proposed CRS1 are increase in COP leading to reduction in annual energy consumption, reduction in the number of compressors, reduction in compressor discharge temperature, reduction in total NH_3 charge, isolation of food from proximity to NH_3 in the plate or blast freezers, and cold storage.

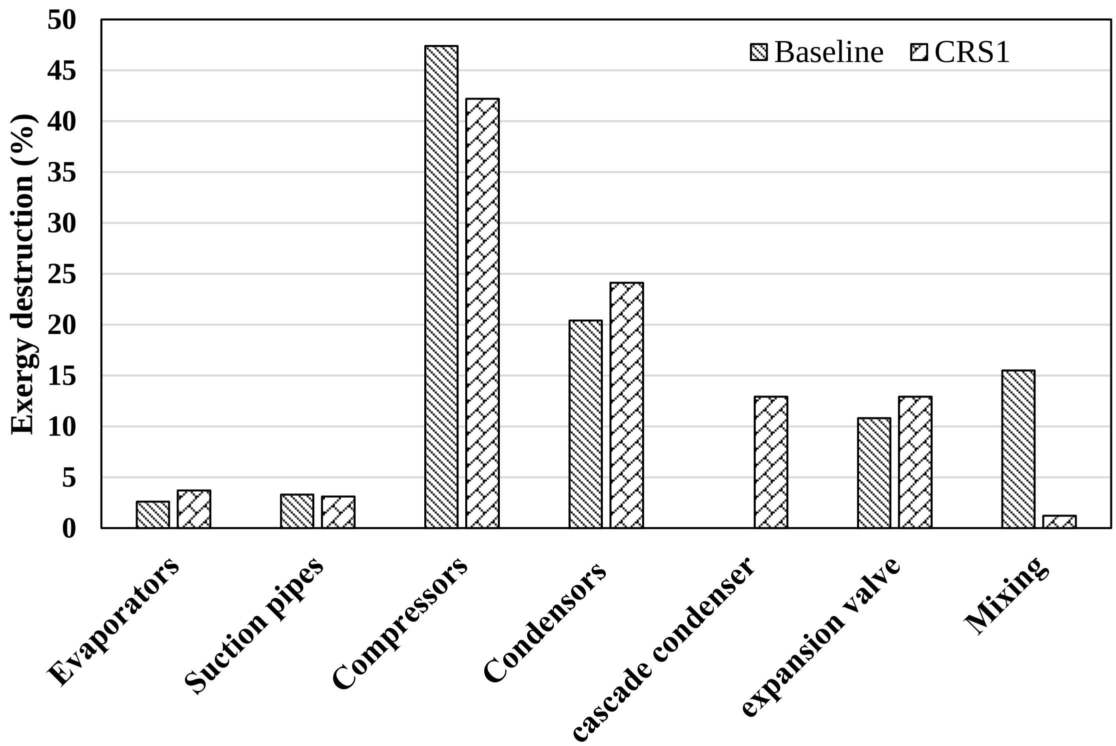


Figure 5.18: Component wise percentage exergy destruction

5.6 Conclusions

Three novel CO_2-NH_3 CRSs; each having four evaporators arranged across the T_{MC} have been proposed for supporting typical cooling demands in various seafood processing plant operating in tropical climate. The key differences among these three systems are the number of compressors used, their distribution of evaporator load above and below the T_{MC}, and the utilization of pumped circulation of refrigerants. The developed system models are validated against the published literature for a single evaporator. CRS system simulation results are compared with that of a simulated conventional NH_3 system for a range of evaporator loads and ambient temperature. The CRS1 configuration exhibited superior performance for the entire range of condensing temperature for usual refrigeration load conditions in a surimi processing plant that typically has a higher cooling load at high temperature evaporator supporting water chilling. The maximum COP advantage using CRS1 achieved is about 11.5% for the surimi processing plant along with an 8.3% reduction in annual average consumption for Mumbai climatic conditions compared to the baseline system. While the CRS3 configuration performed the best in meeting the refrigeration demands in the shrimp/prawn and fillet industry where significant cooling demand is at a lower temperature. The maximum percentage improvement in fish fillets and shrimp/prawn industry for CRS3 is 16.5% and 20.3% along with a 7.5% and 13.2% reduction in annual average consumption respectively over the baseline system. With increase in condenser temperature, the total exergy destruction in both the systems increase while exergy efficiency decreases. The CRS shows about 14% less exergy destruction compared to the baseline system. Further, exergy destruction was compared between the baseline and CRS1 system. In both systems, the compressors contribute to the largest exergy destruction, with an average value of 45%; the evaporators contribute the least exergy destruction, with a modest average value of 4%. This study suggests possible gainful implementation of the natural refrigerant combination of CO_2-NH_3 CRS for a wide range of seafood processing industries in the tropical region. Other benefits in the proposed CRSs are reduction in the number of compressors, possible reduction in pressure ratio in compression, reduction in total NH_3 charge, and isolation of NH_3 from food in cold storage as well as in plate freezers.

CHAPTER 6 Energetic, Environmental and Economic Comparison and Performance Enhancement using IHX

6.1 Abstract

In the seafood industry in India, commonly, a multi-evaporator multi-stage refrigeration system with refrigerants such as R22, R404A, or NH_3 are used. However, R22, and R404A have harmful effects on the environment due to their high global warming potential. This study analyzed and compared performance of an all-natural multi-evaporator CO_2-NH_3 CRS proposed in Chapter 5 with a conventional R22, R404A, and NH_3 systems. The comparison is based on energy, environmental and economic parameters. For analysis, data of refrigeration demands in a surimi processing and storage plant located in Mumbai are utilized. The study revealed that the CRS has higher COP and the lowest annual energy consumption followed by the conventional NH_3 system. The CRS exhibited 6.2%, 12.3% and 3.2% less energy consumption compared to R22, R404A, and NH_3 systems, respectively. Similarly, the CRS also showed the lowest total equivalent warming impact which is 26.8%, 44.3% and 3.2% less compared to R22, R404A, and NH_3 systems, respectively. Furthermore, CO_2-NH_3 CRS also performed better economically in terms of annual cost rate and total life cycle cost estimate.

Further, to enhance the performance of the CRS, integration of an internal heat exchanger (IHX) is proposed. The IHX in the sub-critical CO_2 refrigeration system produces required sub-cooling at the exit of the condenser, increasing the refrigeration effect, and thus the overall performance improves. The cascade temperature is optimized and the performance of the system is found to be superior to the conventional NH_3 system for various operating conditions. The effects of refrigeration loads, evaporation temperatures, and ambient temperature on the performance of the proposed cascade system are also investigated.

6.2 Introduction

The concerns regarding high ODP and GWP from various synthetic refrigerants are to be mitigated by regulating their usage in a time bound manner under the Montreal and Kyoto Protocols and encourage use of environmental friendly refrigerants. Based on a survey, Kumar *et al.* (2018) reported that R22, R134a, R404A and NH_3 are the most commonly used refrigerants by Indian refrigeration industry, and they also estimated the percentage share of these refrigerants across various applications. Among these refrigerants, the first three have high GWP values of 1810, 1430, and 3943, respectively, and thus are under regulation to be phased out from 2028 in India (Chatterjee, 2022). As an immediate replacement of R404A Yang *et al.* (2021) reviewed a few possible alternatives such as R454A, R455A, R457A, R459B, R468A and L40. These refrigerants have almost similar thermophysical properties, and retrofitting may be possible but are not regarded as permanent substitution due to their higher GWP value. Therefore, the leading candidates, as an alternative, with very low GWP are the natural refrigerants like NH_3, CO_2, hydrocarbons and hydrofluoroolefins (HFOs). In this study, the CRS1 that is conceptualized in *Chapter 5* and found to have better thermodynamic performance compared to other systems is used for analysis and is termed as CRS hereafter to make it simple.

Over the years, advancements in the design of refrigeration systems and components have led to enhancement in their efficiency, safety and durability. There are significant environmental and economic benefits of using suitable low GWP alternatives and to explore possible design changes to meet the specific cooling demand. To increase the system energetic efficiency, use of IHX have been endorsed in literature (Liu *et al.*, 2021; Nguyen and Le, 2020). There are, however, two contradicting impacts attributable to the introduction of IHX. While it increases the specific refrigeration capacity in the evaporator, it also increases the compressor work due to increased specific suction volume of refrigerant.

Previous research has shown that the IHX improves overall performance when the working fluid has high heat capacity and only when operates within a range of evaporating and condensing temperatures (Domanski *et al.*, 1994). Klein *et al.* (2000) concluded that installing IHX in the NH_3 refrigeration system reduces the system performance irrespective of operating conditions. Zhang and Xu (2011) theoretically analyzed the effect of an IHX integration with the CO_2 system both in subcritical and transcritical operations. Llopis *et al.* (2015) experimentally validated the results obtained by Zhang and Xu for the subcritical CO_2 cycle and concluded that refrigeration capacity and COP were reduced by 3.5 % and 3.29 %, respectively, at –25 °C evaporation temperature. However, for a lower temperature of –40 °C in the evaporator, the system COP was observed to increase marginally by about 0.45%.

Purohit *et al.* (2018) experimentally investigated the energetic and exergetic performance of the CO_2 chiller system with IHX for extreme warm climatic conditions. It is therefore concluded that there is merit in using the IHX with CO_2 subcritical cycle to enhance performance.

However, there is limited study that reported the benefits of using IHX in CO_2-NH_3 CRS in multi-evaporator system for specific applications. A state-of-the-art multi-evaporator CO_2-NH_3 CRS was studied by Saini *et al.* (2021) for a seafood processing and storage application. However, no study has reported energetic, environmental and economic performance comparison between a multi-evaporator CO_2-NH_3 CRS and a conventional multi-evaporator two-stage VCR system with refrigerants R22, R404A or NH_3.

The novelty of the present study is to compare the performance of a multi-evaporator CO_2-NH_3 CRS and a conventional VCR system operating with R22, R404A or NH_3 in terms of energetic, environmental and economic parameters. The refrigeration systems are dimensioned to meet typical refrigeration demands of a surimi (seafood) processing and storage plant, for operating condition of Mumbai (India). Through thermodynamic modelling and numerical simulations, COP, annual energy consumption (AEC), and TEWI are computed and compared. By estimating various cost components such as initial capital cost, annual operating cost, and environmental cost due to greenhouse gas emissions, annual cost rate and life cycle cost of all the systems are evaluated in order to assess the economic viability of the proposed system.

Although many researchers have analyzed the CO_2-NH_3 refrigerant pair, no study is found in the open literature exploring use of IHX and gas-cooler in a multi-evaporator CO_2-NH_3 CRS meeting the refrigeration demands in surimi processing and storage plant. The possibility of improving the performance by incorporating an IHX and a gas-cooler in a CO_2-NH_3 CRS is also investigated in this study.

6.3 System Modelling

In this analysis, we considered the CO_2-NH_3 CRS and baseline systems analyzed in *Chapter 5* for the comparison; R22, R404A and NH_3 refrigerants are considered for the baseline system. Therefore, the exact thermodynamic models along with refrigeration demands and other operating conditions discussed in the *Chapter 5* are adopted for the comparison. For environmental and economic assessment and comparison of the systems the mathematical model is discussed in the following sub-sections.

6.3.1 Environmental model

Total equivalent warming impact ($TEWI$) is a significant parameter that considers sum of direct ($TEWI_{direct}$) and indirect ($TEWI_{indirect}$) equivalent CO_2 emissions from a refrigeration system. $TEWI_{direct}$ counts the emissions from direct leakage of refrigerants in to atmosphere. Refrigeration is an energy-intensive process and consumes energy from grids, $TEWI_{indirect}$ considers the emissions caused due to electricity generation. The TEWI is calculated using equations 6.1-6.3. Total refrigerant leakage ($M_{leakage}$) is computed from total refrigerant charge and refrigerant leakage rate. In this study, refrigerant charge is considered as 3 kg of refrigerant per kW of refrigeration load and annual refrigerant leakage is taken as 15% (Lata and Gupta, 2020). Recycling factor (α) is considered as 95% while GWP values for NH_3, CO_2, R22 & R404A are considered as 0, 1, 1810 and 3943 respectively. The electricity regional conversion factor (β) is taken as 0.9 kg CO_2 per kWh energy used, that depends upon the regional energy mix (Patel *et al.*, 2019).

$$TEWI = TEWI_{direct} + TEWI_{indirect} \tag{6.1}$$

$$TEWI_{direct} = \left(M_{leakage} * n + M_{charge}(1-\alpha)\right) GWP \tag{6.2}$$

$$TEWI_{indirect} = \beta * AEC * n \tag{6.3}$$

Simulation is carried out to analyze the performance of both the systems for the mentioned refrigeration demands and operating conditions. Performance of R22, R404A & NH_3 are computed using the two-stage VCR system and CO_2-NH_3 refrigerant pair using the CRS. Ambient conditions or condenser temperature is the only parameter that is varied considering the year-round operation of the system.

6.3.2 Economic model

An economic assessment of the analyzed refrigeration systems is presented here. Due to various reasons like industry type, user tendency, the nature of problem and the existence of many cost components, a standard economic model has not yet been established in the industrial sector. From literature it is found that several models have been used for economic analysis of refrigeration systems (Lata and Gupta, 2020; Mosaffa *et al.*, 2016; Roy and Mandal, 2020). Irrespective of the models all of these consider similar factors which affects the capital and operating cost of the system. In the industrial refrigeration systems, the initial capital cost of the system is dominated by the compressor and heat exchanger cost. The constructed economic model consists of an initial capital cost (*ICC*), annual operating cost (*AOC*), maintenance cost (*MC*) and environmental penalty cost due to greenhouse gas emissions. Based on these cost components, annual cost rate (*ACR*) and life cycle cost (*LCC*) are estimated for comparison among the various systems. The *ICC* utilized for the present study comprises of component cost, installation cost and some additional cost for piping, auxiliary instruments etc. System component costs are estimated based on the heat exchanger area (A) for heat exchangers, compressor power consumption ($\dot{W}$) for compressors, while refrigerant mass flow rate ($\dot{m}_{ref}$) is used for computing expansion valves and liquid receiver cost. Component costs are calculated using the cost function given in Table 6.1 (Cui *et al.*, 2019; Patel *et al.*, 2019; Roy and Mandal, 2020).

Table 6.1: Component cost function of the system

Components	Cost function (in US $)
CO_2 compressor	$17547.\dot{W}^{0.4488}$
NH_3, R22 or R404A compressor	$758.15.\dot{W}^{0.8768}$
Evaporator	$331.7.A^{0.85}$
Condenser	$331.7.A^{0.90}$
Cascade condenser	$1874.4.A^{0.9}$
Expansion valve	$817.\dot{m}_{ref}$
Liquid receiver	$2000.\dot{m}_{ref}{}^{0.67}$

The heat transfer area of heat exchangers is calculated using equation 6.4.

$$A = \frac{Q}{U.LMTD} \tag{6.4}$$

The values of U for evaporators, condenser and cascade condenser are considered as 30, 40 and 1000 W m^{-2} K^{-1} respectively (Mosaffa *et al.*, 2016). The installation, piping and additional instrumentation costs (C_{add}) are assumed as 15% of the total component cost. Initial capital cost (*ICC*) is computed by adding all the component costs and the additional cost as given in equation 6.5 (Roy and Mandal, 2020b).

Initial capital cost: $ICC = \sum_i^n C_{comp,i} + C_{add}$ (6.5)

Annual operating cost (*AOC*) is calculated using equation 6.6, where annual energy consumption (*AEC*) is calculated for the temperature bin hour of Mumbai and local commercial electricity price is considered as 0.1 \$ kWh^{-1} (Pandey, 2019).

Annual operating cost: *AOC* = *AEC* × electricity price per unit (6.6)

The rate of penalty cost due to GHG emissions is calculated as using equation 6.7, where cost of CO_2 avoided (C_{CO2}) is assumed as 0.09 \$ kg^{-1} of CO_2 emission (Roy and Mandal, 2020b).

Environmental penalty cost: $C_{env} = TEWI \times C_{CO2}$ (6.7)

In order to recover the investment over a period of years, assessment of annual cost rate of the system is computed using equation 6.8. Maintenance cost (MC) is considered as 5% of *ICC* and capital recovery factor (CRF) is calculated using equation 6.9 discount rate (*i*) is considered as 10%, and system service lifetime (n) is assumed as 15 years (Roy and Mandal, 2020).

Annual cost rate: $ACR = (ICC + TEWI_{direct} \times C_{CO2}).CRF + \mathrm{AOC} + \mathrm{MC} + (TEWI_{direct} \times C_{CO2})$ (6.8)

$$CRF = \frac{i(i+1)^n}{(i+1)^n - 1} \tag{6.9}$$

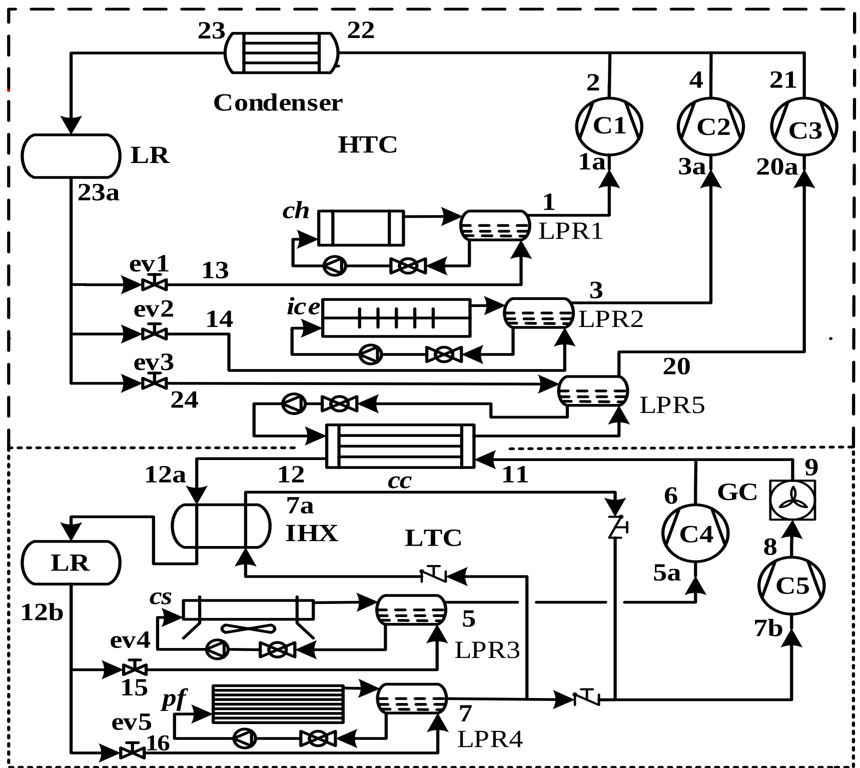
Life cycle cost (*LCC*) is calculated using equation 6.10, where the system service lifetime (n) is considered as 15 years.

Life cycle cost: $LCC = ICC + (AOC + MC).n + C_{env}$ (6.10)

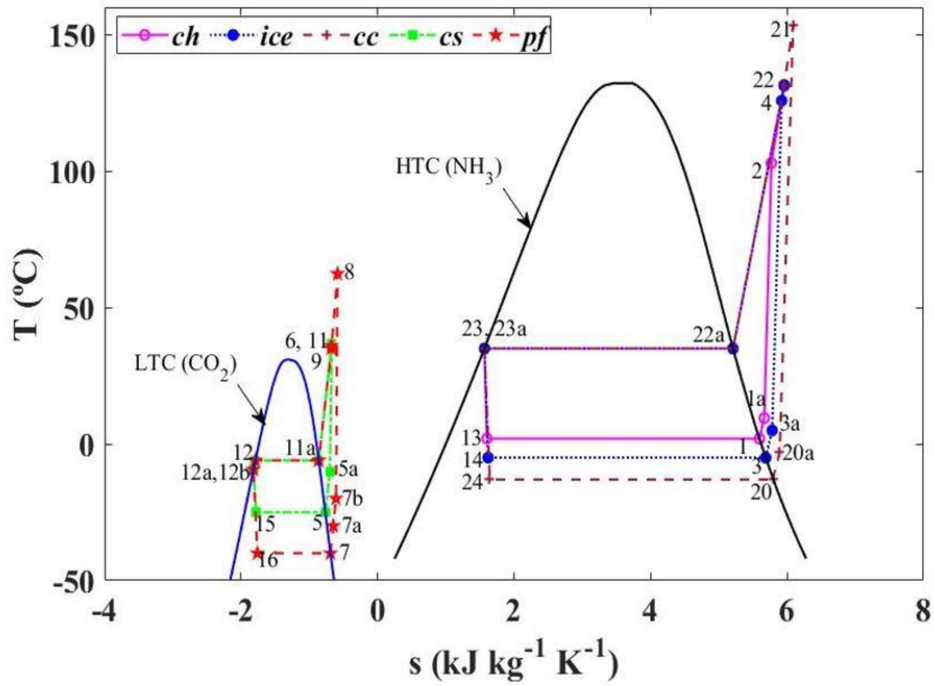
6.3.3 IHX and gas-cooler integration with CRS

The proposed integration of IHX with CRS and its T-s diagram are shown in Figure 6.1. The system consists of two circuits- LTC with CO_2 as the refrigerant, and HTC with NH_3 as the refrigerant. Preliminary computation shows that the temperature of CO_2 at the exit of *pf* compressor is higher than ambient, therefore, an additional gas-cooler is proposed prior to the cascade condenser. This will reduce the amount of heat supplied to the high-temperature circuit, decreasing the compressor work of the high-temperature circuit.

The condenser fan consumes a modest amount of energy to reject the heat to the ambient, which is not considered for the performance calculation in previous sections assuming both the systems reject approximately same amount of heat, thus it does not make significant difference. However, while integrating the gascooler in LTC, it removes some amount of heat before it goes to HTC and thus reduces heat rejection by HTC condenser. Although, the LTC gascooler fan also consumes energy and to include the effect of fan power consumption is added in COP calculation.



a)



b)

Figure 6.1: a) Piping diagram, b) P-h diagram of CRS integrated with IHX & gas-cooler

Fan power consumption is estimated as 3 % of heat rejection by gascooler or condenser (Purohit, Sharma, *et al.*, 2018). Heat rejection by the gascooler to the ambient ($\dot{Q}_{GC}$) and the remaining heat transfer to the HTC from the LTC ($\dot{Q}_{HTC}$) are computed using equations 6.11 and 6.12. The effectiveness (ε) for IHX is defined as a ratio of actual heat transfer to the maximum possible heat transfer. It depends on the mass flow rate of the refrigerant at outlets of condenser and evaporators and enthalpy difference between these points; it is expressed in equation 6.13. COP of the baseline NH_3 system and the CRS integrated with the IHX and gascooler are computed using equation 6.14 and equation 6.15. Percentage improvement in COP from baseline to the proposed CRS is calculated using the equation 6.16.

$$\dot{Q}_{GC} = \dot{m}_7 * (h_8 - h_9) \tag{6.11}$$

$$\dot{Q}_{HTC} = \dot{m}_{cc}(h_{20} - h_{24}) = \dot{m}_{11}(h_{11} - h_{12}) \tag{6.12}$$

$$\varepsilon = \frac{C_c(T_{7a} - T_7)}{C_{min}(T_{12} - T_7)} \text{ or } \varepsilon = \frac{C_h(T_{12a} - T_{12})}{C_{min}(T_{12} - T_7)} \tag{6.13}$$

$$COP_{Baseline} = \frac{\dot{Q}_{ch} + \dot{Q}_{ice} + \dot{Q}_{cs} + \dot{Q}_{pf}}{\dot{W}_{ch} + \dot{W}_{ice} + \dot{W}_{cs} + \dot{W}_{pf} + \dot{W}_{fan}} \tag{6.14}$$

$$COP_{CRS} = \frac{\dot{Q}_{ch} + \dot{Q}_{ice} + \dot{Q}_{cs} + \dot{Q}_{pf}}{\dot{W}_{ch} + \dot{W}_{ice} + \dot{W}_{cs} + \dot{W}_{pf} + \dot{W}_{fan}} \tag{6.15}$$

$$\delta = \frac{(COP_{CRS} - COP_{Baseline})}{COP_{CRS}} * 100 \qquad (6.16)$$

6.4 Results and Discussion

Baseline system with all three refrigerants R22, R404A and NH_3 and the CO_2-NH_3 CRS are compared using parameters such as COP and AEC for the energy performance, TEWI for environmental performance and ACR, LCC and environmental cost for economic viability.

6.4.1 Energetic performance

For the CRS, the COP is calculated at the optimized T_{MC} of the system for all ambient conditions as shown in *Chapter 5*. Figure 6.2 shows the variation of COP with increase in condenser temperature for all the three refrigerants in baseline system and for the CRS. An increase in ambient temperature or condenser temperature leads to increase in the compression ratio, the isentropic efficiency of the compressor is reduced. Among all the analyzed systems, CO_2-NH_3 CRS showed the maximum COP at all condenser temperatures. With increase in condenser temperature from 15 ºC to 45 ºC, the COP of the CRS decreased by 59.2%, which was minimum among all the analyzed systems followed by 62.3% in NH_3 baseline system. It shows that the decrease in COP for baseline system with all considered refrigerants are more pronounced at higher ambient temperature.

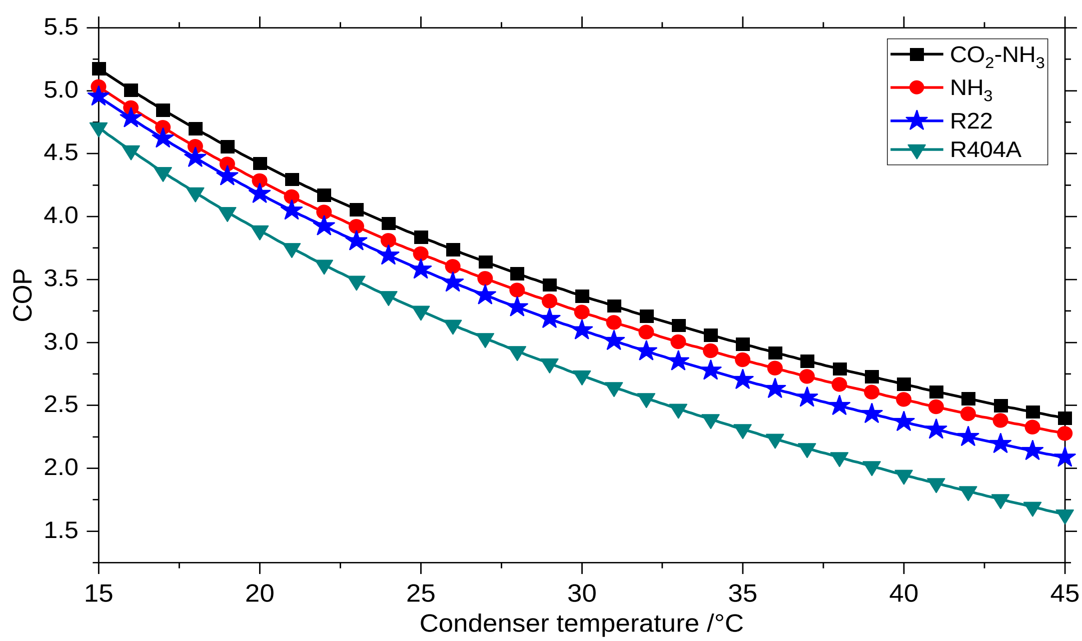


Figure 6.2: Influence of the condenser temperature on COP

In seafood applications, especially in the blast freezer room, or in plate freezers, the temperature is preferred to be below −40 ºC. Decrease in the temperature helps in achieving higher quality products as it increases the freezing rate and smaller crystal formation during freezing. In the analysis, the performance of all the systems is compared at various evaporation temperatures of the *pf* evaporator and the variation in COP as shown in Figure 6.3. As shown, the COP of all the systems decrease with decrease in evaporator temperature. However, COP declining rate for the CO_2-NH_3 CRS is the minimum while it is the maximum for baseline NH_3 system. Decrease in the *pf* evaporator temperature from −35 ºC to −50 ºC results in 11.2% decrease in COP of CO_2-NH_3 CRS while 16.45% decrease in COP for NH_3 VCR system. Thus freezing can be achieved at lower temperatures with higher performance using the CO_2-NH_3 CRS.

The hourly temperature bin data of Mumbai as depicted in Figure 6.4 is used for calculation of AEC. Figure 6.5 shows the AEC estimated for all the analyzed systems for the year-round operation. The AEC value of the CO_2-NH_3 CRS was found to be 929 MWh. This value is lower than the NH_3, R22 and R404A baseline system by 3.2%, 6.2% and 12.3%, respectively. The AEC is directly linked with operating cost of the plant, higher the AEC more will be the operating cost. Thus, CO_2-NH_3 CRS offers the most economic option in terms of annual operating cost (AOC).

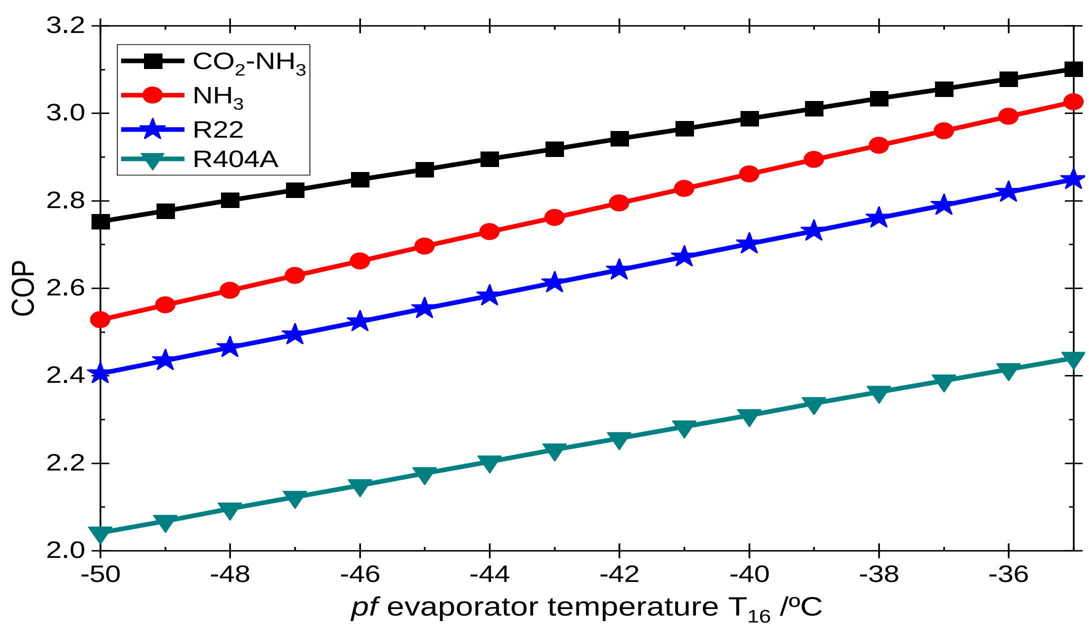


Figure 6.3: Influence of the *pf* evaporator temperature on COP

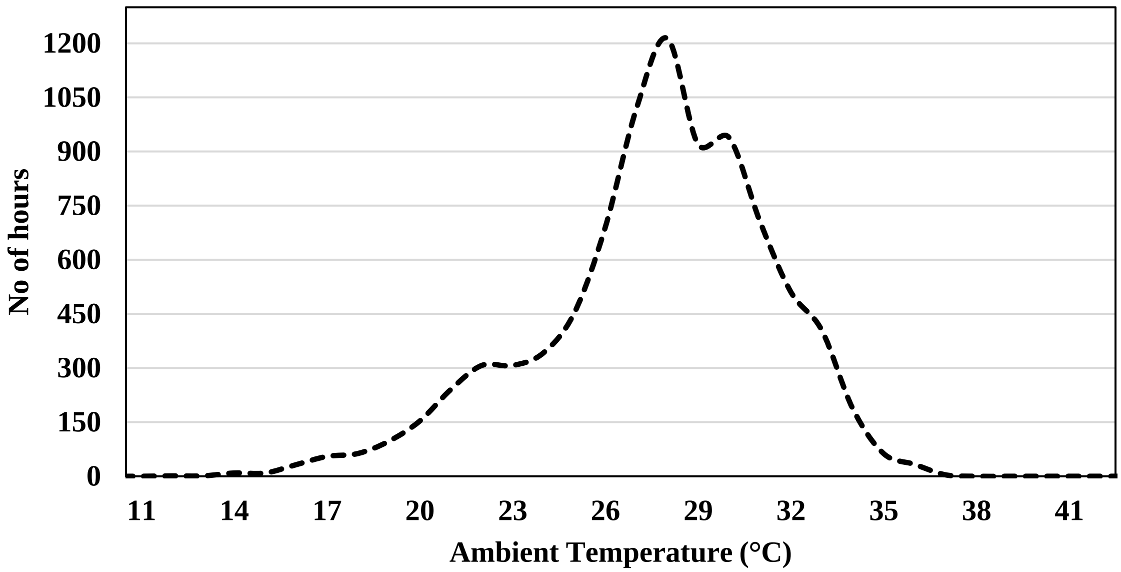


Figure 6.4: Hourly temperature bin of Mumbai, India

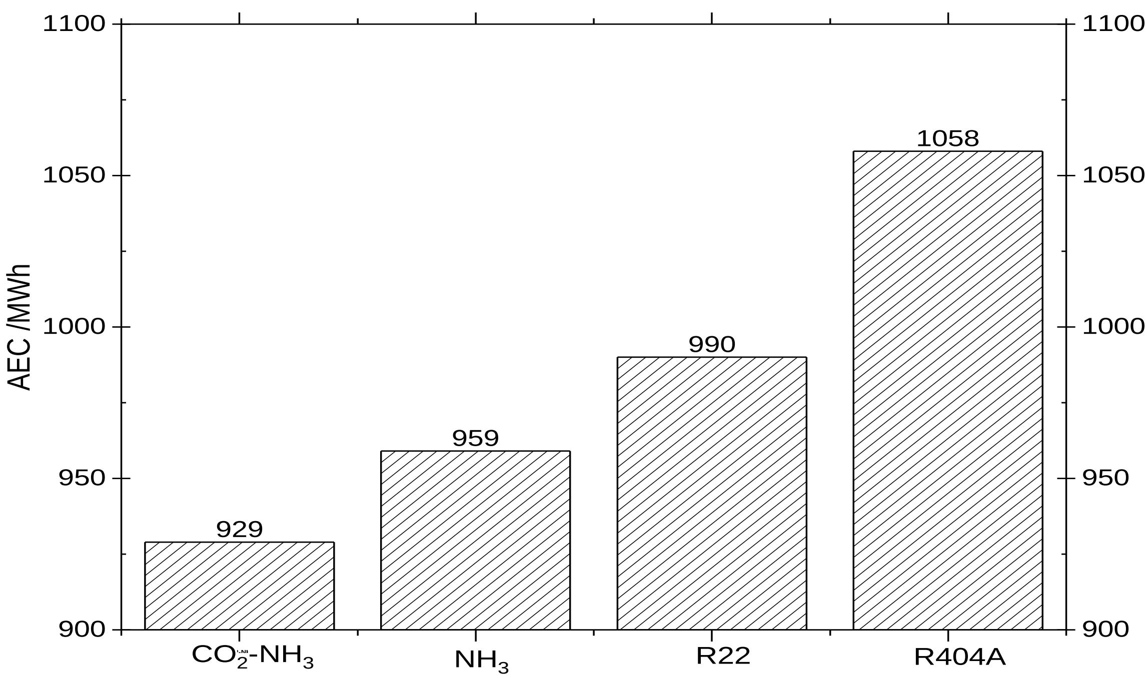


Figure 6.5: The annual energy consumption

6.4.2 Environmental emissions

Figure 6.6 presents the cumulative TEWI for all the systems analyzed. As GWP of NH_3 and CO_2 are 0 and 1 respectively, so the direct TEWI for the NH_3 baseline system and CO_2-NH_3 CRS are almost negligible. The contribution of direct TEWI for R404A and R22 systems are about 36.4% & 21.2% of total TEWI. The lowest value of the AEC of CO_2-NH_3 CRS results in lowest overall TEWI. The CO_2-NH_3 CRS is thus found to be the most environmentally friendly option.

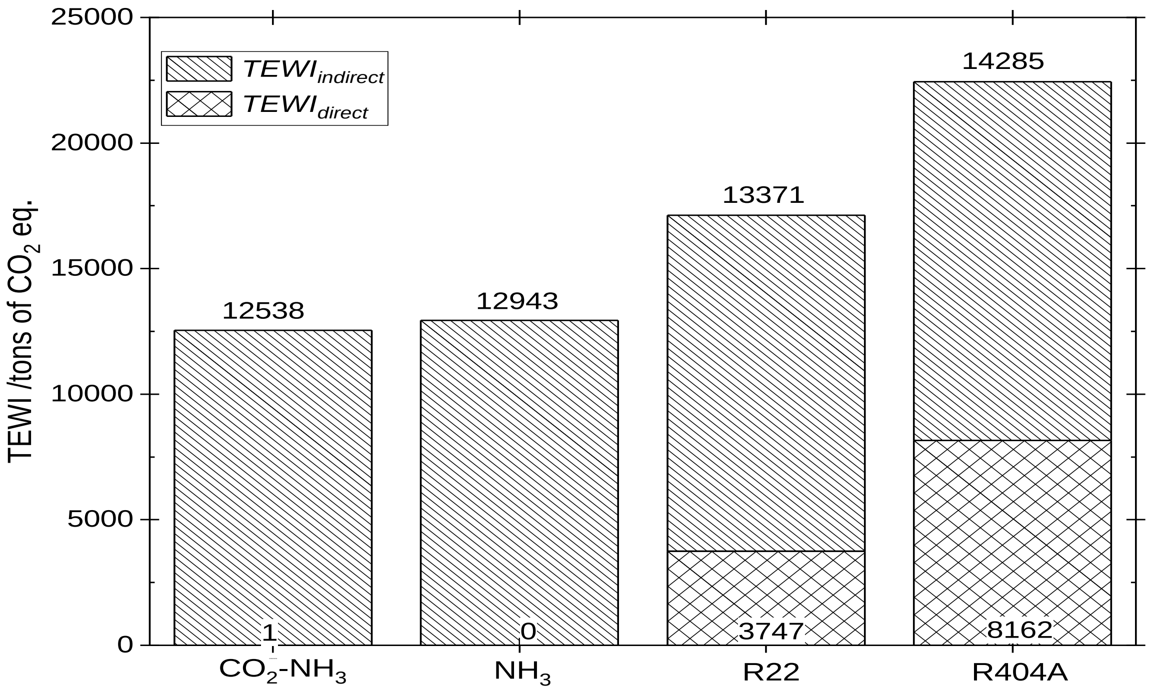


Figure 6.6: The direct & indirect total equivalent warming impact

6.4.3 Economic analysis

Based on the simulations, the refrigerant mass flow rates $\dot{m}_{ref}$ in various loops are determined. The heat exchanger area A and compressor power consumption $\dot{W}_{comp}$ at mentioned operating conditions and 30 ºC ambient condition for all the analyzed systems are computed and tabulated in Table 6.2. Due to the relatively higher heat capacity and higher latent heat of vaporization, the baseline NH_3 is found to have the lowest mass flow rate. For the simulation, condition of constant temperature difference between refrigerant and cooling fluid is assumed and heat transfer properties of refrigerants are avoided thus it could not capture the difference in evaporator size. However, the temperature of compressor discharge depends on the refrigerant properties, and it alters the $LMTD$ for condenser resulting in change in condenser area for all the refrigerants. The higher compression ratio and higher specific heat ratio of NH_3

compared to other refrigerants leads to higher discharge temperature. This results in higher value of *LMTD* and smaller heat exchange area of condenser.

As shown in Figure 6.2, the CO_2-NH_3 CRS has the highest performance and lowest compressor power consumption followed by baseline system with NH_3. R404A has the highest compressor power consumption. In the CRS, as the *cs* and *pf* compressors are only rejecting heat to the *cc* heat exchanger, thus the compressor power consumption of these two compressors are drastically reduced, leading to higher power consumption in the *cc* compressor.

Using the cost component functions mentioned in Table 6.1 and the obtained values of mass flow rates, compressor power consumptions and heat exchanger area in Table 6.2, various cost components computed for all the analyzed systems are presented in Figure 6.7. The total *ICC* of the baseline NH_3 is found to be the minimum due to lower refrigerant mass flow rates, lower heat exchangers surface areas and relatively lower compressor power consumption while for R404A system the same is the maximum. Although, the baseline NH_3 system followed the CO_2-NH_3 CRS but higher cost of CO_2 compressors and additional cost of cascade condenser results in relatively higher *ICC* for CO_2-NH_3 CRS. Due to the lowest *AEC*, the CO_2-NH_3 CRS also showed the lowest *AOC* followed by NH_3 baseline. Similarly, due to the lowest TEWI, the environmental cost of CO_2-NH_3 CRS is the lowest followed by NH_3 baseline, whereas higher TEWI by R22 and R404A systems lead them to higher environmental cost. Figure 6.8 presents the comparison between ACR and LCC of the analyzed systems. Despite having higher *ICC*, CO_2-NH_3 CRS showed the lowest *ACR* thanks to lower TEWI with a minor difference with NH_3 system. Since in the analysis we assume a constant maintenance cost as 5% of the *ICC*, thus it results in higher LCC for compared CO_2-NH_3 CRS compared to NH_3 system.

6.4.4 Performance analysis of CRS integrated with IHX and gascooler

The performance of the CRS integrated with the IHX and gascooler is compared with the baseline system with NH_3 for three different *pf* evaporator temperatures at –35 °C, –40 °C and –45 °C. Heat rejection rate by the gascooler to the ambient ($\dot{Q}_{GC}$) and the remaining heat transfer to the HTC from the LTC ($\dot{Q}_{HTC}$) computed using equations 6.11 and 6.12 are presented in Figure 6.9. The heat rejection rate by the gas-cooler ($\dot{Q}_{GC}$) to the ambient is depicted on the primary vertical axis along with the heat transfer rate to the HTC ($\dot{Q}_{HTC}$) on the secondary vertical axis.

Table 6.2: Refrigerant mass flow rates, heat exchanger areas and compressor power consumptions for analysed systems

Parameters		Unit	Two-stage VCR			CO_2-NH_3
			R22	R404A	NH_3	CRS
Refrigerant mass flow rate	$\dot{m}_1$	$kg\ s^{-1}$	0.7077	0.9992	0.1047	0.1047
	$\dot{m}_3$	$kg\ s^{-1}$	0.3440	0.4928	0.0504	0.0504
	$\dot{m}_5$	$kg\ s^{-1}$	0.4620	0.6949	0.0657	0.2802
	$\dot{m}_7$	$kg\ s^{-1}$	0.4146	0.6523	0.0576	0.2419
	$\dot{m}_{17}$	$kg\ s^{-1}$	0.1031	0.1932	0.0131	-
	$\dot{m}_{18}$	$kg\ s^{-1}$	0.1433	0.2964	0.0172	-
	$\dot{m}_{20}$	$kg\ s^{-1}$	-	-	-	0.1512
Evaporators area	A_{ch}	m^2	531.4	531.4	531.4	531.4
	A_{ice}	m^2	254.2	254.2	254.2	254.2
	A_{cs}	m^2	323.5	323.5	323.5	323.5
	A_{pf}	m^2	277.3	277.3	277.3	277.3
Cascade condenser area	A_{cc}	m^2	-	-	-	300
Condenser area	A_{cond}	m^2	557.9	889.5	319.3	308.3
Compressor power consumption	$\dot{W}_{ch}$	kW	20.29	22.33	19.62	19.62
	$\dot{W}_{ice}$	kW	13.67	14.15	12.31	12.31
	$\dot{W}_{cs}$	kW	33.27	38.44	31.93	8.74
	$\dot{W}_{pf}$	kW	43.79	54.94	40.99	15.67
	$\dot{W}_{cc}$	kW	-	-	-	44.05

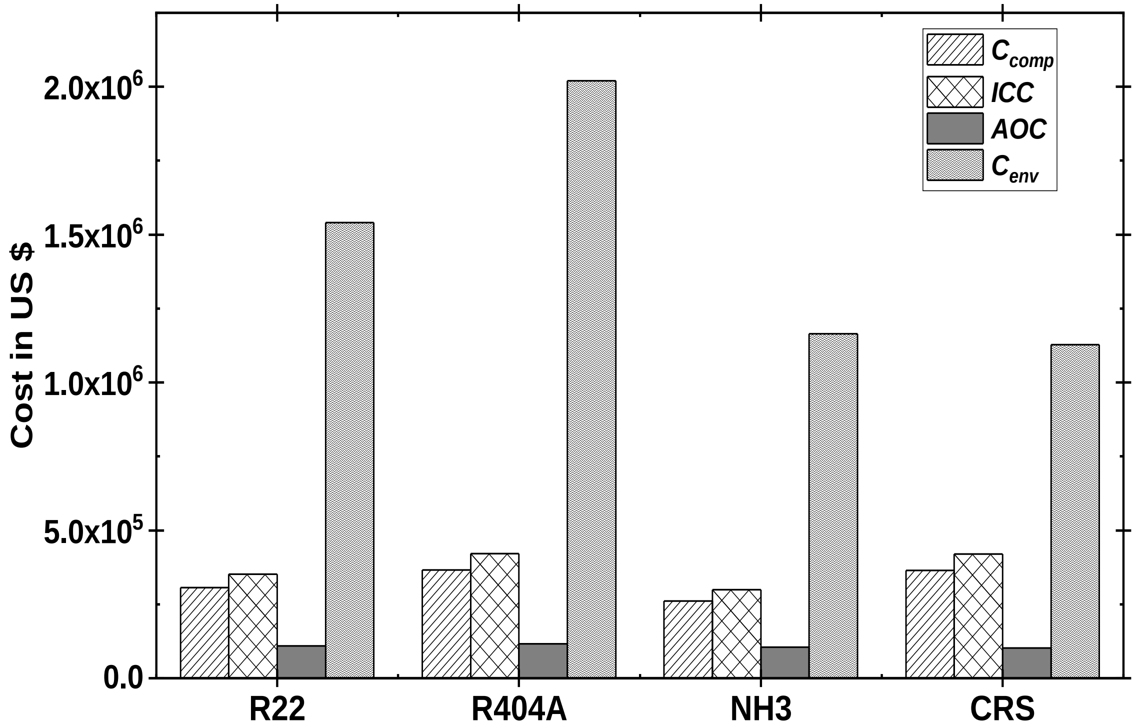


Figure 6.7: Comparison of various cost components between the analyzed systems

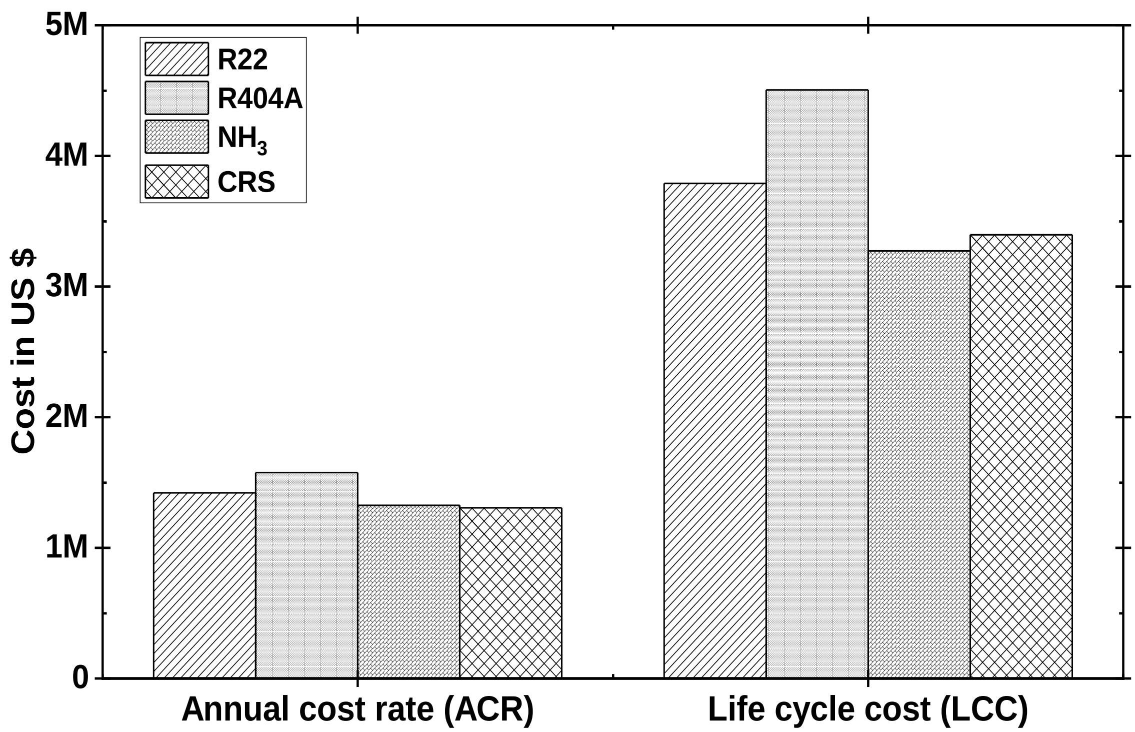


Figure 6.8: Comparison of ACR and LCC between the analyzed systems

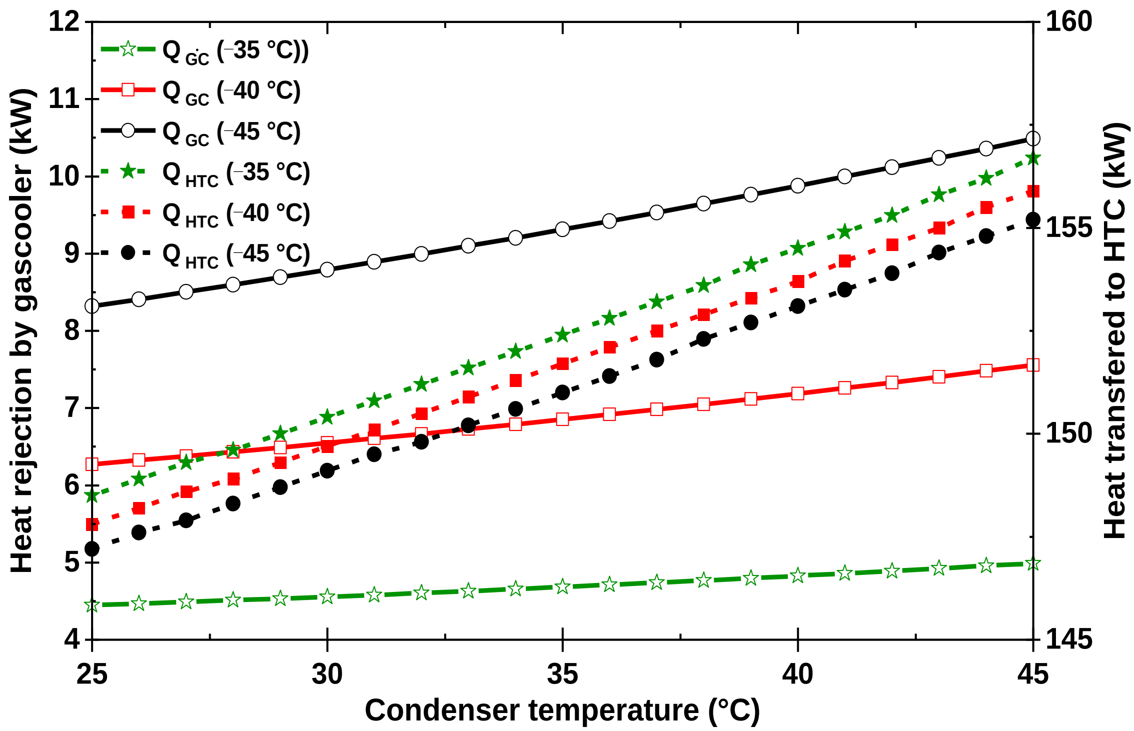


Figure 6.9: Variation of Q_{GC} and Q_{HTC}

With an increase in condensing temperature, $\dot{Q}_{GC}$ increases steadily, however, the rate of increase is highest for $\dot{Q}_{GC}$ for the *pf* evaporator temperature at –45 °C. It is concluded that the increase in the COP in cascade system is a combined effect of the decrease in compressor power consumption due to better thermophysical properties of CO_2 at lower temperature range, coupled with reduced pressure ratio caused by IHX and reduction of the partial thermal load from heat rejection to the ambient in the gas cooler.

Figure 6.10 presents the results of variation of COP at these evaporator temperatures with increase in condenser temperature of HTC. Also, the COP decreases with a decrease in the *pf* evaporator temperature. However, the CRS with the IHX and gascooler shows higher COP compared to the NH_3 baseline system for the full range of T_{cond}. At 25 °C condensing temperature, representing winter operation, the COP of baseline system and CRS with IHX and gascooler are 3.23 and 3.88 respectively while at 45 °C condensing temperature, representing summer operation, they are 1.96 and 2.37 respectively.

The percentage change in COP of the baseline and the CRS with IHX and gascooler are plotted for three different evaporator temperatures (Figure 6.11). It is observed that the minimum improvement of COP in the CRS with IHX and gascooler performance is 17% higher compared to the baseline system. The gain in COP is more pronounced, about 26%, when the *pf* evaporator temperature is –45 °C and for summer time operation. The result establishes the viability of using IHX in spite of its contradicting effects along with a gas-cooler in the CRS system for surimi processing.

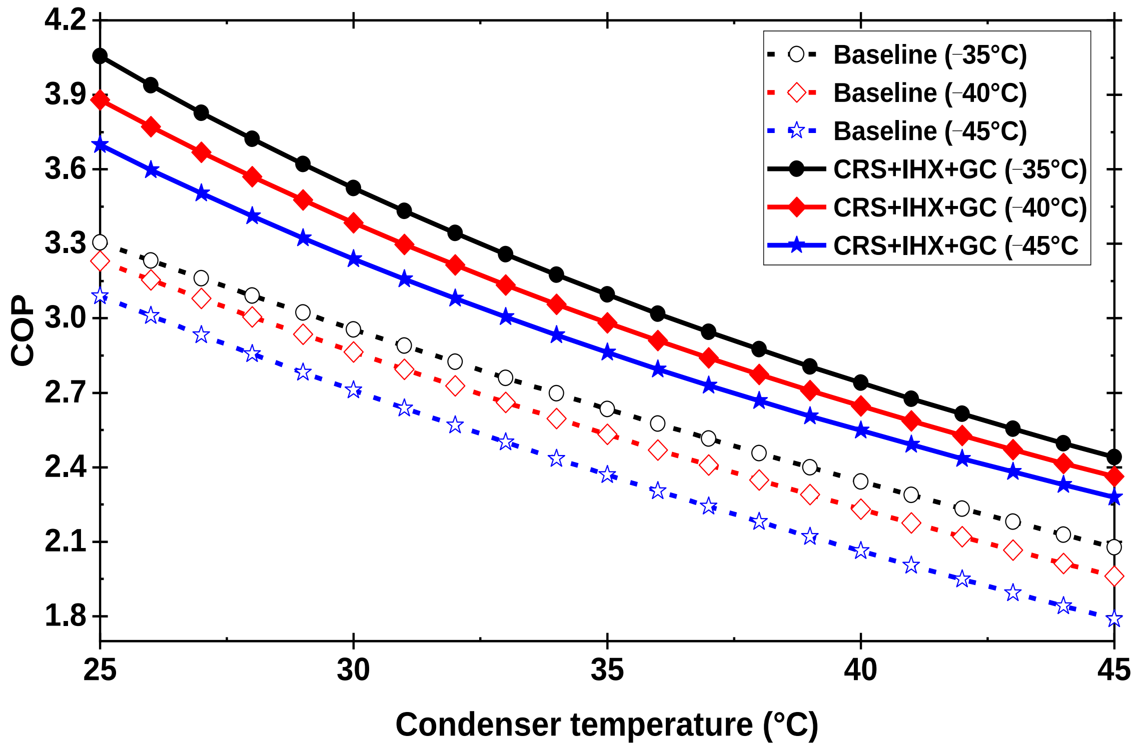


Figure 6.10: COP variation with condenser temperature

In this study, performance improvement by NH_3-CO_2 CRS equipped with IHX and gas-cooler over the conventional NH_3 system is explored. A possible operational benefit of the proposed integration with CRS is established. However, overall economic analysis based on a fixed and variable cost, the influence of local policy and market incentive or penalty, and long-term outlook needs to be examined before implementation.

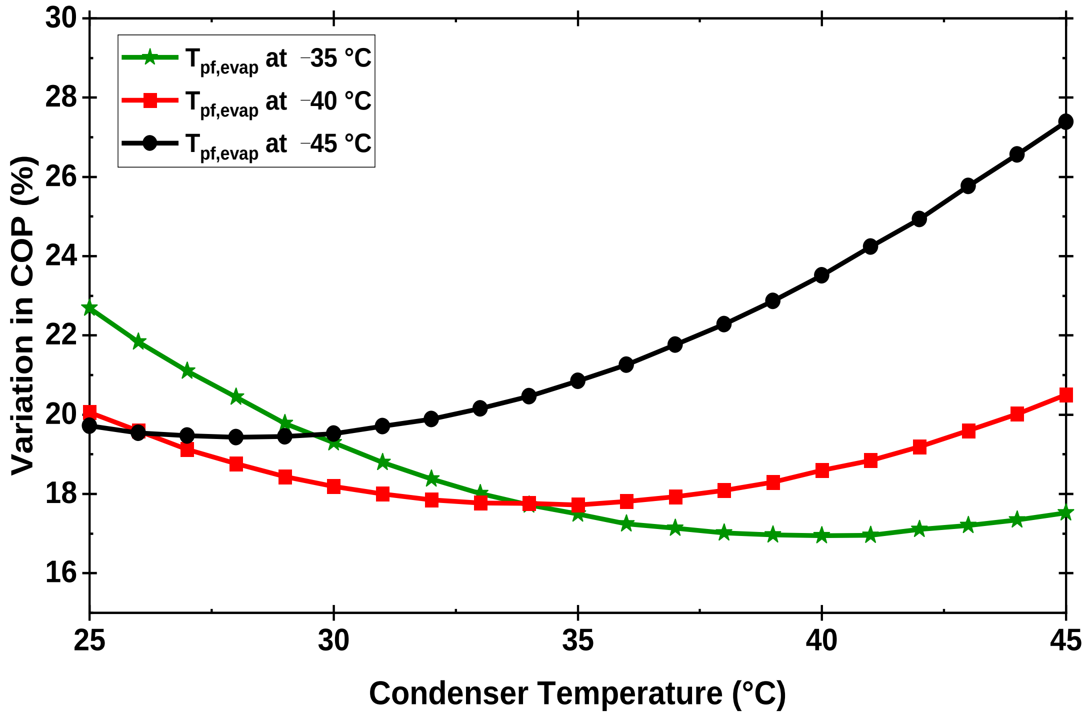


Figure 6.11: Percentage improvement in COP

6.5 Conclusions

This analytical study compared the performance of a state-of-the-art CO_2-NH_3 cascade refrigeration system (CRS) with multi-stage VCR system with working fluids R22, R404A and NH_3. The refrigeration systems are having multi-evaporators intended to meet specific refrigeration demands of a surimi processing and storage industry. The potential benefits of the CO_2-NH_3 CRS with respect to conventional VCR system are noted in terms of energy (AEC), environment (TEWI) and economy (*ICC*, *ACR* & *LCC*). Further, in order to enhance the performance of the CRS, integration of an IHX and a gascooler are proposed and the performance is compared with the baseline NH_3 system. The observations are summarized as follows:

The CO_2-NH_3 CRS yields relatively higher performance at all operating conditions compared to other investigated systems. An increase in the condenser temperature or decrease in freezing temperature, leads to reduction in the COP of all the systems, and the reduction is higher in the conventional baseline systems. At a condenser temperature of 45°C, COP of CO_2-NH_3 CRS is 2.4, which is about 15.0%, 46.8% and 5.3% higher than R22, R404A and NH_3 systems, respectively. Decrease in the *pf* evaporator temperature from −35 ºC to −50 ºC lead to a reduction in COP of CRS by 11.2% and that of NH_3 by 16.45%. Thus, cooling can be achieved at lower temperatures with higher performance by using CO_2-NH_3 CRS. The CRS exhibits 6.2%, 12.3% & 3.2% less AEC compared to the conventional R22, R404A & NH_3 systems, respectively. Similarly, it exhibited 26.8%, 44.3% & 3.2% less TEWI. Reduction in refrigerant mass flow rate leads to relatively compact system components for CO_2-NH_3 CRS compared to R22 and R404A systems. Relatively lower operating and environmental cost are obtained by CO_2-NH_3 CRS owing to higher performance and lower TEWI, but, the CRS has higher initial capital cost compared to the other systems. However, the CRS shows 10.4%, & 24.2% less life cycle cost compared to the conventional R22 & R404A system, but the same is 3.9% higher than NH_3 system. In this study, an integration of an IHX and gascooler with CRS is also proposed to further improve the system performance and its performance is compared against conventional NH_3 system. It has been observed that along with observable improvement in COP in the CRS system the NH_3 charge also can be reduced to 64%. Performance is also evaluated for a lower range of quick freezer operating temperatures at −35 °C, −40 °C, and −45 °C. The proposed CRS system is found more efficient for a wide range of operating conditions. The CO_2-NH_3 CRS presents a low charge NH_3 system, ensuring isolation of toxic and flammable NH_3 from proximity to food in cold storage and freezers. The CO_2-NH_3 CRS has all natural refrigerant configuration, thus, it will be immune to changing environmental regulations in future.

Theoretically, the proposed CRS system offers many advantages over the conventional system. However, a few practical challenges are also associated with the design and operation of such systems which are related to defrost cycle and standstill pressure of CO_2 system, which have not been discussed in this study. In addition to this, the lack of a ready market of system components for CO_2 in India and lack of trained technicians skilled in this technology at present lead to higher overall cost. Moreover, the confidence of vendors about performance of CO_2-NH_3 CRS in Indian context is low, which is also a formidable barrier in its large scale adoption. In view of the above and to study its exact behaviors on energy and other operational performance, future research needs to be focused on experimental investigation of CO_2-NH_3 CRS in the warm ambient conditions prevalent in India.

CHAPTER 7 Potential Low GWP Refrigerants for CRS, Benefits & Challenges in their Adoption

7.1 Abstract

The world is conscious about contribution to global warming from refrigeration and air-conditioning sector. A search is on for energy-efficient refrigeration systems and environmentally friendly refrigerants. Cascade refrigeration system (CRS) has been recognized as a prospective technology to improve energy efficiency while meeting multi-target temperatures. This study investigates various potential environmentally friendly refrigerants suitable for seafood processing and storage applications having multi-target temperature in a warm/hot climate. Fourteen low GWP pure fluids are identified as potential refrigerants based on literature survey and these are evaluated to determine the comparative impact of these refrigerants. For the comparison one popularly used blend, R404A is considered in the analysis. Out of these, six refrigerants (R41, R170, R1270, R404A, CO_2, and N_2O) were earmarked for the low-temperature circuit and ten (R152a, R161, RE170, R290, R1270, R1225ye(Z), R1234yf, R1234ze(E), R1243zf, and NH_3) are identified for the high-temperature circuit of the CRS based on their thermophysical properties. Note that R1270 appears in both the lists. Suitable modelling and simulation techniques were employed to investigate the performance parameters namely COP, annual energy consumption (AEC), TEWI, compressor volumetric displacement, compressor discharge temperature, and compression ratio. The refrigerant pair R1270-RE170 exhibited the best overall COP and the lowest total equivalent CO_2 emission for the application. CO_2-NH_3, the most studied refrigerant pair in the literature, showed marginally lower COP. However, it has the lowest compressor volumetric displacement leading to a compact system with the minimum refrigerant charge. NH_3 exhibited a higher compressor discharge temperature in all CRS, which can provide an opportunity for heat recovery.

Furthermore, this study presents a comprehensive assessment of energetic and environmental benefits from adoption of suitable natural refrigerants at various stages of entire surimi and shrimp cold chains in the Indian scenario. This includes refrigeration demands during fish harvesting, transport of fresh catch, sorting and processing, and long-time storage. For each

stage, the refrigeration demands are estimated and simple refrigeration cycle configurations are simulated to analyze the AEC and TEWI. Three scenarios of refrigerant use are analyzed and compared. Field data from a surimi supply chain in west coastal region of India and shrimp supply chain in east coastal region of India are used for the study. In addition, the various barriers in terms of technological, economic and policy related are highlighted.

7.2 Introduction

Adoption of a CRS could positively benefit over a conventional multi-stage VCR system in terms of energy, environment and economics in a seafood processing and storage application as has been established in *Chapter 5* and *6*. However, selection of refrigerant is an important aspect for its successful commercial implementation with a long term outlook. Apart from thermodynamic properties, their safety aspects such as toxicity, flammability, chemical stability, and also regulatory compliance play an important role (McLinden *et al.*, 2017). The most widely used refrigerants in industrial refrigeration application in India at present are HCFCs like R22, HFCs like R404A, R134a, R32 or other blends and natural refrigerant NH_3 (Kumar *et al.*, 2018). Most of them have very high global warming potential (GWP), except NH_3 (Zhu *et al.*, 2021). As per Kigali Amendment to the Montreal Protocol, HCFC and HFC group with high GWP are on planned phasedown world over (Heath, 2017). As a result, research for the next generation of refrigerants with low GWP has become a mandate. A number of studies have been published in recent years that focused on the performance analysis of various refrigerant pairs in CRS configuration and benefits of specific refrigerant pairs in CRS are summarized in *Chapter 2*.

Based on the literature survey, it is concluded that most of the reported studies considered only a single evaporator in the CRS. However, a seafood processing unit require simultaneous operation of multi-target temperature evaporators maintained at three to four different evaporation temperatures (Saini *et al.*, 2021). This application is thus unique, and there are no established results on the relative performance of various refrigerant pairs for such applications. Further, most of the studies explored only a few refrigerants like NH_3, CO_2, R290, R1270, R41, R161 etc. and left other potential low GWP refrigerants like RE170, R170, N_2O, and R152a from comparison. Moreover, majority of the studies preferred CO_2 in the LTC over other refrigerants. The reported studies mostly looked at parameters such as COP, energy consumption, exergy destruction, and emissions, leaving out parameters such as cost and availability, safety, system compactness, discharge temperatures, etc., that also play an important role in refrigerant selection (Ciconkov, 2018; Kim *et al.*, 2004). The current research work attempts to bridge the above research gaps.

With highlighted research gaps, this study aims to identify low GWP and better-performing refrigerant pairs suitable to meet the refrigeration demands of a seafood processing plant which can operate in high ambient temperature conditions. A comparative assessment of various parameters which can help in selecting the refrigerants is carried out based on their performance. A suitable research methodology is developed to determine the relative merits of various refrigerant combinations in a CRS. Selected parameters for the comparison are COP, AEC, TEWI, compressor volumetric displacement, compression ratio, and compressor discharge temperature.

Furthermore, three scenarios of refrigerant mix employed in India are analyzed to highlight the benefits of their adoption with an example of surimi and shrimp cold chains. Simple refrigeration cycle-configurations along with the refrigerants employed are simulated to meet refrigeration demands at various stages of the cold chains. The study is based on field visits, and surveys conducted in India at Mumbai for surimi, Kolkata for shrimp, and Udupi for rest raw material. The study also highlighted the challenges associated with adoption of natural refrigerants like CO_2, NH_3 and hydrocarbons R170, R290 and R1270 with focus on India.

7.3 Methods and Materials

The two broad aspects of this study are a) refrigerant selection and performance analysis of refrigerant pairs with CRS and b) analysis of refrigerant use scenarios in applications of surimi and shrimp cold chains. Methodology adopted for both the analyses are discussed in the following sub-sections.

7.3.1 Refrigerant selection

For refrigerant selection, the natural refrigerants and other potential refrigerants having GWP lower than 150 are selected from the literature as elucidated in *Chapter 2*. This GWP limit is adopted from the EU F-gas Regulation no 517/2014 which outlaw the use of HFC refrigerants having GWP 150 or more in the commercial refrigeration with a capacity more than 40 kW

(Regulation EU no. 517/2014). The methodology followed for the refrigerants selection is summarized in Figure 7.1.

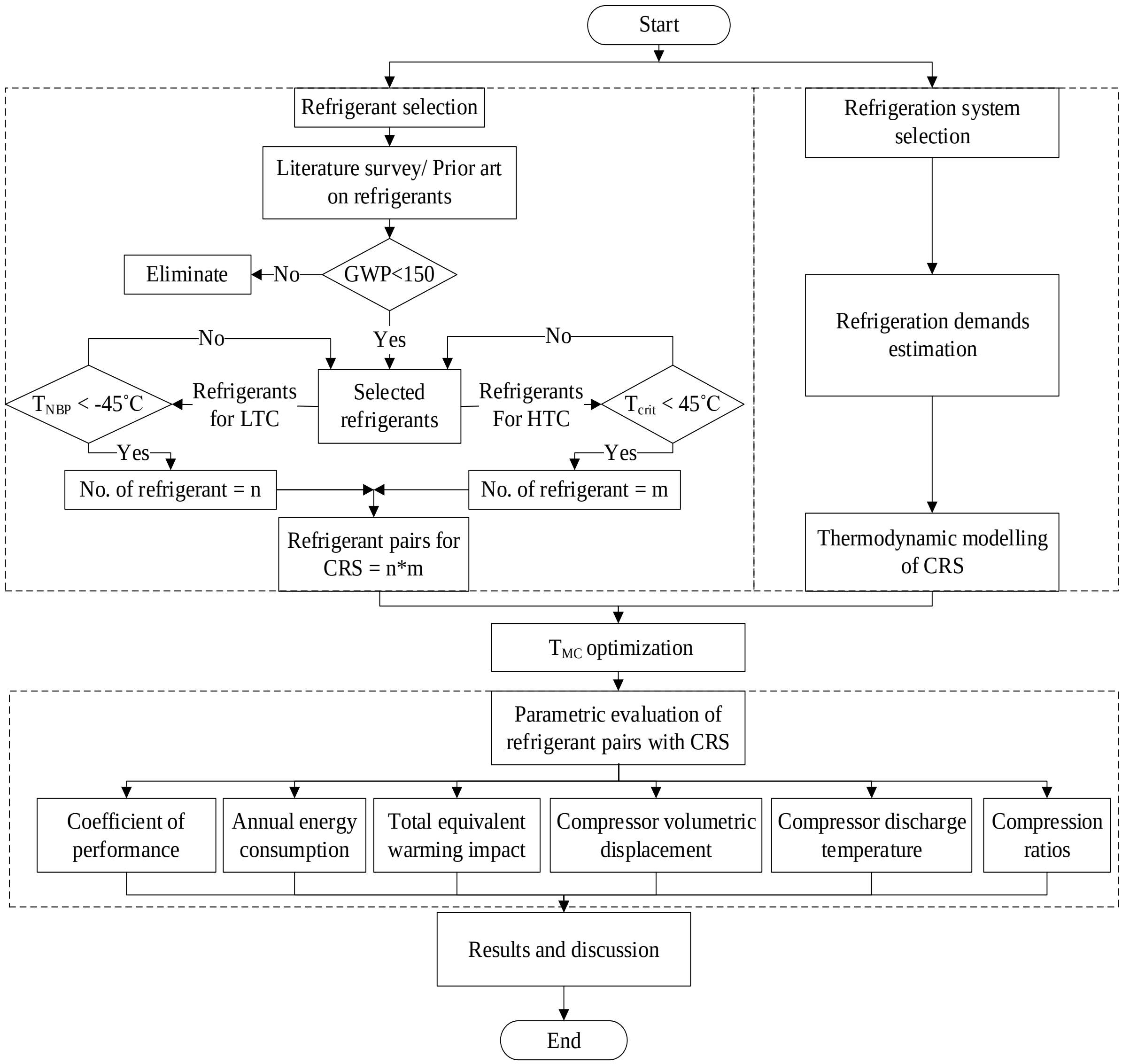


Figure 7.1: The research flow process chart

Out of the 27 potential refrigerants recommended by McLinden *et al.* (2017), 14 refrigerants having GWP over 150 are dropped similarly those which have not been commercialized yet are also omitted, leaving out 13 for this study. These are R41, R152a, R161, R170, R E170, R290, R1234yf, R1234ze (E), R1243zf, R1225ye (Z), R1270, NH_3, and CO_2. Two additional contenders, N_2O and R404A, are included in the study. Where N_2O, a natural refrigerant, which is not restricted by the regulation and has comparable thermophysical properties to CO_2

(Bhattacharyya *et al.*, 2009; Megdouli *et al.*, 2017) is included in the present study. R404A is added as a base case for comparison as it is one of the most commonly used refrigerant in industrial refrigeration for low temperature applications (Zhu *et al.*, 2021). Important thermodynamic and physical properties of the selected refrigerants are compiled in Table 2.3 of *Chapter 2*.

Subsequently, the refrigerants are earmarked for use in LTC and/or HTC based on their thermophysical properties and thermodynamic performance. Refrigerants with a NBP below −45 ºC are chosen for LTC to avoid vacuum operation to meet freezer application. Further, refrigerants with critical point above 45 ºC are selected for HTC to avoid high pressure transcritical operation in high ambient temperature conditions. This way 6 refrigerants – R41, R170, R1270, R404A, CO_2, and N_2O are found suitable for LTC, and 10 refrigerants – R152a, R161, RE170, R290, R1270, R1225ye(Z), R1234yf, R1234ze(E), R1243zf, and NH_3 were identified for HTC. R1270 qualified for both LTC and HTC. A total of 60 refrigerant pairs resulted from this procedure, and the performance of these refrigerant pairs is analyzed for the selected refrigeration system.

7.3.2 Refrigeration system and modelling

The refrigeration system analyzed in *Chapter 5* as CRS1 is adopted for the analysis in this study and also depicted in Figure 7.2, and referred as CRS hereafter. This system is intended to meet refrigeration demands of a seafood processing and storage industry.

As the considered refrigeration system is itself too complex, moreover 60 refrigerant pairs have been considered for performance investigation. For a critical analysis, a unique model needs to be developed which considers thermodynamic properties of particular refrigerant and the design of various components like compressor and heat exchanger accordingly, however, it will make the analysis multifaceted. Thus the simple lumped thermodynamic model presented in previous chapters is used for of all the refrigerant pairs and the simulations are carried out on the *EES* platform. The equation employed in the thermodynamic and environmental models are discussed in *Chapters 5* & *6* and the same are adopted in this study.

For a compressor, the isentropic efficiency(η_s) depends on compressor pressure ratios (R) and require different correlations based on compressor type, refrigerant use and operating condition. Considering the large number of refrigerant pairs under investigation, a common correlation (Sun *et al.*, 2019) for the refrigerants is followed given by:

$$\eta_s = 0.874 - 0.0135 \times R \qquad (7.1)$$

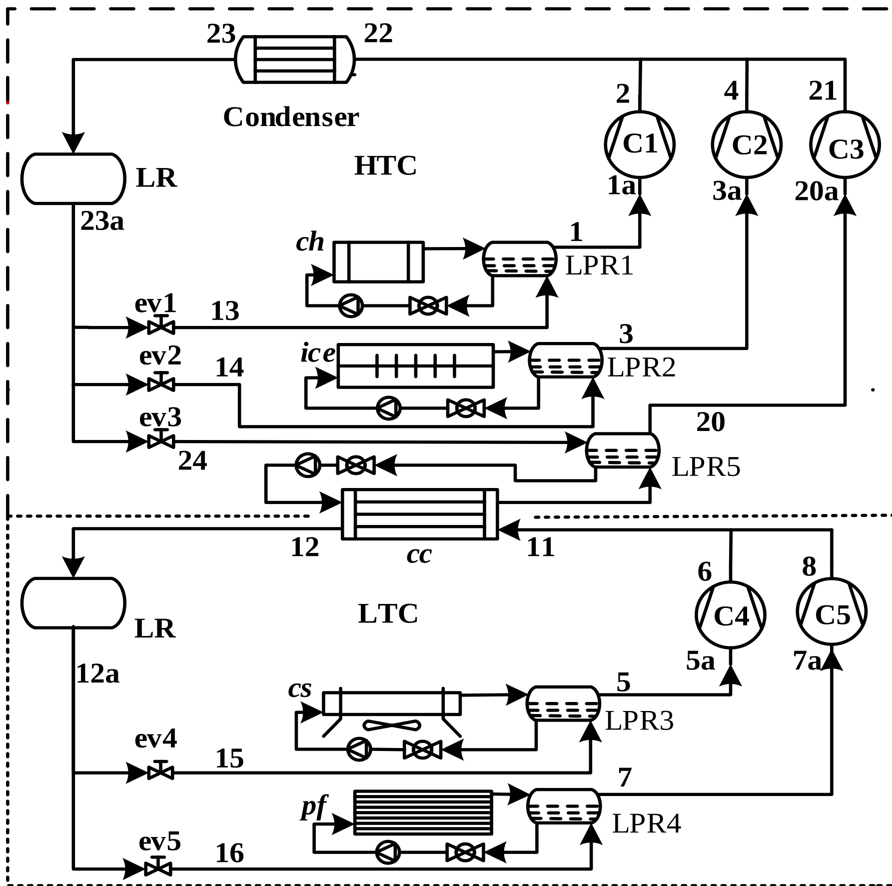


Figure 7.2: Schematic diagram of the CRS

Compressor volumetric displacement ($m^3\,h^{-1}$) is estimated employing equation 7.2, where d_i (kgm^{-3}) is the gas density of refrigerant at compressor inlet. The compressor discharge temperature and compression ratio are determined based on the state point properties iterated from simulated results. All these parameters are calculated for a constant condenser temperature of 35 ºC.

$$Vol.Displacement = \frac{\dot{m}}{d_i} * 3600 \qquad (7.2)$$

In the refrigerant analysis part, refrigeration load, the evaporation temperature and other operating conditions are assumed in a similar manner as discussed in *Chapter 5*. Selected parameters for comparison are COP, AEC, TEWI, compressor volumetric displacement, compression ratio, and compressor discharge temperature.

7.3.3 Refrigeration demands

For analysis of the possible energy and environmental benefits in the entire surimi and shrimp cold chains, the approximate refrigeration demands at all the stages of the cold chains are estimated based on annual production data. These are discussed in the following sub-sections.

a) Refrigeration demands in surimi cold chains

The export share of surimi out of the total seafood from India in 2020-21 was about 13% (1,42,975 tonnes) by volume and contributed to export earnings of 378.3 million USD (MPEDA, 2021). Most of the surimi processing plants are located on the western coast of India in the state of Gujarat and Maharashtra. Surimi production yields about 25% of the raw material (Park, 2015), and hence about 5,71,900 tonnes of raw material is assumed to be processed annually to produce the aforementioned 1,42,975 tonnes of surimi. India has a fishing ban of two months every year during the monsoon to allow fish to breed and repopulate the water. Thus, fish harvesting is carried out for about 304 days in a year. Therefore, daily production of surimi from India is about 470 tonnes from 1880 tonnes raw material. The five major stages of the supply side of surimi cold chain as discussed in *Chapter 1*, are: a) Fish harvesting, b) Fish handling and pre-processing, c) Transport to processing plants, d) Processing in factory and e) Storage. The large cooling demands in surimi cold chain at various temperatures have a lot of scope of improvement that can contribute to overall quality improvement as well as yield. Typical refrigeration demand at each stage is discussed in *Chapter 2*. These refrigeration demands are scaled up to match the total production and are tabulated in Table 7.1.

b) Refrigeration demands in shrimp cold chains

Shrimp supply chain has some similarity with the surimi supply chain; however, the harvesting is mostly from ponds and hatcheries where shrimp is farmed. Thus, the time spent between post-harvest and pre-processing is smaller which helps quality assurance. India is the global leader in shrimp farming; the total production in 2020-21 was about 7,19,847 tonnes out of which 82% was exported leading to export earnings of 4.43 billion USD (MPEDA, 2021). The shrimp farms are located mostly in four states on the eastern and the south-eastern coasts of India: West Bengal, Orissa, Andhra Pradesh, and Tamil Nadu. Field survey for data collection is carried out at a shrimp processing plant located at Kolkata (West Bengal) for this study. During processing, yield of shrimp meat is 45% of raw material (Venugopal, 2021). Assuming 304 days per year of production, average daily production is about 2370 tonnes for which 5260 TPD of shrimp is harvested. For the estimation of cooling load, based on the field visits, ice usage for 1 tonne of raw material for shrimp in the harvesting, handling and pre-processing, and transportation and waiting stages are considered as 1 tonne, 0.2 tonne and 1 tonne, respectively. At the processing stage, the estimated chilled water requirement is about 4 times the raw material weight. Quick deep freezing is the popular method for finished product at a low temperature of –35 °C. The average time of cold storage for shrimp is estimated as 1 month. The refrigeration demands throughout the entire shrimp cold chain are tabulated in Table 7.1.

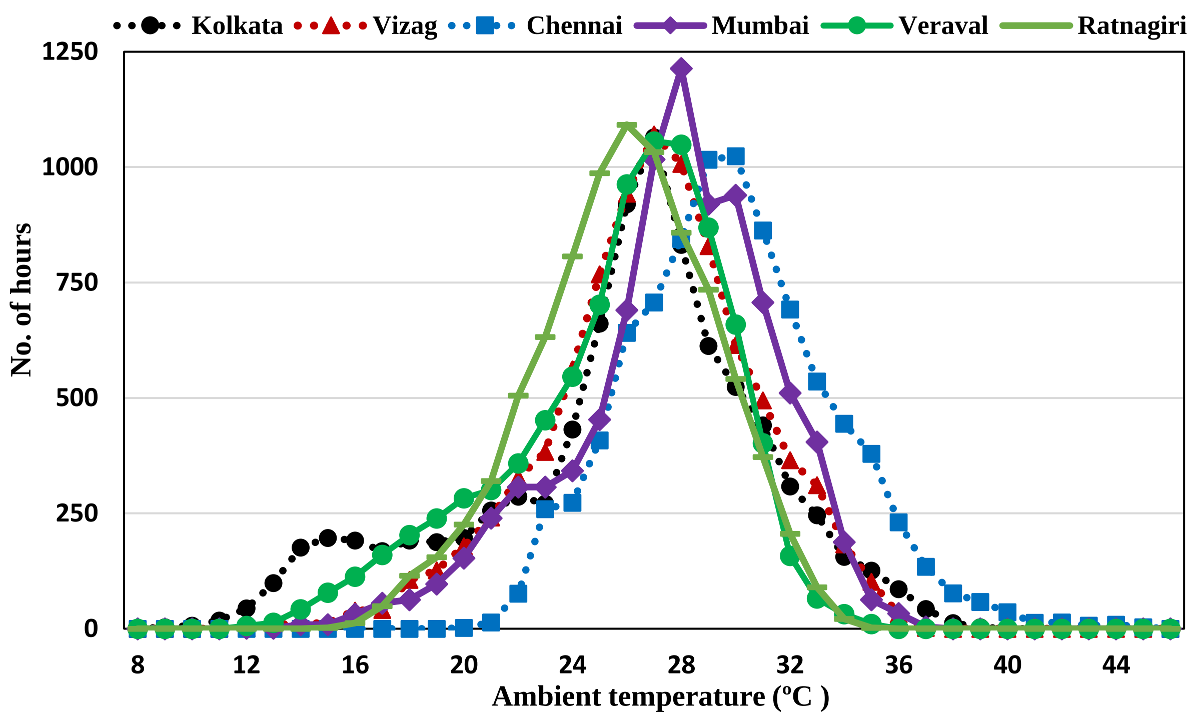


Figure 7.3: Hourly ambient temperature bin data of coastal sites in India

The hourly averaged temperature bin data of Kolkata, Vishakhapatnam, and Chennai (Eastern and Southeastern coast), and Mumbai, Veraval & Ratnagiri (Western coast), are exhibited in

Figure 7.3. All the coastal seafood producing sites have similarity in temperature profiles and the variation in temperature bin data is not large. Hence, for the estimation of annual energy consumption and total equivalent warming impact, climatic conditions of Mumbai on the western coast for surimi and Kolkata on the eastern coast for shrimp are considered. While for cooling demand estimation, the ambient temperature is considered fixed at 30 °C. Estimated cooling demands to meet 470 TPD surimi production and 2370 TPD shrimp production including required ice, chilled water, deep freezing and cold storage in the downstream supply chains are presented in Table 7.1.

7.3.4 Refrigerant usage scenario

The Alliance for an Energy Efficient Economy (AEEE), carried out an assessment of India's nationwide cooling demands for 2027, under a project by the Indo-German energy forum (Kumar *et al.*, 2018). This study reported cooling demands and refrigerants used in various sectors and future growth in these sectors in India. It also estimated the energy consumption and total CO_2 emissions. The study recorded some assessment of refrigerants used in India during 2017 and made a projection for 2027 for a business as usual scenario, and an *intervention* scenario. The intervention scenario is based on energy efficiency, changes in technologies as well as refrigerants, development of the cooling sector etc. for 2027. Further, the ozone cell, Ministry of Environment, Forest & Climate Change (MoEF&CC), Government of India along with the United Nations Environment Programme (UNEP) published a report on promotion of low GWP refrigerants for sustainable cooling technologies that have potential to be introduced at various stages of cold chains in India. This study suggested solutions such as low charge ammonia systems and CRS based on some case studies (Ozone cell, 2021). In another study, Saini et al. (2021) discussed how CRSs are more energy efficient for seafood processing and storage. In the current study, the energy consumption and emissions for three scenarios, *Scenario 0*: refrigerant mix as estimated by Kumar *et al.* (2018), *Scenario 1*: refrigerant mix as predicted for 2027 with interventions and *Scenario 2*: use of fully natural refrigerants mix are discussed. All the three scenarios are summarized in Table 7.2.

Table 7.1: Cooling demands in surimi and shrimp cold chains

Refrigeration demands	Surimi			Shrimp		
	Amount	Cooling load $\dot{Q}$ (kW)	Load %	Amount	Cooling load $\dot{Q}$ (kW)	Load %
Chilled water (7 ºC)	4703 TPD	5250	12.0	9470 TPD	10550	11.6
Ice (0 ºC)	6020 TPD	33090	75.3	11580 TPD	63650	69.8
Cold storage capacity (–20 ºC)	70,500 tonnes	2760	6.3	71100 tonnes	2790	3.1
Freezing (–35 ºC)	470 TPD	2820	6.4	2370 TPD	14220	15.6

Table 7.2: Refrigeration type and refrigerant use in various applications

Scenarios	Refrigerant mix		
	Chilled water & Ice	Cold storage	Deep freezing
Scenario 0	Single-stage: R134a: 20%, R22: 10% R404A: 5%, NH_3: 65%	Double-stage: R134a: 5%, R22: 10%, R404A: 10%, NH_3: 75%	Double-stage: R134a: 65%, R404A: 20%, R407/R410A: 10%, NH_3: 5%
Scenario 1	Single-stage: R134a: 10%, R404A: 5%, NH_3:85%	Double-stage: R134a: 5%, R404A: 5%, NH_3: 90%	Double-stage: R134a: 50%, R404A: 5%, R407/R410A: 30%, CO_2: 15%
Scenario 2	Single-stage: NH_3: 85%, R290/R1270:15%	Cascade system: CO_2-NH_3: 60%, R1270-NH_3: 20%, R170-NH_3: 20%	Cascade system: CO_2-NH_3: 60%, R1270-NH_3: 20%, R170-NH_3: 20%

Single-stage vapour compression refrigeration systems are considered for refrigeration in chilled water and ice production for all the three scenarios. While, for cold storage and freezing applications, double-stage vapour compression refrigeration systems with intercooling were considered for *Scenario 0 and Scenario 1*. For CO_2 in *scenario 1*, a transcritical parallel compression booster refrigeration system is considered (Purohit *et al.*, 2018). For *Scenario 2* CRS having either CO_2, R1270 or R170 in LTC and NH_3 in HTC for both cold storage and freezing application (Saini *et al*., 2021).

7.4 Results and Discussion

This study aims to determine suitable refrigerant pairs by comparing a set of evaluation criteria mentioned earlier. Simulations were carried out on the EES platform for the mentioned cooling demands using a model developed for all the refrigerant pairs, and the results are presented and discussed in this section. The comparative study of various performance parameters is carried out for each refrigerant pair at optimum T_{MC} and is shown in following sections.

As discussed earlier in *Chapter 5*, the trends of COP variation for all the refrigerant pairs are found similar to that of CO_2-NH_3 in T_{MC} optimization. At a fixed T_{cond}, the COP_H increases with an increase in T_{MC} due to lowering of compression ratio while COP_L decreases due to the opposite reason. This variation of COP_L and COP_H results in a nonlinear change in overall COP (Ustaoglu *et al.*, 2020). Starting from a lower value of $T_{MC,}$ the overall COP gradually increases with an increase in T_{MC} and beyond an optimum value, it starts decreasing. Using the *Golden section search* method in *EES*, T_{MC} is optimized to obtain the maximum COP. The optimum T_{MC} values are simulated at a range of condenser temperatures. Figure 7.4 presents the optimum T_{MC} for all refrigerant pairs at T_{cond} 35 ºC and other mentioned operating conditions. It is observed that for any refrigerant in HTC, the optimum T_{MC} is the lowest when paired with CO_2 in LTC, implying large penalty for operating CO_2 system with high pressure ratio. Again, for any refrigerant in LTC, optimum T_{MC} is the lowest when paired with RE170 in HTC implying its performance is least affected by increase in pressure ratio. This wide range of T_{MC} provides an opportunity to select refrigerants based on the application requirements at different levels.

7.4.1 COP and annual energy consumption

Overall COP of the CRS for all the refrigerant pairs is compared in Figure 7.5. It is observed that for any refrigerant in HTC, R1270 in LTC provides higher COP and the reference refrigerant R404A provides the second-highest COP, while RE170 and NH_3 in HTC in general provide higher COP for any refrigerant in LTC. The refrigerant pair R1270-RE170 is adjudged the best performing pair in terms of COP (3.18), while R170-R1234yf is adjudged the worst-performing, whose COP is about 14% lower than the best performing pair. In the Indian seafood industry, NH_3 and R404A are the most utilized refrigerants. Both of them have relatively higher COP in their respective circuit. CO_2-NH_3, the most studied CRS refrigerant pair in the literature, shows about 6.5% lower COP than the best COP option. HC RE170 has high flammability and is in the A3 safety group, while natural refrigerant NH_3 is in the B2L safety group, with mild toxicity and low flammability. It is noted that, for the HTC refrigerants, all the HFOs exhibit comparatively lower COP.

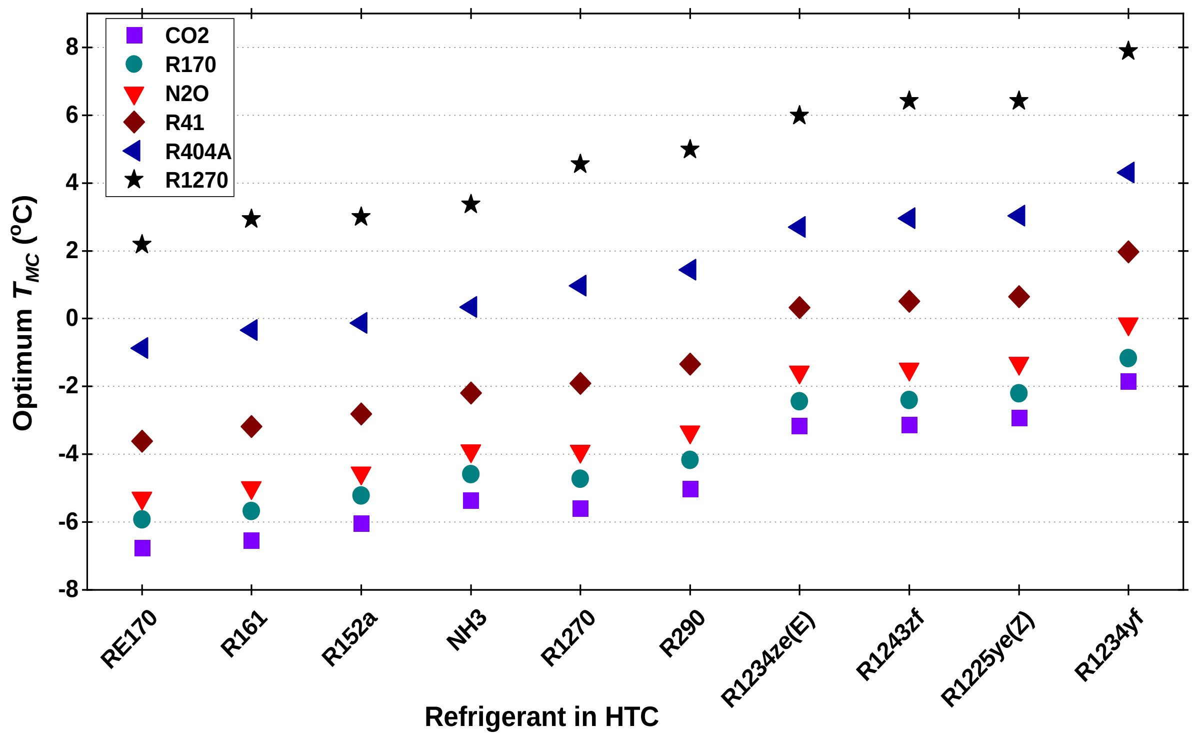


Figure 7.4: Optimum T_{MC} of refrigerant pairs at T_{cond} 35 °C

An assessment of the annual energy consumption (AEC) is done based on the hourly temperature bin data of Mumbai using the ISHRAE database (Mathur *et al.*, 2017). AEC for all the refrigerant pairs is estimated and the results are illustrated in Figure 7.6.

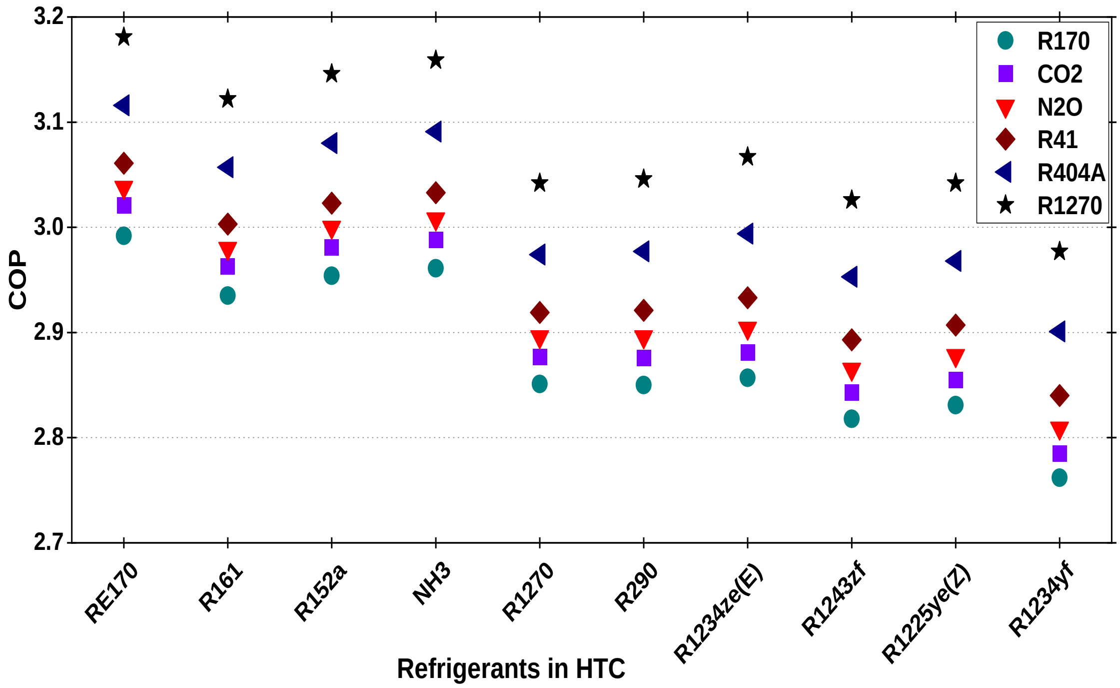


Figure 7.5: COP of refrigerant pairs at T_{cond} of 35 °C

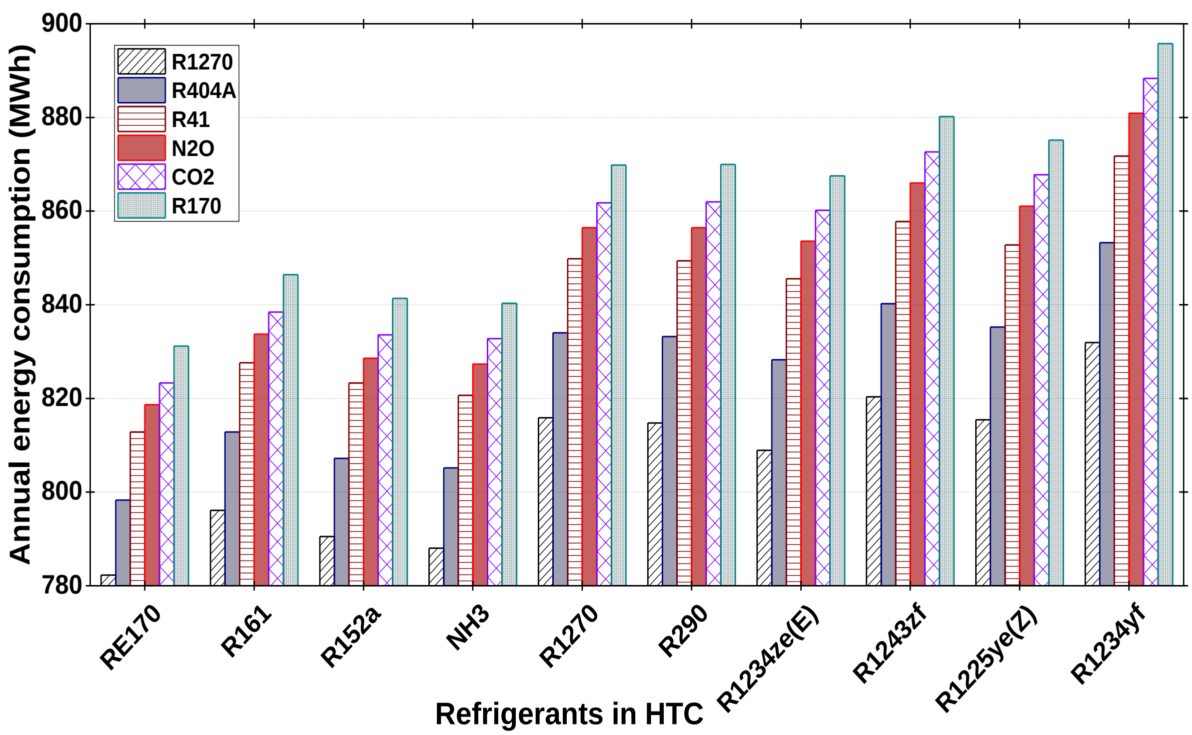


Figure 7.6: Annual energy consumption

The AEC of the refrigerant pair follows the reverse trend of their COP value; therefore, refrigerant pairs R1270-RE170 and R170-R1234yf, respectively, yield the minimum AEC of 782.3 MWh and the maximum AEC of 895.8 MWh.

7.4.2 Total equivalent warming impact (TEWI)

Figure 7.7 illustrates the TEWI values for all the refrigerant pairs. As anticipated, the reference fluid R404A exhibits the highest warming impact paired with any other refrigerant. While RE170 and NH_3 used in HTC generally have the lowest TEWI when paired with other refrigerants. Among the LTC refrigerants, R1270 exhibits the lowest and R41 the second lowest value of TEWI paired with any HTC fluid. However performance of R1270 as HTC fluid is not as good. Refrigerant pair R1270-RE170 showed the minimum TEWI of 10560 tonnes of CO_2 equivalent for the cooling load. Although refrigerants NH_3 and CO_2 have the lowest GWP among HTC and LTC refrigerants, due to marginally higher AEC, the pair CO_2-NH_3 yields relatively higher $TEWI_{indirect}$ resulting in 6.44% higher total TEWI than the best performing pair.

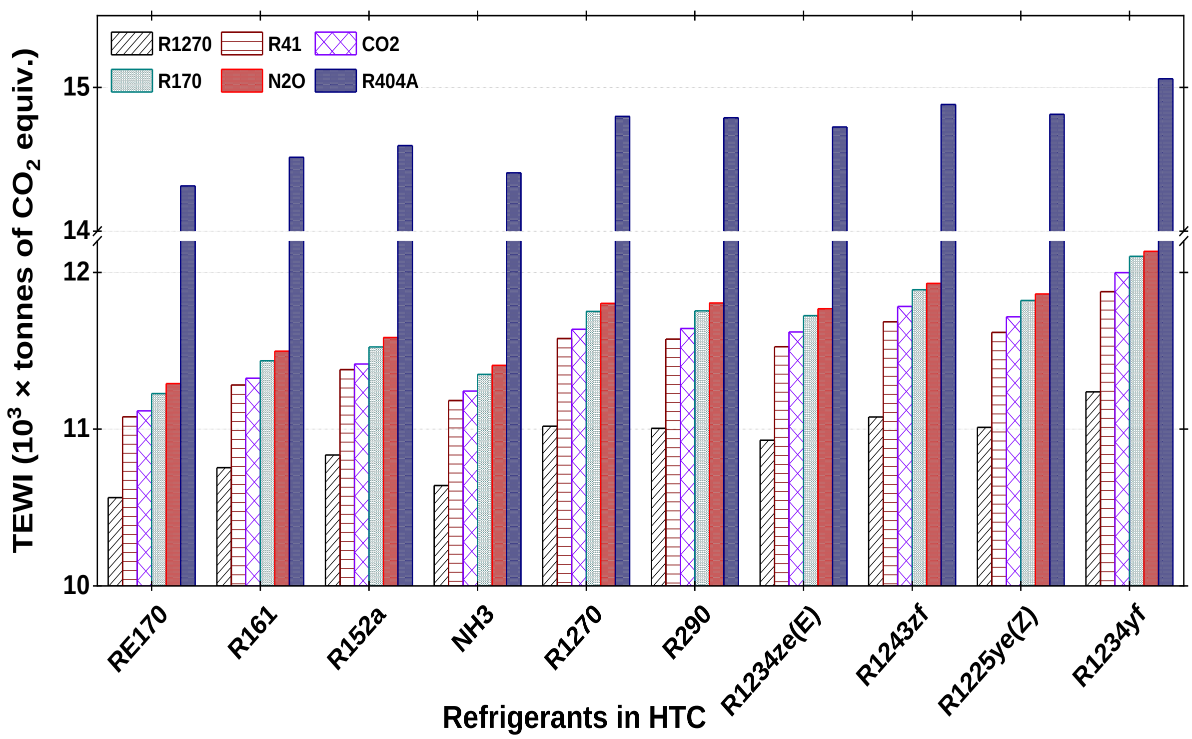


Figure 7.7: TEWI of refrigerant pairs

7.4.3 Compressor volumetric displacement

Compressor volumetric displacement is an important influencer on the size of refrigeration system components. Low volumetric displacement makes the system compact and more preferable for certain applications. The refrigerant properties and the operating conditions affect the volumetric displacement. Variation in T_{MC} is also viewed as a change in operating conditions for both the LTC and HTC compressors. The refrigerant combination in a pair dictates the optimum T_{MC} when other parameters are constant. In the CRS (Figure 7.2), *ch* and *ice* compressors are not connected to LTC; therefore, the volumetric displacement of these compressors vary only with the HTC refrigerant. Thus, the volumetric displacement of these two compressors are studied independent of T_{MC} or the LTC refrigerants and is shown in Figure 7.8. It is observed that, within the HTC refrigerant choices, the NH_3 system has the lowest volumetric displacement in *ch* and *ice* compressors, followed by R1270. Due to the lower refrigeration capacity, HFOs have comparatively higher volumetric displacement. Although RE170 performed better than NH_3 in terms of COP, it has about 84.3% higher volumetric displacement than NH_3. R1234ze(E) has the highest volumetric displacement among HTC refrigerants which is about 134.5% higher than NH_3.

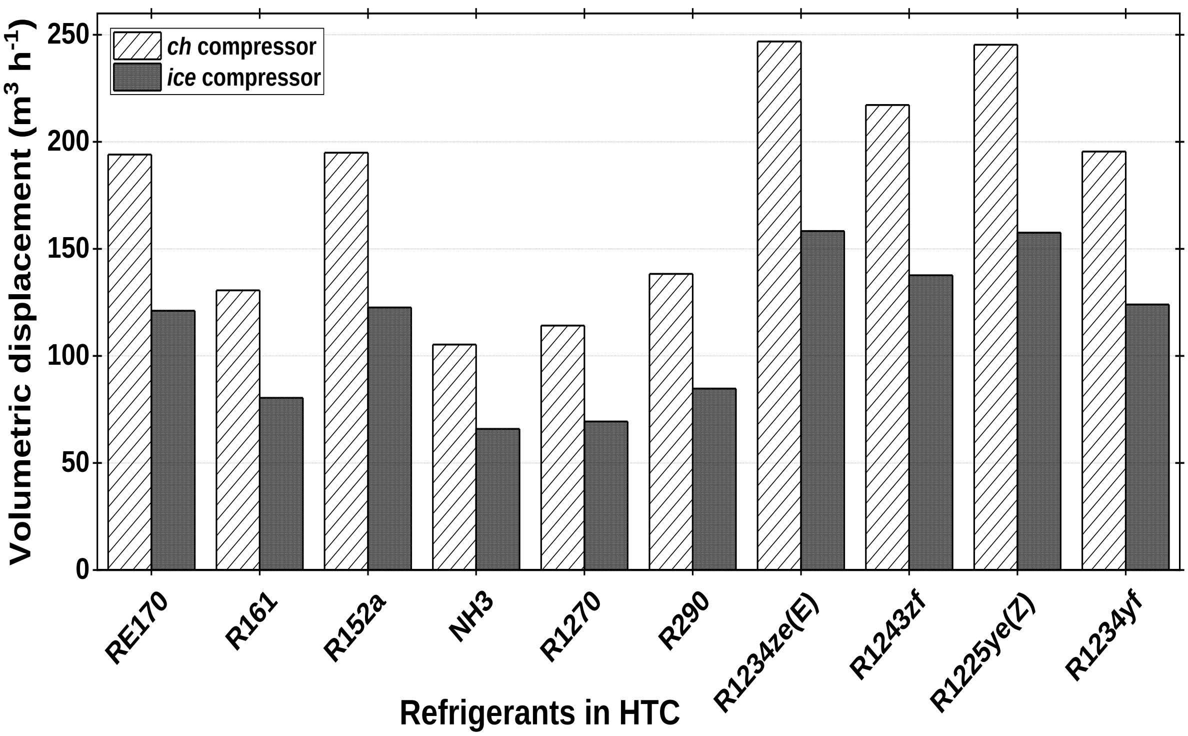


Figure 7.8: Volumetric displacement of *ch* and *ice* compressors at T_{cond} 35 ºC

The variation in T_{MC} influences the volumetric displacement of the *cc*, *cs*, and *pf* compressors. However, from the simulation, we observed that the effect of change in T_{MC} on the volumetric displacement of *cs* and *pf* compressors is below 2%. Hence, the volumetric displacement of *cs* and *pf* compressors are studied independent of HTC refrigerant and is shown in Figure 7.9. It is observed that within the LTC refrigerant choices, CO_2 has the lowest volumetric displacement in *cs* and *pf* compressors which are 25.7 and 37.8 $m^3\ h^{-1}$, respectively. The same is about 82% lower than the refrigerant R1270 which has the highest volumetric displacement.

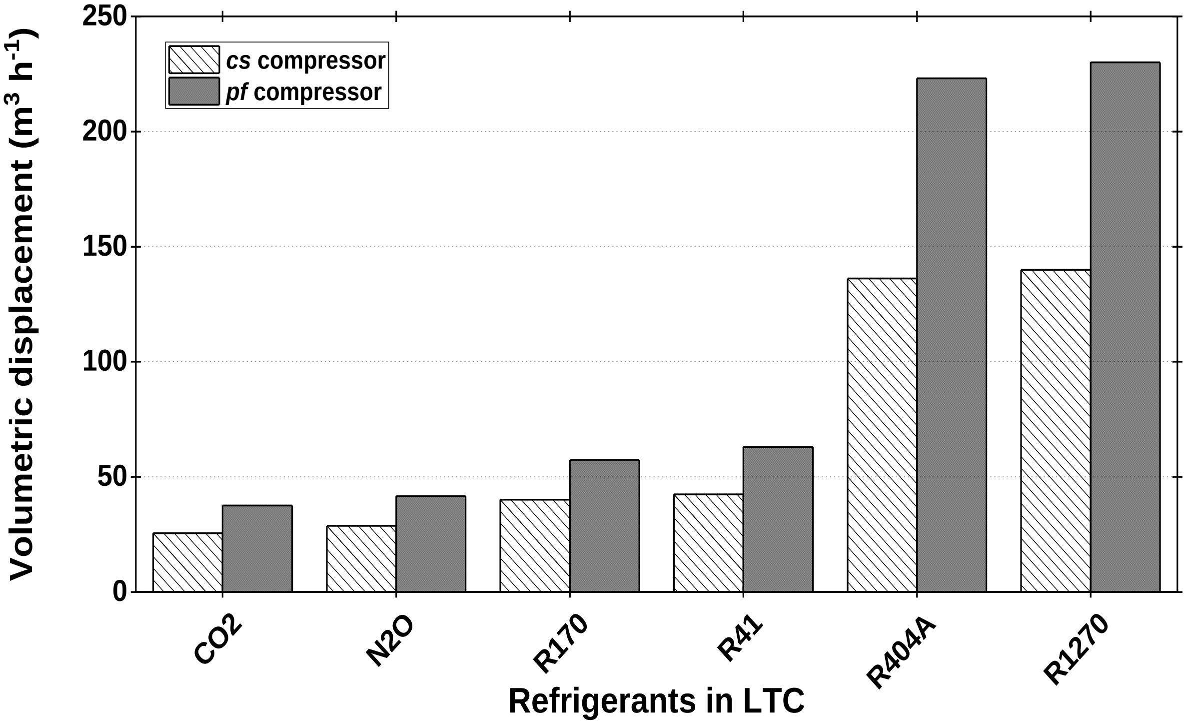


Figure 7.9: Volumetric displacement for *cs* and *pf* compressors at T_{cond} 35 ºC

From the simulation we find that T_{MC} has a more profound effect on the volumetric displacement of the *cc* compressor. Figure 7.10 shows the consequences of variation in the optimum T_{MC} on the volumetric displacement of all five compressors for a case when NH_3 in HTC is paired with various LTC refrigerants. Volumetric displacement of all the other HTC refrigerants follows this trend in the *cc* compressor. As T_{MC} increases from –5.4 ºC for CO_2-NH_3 pair to 3.4 ºC for R1270-NH_3 pair, volumetric displacement of NH_3 in *cc* compressor decreases by 26.1% from 247 $m^3\ h^{-1}$. Figure 7.10 also shows that volumetric displacement of *ch* and *ice* compressors remains constant for a particular refrigerant in HTC. Refrigerants NH_3 and CO_2 have the lowest volumetric displacements in their respective circuits, thus CO_2-NH_3 has the lowest total volumetric displacement, resulting in the most compact system.

The refrigeration capacity of a refrigerant has profound effect on compressor volumetric displacement. Low refrigeration capacity leads to higher volumetric displacement and high refrigerant charge in a system. The high volumetric displacement increases the compressor duty time and also the compressor power consumption (Sánchez *et al.*, 2019). In this study, the isentropic efficiency of compressor is taken as a function of compression ratio only, as a consequence, the effect of higher volumetric displacement on COP or AEC has not been captured.

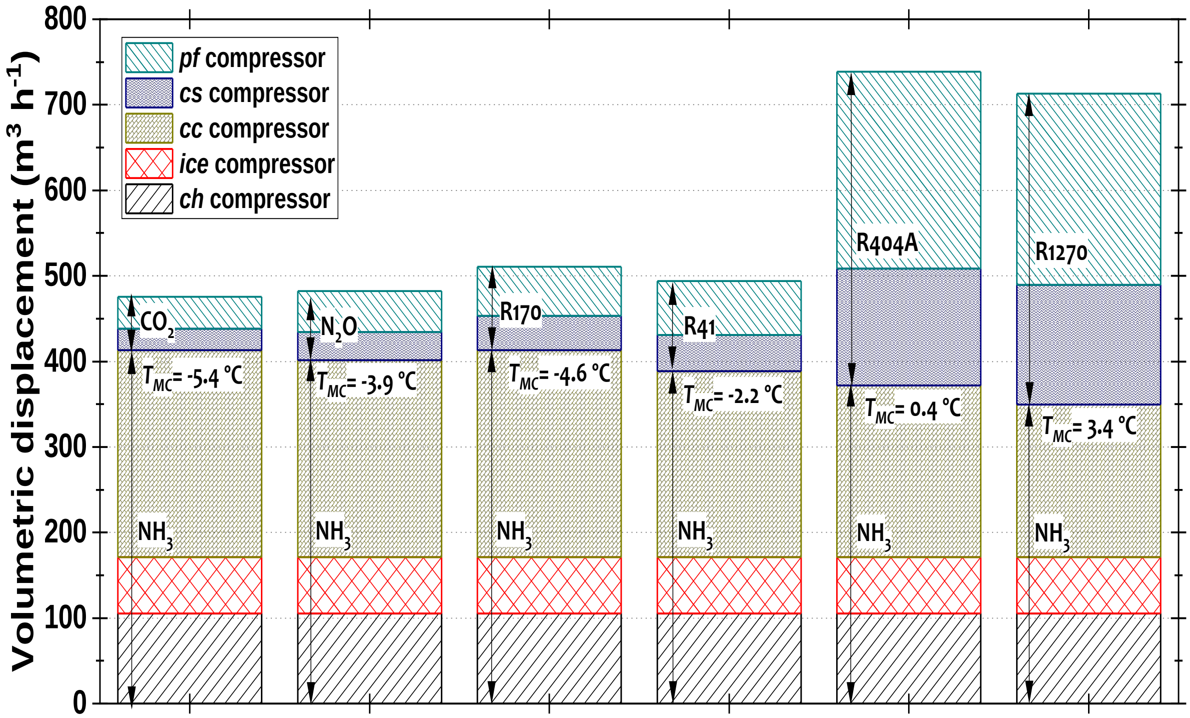


Figure 7.10: Influence of T_{MC} on the volumetric displacement of *cc* compressor, NH_3 in HTC paired with various LTC refrigerants at T_{cond} 35 °C

7.4.4 Compression ratio and discharge temperature of compressors

Based on state point properties iterated from the simulation at T_{cond} 35 ºC, two parameters compression ratio and discharge temperature are determined for all five compressors and all the refrigerant pairs. The compression ratio and discharge temperature are illustrated in Figure 7.11 and Figure 7.12, respectively. For the *ch* and *ice* compressors, both the parameters are observed to vary when HTC refrigerant is changed but remains unaffected when LTC refrigerant is changed. In contrast, for the *cc*, *cs* and *pf*, these two parameters are observed to vary with both the LTC and HTC refrigerants. The value of compression ratio for most of the compressors, irrespective of refrigerant pairs, are found to be below 4.5 except for the *cc* and *pf* compressors. Due to significant differences in their evaporator and condenser temperatures, the *cc* compressor in HTC and *pf* compressor in LTC exhibited higher compression ratios.

Among the HTC refrigerants, NH_3 is found to have the highest compression ratio, and R1270 has the lowest. Correspondingly, among the refrigerants in LTC, R1270 has the highest compression ratio and R170 has the lowest. NH_3 also has the highest specific heat ratio among all the HTC refrigerants. The combination of high compression ratio and high specific heat ratio of NH_3 result in higher discharge temperature. The discharge temperature of all the compressors is found to be below 80 ºC except for NH_3, which reaches up to 137.3 ºC. A high compression ratio implies lower isentropic efficiency, thus further lowering the actual COP from the theoretically calculated value. High discharge temperatures may also contribute to accelerated wear of compressor components and degradation of the lubricant. However, the temperature range for this application is observed to be well within the allowable limit (Stewart, 2018). The higher discharge temperature of NH_3 can be explored for heat recovery opportunities which can improve overall system performance.

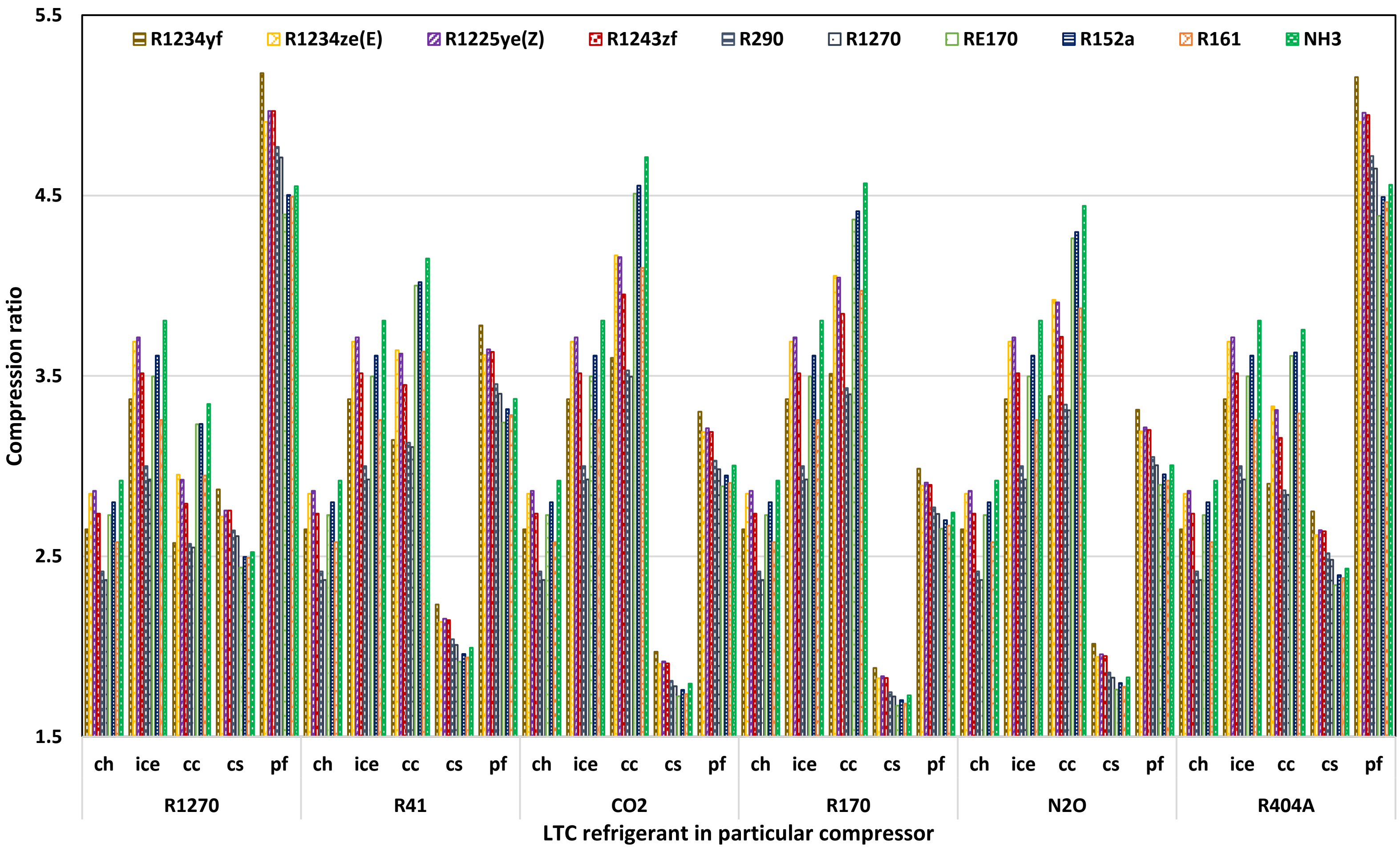


Figure 7.11: Compression ratio at T_{cond} 35 °C

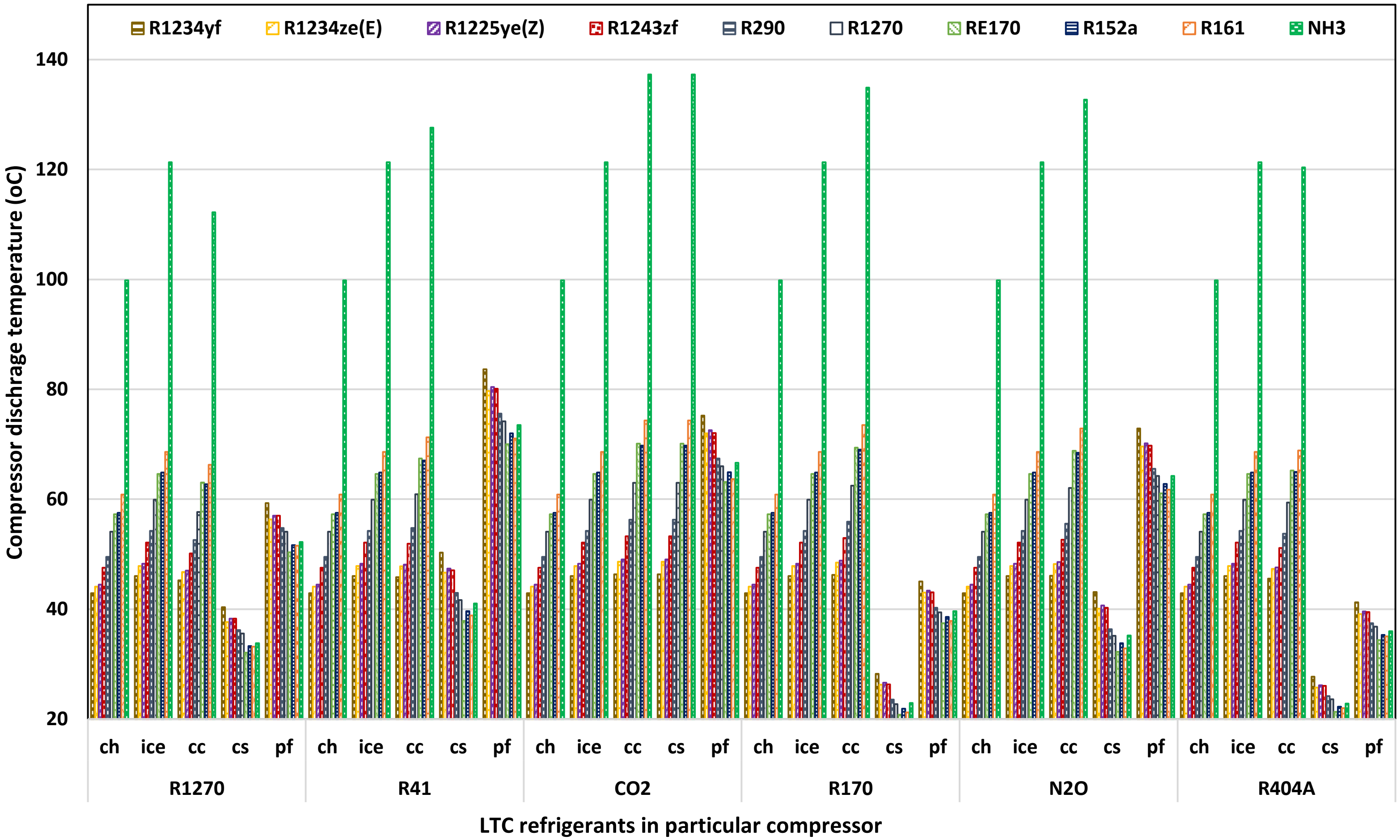


Figure 7.12: Compressor discharge temperature (°C) at T_{cond} 35 °C

7.4.5 COP analysis of refrigerants in various applications of the cold chain

Performance of all the systems with the mentioned refrigerants for particular application are estimated through simulation for the ambient conditions of Mumbai and Kolkata. COP at an ambient temperature of 30 ºC for all the refrigerants along with the analyzed refrigeration systems with various refrigerants are given in Table 7.3. Among all the analyzed refrigerants NH_3 has the highest COP for all the applications. Although COP of cascade system in freezing application is higher compared to NH_3 from other investigated HTC refrigerants.

Table 7.3: COP of refrigeration systems with different refrigerants at 30 °C

Application/ Refrigerant mix	Single-stage system		Two-stage system		Cascade system	
	Chilled water	Ice	Cold storage	Deep freezing	Cold storage	Deep freezing
NH_3	5.86	4.08	2.31	1.51	-	-
R134a	5.04	3.87	1.96	1.22	-	-
R404A	4.56	3.48	1.65	0.99	-	-
R22	5.08	3.95	2.08	1.34	-	-
R410A	4.69	3.62	1.84	1.16	-	-
R407C	5.33	4.11	2.09	1.34	-	-
R290	5.56	4.24	-	-	-	-
R1270	5.54	4.24	-	-	-	-
CO_2	-	-	-	0.95	-	-
CO_2-NH_3	-	-	-	-	2.25	1.89
R1270-NH_3	-	-	-	-	2.46	2.04
R170-NH_3	-	-	-	-	2.22	1.87

7.4.6 AEC and TEWI in scenario 0

AEC and TEWI values per tonnes of products for all the four applications of surimi and shrimp cold chain for Scenario 0 are presented in Figure 7.13 and Figure 7.14. AEC per tonne of product for surimi and shrimp cold chains are 569 kWh and 267 kWh. In surimi cold chain AEC for ice production is the highest with a share of 63.4% and that of chilled water is the lowest 7.6%. Similarly, for shrimp cold chain AEC for ice production is the highest with a share of 50.2%, and cold storage has the lowest 4.2%. The TEWI per tonne of product for surimi and shrimp cold chains are 9255 kgs of CO_2 equivalent and 4352 kgs of CO_2 equivalent, respectively. Percentage share of TEWI in both cold chains are almost similar to the AEC share. In *Scenario 0*, the net AEC for surimi and shrimp cold chains are estimated as 81.27 GWh and 192.25 GWh, respectively based on simulation. Similarly, the estimated net TEWI for surimi and shrimp cold chains are 1322×10^6 kgs of CO_2 equivalent and 3135×10^6 kgs of CO_2 equivalent, respectively.

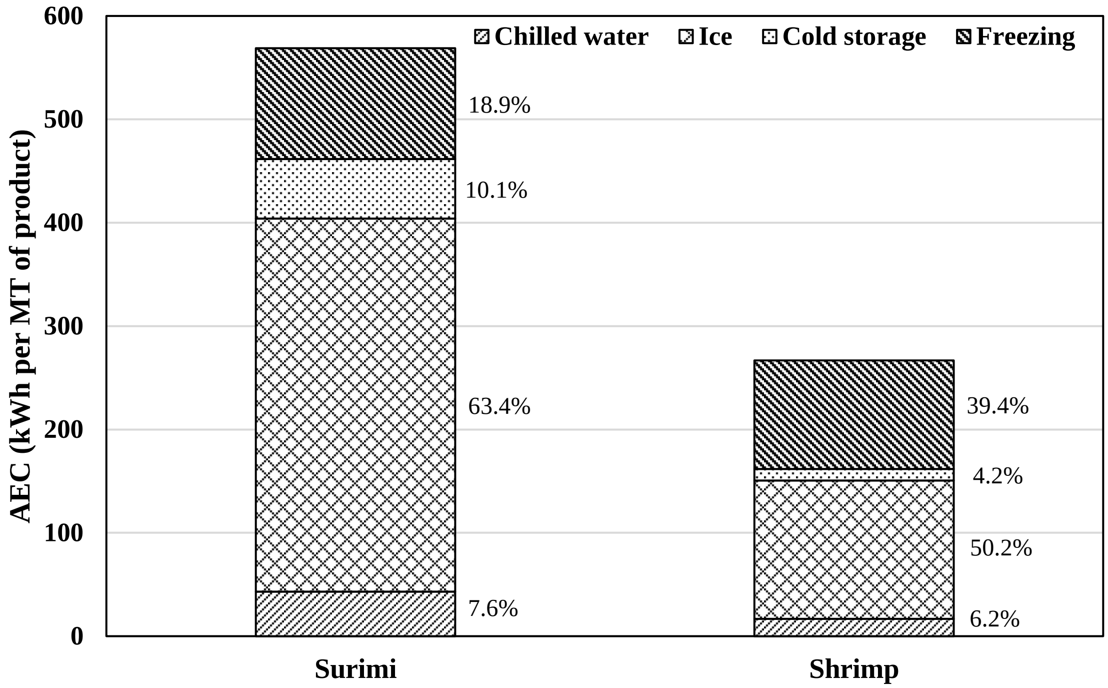


Figure 7.13: Energy footprint of supply side of cold chain in *Scenario0*

Due to higher cooling requirement in cold chain at all the stages, surimi has about double AEC and TEWI value per tonne of product. Ice production consumes the highest AEC and contribute the highest emissions. To reduce the energy consumption and emissions, cold chain stages where ice requirement are high need interventions. In surimi, the harvesting stage consumes the maximum ice, and hence on-board RSW system can be deployed which can reduce ice requirements and improve quality of catch (Saini *et al.*, 2019).

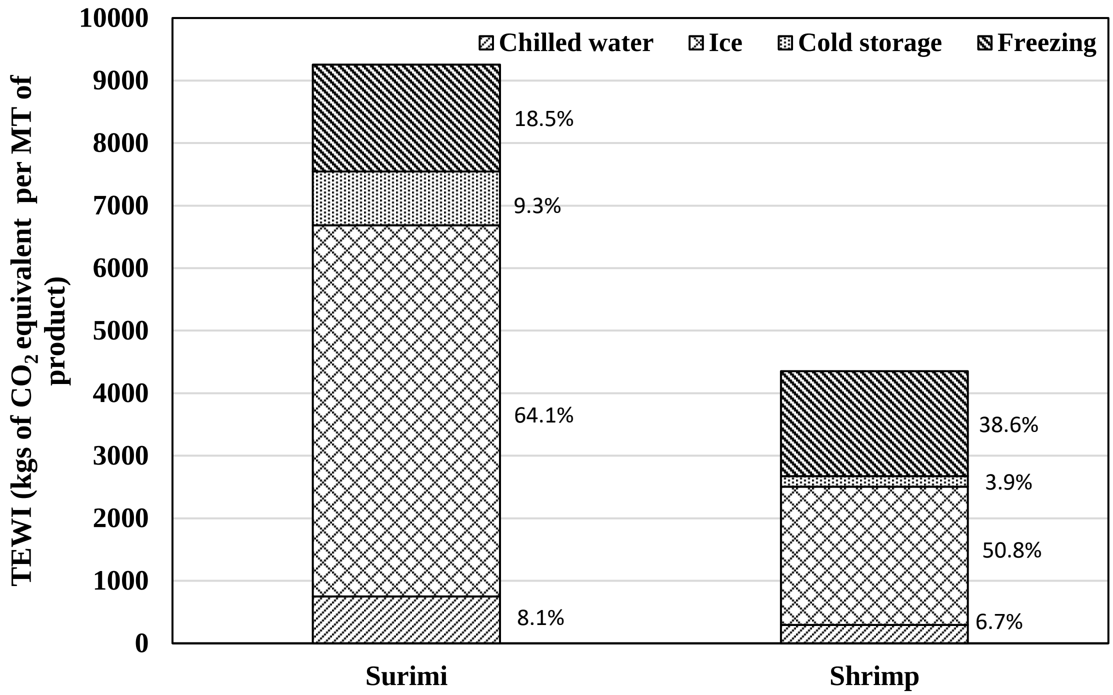


Figure 7.14: Environmental footprint of supply side of cold chain in *Scenario 0*

7.4.7 Comparison of AEC and TEWI for all scenarios

The estimated AEC and TEWI of the surimi and shrimp cold chains per tonne of product for all the three considered scenarios of refrigerant mix are presented in Figure 7.15 and Figure 7.16. Change in AEC between the *Scenario 0* and *Scenario 1* are negligible but AEC for Scenario 2 is about 6.6% and 3.5% less compared to *Scenario 0* for surimi and shrimp respectively (Figure 7.15). This reduction in AEC values are a result of high performance of the cascade system at the lower cooling temperature of cold storage and freezing applications. TEWI of surimi cold chain in *Scenario 1* and *Scenario 2* are about 7.5% and 22.5% less compared to *Scenario 0* (Figure 7.16). Similarly, TEWI of shrimp cold chain in *Scenario 1* and *Scenario 2* are about 5.7% and 20.1% less compared to *Scenario 0*. This change in TEWI are a result of the combined effect of change in GWP value of refrigerant mix and the AEC value in a particular cold chain.

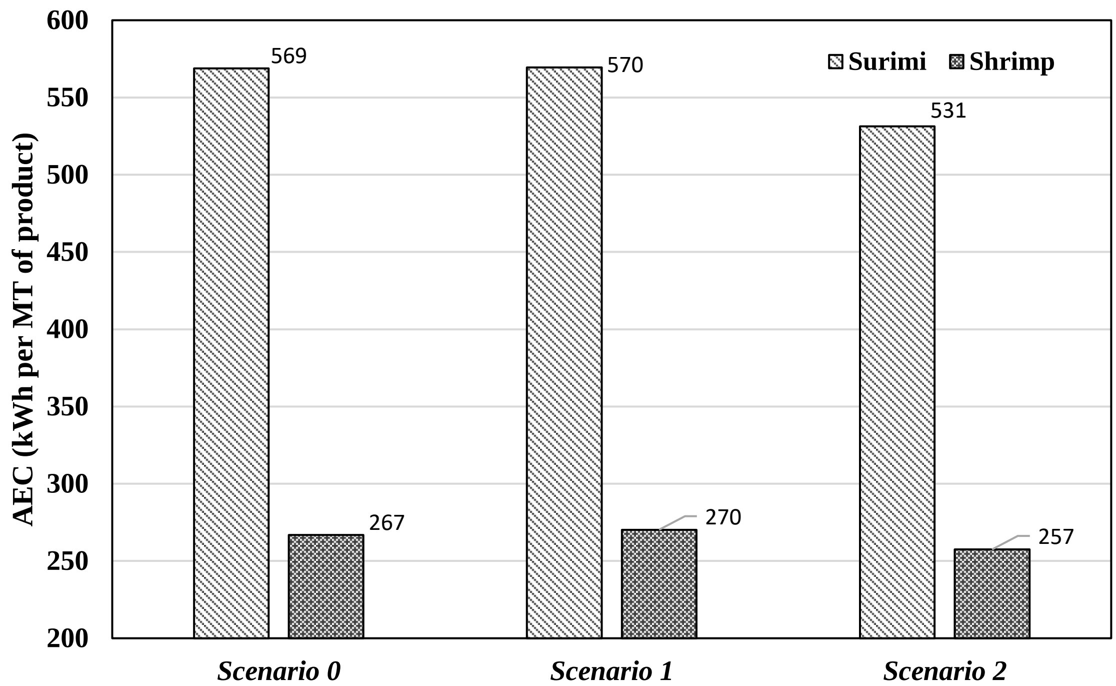


Figure 7.15: AEC per tonne of product for all three analysed *Scenario*

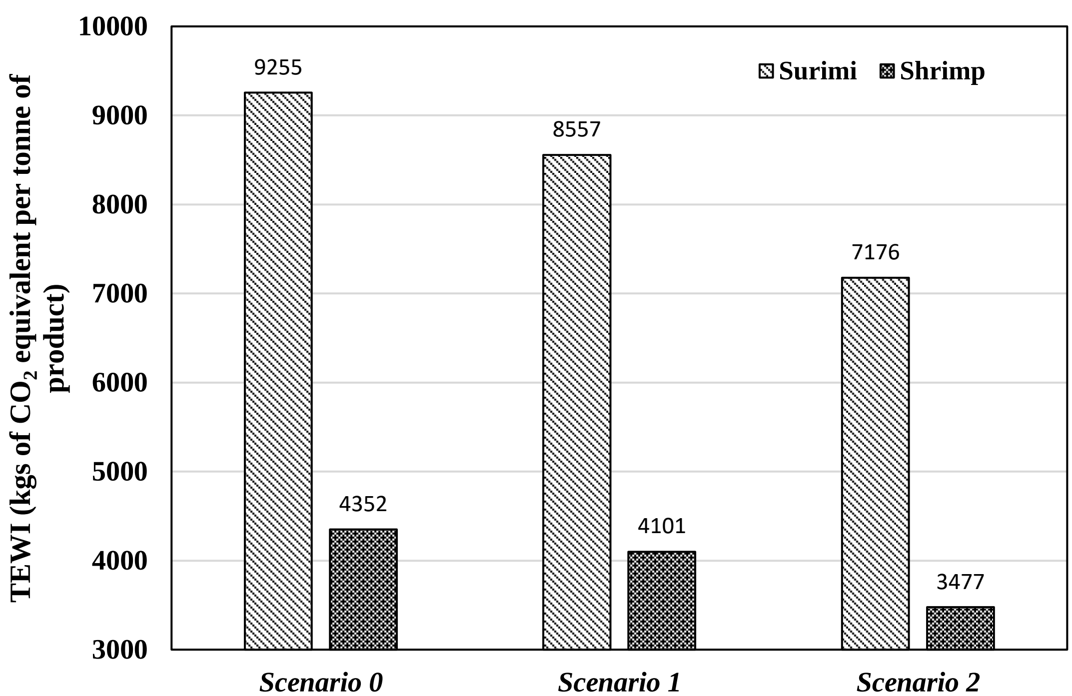


Figure 7.16: TEWI per tonne of product for all three analyzed cases

7.5 Challenges in Adoption of Potential Refrigerant

As emphasized in the introduction section, India has put forward concrete steps to combat the rising climatic changes by phasing down the HFC refrigerants. However, it is contextual to mention here that there are many barriers to the intended transition of the Indian refrigeration industry. Based on a survey conducted in India, Bhattacharyya, (2010) highlighted various constraints in adoption of low GWP natural refrigerants such as *legislation for standards and policies*, *lack of funding and financial support*, *non-availability of technology and safety measures, insufficient training for technicians*, and *absence of markets and marketing of appliances using natural refrigerants*. Sanguri *et al.* (2021) also reported a more recent study to ascertain the major barriers in adoption of low GWP refrigerants in India. They highlighted 20 major barriers and categorized them into nine levels and concluded that *lack of government incentives* is the most significant barrier faced by the industry followed by *lack of R&D facilities for the development of low GWP refrigerants*. G*overnment's inefficiency to enforce regulations uniformly*, and *ambiguity in regulations defined by the government* are the other prominent barriers. With no formal legislation mandating the use of natural refrigerants for refrigeration applications in the industrial, commercial and domestic sectors, the uptake of natural refrigerants becomes voluntary, and hence has not been very significant. The eventual transitions of the refrigeration market towards low GWP refrigerants have many other challenges related to cost, safety, and availability. Moreover, the levels of knowledge and confidence about performance of various alternative refrigerants in Indian context is lacking, which is also a formidable barrier in their adoption.

The commercial success or acceptance of a refrigerant depends on several factors other than the system's energetic performance. For example, one prominent concern in a commercial application is safety, including concern over toxicity and flammability. Flammability is one of the most common concerns about alternative refrigerants (ASHRAE, 2021; Yang *et al.*, 2021). All the hydrocarbons, NH_3, and a few HFOs are considered future refrigerants and viable substitutes but are categorized either as highly or mildly flammable. Therefore, to mitigate fire hazards, refrigerants with low volumetric displacement can be recomended for safer operation. Further, NH_3 has toxicity but is considered safe below 100 PPM level concentration.

The unavailability of a refrigerant and suitable compressor for the same along with other auxiliary components in a geographic location and their cost can also pose a serious challenge in its adoption. As part of this dissertation, a limited-scale survey was conducted covering India's north, south, and west zones (Delhi, Chennai, and Mumbai). Six out of the 15 refrigerants R41, R161, RE170, R1225ye(Z), R1234zf, N_2O are found *unavailable* across all zones, whereas R170, R1234yf and R1270, are found to have much higher costs compared to NH_3 (Table 7.4). While, the author did not find any regulatory restrictions, it was concluded that the reason for unavailability is a lack of demand. However, if demand for a certain refrigerant increases, availability should improve.

One significant concern in adopting new alternative refrigerants is the perceived uncertainties due to ever-changing laws and regulations across multiple countries. The refrigerant industry has progressed in terms of safety and efficiency regulations but still has many uncertainties regarding the end goal of GWP. Therefore, in this transition phase, the industry is waiting for long-term solutions instead of investing in short-term technologies. In addition, many new refrigerants and related technologies have only been tested on lab-scale prototypes; therefore, large capacity systems and availability of trained technicians for a particular technology are still lacking. Natural refrigerants, which have been used for many decades, have already known health and safety issues, therefore, will be easier to implement, when the thermodynamic and energetic properties are suitable for the purpose.

Table 7.4: Survey results on cost and availability of refrigerants in India

	Cost (INR per kg)			
Refrigerants	Supplier 1 (Delhi)	Supplier 2 (Mumbai)	Supplier 3 (Mumbai)	Supplier 4 (Chennai)
R41	NA	NA	NA	NA
R152a	340	310	310	260
R161	NA	NA	NA	NA
R170	13500	NA	NA	NA
RE170	NA	NA	NA	NA
R290	1460	NA	NA	1740
R1225ye(Z)	NA	NA	NA	NA
R1234yf	8200	9000	8500	10600

R1234ze(E)	1800	1650	NA	NA
R1243zf	NA	NA	NA	NA
R1270	9250	NA	NA	8700
R404A	460	450	450	480
CO_2	40	40	45	40
NH_3	70	65	75	70
N_2O	NA	NA	NA	NA

7.6 Conclusions

For the refrigeration demands in a seafood processing and storage application at high ambient temperature conditions, 60 low GWP refrigerant combinations are comparatively investigated for a multi-target-temperature evaporators based cascade refrigeration system (CRS). A comparative analysis of their energetic, environmental, and operational parameters are presented in this study. For the assessment of energy consumption and environmental impact in the entire surimi and shrimp cold chain in India, three scenarios are evaluated. Two of the scenarios are based on refrigerant mix use at present and forecasted partial adoption of low GWP refrigerants and a third scenario of shifting fully to natural refrigerants, are analyzed. Performance of these refrigerants are computed by simulating refrigeration system for particular application. AEC and TEWI are determined assuming ambient conditions of coastal sites for surimi and shrimp production. Based on the analytical study, the following conclusions were drawn:

- The performance of the CRS depends on the cascade condenser temperature *(*T_{MC}*)* for a particular refrigerant pair. To achieve maximum COP for any refrigerant pair, T_{MC} needs to be optimized for all operating conditions.
- With an increase in condenser temperature (T_{cond}), the optimum T_{MC} increases, whereas COP decreases. At T_{cond} 35 °C, the optimum T_{MC} for refrigerant pair CO_2-RE170 has the lowest (–6.8 ºC), and refrigerant pair R1270-R1234yf has the highest 7.9 ºC value.
- Among all the HTC refrigerants, RE170 showed the maximum COP closely followed by NH_3. Among LTC refrigerants, R1270 showed the maximum COP, and was above

the reference fluid R404A. The refrigerant pair R1270-RE170 showed the best overall COP of 3.18, while COP of CO_2-NH_3 was about 6.45% lower than the best.

- The AEC of the refrigerant pair follows a reverse trend of their COP value, thus, refrigerant pairs R1270-RE170 and R170-R1234yf, yielded the minimum AEC of 782.3 MWh and the maximum AEC of 895.8 MWh, respectively.
- Despite having marginally high GWP, refrigerants such as R41 in LTC and R152a in HTC yielded low TEWI because of their comparatively higher COP and lower AEC than various other refrigerants.
- At present, R404A dominates industrial refrigeration because of its high energetic performance and better safety features. TEWI of R404A is much higher than all other refrigerants compared, while R1270 additionally has better energetic performance. Hence R404A can be replaced by various competitive options.
- Among the HTC refrigerant choices, the NH_3 system has the lowest volumetric displacement, followed by R1270. Due to the lower refrigeration capacity, the HFOs have comparatively higher volumetric displacement. It is observed that within the LTC refrigerant choices, CO_2 has the lowest volumetric displacement in LTC compressors and is about 82% lower than the refrigerant R1270 which has the highest volumetric displacement.
- NH_3 in HTC and CO_2 in LTC showed the minimum volumetric displacements due to their suitable thermodynamic properties, implying most compact system with minimum charge can be designed with this refrigerant.
- Among the HTC refrigerants, NH_3 has the highest compression ratio, and R1270 has the lowest. Whereas among the refrigerants in LTC, R1270 has the highest compression ratio and R170 has the lowest. The high compression ratio and high specific heat ratio of NH_3 result in higher discharge temperature. At 35 ºC T_{cond}, the discharge temperatures of all the compressors are found to be below 80 ºC except for NH_3, which reaches up to 137.3 ºC.
- Advantage of using natural refrigerants in terms of AEC is found to be a reduction of

6.5% for surimi and 3.5% for shrimp cold chain. Furthermore, a reduction in TEWI is found to be 22.5% for surimi and 20.1% for shrimp cold chain.

The study is expected to help policy makers frame regulations to encourage the use of sustainable refrigerants in the near future. However, adoption of these natural refrigerants by the Indian refrigeration industry may not be smooth as there are various prominent barriers, some of which are discussed here. Major challenges identified in adoption of alternative refrigerants are their safety features, restricted availability of the refrigerants as well as components in a geographic region due to various reasons including current lack of demand.

CHAPTER 8 Conclusions and Future Work

8.1 Conclusions

The work deals with various cooling solutions to improve seafood cold chains at different stages. A review of the existing cooling methods at various stages of the seafood cold chain in India is carried out. The state-of-the-art cooling technologies and the transition in use of refrigerants are also tracked. Simultaneously, various research gaps were identified and highlighted for further study. Particular emphasis was placed on improving the performance of cooling methods and systems through the use of low cost technologies for use in the Indian scenario. An on-board compensatory cooling system for the fish holds in small motorized boats is designed and analyzed. Further, for an integrated surimi processing and storage plant, a cascade refrigeration system (CRS) is designed in place of a conventional multi-stage cooling systems. The benefits that CRS provides to the seafood processing plants and cold storage include isolation of food toxic refrigerant from possible direct contact with seafood, reduced energy consumption, improved exergy efficiency and reduced emissions to the environment. An economic analysis of the proposed systems is also presented to demonstrate the economic viability of the proposed system configurations. Furthermore, the thesis presents a study on the analysis of various low global warming potential (GWP) refrigerants including CO_2 and NH_3, and their relative energy and environmental benefits over conventional systems, for both surimi and shrimp cold chain. Towards these above stated objectives, in-depth studies were carried out following well-defined methodologies. Key conclusions and highlights from that have been drawn from *Chapters 4* and *5* are:

- An on-board novel compensatory refrigeration system is designed to compensate for the heat ingress into the insulated storage compartments and ensure storage near at 0 °C. Diesel engine exhaust of the fishing boat serves as heat source for the absorption cooling device and sea water serves as heat sink. The payback period of the vapour absorption refrigeration (VAR) system is computed to be around 1 year 4 months.

- Three novel CO_2-NH_3 CRSs; each having four evaporators have been proposed for supporting the typical cooling demands in various seafood processing plants operative in the tropical climate, taking Mumbai coastal environmental data as reference.
- The maximum coefficient of performance (COP) improvement from the proposed CRS attained is about 11.5% for the surimi processing plant along with an 8.3% reduction in annual energy consumption (AEC) compared to the multi-stage NH_3 system considered as a baseline system. The maximum COP improvement in fish fillets and shrimp/prawn industry obtained with specifically designed CRS was 16.5% and 20.3% along with a 7.5% and 13.2% reduction in AEC respectively over the baseline system.
- The proposed CRS also shows about 14% less exergy destruction compared to the baseline system. Component wise, the compressors contribute to the largest exergy destruction, with an average value of 45%, while the evaporators contribute the least with a modest average value of 4%.

The foremost conclusions drawn from the *Chapter 6* of the thesis are:

- The study compared the performance of a proposed CO_2-NH_3 CRS with conventional systems having working fluids as various refrigerants such as R22, R404A and NH_3. The CRS showed about 15.0%, 46.8% and 5.3% higher COP than conventional. The advantage further gets enhanced with decrease in the freezer temperature leading to superior performance of the CRS at lower temperatures.
- The CRS exhibited lower environmental emissions of 26.8%, 44.3% and 3.2% compared to the conventional R22, R404A & NH_3 systems owing to higher performance and lower GWP of refrigerants.
- The CRS shows lower life cycle cost (*LCC*) compared to the R22 and R404A system, but higher *LCC* compared to NH_3 only system.
- Effect of integration of an internal heat exchanger (IHX) and a gascooler with the CRS are further analyzed towards improvement of system performance. The minimum improvement in COP of the proposed system was 17% higher compared to the baseline system when augmented with IHX and gascooler.

The key conclusions drawn from *Chapter 7* are:

- Applicability of various low GWP refrigerants in CRS configuration was studied. The refrigerant pair R1270-RE170 showed the best overall COP of 3.18, while COP of CO_2-NH_3 was about 6.45% lower than the best.
- Despite having marginally high GWP, refrigerants such as R41 in LTC and R152a in high temperature circuit (HTC) yielded lower TEWI owing to comparatively higher COP and lower AEC than other refrigerants.
- Although R404A dominates industrial refrigeration because of its perceived high energetic performance and better safety features. TEWI of R404A is found to be much higher than all other refrigerants compared, while R1270 additionally has better energetic performance. Hence R404A can be replaced by various competitive options.
- In HTC, the NH_3 has the lowest volumetric displacement, followed by R1270. Due to the lower refrigeration capacity, the Hydrofluoro Olefins (HFOs) have comparatively higher volumetric displacement. CO_2 has the lowest volumetric displacement in low temperature circuit (LTC) compressors.
- Advantage of using natural refrigerants in terms of AEC is found to be a reduction of 6.5% for surimi and 3.5% for shrimp cold chain. Furthermore, a reduction in TEWI is found to be 22.5% for surimi and 20.1% for shrimp cold chain.
- Various prominent barriers in adoption of the proposed systems and refrigerants are also presented. Major challenges identified in adoption of alternative refrigerants are their special safety features, limited availability of both refrigerant fluid and components in a geographic region due to lack of demand. Additionally, the lack of ready market of system components for CO_2, lack of trained technicians in this technology at present lead to high overall establishment cost. Moreover, the confidence of vendors about performance of CO_2-NH_3 CRS in Indian context is low, which is also a formidable barrier in its large scale adoption.

This dissertation embodied a comprehensive research effort and the results provided included some very important imperatives for the relevant industry as well as for environmental sustainability initiatives.

8.2 Future Work

This has been a structured endeavor of exploring various environmentally friendly refrigerants and energy efficient refrigeration systems which, if taken forward towards implementation, can have extensive contribution towards making Indian seafood cold chain green and clean. Theoretically, the proposed CO_2-NH_3 CRS offers many advantages over the conventional system. However, a few practical challenges are also associated with the design and operation of such a system and its component selection. Additional future studies can be carried out related to efficient defrost system for such configurations, mitigation of standstill pressure issues for CO_2 system for high ambient temperature operation, compressor lubricating oil separation and contamination issues etc. Overall, future research needs to be focused on experimental investigation of CO_2-NH_3 CRS in the warm ambient conditions of India.

A few other potential future works can be on the following:

- Vapour absorption refrigeration system utilizing waste heat is a matured technology and demonstration of this technology on motorized small fishing boats.
- Performance optimization and correlation development among the COP, T_{MC}, evaporation temperature and refrigeration load at various evaporators in a CRS can be attempted, this will make scaling of the system easier.
- Seafood processing industry produces a large amount of waste cold water. In addition, Also, refrigeration system rejects substantial amount of heat at the condenser. Suitable integration of various waste heat and cold energy streams can help improve the overall energy performance of the processing plants.

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List of Publications

International Journals

1. **Saini, S. K.**, Dasgupta, M. S., Widell, K.N., & Bhattacharyya, S. *Comparative analysis of a few novel multi-evaporator CO_2-NH_3 cascade refrigeration system for seafood processing and storage.* **International Journal of Refrigeration** (2021), 131, 817-825. [SCI; IF: 4.14; H-Index: 125]. https://doi.org/10.1016/j.ijrefrig.2021.07.017
2. **Saini, S. K.**, Dasgupta, M. S., Widell, K.N., & Bhattacharyya, S. *Energetic, environmental and economic assessment of multi-evaporator CO_2-NH_3 cascade refrigeration system for seafood application.* **Journal of Thermal Analysis and Calorimetry** (2022). [SCI; IF: 4.76; H-Index: 101]. https://doi.org/10.1007/s10973-022-11619-7
3. **Saini, S. K.**, Dasgupta, M. S., Widell, K.N., & Bhattacharyya, S. *Comparative investigation of low GWP pure fluids as potential refrigerant options for a cascade system in seafood application*. **Mitigation and Adaptation Strategies for Global Change** (2022), 27, 57. [SCI; IF: 3.93; H-Index: 76]. https://doi.org/10.1007/s11027-022-10036-3
4. Vaishak, S., Singha, P., Dasgupta, M. S., Hafner, A., Widell, K., Bhattacharyya, S., **Saini, S. K.**, ... & Ninan, G. (2023). *Performance analysis of a CO2/NH3 cascade refrigeration system with subcooling for low temperature freezing applications.* **International Journal of Refrigeration.** https://doi.org/10.1016/j.ijrefrig.2023.05.013

International Conference papers

1. **Saini, S.K.**, Dasgupta, M.S., Widell, K.N., Bhattacharya, S. *Thermal and economic analysis of an on-board compensatory refrigeration system for small fishing boats*. **25th National and 3rd International ISHMT-ASTFE Heat and Mass Transfer Conference (IHMTC-2019)**, Roorkee, India. http://dx.doi.org/10.1615/IHMTC-2019.1690

2. **Saini, S.K.**, Dasgupta, M.S., Widell, K.N., Bhattacharya, S. *Performance evaluation of a multi-evaporator CO_2-NH_3 cascade refrigeration system with IHX for seafood processing industry*, **14th IIR-Gustav Lorentzen Conference on natural refrigerants (GL-2020)**, Kyoto, Japan. http://dx.doi.org/10.18462/iir.gl.2020.1089
3. **Saini, S.K.**, Dasgupta, M.S., Widell, K.N., Bhattacharya, S. *Comparative study of exergetic and economic analysis of multi-evaporator NH_3 and CO_2-NH_3 CRS for a seafood processing plant*. **18th International Refrigeration and Air Conditioning Conference (Herrick Conf.-2021)**, Purdue, USA.
4. **Saini, S.K.**, Dasgupta, M.S., Widell, K.N., Bhattacharya, S. 2022. *Assessment of benefits of employing natural refrigerants in seafood cold chain in India*. **15th IIR-Gustav Lorentzen Conference on natural refrigerants (GL-2022)**, Trondheim, Norway. http://dx.doi.org/10.18462/iir.gl2022.0190

National Conference papers

1. **Saini, S.K.**, Dasgupta, M.S., Widell, K.N., Bhattacharya, S. *Natural refrigerant based cascade system for seafood application.* **48th National conference on Fluid Mechanics and Fluid Power (FMFP-2021)**, Pilani, India.

Book Chapters / Technical Reports

1. Dasgupta, M.S., Routroy, S., Bhattacharyya, S., Sultan, A., **Saini, S.K.**, Gupta, K., Kaushik, N., Widell, K.N., Tveit, G.M. and Thakur, M. *ReValue project Report– Deliverable 1.1- Value stream map and supply chain interdependencies in India - Surimi case*, (2020). **ISBN 978-82-7174-375-8**.
2. Dasgupta, M.S., Routroy, S., Bhattacharyya, S., Sultan, A., **Saini, S.K.**, Gupta, K., Kaushik, N., Widell, K.N., Tveit, G.M. and Thakur, M. *ReValue project Report– Deliverable 1.2-Logistics and Cold Chain Management Concepts*. OC2020 A-094 (2020). **RCN 978-82-7174-393-2**.

3. Dasgupta, M.S., Routroy, S., Bhattacharyya, S., Sultan, A., **Saini, S.K.**, Widell, K.N. and Thakur, M. *ReValue project Report–D1. 3: Report on energy efficient refrigeration systems-Surimi case* (2020). **ISBN 978-82-7174-394-9**.

Author Biography

Santosh Kumar Saini

Santosh Kumar Saini is a PhD scholar in Mechanical Engineering Department of BITS Pilani India, his area of research is Refrigeration and seafood cold chain. He joined the PhD program of BITS Pilani in August 2018. He received his MTech degree in Renewable Energy from Centre for Energy and Environment, MNIT Jaipur in year 2018 and his BTech degree in Mechanical Engineering from Rajasthan Technical University, Rajasthan in year 2015.

Santosh has research interest in thermal science & engineering, refrigerants, refrigeration & air-conditioning and cold chains, waste heat recovery, building energy simulation, renewable energy integration and has competencies in HVAC & refrigeration system conceptualization, design, modelling & simulation, experiments and data analysis. He has experience of data collections through surveys and site visits, conduct effective literature reviews and has good knowledge of renewable energy integrations with buildings and passive heating & cooling strategies for building. He is also proficient in environmental emission estimation and life cycle assessment.

Santosh has published more than 10 research articles in various peer reviewed international conferences and journals. He has successfully executed assigned role in two international projects: CBERD & ReValue and has received two travel support grants from SERB, and DBT, Govt. of India.

Biography of Supervisors

Prof. Mani Sankar Dasgupta

Prof. M S Dasgupta has about thirty years of teaching and research experience at BITS Pilani, India at Mechanical Engineering Department. He has mentored and motivated several students who subsequently grew to be among the highest achievers across the globe, both in corporate world and in research. Prof. Dasgupta served as Department Head of Mechanical Engineering

For 8 years in two double terms and as Faculty in-charge of Campus placement for 8 years. He is currently serving as Coordinator IQAC for the University and had served as the same capacity earlier for 2 years. He had participated in 10 NAAC peer team visits in various capacities.

Prof. Dasgupta has primary research interest is environmentally friendly technologies, specifically in CO_2 Trans-critical systems and natural refrigerants, has worked in several funded research projects including two in partnership with SINTEF Ocean Norway and NTNU Norway. He has more than 75 publications in peer reviewed international Journals and conferences and have given keynote address at various conferences in India and abroad.

Prof. Souvik Bhattacharyya

BME (Jadavpur), MS (Cincinnati), PhD (Texas A&M)

Souvik Bhattacharyya is the Vice Chancellor of BITS Pilani since 2016. Following a post-doctoral research stint at Texas A&M, he joined IIT Kharagpur in 1991 as a faculty member where he has been a Professor (2003-21), Dean (2009- 12) and Deputy Director (2013-16). He also served as the Vice-Chancellor of Jadavpur University and held faculty position at University of Canterbury, New Zealand during 1998-2000. He has published over 250 research articles, is a co- author of the Heat Transfer text with J P Holman (McGraw Hill) and adaptation author of Thermodynamics by Borgnakke and Sonntag (Wiley India).

Prof Bhattacharyya has been a member of the Editorial Boards of several reputed international journals. He is currently a member of the Expert Advisory Committee of DST Advanced Manufacturing Technologies Program and IIT Jodhpur Senate. He has been the Chairperson of the SERB TARE Expert Committee and SERB TARE Screening Committee, member of SERB Civil & Mechanical PAC, DST Fast-Track Committee, Research Council of CSIR-CGCRI, and the Court of Indian Institute of Science, Bangalore.

Prof Bhattacharyya is a co-Chair of the FICCI Higher Education Committee and the Member-at-large of the Executive Council, Shastri Indo-Canadian Institute (SICI).

He is a recipient of the Outstanding Teacher Award of Indian National Academy of Engineering (INAE) in its inaugural year (2013) and an elected Fellow of INAE, National Academy of Science, India (NASI) and West Bengal Academy of Science and Technology (WAST).

Dr. Kristina Norne Widell

Dr. Kristina Norne Widell is a senior research scientist at SINTEF Ocean in Trondheim, Norway, working in the department of fisheries and new bio-marine industry. SINTEF is one of Europe's largest independent research institutes. Kristina received her Ph.D. degree in Energy and process engineering from the Norwegian University of Science and Technology (NTNU). The title of her thesis was "Energy efficiency of freezing tunnels – towards an optimal operation of compressors and air fans", in which she explored industrial fish freezing systems.

Her research addresses mainly processing technology and systems for food industry, especially related to refrigeration and food cold chains.

Kristina is leading the project CoolFish, which develops technologies and concepts for more integrated, energy-efficient and environmentally friendly cooling, freezing and heating onboard fishing vessels. The transition from climate and environmentally harmful to natural working fluids is a central mission in the project. She is also the coordinator of the EU funded project ENOUGH, which will provide a range of tools helping the European food industry to reduce their emissions, which is within the EU 'Farm to Fork' strategy. The project has 30 European partners from academia, industry and international organizations. Another topic that is central in her work is thermal energy storage. It is a topic that gets more attention now, when the electricity prices are higher and more electricity comes from renewable/intermittent sources.

She is also working with increasing the knowledge transfer between research and industry, both nationally and internationally. Food product quality is a central value in many of the projects she participates in and this must be linked with energy demand and resource utilization. High food quality and good procedures are necessary to avoid food loss and waste.