Trans-critical CO_2 System for Warm Climate: An Evaluation of System Configurations and Scroll Expander

THESIS

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Simarpreet Singh

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Prof. M.S. Dasgupta

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Certificate

This is to certify that the thesis entitled **"Trans-critical CO_2 System for Warm Climate: An Evaluation of System Configurations and Scroll Expander"** and submitted by **Mr. Simarpreet Singh**, ID No. **2013PHXF0001P** for award of Ph. D. of the institute embodies original work done by him under my supervision.

Prof. M. S. Dasgupta
Department of Mechanical Engineering,
Birla Institute of Technology and Science Pilani,
Pilani Campus, Rajasthan, India (333031)

Date:

Abstract

CO_2 trans-critical systems, in recent years, have attracted attention for a wide variety of refrigeration applications. Natural refrigerants are perceived to be potential permanent solution in the face of progressive restrictions being imposed upon the use of synthetic refrigerants. Revival of interest in natural working fluids for refrigeration and air conditioning application in the early 1990's is credited to environmental concern over ozone depletion potential (ODP) and later, on global warming potential (GWP) of the various synthetic refrigerants like CFCs, HCFCs and HFCs. Natural refrigerants such as air, water, ammonia, CO_2, isobutene, propane etc. are economic, ecologically safe, and have zero ODP and low GWP. CO_2 (R744) in particular has unit GWP. COP of CO_2 system decreases drastically for high temperature application. Reason for low COP are high pressure operation beyond critical point (7.1MPa & 31.1°C) and large throttling losses associated with expansion from a high super-critical pressure to a sub-critical pressure. There are, however, unique opportunities associated with the high ambient temperature operation. A large number of modifications are feasible and have been explored beyond the basic trans-critical CO_2 refrigeration cycle to improve the performance of the system and also to adapt to the various working conditions. Researchers have explored working on several application areas like automotive air conditioning, heat pump water heater, commercial refrigeration, large commercial space cooling etc. Use of work recovery expander as a replacement of expansion valve is emerging as a promising option to achieve commercial success. The main objective of this study is to identify optimum operating and design conditions for work recovery expander at high ambient conditions in order to improve the overall performance of CO_2 system operating in trans-critical mode.

During this study, literature on work recovery expander designs in CO_2 trans-critical system were tracked for the last 15 years to find a research trend. The trend of research is found to be intermittent & scattered across nineteen universities worldwide. The qualitative data obtained from literature survey is further put through a set of selected multi attribute decision making methods (MADM). Out of the various methods, the ones identified as most promising are scroll expander, piston expander and screw expander.

A comparative study is undertaken on six prominent modifications and their suitability for high ambient temperature operation (35-55°C) is explored. These modifications are basic trans-

critical CO_2 refrigeration system incorporating internal heat exchanger (IHX), system with work recovery expander, IHX with expander, inter-cooler (IC), flash gas bypass (FGB) and flash gas inter-cooler (FGI). Applications like chiller, domestic refrigeration and air cooling are explored. In general, it was observed that the cycle modifications have a positive effect on the overall COP of the system. However, to comprehend practicability of these modifications for the three application areas, a few other parameters which affect design and operation are also included in the study. These are compressor discharge pressure and temperature, mass flow rate, inter-stage pressure for multi-stage operation and exergy destruction. Effect of real time constraints like approach temperature, pressure drop in gas cooler, compressors efficiency, degree of superheat, expander efficiency and effectiveness of intermediate heat exchanger are also explored.

In warm weather, the compressor needs to be operated with higher pressure ratio to reject heat to the surroundings. This implies enhanced power requirement and reduction in coefficient of performance (COP). This challenging situation, however, provides an opportunity of energy recovery from the system in the form of heat and work. A work recovery expander can be employed to improve the overall performance of the system. Scroll work recovery expander was selected for study based on its unidirectional, smooth & continuous rotary operation which imparts low irreversibility, high reliability, robustness and ability to handle high pressure ratio. Performance of a trans-critical CO_2 refrigeration cycle equipped with a scroll work recovery expander was investigated using a semi-empirical model. Based on year- round ambient temperature data, at New Delhi (India), it was observed that, about 20% of the total energy consumed is recovered with the help of scroll expander during trans-critical operation. An economic analysis is carried out for installation of work recovery scroll system. Total payback period (PBP) for scroll expanders is found to be about 2 years or less, which is encouraging. Total Equivalent Warming Impact (TEWI) of the system is also computed and compared with that of a conventional trans-critical CO_2 refrigeration system and a system running on low GWP hydrofluoroolefin (R1234yf) refrigerant. Trans-critical CO_2 system equipped with work recovery scroll expander appears as a promising option, as the same has about 15% lower TEWI compared to conventional trans-critical CO_2 system.

Field data from a medium scale ammonia based milk refrigeration plant is collected and a trans-critical CO_2 refrigeration system is conceptualized to replace the ammonia based refrigeration plant. Another scheme evaluated was a CO_2 based heat pump system for waste heat utilization within the plant and there by improve overall COP. A thermodynamic model of the refrigeration system was built & simulated for year-round field data and the performance of both were compared. An economic analysis is also carried out to establish feasibility of the proposed systems.

Recovery of expansion work in high pressure refrigeration system has been viewed as a workable solution but has not become common yet. Some of the barriers are relatively low efficiency of work recovery device, high cost of expander and issues associated with gainful utilization of the recovered energy in site. An expander was developed for the purpose of experimental study through internal modification from a commercial scroll compressor with very less additional cost involvement. The mechanical working process for conversion is also discussed. In order to examine performance of the fabricated scroll expander, it is tested in an open loop test rig. Influence of the various operating parameters like mass flow rate, supply pressure, pressure ratio and rotating speed on the overall performance of the system in sub-critical mode are examined. A back propagation artificial neural network (ANN) was also trained using experimental data. The trained ANN may be used for predicting behavior of the work recovery expander for various pressure ratio.

Student Declaration

I hereby declare that the research work entitled **"Trans-critical CO_2 System for Warm Climate: An Evaluation of System Configurations and Scroll Expander"** submitted by me for the partial fulfillment of the requirements for the degree of Doctor of Philosophy (Ph. D) in the Department of Mechanical Engineering, BITS Pilani, Pilani Campus under the supervision of Prof. M.S. Dasgupta.

I further declare that to the best of my knowledge this is my original work and has not been submitted earlier to any other university or institution.

Place: Pilani, Rajasthan

Date:

(Simarpreet Singh)

Signature of the candidate

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Simarpreet Singh

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Nomenclature

A	Area	(m^2)
c	Specific heat	(J $kg^{-1}K^{-1}$)
E	Energy consumption	(kWh)
EQP	Equipment price	($)
G	Stored product	(kg)
h	Specific enthalpy	(kJ kg^{-1})
I	Irreversibility	(kW)
IC	Total installation cost	($)
$INST$	Installation price	($)
Q	Heat transfer coefficient	(W m^{-2} K^{-1})
l	Leakage rate	(%)
$\dot{m}$	Mass flow rate	(kg s^{-1})
MC	Maintenance cost	($)
MT	Million tones	
n	Number of years	
N_l	Lightening power	(W)
OP	Operational cost	($)
p	Pressure	(MPa)
pr	Pressure ratio	
r	Radius	(mm)
R	Coefficient of determination	
RC	Repair cost	($)
RR	Reduction rate	
s	Entropy	(kJ kg^{-1} K^{-1})
t	Time	(h)
t_l	Average working time of lightening	(h day^{-1})
T	Temperature	(ºC)
W	Compressor work	(kW)
$\widetilde{W}$	Fuzzy weight vector	

$\tilde{X}$	Matrix	
X	Material thickness	(m)
z	Daily number of air exchanging	

Greek symbol

α	Recycling factor	(%)
α_o	Coefficient of heat transfer outside air	($W\ m^{-2}\ K^{-1}$)
α_i	Coefficient of heat transfer inside air	($W\ m^{-2}\ K^{-1}$)
β	Indirect emission	($kg\ kWh^{-1}$)
l	Conductivity	($S\ cm^{-1}$)
$\prod$	Preference index	
φ	Outranking relation	
$\eta, \mathcal{E}$	Isentropic efficiency	
Υ_i	Thermal conductivity of insulation	($W\ m^{-2}\ K^{-1}$)
II	Second law	
ρ	Density	($kg\ m^{-2}$)

Subscripts and superscripts

ad	Adapted
amb	Ambient
b	Base circle
bc	Basic cycle
cond	Condenser
comp, c	Compressor
e	End, Expander
el	Electrical
ev, evap	Evaporator
ex	Exhaust
exp	Expansion
fgb	Flash-gas bypass
fgi	Flash-gas intercooler

gc	Gascooler
gm	Geometrical mean
i	Inner, Index
ic	Inter-cooler
ih	Infiltration heat
ihx	Internal heat exchanger
int	inter-stage
is	Isentropic
leak	Leakage
max	Maximum
meas	Measured
ms	Multi-stage
n	Matrix size
o	Outer
ph	Product heat
rot	Rotation
rv	Volumetric ratio
s	Start
sh	Shaft
sim	Simulation
ss	Single stage
su	Suction
sys	System
t	Thickness, Total
th	Transmission heat
thru	Throat
uh	Unexpected heat
v	Volumetric
w	Wall

Acronyms

ANN	Artificial neural networking
APH	Air-pre heater
C	Criterion
CC	Closeness coefficient
CFC	Chlorofluorocarbon
CI	Critical index
CIP	Clean in place
CO_2	Carbon dioxide
COP	Coefficient of performance
CR	Consistency ratio
D	Design
E	Electricity
EEV	Electronic expansion valve
FAO	Food and agriculture organization
FAHP	Fuzzy alternative hierarchy process
FGB	Flash gas bypass
FGI	Flash gas inter-cooler
GWP	Global warming potential
HCFC	Hydrochlorofluorocarbons
HFC	Hydroflourocarbons
HFO	Hydrofluoroolefin
HSD	High speed diesel
HT	High temperature
HVAC	Heating, ventilation and air-conditioning
IC	Inter-cooler
IHX	Internal heat exchanger
IT	Intermediate temperature
L	Leakage
LT	Low temperature

M	Maintenance
MADM	Multi-attribute decision making
N	Noise
ODP	Ozone depletion potential
OT	Overlap temperature
PBP	Payback period
PF	Preference function
R	Refrigerant
RAC	Refrigeration and air-conditioning
RI	Relative index
S	Separation measures
TEWI	Total equivalent warming impact
TR	Tones of refrigeration
V	Weighted normalized decision
WPH	Water-pre heater
WR	Work recovery

CHAPTER 1

Introduction

This chapter presents motivation and overview of my thesis work carried out and background theories along with work objective. Key challenges associated with the applicability of CO_2 in warm climate and a few strategies to overcome those challenges are also included. The chapter also presents how the thesis work is broken down into various chapters.

1.1. Motivation

Built environment is one of the major requirements of healthy and comfortable living condition and its demand is ever increasing. The energy demand arising out of this and increasing environmental foot print of the same is a matter of concern globally. Revival of interest in natural working fluid for refrigeration and air conditioning applications in recent year is triggered by environmental concern of ozone depletion potential (ODP) and global warming potential (GWP) of the various synthetic refrigerants like chlorofluorocarbon (CFCs), hydro fluorocarbon (HFCs) and hydro chlorofluorocarbon (HCFCs), being used in HVAC systems. CFCs have been phased out by 1996 in developed countries and is ear marked for abolition by 2010 in developing countries. Initial alternative to CFCs included HCFCs but they were also being found to be having harmful effect due to high GWP. The HCFCs are targeted to be phased out internationally by the year 2030. At present, HFCs as a refrigerant are found to be the leading replacement of CFC and HCFC. However, even the most popular HFCs are found to decompose at low pressure and in the presence of sunlight at tropospheric layer and contribute to acid rain and poisonous substances. So, there is a need to look for safer alternatives.

The synthetic refrigerants thus have inherently non-ecofriendly characteristics. A large number of studies are in progress at present to improve technology around natural refrigerants like air, water, ammonia, CO_2 and hydrocarbons that are naturally present in biosphere and ecologically safer. CO_2 as refrigerant received a boost as trans-critical cycle in 1991 by acclaimed Professor Gustav Lorentzen. CO_2 has many favorable thermo-physical properties and is viewed by a prominent section of researchers as a viable and long term alternative to synthetic refrigerants [1]. Benefits of CO_2 as a natural refrigerant are: high volumetric refrigeration capacity, non-flammability, non-toxicity, availability and low cost. The disadvantage and challenges includes low COP at high ambient temperature and issued related to high pressure handling during super-critical operation.

1.2. Natural Refrigerants

Natural refrigerants are those that are available in our biosphere. They include a range of organic and inorganic compounds suitable for use in a variety of HVAC applications. Common natural refrigerants are ammonia, natural hydrocarbons, water, carbon dioxide, etc. They typically have very low or zero OPD and GWP. Natural refrigerants are perceived by business houses as potential permanent solution in the face of progressive phase out of synthetic refrigerants.

Ammonia (NH_3)

Ammonia has been in consistent use as a refrigerant since the 1800s. Ammonia is very efficient alternative to fluorocarbon refrigerants in a number of applications, and is considered as prominent option in some areas such as ice plants, industrial refrigeration, etc. The environmental properties of ammonia are also favorable. Concerns do exist regarding the safe storage of ammonia due to its toxicity and flammability and additionally due to its pungent odor. Ammonia also has low thermal capacity so the system become bulky. The use of ammonia as refrigerant is not recommended for space that has high population density. Advancement have been reported in recent years to minimize these risks by designing low charge ammonia system. Some refrigeration systems have employed cascade systems with other refrigerants in order to reduce and isolate the ammonia charge. Around the world, large ammonia systems are being subjected to increasingly stringent safety regulations.

Hydrocarbon

Hydrocarbon based refrigerants are obtained from natural sources during oil and natural gas extraction. Hydrocarbon refrigerants have excellent environmental, thermodynamic, and thermo-physical properties; however, they are highly flammable. In the past, hydrocarbon refrigerants have had limited applications primarily within the petrochemical industry to provide industrial chilling and process refrigeration. With the phase-out of the CFCs, hydrocarbon refrigerants have come into the focus more prominently. The hydrocarbons most commonly used as refrigerants are: methane (R50), ethane (R170), propane (R290), butane (R600), isobutane (R600a), ethylene (R1150) and propylene (R1270).

Water

Water is the safest natural refrigerant. Water is widely used as a refrigerant in high temperature vapor absorption refrigeration system such as lithium bromide/water ($LiBR/H_2O$) absorption chillers where water is the absorbent and lithium bromide is used as an absorbent. The challenge for absorption chillers is that even a double effect absorption cycle has a COP only slightly greater than 1. It is far less common to find water in use within a vapor compression refrigeration system, although it does have one particularly noteworthy attribute; its thermo-physical property enable it to achieve high COP. Water systems have a number of technological characteristics that have, limited its growth in industry. First, the operating pressures for water based refrigeration systems are extremely low approaching a near vacuum making their continued operation free of contaminants (air) difficult. Second, the density of water vapor is extremely low; thereby, necessitating compressors capable of processing extremely high volumetric flow rates. Lastly, water is inherently limited to refrigeration applications for high temperature only. However, research and development continues in the field of application chillers in large sizes that could one day become a significant part of the chiller and ice water markets.

Carbon Dioxide (CO_2)

CO_2 has been used in the refrigeration industry since 1860s but was completely abandoned later due to introduction of synthetic refrigerants. Interest in the use of CO_2 as a refrigerant resurfaced in the 1990s, due to introduction of regulation to phase out ozone depleting CFC refrigerants. CO_2 is cheap & is easily available refrigerant and many experts recommends it for having a unique set of properties which make it an ideal refrigerant in certain conditions. In addition to its favorable thermodynamic properties, CO_2 is non-toxic, and it carries an A1 safety classification. CO_2 is also heavier than air and therefore any contamination is retained. It is this property that can cause some safety issues, if enough CO_2 builds up in an enclosed space it will begin to displace oxygen and can cause asphyxiation if anyone is present within the space.

Characteristic properties of CO_2 with some other traditional refrigerants are tabulated in Table 1.1.

CO_2 systems typically operates at 5 to 10 time's higher pressure than fluorocarbons and other refrigerants, when adopted in vapor compression system. Due to high pressure operation, some design challenges are present and also the work required to drive the compressor, tend to reduce the COP. Application of CO_2 in cascade system with other refrigerants have also been explored extensively to reduce a few of the disadvantages.

Table 1.1: Properties of CO_2 and a few traditional refrigerants.

Fluid	Cr. Temp. (°C)	Critical Pressure (bar)	Sat. Pressure (bar) at –20°C	at + 30°C	Volumetric Latent Heat at –20 °C	Molecular Mass (kg/km)
CO_2	31.06	73.84	19.67	72.05	14592	44.01
R-22	96.15	49.90	2.453	11.92	2371	86.47
R-134a	101.06	40.59	1.327	7.702	1444	102.03
R-410A	71.36	49.03	4.007	18.89	3756	72.59
NH_3	132.25	113.33	1.901	11.672	2131	17.03

1.3. Trans-critical CO_2 cycle

CO_2, owing to its low critical temperature (31°C) and relatively high critical pressure (7.1MPa), is frequently operated above its critical pressure for part of the cycle. The relevant P-h graph of the trans-critical CO_2 cycle operation is shown in Figure (1.1).

In a trans-critical CO_2 refrigeration cycle, the heat rejection takes place at super-critical state. The fluid above the critical point is treated as gas and the heat exchanger in which cooling of CO_2 gas takes place is called a gascooler. A gascooler replaces the condenser in a CO_2 trans-critical system. In a vapor compression system, the condensing temperature is chosen based on coolant temperature in the condenser and the corresponding saturated pressure is taken as the condensing pressure. It is note-worthy that, in super-critical heat rejection, no saturation point exists, so the gascooler pressure is independent of the refrigerant temperature at gascooler exit.

Since the throttle valve inlet condition determines the specific refrigeration effect, it is necessary to optimize the high side pressure of the CO_2 during trans-critical operation.

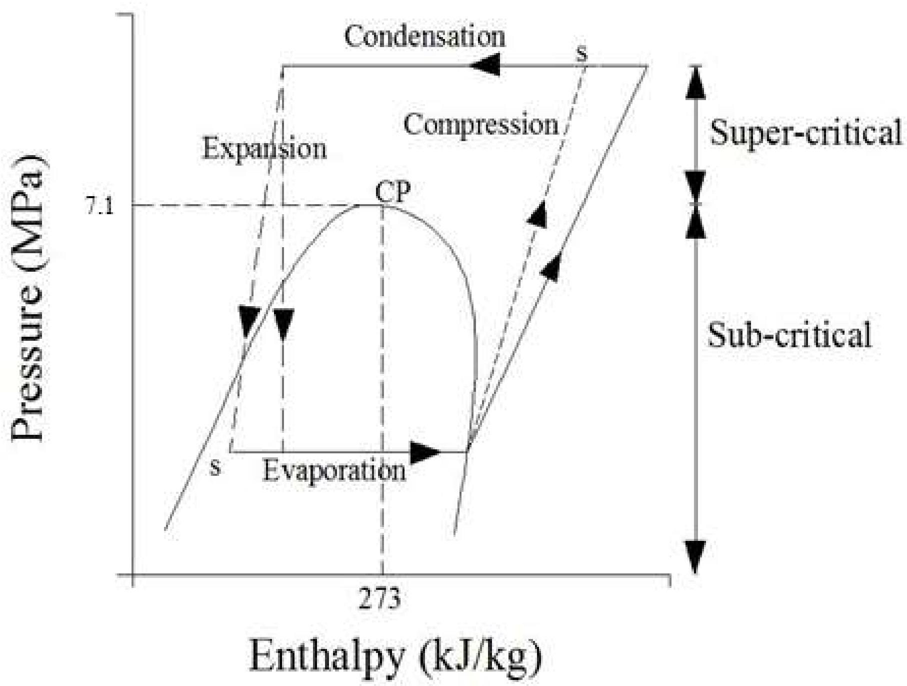


Figure 1.1: Trans-critical CO_2 cycle.

1.4. Key Challenges, Improvement Options and Contribution

CO_2 refrigeration system based on conventional cycle design don't give performance that is commercially competitive. The COP of the system is much lower than commercially popular synthetic refrigerants, more so for high ambient conditions. The reason for the same mainly are high pressure operation and resulting in large thermodynamic losses associated with throttling process. In order to improve the cycle performance, various possible modifications have been explored, like multi-staging of compression with intercooler, use of internal heat exchanges, use of work generating expansion devices, use of ejectors and use of more efficient heat exchanger.

Researchers have worked over wide range of operating temperatures & pressures and reported various degrees of success in improving the COP of CO_2 refrigeration system through cycle modification. The potential of work recovery and resulting COP improvement is viewed as crucial to its commercial success. Higher ambient temperature operation necessitates operation of CO_2 cycle in high pressure with trans-critical mode which potentially provides opportunity for higher work recovery. Various work recovery devices have been explored; for example reciprocating piston expander, rolling piston expander, scroll expander, screw expander, turbo expander and vane expander.

An attempt will be made to find the potential system modification and impact of implementing the same in existing system. A work recovery expander, observed as the potential modification will be designed and tested to observe the overall gain.

1.5. Contents and Thesis Structure

The thesis is organized into seven chapters. Chapter 1, gives introduction about refrigeration and the medium used for the refrigeration process and discusses the nomenclature and refrigerant selection criteria. It also introduces CO_2 as a prominent option as a natural refrigerant, it has summarized the key challenges associated with its use, mentions a few COP improvement strategies and contribution that are being investigated.

A literature review is presented in chapter 2. The reported literature is presented in a way that also gives background history of CO_2 as a refrigerant. The different paragraph is written with respect to application area, various modifications to the basic cycle and components of trans-critical CO_2 system. It also discusses the brief, research reported on CO_2 application areas like supermarket & heat pump system. Literature on various work recovery expander like piston, rolling piston, screw, scroll, turbo & vane are represented in detail.

From the literature, four emerging areas, namely multi-staging, waste heat recovery, work recovery expander and use of IHX are prominent options for enhancement of COP of the CO_2 systems at high ambient context. Chapter 3 summarizes the study objectives defined after analyzing the gaps areas in the literature.

Chapter 4 discuss thermodynamic modeling and simulation. Basic configuration of CO_2 refrigeration system along with different modifications to the basic cycle with energetic and exergetic point of view is presented. The chapter also includes simulation of a cascade refrigeration system determination of minimum TEWI. Moreover, it deals with the simulation of scroll work recovery expander using semi-empirical model. A comparative study of trans-critical CO_2 refrigeration system with and without scroll expander is presented. Thermo-economic analysis of the trans-critical CO_2 refrigeration system is carried out along with the total work recovery by the system for a high temperature city, Delhi.

Chapter 5 presents a case study on Indian dairy industry for possible implementation of CO_2 system for simultaneous heating and cooling application. Field data on ambient condition, milk handling and for year-round operation is taken up for this study. We evaluated two proposal from thermo-economic stand point. One is replacement of the existing ammonia based refrigeration plant with a CO_2 booster system and the other is use of a CO_2 heat pump for both waste heat recovery and water heating application in Dairy industry.

Chapter 6 presented an experimental study carried out using a scroll work recovery expander to study the overall performance of component and its working characteristics. The experimental study includes the scroll expander fabrication, experimental setup, calibration, experimental procedure and data reduction. Comparison and validation is also presented for the various performance parameters like mass flow rate, pressure ratio, suction temperature and shaft work using ANN.

Chapter 7 summarizes the major conclusions and recommendations for future work through the study.

CHAPTER 2

Literature Review

Carbon dioxide has been one of the most popular refrigerants in early 1900s, however, introduction of synthetic refrigerants later led to abolition of CO_2 as refrigerant. Increased awareness about harmful effects of synthetic refrigerants on environments in recent years is leading to shift of preference towards natural refrigerants. The thermo-physical properties of CO_2 are quite favorable among the natural refrigerants. However, its low critical temperature (31.1ºC) & relatively higher critical pressure (7.1 MPa) introduces additional challenges for use at high ambient temperature. This chapter presents a literature survey on trans-critical CO_2 systems as a resurgent refrigerant. The literature is organized in various themes like 'Historical Development', 'Mitigating challenges of high ambient temperature', 'Measure of cycle effectiveness', 'Modifications of basic CO_2 cycle' and 'Improving existing system with CO_2 trans-critical systems' sections, focused around possible system modifications, various components of the system and CO_2 system as replacement for existing systems.

2.1. Historical Development

As we go back well over a century, the advent of refrigeration and air conditioning in the late 1800s, had an enormous impact on various industries. CO_2 was first proposed as a possible refrigerant in 1850 [3]. Following a period of further development, the first successful commercial CO_2 was built in 1881 and later, in the year, 1890, a CO_2 based marine plant was established in UK. In USA, the first continuous production of CO_2 refrigerating equipment started in 1897 and more than 50% of all refrigeration systems in service at that time was based on CO_2 refrigerant. In 1905, Voorhees developed what is now known as the multiple effect cycle, which involves separation of liquid and vapor at an intermediate stage in the expansion process. In Europe CO_2 machines were often the only choice considered at that time.

In 1930−1940s the fluorocarbon based refrigerants were introduced with a massive advertisement campaign because of commercial interest and quickly took over a large part of market. Problems like high pressure containment capacity and efficiency loss at high discharge temperature of CO_2 system, lead to its decline assisted aggressive marketing of CFC products highlighting is low cost tube assembly and higher efficiency led to quick decline in popularity of CO_2 system. And by 1950s CO_2 was almost completely phased out [2]. Ammonia, however, remained the preferred refrigerant for the large industrial machines. All other conventional refrigeration applications became completely dominated by the various types of CFCs and HCFCs. System design with the new fluids were modern, more compact and clean.

Beyond 1950s, the HFCs once considered as the most prominent long term solution, have then been proven to be harmful owing to its high Global Warming Potential (GWP). Attention was focused to the long term harmful issues related to GWP and overall reduction of greenhouse gas emissions. Compared to CO_2, GWP of HFCs is higher by many folds. HFCs, are included in the Kyoto agreement (1992) as materials which are to be regulated. Further an assessment report released by the Intergovernmental Panel on Climate Change (IPCC) led to accelerate the phase-out of HCFCs in the wake of findings that action under the ozone layer treaty could do more to combat global warming than the Kyoto Protocol. Damage to stratospheric ozone layer has led to the Montreal Protocol and universal banning of the most CFCs and HCFCs compounds. R134a is one of the HFC refrigerant extensively used in automobile air conditioning. But, R134a may be decomposed by sunlight in the troposphere resulting in formation of acid and poisonous substances.

This has forced refrigeration engineers to search for new refrigerants such that there is no degradation of performance compared to that of proven CFC and HCFC technology. HFCs are presently used in newly produced refrigeration and air conditioning systems. HFCs are substances with zero ODP, and exhibit thermal and transport properties similar to CFCs and HCFCs. Hence this new class of fluids may be used with machinery already designed for CFCs and HCFCs with minor modifications. CO_2 is one such naturally available old refrigerant, which has been completely abandoned for more than 40 years, and is arguably the best future refrigerant both in air conditioning and heat pump applications due to its environmentally benign nature [3].

2.2. Mitigating Challenges of High Ambient Temperature

In 1993, Lorentzen and Pettersen, proposed a work around for the difficulties of low critical temperature of CO_2 refrigerant (31°C) by successfully operating the cycle in trans-critical mode and took advantage of its typical properties above the critical zone. Both temperature and pressure can be controlled independently above critical zone to obtain optimum performance of the system [4]. Above the critical temperature, refrigerant is treated as gas and the condenser is replaced by a gascooler. In 1994, Pettersen et al. put forward that the gascooler outlet pressure is the most influential parameter that effects the overall performance of the system. It was also concluded that increase in gascooler pressure does not always lower the COP of the tans-critical CO_2 cycle [5]. This can be attributed to the unique behavior of CO_2 near the critical point zone and beyond. The study also mentioned that, increase in the higher side pressure, increases the COP initially and then starts decreasing because beyond a certain pressure limit, increase in compressor work overtakes the gain in the refrigeration capacity due to the steep isotherms in super-critical region.

Several authors have since focused on the optimization problem of the trans-critical CO_2 cycle at high pressure and have put forward various expressions to define the optimal high pressure as a function of gascooler temperature and evaporator temperature. At high ambient temperature, the CO_2 cycle is required to be operated high above the super-critical zone to achieve effective heat transfer between the gascooler and ambient. CO_2 trans-critical cycles operate at much higher pressure than conventional refrigeration systems, typically 5 to 10 times. Literature available on both experimental and simulation based study of the CO_2 systems reveals that the cooling COP of trans-critical CO_2 system is more sensitive to the variation of ambient temperature, than with the conventional refrigerants. It is also reported that simple CO_2 cycle is superior in performance at moderate & low ambient temperature but inferior at high ambient temperature applications.

2.2.1. Measure of cycle effectiveness

COP measure is considered as well established parameter, to evaluate performance of a trans-critical CO_2 refrigeration cycle. However, a number of researchers have proposed various other measures for system comparison depending upon circumstances, for example: energy and exergy, lowest total equivalent warming impact (TEWI) and minimum payback period (PBP).

2.2.1.1. Energy and exergy

Energy based measure is a result of first law of thermodynamics while exergy based concept is based on second law of thermodynamics. Energy is a measure of only the quantity whereas exergy also takes into account the quality of energy. Performance of the various components of refrigeration system are computed using the parameters like enthalpy, entropy, mass flow rate, reference temperature, refrigeration effect and work done by the compressor at various positions in the cycle and the relevant equations used (article 4.1.1 & 4.1.2) to evaluate the same is given in literature [6–9].

2.2.1.2. Total equivalent warming impact (TEWI)

TEWI is a parameter which assesses both direct and the indirect emission of greenhouse gasses, essentially including emission associated with the process of electricity generation. Concept of TEWI was developed by Fischer [10] as a measure of combined global warming impacts of the refrigerant losses to the atmosphere and the CO_2 emissions from fossil fuel to generate power to run the refrigeration and air-conditioning systems. A few researchers also used the same method to identify the TEWI with respect to a supermarket HVAC processes [11] and low carbon emission from a refrigeration system [12]. TEWI is computed using Eqs. (2.1−2.3).

$$TEWI = TEWI_{direct} + TEWI_{indirect} \tag{2.1}$$

$$TEWI_{direct} = (GWP * R * l * n) + (GWP * R_{ch} * (1 - \alpha)) \tag{2.2}$$

$$TEWI_{indirect} = E * \beta * n \tag{2.3}$$

2.2.1.3. Payback period (PBP)

The economic implication of introduction of various possible modifications for trans-critical CO_2 refrigeration system is attempted to be measured in terms of PBP [13]. The PBP of the modification is computed using Eq. (2.4).

$$Payback\ period\ (PBP) = \frac{Component\ or\ modification\ price\ (\$)}{power\ saving\ (kWh) * Running\ hours\ yr^{-1} * teriff(\$\ kW^{-1})} \tag{2.4}$$

2.3. Modifications of Basic CO_2 Trans-critical Refrigeration Cycle

The basic single stage trans-critical refrigeration cycle may need to be modified for various reasons such as COP improvement, capacity enhancement, etc. In general, a number of possible modifications, including multi-staging, use of internal heat exchanger (IHX), staging of compression as well as expansion, splitting of flows and use of work generating expander.

2.3.1. Use of IHX

IHX contributes by improving the vapor quality at compressor suction, thereby reducing the compression work. In 1998, Robinson and Groll studied a trans-critical CO_2 system equipped with IHX using a thermodynamic modeling approach. It was reported that the IHX can contribute to increase system COP to the extent of 7% [14]. The effect of IHX on the overall system performance of the trans-critical CO_2 system in mobile air conditioning application was experimentally investigated and reported in 2001 by Boewe et al. [15]. It was reported that IHX significantly improves (upto 25%) the system efficiency. In 2004, Sarkar et al., however, reported a minor influence of IHX on the overall system performance and supported the same through experimental study at low and moderate gascooler exit temperature [16]. In the same year, Kim et al. reported that the enhancement in overall performance of the trans-critical CO_2 system is observed by using IHX for water heating application [17].

In 2005, Chen and Gu reported correlations between the optimum pressure and other system parameters such as ambient temperature, evaporator temperature and working pressure ratio. A correlation was also proposed for determining optimum high pressure of the CO_2 system. The correlations could predict with deviation of less than 3.6% at 5.3ºC, the evaporator temperature [18]. In the same year, a study based on both simulation and experiment was reported by Rigola et al., focusing on influence of IHX on the overall system performance with ambient temperature range 35ºC to 43ºC. An improvement in cooling capacity and system COP is reported with use of IHX in the system [19]. In 2007, in a more detailed study, improvement in cooling capacity of a CO_2 refrigeration cycle was reported, achieved by varying the refrigerant mass flow rate, compressor frequency, EEV opening, and length of an IHX by Cho et al. It was reported that a reduction in optimal gascooler discharge pressure upto 0.5MPa is achieved due to use of IHX. Further, COP improvement in the range 7.1% to 9.1% was reported [20].

In 2008, Aprea and Maiorino reported performance evaluation of a trans-critical CO_2 cycle using IHX for residential application. Performance of CO_2 systems with and without IHX were compared in term of COP. Significant improvement in the system COP was reported [21].

Zhang et al. in 2011 reported effect of IHX on the overall performance of trans-critical CO_2 system. Performance of IHX working in both sub-critical and super-critical zone was discussed. It was concluded that the COP of the system reduces slightly due to IHX when working in sub-critical zone [22].

An experimental study was reported by Nakagawa et al. in 2011 studying the effect of IHX on the overall system performance of an ejector expansion refrigeration system. Various system operating parameters such as pressure ratio, ambient temperature was varied and experiment carried out with two different IHX length of 30cm and 60cm. Maximum 27% COP improvement was reported using 60cm length IHX, compared with conventional system without IHX using R22 as a working fluid [23]. In 2011, a comparative study of the trans-critical CO_2 system with and without IHX was reported by Sanchez et al. using field data. A simulation model was proposed and validated using the field data at three different evaporative temperatures -5ºC, -10ºC & -15ºC and two different gascooler temperatures 31ºC & 34ºC. A few correlation were also reported relating effectiveness of IHX and the various operating parameters such as pressure ratio and ambient temperature [18].

In 2012, Cabello et al. reported an experimental study and projected the effectiveness of IHX on the overall performance of the CO_2 refrigeration system. Three different injection points were arranged: before IHX, after IHX and just before the suction chamber of the compressor. It was reported the cooling capacity and COP were enhanced by 9.81% and 7.01% respectively [25].

2.3.2. Multi-staging of CO_2 trans-critical refrigeration system

As the temperature lift increase i.e. difference between condenser and evaporator temperature increases, single stage refrigeration systems become inefficient and impractical due to high compression work and requirement of very high pressure component. The temperature lift may become large, either due to the requirement of very low evaporator temperature or very high ambient temperature. In multi-stage compression system configuration, generally two different refrigerants are used in series with two single stage refrigeration system that are coupled at an intermediate temperature. The intermediate pressure and intermediate temperature are design parameters for a multi-stage system, Missimer's Eq. (2.5) & Schmidt's Eq. (2.6) are classically used to determine them.

$$\frac{p_{cond,LTC}}{p_{evap,LTC}} = \frac{p_{cond,HTC}}{p_{evap,HTC}} \tag{2.5}$$

$$T_{evap,HTC} = \sqrt{T_{cond,HTC} * T_{evap,LTC}} \tag{2.6}$$

In multi-staging although, the initial capital investment increases, but it reduces compression work in both the single stage systems and also simplifies component design.

In 1996, Chen and Wu reported a technique for the optimization of intermediate pressure on the basis of minimum work requirement. A fair agreement was projected with classical estimate. It was reported that the high discharge temperature is the disadvantage to COP for CO_2 heat pump system, which can be overcome by employing a two-stage system with intercooler in between the compressor stages in parallel with the gascooler [26]. In 2004, a thermodynamic simulation model was reported by Bell, in order to investigate a two-stage parallel compression CO_2 refrigeration cycle with economizer [27]. Both theoretical and experimental study was reported by Cafvallini et al. in 2005, in order to optimize two-stage CO_2 system with intercooler for an air conditioning application. Overall 25% COP improvement was reported for a two-stage compression scheme [28].

During super-critical operation, in order to manage the high pressure in evaporator coil of a trans-critical CO_2 system, multi-pass tubes are considered as a prominent option. The vapor and liquid distribution is difficult to maintain or predict due to combined effect of surface and gravitational force. This problem appears frequently in CO_2 systems due to high pressure operation. Bypassing the vapor around the evaporator and allowing only liquid refrigerant to flow towards the evaporator is an effective method to overcome this problem. In 2004, Elbel and Hrnjek reported an experimental study using Flash gas bypass in trans-critical CO_2 refrigeration system. It was reported that the cooling capacity and COP improved by 9% and 7% respectively [29]. In the same year, in order to evaluate the performance of a trans-critical CO_2 refrigeration system for a two-stage compression, a theoretical study was reported by Hwang and Radermcher for four system configurations and three different temperature levels 7.2ºC, -6.7ºC and -23.3ºC. It was reported that the system configurations with IHX, possesses the maximum COP [30].

In 2005, a CO_2/C_3H_8 cascade system based simultaneous heating and cooling application was reported by Bhattacharyya et al. The system was equipped with CO_2 in high temperature cycle (HTC) and propane in low temperature cycle (LTC). The cascade system configuration was equipped with IHX in both HTC and LTC. It was reported that the influence of IHX in LTC was marginal where as in HTC, IHX contributed significantly towards COP improvement [31]. In order to evaluate the optimal intermediate temperature (IT) of a cascade condenser, both theoretical and experimental study was reported by Lee et al. in 2006 at a fixed evaporator temperature of -50ºC. The heat rejection temperature was also fixed at 32ºC and IT range was selected within -5ºC to 5ºC, on the basis of minimum exergy losses by the system. it was reported that the maximum COP of the cascade system was achieved for IT of 0ºC [32]. In 2006, an experimental study was reported by Kruse and Ru, using CO_2/N_2O combination of refrigerants in a cascade system for low temperature applications of -50ºC, -70ºC and -88ºC. It was reported that the maximum COP of the cascade system was in the range 0.38 to 0.82 [33].

In 2007, a simulation based study was reported for a two-stage trans-critical CO_2 system using flash intercooling in between the two-parallel compressor system for a heat pump by Agarwal and Bhattacharyya. The overall results exhibits that the flash intercooling technique was not economical with refrigerant like CO_2 [34]. In 2009, a theoretical study using CO_2/NH_3 cascade system was reported by Alberto et al. Exergy analysis was carried out varying various operating parameters such as evaporator temperature, condenser temperature and IT in order to study the overall performance of the system. A correlation was also proposed for the design to optimize the overall performance of the system [35]. In the same year, Bhattacharyya et al. reported a CO_2/N_2O cascade system for simultaneous heating and cooling application. N_2O was selected to fulfill the ultra-low freezing temperature demand of -65ºC. The cascade system configuration was equipped with two suction line IHX in both HTC and LTC. The influence of variation in operating parameters like evaporator temperature, condenser temperature and cascade condenser temperature on the overall system performance, was studied. It was observed that the IHX in suction line had a marginal influence on the overall performance of the system and also the system exhibited comparable behavior for swapping of fluids in LTC & HTC because of similarity in thermodynamic properties [36]. Later, in the same year, a comparative study was reported by Bingming et al., to evaluate the performed of the cascade system using three combinations of refrigerants i.e. NH_3/CO_2, NH_3/NH_3 and NH_3 system with economizer in cascade refrigeration system configuration. A comparison was made at a constant evaporator temperature of -40°C between the three configurations. It was concluded that the NH_3/CO_2 cascade system is very competitive for low temperature application [37].

In 2011, an experimental work was reported by Dopazo and Seara for a freezing process applications using CO_2/NH_3 cascade refrigeration system configuration. Tests were conducted at four different evaporator temperature levels i.e. -50°C, -45°C, -40°C and -35°C keeping IT within range -17.5°C to -7.5°C. Performance of the system was further compared with the baseline two-stage NH_3 refrigeration system. It was reported that the COP of the cascade refrigeration system at -40°C evaporator temperature, was 19.8% more than the baseline system configuration [38]. In 2011, the performance and operating characteristics of a two-stage CO_2 cycle using gas injection was investigated by Heo et al. The performance was measured in terms of the amount of refrigerant charge, compressor frequency, EEV opening for the cooling mode operation. Cooling COP with 16.5% enhancement was reported [8]. An optimum cycle control strategy for the intermediate pressure as reported by Heo in 2012 with a refrigerant injection heat pump having a double expansion sub-cooler based on the injection ratio. The optimum sub-cooler pressure ratio reported was within range 0.4 to 0.7 [39].

In 2014, an experimental investigation was reported by Sanz-Kock with R134a/R717 cascade refrigeration system for three different low temperature applications i.e. -30ºC, -35ºC and -40ºC. Simple cascade system configuration without IHX was tested using R134a in HTC and R744 in LTC. Experiment was carried out using nine different parametric combinations and for a range of IT. The maximum COP of the system was obtained at -30ºC IT for the range between 5ºC to 0ºC, while -35ºC was recommended for the range between -11ºC to 6ºC and -40ºC for the range between -15ºC to11ºC. The reported COP range was between 1.38 to 1.43 with an average of 1.41 at -30ºC, 1.19 to 1.25 with an average of 1.23 at -35ºC and 1.02 to 1.08 with an average of 1.05 at -40ºC [40].

In 2016, Zhang et al. reported four system configurations with two-stage compressor using IC, IC with expander, FGI and FGI with expander, at a fixed evaporator temperature of 0°C, while varying the gascooler outlet temperature within range 32°C to 48°C. Performance comparison of the four systems were made with respect to basic conventional system. The system with IC and expander yields the maximum COP as reported [41].

2.3.3. Expansion with work recovery

For effective heat rejection in CO_2 refrigeration system at high ambient temperature, the compressor needs to be operated at high pressure ratio. This implies enhanced power requirement and reduction in COP. This challenging situation, however also provides an opportunity of energy recover from the system in the form of heat and work. Use of work recovery expander in place of expansion valve is a promising option to achieve commercial success for trans-critical CO_2 refrigeration and air conditioning system by enhancing overall COP. Two simultaneous functions are carried out by the refrigeration system equipped with work recovery expander during the operation i.e. expansion of the high-pressure vapor and constant work generation, results improvement in the overall performance of the system.

In 1998, a potential improvement scheme, to improve the overall performance of the trans-critical CO_2 refrigeration system was proposed by Robinson and Groll, by replacing the expansion valve with work recovery expansion turbine. Initial investigation was carried out to investigate the heat transfer properties of CO_2 and the thermodynamic modeling of trans-critical CO_2 refrigeration system for air conditioning application equipped with suction line IHX. Performance of the system was also compared to the baseline R22 refrigeration system. An optimum heat rejection pressure with respect to maximum COP was contrasted. Also, it was reported that the use of IHX with work recovery expander, reduces the overall performance of the system by about 8%. Major conclusion drawn from the study was, replacement of the expansion valve with work recovery expander potentially reduces 35% of the total cyclic irreversibility [14].

Based on the first law and second law of thermodynamics, a comparative study was carried out by Yang in 2007, using a trans-critical CO_2 refrigeration system with and without work recovery expander. Performance of the both systems were compared and evaluated on the basis of various parameters like evaporator temperature, gascooler outlet pressure, total exergy losses and overall COP of the system. It was observed that 38% of the total cycle irreversibility occurs during the throttling process. Comparative analysis showed that the COP and exergy of the cycle, on an average, were 33% and 30% higher than the system equipped with throttling valve configuration [42]. Various work recovery expanders like piston expander, rolling piston expander, scroll expander, screw expander, turbo expander and vane expander have been exploited by various researchers with an objective to improve the overall performance of the CO_2 system.

2.3.3.1. Piston expander

In 1999, a piston expander compressor unit as a free piston machine was tested by Heyl et al. Several modifications of the process were investigated for improvement of efficiency. The design was very simple and was tested at full pressure mode principle. 78% of the expander power was recovered [43]. A new design of expander-compressor unit was investigated for variable speed in 2002 by Nickl et al. They categorized the mechanism to be very complex and only 10% further improvement in work recovery was registered [44]. In the same year, a piston-cylinder type work output device having a two-piston engine was designed, fabricated and tested for four different cycle modifications by Baek et al. The reported performance enhancement from previous design was 10% [45]. Another piston type expander was tested and reported by Baek et al. in 2005 and were operated and performance was evaluated at three different operating conditions. Limited operating characteristics were identified and only about 10% increment in COP was reported [46].

Later, in the year 2005, a three-stage expander modification was developed on the free piston expander by Nickl et al. In this modified design, a vapour-liquid separator was installed between the second and third stage of expansion and due to which some increment in efficiency was identified [47]. In 2006, a prototype of CO_2 swing piston expander integrated with shaft was tested by Guan et al. During the analysis, it was found that friction and leakage accounts for most of the losses. A material with high elastic modulus was adopted to decrease the leakage rate at the gap. They also optimized the expander geometry to minimize the frictional and leakage losses. The expander investigated registered 28% to 44% efficiency [48].

In 2006, a design of a double acting free piston expander was investigated by Kohsokabe et al. They analysed the dynamic pressures in the expander chamber combined with compressor on a common shaft. They also considered the effect of heat transfer and leakage in the expander.

The expander was tested at various speeds and concluded that the prototype could be stable for a wide range of pressure [49]. An expander coupled with an auxiliary compressor to the trans-critical cycle was designed and tested by Zhang et al. in 2007. The design was developed taking due care of geometry constraints and losses due to leakage and heat transfer. 62% efficiency was reported and the researchers concluded that the expander could work in wide range of pressure ratio [50].

In 2011, an investigation on the influence of non-condensable gas on the expander performance was carried out by Tian et al. Nitrogen gas was selected due to its stable chemical properties. With various ratio of nitrogen gas mass introduced into the system, the resulting change in frictional loss and efficiency was studied. It was concluded that presence of nitrogen decreases the frictional loss and as a result efficiency improved [51]. In 2014 a reciprocating piston expander with a geometry having four cylinders arranged radially was investigated by Fukutaa et al. O-rings were adopted to reduce the friction between the moving parts and efficiency of 40% was reported [52].

2.3.3.2. Rolling piston expander

In 2003, a theoretical model of rolling piston expander was analysed to determine the required design parameters for construction by Li et al. The expansion process was visualised in two parts; first due to work done by pressure and second due to volume expansion. They reported efficiency as 50% for the expander for various losses due to leakage and friction [53]. In the same year, a prototype of rolling piston expander was tested by Zha and Ma. The rolling piston expander was selected on the basis of ease of fabrication. The efficiency of the expander was found to vary within range 23% to 58% [54].

In 2006, a new design of two cylinder rolling piston expander was proposed by Yang et al. They proposed that the suction valve could be removed and as a result the expansion process can become continues. The expander was divided into two units but both were coupled with a crank shaft. It was observed that the rotation angle of the crank shaft is influenced by the torque due to refrigerant pressure [55]. In 2008, an another work on two cylinder rolling piston expander was reported by Matsui et al., where a 6% improvement in COP was achieved by optimizing clearance between the parts [56].

In 2009, a two-stage rotary expander was developed and optimized by Matsui et al. An analytical model was developed by combining dynamic mechanical analysis and refrigerant pressure analysis. During analysis of the model, leakage was also taken care of. After optimization, 54% expander efficiency was achieved [57]. In the same year, an another work was reported in the rolling piston expander by Li et al. to study the effect of parameters like mass flow rate, volume of cylinder, suction and discharge port. Experimental results showed that the suction control system and expander could work normally under large pressure difference. Isentropic efficiency of 58.7% was reported [58].

In 2013, a prototype of rolling piston expander was tested by Zhang et al., reduction of leakage and friction loss, targeted by simultaneously decreasing the length and diameter of the connecting hole. Maximum 42.3% expander efficiency was reported [59]. In 2015, two efficiency improvement measures were tried out on the two-rolling piston expander by Hu et al. The efficiency improvement measures were, increasing the expansion ratio and adopting more efficient sealing measures. Reported maximum system efficiency was 77%, which is a substantial improvement over their earlier design [60].

2.3.3.3. Screw expander

In 2002, a twin screw combined compressor and expander model was developed by Stostic et al. They examined how the rotor forces created by compression and expansion could partially be balanced in order to eliminate the axial forces and reduce the radial bearing forces. The resulting COP increment reported was from 2.79 to 4.8 [61]. Later a numerical analysis of the same design was conducted in 2006 by Smith et al. During the analysis, they developed a three-dimensional computational model, comprising of both fluid and structural deformations to investigate fluid solid interaction. They concluded that the arrangement can reduce radial and axial forces of a rotor [62].

2.3.3.4. Scroll expander

In 2004, a new and modified model of scroll expander was developed by Westphalen et al., and analysed for high ambient conditions. Analysis showed that the most attractive approach for utilization of the expander shaft power is to integrate it with compressor/expander unit. They also reported suitable design operating conditions like machine speed, displacement, volume ratio and outer diameter of scroll flank and also estimated the leakage losses as 20% and reported 72% expander efficiency [63]. In 2006, another scroll expander was investigated both theoretically and experimentally by Fukuta et al. In theoretical study, they introduced a calculation method for leakage flow rate. Theoretical expander efficiency of 60% at 3600 rpm was reported while during testing 55% efficiency was obtained [64].

A prototype of scroll expander was tested by Kohsokabe et al. in 2008. The performance parameters i.e. rotational speed, inlet pressure & temperature, exit pressure and mass flow rate of the expander were analysed. They found that the optimum pressure ratio was larger than the ideal pressure ratio and an efficiency of 83% was reported [65]. In 2010, another prototype of scroll expander was developed and tested by Nagata et al. The prototype was having an intercooler in which a sub-compressor is allocated after the intercooler. The expander directly drives the sub compressor. They analysed the pressure and leakage losses and concluded that the power consumed by the main compressor was successfully reduced due to the usage of expander [66].

In the same year, an attempt was made by Hiwata et al. to redesign the gap profile of a scroll expander in order to make use of over-expansion and to control the axial force on thrust bearing. They found that with the help of oil film pressure-force of the main and eccentric bearing, the radial force between the scroll and wrap could be controlled. They concluded that the pressure force contributed to the reduction in leakage loss and 96% of volumetric efficiency was obtained [67].

2.3.3.5. Turbo expander

In 1999, the first prototype of turbo expander on CO_2 trans-critical system was tested by Alvarez. The model was simple, running at high speed and 50% efficiency was reported [68]. In 2004, a turbine compressor mechanism was patented with axial flow design by Hays and Brasz. During optimization, the effect of rotor speed was analysed and 69% efficiency was reported at high speed [69]. In 2006, a radial outflow impulse turbo expander was tested by Tondell and concluded that the expander exhibited rather low efficiency due to large diameter and frictional losses and the concept of radial outflow turbine was incompatible. They suggested use of radial inward flow direct impulse turbine [70].

2.3.3.6. Vane expander

A vane expander was tested by Fukuta et al. in 2006 considering internal leakage. They analysed the pressure inside chamber with sensors attached to the expansion chamber and graphically represented pressure v/s rotational angle plot and compared the same with ideal solution. They found that the rapid decrease in pressure during expansion process was responsible for major leakage. A tight contact arrangement with springs on vanes and the cylinder wall were proposed to improve the efficiency and subsequently an improvement of volumetric efficiency from 17% to 30% at a speed of 800 rpm was reported [71].

In 2010, a revolving vane expander was examined by Subiantoro and Ooi. They analysed the model with some assumptions like perfect sealing, shaft rotation at constant angular velocity and zero friction loss at shaft bearing. They concluded that the vane should be attached to the driving component in order to minimize the friction loss [72]. A prototype of rotary vane expander was tested in 2011 by Jia et al., that incorporated practical considerations like leakage due to loss of contact between the vane tip and cylinder wall. For efficiency improvement, they introduce high pressure gas into the vane slot and compared this modified prototype with the original prototype and 15% to 45% improvement under similar working condition was reported [73].

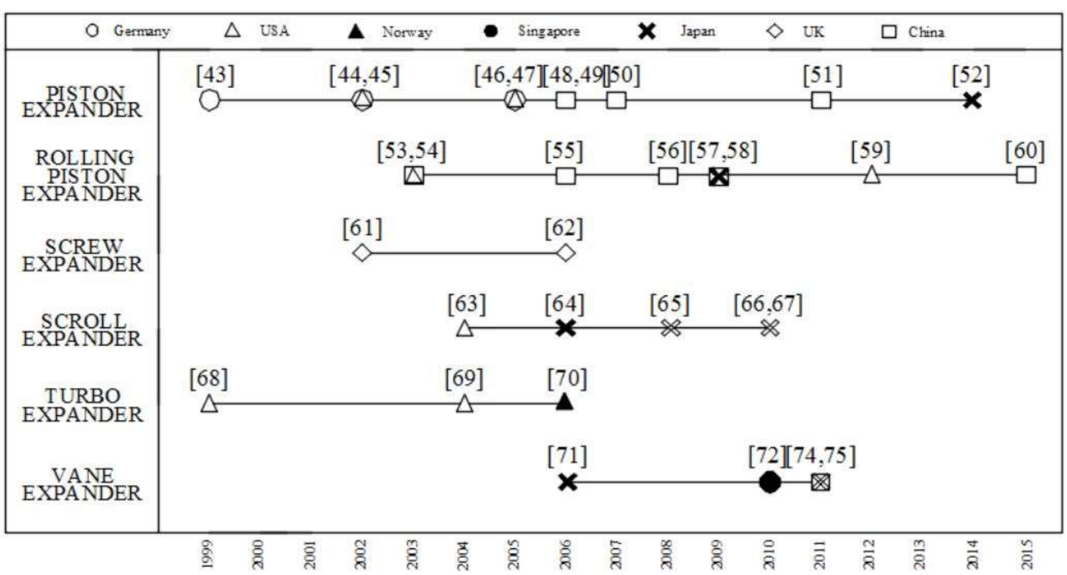


Figure 2.1: Reported research in trans-critical CO_2 with work recovery expanders.

A new small and simple prototype was tested by Fukutaa et al. in 2011. They introduced a concept of vane back pressure component to reduce leakage. O-rings were used as shaft seal. The influence of two parameters namely inlet temperature and the flash inception on expansion process were observed and they concluded that further study is needed to examine the influence of the same on expander's efficiency [74]. The trend of the reported research for work recovery expander is shown in Figure (2.1).

2.4. Trans-critical CO_2 Heat Pump System

A novel fan-less CO_2 heat pump system was suggested [75] for domestic room heating application, which provided a few benefits like low noise, less power consumption and increased comfort due to reduce air draft in the room. Effect of IHX on the overall performance of the trans-critical CO_2 heat pump system was studied [76] and reported that the system performance was improved by 16.5% at high evaporation temperature. In the same year, a CO_2 heat pump for household cloth drier application was also reported [77]. A comparison was also made with the conventional R134a system for same application, using a fixed approach temperature in heat exchanger. It was reported that the CO_2 system consumes lesser power.

Investigation of influence of various operating parameters on a CO_2 heat pump system at a fixed water inlet temperature was carried out for water heating application [78]. A prototype was tested within a range of ambient temperature (-15°C to -35°C) and water outlet temperature (55°C to 80°C). A correlation was developed for the optimum discharge pressure as a function of ambient temperature and water outlet temperature. A comparative study was carried out with three different vapor injection techniques namely 'flash tank vapor injection', 'sub-cooler vapor injection' and 'flash tank vapor injection with suction line heat exchanger' at low ambient temperature [79]. The sub-cooler vapor injection technique was found to be having the maximum COP. A quasi-transient model for direct expansion CO_2 heat pump system for integrated space and water heating application was reported [80].

Effect of bore-hole length on system efficiency was studied and an optimum bore-hole length was suggested. In a recent study, an energy assessment method for a water/water CO_2 heat pump system was proposed for winter heating and summer cooling application [81]. It was concluded that a CO_2 system is very efficient in water heating application, but is less efficient in a combined heating and cooling application. In the same year, an ammonia based and two CO_2 based heat pump systems were proposed to fulfill the demand of cold water, warm water and hot water in a food processing industry [9]. The annual primary energy saving and saving of operating cost was also computed for four cities of US. Maximum annual primary energy saving reported using the CO2 heat pump range from 56% to 65% and using the NH3 heat pump was 44% with respect to existing system at the plant.

2.5. Replacing Existing Conventional System with CO_2 Trans-critical System

CO_2 trans-critical systems have attracted attention for application in wide variety of refrigeration applications. Natural refrigerants are perceived to be potential permanent solution in the face of progressive restrictions imposed upon use of synthetic refrigerants. At present, a number of medium to large commercial refrigeration setups, supermarkets are running on natural refrigerants.

In 2004, a comparative study was reported between performance of CO_2 trans-critical refrigeration system and R404A based large capacity refrigeration system by Girotto et al. Five supermarkets running with CO_2 refrigeration system in north Italy were selected for the study. It was observed that the total installation cost of the system was comparable but the total annual electricity consumption by the CO_2 refrigeration system was 10% higher. Some strategies to reduce total electricity consumption by the CO_2 refrigeration systems improving the component design, recovering thermal energy, integrating the HVAC system with medium temperature and low temperature refrigeration plants and reducing thermal loads on refrigerated display cases [82].

In a study reported in two parts by Ge & Tassou development of a modeling procedure "SuperSim" was elaborated in 2011. It involved evaluation of the performance of a supermarket, involving building envelope and installed HVAC systems. Results from "SuperSim", was validated against the R404A, as the baseline system. Considering high head pressure as an important parameter for the performance evaluation at higher ambient temperature, the system was simulated for the period of one-year. It was reported that there is potential improvement in the system performance by employing heat recovery system at lower ambient temperature conditions. Among the various system configurations observed, trans-critical CO_2 booster refrigeration system was the best option for the large refrigeration systems for supermarket applications [83,84].

In 2012, three different system configurations of trans-critical CO_2 refrigeration systems were investigated by Cecchinato et al. for supermarket application, to find the potential improvements over the traditional R404A refrigeration system, in various climatic conditions. Heat pump system using R404A as a working refrigerant, chillers using R410A and low temperature system using R404A and a R744 refrigeration systems were analyzed. Using the field data, total energy consumption and overall saving were predicted. It was reported that the total annual energy saving with R744 refrigeration system was about 15% over the baseline R404A system [85]. In the same year, a simulation model was developed by Ommen and Elmegaard, for commercial trans-critical/sub-critical booster refrigeration system to carry out the thermo-economic diagnosis of the system cooling at individual temperature levels. Main objective of the model was to find ways to reduce the total electricity consumption and running cost. It was observed that the running cost is double, when the frozen temperature of the goods are achieved [86].

In 2014, various CO_2 trans-critical and cascade/secondary loop refrigeration system configurations were analyzed and reported by Sharma et al., for supermarket application. Selected configurations and their performance were compared with the baseline R404A direct expansion system. Among the various supermarket configurations, the booster system with bypass compressor was found to have the lowest energy consumption and performance superior to the R404A system [87]. In 2015, a study was carried out at Sweden, based on five supermarkets installed between the year 2007 to 2010, using field data measurements by Swalah et al. A comparative study was reported between the various configurations. Trans-critical CO_2 booster system using the intermediate vessel to bypass the vapors, was reported to have the highest COP [88].

2.6. Summary and Gap Areas

A detailed literature review is carried out in order to identify the recent trends and gaps in research for CO_2 systems. Research trend at present is focused on the improvement of system efficiency through various modifications of a simple cycle, for example multi-staging, use of work recovery expander, cascade configuration, use of IHX, etc. The commercial success of the CO_2 systems is still a challenge at high ambient context. Moreover, a few low GWP synthetic refrigerants are also recently introduced which could work efficiently at high ambient temperature.

The gap areas identified from the present literature review are as following:

- Work recovery expander appeared as a promising option to improve overall performance of the CO_2 system (theoretically and practically), but the advantage is prominent only at high pressure range. Two phase behavior of the refrigerant during the expansion process is still unpredictable and which is greatly influenced by the expansion profile.
- Screw and Scroll expanders are popular due to high efficiency & high reliability and can be viewed as a prominent option. But, the major drawback is their very high component cost and availability off-the-shelf. So, there is a need to find an alternate economical method to develop a work recovery expander. Some researchers reported reverse engineering of expander from compressor with various degree of success.
- CO_2 heat pump is employed in various process industries to partially fulfill both heating and cooling demands. But, a very few studies were reported for waste heat recovery form the process industries by the CO_2 system.
- Various supermarkets running with CO_2 systems in colder region are successful and viewed as permanent solution for clean environment. However, for warm climatic condition, replacement of a conventional system with CO_2 system has not been found to be commercially viable.

CHAPTER 3
Objectives

Considering CO_2 as a potential solution for various HVAC&R applications, an extensive literature review is conducted and systematically analyzed in order to extract the research gap areas and the same are reported in the literature review chapter. From the literature, four emerging areas, namely multi-staging, waste heat recovery, work recovery expander and use of IHX are prominent options for enhancement of COP of the CO_2 systems at high ambient context. This chapter summarizes the study objectives defined after analyzing the gaps areas in the literature.

3.1. Study Objectives

Major objectives of the present work are:

- To identify the most prominent options among the various work recovery expander designs based on literature survey and validate based on data analysis.
- Simulate various CO_2 trans-critical refrigeration system configurations, including single as well as multi-stage systems to find optimum system configurations for warm climate.
- Model and simulate a potential implementation of CO_2 system for warm climate. A medium capacity commercial milk chilling plant is identified for developing the case study.
- Investigate working of a work recovery expander and its potential advantage, including study of both economic and environmental aspect.
- Attempt to re-engineer a scroll compressor to work as expander and experimentally investigate it's working.

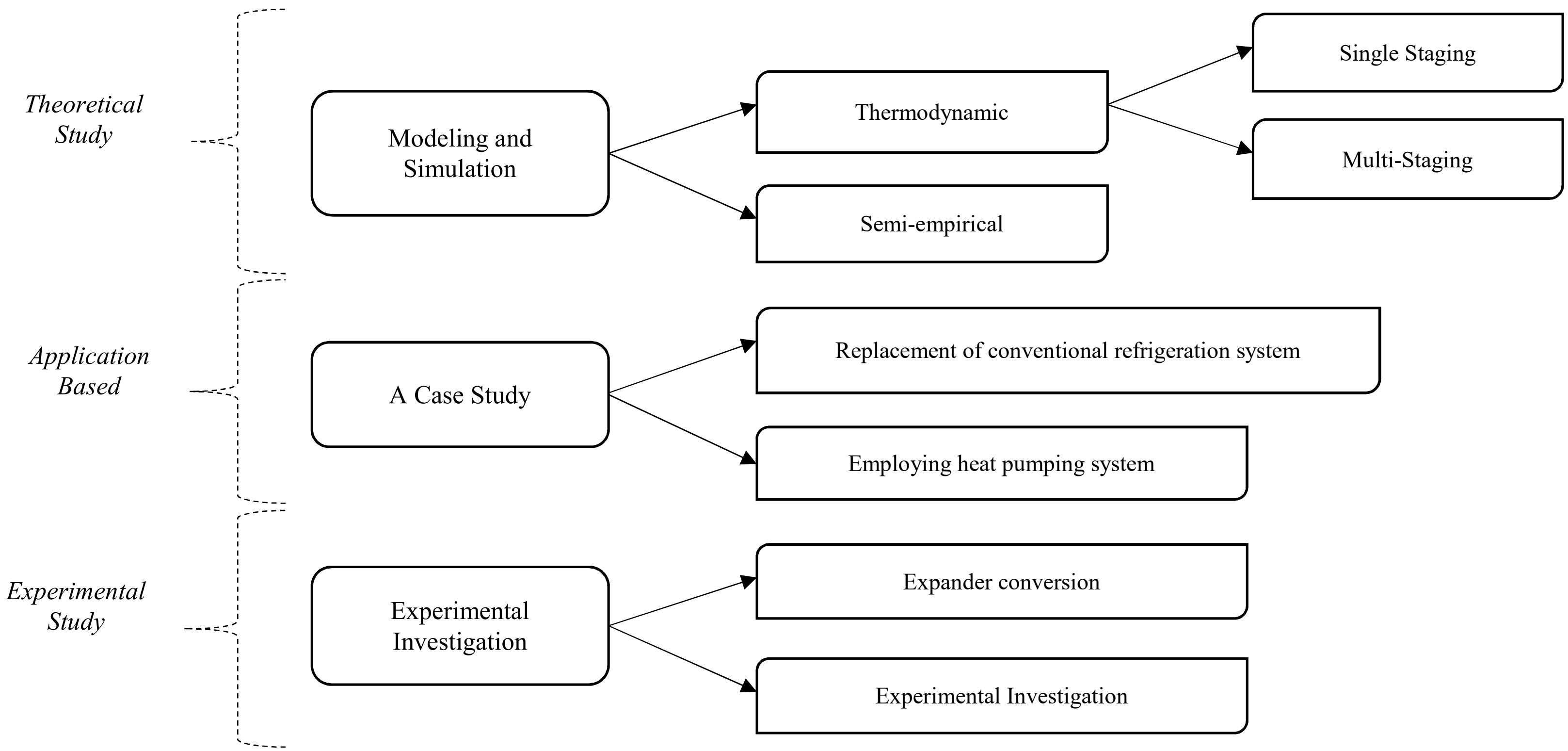


Figure 3.1: Flow chart representing the study objectives.

CHAPTER 4
Modeling and Simulation

The performance of CO_2 refrigeration system is highly susceptible to ambient temperature. In this chapter, six modifications are investigated for high ambient temperature condition (typical in India). The Chapter also includes, energy as well as exergy analysis of the systems, against range of various operating parameters like evaporator temperature, gascooler outlet temperature and compressor outlet pressure. Effect of a few real-time constraints are also analyzed. Real-time constraints studied are approach temperature, pressure drop in gascooler, compressors efficiency, degree of superheat, expander's efficiency and effectiveness of intermediate heat exchanger on the system behavior. An innovative minimum TEWI based scheme is proposed for intermediate temperature selection, while study of cascade system as one of the cycle modification. One synthetic refrigerant with low ODP and GWP introduced relatively recently in the market R1234yf, is also analyzed.

This chapter also details analytical study of performance of a scroll work recovery expander using a semi-empirical model for CO_2 trans-critical flow. Total energy consumption and work recovery are compared. The chapter includes economic analysis of five different capacity of scroll expanders. Also, a case study is carried out to compute the total payback period of a scroll expander suitable for development at a high ambient city, Delhi.

4.1. Trans-critical CO_2 Refrigeration System and Modifications

A comparative study of performance of six prominent modifications to the basic trans-critical CO_2 refrigeration system (Figure 4.1,a) is made to comprehend their suitability to high ambient temperature applications (35–55°C), and the same are listed below:

- Trans-critical CO_2 refrigeration cycle using internal heat exchanger (IHX).
- Trans-critical CO_2 refrigeration cycle using work recovery expander.
- Trans-critical CO_2 refrigeration cycle using work recovery expander with IHX.
- Trans-critical CO_2 refrigeration cycle using inter-cooler (IC).
- Trans-critical CO_2 refrigeration cycle using flash gas inter-cooler (FGI).
- Trans-critical CO_2 refrigeration cycle using flash gas bypass (FBG).

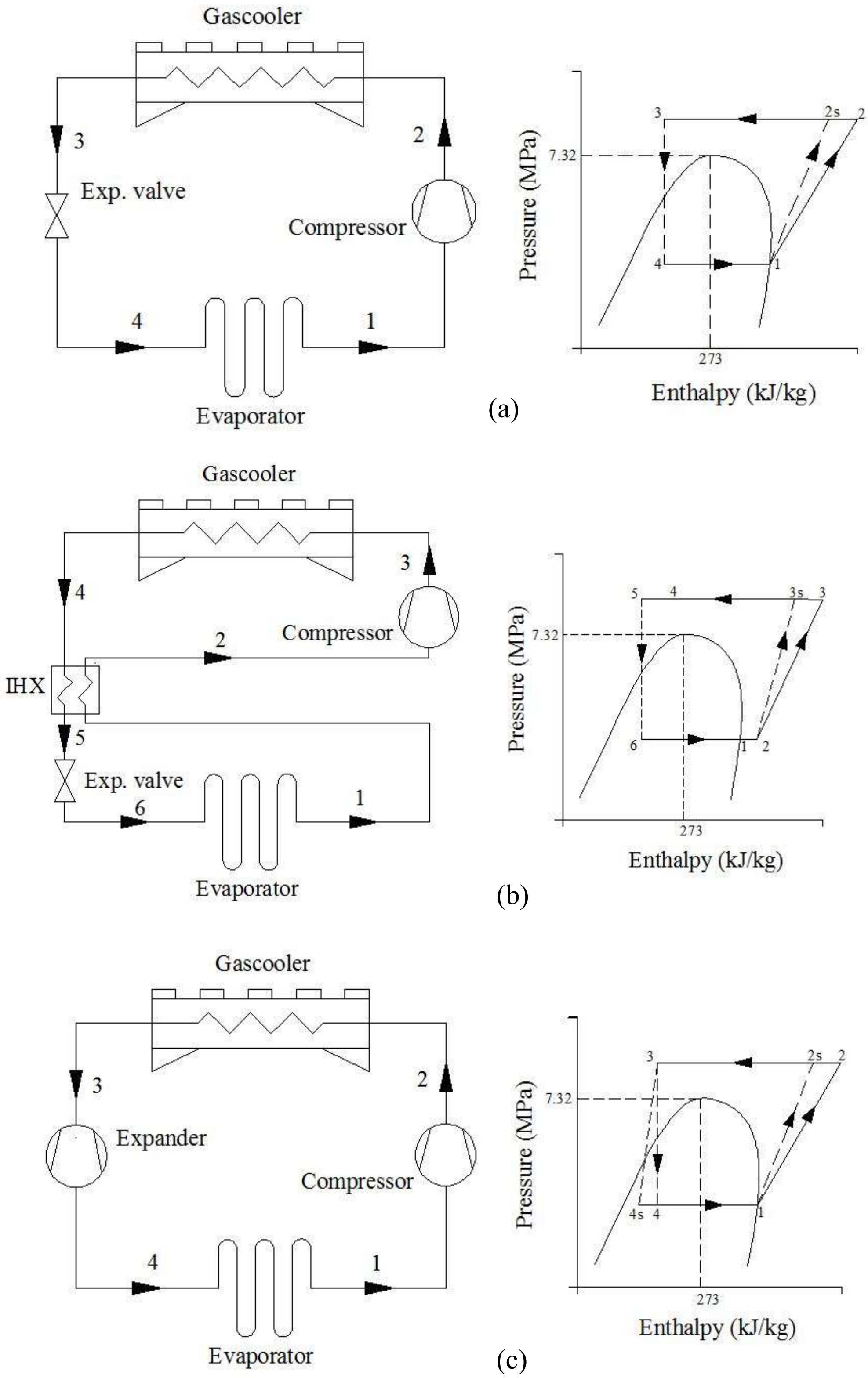
The range of applications covers are chiller, domestic refrigeration & air conditioning system and commercial refrigeration with chosen evaporator temperature of -10°C, 0°C and 10°C. A few other parameters which affect design and operation are also included in the study such as compressor discharge pressure, temperature, mass flow rate and exergy destruction. Effect of real time constraints like approach temperature, pressure drop in gascooler, compressors efficiency, degree of superheat, expander's efficiency and effectiveness of IHX are also incorporated. Schematic diagram and corresponding p-h diagrams, developed using Refprop (9.0) for various cycle modifications and are shown in Figure (4.1,b-g).

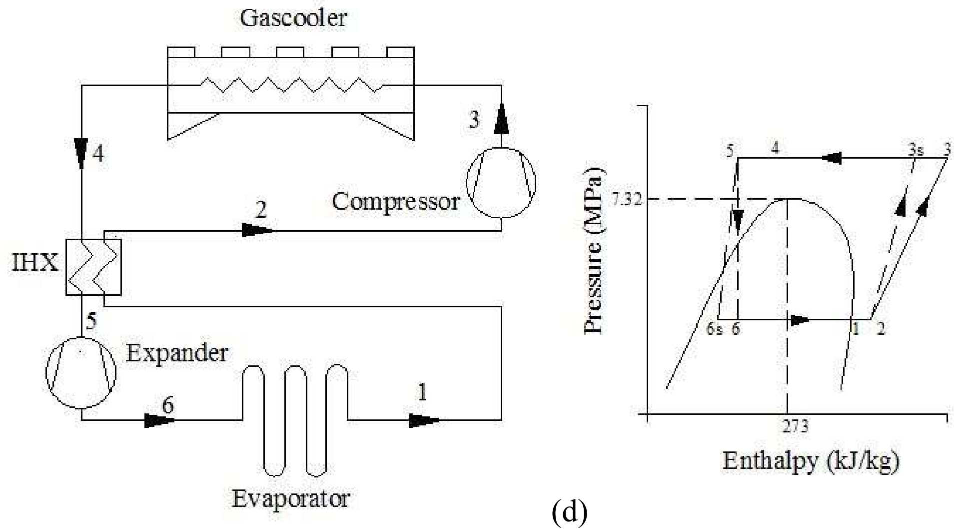
Figure (4.1,b) shows incorporation of IHX in basic trans-critical CO_2 refrigeration cycle. IHX is used in the cycle to increase the degree of sub cooling at the gascooler outlet, which enhances the refrigeration effect at the evaporator. Although due to the degree of superheating with the IHX, work consumption for compressor also increases.

Figure (4.1,c) shows work recovery expander, incorporated in the basic trans-critical CO_2 refrigeration cycle replacing expansion valve. The expander is used in the cycle for mechanical power generation, which can be further utilized for partially driving the compressor. The power generated contributes towards decreasing the system's overall power consumption. The initial investment for exploring work recovery expander is higher and an additional issue of gainful utilization of the work generated needs to be addressed.

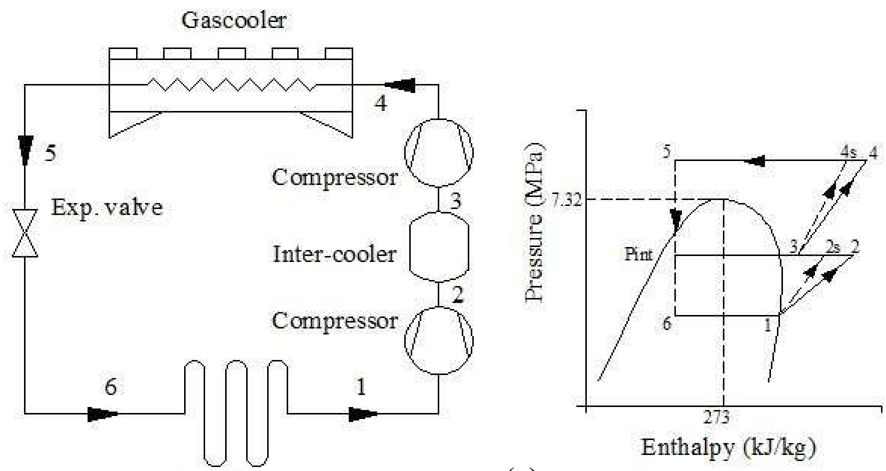
Figure (4.1,d) shows cycle incorporating IHX with work recovery expander in basic trans-critical CO_2 refrigeration cycle. This configuration contributes towards increase in the refrigeration effect by increasing both the sub cooling of vapors at the exit of the gascooler and mechanical power generation by work recovery expander during expansion or throttling process.

Figure (4.1,e) shows the incorporation of multi-stage compressor in basic trans-critical CO_2 refrigeration cycle along with IC using air as external fluid. Work input to the compressor 2 (Figure 4.1,e) is reduced due to a reduction in the specific volume of the refrigerant caused by cooling of vapor at the exit of compressor 1.

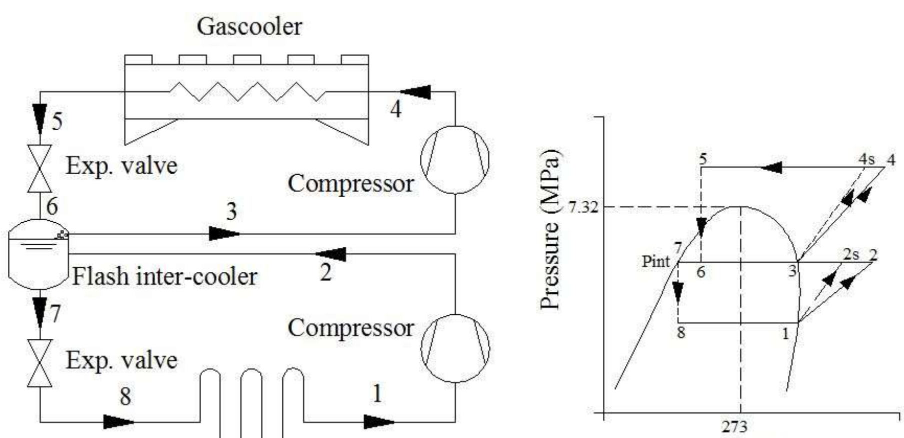

Gascooler
Exp. valve
Compressor
Evaporator
Pressure (MPa)
Enthalpy (kJ/kg)
7.32
273
(a)
Gascooler
IHX
Compressor
Exp. valve
Evaporator
Pressure (MPa)
Enthalpy (kJ/kg)
7.32
273
(b)
Gascooler
Expander
Compressor
Evaporator
Pressure (MPa)
Enthalpy (kJ/kg)
7.32
273
(c)


Gascooler
Compressor
IHX
Expander
Evaporator
1
2
3
4
5
6
Pressure (MPa)
Enthalpy (kJ/kg)
7.32
273
5
4
3s
3
6s
6
1
2

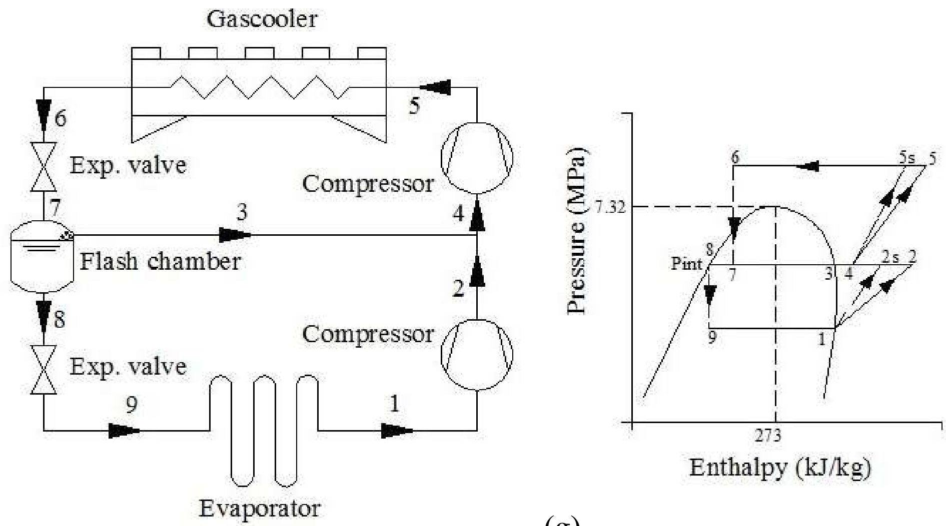
(d)


Gascooler
Compressor
Inter-cooler
Compressor
Exp. valve
Evaporator
1
2
3
4
5
6
Pressure (MPa)
Enthalpy (kJ/kg)
7.32
273
Pint
5
4s
4
3
2s
2
6
1

(e)


Gascooler
Compressor
Compressor
Exp. valve
Exp. valve
Flash inter-cooler
Evaporator
1
2
3
4
5
6
7
8
Pressure (MPa)
Enthalpy (kJ/kg)
7.32
273
Pint
5
4s
4
7
6
3
2s
2
8
1

(f)



(g)

Figure 4.1: Trans-critical CO_2 refrigeration cycles.

Figure (4.1,f) shows the introduction of FGI in the basic trans-critical CO_2 refrigeration cycle. The function of FGI is to interrupt the vapor at inter stage pressure and de-superheat the vapor back to the saturation line which further reduces the compression effort by compressor 2 (Figure 4.1,g).

Figure (4.1,g) shows the introduction of FGB in the basic trans-critical CO_2 refrigeration cycle. The function of FGB is to increase the refrigeration effect, reducing the amount of flash gas entering in the evaporator. It reduces the effort of compressor 1 (Figure 4.1,g) and their by reduces the power consumption of the system.

Steady flow energy equation, mass balance and exergy equations are used to make a simulation based study for comparative analysis of various modification options of trans-critical CO_2 refrigeration system at high ambient context (typical Indian conditions). The basic cycle and its six major modifications are analyzed to identify their effect on systems COP at high ambient temperature. In multi-stage trans-critical CO_2 refrigeration cycle, the choice of intermediate pressure is also very important in improving COP. However, for the systems using FGI and FGB, the intermediate pressure is optimized simultaneously along with the gascooler pressure. In practice this will call for a robust control system to be implemented. In order to simplify the computational modeling, following assumptions are made:

- Heat loss from the various components to the surroundings is negligible.
- Single phase heat transfer occurs with the external fluid in IC.
- Evaporation, gas cooling and intercooling processes are isobaric.
- A correlation for isentropic efficiency for compressor is used [13].

4.1.1. Energy analysis of the system and its components

Energy analysis of the various components of trans-critical CO_2 refrigeration cycle is carried out by using the Eqs. (4.1–4.17):

Compressor work

$$W_{comp,ss(bc,e)} = m(h_2 - h_1) \tag{4.1}$$

$$W_{comp,ss(ihx,eihx)} = m(h_2 - h_1) \tag{4.2}$$

$$W_{comp,ms(ic)} = m((h_2 - h_1) + (h_4 - h_3)) \tag{4.3}$$

$$W_{comp,ms(fgi)} = m_{1(fgi)}(h_2 - h_1) + m_{2(fgi)}(h_4 - h_3) \tag{4.4}$$

$$W_{comp,ms(fgb)} = m_{1(fgb)}(h_2 - h_1) + m_{2(fgb)}(h_5 - h_4) \tag{4.5}$$

Refrigeration effect of evaporator

$$Q_{ev,ss(bc,e)} = m(h_1 - h_4) \tag{4.6}$$

$$Q_{ev,ss(ihx,eihx)} = m(h_1 - h_6) \tag{4.7}$$

$$Q_{ev,ms(ic)} = m(h_1 - h_6) \tag{4.8}$$

$$Q_{ev,ms(fgi)} = m_{1(fgi)}(h_1 - h_8) \tag{4.9}$$

$$Q_{ev,ms(fgb)} = m_{1(fgb)}(h_1 - h_9) \tag{4.10}$$

Gascooler capacity

$$Q_{gc,ss(bc,e)} = m(h_2 - h_3) \tag{4.11}$$

$$Q_{gc,ss(ihx,eihx)} = m(h_3 - h_4) \tag{4.12}$$

$$Q_{gc,ms(ic)} = m(h_4 - h_5) \tag{4.13}$$

$$Q_{gc,ms(fgi)} = m_2(h_4 - h_5) \tag{4.14}$$

$$Q_{gc,ms(fgb)} = m_2(h_5 - h_6) \tag{4.15}$$

Isentropic efficiency of compressor

$$\eta_{is=}\ 0.815 + 0.022\left(\frac{P_2}{P_1}\right) - 0.0041\left(\frac{P_2}{P_1}\right)^2 + 0.0001\left(\frac{P_2}{P_1}\right)^3 \tag{4.16}$$

System performance.

$$COP = \frac{Q}{W} \tag{4.17}$$

4.1.2. Exergy analysis of the system

Exergy analysis of the various components of trans-critical CO_2 refrigeration cycle is done by using the following Eqs. (4.18–4.33):

Irreversibility of compressor and expander

$$I_{c,ss(bc,e)} = m(To(s_2 - s_1)) \tag{4.18}$$

$$I_{c,ss(ihx,eihx)} = m(To(s_3 - s_2)) \tag{4.19}$$

$$I_{c,ms(ic)} = m(To(s_2 - s_1) + To(s_4 - s_3)) \tag{4.20}$$

$$I_{c,ms(fgi)} = (m_{1(fgi)}(To(s_2 - s_1))) + (m_{2(fgi)}(To(s_4 - s_3))) \tag{4.21}$$

$$I_{c,ms(fgb)} = (m_{1(fgb)}(To(s_2 - s_1))) + (m_{2(fgb)}(To(s_5 - s_4))) \tag{4.22}$$

Irreversibility of inter-cooler

$$I_{c,ms(ic)} = m((h_2 - h_3) - To(s_2 - s_3)) \tag{4.23}$$

Irreversibility of gascooler

$$I_{gc,ss(bc,e)} = m((h_2 - h_3) - To(s_2 - s_3)) \tag{4.24}$$

$$I_{gc,ss(ihx,eihx)} = m((h_3 - h_4) - To(s_3 - s_4)) \tag{4.25}$$

$$I_{gc,ms(ic)} = m((h_4 - h_5) - To(s_4 - s_5)) \tag{4.26}$$

$$I_{gc,ms(fgi)} = m_{2(fgi)}((h_4 - h_5) - To(s_4 - s_5)) \tag{4.27}$$

$$I_{gc,ms(fgb)} = m_{2(fgb)}((h_5 - h_6) - To(s_5 - s_6)) \tag{4.28}$$

Irreversibility of evaporator

$$I_{ev,ss(bc,e)} = m(To(s_1 - s_4) - q_{ev}\left(\frac{To}{Tr}\right)) \tag{4.29}$$

$$I_{ev,ss(ihx,eihx,ic)} = m(To(s_1 - s_6) - q_{ev}\left(\frac{To}{Tr}\right)) \tag{4.30}$$

$$I_{ev,ms(fgi)} = m_{1(fgi)}(To(s_1 - s_9) - q_{ev}\left(\frac{To}{Tr}\right)) \tag{4.31}$$

$$I_{ev,ms(fgb)} = m_{1(fgb)}(To(s_1 - s_9) - q_{ev}\left(\frac{To}{Tr}\right)) \tag{4.32}$$

Second law efficiency

$$\eta_{II} = \frac{W_{comp} - I_t}{W_{comp}} \tag{4.33}$$

Based on the Eqs. (4.21–4.33), simulation models are developed for the six modifications of basic trans-critical CO_2 refrigeration cycle in Matlab platform. The program takes refrigerant property data from Refprop (9.0) automatically during computation.

The simulation is so designed as to extract the maximum COP and corresponding operating parameters (COP, discharge pressure, compressor discharge temperature, mass flow rate and total exergy destruction) for complete range of gascooler outlet temperature and pressure. The evaluation of all investigated cycles in our work is carried out for one fixed cooling capacity of 10 kW or 3.12 Ton. The upper limit of the inter-stage pressure for FGB and FGI multistage systems are restricted below the critical pressure. However, in IC multi-stage system, the inter-stage pressure is kept free to optimize below or above the critical pressure corresponding to maximum COP.

4.1.3. Evaluation of the various modifications of the system

The range and values of the fixed and variables parameters used in the simulation based study is tabulated in Table (4.1) and the performance evaluation of the modified systems are tabulated in appendix (AT2).

Table 4.1: Parameter variation for the simulation.

Parameters	Value/Range
Evaporator temperature	-10°C, 0°C, 10°C
Gascooler outlet temperature	35°C to 55°C
Compressor outlet pressure	8MPa to 12MPa
Effectiveness of IHX	70% [19]

4.1.3.1. Impact on COP

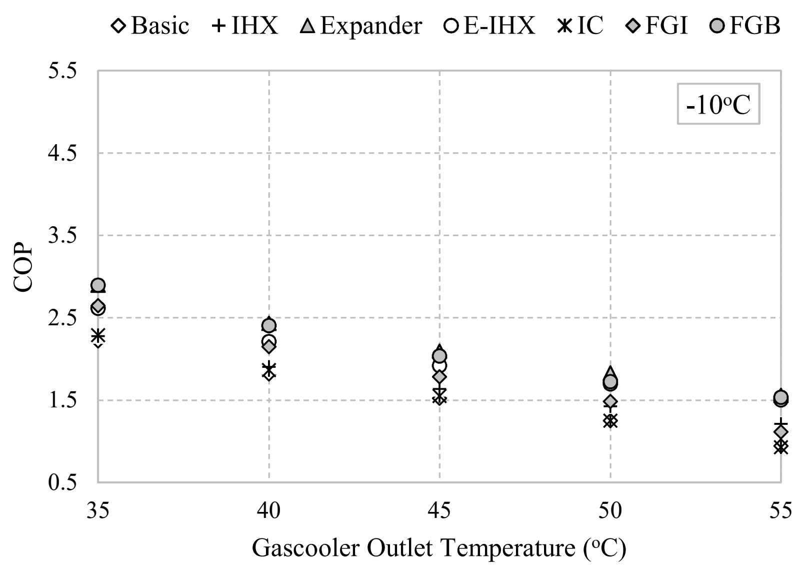
Figure (4.2) shows the impact on COP with gascooler outlet temperature for the different evaporator temperature at optimum gascooler pressure. The approach temperature is taken to be zero implying the ambient air temperature to be equal to the gascooler outlet temperature.

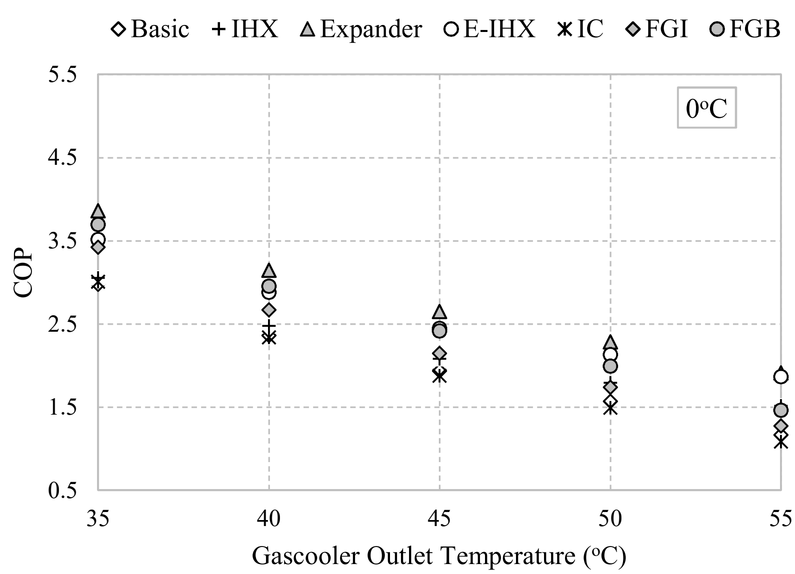
Observations made from the study on variation of COP with gascooler outlet temperature are as follows:

- A sharp decrement in system's COP is observed with increasing gascooler outlet temperature across all evaporator temperature. This can be attributed to the fact that increase in gascooler outlet temperature lead to increase in throttling losses which further results into higher irreversibility.
- With increase in evaporator temperature the systems COP increases. This attributed, mainly due to the fact that increase in evaporator temperature, decreases the required

compression and also as for same gascooler pressure the temperature lift between evaporator temperature and compressor discharge temperature decreases.

- Across the range, the COP of the system with work recovery is comparatively high due to work recovery in place of throttling losses. Also, the performance of systems with IHX is comparatively higher with respect to basic system.





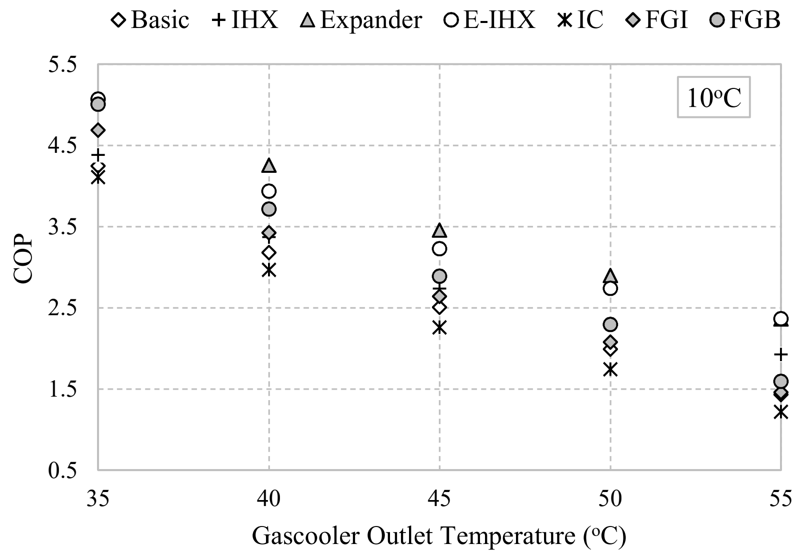


Figure 4.2: Variation of COP with gascooler outlet temperature.

- Among the multi-stage systems, the performance of FGI is better for analyze evaporator temperatures due to the flash intercooling in the system which results in less flash gas being available for second stage compressor as compared with FGB system.

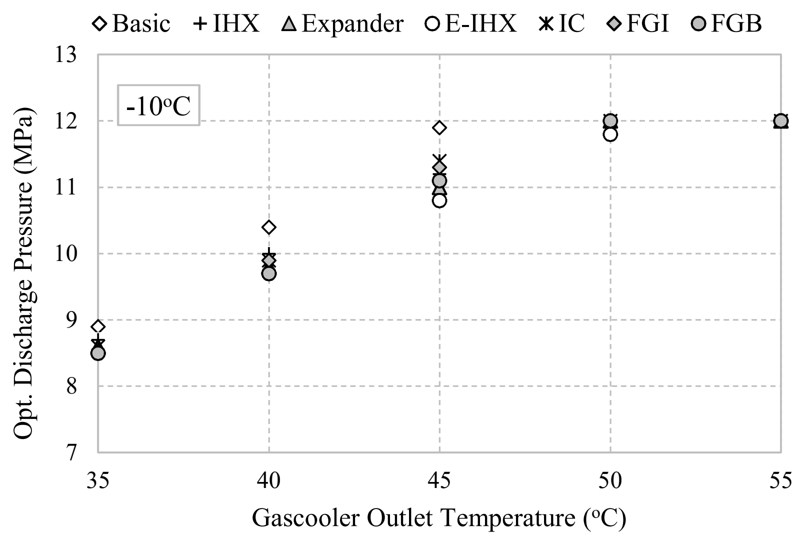
4.1.3.2. Discharge pressure variation

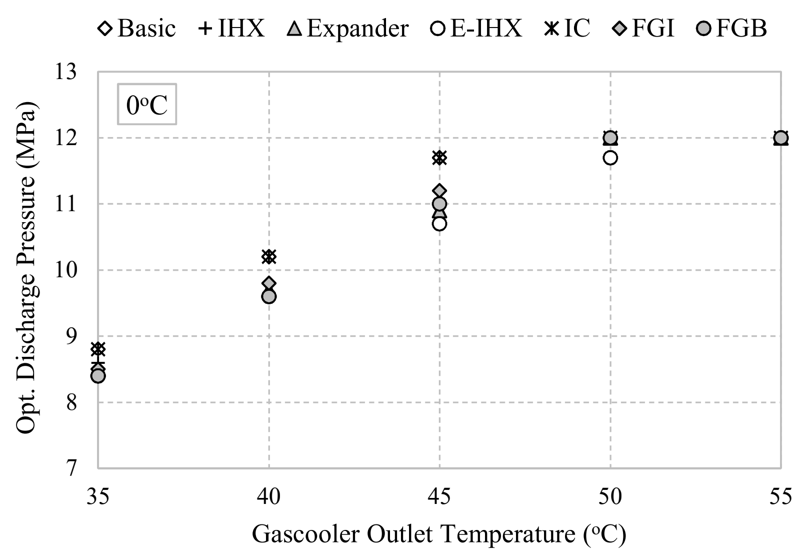
One of the major challenges associated with trans-critical CO_2 refrigeration cycle is requirement of higher discharge pressure at higher ambient temperature. As the operating range for our simulation is 35°C to 55°C, we have operated the cycles for minimum 8MPa gascooler inlet pressure, below this range the cycle can't be operated for full range of ambient temperature. Figure (4.3) shows, the variation of optimum discharge pressure with gascooler outlet temperature.

Observations made from the study of variation of recommended discharge pressure with gascooler outlet temperature for maximum COP are:

- Near the critical temperature zone, the operating pressure of gascooler is 8MPa for all the investigated systems.

- A sharp increment in compressor discharge pressure is observed with increase in gascooler outlet temperature beyond the critical temperature. As the evaporator temperature increases, there is a marginal decrease observed in discharge pressure.
- At high ambient temperature, the discharge pressure is found to be the lowest for the systems using expander with IHX. Also, the systems with IHX are found to have comparatively lower discharge pressure then rest. This is due to the fact that for a fixed cooling capacity, a lower mass flow rate results in lower operating pressure.





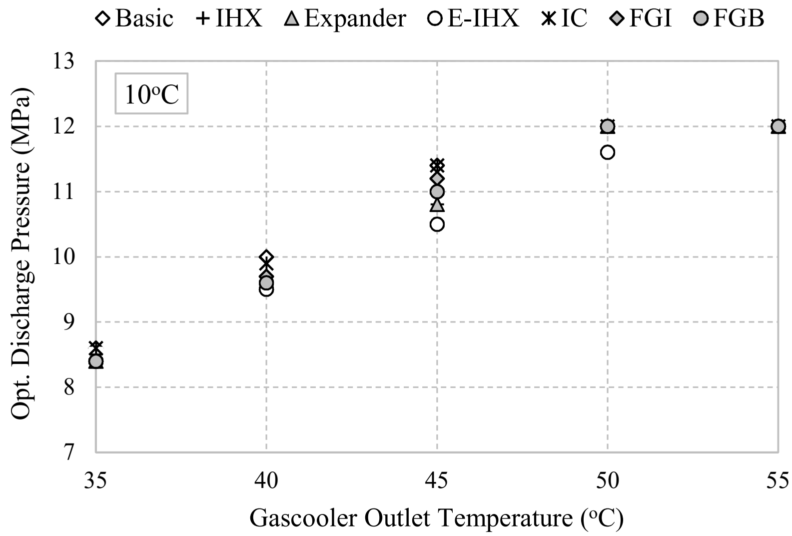


Figure 4.3: Variation of optimum discharge pressure with gascooler outlet temperature.

- For high evaporating temperature, the discharge pressure for FGB and FGI multi-stage systems are found to be similar and they also exhibit similar behavior as the basic system.

In all investigated cases, the discharge pressure for multi-stage systems with IC are found to be comparatively higher because the optimum inter-stage pressure is near or above the critical point. Which results in higher operating pressure due to the typical S-shaped isotherms of CO_2.

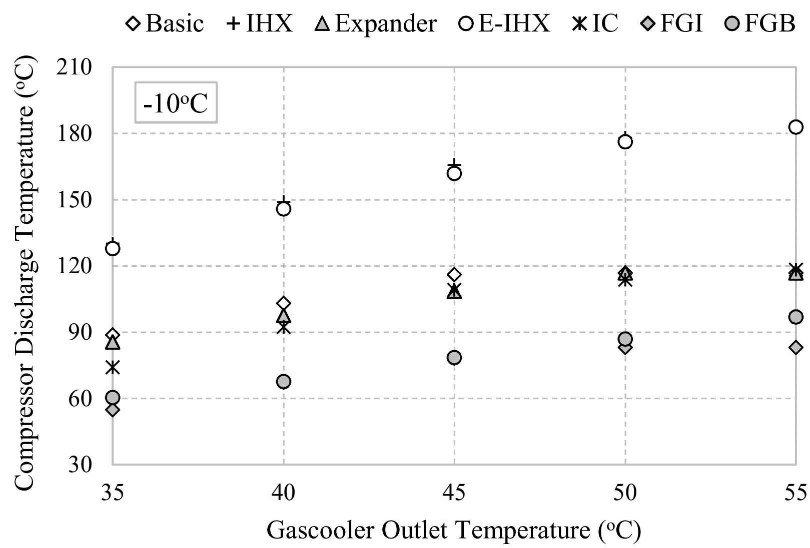
4.1.3.3. Compressor discharge temperature variation

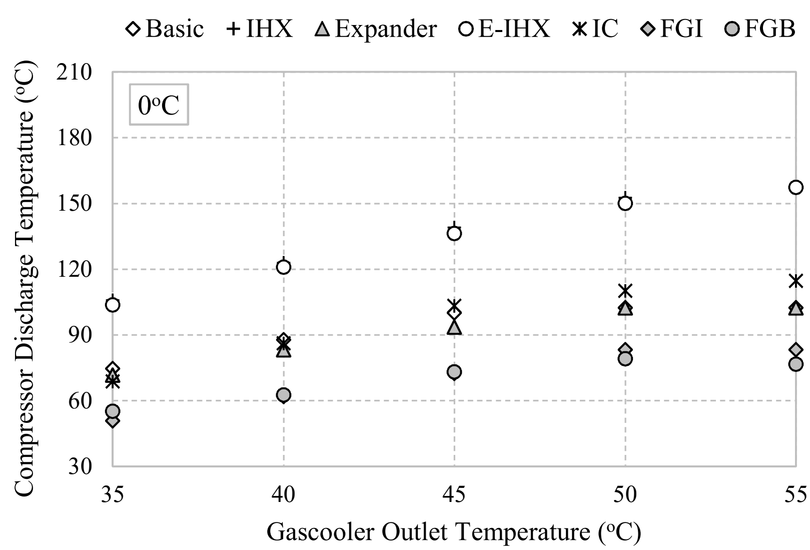
Figure (4.4) shows the variation of compressor discharge temperature with gascooler outlet temperature. The compressor discharge temperature plays a vital role in the evaluation of component design and system performance. In practice, the quality of lubricating oil degrades at high discharge temperatures. This results in reduction in compressor performance.

Observations made from the variation of compressor discharge temperature with gascooler outlet temperature for maximum COP are:

- The compressor discharge temperature is found to be higher for the systems with IHX for all the three analyzed evaporator temperatures. This is ascribed to the super-heating effect to the vapors at the suction port of the compressor. However, the systems with IHX are also found to have a comparatively better COP for all operating parameters.

- The discharge temperatures are found to be nearly similar for both the compression stages of the IC multi-stage system due to the higher optimized inter-stage pressure as discussed in article 3.





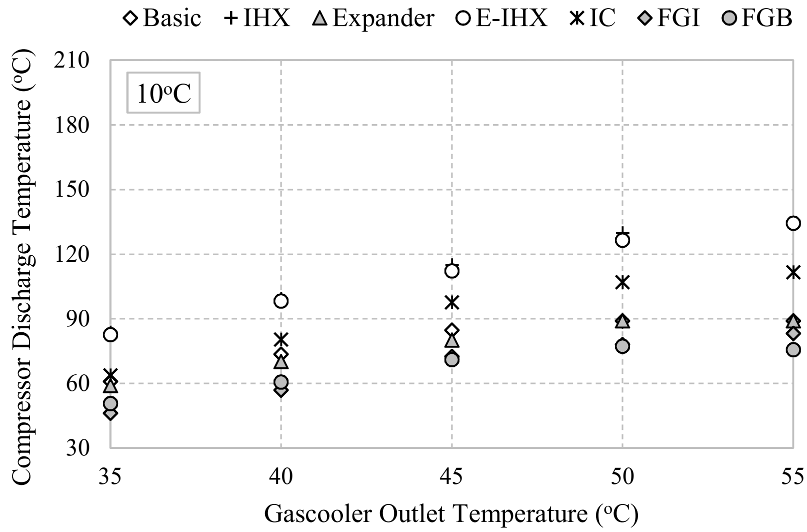
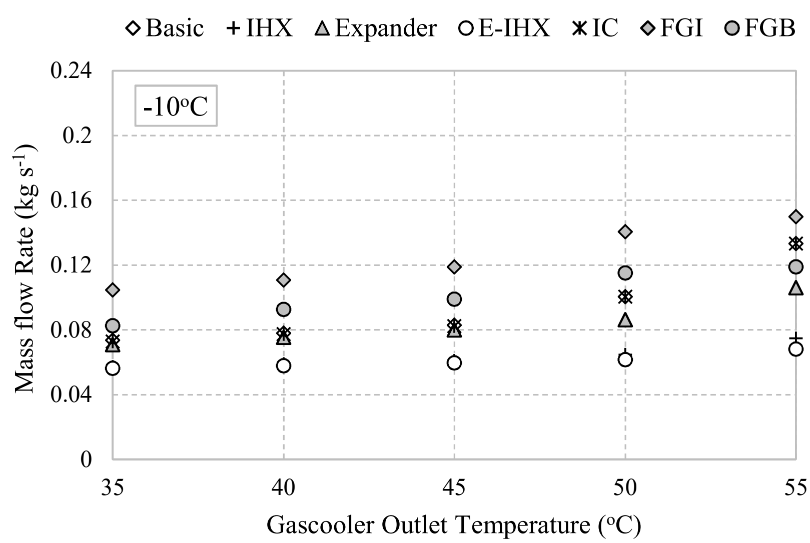
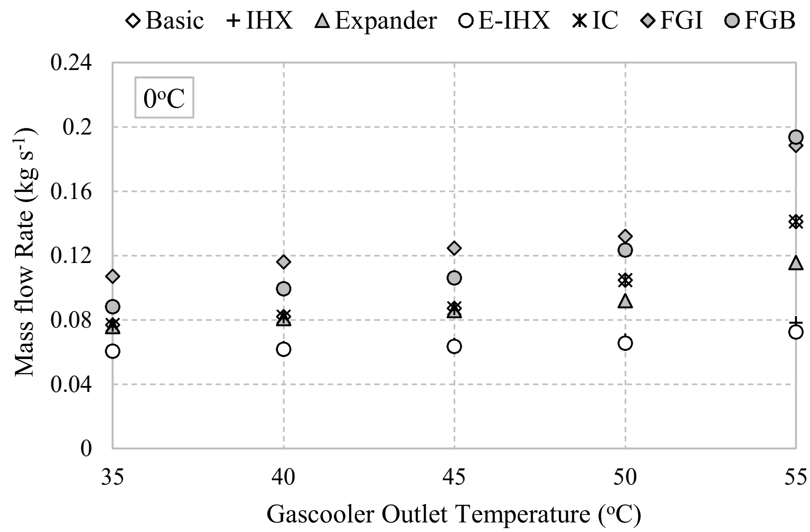


Figure 4.4: Variation of compressor discharge temperature with gascooler outlet temperature.

- It is observed that for all the evaporator temperatures, in FGB and FGI multi-stage systems, the discharge temperature of second stage compressors decreases slightly near the critical region, and then there is a sharp increment. The unique properties of CO_2 above the critical point are the most likely reason for the same.

4.1.3.4. Rate of mass flow variation





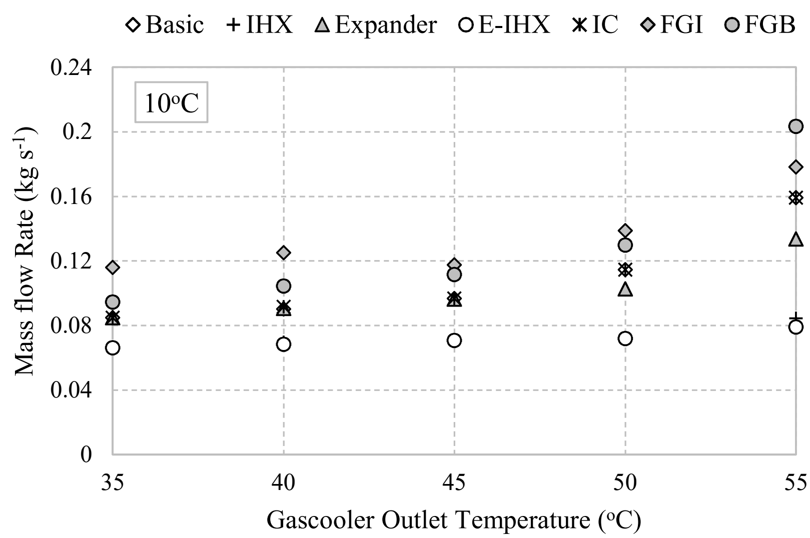


Figure 4.5: Variation of mass flow rate with gascooler outlet temperature.

Figure (4.5) shows the variation of mass flow rate with gascooler outlet temperature. Observations made from the variation of mass flow rate with gascooler outlet temperature for maximum COP are:

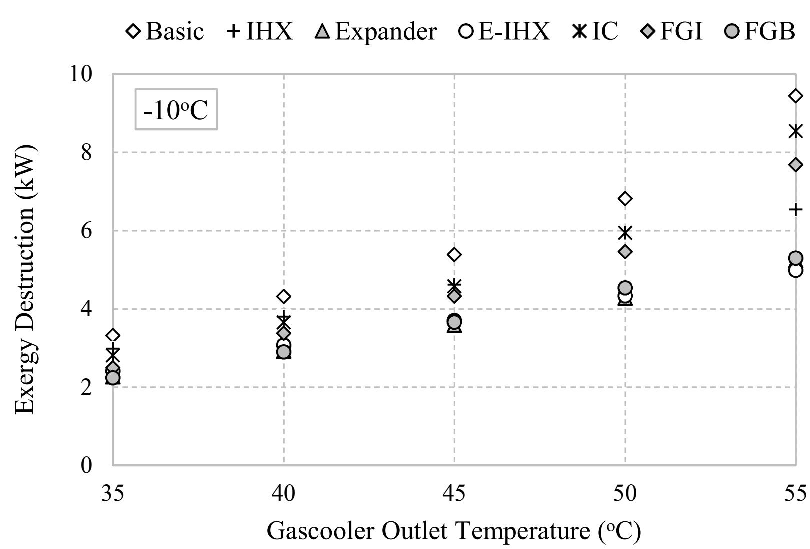
- To attain desired cooling capacity, lower mass flow rate is required for system with IHX. This is due to the presence of sub-cooling in IHX. This in-turn reduces the gascooler operating pressure as observed in Figure (4.3).

- In a system with IC and a system with expander, the mass flow rate is found to be nearly same as the basic CO_2 trans-critical system. In system configurations with expander, due to imposed isentropic efficiency of expander, the mass flow rate does not show improvement from basic cycle. In system with IC, for reduction in mass flow rate, more effective external fluid, in place of ambient air cooling can be explored.
- In FGB and FGI multi-stage systems, the mass flow rate in the first stage is observed to increase with increase in evaporator and gascooler outlet temperatures. In second stage, the mass flow rate fluctuates for all the three evaporator temperatures. Note that for both FGB and FGI the systems, the mass flow rate in second stage is comparatively higher than that in first stage due to the vapor coming from the flash chamber.

4.1.3.5. Exergy destruction

Exergy analysis is carried out for the evaluation of refrigeration systems based on the second law of thermodynamics. The variation of total exergy destruction with gascooler outlet temperature is shown as in Figure (4.6).

Observation from the total exergy destruction with gascooler outlet temperature for maximum COP are as follows:



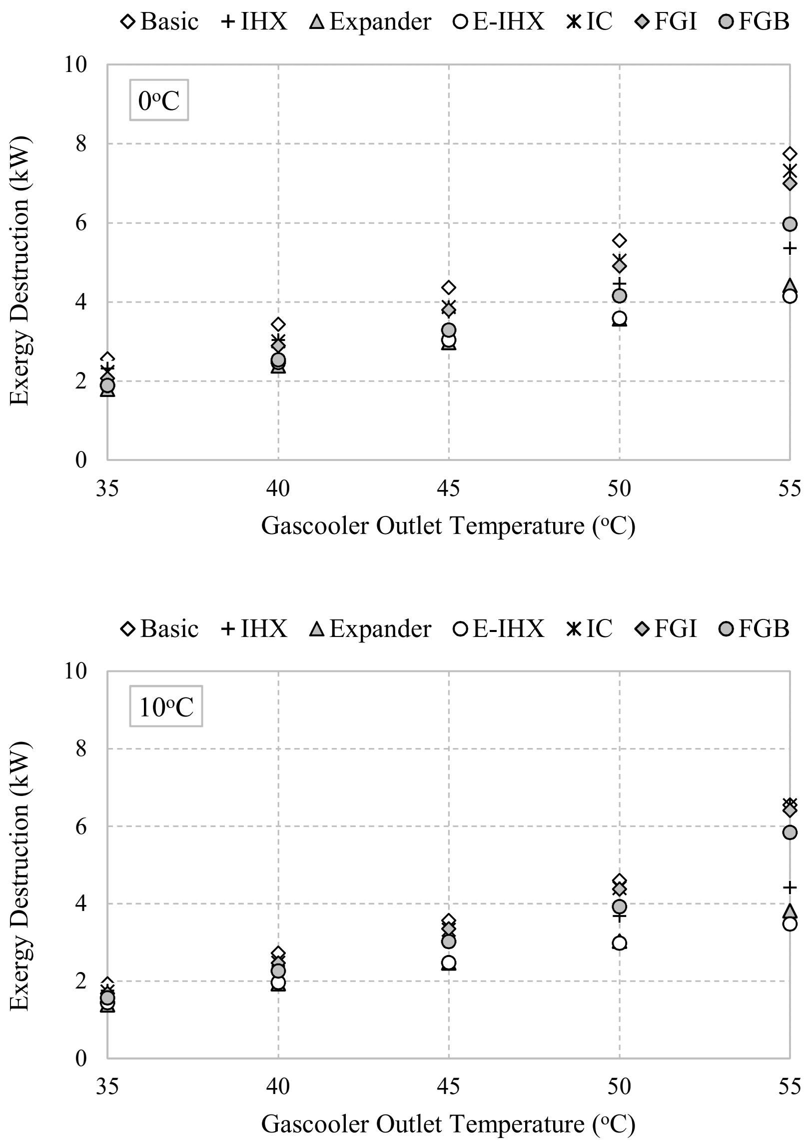


Figure 4.6: Variation of total exergy destruction with gascooler outlet temperature.

- As the gascooler outlet temperature increases the exergy losses in the system increases due to the increase in the optimum operating pressure ratio of the system.
- The exergy destruction of systems with expander is comparatively lower for all evaporating temperatures due to the work recovered from the system itself using work recovery expander.

- The exergy destruction in FGB and FGI multi-stage systems, increases sharply at high gascooler outlet temperature, for all evaporator temperatures. This is due to the increased mass flow rate in the second stage compression. Among the multi-stage systems, use of IC is found to have comparatively lower exergy destruction.

4.1.3.6. Effect of a few real-time constraints

Effect of the real-time constraints such as approach temperature (difference between gascooler outlet temperature and ambient temperature), pressure drop in gascooler, compressors efficiency, degree of superheat, expanders efficiency and effectiveness of IHX play an influential effect on systems COP as projected in Figure (4.7).

Real time constraints adopted in the study is to relate the thermodynamic analysis of six modifications of basic CO_2 refrigeration system with real time practicability for a particular domestic refrigeration application at 0°C. The range and values of the real-time constraints used in simulation based study is listed in Table (4.2) and performance evaluation of the modified systems using real time constraints is tabulated in appendix (AT3).

Table 4.2: Real time constraints variation for the simulation.

Parameters	Value/Range
Compressor outlet pressure	8MPa to 12MPa
Approach temperature	0°C to 5°C
Pressure drop in gascooler	0.1MPa to 0.5MPa
Isentropic efficiency of compressor	60% to 85%
Isentropic efficiency of expander	50% to 75%
Effectiveness of IHX	60% to 85%
Superheating at compressors intlet	0°C to 5°C

Observation from the variation of real time constraints with gascooler outlet temperature are as follows:

- Systems COP decreases with increase in approach temperature.

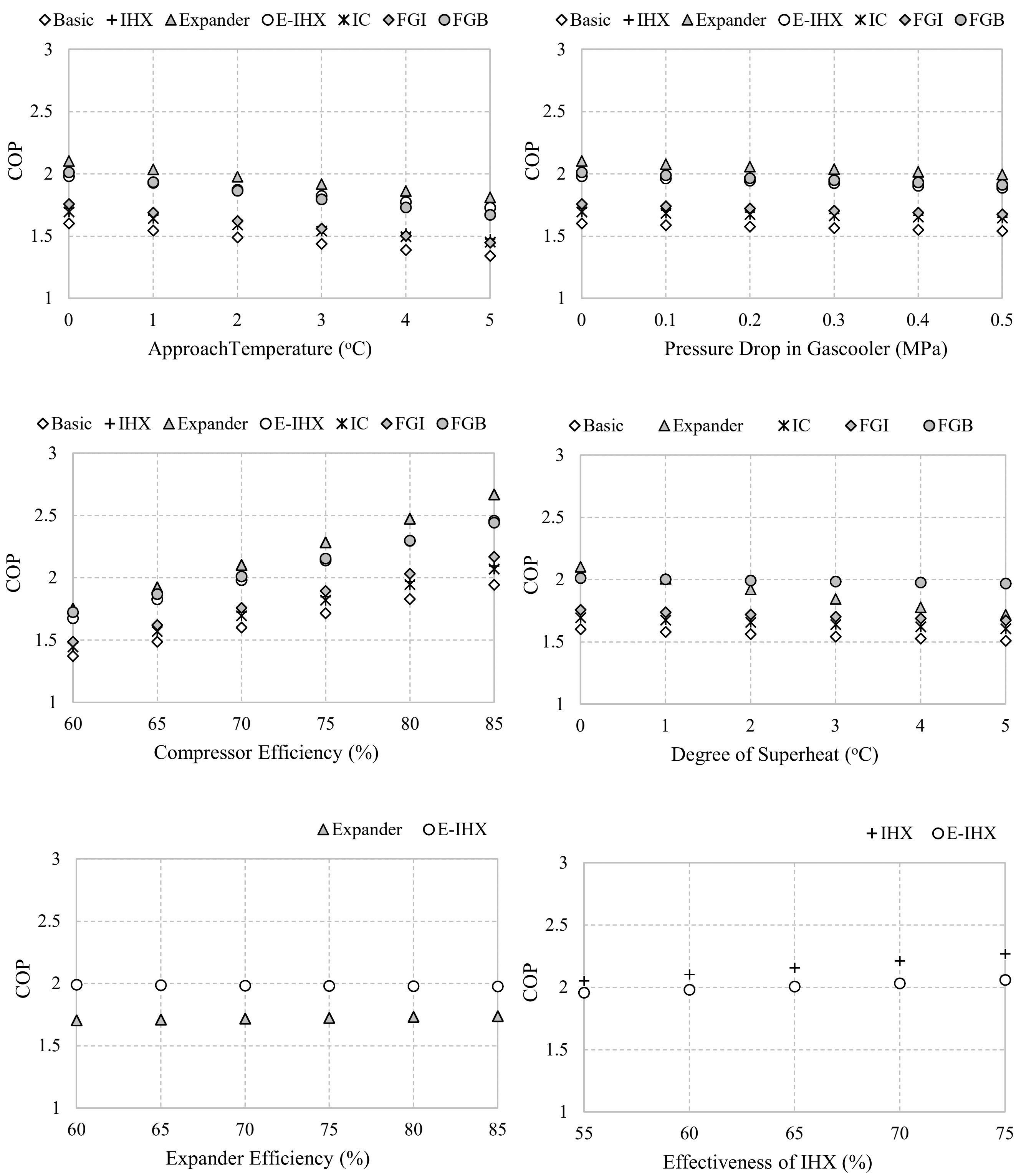


Figure 4.7: Variation of real time constrains with gascooler outlet temperature.

- Pressure drop in gascooler contributes towards reduction in systems COP. However, the variation of COP with pressure drop is less significant as compare to that with approach temperature.
- Compressor and expander efficiency influences the systems COP and with their increment, performance of the cycle improves owing to the reduction in irreversibility.
- In practical systems, degree of superheat ensures dry vapors to the compressors while it has negative effect on systems COP. As the degree of superheat increases the COP decreases. However, the rate of decrement is insignificant.
- At higher ambient operating temperatures, the use of IHX in basic cycle enhances the system performance and increases with increase in effectiveness of IHX.

  The effect of expander efficiency on system's COP is found insignificant.

4.2. Cascade Refrigeration System

The COP of CO_2 trans-critical system decreases drastically for its high temperature trans-critical application. So, it will be advantageous to use CO_2 in cascade refrigeration system for low temperature applications. A suitable selection of refrigerant for the high temperature cycle (HTC) of a cascade system can provide high overall system efficiency with minimum harm to environment.

A cascade system with low GWP refrigerants R1234yf is proposed and analyzed. R1234yf is one of the two relatively recent synthetic refrigerants introduced by Honeywell aving very low GWP. Energy based analysis of the cascade system configuration is carried out for low temperature application running at high ambient temperature. Intermediate temperature (IT) of the cascade condenser is optimized on the basis of lowest TEWI on the environment. Influence of condenser temperature, evaporator temperature, efficiency of IHX, IT and OT of the cascade condenser/evaporator on the system performance is contrasted.

Figure (4.8) shows schematic of an R1234yf/R744 (CO_2) cascade refrigeration system configuration, consisting of two single stage vapor compression systems. R1234yf is used in the HTC and R744 (CO_2) is used in LTC. IT of cascade condenser is an important parameter, to get the optimum COP.

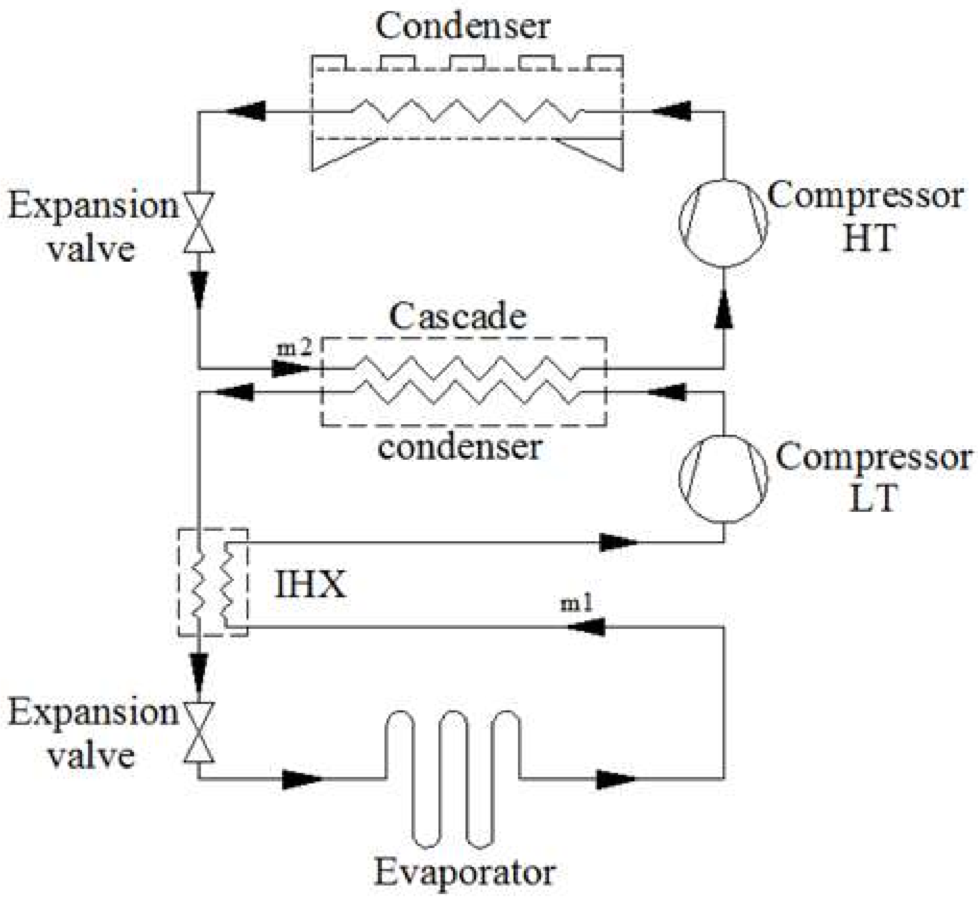


Figure 4.8: Schematic of R1234yf/R744 (CO_2) cascade system.

Additionally, an IHX is employed in the LTC. The configuration with IHX is an established modification which contributes towards improvement in refrigeration effect by increasing the sub cooling of vapors at the exit of the cascade condenser. Steady flow energy equation and mass balance equation are used to conduct a simulation based study of R1234yf/R744 (CO_2) cascade refrigeration system at high ambient temperature for low evaporator temperature applications.

In order to simplify the computational model, following assumptions are made.

- Heat loss is negligible from piping and compressor.

- Evaporation and condensing processes are isobaric.

Further, thermodynamic analysis of the various components, is carried out using Eqs. given in section 4.1.1. The value/ range of the parameters used during simulation based study is listed in Table (4.3) and performance evaluation of the cascade system is tabulated in appendix (AT4).

Table 4.3: Parameters used for simulation of cascade system.

Parameter	Value/Range
Condenser temperature	30ºC to 50ºC
Evaporator temperature	-40ºC to -20ºC
Cascade intermediate temperature	-30ºC to 30ºC
Overlap temperature	1ºC to 10ºC
Efficiency of IHX	50% to 100%
Cooling capacity	40kW
Overlap temperature	5ºC

TEWI scheme explained along with the mathematical equations in the article (2.2.2) is used to evaluate the performance of the cascade refrigeration system configuration. The proposed minimum TEWI scheme to fix the IT of cascade configuration, is validated through comparison with 56 different refrigerant combinations, tabulated in appendix (AT5). Analysis establishes the advantage of proposed scheme over existing scheme in each case.

Following assumptions are taken as per Emerson climate Technology [12]:

- Annual leakage rate 15%, for both CO_2 and HFO system.
- Total operational life of both systems are 10 years.
- 95% of the refrigerant is assumed to be recycled.
- Emission due to electricity generation 0.92 kg kWh^{-1}.

4.2.1. Variation of TEWI with IT of cascade system

Influence on TEWI depending upon selected IT of the R1234yf/R744 cascade system configuration is shown in Figure (4.9). To compute the influence on TEWI, condenser and evaporator temperature of overall system are kept constant at 50ºC and -40ºC respectively. The IT of cascade condenser is varied for a large range (-30 to -50ºC). It is observed that the minimum TEWI of the cascade system is lowest near 0ºC.

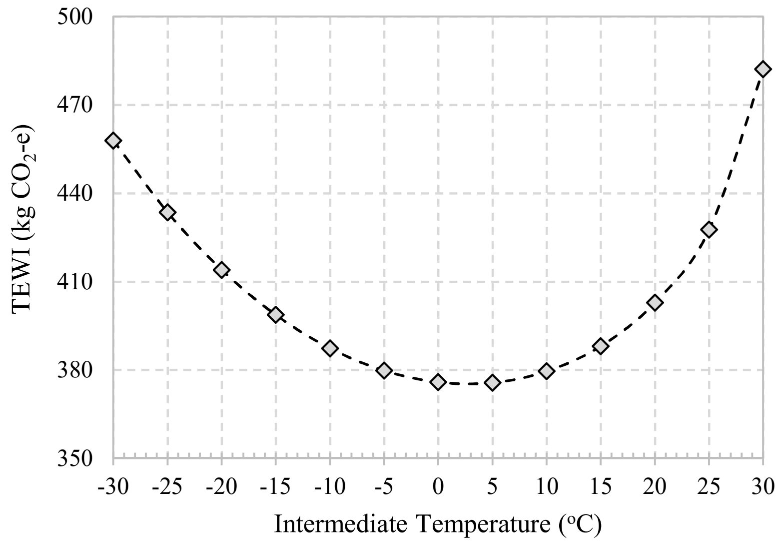


Figure 4.9: Variation of TEWI with IT of the cascade system.

4.2.2. Influence of condenser and evaporator temperature

To understand the effect of COP_{LTC} and COP_{HTC} on the overall system performance, both cascade and evaporator temperatures are varied independently.

Figure (4.8) shows the variation of COP with evaporator temperature, keeping condenser temperature and IT constant at -40ºC and 0ºC respectively. It is observed that the COP_{LTC} as well as COP_{sys} increases as the temperature lift increase, due to the reducing compressor work by the system results increasing system refrigerating effect.

Figure (4.9) shows the variation of COP with condenser temperature, keeping evaporator and IT temperature constant at 50ºC and 0ºC respectively. It is observed that the COP_{HTC} decrease, as the condenser temperature increase and contribute towards decrease in overall COP_{sys}. As the condenser temperature increase, the pressure lift between cascade condenser/evaporator increases, which increases in compressor work in HTC, hence COP decreases.

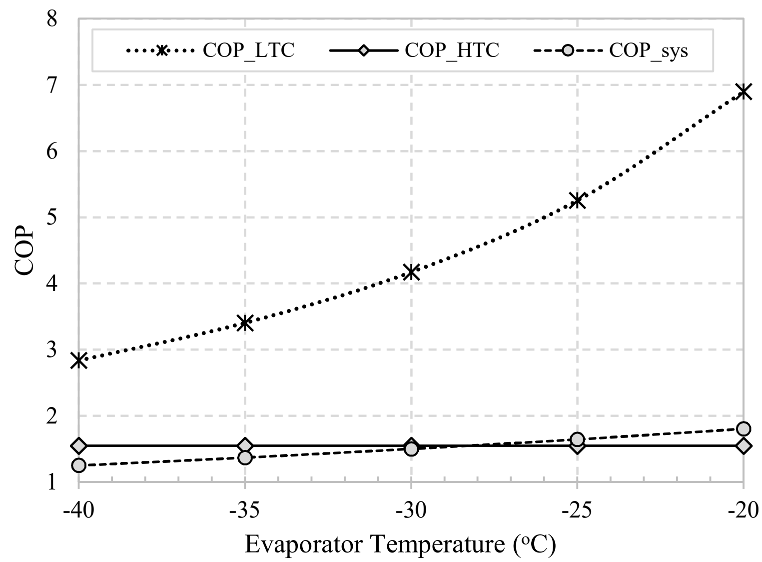


Figure 4.8: Variation of COP with evaporator temperature.

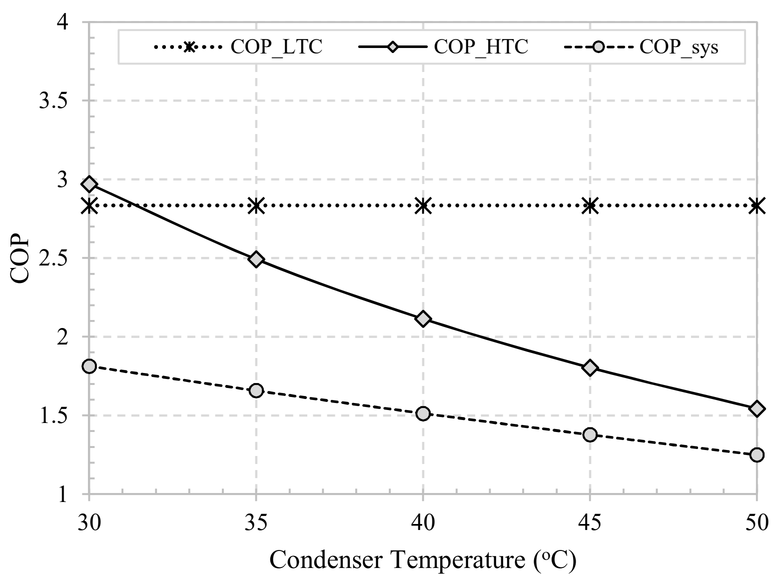


Figure 4.9: Variation of COP with condenser temperature.

4.2.3. Influence of IHX

Figure (4.10) shows the variation of COP with effectiveness of IHX, keeping evaporator, condenser and IT constant at -40ºC, 50ºC and 0ºC respectively. It is observed that the efficiency of the IHX has a marginal effect on the COP_{sys}.

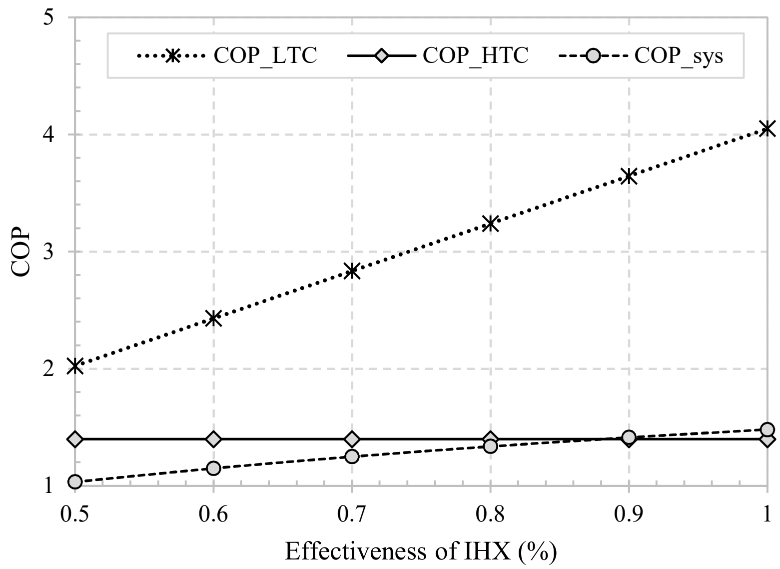


Figure 4.10: Variation of COP with efficiency of IHX.

4.2.4. Influence of IT and OT on COP_{sys}

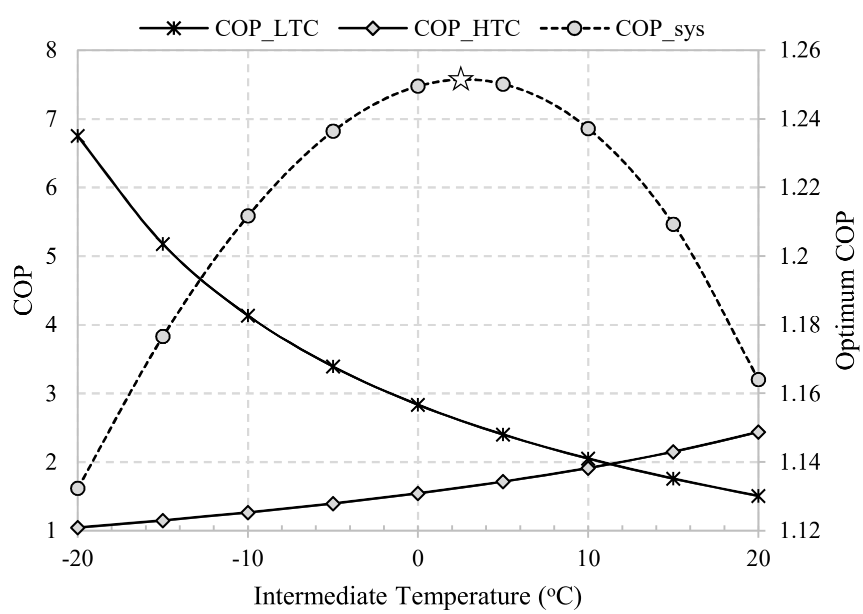


Figure 4.11: Variation of COP with IT.

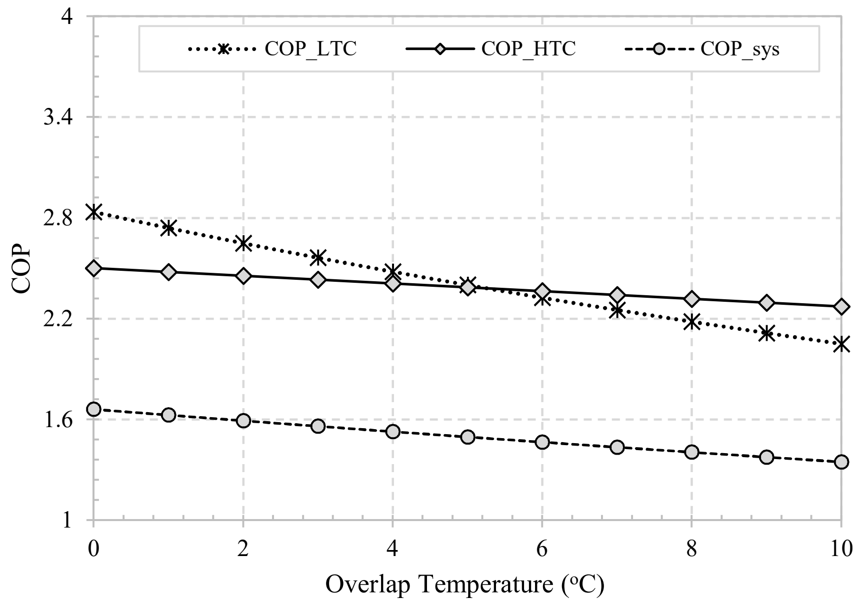


Figure 4.12: Variation of COP with OT.

From Figure (4.11) it is observed that, as IT increases, the temperature lift in LTC increases and HTC decreases, leading to increase in COP_{HTC} and decrease in COP_{LTC}. Similarly, when IT decreases, the temperature lift of HTC increases and LTC decreases, leading to increase in COP_{LTC} and decrease in COP_{HTC}. These two opposite phenomena result in a marginal increment initially in COP_{sys} and then a reduction, beyond optimum value of COP. The results also suggest that IT near 0ºC gives the maximum COP_{sys}.

From Figure (4.12) it is observed that, with the increase in OT, the overall COP_{sys} decreases. This implies, increasing OT leads to reduction in cascade condenser/evaporator effectiveness i.e. reducing heat transfer between the both refrigerants. As OT increases, COP_{HTC} & COP_{LTC} decreases, leading to decrease in COP_{sys}.

4.3. Scroll Work Recovery Expander

One prominent option and modification to enhance COP of trans-critical CO_2 system is to adopt a work recovery expander. Among the various work recovery expander options available, it is established in Chapter 2, that the scope of scroll expander is highest.

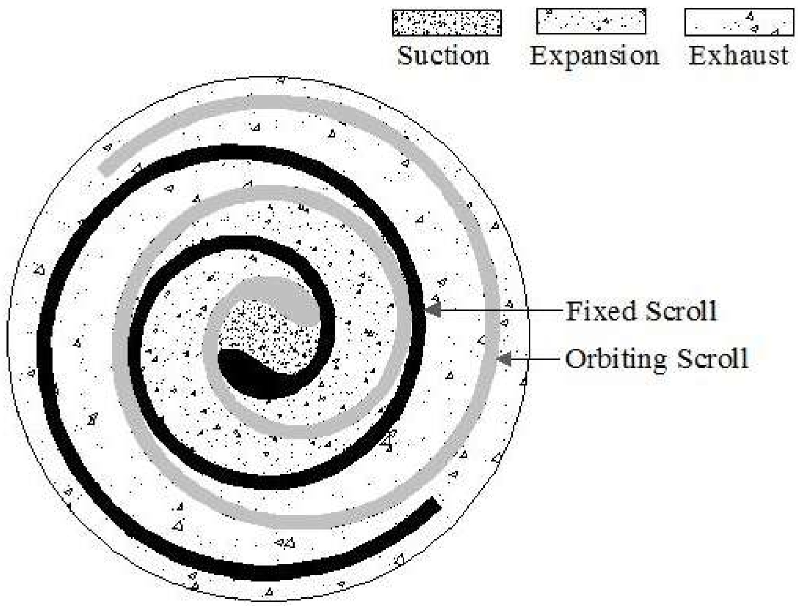


Figure 4.13: Scroll expander.

Working of scroll expander is based on two nested spirals, often called scrolls, one fixed while the other is imparted an orbital motion as CO_2 expands inside the device and produces work. The suction, expansion and exhaust process within a scroll expander during a single rotation of shaft are illustrated in Figure (4.13).

4.3.1. Semi-empirical modeling

In 2009, a semi-empirical model of scroll was developed to study the effect of design on the efficiency of scroll expander using synthetic refrigerants by V. Lemort [89]. An experimental study using R123 as working fluid in an organic Rankine cycle. Was reported. The semi-empirical model developed and reported is used by many other researchers later.

In the present work, in order to predict performance of scroll work recovery expander using CO_2 as a working fluid under trans-critical conditions the model is modified. The conceptual scheme is projected in Figure (4.14). For CO_2, the expansion process within the scroll will be trans-critical i.e. single phase to double phase. The model is suitably modified such that at the suction of the scroll expander, pressure and temperature are optimize strategically according to the gascooler outlet temperature to generate the maximum shaft work and maximize the COP of the system.

Flow through work recovery expander constitutes:

- Adiabatic supply pressure drop (*su*—*su1*).
- Isobaric supply cooling (*su1*—*su2*).
- Adiabatic and reversible expansion to the "adapted" pressure (*su2*—*ad*).
- Adiabatic expansion at a constant machine volume (*ad*—*ex2*).
- Adiabatic mixing between supply and leakage flow (*ex2*—*ex1*).

  Isobaric exhaust cooling-down or heating up (*ex1*—*ex*).

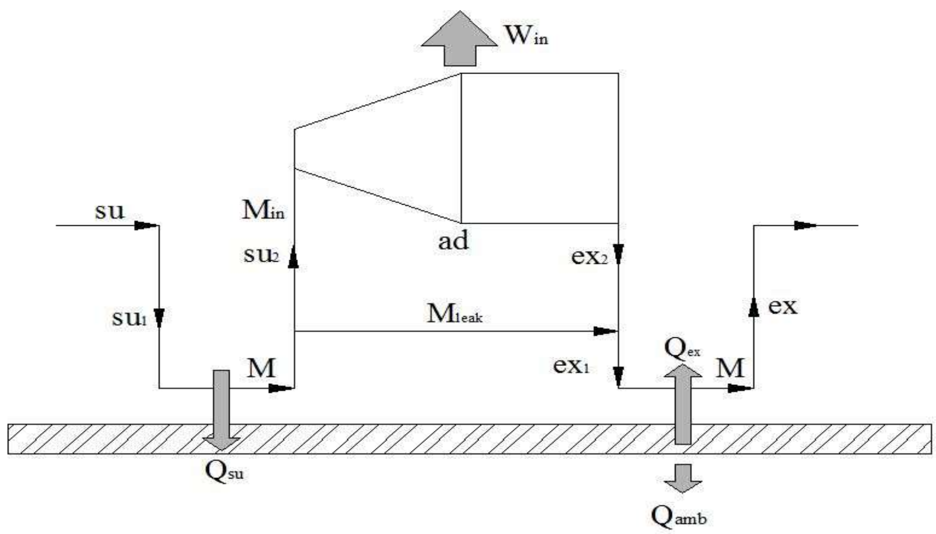


Figure 4.14: Conceptual scheme of scroll expander.

The discretization of the expander is elaborated as following.

Adiabatic supply pressure drop (su—su1): Pressure losses between the expander suction port and the suction chamber is accounted for with this process and resolved as an isentropic flow through the nozzle.

$$\dot{m} = \frac{A_{su}}{v_{thr,su}} \sqrt{2\left(h_{su} - h_{thru,su}\right)} \tag{4.34}$$

Isobaric supply cooling, $\dot{Q}_{su}$ (su1—su2): This process covers the heat transfer between fluid entering the suction chamber and the outer envelope of expander.

$$\dot{Q}_{su} = \dot{m}(h_{su1} - h_{su}) = \left[1 - e^{\frac{-k_{su}}{\dot{m}\,cp}}\right] \dot{m} \times C_p\,(T_{su} - T_w) \tag{4.35}$$

Adiabatic and reversible expansion upto throat of the expander called "adapted" *(su2—ad):* is related to the built-in volume ratio of the expander.

$$v_{in} = r_v \times v_{su2} \tag{4.36}$$

Adiabatic expansion at a constant volume inside machine (ad—ex2): The under and over expansion occurs at this stage when the adapted pressure is higher or lower respectively, compares to the system pressure at the expander outlet.

Isobaric exhaust cooling-down or heating-up, $\dot{Q}_{ex}$ (ex1—ex): This process covers the heat transfer between fluid leaving the discharge chamber and the outer envelope of expander.

$$\dot{Q}_{ex} = \dot{m}(h_{ex} - h_{ex1}) = \left[1 - e^{\frac{-k_{ex}}{\dot{m}\,cp}}\right] \dot{m} \times C_p\,(T_w - T_{ex}) \tag{4.37}$$

4.3.1.1. Performance evaluation of scroll expander

The total mass displaced in the expander ($\dot{m}$) is associated with mass in the scroll and mass leakage from the scroll due to the gap between top and bottom flank.

$$\dot{m} = \dot{m}_{in} + \dot{m}_{leak} = \frac{N}{v_{su2}} \frac{V_s}{r_{v,in}} + \dot{m}_{leak} \tag{4.38}$$

The heat transfer coefficient $\left(k_{CO_2}\right)$ is re-calculated for CO_2 refrigerant for high ambient conditions as in Eq. (6.6).

$$k_{CO_2} = k_{R-123} \left(\frac{\rho_{CO_2}}{\rho_{R123}}\right)^{0.8} \left(\frac{\mu_{R123}}{\mu_{CO_2}}\right)^{0.4} \left(\frac{Cp_{CO_2}}{Cp_{R123}}\right)^{0.4} \left(\frac{k_{CO_2}}{k_{R123}}\right)^{0.6} \tag{4.39}$$

Internal power $\left(\dot{W}_{in}\right)$ is produced in the expander is viewed as a combination of the suction power, expansion power and discharge power.

$$\dot{W}_{in} = \dot{m}_{in}[(h_{su2} - h_{in}) + v_{in}(p_{in} - p_{ex})] \tag{4.40}$$

The power transferred by the shaft connected to the main unit of scroll expander as an output is considered as a shaft power ($\dot{W}_{sh}$).

$$\dot{W}_{sh} = \dot{W}_{in} - 2 \times \pi \times N_{rot} \times \tau \tag{4.41}$$

Overall isentropic efficiency (η_{meas}) is the ratio of actual power produced to the ideal power produced for the same pressure ratio i.e. difference between gascooler pressure and evaporator pressure.

$$\eta_{meas} = \frac{\dot{W}_{el}}{\dot{m}(h_{su} - h_{ex})} \tag{4.42}$$

The model is developed and simulation is carried out in Matlab platform using Eqs. (4.34–4.42) and the properties of the working fluids are extracted from Refprop. Input parameters used for the analysis are supply pressure, supply temperature, exhaust pressure and shaft speed, as tabulated in Table (4.4).

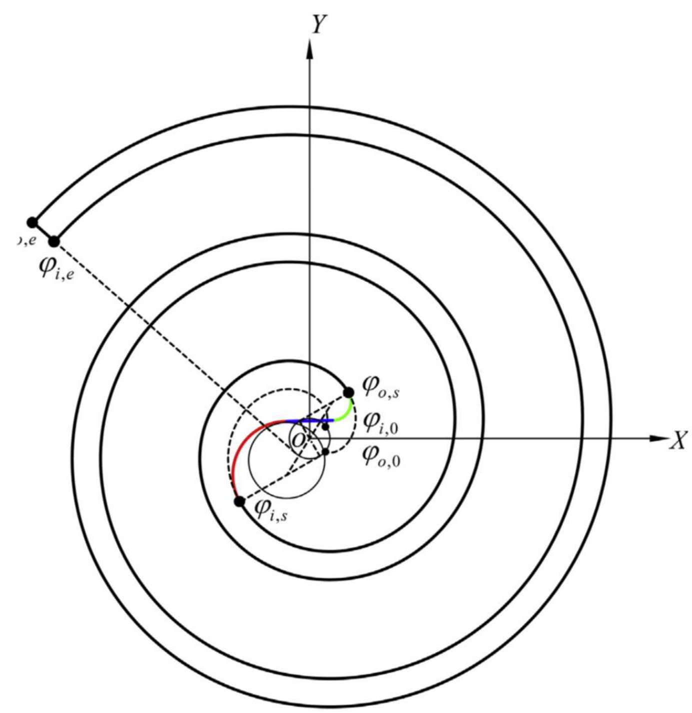


Figure 4.15: Scroll profile with involute angles.

Internal design fixed parameters of the scroll involute used for simulation are taken for a 10kW rating work recovery scroll expander that was originally a scroll compressor of Hitachi make (Model No. 503DH─83D2), having 7.37cm^3 suction volume and 4.2 geometrical volume expansion ratio. The scroll profile with involute angles are projected in Figure (4.15). Internal geometry and design parameters are tabulated in Table (4.5). The output of simulation model is shaft power generated and isentropic efficiency of the scroll work recovery expander.

Table 4.4: Input variables for a scroll expander

Parameter	T_{su} (°C)	P_{su} (MPa)	N (rpm)
Value/Range	32─48	8─12	1500─2500

Table 4.5: Internal design parameters of a scroll expander.

Variable	r_b (mm)	h (mm)	$\Phi_{o,0}$ (rad)	$\Phi_{i,0}$ (rad)	$\Phi_{o,S}$ (rad)	$\Phi_{i,S}$ (rad)	Φ_e (rad)	r_o (mm)
Value	3.2	3.3e+2	-6.9e-1	6.9e-1	2.6	5.7	1.70e+1	5.5

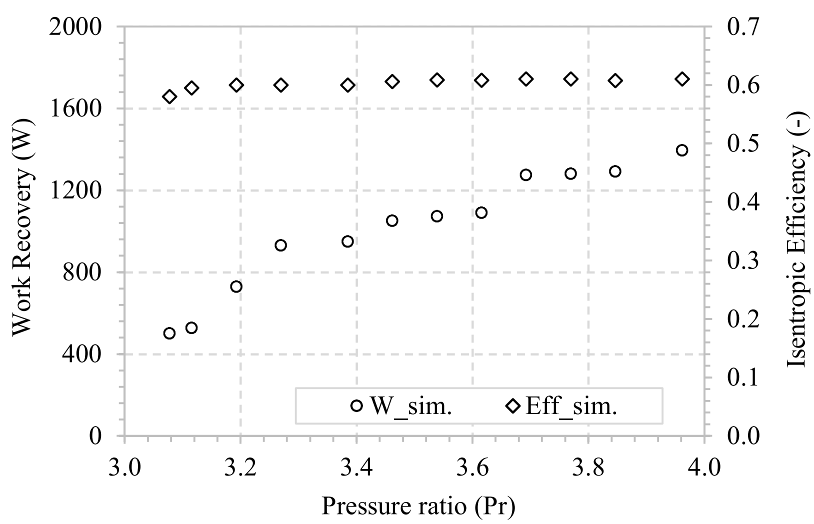


Figure 4.16: Variation in isentropic efficiency and shaft work with pressure ratio.

Figure (4.16) shows the overall isentropic efficiency of the scroll work recovery expander at various pressure ratios. It was observed that the efficiency is lower for low pressure ratios, increases as pressure ratio increases and then decreases beyond a pressure ratio of ~4. The observed behavior can be ascribed to over-expansion losses at low pressure ratio and under-expansion losses at high pressure ratio.

At higher pressure ratio, other factors like leakage loss, heat transfer loss, friction loss etc. also may increase and they may combine to contribute towards the sharper decrease in efficiency. In the present analysis, the effect of these factors are not incorporated. It is observed that the efficiency of the scroll expander varies between 58% to 61% and maximum 55% to 61% is obtained between 3.2 to 3.7 pressure ratio.

To observe the changing behavior of isentropic efficiency and shaft work with pressure ratio, regression analysis is carried out and it is found to have a R^2 value of 0.96, for a second order polynomial function as presented in Eqs. (4.43) & (4.44). The motive behind this regression analysis is to apply these equations further, to predict the isentropic efficiency and shaft work for a known pressure ratio.

$$\eta_{is} = -0.0439(pr)^2 + 0.328(pr) + 0.1138 \tag{4.43}$$

$$W_{sh} = -533.04(pr)^2 + 4722.2(pr) - 8746.7 \tag{4.44}$$

4.3.1.2. Trans-critical CO_2 refrigeration system using scroll expander

Figure (4.17) shows a schematic of work recovery scroll expander, incorporated into a conventional trans-critical CO_2 refrigeration cycle. The work recovered is used to partially drive the compressor with a suitable technology. The shaft work generated thus contributes towards decrease in the system's overall power consumption.

Likewise, an expansion valve is set to carry out the expansion process, when the ambient temperature is below 31ºC or under sub-critical in operation. Steady flow energy equation and mass balance equation are used to conduct a simulation based study for comparative analysis of trans-critical CO_2 refrigeration system at high ambient temperature.

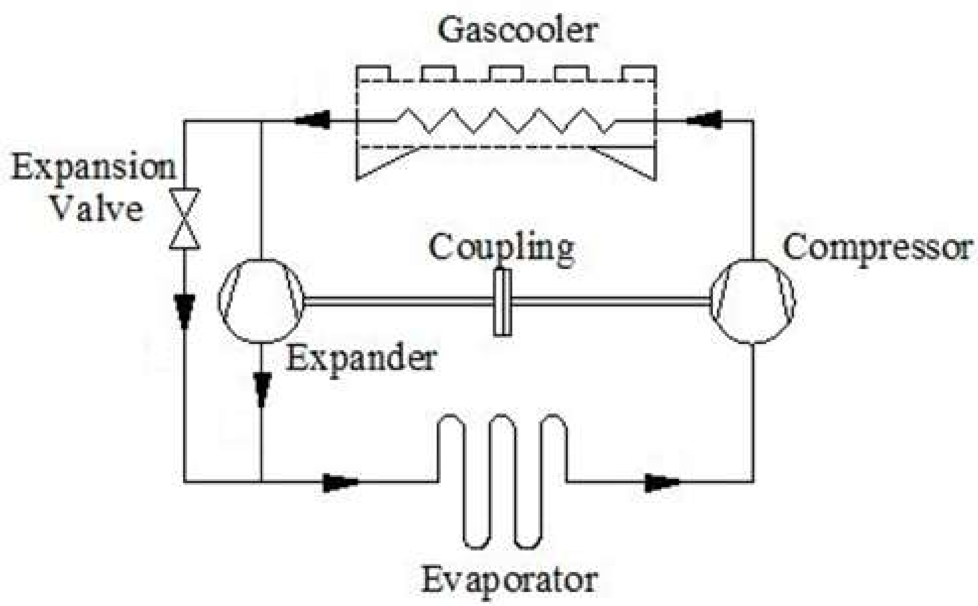


Figure 4.17: Schematic of trans-critical CO_2 refrigeration system.

Table (4.6) summarizes the range of fixed and variable parameters used for simulation based study of trans-critical CO_2 refrigeration system.

In order to simplify the computational model, following assumptions are made.

- Heat loss is negligible from piping, compressor and expander.
- Evaporation and gas cooling processes are isobaric.
- Ambient temperature is equal to gascooler outlet temperature.

Table 4.6: Parameters for simulation.

Parameter	Value/Range
Evaporator temperature	-10°C
Gascooler outlet temperature	32°C to 48°C
Compressor outlet pressure	8MPa to 12MPa
Approach temperature	0°C
Pressure drop in gascooler	0MPa
Superheating at compressor outlet	0°C

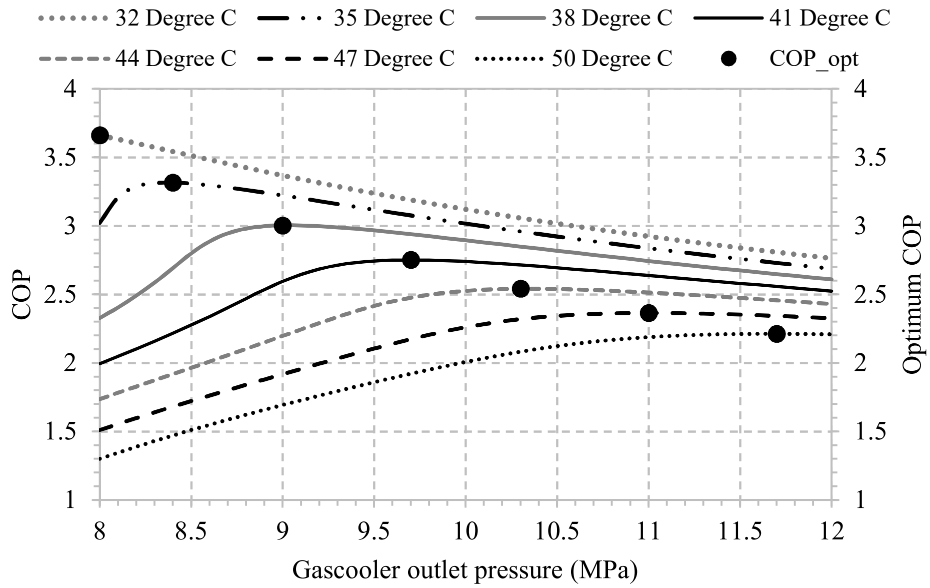


Figure 4.18: Variation of system COP with gascooler outlet pressure at various gascooler outlet temperature.

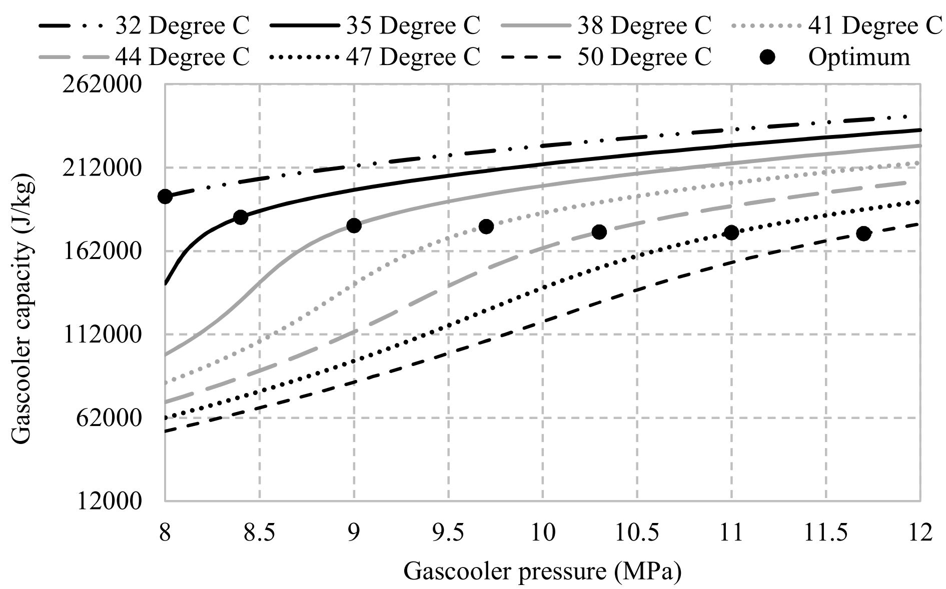


Figure 4.19: Variation of gascooler capacity with gascooler outlet pressure at various gascooler outlet temperature.

Figure (4.18) shows the variation of COP of trans-critical CO_2 refrigeration system at various gascooler outlet pressure and ambient temperature. Optimum COP at various gascooler outet temperature is also projected. It can be observed that as the ambient temperature increases the systems COP reduces. This ascribed to the fact that as the gascooler outlet pressure increase the required compression increases. The optimum COP is identified to maximize the system performance at various ambient temperature above critical point. The S-shape isotherms restricted the system performance after a certain range and resulting in COP reduces constantly beyond that.

Figure (4.19) shows the variation of gascooler capacity with gascooler outlet pressure at various gascooler outlet temperature. The optimum conditions for various ambient temperature is also identified (maximum COP of the system). It is observed that, as the pressure increase the gascooler capacity increases at various ambient temperature.

4.3.1.3. Energy consumption & work recovery

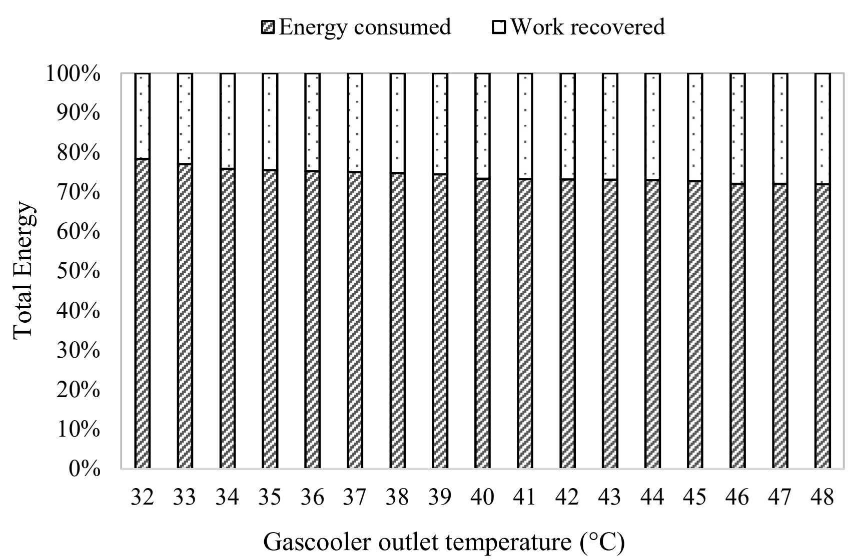


Figure 4.20: Work recovered in scroll as percentage of energy consumed in compressor.

Figure (4.20) shows the energy recovered by a work recovery scroll expander as percentage of energy consumed by the compressor work for trans-critical operation. As the gascooler outlet temperature increases, the percentage work recovery by the expander also increases and about 20% to 30% of the overall energy consumption by the compressor is recovered.

4.3.1.4. Economic analysis

The economic implication of introduction of a scroll work recovery expander for trans-critical CO_2 refrigeration system is attempted to be measured in terms of payback period (PBP). To compute PBP for the CO_2 based refrigeration system for a city with high ambient temperature, data for New Delhi is considered as a case study. The number of hours at a particular ambient temperature round the year is computed for the target city (New Delhi).

In Figure (4.21), the shaded area (beyond 31°C) represents the total number of hours in which the work recovery expander (trans-critical operation) will be effective for a year-round operation of the system. The total number of working hours of scroll expander is found to be 2434 (~27.8% of total time in a year) and 0.225$ (@ 1 INR=0.015$ conversion rate) is assumed as local domestic electricity tariff, including taxes. The PBP of the scroll work recovery expander is computed using Eq. (2.4). The necessary supporting data for computing the PBP of scroll work recovery expander of various capacities are tabulated in Table (4.7).

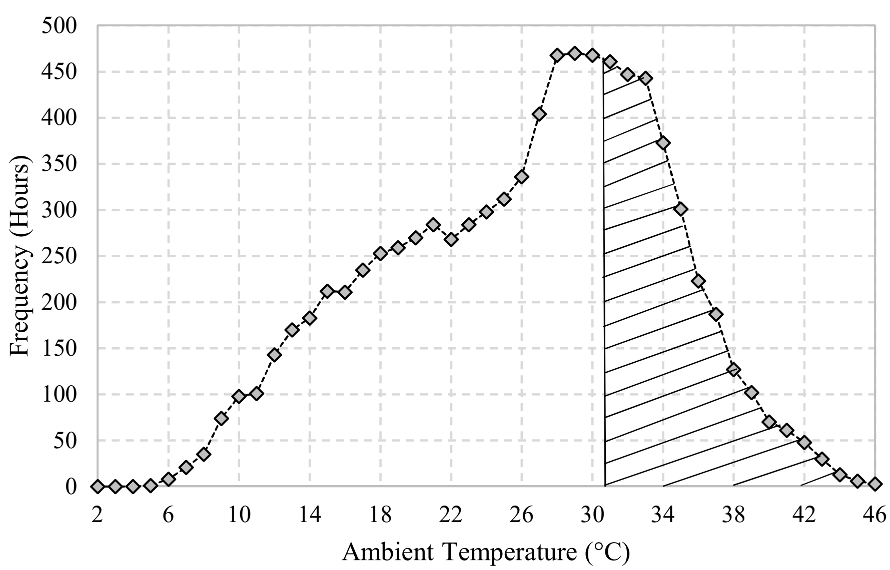


Figure 4.21: Number of hours in a year (2015) for a particular ambient temperature.

Four high capacity scroll compressors are selected to assess PBP of work recovery expander for trans-critical CO_2 refrigeration systems for operation in ambient conditions of sample city (New Delhi). Assumption is made based on single point actual data that 25% additional cost for re- working to covert compressor to expander.

Table 4.7: Various capacities of scroll expander.

Compressor rating	Denfoss	Base price of scroll compressor	VAT @14%	Reworking @25%	Net expander cost
(kW)	Model	($)	($)	($)	($)
10	SM084	600.91	84.14	150.25	835.38
15	SM120	661.09	91.89	165.27	918.92
20	SM147	773.78	108.33	1891.25	1075.55
25	SM185	976.61	136.73	244.15	1357.49
33	SY240	1945.72	272.40	486.43	2704.55

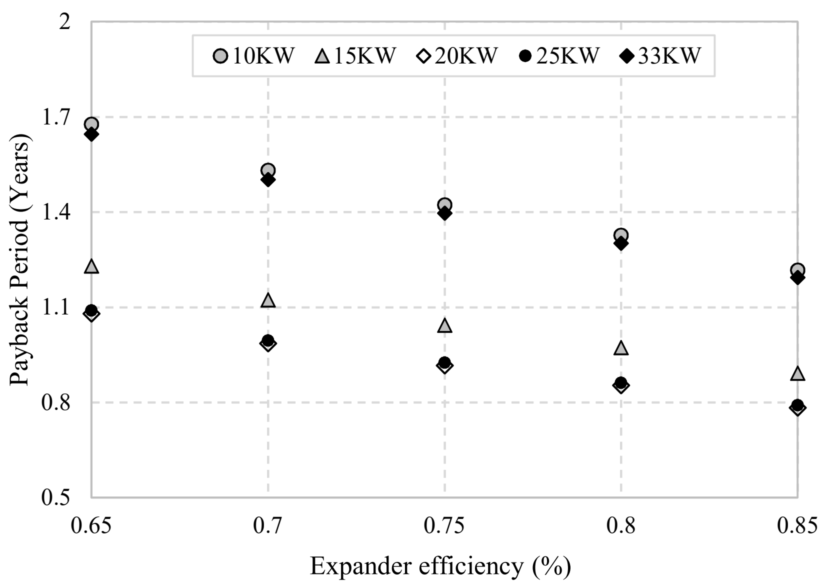


Figure 4.22: PBP at various expander efficiency and cooling capacity.

Figure (4.22) shows the total PBP of scroll expander at various expander efficiency and cooling capacity. It is observed that the PBP of the medium cooling capacity is the lowest. Overall PBP range of 1 to 1.7 years appears reasonable for investment.

Figure (4.23) shows the TEWI for three systems namely CO_2 refrigeration system, CO_2 refrigeration system with expander and R1234yf refrigeration system. It was observed that, adopting work recovery expander in the trans-critical CO_2 system not only reduces the energy consumption by the system, but also reduces the emission of greenhouse gasses directly and indirectly. It is observed that the system with work recovery expander potentially reduce ~40% and ~10% of the total greenhouse gas emission compared to conventional trans-critical CO_2 refrigeration system and R1234yf system respectively during trans-critical operation.

Aggregate direct and indirect TEWI of the three analysed systems over 10 years life span for the target city however, throws a very different picture, Figure (4.24). Over the projected life span, it was observed that, CO_2 system with or without work recovery expander has substantially high TEWI for 10 years long operation compared to R1234yf system.

However, the work recovery expander potentially saves indirect TEWI to the extent of ~40% for trans-critical operation and the direct contribution to TEWI is also about 10% less when compared to that of a low ODP fluid R1234yf. TEWI contribution is higher for CO_2 cycle and for about 70% of the time span during the year CO_2 cycle runs as sub-critical where there is limited scope of work recovery.

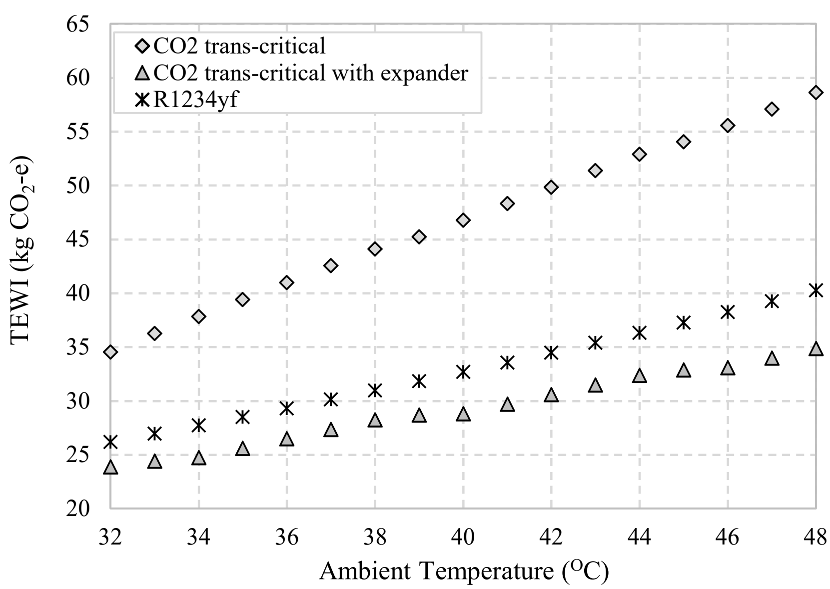


Figure 4.23: Total TEWI at high ambient temperature.

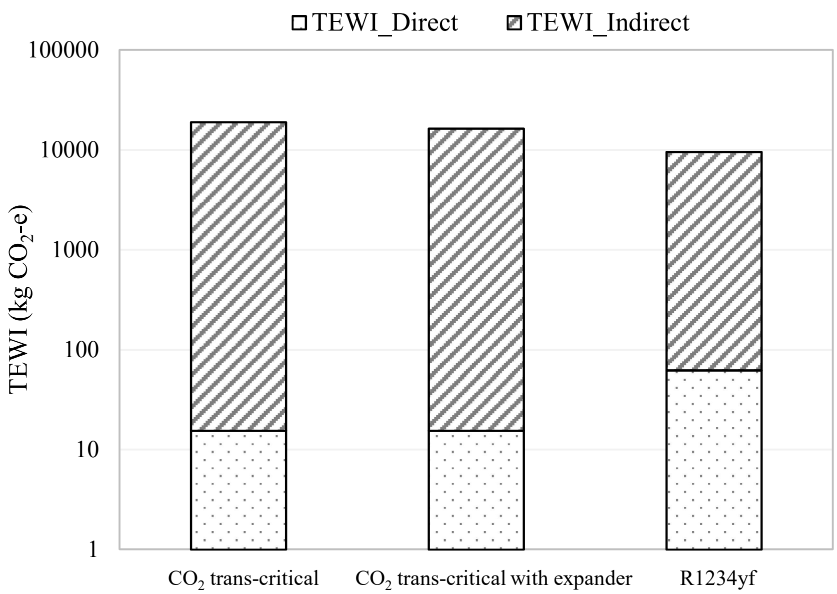


Figure 4.24: Aggregate TEWI of the refrigeration systems.

CHAPTER 5

Indian Dairy Industry – A Case Study

A detailed analysis is carried out on a northern India based mid-sized milk & milk product handling plant, analyzing the state of the art in heating and cooling demand and cold storage system. The applicability of a trans-critical CO_2 booster system is explored for the plant. Two schemes, one of total replacement of existing ammonia system with CO_2 booster system and other of CO_2 heat pump system for waste heat utilization are analyzed. The chapter also evaluates economic analysis of the proposals. One year long field data from the plant and are taken into consideration for the analytical study.

5.1. Existing Plant Description

The milk plant, selected in northern India established under Verka banner, in 1963 and is located at Amritsar (Punjab). The currently operating boiler unit in the plant was established in 2008 and the ammonia based refrigeration system in 2011. The ammonia based refrigeration plant and the coal fired boiler are installed to fulfill the continuous fluctuating cooling and heating demand respectively.

The refrigeration system is equipped with two screw compressors (main & standby) each having refrigeration capacity 140 TR. The plant handles one medium temperature and one low temperature cooling chambers, maintained at 4ºC and -10ºC respectively. The main components of the refrigeration unit are; an economizer, a receiver, a low pressure (LP) and a high pressure (HP) compressor. Heat from the system is rejected through evaporative cooling arrangement. Schematic diagram of the ammonia based refrigeration system is shown in Figure (5.1) and complete refrigeration plant layout is shown in appendix (BF1) along with a few representative photographs of the plant in appendix (BF2−BF5).

Pressure sensors are fitted (at inlet and exit position) at low and medium temperature evaporators, at economizer and at condenser. Temperature sensors are installed at the inlet and exit of all the main components in the system. Data scan is performed for every parameter at one-hour interval. Outdoor (ambient) temperature data for the study is taken from weather station online repository in absence of reliable measurement within plant. The daily averaged temperature data thus obtained various from 0°C to 40°C during the study period. Milk handling data is available on a per day basis. Milk handling by the plant is intermitted and the major load is during early morning hours. Milk & milk products like butter, ice-cream, curd, condensed milk, flavored milk, cheese, etc. are introduced into the plant between 9 AM to 11 AM. Refrigerated products are taken out between 4 PM to 6 PM. Generally, one day’s stock is maintained within the cooling chambers. Thus, the over-night heat load on the plant is comparatively small. Due to supply as well as demand fluctuation, the milk handling load is also not constant through-out the year, typically in winter days it increases upto 1,50,000 liters day^{-1} and decreases to 70,000 liters day^{-1} during summer days. A sample data sheet of the refrigeration system for a particular day, is tabulated in appendix (AT6).

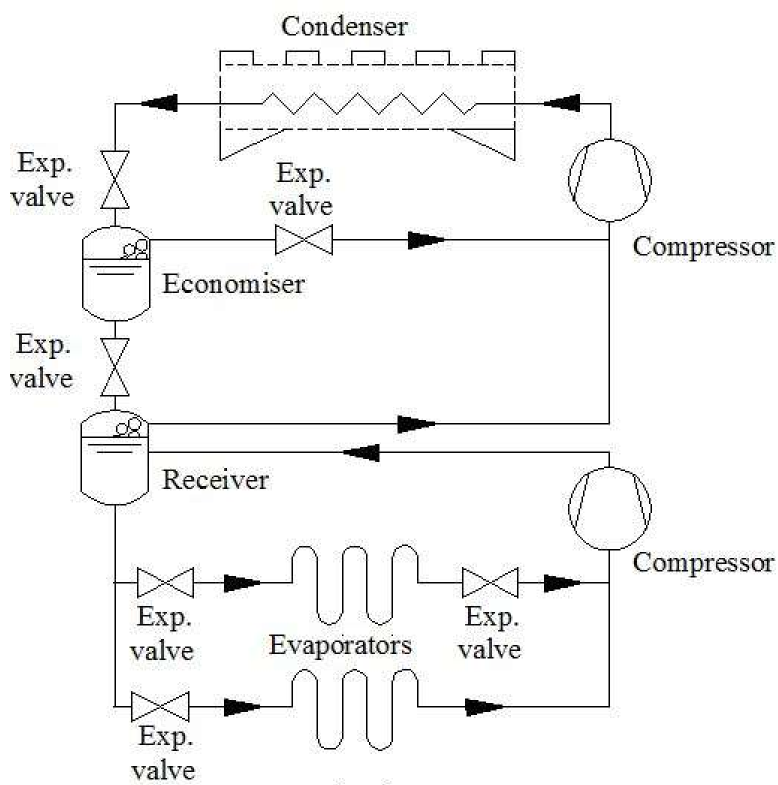
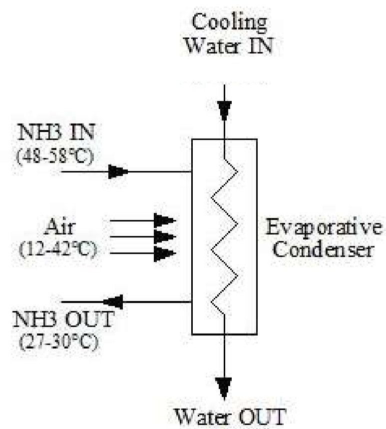
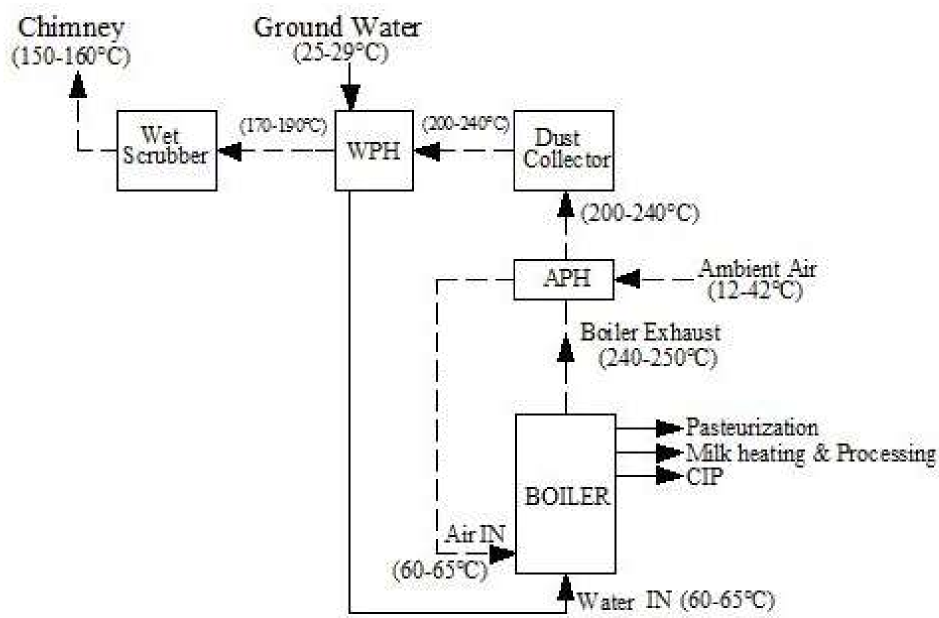


Figure 5.1: Ammonia refrigeration system.

Heat from the ammonia based refrigeration system is rejected using evaporative cooling arrangement as shown in Figure (5.2). Forced air draught fans and continuous water circulation are arranged for efficient heat rejection. Every seven days it is required to replenish 5000 to 6000 liters of water in a tank with ground water to compensate consumption during evaporative cooling process.



Evaporative condenser of an ammonia based refrigeration system



Boiler

Figure 5.2: Schematic of existing heating and cooling system.

Ammonia based vapor compression refrigeration systems are by far the most preferred mode for cooling in milk processing plants. A variety of products are handled by these plants like butter, ice cream, curd, condensed milk, butter milk, flavored milk, cheese, etc. There is also a large seasonal variation in milk supply quantity as well as its demand.

Operations in dairy plants demand thermal energy, electrical energy and copious amounts of water. Typically, in India, low grade coal is used to run the boilers to fulfill the heating demand. These boilers have historically proven to be effective sources of energy in dairy industries. Significant proportion of thermal energy demand (about 75%) for heating and steam generation purpose in dairy plants are met by the coal fired boilers. The rest (about 25%) is met through thermo-packs using HSD and heaters using electricity. Electricity is also used for running the refrigeration plant, for lighting and automation.

Elsewhere in the plant, the heating demand for milk processing is met by a coal fired tube-in-tube boiler of 3 Ton capacity. Boiler is employed to generate high pressure steam (7.5 kg cm^{-2}) for pasteurization, milk heating & processing and CIP. In order to increase the thermo-economic efficiency of the boiler, the same is equipped with an air pre-heater (APH), a water pre-heater (WPH), a dust collector and a wet scrubber as shown in Figure (5.2). After passing through the dust collector, the WPH and wet scrubber, the cleaned & low heat content air is exhausted through a chimney.

5.1.1. Heating and cooling demand in the plant

The boiler meets about 183116 MJ of heating demand per day in the plant. Where about 36 TR of cooling demand per day is met by electric power supply. Heating demand is in milk pasteurization, milk condensing, milk drying, clarified butter (ghee) processing and CIP. Cooling demand is in milk chilling, milk pasteurization, product processing, prepack milk pouch filling, milk cold storage and powder milk production as shown in Figure (5.3).

Pasteurization is a process, which require both cooling and heating, where milk temperature is raised to minimum 71.5°C in 14 seconds, held for about 15 seconds and thereafter, chilled to 4°C in 12 seconds. The purpose of pasteurization of milk is to destroy pathogens present and thereby increase shelf life.

5.1.2. Energy consumption

Methodology adopted for measuring current operational energy consumption during various processes is shown in Figure (5.4) and instrumentation used for the same is tabulated in Table (5.1). Due permission and support from officials/supervisors is taken to collect information about the various processes, milk load distribution within the plant and operational data. The data was analyzed to arrive at a base line energy consumption pattern and to identify the energy usage pattern and losses in the system. Measurements and monitoring is carried out with appropriate instruments including continuous and time lapsed recording.

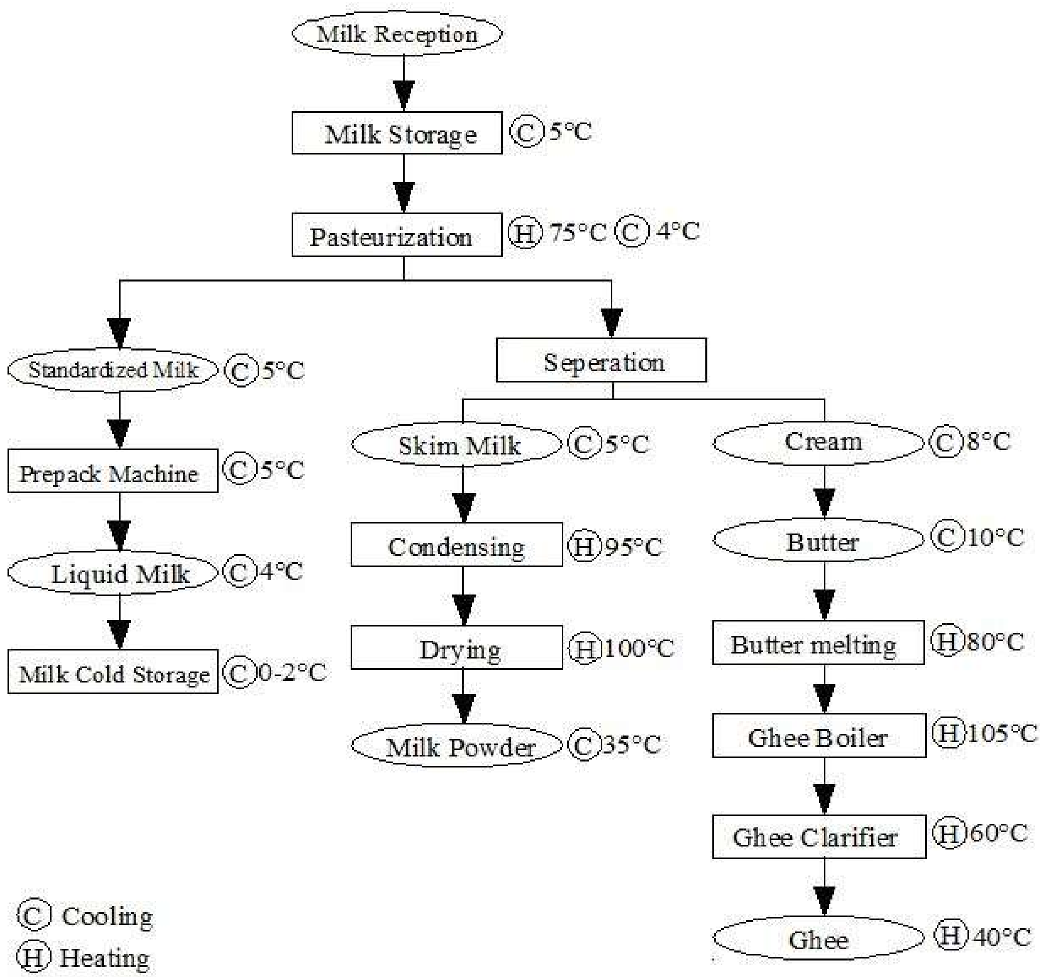


Figure 5.3: Various operations carried out in milk processing.

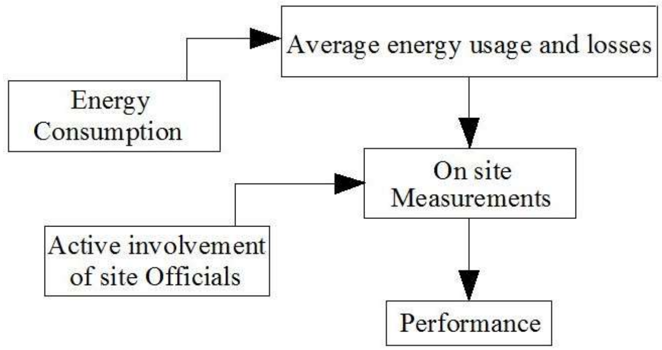


Figure 5.4: Energy consumption measurement.

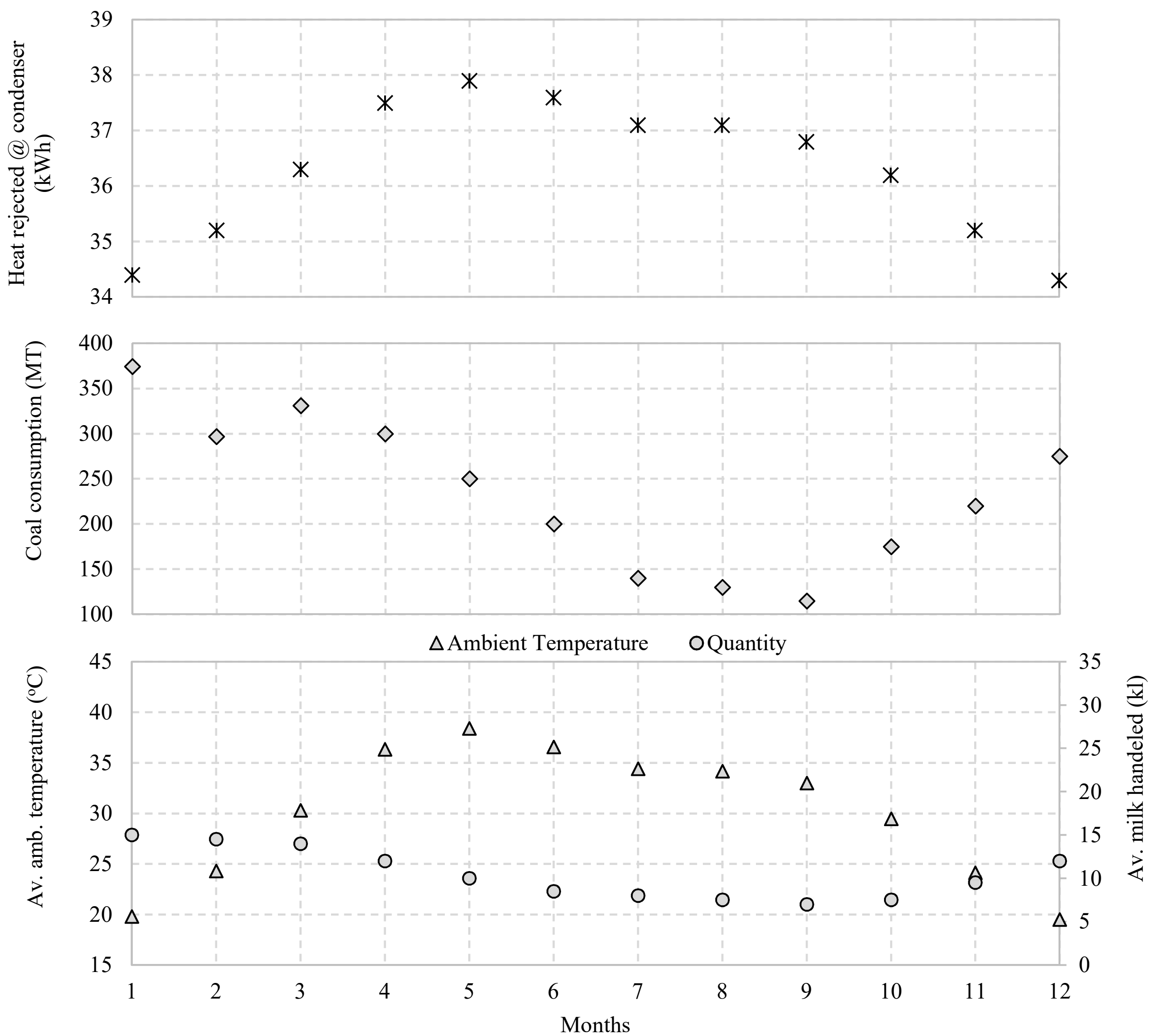


Figure 5.5: Heat available, coal consumed, average ambient temperature and milk handling.

Table 5.1: Instruments used for measurements.

Instrument	Parameter	Measuring Range	Accuracy
Combustion analyzer	Oxygen	0 to 21%	± 0.2%
	Carbon monoxide	0 to 4000 ppm	± 20 ppm (< 400 ppm)
			± 5% (> 400 ppm)
Digital manifold	Temperature	0 to 600°C	± 0.3%
Anemometer	Air velocity	0.5 to 89 miles hr^{-1}	± 3%
Digital pressure meter	Pressure	0 to 350 mbar	± 0.2%
Thermocouple	Temperature	-50°C to 450°C	0.5°C

Table 5.2: Boiler data.

Variable	Unit	Value/Range
Fuel		Coal
Flue gas temperature before APH	°C	294
Flue gas temperature after APH	°C	208
Ambient temperature (Daily averaged)	°C	15 to 48
Calorific value of fuel	kcal kg^{-1}	3500
Average oxygen percentage in flue gases	%	9.7
Operating steam pressure	kg cm^{-2}	7.5
Sensible heat loss		
Excess air	%	86
Theoretical air required to burn 1 kg of fuel	kg kg^{-1} of fuel	4.86
Total air supplied	kg kg^{-1} of fuel	9.04
Weight of flue gas	kcal kg^{-1} °C	10.04
Specific heat of flue gases	kcal kg^{-1} of fuel	603
Heat loss in flue gases	%	17.2
Heat loss due to evaporation of moisture in fuel		
Moisture in fuel	kg kg^{-1} of fuel	0.31
Specific heat	kcal kg^{-1} of fuel	0.45
Heat loss due to moisture in fuel	%	6.2
Heat loss due to evaporation of water formed due to hydrogen in fuel		
Hydrogen in fuel	kg kg^{-1} of fuel	0.02
Specific heat	kcal kg^{-1} of fuel	0.45
Heat loss due to hydrogen in fuel	%	3.6
Other losses (radiation losses, blow down, heat losses due to unburnt carbon in ash etc.)	%	3
Total losses	%	30
Boiler efficiency	%	70

Figure (5.5) shows monthly averaged variation of maximum heat rejected from condenser of the ammonia based refrigeration system, coal consumption variation in the boiler and the variation in average milk handling, months 1 to 12 represents January to December. The heating demand at various processes within the milk plant is handling by steam from boiler which uses ground water as boilers feed available in temperature range 25°C to 29°C round the year. The boiler related information required for boiler efficiency calculation is tabulated in Table (5.2). The daily average flow rate of water for boiler facility is 3.5 to 4 m^3 h^{-1}. The boiler tank water holding capacity is 11 m^3. Approximately, 4.5 liters of water is supplied for per kg of coal consumption.

5.2. Trans-critical CO_2 Booster Refrigeration System

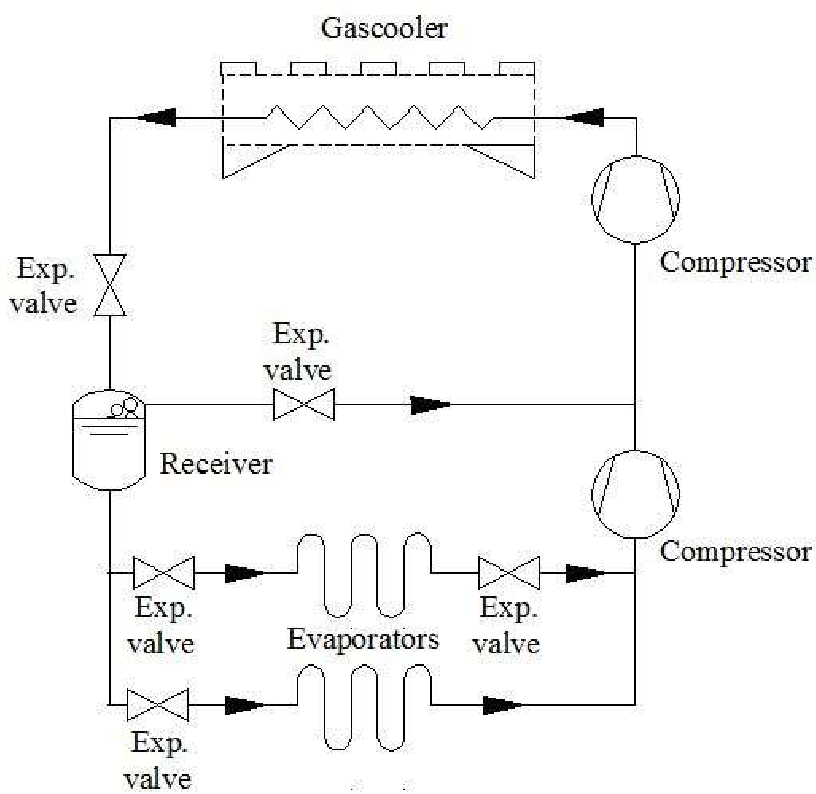


Figure 5.6: Trans-critical CO_2 booster refrigeration system.

Figure (5.6) shows schematic for a CO_2 booster system or the proposed replacement plant. In order to construct thermodynamic model, four regions in the overall cycle are identified. The high side pressure region is defined as one extending from high stage compressor to the gascooler or condenser depending upon the ambient condition.

Intermediate pressure region begins at outlet of the receiver of the expansion valve to the medium and low temperature evaporator coil. While medium pressure region begins at the outlet of the medium temperature expansion valve and extends through the evaporator coils to the medium temperature evaporator. Low pressure region starts at the outlet of the low temperature expansion valve and extends through the evaporator coil upto the low-pressure compressor.

The various fixed parameters used as input for thermodynamic modeling of the proposed trans-critical CO_2 booster refrigeration system is tabulated in Table (5.3). Primary parameters used for the evaluation of the ammonia based refrigeration system are cooling capacity and COP, calculated at one-hour interval during the operative hours, and then averaged to monthly values. COP of the system is calculated based on energy meter reading and is taken as the ratio of cooling capacity to the total electrical power consumption. Cooling capacity is computed by estimating mass flow rate of refrigerant and refrigerant enthalpy at the inlet and outlet points of the evaporators. Indirect method is adopted to measure mass flow rate, using performance values from design data supplied by manufacturer and energy balance around the compressor.

Table 5.3: Parameters of trans-critical CO_2 booster refrigeration system.

Variable	Value
Medium temperature cooling chamber	4ºC
Low temperature cooling chamber	-10ºC
Superheating	5ºC

The design data for compressor is used to calculate the efficiency of the compressor at various operating conditions and is computed by using Eq. (5.1):

$$\eta_c = \frac{\dot{m}_r(h_{comp,out,is} - h_{comp,in})}{\dot{E}_{el}} \tag{5.1}$$

where, the mass flow rate of refrigerant is calculated by using Eq. (5.2):

$$\dot{m}_r = \frac{\dot{Q}_{total,ev}}{\Delta h_{ev}} \tag{5.2}$$

COP of refrigeration system is the ratio of cooling capacity to electrical power consumption and is calculated by using Eq. (5.3):

$$COP = \frac{\dot{Q}_{total}}{\dot{E}_{el}} \tag{5.3}$$

Total heat load calculation for the system was performed, as suggested [13] . The heat addition through the walls, floor and ceiling of the milk cold storage room are considered under transmission heat load and is computed by using Eq. (5.4):

$$\dot{Q}_{th} = KA(T_{out} - T_{in}) \tag{5.4}$$

Entry of air into the milk plant during manual loading & unloading is considered under infiltration heat and is calculated by using Eq. (5.5):

$$\dot{Q}_{ih} = CzV(T_{out} - T_{in}) \tag{5.5}$$

The major part of heat load is due to the product heat and is calculated by using Eq. (5.6):

$$\dot{Q}_{ph} = \frac{GC_{ph}(T_{out}-T_{in})}{\Delta t_s/3600} * LF \tag{5.6}$$

Further, heat added from unknown and unforeseen sources are lumped together and computed by using Eq. (5.7):

$$\dot{Q}_{uh} = \left(N_p * C_p\right) + (N_l) \tag{5.7}$$

At first, study was performed to determine the effect of various operating parameters on the energy efficiency of the system. The overall COP of the refrigeration system is computed as a ratio of refrigeration effect and the total work done by the compressors and pumps, as calculated using Eq. (5.8):

$$COP = \frac{\dot{Q}}{\dot{W}_{total}} \tag{5.8}$$

The following assumptions are made for the analysis.

- Pressure drop and heat gain/loss are neglected in pipes.
- Only saturated vapor and saturated liquid exits the receiver.
- Fan power consumption in condenser or gascooler are negligible.

Performance evaluation of the system is carried out for the full study period of two years using field data of ambient temperature but representative plot for one year is shown removing repetition.

5.2.1. System performance comparison

Variation of COP of the ammonia based plant for a typical day in summer and winter month is plotted from field measurement data as well as from the constructed thermodynamic model using daily average temperature and total heat load handled as input and is shown in Figure (5.7). It is evident that apart from ambient temperature, the COP is influenced by other factors; and it is found that thermodynamic modelling is more or less able to follow the field data.

Figure 5.7: Hourly variation of COP on a typical day of summer and winter.

Figure (5.8) represents monthly variation of COP of ammonia plant with variation of ambient temperature and volume of milk handled for a period of one year, months 1 to 12 in the plot represents January to December. An average monthly data sorted for a particular peak hour load time for a day, is tabulated in appendix (AT7). Each point in the graph represents the monthly average value of the corresponding parameters. We see that, as the volume of milk handled increases, the deviation of simulation data from field data increases. Excess milk handling may be associated with repeated and prolonged opening of door etc. resulting in higher infiltration.

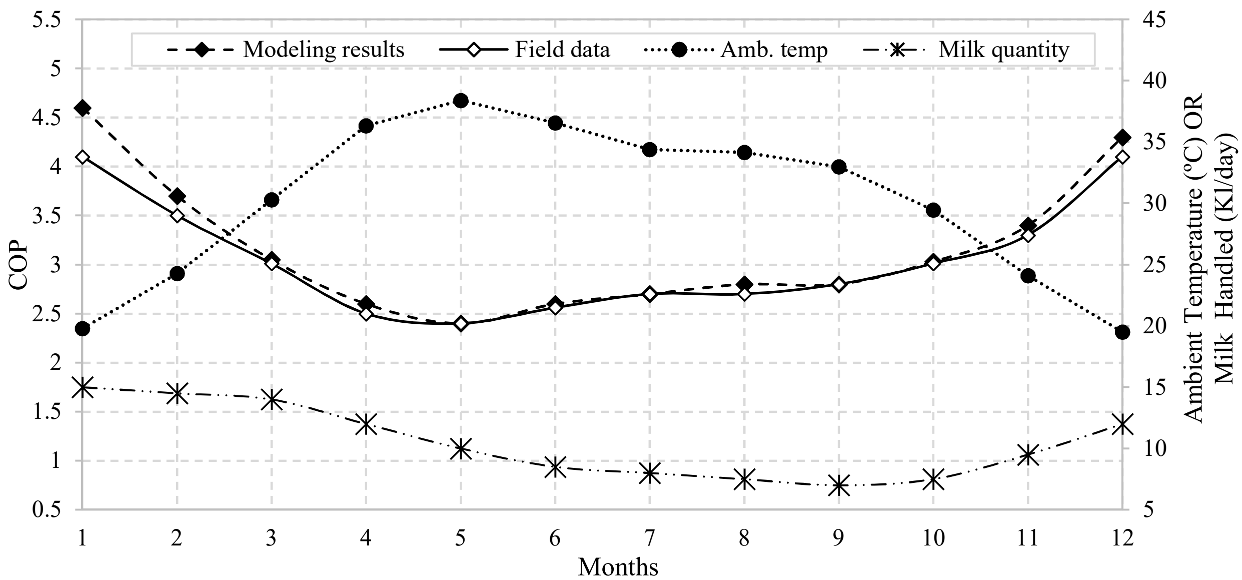


Figure 5.8: Yearly variation of COP from modeling & field results for an ammonia system.

A correlation factor (CF) is computed using extrapolation function in excel between the difference of simulation data from field data to the quantity of milk handled, by using regression analysis and it is found to have R^2 value of 0.91, for a fifth order polynomial function. The same CF is used later to correct simulation based prediction of COP from CO_2 booster refrigeration system against effect of variation of quantity of milk handled.

Figure (5.9) represents comparison between COP of ammonia based refrigeration system and proposed trans-critical CO_2 booster refrigeration system with variation of ambient temperature and with and without CF applied. It is observed that the COP of the proposed CO_2 system is lower round the year. It is also observed that as the ambient temperature increases beyond critical temperature of CO_2 (31.1ºC), COP falls drastically. The same is ascribed to the requirement of high pressure operation and high heat rejection by the gascooler of the system for trans-critical operation.

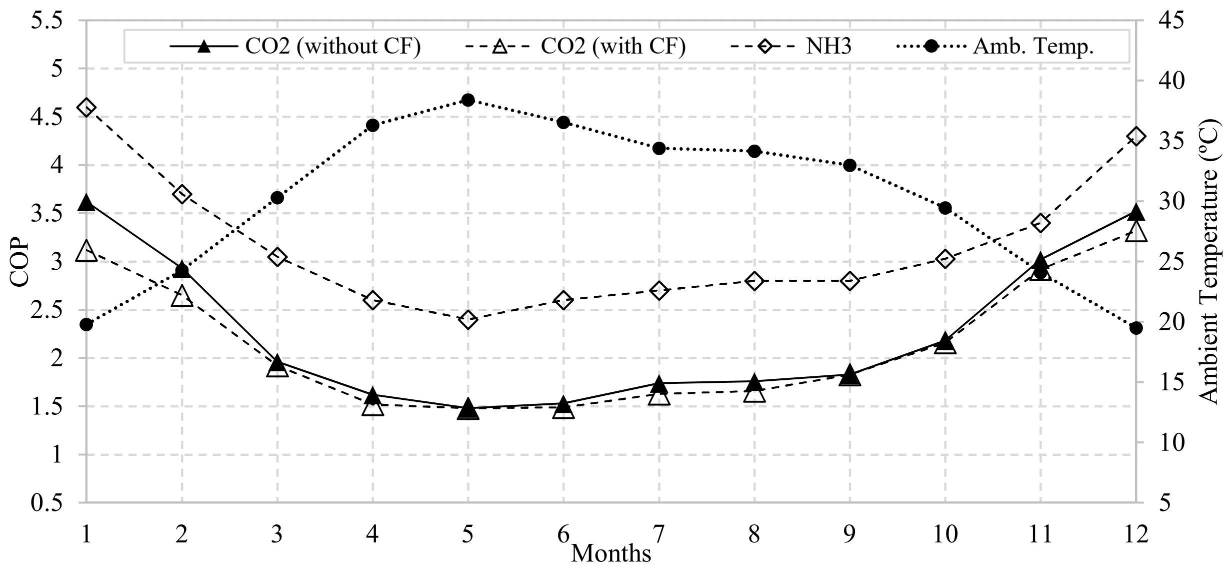


Figure 5.9: Comparison of COP for an ammonia based refrigeration system and trans-critical CO_2 booster refrigeration system with ambient temperature.

A comparative study of cycle operations at various ambient temperature for both ammonia and CO_2 system are presented in Figure (5.10). It is observed that comparatively higher discharge pressure is required for trans-critical CO_2 booster refrigeration system to generate effective heat rejection through the gascooler at various ambient temperature. Comparing ammonia based refrigeration system and CO_2 booster refrigeration system at various ambient temperatures, it is observed that the required amount of heat rejection for CO_2 system is much higher for all ambient temperatures, at each step (20ºC, 25ºC, 30 ºC, 35ºC & 40ºC). This partially explains lower COP of CO_2 system round the year. Further, thermo-economic analysis is carried out for both the systems in order to have a clear understanding of lower COP of CO_2 system.

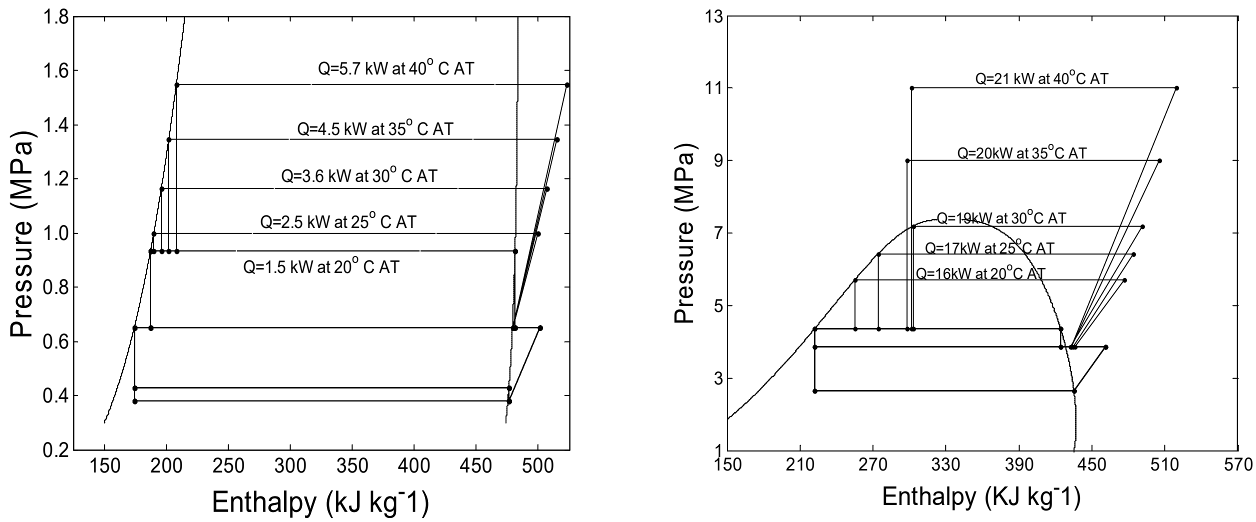


Figure 5.10: Heat rejection in condenser/gascooler at various ambient temperatures.

5.2.2. Thermo-economic analysis.

For thermo-economic analysis of both the systems, exergy destruction for the major components of the systems are determined at two reference temperatures. The two reference ambient temperatures (20°C & 40°C) are selected in order to cover sub and trans-critical operation of CO_2 booster refrigeration system. A study proposed [86], presented a thermo-economic analysis of trans-critical CO_2 booster system for maintaining a medium and a low temperature cooling chambers. The study was to establish a relationship between cooling costs at each individual cooling chambers. Exergy destruction for the ammonia based refrigeration system and the proposed trans-critical CO_2 refrigeration system are evaluated using Eq. reported in Section (4.1.1). NTP conditions is taken as dead state. The cost per exergy unit is calculated by using Eqs. (5.10) & (5.11).

$$\dot{c}_i = \dot{c}_f + \dot{c}_{O\&m} + \dot{c}_{Cl} \tag{5.10}$$

$$\dot{C}_i = \dot{c}_i \dot{E}_i \tag{5.11}$$

where, current local $\dot{C}_\iota$ is INR 11 per unit (commercial rates) of electrical energy and is equivalent to about \$0.162.

A comparison of exergy destruction of all the main components of the ammonia based refrigeration system and trans-critical CO_2 booster refrigeration system at 20ºC and 40ºC ambient temperature is listed in Table (5.4) & (5.5).

Table 5.4: Thermo-economic variables of NH_3 system for components evaluation.

Components	Sub-critical (20ºC)					Sub-critical (40ºC)				
	$E_{D\&L}$ kW	$E_{D\&L}$ %	$Y_{D\&L}$ %	$C_{D\&L}$ ₹ h^{-1}	$C_{D\&L}$ \$ h^{-1}	$E_{D\&L}$ kW	$E_{D\&L}$ %	$Y_{D\&L}$ %	$C_{D\&L}$ ₹ h^{-1}	$C_{D\&L}$ \$ h^{-1}
LP compressor	0.009	1.52	17.31	0.095	0.001	0.001	0.15	10.00	0.099	0.001
HP compressor	0.005	0.84	9.62	0.055	0.001	0.001	0.15	10.00	0.057	0.001
Condenser	0.405	68.52	78.50	4.455	0.067	0.463	71.34	463.0	5.093	0.076
LT evaporator	0.001	0.16	1.92	0.011	0.000	0.001	0.15	10.00	0.011	0.000
HT evaporator	0.002	0.33	3.85	0.022	0.000	0.002	0.31	20.00	0.022	0.000
Exp. valve 1	0.002	0.33	3.85	0.022	0.000	0.002	0.31	20.00	0.022	0.000
Exp. valve 2	0.001	0.16	1.92	0.011	0.000	0.001	0.15	10.00	0.011	0.000
Exp. valve 3	0.001	0.16	1.92	0.011	0.000	0.001	0.15	10.00	0.011	0.000
Exp. valve 4	0.165	27.91	18.23	1.815	0.003	0.177	27.27	1770	1.947	0.029
Total	0.591			6.501	0.0717	0.649			7.139	0.107

For component evaluation, a few additional thermo-economic parameters such as exergy destruction and loss ratio, cost rate of exergy destruction and loss are also presented. From Table (5.4), it is evident that the exergy destruction and its corresponding cost, increases considerably in condenser and expansion valve of the ammonia based refrigeration system with increase in ambient temperature. This is mainly due to the increase in compressor discharge temperature and pressure ratio between evaporator and condenser.

Table (5.5), summarizes corresponding exergy destruction and associated cost components for the replacement CO_2 system. Comparatively higher exergy destruction is observed in HP compressor, gascooler and expansion valve. It is also observed that the exergy destruction and corresponding cost for trans-critical CO_2 booster refrigeration system are about 3 times more during working in super-critical operation than in sub-critical operation and about 10 times more than the corresponding value for ammonia based refrigeration system. The high exergy destruction and loss are mainly due to the high-pressure lift developed by HP compressor and high heat rejection required by the gascooler. As ambient temperature increases the output from gascooler remains at higher temperature, this increases the amount of vapor returning to HP compressor, leading to increase in energy consumption at compressor.

Table 5.5: Thermo-economic variables of CO_2 system for components evaluation.

Components	Sub-critical (20ºC)					Trans-critical (40ºC)				
	$E_{D\&L}$ kW	$E_{D\&L}$ %	$Y_{D\&L}$ %	$C_{D\&L}$ ₹ h^{-1}	$C_{D\&L}$ \$ h^{-1}	$E_{D\&L}$ kW	$E_{D\&L}$ %	$Y_{D\&L}$ %	$C_{D\&L}$ ₹ h^{-1}	$C_{D\&L}$ \$ h^{-1}
LP compressor	0.21	9.73	7.90	2.33	0.035	0.22	3.60	2.93	2.50	0.038
HP compressor	1.38	63.49	51.53	15.26	0.229	3.37	53.42	43.48	37.08	0.556
Gascooler	0.14	6.44	5.23	1.54	0.023	1.15	18.36	14.95	12.75	0.191
LT evaporator	0.08	3.74	3.03	0.89	0.013	0.08	1.38	1.13	0.96	0.014
HT evaporator	0.16	7.59	6.16	1.82	0.027	0.17	2.81	2.29	1.95	0.029
Exp. valve 1	0.06	3.19	2.59	0.76	0.011	0.07	1.18	0.96	0.82	0.012
Exp. valve 2	0.03	1.38	1.12	0.33	0.005	0.03	0.51	0.42	0.35	0.005
Exp. valve 3	0.02	1.15	0.94	0.27	0.004	0.17	2.73	2.22	1.89	0.028
Exp. valve 4	0.07	3.28	2.66	0.78	0.012	1.00	15.99	13.02	11.10	0.167
Total	2.185			23.98	0.359	6.312			69.4	1.04

Exergy destruction for all major components of the trans-critical CO_2 booster refrigeration system is plotted in Figure (5.11) for a period of 1 year. This shows variation of cost for the proposed replacement plant handling same quantity of milk at corresponding ambient temperature, as the ammonia based plant.

From Figure (5.12), it is observed that there is an abrupt increase of exergy destruction in HP compressor, gascooler and expansion valve occurring during summer months. The observation guides us to examine the challenges associated with high exergy destruction occurring in the system and explore possible alternatives in order to enhance system efficiency.

A few possibilities that can be explored for COP improvement are deployment of more efficient cycle, effective use of work recovery expander in place of expansion valve, employing a heat recovery system for gainful utilization of rejected heat from the gascooler etc. From a realistic stand point assuming simultaneous deployment of a work recovery expander with 65% efficiency and recovery of about 50% heat rejected by gascooler, for gainful use within the plant, further simulation is carried out keeping milk handling quantity as 0.1 million liters per day.

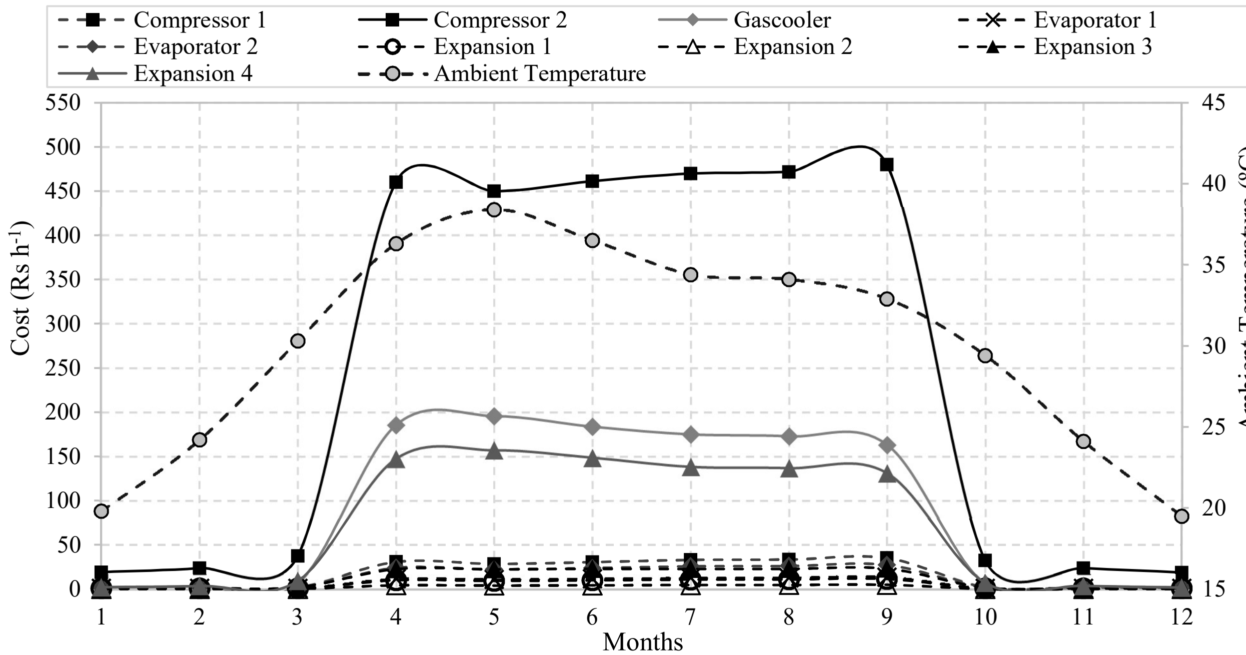


Figure 5.11: Exergy cost distribution.

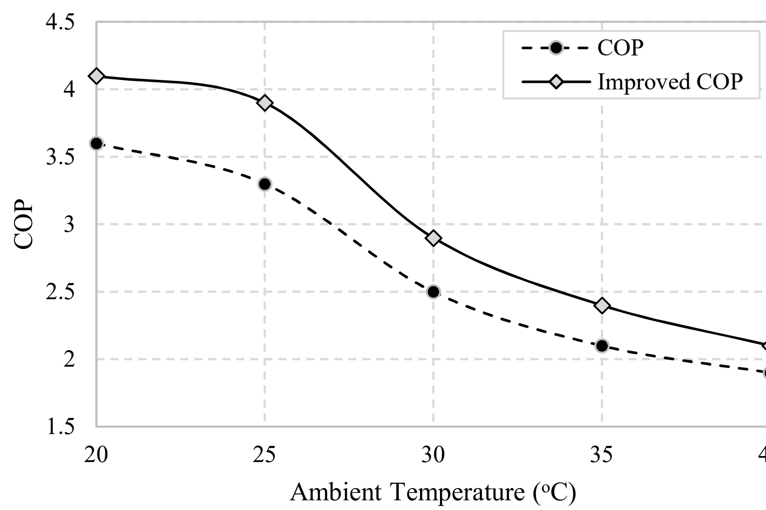


Figure 5.12: Variation in COP of trans-critical CO_2 booster refrigeration system (with and without system improvements) at various ambient temperature.

The result is summarized as in Figure (5.12). From the figure, it is seen that COP improves by about 30% reducing the maximum COP gap with ammonia system at corresponding temperature to about 0.5 only, which is an encouraging result.

5.3. Trans-critical CO_2 Heat Pump System for Waste Heat Utilization

We propose to incorporate a CO_2 trans-critical heat pump system to utilize the rejected heat from the ammonia based refrigeration plant in order to preheat boiler feed water and thereby reduce coal consumption. The boiler feed water is ground water that is available round the year at more or less constant temperature. The condenser of ammonia based refrigeration system is coupled with evaporator of the heat pump system such that the evaporative cooling system is eliminated resulting in saving of some ground water.

The CO_2 system COP is now coupled to ground water that is available at around 27°C. The objective of this study is to evaluate the feasibility of such a plant for a milk refrigeration unit using the actual heating & cooling demand data from plant and its seamless utilization with on-site installations. The resulting change in primary operating cost and energy saving thereof are also computed.

5.3.1. Incorporation of trans-critical CO_2 heat pump system

Figure (5.13) shows the proposed trans-critical CO_2 heat pump system with IHX, coupled with existing ammonia based refrigeration and boiler system. The trans-critical CO_2 heat pump meets simultaneous heating and heat recovery demand. The evaporator of the heat pump system takes up heat from the ammonia cycle and boiler feed water is preheated utilizing the heat rejected by the gascooler.

As per the required cooling and heating demand in the milk refrigeration plant, heat pump system parameters such as discharge pressure, suction temperature, gascooler outlet temperature, etc. are to be optimized for maximum COP.

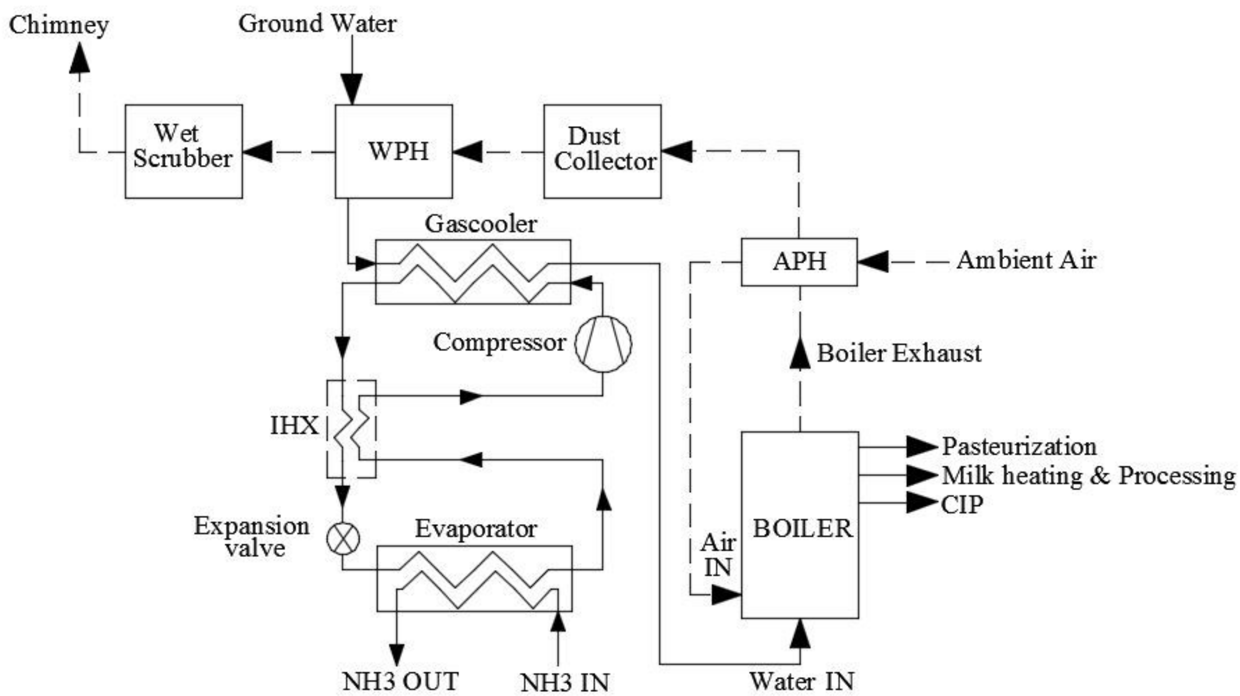


Figure 5.13: Schematic of proposed trans-critical CO_2 heat pump system.

5.3.2. CO_2 heat pump performance evaluation

Steady flow energy and mass balance equations are used to build a simulated model for analysis. The CO_2 heat pump system is simulated to operate at a range of gascooler outlet pressure (8−12 MPa) to identify the optimum value to gascooler pressure for a particular gascooler outlet temperature to compute the heating capacity. The model of CO_2 heat pump is developed and evaluated using Eqs. (5.12−5.14). Range and value of the variables/constants used in the study are tabulated in Table (5.6).

For the simulation, compressor power input is computed using Eq. (5.12).

$$\dot{W}_{c,hp} = \dot{m}(h_{c,out} - h_{c,in}) \tag{5.12}$$

The temperature at the suction of the compressor is computed using effectiveness of IHX, Eq. (5.13).

$$\varepsilon_{IHX} = \left(\frac{T_{c,in} - T_{ev,out}}{T_{gc,out} - T_{ev,out}}\right) \tag{5.13}$$

COP of the CO_2 heat pump system is computed using Eq. (5.14).

$$COP_{hp} = \left(\frac{h_{gc,in} - h_{gc,out}}{h_{c,out} - h_{c,in}}\right) \tag{5.14}$$

Table 5.6: Parameters for trans-critical CO_2 heat pump system.

Parameter	Value/Range
Evaporator temperature	20°C
Gascooler outlet temperature	35°C to 39°C
Gascooler outlet pressure	8 to 12MPa
Ground water temperature	25°C to 29°C
Effectiveness of IHX	70
Approach temperature	10°C

Figure (5.14) shows the variation of COP with gascooler outlet pressure at various gascooler outlet temperatures, for the CO_2 system assuming 10°C approach temperature over and above the available ground water temperature round the year.

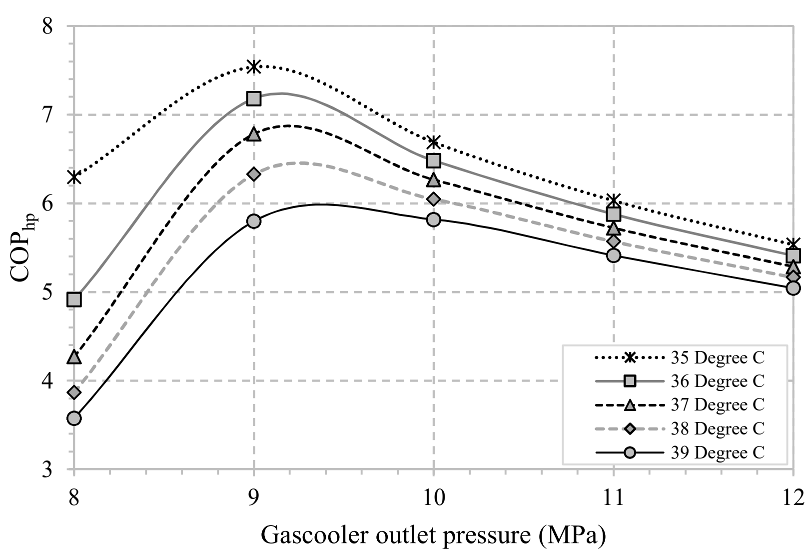


Figure 5.14: Variation of cooling COP and heating COP with gascooler outlet pressure at various gascooler outlet temperature

A steady decrease in COP is observed with increasing gascooler outlet pressure across various gascooler outlet temperatures beyond a maximum at around 9MPa. It is also observed that, with decrease in gascooler outlet temperature, the COP of the CO_2 heat pump system increases, as the pressure difference between gascooler and evaporator decreases.

Figure (5.15) shows the maximum specific heat available for feed water pre-heating at the optimum gascooler outlet pressure 9MPa and total energy consumption of the system at various gascooler outlet temperature. As expected, heat transfer to boiler feed water and total energy consumption by the system increases with increase in gascooler outlet temperature.

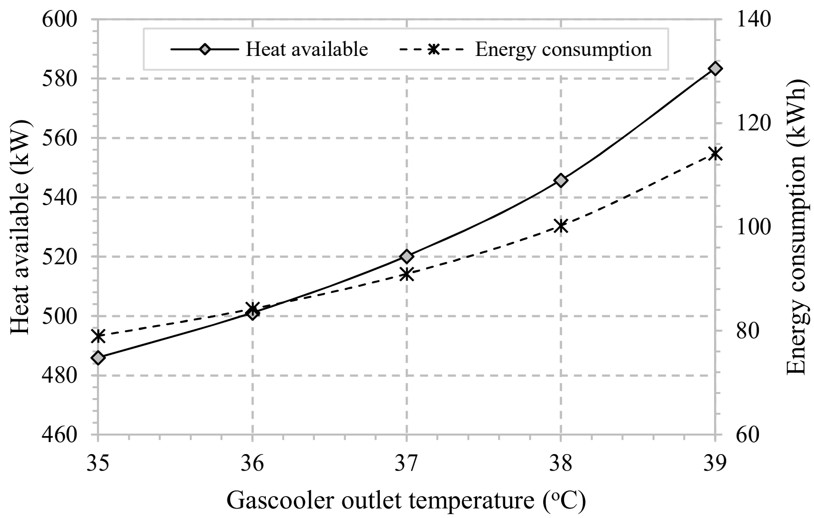


Figure 5.15: Variation of max heat available at gascooler with energy consumed.

5.3.3. Economic analysis

Change in energy consumption across the plant is assessed to identify potential cost saving and reduction in CO_2 emission due to introduction of the trans-critical CO_2 heat pump. Heat rejection from the gascooler is used to pre-heat the boiler feed water which reduces coal consumption resulting in saving of total fuel cost and CO_2 emission. In order to compute the overall effect of the proposed heat pump system, the electricity and coal consumption are computed. Annual demand of electricity and coal is computed on the basis of 16 hours per day plant operation.

Total electricity consumed by the proposed system is observed to increase (Table 5.7) due to additional compressor employed in the CO_2 heat pump system. The quantity of coal consumed reduces due to enhancement in feed water pre-heating from the CO_2 heat pump. Further, evaporative condenser is eliminated resulting in reduction of ground water used. About 5000 liters of water per week is saved. Rate of electricity consumed is taken as 0.12$ (8.17₹) kWh^{-1} and for coal is taken as 0.12$ kg^{-1} [90].

Table 5.7: Energy consumption using current & proposed trans-critical CO_2 heat pump system.

			Unit	Current	Proposed	RR
ENERGY CONSUMPTION	ELECT.	Heat pump elect. consumption	kWh		80	
		Cooling capacity	kW	37	37	
		Chiller	COP	2.3	2.3	
		Chiller electricity consumption	kWh	16.3	16.3	
		Total electricity consumption	kWh	16.3	96.3	
	COAL	Boiler heat capacity	kW	59	59	
		Calorific value of coal	kcal kg^{-1}	3500	3500	
		Density of coal	kg l^{-1}	0.866	0.866	
		Effective utilization ratio	%	0.5`	0.5`	
		Consumption of coal	kg h^{-1}	12.3	6.15	
		Water for evaporative cooling	Liter	5000		
ANNUAL ENERGY CONSUMPTION	ELECT.	Electricity consumption	MWh	101	525.6	
		CO_2 emission	t-CO_2 kWh^{-1}	0.93	0.93	
		Total CO_2 emission	t-CO_2 kWh^{-1}	93930	488808	
		Electricity cost	$ kWh^{-1}	0.12	0.12	
		Total electrical cost	$	12076.15	63072	
	COAL	Coal consumption	Tones	2807	1360	
		CO_2 emission	t-CO_2 kg^{-1}	2.71	2.71	
		Total CO_2 emission	t-CO_2 kg^{-1}	7606970	3685600	
		Coal cost	$ kg^{-1}	0.12	0.12	
		Total coal cost	$	329747.9	163200	
		Total CO_2 emission	t-CO_2	7700900	4174408	45.7%
		Total fuel cost	$	341824.1	226272	33.8%
		Savings	$		115552	

Total energy cost of electricity and coal are computed as sum of energy cost for both the fuels. CO_2 emission for direct coal burning in boiler is taken as 2.71 t-CO_2 kg^{-1} and for electricity consumption is taken as t-CO_2 kWh^{-1} [90]. Total CO_2 emission annually for electricity consumed and coal usage is computed as the sum of CO_2 emissions. Total reduction rate (RR) of electricity consumed and CO_2 emission for the system is computed using Eq. (5.15).

$$\text{Reduction rate (RR)} = \left(\frac{\text{Current System - Proposed System}}{\text{Current System}}\right)*100 \tag{5.15}$$

It is found that the total CO_2 emission is reduced by about 45.7% because of overall reduction in coal consumption and net cost of energy consumption cost is reduces by about 33.8%. The available heat can be used to preheat the water upto 70°C before feeding into the boiler. The overall impact of the proposed heat pump system is shown in Figure (5.16). Product & process costing analysis and fuel impact sheet are tabulated in appendix (AT8) & (AT9).

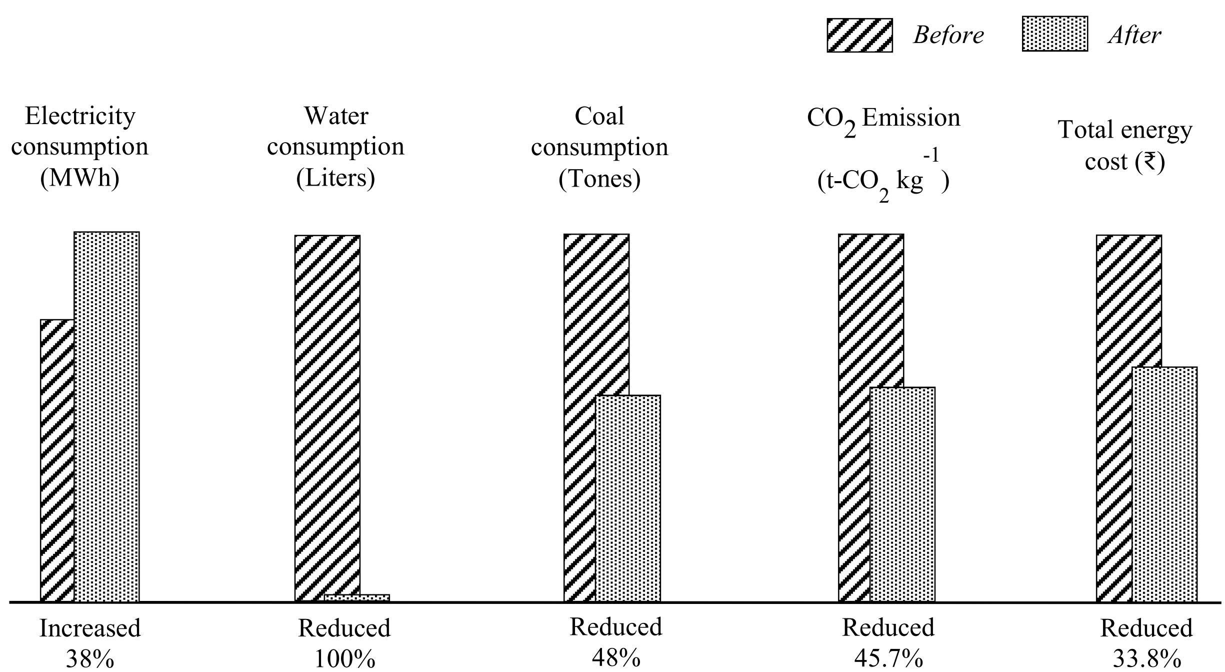


Figure 5.16: Overall impact of the heat pump system employed.

Economic assessment of the proposed trans-critical CO_2 heat pump system is carried out on the basis of PBP and LCC approach [13]. Various factors involved during the assessment are shown in Figure (5.17).

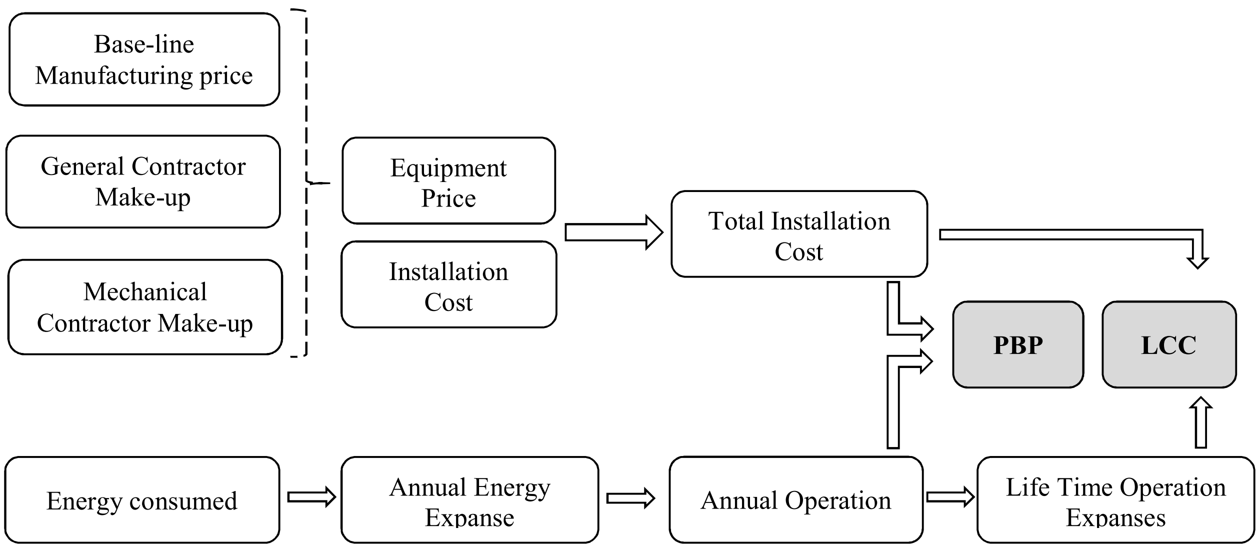


Figure 5.17: Economic assessment.

PBP is the measure of time it takes in years to recover the total additional expenditure for heat pump system and is computed using Eq. (5.16).

$$PBP = \frac{LCC}{Savings} \tag{5.16}$$

where, LCC is the life cycle cost of CO_2 heat pump system with respect to the savings by employing the CO_2 heat pump system. The total expense over the life of the trans-critical CO_2 heat pump system includes total installation cost (IC), equipment price (EQP), operational costs (OC), discount rate (r) and total life span (t) and the same is computed using Eq. (5.17).

$$LCC = IC + \sum_{t=1}^{N} \frac{OC}{(1+r)^t} \tag{5.17}$$

where, IC and OC are computed using Eq. (5.18) and (5.19) respectively.

$$IC = EQP + INST \tag{5.18}$$

where, IC is the sum of total equipment cost (EQP) and installation price (INST) associated with the installation of the CO_2 heat pump system.

$$OC = EC + RC + MC \tag{5.19}$$

where, OC is the sum of total electricity consumed (EC), repair cost (RC) associated with the component failure, computed using Eq. (5.20) and maintenance cost (MC) which is the service cost for maintaining continuous operation of the equipment.

$$RC = \frac{0.5\ EQP}{Life\ span} \tag{5.20}$$

Thus, the various inputs required to compute LCC are INST, EQP, IC, EC, MC, RC and total life span of the proposed trans-critical CO_2 heat pump system are taken as per standards [91], as tabulated in Table (5.8). LCC of the proposed CO_2 heat pump system involves the contribution of various costs (standard and fluctuating) and the same is computed using Eq. (5.17).

Table 5.8: LCC and PBP assessment.

Variable	Cost ($)	@ Rate	Total ($)
Installation price (INST)	1585		1585
Equipment (EQP)	2540		2540
Installation cost (IC)			3740
Electricity consumed (EC)	525.6 MWh	0.12$ kWh^{-1}	63072
Maintenance cost (MC)		10% of IC	374
Repair cost (RC)		4.9% of EQP	127
LCC			401052.5
PBP (in months)			~40

CHAPTER 6

Work Recovery Expander

Work recovery expander of various design have been reported in literature and can potentially enhance the overall COP of a refrigeration system particularly at high ambient. A multi attribute based decision making (MADM) study is carried out to determine the optimum option in a given situation. This chapter also reports an experimental investigation carried out to evaluate the performance of a scroll work recovery expander, using CO_2 as a working fluid. An open set-up is fabricated in order to test the performance of the scroll work recovery expander. The expander is redesigned and fabricated out of a scroll compressor. The chapter includes comparison of experimental and simulation based study results using a quasi-dimensional thermodynamic model.

6.1. Analysis of Literature on Work Recovery Expander

The status of research in work recovery expanders for trans-critical CO_2 refrigeration system is not mature enough to be recommendatory. Several options among which the selection is carried out, is reported in the literature review. Selection of the appropriate work recovery expander for a particular application is an important issue. A proposed selection frame work based on multi-attribute decision making (MADM) techniques, to identify the optimum solution from various options that are fuzzy in nature and has been applied in various domain is carried out. It is probably first time that MADM has been employed for decision making based on fuzzy information granules out of literature survey. Choice of refrigerant for a particular application is an area that goes by tradition but merits discretion based on ever changing environmental regulations and operating conditions. Seven criteria based selection framework is attempted to be established for ranking of refrigerants based on MADM techniques. Selection of refrigerant for specific application based on eight compering criteria are also carried out using popular MADM technique.

6.1.1. Multi-attribute decision making (MADM)

The research reported in article (2.3.3) regarding work recovery expander, had been intermittent, scattered across continents and parallel, as is evident from Figure (2.1). The trend of the research work must have been influenced by many practical factors which are difficult to track. However, it is felt that in order to achieve significant progress in shorter time, there is a need to consolidate the future research effort. An attempt is made here to compile the relative merits of six different work recovery expanders in qualitative and quantitative terms. The extracted mixed data from the literature survey is analyzed with the help of various approaches of MADM to extract conclusions. Based on information gathered out of literature survey and through peer group discussion, the qualitative data for CO_2 work recovery expanders is extracted and categorized within different aspects like maintenance, design, leakage, overall losses, efficiency, noise and cost. The qualitative data for the same are scouted from the literature. The linguistic grades imparted to the criteria studied as given in Table (6.1).

Table 6.1: Linguistic grades imparted to the criteria.

Grade criteria	A	B	C	D	E
Maintenance	Very high	High	Moderate	Low	Very low
Design	Very good	Good	Average	Poor	Very poor
Leakage	Very high	High	Moderate	Low	Very low
Losses	Very high	High	Moderate	Low	Very low
Efficiency	Very high	High	Moderate	Low	Very low
Noise	Very high	High	Moderate	Low	Very low
Cost	Very high	High	Moderate	Low	Very low

In order to draw conclusion from the set of qualitative data three optimization techniques namely FAHP, TOPSIS and PROMETHEE are selected. These are well proven techniques, used by many researchers in many different situations and these can handle mixed mode data very efficiently.

6.1.1.1. Fuzzy alternative hierarchy process (FAHP)

In traditional AHP method, the scale of pair wise comparison among criteria are restricted to crisp numbers and the method does not take into account the uncertainty associated with the mapping of one's judgment to a number. The principle of fuzzy logic was introduced into the AHP for MADM to take care of the same. This process makes it possible to adapt the

AHP in an environment in which the initial information, or the relations among criteria and alternatives, are qualitative, uncertain or imprecise. The steps recommended [92] is followed.

Step 1: Initial data collections, categorization with their critical output is produced in Table (6.1).

Step 2: Fuzzy AHP elements are created from initial data of Table (6.1). Subsequently establish the pair-wise comparison matrix established form fuzzy AHP element matrix. In this study, trapezoidal fuzzy members are used to represent pairwise comparison as given in Table (6.2).

Step 3: The conversion of crisp number into fuzzy numbers is calculated by using Eq. (6.1).

$$\tilde{A} = \left(x - 1, x - \frac{1}{2}, x + \frac{1}{2}, x + 1\right) \tag{6.1}$$

A comparison matrix $\tilde{X}$ is constructed from the relative importance matrix according to the pair-wise comparison.

$$\tilde{X} = \begin{array}{c} C_1 \\ C_2 \\ C_3 \\ \cdots \\ C_n \end{array} \begin{bmatrix} \tilde{x}_{11} & \tilde{x}_{12} & \cdots & \cdots & \tilde{x}_{1n} \\ \tilde{x}_{21} & \tilde{x}_{22} & \cdots & \cdots & \tilde{x}_{2n} \\ \tilde{x}_{31} & \tilde{x}_{32} & \cdots & \cdots & \tilde{x}_{3n} \\ \cdots & \cdots & \cdots & \cdots & \cdots \\ \tilde{x}_{n1} & \tilde{x}_{n2} & \cdots & \cdots & \tilde{x}_{nn} \end{bmatrix} \tag{6.2}$$

Step 4: Checking of consistency of the pair wise comparison matrix using Eq. (6.3).

$$\mathrm{CR} = \frac{CI}{RI} = \frac{\lambda_{max} - n}{RI(n-1)} \tag{6.3}$$

As a rule, only if CR < 0.10, the consistency of the matrix is considered as acceptable, otherwise the pair-wise comparisons matrix $\tilde{X}$ needs to be revised. For 7×7 matrix used, the relative index (RI) is found to be 1.32.

Step 5: Calculating weights based on the pair wise comparison matrix $\tilde{X}$, by using set of Eq. (6.4) and the matrix along with their calculated weights are produced in Table (6.3).

$$\alpha_j = \left[\prod_{j=1}^{n} l_{ij}\right]^{1/n} \quad \beta_j = \left[\prod_{j=1}^{n} m_{ij}\right]^{1/n} \quad \gamma_j = \left[\prod_{j=1}^{n} n_{ij}\right]^{1/n} \quad \delta_j = \left[\prod_{j=1}^{n} s_{ij}\right]^{1/n} \tag{6.4}$$

and

$\alpha = \sum_{j=1}^{n} \alpha_j \qquad \beta = \sum_{j=1}^{n} \beta_j \qquad \gamma = \sum_{j=1}^{n} \gamma_j \qquad \delta = \sum_{j=1}^{n} \delta_j$

Fuzzy weight vector $\widetilde{W}$ is constructed by using Eq. (6.5) and produced in Table (6.5).

$$\widetilde{W} = \left(\alpha_j \delta^{-1}, \beta_j \gamma^{-1}, \gamma_j \beta^{-1}, \delta_j \alpha^{-1}\right) \tag{6.5}$$

Table 6.2: Initial data collection of work recovery expander.

Expander	Maintenance	Design	Leakage	Losses	Efficiency	Noise	Cost
Piston	D	A	D	D	C	C	C
Rolling piston	D	B	B	D	C	A	C
Screw	D	B	B	C	C	D	B
Scroll	D	B	D	D	B	E	B
Turbo	B	C	C	C	C	B	A
Vane	B	B	C	C	D	C	A

Table 6.3: Scale of relative importance used in pair wise comparison of FAHP.

Scale of relative importance (crisp number)	Standard trapezoidal fuzzy number	Linguistic variable
1	(1, 1, 1, 1)	Equally important
3	(2,5/2,7/2,4)	Weakly important
5	(4,9/2,11/2,6)	Essentially important
7	(6,13/2,15/2,8)	Very strongly important
9	(8,17/2,9,9)	Absolutely important

Table 6.4: Pair wise comparison of relative importance matrix including their weights.

	αj	βj	γj	δj
Maintenance	0.33101	0.38808	0.49537	0.55218
Design	1.91897	2.18893	2.70161	2.96402
Leakage	0.51242	0.59219	0.78347	0.92112
Losses	0.62460	0.69277	0.87684	1.01696
Efficiency	3.53423	3.88774	4.57475	4.91016
Noise	0.21073	0.22616	0.27150	0.30772
Cost	1.32329	1.50349	1.90086	2.15457
∑	8.45525	9.47937	11.60440	12.82673

Table 6.5: Fuzzy weights and crisp weights.

Fuzzy weights					Crisp weights
w1	0.02581	0.03344	0.05226	0.06531	0.04375
w2	0.14961	0.18863	0.28500	0.35055	0.24124
w3	0.03995	0.05103	0.08265	0.10894	0.06938
w4	0.04870	0.05970	0.09250	0.12028	0.07889
w5	0.27554	0.33502	0.48260	0.58072	0.41525
w6	0.01643	0.01949	0.02864	0.03639	0.02485
w7	0.10317	0.12956	0.20053	0.25482	0.16969

Table 6.6: Normalized matrix.

Expander	M	D	L	Loss	Eff.	N	C
Piston	0.08824	0.29412	0.08824	0.08824	0.14706	0.14706	0.14706
Rolling piston	0.07143	0.19048	0.19048	0.07143	0.11905	0.23810	0.11905
Screw	0.07500	0.20000	0.20000	0.12500	0.12500	0.07500	0.20000
Scroll	0.08824	0.23529	0.08824	0.08824	0.23529	0.02941	0.23529
Turbo	0.17391	0.10870	0.10870	0.10870	0.10870	0.17391	0.21739
Vane	0.20513	0.07692	0.12821	0.12821	0.07692	0.12821	0.25641

Table 6.7: Product of normalized matrix and crisp weights.

Expander	M	D	L	Loss	Eff.	N	C
Piston	0.00386	0.07095	0.00612	0.00696	0.06107	0.00365	0.02496
Rolling piston	0.00313	0.04595	0.01321	0.00564	0.04943	0.00592	0.02020
Screw	0.00328	0.04825	0.01388	0.00986	0.05191	0.00186	0.03394
Scroll	0.00386	0.05676	0.00612	0.00696	0.09771	0.00073	0.03993
Turbo	0.00761	0.02622	0.00754	0.00858	0.04514	0.00432	0.03689
Vane	0.00897	0.01856	0.00889	0.01011	0.03194	0.00319	0.04351

Step 6: Calculating relative weights of all the sub criteria and then de-fuzzifying using Eq. (6.6), the normalized matrix produced after de-fuzzifying is given in Table (6.6).

$$N = \frac{l+2m+2n+s}{6} \tag{6.6}$$

Thereafter the de-fuzzified weights of the criteria are multiplied with their sub-criteria to find out the relative weights in Table (6.7) while their ranking is shown in Table (6.8).

Table 6.8: Ranking of the expanders.

Expander	Relative weights	Ranking
Piston	0.17757	2
Rolling piston	0.14348	4
Screw	0.16297	3
Scroll	0.21207	1
Turbo	0.13629	5
Vane	0.12518	6

6.1.1.2. Technique of order preference by similarity to ideal solution (TOPSIS)

TOPSIS is a MADM technique, which was originally developed by Hwang and Yoon in 1981. TOPSIS is based on the concept that the chosen alternatives should have the shortest geometric distance from the positive ideal solution and the longest geometric distance from the negative ideal solution. It is a method of compensatory aggregation that compares a set of alternatives by identifying weights for each criterion, normalizing scores for each criterion and calculating the geometric distance between each alternative and the ideal alternatives that is having the best score in each criterion.

Step 1: Table (6.1) is the input for formation to the decision matrix.

Step 2: Normalization of the decision matrix is done by using Eq. (6.7) and the normalized matrix is given in Table (6.9).

$$r_{ij}=\frac{X_{ij}}{\sqrt{\sum_{i=1}^{n} X_{ij}^{2}}} \tag{6.7}$$

Step 3: Construction of the weightage normalizing decision matrix by multiplying the normalized decision matrix by its associate weights. The weights of normalized values are calculated by using Eq. (6.8) and produced in Table (6.10).

$$V_{ij} = W_{ij} r_{ij} \tag{6.8}$$

Step 4: Determination of the positive ideal solution and negative ideal solution of the weighted normalized decision matrix by using Eqs. (6.9) & (6.10) and given in Table (6.11).

$$A^{+} = \left\{ \left(\frac{\max v_{ij}}{j \in J}\right), \left(\frac{\min v_{ij}}{j \in J'}\right) \right\} \tag{6.9}$$

$$A^{-} = \left\{ \left(\frac{\min v_{ij}}{j \in J}\right), \left(\frac{\max v_{ij}}{j \in J'}\right) \right\} \tag{6.10}$$

Step 5: Calculation of the separation measures of normalized decision matrix. The separations of each alternative from the positive and negative ideal solution are given by using Eqs. (6.11) & (6.12) and produced in Table (6.12).

$$Si^{+} = \sqrt{\sum_{j=1}^{n} (Vij - vj^{+})^{2}} \tag{6.11}$$

$$Si^{-} = \sqrt{\sum_{j=1}^{n} (Vij - vj^{-})^{2}} \tag{6.12}$$

Table 6.9: Normalized matrix for TOPSIS.

Expander	Maintenance	Design	Leakage	Losses	Efficiency	Noise	Cost
Piston	0.234	0.55	0.214	0.29	0.38	0.33	0.257
Rolling piston	0.234	0.443	0.571	0.29	0.38	0.67	0.257
Screw	0.234	0.443	0.571	0.49	0.38	0.201	0.411
Scroll	0.234	0.443	0.214	0.29	0.608	0.067	0.411
Turbo	0.625	0.277	0.357	0.49	0.38	0.53	0.514
Vane	0.625	0.166	0.357	0.49	0.228	0.33	0.514

Table 6.10: Calculated geometrical mean & weights.

Expander	Geometrical mean	Weights
Piston	0.307	0.145
Rolling piston	0.378	0.178
Screw	0.367	0.173
Scroll	0.271	0.128
Turbo	0.438	0.207
Vane	0.354	0.167
∑	2.115	

Table 6.11: Positive and negative ideal solution.

Expander	Maintenance	Design	Leakage	Losses	Efficiency	Vibration	Cost
Piston	0.033	0.079	0.031	0.042	0.055	0.047	0.037
Rolling piston	0.041	0.078	0.101	0.051	0.067	0.119	0.045
Screw	0.040	0.076	0.098	0.084	0.065	0.034	0.071
Scroll	0.029	0.056	0.056	0.037	0.077	0.008	0.052
Turbo	0.129	0.057	0.077	0.101	0.078	0.109	0.106
Vane	0.104	0.027	0.062	0.081	0.038	0.055	0.085
V+ve	0.029	0.079	0.031	0.037	0.078	0.008	0.037
V-ve	0.129	0.027	0.101	0.101	0.055	0.119	0.106

Table 6.12: Positive and negative separation measures.

Expander	S +ve	S -ve
Piston	0.092	0.206
Rolling Piston	0.017	0.0167
Screw	0.084	0.290
Scroll	0.037	0.180
Turbo	0.177	0.045
Vane	0.125	0.085

Step 6: Calculation of the relative closeness to the ideal solution from the positive and negative separation alternatives by using Eq. (6.13) and the same is produced in Table (6.13).

$$Ci^{+} = \frac{Si^{-}}{Si^{+} + Si^{-}} \tag{6.13}$$

The larger the Critical index (Ci^{+}) value, the better is the performance of the alternatives.

Step 7: Ranking of relative closeness matrix according to the performance order is produced in Table (6.13).

Table 6.13: Relative closeness and ranking of expanders.

Expander	Relative closeness	Ranking
Piston	0.69	3
Rolling piston	0.48	4
Screw	0.77	2
Scroll	0.82	1
Turbo	0.20	6
Vane	0.40	5

6.1.1.3. Preference ranking organization method for enrichment of evaluations (PROMETHEE)

The PROMETHEE method was developed at the beginning of the 1982; it can handle qualitative, quantitative and mixed data. Rather than pointing out the "right" decision, the PROMETHEE method helps decision makers to find the alternatives that best suits their goal and in understanding of the problem. The proposed decision making framework using PROMETHEE method provides a complete ranking of the alternatives from the best to the worst one using the net flows and ranking [93].

Step 1: Identifying the selection criteria for the considered decision making problem and short listing of the alternatives on the satisfying criteria produced in Table (6.1).

Step 2: Selection criteria is then used for preparation of decision matrix including the measures of values of all criteria for the alternatives.

Step 3: Construction of decision maker preference function and contribution of the alternatives to the decision matrix in terms of each separate criterion ranging 0 to 1 and the same is produced in Tables (6.14–6.20).

Table 6.14: Decision preference for maintenance of expander.

`	Piston expander	Rolling piston	Screw expander	Scroll expander	Turbo expander	Vane expander
Piston	-	0	0	0	0	0
Rolling piston	1	-	0	0	0	0
Screw	1	1	-	0	0	0
Scroll	1	1	1	-	0	0
Turbo	1	1	1	1	-	0
Vane	1	1	1	1	1	-

Table 6.15: Decision preference for expander design.

Design	Piston expander	Rolling piston	Screw expander	Scroll expander	Turbo expander	Vane expander
Piston	-	1	1	1	1	1
Rolling piston	0	-	0	0	1	1
Screw	0	1	-	0	1	1
Scroll	0	1	1	-	1	1
Turbo	0	0	0	0	-	1
Vane	0	0	0	0	0	-

Table 6.16: Decision preference for leakage in expander.

Leakage	Piston expander	Rolling piston	Screw expander	Scroll expander	Turbo expander	Vane expander
Piston	-	0	0	0	0	0
Rolling piston	1	-	0	1	1	1
Screw	1	1	-	1	1	1
Scroll	1	0	0	-	0	0
Turbo	1	0	0	1	-	0
Vane	1	0	0	1	1	-

Step 4: Calculating the outranking relations of the decision maker preference function, the leaving flow, entering flow and the net flow for alternatives are calculated by using Eq. (6.14|) to (6.16) and produced in Table (2.21) along with their ranking.

$$\varphi^{+}(a) = \sum_{x \varepsilon A} \prod_{xa} \tag{6.14}$$

$$\varphi^{-}(a) = \sum_{x \varepsilon A} \prod_{ax} \tag{6.15}$$

$$\varphi(a) = \varphi^{+}(a) - \varphi^{-}(a) \tag{6.16}$$

Table 6.17: Decision preference of loss from the expander.

Loss	Piston expander	Rolling piston	Screw expander	Scroll expander	Turbo expander	Vane expander
Piston	-	0	0	0	0	0
Rolling piston	1	-	0	0	0	0
Screw	1	1	-	0	0	0
Scroll	1	1	0	-	0	0
Turbo	1	1	1	1	-	0
Vane	1	1	1	1	1	-

Table 6.18: Decision preference for the expander efficiency.

Efficiency	Piston expander	Rolling piston	Screw expander	Scroll expander	Turbo expander	Vane expander
Piston	-	0	0	0	0	1
Rolling piston	1	-	0	0	0	1
Screw	1	1	-	0	0	1
Scroll	1	1	1	-	1	1
Turbo	1	1	1	0	-	1
Vane	0	0	0	0	0	-

Table 6.19: Decision preference of vibration from the expander.

Vibration	Piston expander	Rolling piston	Screw expander	Scroll expander	Turbo expander	Vane expander
Piston	-	0	1	1	0	1
Rolling piston	1	-	1	1	1	1
Screw	0	0	-	1	0	0
Scroll	0	0	0	-	0	0
Turbo	1	0	1	1	-	1
Vane	1	0	1	1	0	-

Table 6.20: Decision according to the cost of the expander.

Cost	Piston expander	Rolling piston	Screw expander	Scroll expander	Turbo expander	Vane expander
Piston	-	0	0	0	0	0
Rolling piston	1	-	0	0	0	0
Screw	1	1	-	0	0	0
Scroll	1	1	1	-	0	0
Turbo	1	1	1	1	-	0
Vane	1	1	1	1	0	-

Table 6.21: Weighted matrix, net dominance and ranking of the expanders.

	Piston	Rolling	Screw	Scroll	Turbo	Vane	$\varphi+$	Net flow	Rank
Piston	-	0.25	0.5	0.5	0.25	0.5	2	-4.75	6
Rolling piston	1.5	-	0.25	0.5	0.75	1	4	-0.75	5
Screw	1.25	1.5	-	0.75	0.5	0.75	4.5	0.25	2
Scroll	1.25	1.25	1	-	0.5	0.5	4.75	0.5	1
Turbo	1.5	1	1.25	1.25	-	0.75	2.25	0.25	3
Vane	1.25	0.75	1	1.25	1	-	5.25	-0.25	4
$\varphi-$	6.75	4.75	4	4.25	2	5.3			

The ranking obtained for the six expander designs from application of FAHP, TOPSIS and PROMETHEE optimization techniques is shown in Figure (6.1). Optimized rankings of the expanders are reversed in the plot to depict them effectively. It has been observed that the scroll type work recovery expander is at the highest ranking following by screw type work recovery expander and piston type work recovery expander.

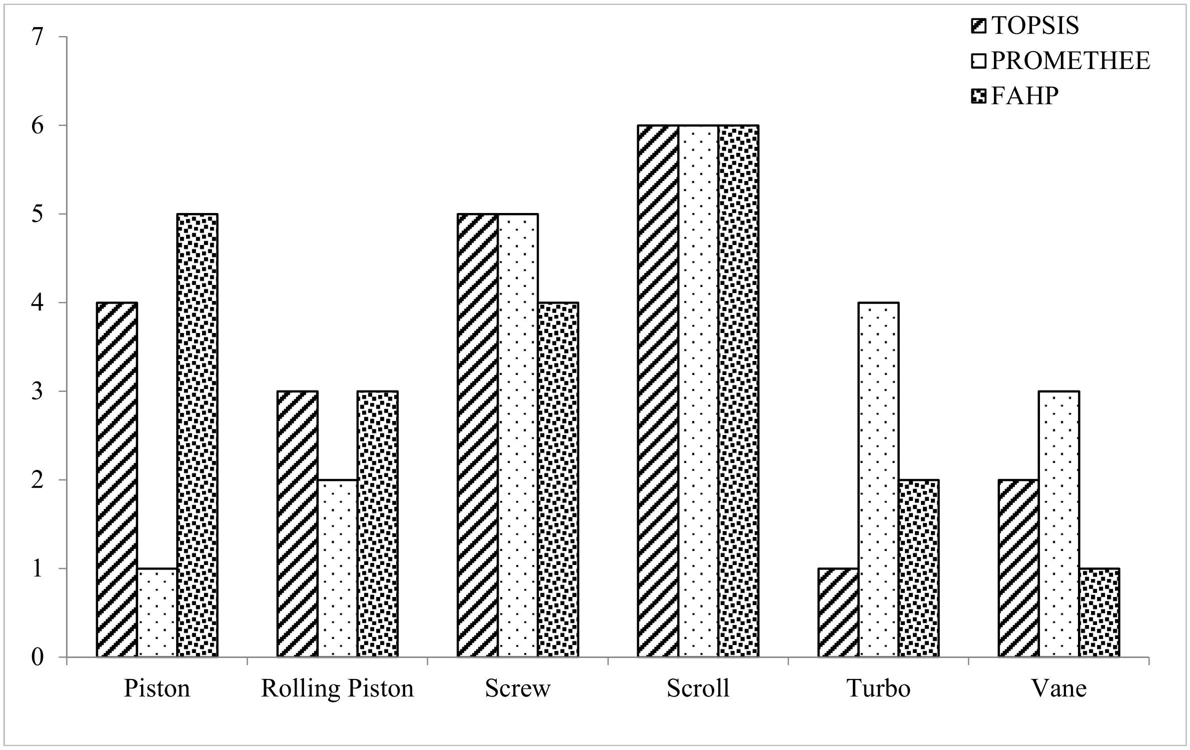


Figure 6.1: Ranking of expanders.

6.2. Experimental Investigation

For effective heat rejection from gascooler in CO_2 refrigeration system at high ambient temperature, the compressor is operated with high pressure lift. This implies enhanced power requirement and reduction in coefficient of performance (COP) of the system. High pressure operation of CO_2 system provide opportunity for work recovery through expander during throttling process.

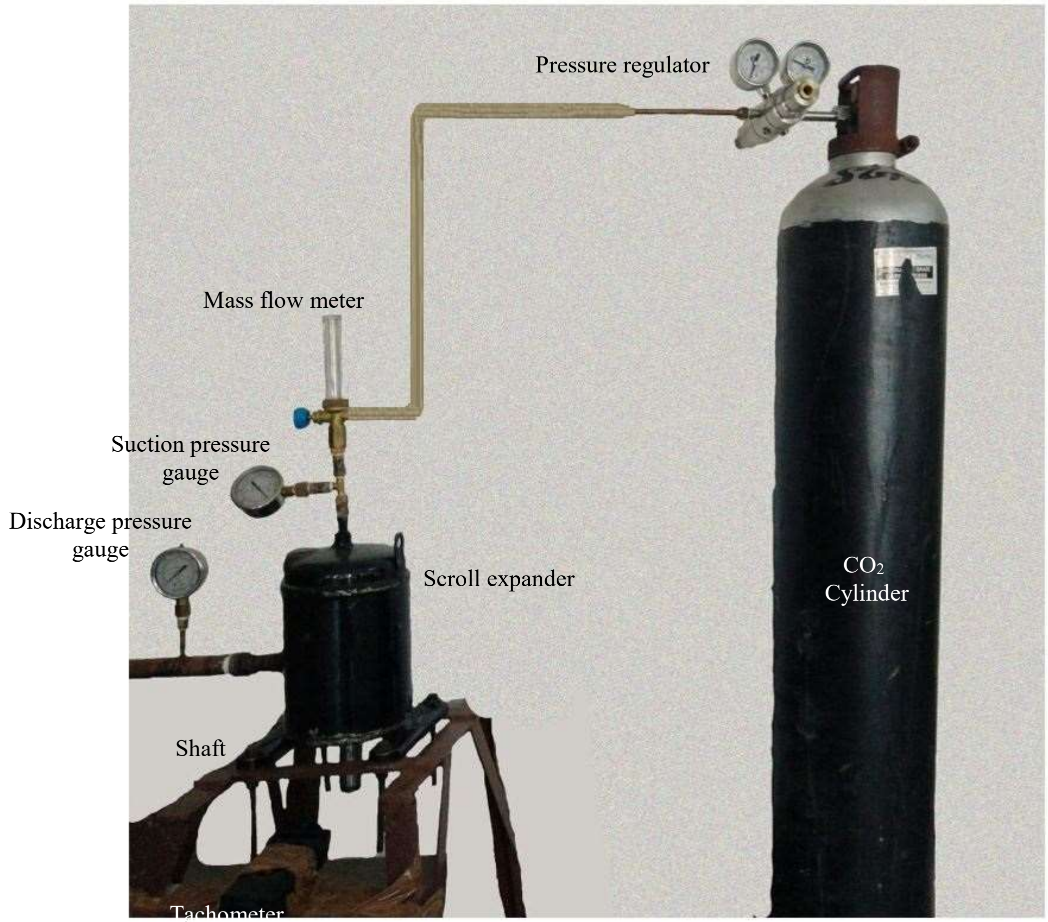


Figure 6.2: Experimental setup.

CO_2 trans-critical refrigeration systems operate in sub-critical zone for major part of the up-time even for warm climate region (as seen in article 4.3.1.2). Recovery from the expansion work from CO_2 refrigeration systems is viewed as a workable solution to tide over the challenge of typical low COP of such systems for warm climate context. Work recovery expander application of scroll device is not a very commercially ready technology, as a result, off-the-shelf devices are available but very costly. Some of the barriers for wide spread implementation of expanders are; relatively low work recovery and high initial investment. For the present work, an open loop arrangement is fabricated in order to evaluate the performance of a CO_2 driven scroll work recovery expander.

The open loop experimental set-up used for evaluation of performance of scroll work recovery expander is shown in Figure (6.2). The main components of the open loop CO_2

experimental setup are: a high-pressure CO_2 cylinder, a scroll expander, three temperature sensors, three pressure gauges, flow valve, tachometer and torque meter. Variation of pressure and temperature are measured at important locations, as shown in Figure (6.2). Rotational speed is measured using proximity tachometer.

6.1.1. Scroll work recovery expander design modification and fabrication

The scroll work recovery expander used in the open loop test rig is modified from a Hitachi, model 503DH−83D2 scroll compressor, having 7.37 cm^3 suction volume. In order to prevent refrigerant back flow, a check valve is arranged in the scroll compressor assembly. The scroll mechanical unit arrangement holds both stationary and moving scroll involute, Oldham coupling, sealing unit mounted on housing case with shaft, as shown in Figure (6.3). Involute profile angles of the scroll compressor used, are tabulated in Table (6.22) and illustrated in Figure (4.19).

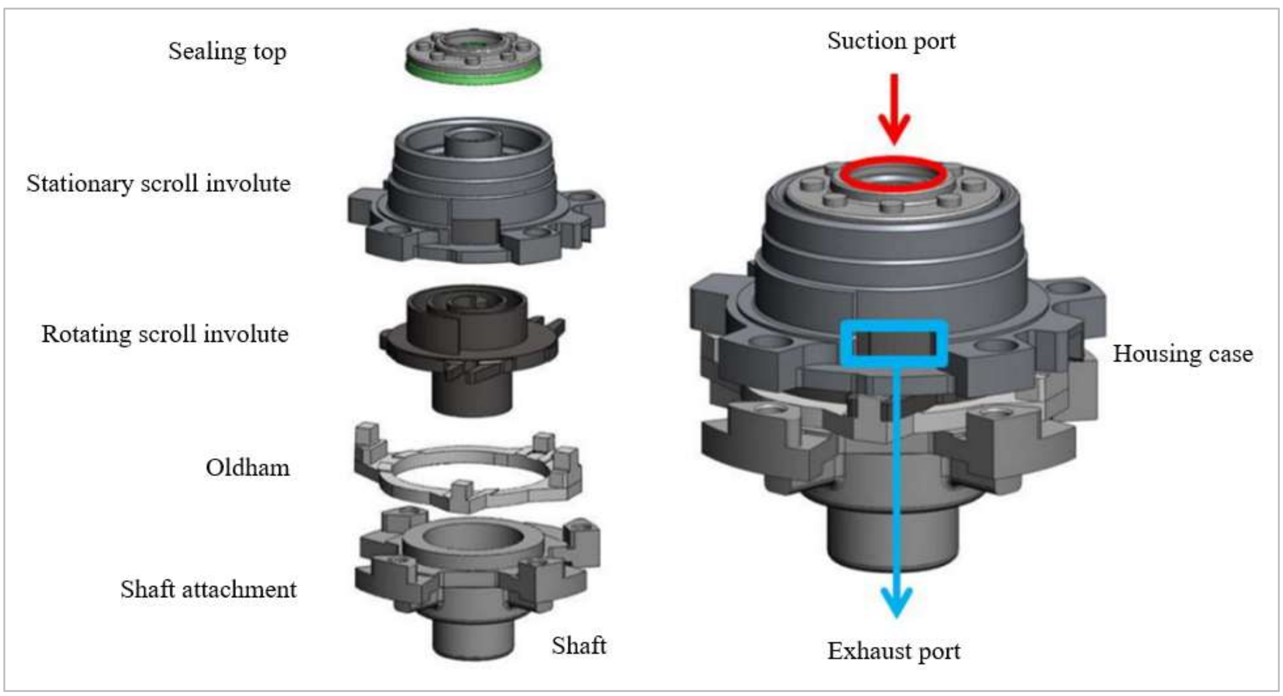


Figure 6.3: Parts of a scroll expander.

Motion of the orbiting scroll involute mounted on a shaft is kept constrained with the help of an Oldham against stationary involute. Three bolts are employed to hold the stationary scroll involute to the scroll base, this also prevents its radial motion. Two important functions performed by the shaft are: working as a medium to transfer mechanical power and working as oil pumping system.

During the conversion of scroll compressor into expander, the check valve arranged in the assembly is removed to allow refrigerant flow to the low-pressure chamber from the high-pressure chamber. An insulating layer is inserted between the moving and stationary involute casing to reduce friction and heat loss. A mechanical coupling is created for the direct drive operation, using a key joint assembly to the shaft end.

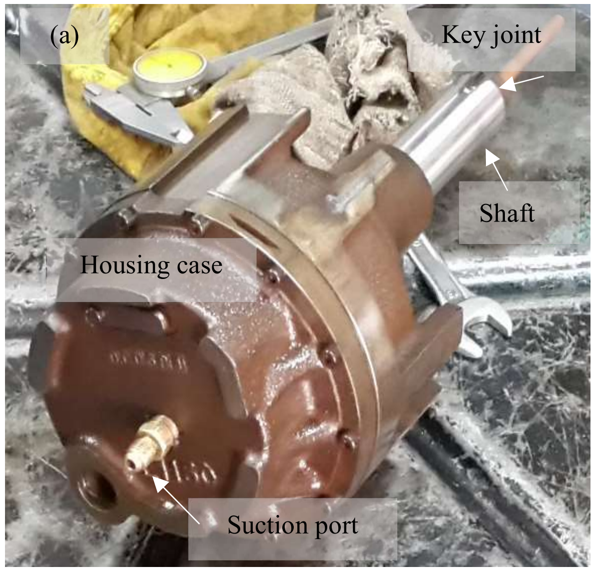
Table 6.22: Internal geometry of scroll expander.

Variable	r_b (mm)	h (mm)	$\Phi_{o,0}$ (rad)	$\Phi_{i,0}$ (rad)	$\Phi_{o,S}$ (rad)	$\Phi_{i,S}$ (rad)	Φ_e (rad)	r_o (mm)
Value	3.2	3.3e+2	-6.9e-1	6.9e-1	2.6	5.7	1.70e+1	5.5

Suction and discharge ports of the compressor are also interchanged to reverse the refrigerant flow direction. The sealing cup at the top of the scroll expander is used as suction port holder. After conversion, the refrigerant flow direction is altered and the refrigerant suction is axial while discharge is radial during the expansion process. A port is created on the outer jacket (housing) which acts as the discharge port. The mechanical work generated during the expansion process is transferred to the shaft attached to the moving scroll involute. Picture of internal mechanical component of the scroll expander with housing and oil chamber is shown in Figure (6.4,a) and (6.4,b) respectively.

6.1.2. Measuring devices

The open loop experimental setup is used to determine the effect of four operating parameters: suction pressure, exhaust pressure and rotational speed. Variation of pressure is measured at important locations such as: at suction and discharge of scroll expander. Flow meter is mounted on the top at the section port, in order to measure the mass entering.



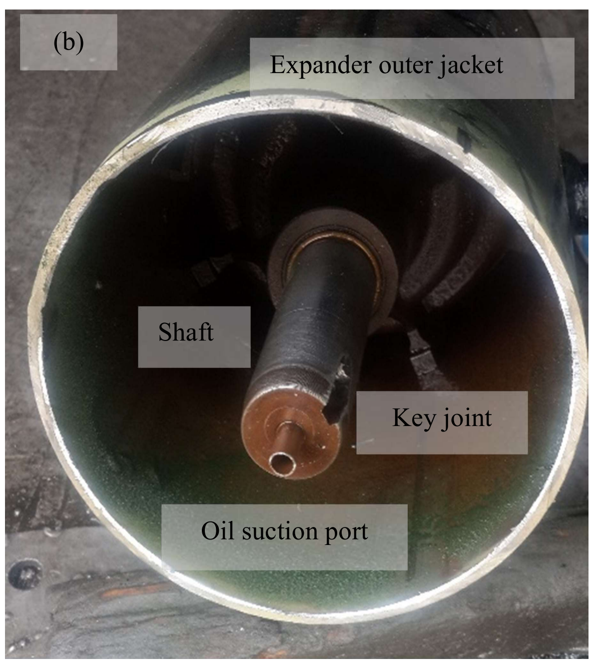


Figure 6.4: (a) Mechanical unit of scroll expander, (b) Oil chamber and suction port.

In-direct contact type tachometer is used to measure the rpm of the rotating shaft coupled with the scroll expander. Characteristic of the measuring devices used in the experimental setup are tabulated in Table (6.23). The range of the various parameters used during testing are tabulated in Table (6.24).

Table 6.23: Range of the measuring devices.

Variable	Specifications	Measuring range	Uncertainties
Pressure (bar)	Dial gauge (Glycol fix)	0 to 60	0.1 bar
Torque meter (rpm)	Dial photo type	60 to 100000	±0.05%
Flow meter (kg min^{-1})	Rasemount rotameter	0.25 to 13	±1%

Table 6.24: Value/range of parameters.

Parameter	P_{su} (MPa)	M_{meas} (kg s^{-1})	N (rpm)
Range	3.5−1.5	0.002−0.017	800−2500

6.3. Experimental Evaluation of Scroll Expander Performance

In order to evaluate the performance of the scroll work recovery expander, under various input parameters, further experiments are carried out. The shaft work is calculated using Eq. (6.17).

$$W = 2\pi N\tau/60 \tag{6.17}$$

Fig. (6.5) shows the variation in shaft work generated with respect to pressure ratio. In the same plot, corresponding variation in shaft speed is also shown. Work generated by the shaft is in the range of 300−2800 W. It is observed that, with increase in suction pressure i.e. increase in pressure ratio, the shaft work increase. This is ascribing to the fact that, as the pressure ratio increases, the work generated during the expansion process with in the scroll increases. From the above observation, it is also concluded that the shaft speed has major influence on work generated.

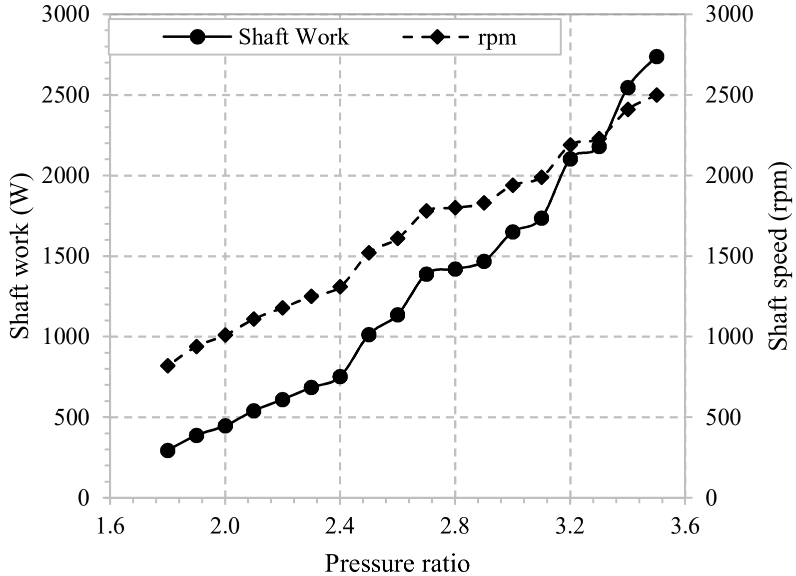


Figure 6.5: Variation of shaft work and shaft speed w.r.t. pressure ratio.

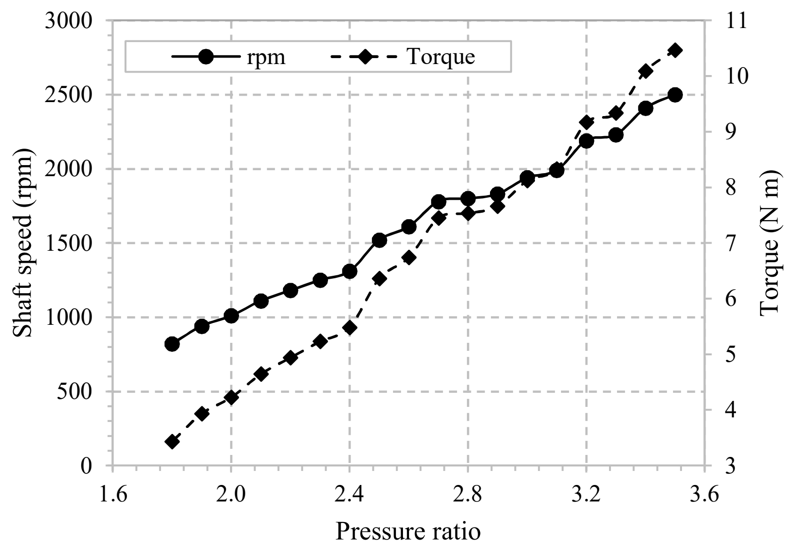


Figure 6.6: Variation of shaft speed and torque w.r.t. pressure ratio.

Fig. (6.25) shows the variation of shaft speed with respect to pressure ratio. In the same plot, variation in torque shown. It is observed that, the toque increases as the load increases on the shaft. The shaft work obtained during the tested range is between 800-2500 rpm. Error analysis with experimental data is also carried out using a method recommended by ASHRAE [2]. The standard error and the standard deviation are determined using Eq. (6.18).

$$\mathcal{E} = \frac{1}{n}\sum_{i=1}^{n}(V_{meas} - V_{sim}) = \frac{1}{n}\sum_{i=1}^{n}(\mathcal{E}_i) = \left[\frac{1}{n}\sum_{i=1}^{n}(\mathcal{E}_{meas} - \mathcal{E}_{sim})^2\right]^{0.5} \tag{6.18}$$

Table 6.25: Error analysis.

Variable	Standard error	Standard deviation	Minimal deviation	Maximal deviation
Mass flow rate (g s^{-1})	0.52	2.226	-7.2%	+4.8%
Shaft work (W)	173	738	-4.6%	+12.3%

6.4. Artificial Neural Network (ANN)

Keeping in mind the high cost of scroll work recovery devices, an attempt is made to establish an ANN model of the same. The ANN has three nodes in input layer to read input parameters: mass flow rate, suction pressure and suction volume and two nodes in output layer for parameters: rotational speed and shaft work. The ANN has a single hidden layer with adequate nodes as shown Figure (6.7).

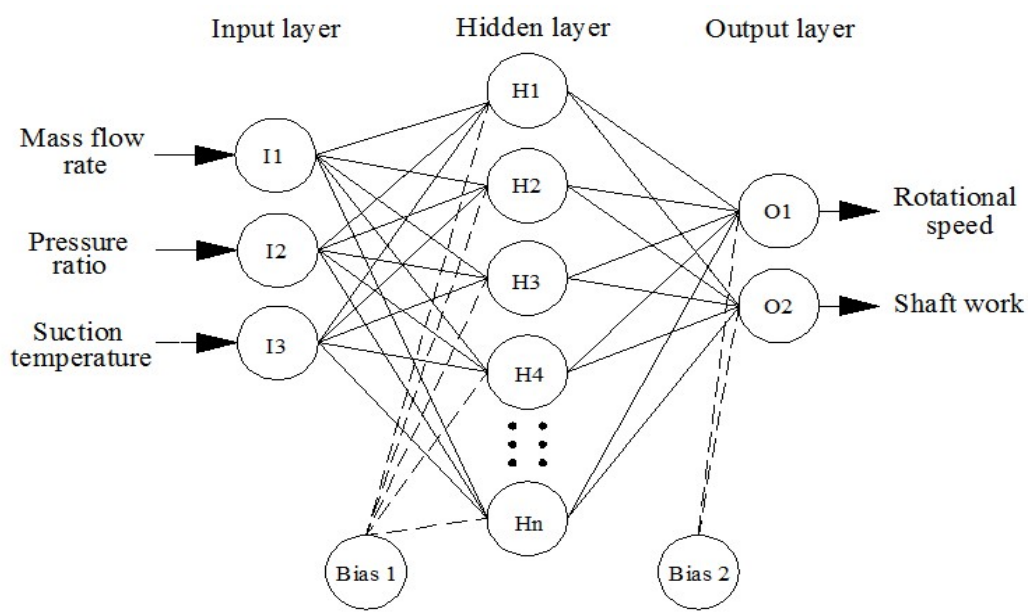


Figure 6.7: ANN model structure.

Neural network tool in SPSS platform, is used to train and validate the ANN over range of experimental data. Feed forward back propagation neural network (FFBPNN) with Levenberge Marquardt algorithm is used because of its simplicity and popularity. [94]. The number of neurons in hidden layer is an important issue for error minimization. We arrived at the best fit through trial and error. Performance of the ANN for number of neurons from 6 to 15 are compared through mean square error (E^2), computed using Eq. (6.19) and coefficient of determination (R^2), computed using Eq. (6.20), as tabulated in Table. (6.26).

$$E^2 = \frac{\sqrt{\sum_{m=1}^{n}\left(\frac{y_{output}-y_{actual}}{y_{output}}\right)^2}}{n} \tag{6.19}$$

$$R^2 = \left[\frac{\sum_{m=1}^{n}(y_{output}-\bar{y}_{output})(y_{actual}-\bar{y}_{actual})}{\sqrt{\sum_{m=1}^{n}(y_{output}-\bar{y}_{output})^2 \sum_{m=1}^{n}(y_{actual}-\bar{y}_{actual})^2}}\right]^2 \tag{6.20}$$

Table 6.26: Comparison using various hidden neurons in ANN model.

Neuron	MSE (E^2)	COD (R^2)	
		Shaft speed	Shaft work
6	0.014	0.987	0.984
7	0.028	0.978	0.980
8	0.014	0.986	0.984
9	0.020	0.981	0.980
10	0.016	0.981	0.980
11	**0.019**	**0.989**	**0.985**
12	0.017	0.982	0.981
13	0.025	0.981	0.977
14	0.014	0.982	0.983
15	0.019	0.983	0.982

The model with the larger R^2 and smaller E^2 are the better. 11 number of neurons in hidden layer is found to be optimal. We have total 171 experimental data points out of which 70% randomly selected values are used rest for training and validation. The trained ANN is shown in Figure (6.8) and residual obtained from the trained ANN model is shown in Figure (6.9).

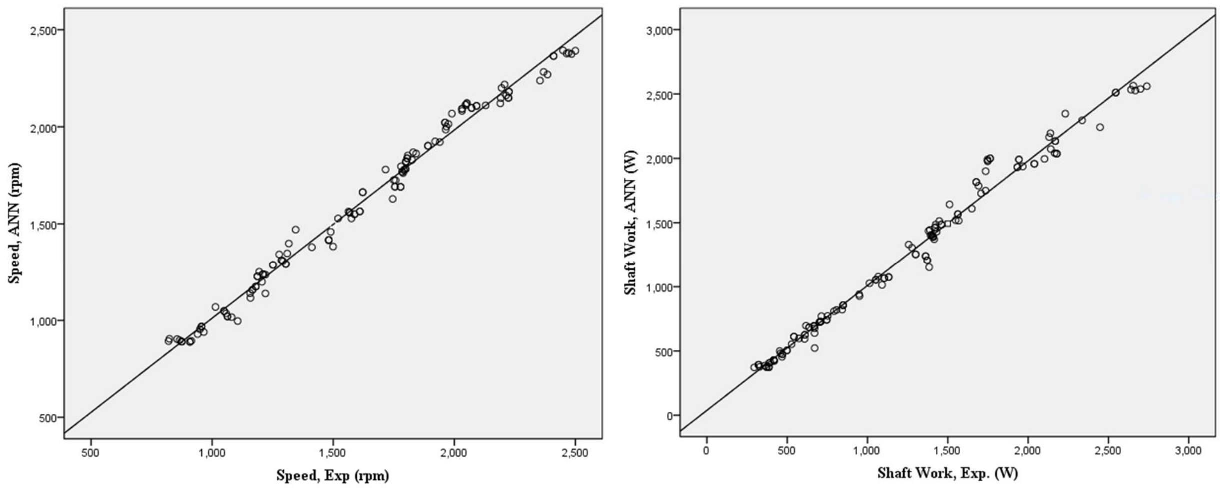


Figure 6.8: ANN model training and validation for speed and shaft work using 11 neurons.

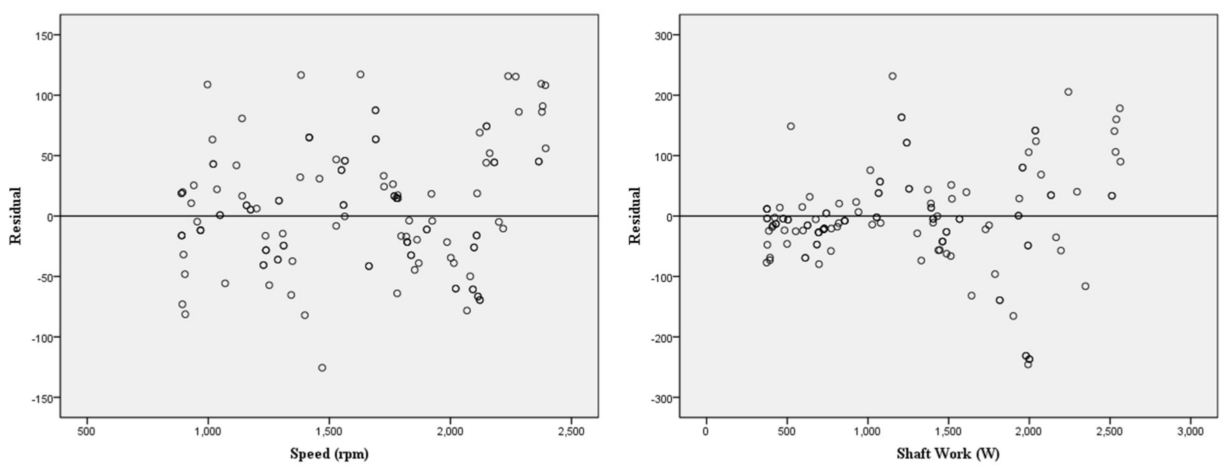


Figure 6.9: ANN model residue of speed and shaft work using 11 neurons.

6.4.1. ANN model validation

The remaining 30% data points are again used to validate the ANN model. Good agreement is found between ANN model prediction and experimental data. It is observed that the maximum deviation between experimental and simulation results are within ±7.5% and ±11.1% for rotational speed and shaft work, as shown in Fig. (6.10).

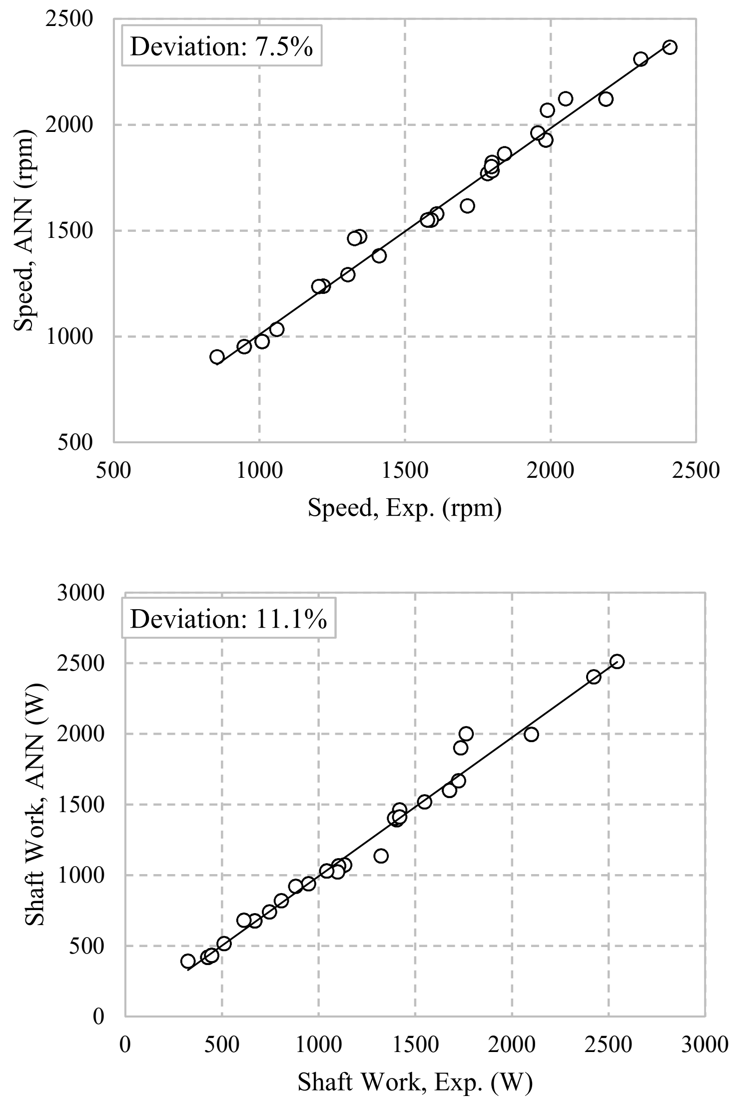


Figure 6.10: Comparison between measured and predicted rotational speed & shaft work using ANN.

The ANN very well from the scroll work recovery expander with CO_2 as refrigerant. This ANN closed mimic an actual scroll expander and can be used in a simulated closed loop study of CO_2 refrigeration system in sub-critical zone.

CHAPTER 7

Conclusions and Future Scope of Work

The study covered theoretical analysis of trans-critical CO_2 systems both for refrigeration and heat pump applications. Specific example from a dairy industry is taken to evaluate both the applications in warm weather. Experimental investigation of a CO_2 scroll expander is carried out for work recovery.

This chapter compiles the overall conclusions drawn from the study. The chapter also incorporate recommendation for future scope of work.

7.1. Conclusions

Revival of carbon dioxide (CO_2) as a preferred working fluid for various Heating, Ventilation and Air Conditioning (HVAC) applications took place post 1990s due to enhanced awareness about direct or primary environmental impact like ozone depletion and greenhouse effect caused by the synthetic refrigerants when released into atmosphere due to leakage at various stages of its production and handling and also secondary impact of carbon release due to substantial share of power consumption by HVAC sector. Certain synthetic refrigerants also have harmful water and soil contaminating potential. While CO_2 is a biosphere gas having unit greenhouse effect. Conventional CO_2 refrigeration cycles, however, have lower COP, implying higher secondary impact on environment, more so for higher ambient operation. It is imperative, therefore, to probe various possible means of enhancing COP of CO_2 cycles, explore newer application areas as well as investigate road blocks implementation in warm climate.

7.1.1. Modeling and simulation

Energy and exergy analysis of five different evolving configurations of CO_2 systems were carried out for warm weather applications (35─50ºC), including application in single as well as multi-stage systems. The comparative study also brings out relative advantages of work recovery expander on the overall performance of the CO_2 system. A minimum TEWI (Total Equivalent warming impact) based criteria for the selection of intermediate pressure and temperature is also suggested. A semi-empirical model of a scroll work recovery expander was built for CO_2

refrigeration system. The model was validated in the trans-critical range from published experimental data. Major conclusions drawn from the simulation study are following:

- A cycle with IHX is found to have a lower mass flow rate at higher ambient temperature. This implies that, systems can be operated effectively at a lower gascooler pressure. Further, performance of system with IHX is found to be comparatively better at higher ambient temperature.
- For applications like refrigeration and air cooling, the multi-stage systems are found to have similar performance as the basic system. However, the FGI multi-stage system is found to have a comparatively better performance at all investigated gascooler outlet temperatures.
- The operating pressure and mass flow rate are found to be higher for multi-stage systems although, compressor discharge temperature is found to be lower. The inter cooler (IC) multi-stage system is found to have higher operating pressure; while this may be improved upon by utilizing more effective IC fluid than air.
- The effect of real time constraints like pressure drop in gascooler, compressor efficiency, approach temperature, expander efficiency, effectiveness of IHX and degree of superheat on the system performance reveals that the efficiency of compressor has dominating effect and efficiency of expander has minor effect of system's COP among the set.
- A combination of two low GWP fluids R1234yf / R744 are recommended for use in cascade system for low overall environment impact & high COP. Minimization of TEWI is recognized as an effective way to decide the IT for the cascade system.
- Effect of independent variation of condenser and evaporator temperature on COP_{HTC} and COP_{LTC} respectively are found to be high but effect on COP_{sys} is comparatively lower.
- Energy analysis of a scroll work recovery expander is carried out in the gas cooler outlet temperature range 32ºC to 48ºC for -10ºC evaporator temperature. Maximum 71% isentropic efficiency is achieved at pressure ratio ~3.7 based on a semi empirical model. Optimum range of pressure ratio recommended for maximizing the work recovery is in the range 3.3 to 4.2.
- A trans-critical CO_2 refrigeration system with work recovery scroll expander is found to have a high COP and low optimum discharge pressure. It is demonstrated that the

expander is capable of potentially recovery of about 20% to 30% of energy consumed by the compressor.

- Based on the year around data of ambient temperature at New Delhi, economic analysis reveals that the total PBP from a work recovery scroll expander of various capacity are within range 1 to 1.7 years. The same is minimum for a medium range capacity scroll expander. The PBP duration appears reasonable for initial investment.

7.1.2. Indian dairy industry- A case study

As a case study is developed to evaluate feasibility of CO_2 trans-critical system for warm weather conditions. Field data from a mid-sized milk & milk product handling plant located in the Northern Indian state of Punjab is taken for analysis. Year-round fluctuation of milk load, heating & cooling demand and ambient temperature are noted. Two schemes are conceptualized and evaluated, one for total replacement of the existing ammonia based refrigeration system with a CO_2 booster refrigeration system and another, a CO_2 heat pump system retrofitted above the ammonia refrigeration system for utilization of the waste heat. Year-round ambient temperature data suggests that a replacement plant will run under trans-critical condition for 26.3% of time and in sub-critical condition for 73.6% of time. Major conclusions drawn from a case study are as following:

- COP of the trans-critical CO_2 booster refrigeration system is found to be lower than the ammonia based refrigeration system, round the year. It is also observed that as ambient temperature increases further beyond critical temperature (of CO_2), the COP falls more drastically.
- Heat and work recovery are distinct possibilities for super-critical operation of CO_2 cycle.
- From exergy analysis, it is also revealed that the exergy destruction and its corresponding cost for trans-critical CO_2 booster refrigeration system is 3 times more than the NH_3 based refrigeration system during super-critical operation.
- Incorporation of a heat recovery system that is only 50% efficient and a work recovery expander that is 65% efficient in place of expansion valve can leads to potential COP improvement by about 30%.

- By employing the proposed scheme of a trans-critical CO_2 heat pump system, ~37 kWh of hear can be recovered coupled with saving of ~5000 liters/week of ground compare to the present consumption rate.
- While overall coal consumption is reduced 48% but total electricity consumption is increased by ~ 50%.
- Reduction in CO_2 emission achieved is about 45.7% and Net expenditure on energy is reduced by about 33.8%. These lead to PBP of 40 months for installation of the proposed trans-critical CO_2 heat pump system.

7.1.3. Experimental investigation of a scroll expander

Performance analysis of a scroll expander was carried out in an open loop test set-up for a limited range of pressure (sub-critical) available from a CO_2 cylinder. A back propagation Artificial Neural Network (ANN) was trained and validated using the experimental data which can be further used for prediction of work recovery for a wider range of sub-critical temperature with CO_2 as working fluid. Major conclusions from the study are as following:

- The maximum deviation between experimental and simulation based outcome for rotational speed and shaft work are within ±7.5%, and ±11.1% respectively.
- Work recovery system is found to be most effective in the pressure ratio range of 2 to 3.3. Maximum shaft work generation recorded is 2800 W at pressure ratio 3.2.
- An ANN model trained and validated for the whole range of sub-critical operation of CO_2 scroll expander is found to have good agreement with experimental data. This should be useful to easily assess the actual work recovery from a scroll expander while operating in a CO_2 refrigeration system during sub-critical operation.

7.2. Future Scope of Work

India has signed the Montreal Protocol along with various other developing countries in October 2016. A further drop in the intake and production of synthetic refrigerants on global perspectives is offered by the recent (December 2016) adoption of the Kigali Amendment to the Montreal Protocol. Under these Protocol, India is bond to phase out synthetic refrigerants by 2030. The success of sub-critical CO_2 system in European countries having cold ambient cannot be transplanted in India having climate pre-dominantly in the super-critical zone for CO_2. This require tremendous in-house research focus. The commercial success of CO_2 system in India is a precursor to its wide spread use as a replacement for the synthetic refrigerants. So, maximum focus ought to be on improving the COP of trans-critical CO_2 system in future research.

- New and innovative multi-stage system configurations to be explored and evaluated for various applications at warm weather conditions using work recovery expander, ejector or new technology.
- Two-phase flow characteristic inside scroll expander can be studied during trans-critical operation, in order to improve the design of the same that is more suitable for high pressure (warm climate) operation.
- Pilot implementation of CO_2 heat pump system in dairy industry for waste heat recovery to be explored.

Thesis title: Trans-critical CO_2 System for Warm Climate: An Evaluation of System Configurations and Scroll Expander

Chapter 1: Introduction

CO_2 trans-critical systems, in recent years, have attracted attention for a wide variety of refrigeration applications. Natural refrigerants are perceived to be potential permanent solution in the face of progressive restrictions being imposed upon the use of synthetic refrigerants. Revival of interest in natural working fluids for refrigeration and air conditioning application in the early 1990's is credited to environmental concern over ozone depletion potential (ODP) and later, on global warming potential (GWP) of the various synthetic refrigerants like CFCs, HCFCs and HFCs. Natural refrigerants such as air, water, ammonia, carbon dioxide, isobutene, propane etc. are economic, ecologically safe, and have zero ODP and low GWP.

CO_2 (R744) in particular has unit GWP. COP of CO_2 systems however, decreases drastically for high temperature application. Reason for low COP are high pressure operation beyond critical point (7.1MPa & 31.1°C) and large throttling losses associated with expansion from a high super-critical pressure to a sub-critical pressure. There are, however, unique opportunities associated with this high ambient temperature operation. A large number of modifications are feasible and have been explored beyond the basic trans-critical CO_2 refrigeration cycle to improve its performance and also to adapt to the various working conditions. Researchers have explored several application areas like automotive air conditioning, heat pump for domestic water heating, commercial refrigeration, large commercial space cooling etc. Use of work recovery expander as a replacement of conventional expansion valve is also emerging as a promising option to achieve commercial success. The main objective of this study is to identify optimum operating and design conditions for CO_2 system operating in trans-critical mode at high ambient conditions in order to improve the overall performance.

Chapter 2: Literature Review

A detailed literature review is carried out in order to identify the recent trends and gaps in research for CO_2 systems. Research trend at present is focused on the improvement of system efficiency through various modifications of a simple cycle, for example multi-staging, use of work recovery expander, cascade configuration, use of IHX, etc. The commercial success of the CO_2 systems is still a challenge at high ambient context. Moreover, a few low GWP synthetic refrigerants are also recently introduced which could work efficiently at high ambient temperature.

The gap areas identified from the present literature review are as following:

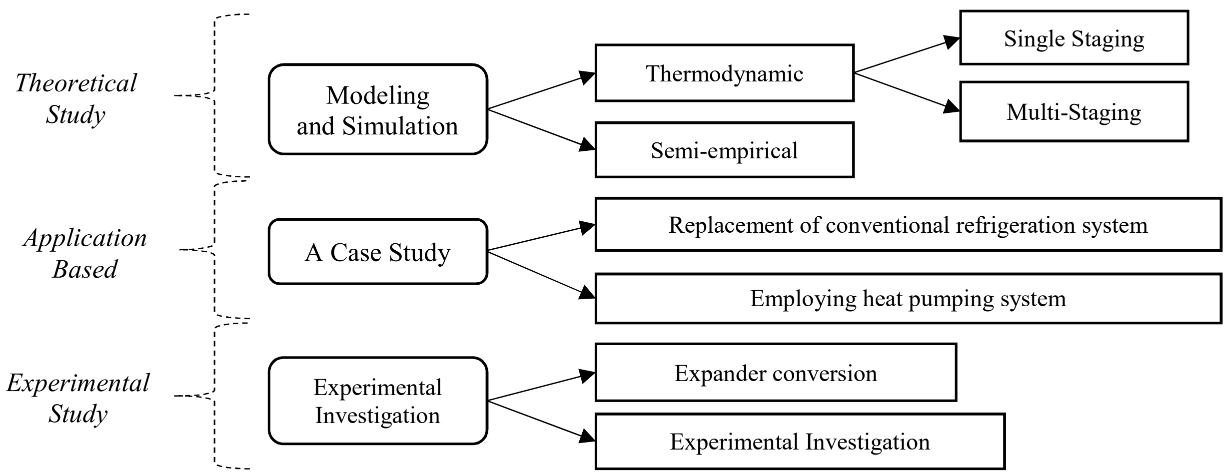
- Work recovery expander appeared as a promising option to improve overall performance of the CO_2 system (theoretically and practically), but the advantage is prominent only at high pressure range. Two phase behavior of the refrigerant during the expansion process is still unpredictable and which is greatly influenced by the expansion profile.
- Screw and Scroll expanders are popular due to high efficiency & high reliability and can be viewed as a prominent option. But, the major drawback is their very high component cost and availability off-the-shelf. So, there is a need to find an alternate economical method to develop a work recovery expander. Some researchers reported reverse engineering of expander from compressor with various degree of success.
- CO_2 heat pump is employed in various process industries to partially fulfill both heating and cooling demands. But, a very few studies were reported for waste heat recovery form the process industries by the CO_2 system.
- Various supermarkets running with CO_2 systems in colder region are successful and viewed as permanent solution for clean environment. However, for warm climatic condition, replacement of a conventional system with CO_2 system has not been found to be commercially viable.

Chapter 3: Study Objectives

Major objectives of the present work are:

- To identify the most prominent options among the various work recovery expander designs based on literature survey and validate based on data analysis.
- Simulate various CO_2 trans-critical refrigeration system configurations, including single as well as multi-stage systems to find optimum system configurations for warm climate.

- Model and simulate a potential implementation of CO_2 system for warm climate. A medium capacity commercial milk chilling plant is identified for developing the case study.
- Investigate working of a work recovery expander and its potential advantage, including study of both economic and environmental aspect.
- Attempt to re-engineer a scroll compressor to work as expander and experimentally investigate it's working.



Chapter 4: Modeling and Simulation

Energy and exergy analysis of five different evolving configurations of CO_2 system were carried out for warm weather application (35─50ºC), including single as well as multi-stage systems. The comparative study also brings out relative advantages of work recovery expander on the overall performance of the CO_2 system. Usefulness of internal heat exchanger (IHX) for high ambient application is also established. Among the multi-stage systems, flash gas intercooler (FGI) configuration was found to be better at high ambient.

The study also revealed that the optimum inter-stage pressure deviates significantly from the conventional inter-stage pressure prediction, given by Missimer's Eq. & Schmidt's Eq. A minimum total equivalent warming impact (TEWI) based criteria for the selection of intermediate pressure is suggested. The advantage of the same for the overall COP improvement is also demonstrated with the help of 56 refrigerant combinations in a cascade refrigeration system.

Chapter 5: Indian Dairy Industry – A Case Study

As a case study, for the evaluation of CO_2 trans-critical system for warm weather conditions, field data from a mid-sized milk & milk product handling plant located in Northern Indian state of Punjab is taken. Year-round fluctuation of milk load, heating & cooling demand and ambient temperature are noted. Two schemes are tested, one for total replacement of the existing ammonia based refrigeration system with a CO_2 booster refrigeration system and another, a proposed CO_2 heat pump system over and above the ammonia refrigeration system for utilization of waste heat. The year-round ambient temperature data suggests that a replacement plant will run under trans-critical condition for 26.3% of time and in sub-critical condition for 73.6% of time. The replacement CO_2 refrigeration plant was found to have lower COP and high initial investment. However, the proposed CO_2 heat pump system combined with the existing ammonia system was found to be very effective and potentially reduce energy cost by 47.6% and CO_2 emission by 41.2%. Further, use of ground water was reduced by 5000 liters per week. Payback period for the proposed heat pump system was found to be about 20 months.

Chapter 6: Experimental Investigation of a Scroll Expander

A semi-empirical model of a scroll work recovery expander was built for CO_2 refrigeration system. The model was validated in the trans-critical range from published experimental data. Limited CFD of analysis flow of CO_2 through the scroll involute was also carried out. A scroll compressor of 10kW capacity is purchased and reworked, as per plan to make it work as a scroll expander. The reworking process gave us a working scroll work recovery expander at a much cheaper price than an off-the-shelf scroll expander. Performance analysis of the same was carried out in an open loop test set-up for a limited range of pressure (sub-critical) available from a CO_2 cylinder. Work generated is found to be high in the pressure ratio range of 2 to 3.3. Maximum shaft work generated is 2800 W at 3.2 pressure ratio.

A back propagation Artificial Neural Network (ANN) was also trained and validated using the experimental data which can be further used for work recovery prediction for a wider range of sub-critical temperature with CO_2 as working fluid.

Chapter 7: Conclusions and Future Scope of Work

Overall, it is concluded that, using IHX in the basic system configuration can improve the COP of the CO_2 refrigeration system at warm climate. Approach temperature has dominating effect on the overall performance of the system. Simultaneous optimization of the inter-stage pressure and gas-cooler pressure can potentially improve the COP of the multi-stage system. Minimum TEWI is found to be an effective way to decide the inter-stage pressure. It is observed that, an expander can potentially recover about 20% to 30% of energy consumed by the compressor. Implementation of CO_2 heat pump system in dairy industry appeared as a promising option to improve performance and waste minimization.

India has signed the Montreal Protocol along with various other developing countries in October 2016. Under this Protocol India is bond to phase out synthetic refrigerants by 2030. The success of sub-critical CO_2 system in European countries having cold ambient cannot be transplanted in India having climate pre-dominantly in the super-critical zone for CO_2. This require tremendous research focus within India. The commercial success of CO_2 system in India is a precursor to its wide spread use as a replacement for the synthetic refrigerant. So, maximum focus should be on improving the COP of trans-critical CO_2 system in future research.

- New and innovative multi-stage system configurations to be explored and evaluated for various applications at warm weather conditions.
- Two-phase flow characteristic inside scroll expander can be studied during trans-critical operation, in order to improve the design suitable for warm climate.
- Pilot implementation of CO_2 heat pump system in dairy industry for waste heat recovery to be explored.

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Trans-critical CO_2 System for Warm Climate: An Evaluation of System Configurations and Scroll Expander

THESIS

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Simarpreet Singh
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Under the Supervision of
Prof. M.S. Dasgupta

BIRLA INSTITUTE OF TECHNOLOGY & SCIENCE, PILANI

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CHAPTER 7

Conclusions and Future Scope of Work

The study covered theoretical analysis of trans-critical CO_2 systems both for refrigeration and heat pump applications. Specific example from a dairy industry is taken to evaluate both the applications in warm weather. Experimental investigation of a CO_2 scroll expander is carried out for work recovery.

This chapter compiles the overall conclusions drawn from the study. The chapter also incorporate recommendation for future scope of work.

7.1. Conclusions

Revival of carbon dioxide (CO_2) as a preferred working fluid for various Heating, Ventilation and Air Conditioning (HVAC) applications took place post 1990s due to enhanced awareness about direct or primary environmental impact like ozone depletion and greenhouse effect caused by the synthetic refrigerants when released into atmosphere due to leakage at various stages of its production and handling and also secondary impact of carbon release due to substantial share of power consumption by HVAC sector. Certain synthetic refrigerants also have harmful water and soil contaminating potential. While CO_2 is a biosphere gas having unit greenhouse effect. Conventional CO_2 refrigeration cycles, however, have lower COP, implying higher secondary impact on environment, more so for higher ambient operation. It is imperative, therefore, to probe various possible means of enhancing COP of CO_2 cycles, explore newer application areas as well as investigate road blocks implementation in warm climate.

7.1.1. Modeling and simulation

Energy and exergy analysis of five different evolving configurations of CO_2 systems were carried out for warm weather applications (35─50ºC), including application in single as well as multi-stage systems. The comparative study also brings out relative advantages of work recovery expander on the overall performance of the CO_2 system. A minimum TEWI (Total Equivalent warming impact) based criteria for the selection of intermediate pressure and temperature is also suggested. A semi-empirical model of a scroll work recovery expander was built for CO_2

refrigeration system. The model was validated in the trans-critical range from published experimental data. Major conclusions drawn from the simulation study are following:

- A cycle with IHX is found to have a lower mass flow rate at higher ambient temperature. This implies that, systems can be operated effectively at a lower gascooler pressure. Further, performance of system with IHX is found to be comparatively better at higher ambient temperature.
- For applications like refrigeration and air cooling, the multi-stage systems are found to have similar performance as the basic system. However, the FGI multi-stage system is found to have a comparatively better performance at all investigated gascooler outlet temperatures.
- The operating pressure and mass flow rate are found to be higher for multi-stage systems although, compressor discharge temperature is found to be lower. The inter cooler (IC) multi-stage system is found to have higher operating pressure; while this may be improved upon by utilizing more effective IC fluid than air.
- The effect of real time constraints like pressure drop in gascooler, compressor efficiency, approach temperature, expander efficiency, effectiveness of IHX and degree of superheat on the system performance reveals that the efficiency of compressor has dominating effect and efficiency of expander has minor effect of system's COP among the set.
- A combination of two low GWP fluids R1234yf / R744 are recommended for use in cascade system for low overall environment impact & high COP. Minimization of TEWI is recognized as an effective way to decide the IT for the cascade system.
- Effect of independent variation of condenser and evaporator temperature on COP_{HTC} and COP_{LTC} respectively are found to be high but effect on COP_{sys} is comparatively lower.
- Energy analysis of a scroll work recovery expander is carried out in the gas cooler outlet temperature range 32ºC to 48ºC for -10ºC evaporator temperature. Maximum 71% isentropic efficiency is achieved at pressure ratio ~3.7 based on a semi empirical model. Optimum range of pressure ratio recommended for maximizing the work recovery is in the range 3.3 to 4.2.
- A trans-critical CO_2 refrigeration system with work recovery scroll expander is found to have a high COP and low optimum discharge pressure. It is demonstrated that the

expander is capable of potentially recovery of about 20% to 30% of energy consumed by the compressor.

- Based on the year around data of ambient temperature at New Delhi, economic analysis reveals that the total PBP from a work recovery scroll expander of various capacity are within range 1 to 1.7 years. The same is minimum for a medium range capacity scroll expander. The PBP duration appears reasonable for initial investment.

7.1.2. Indian dairy industry- A case study

As a case study is developed to evaluate feasibility of CO_2 trans-critical system for warm weather conditions. Field data from a mid-sized milk & milk product handling plant located in the Northern Indian state of Punjab is taken for analysis. Year-round fluctuation of milk load, heating & cooling demand and ambient temperature are noted. Two schemes are conceptualized and evaluated, one for total replacement of the existing ammonia based refrigeration system with a CO_2 booster refrigeration system and another, a CO_2 heat pump system retrofitted above the ammonia refrigeration system for utilization of the waste heat. Year-round ambient temperature data suggests that a replacement plant will run under trans-critical condition for 26.3% of time and in sub-critical condition for 73.6% of time. Major conclusions drawn from a case study are as following:

- COP of the trans-critical CO_2 booster refrigeration system is found to be lower than the ammonia based refrigeration system, round the year. It is also observed that as ambient temperature increases further beyond critical temperature (of CO_2), the COP falls more drastically.
- Heat and work recovery are distinct possibilities for super-critical operation of CO_2 cycle.
- From exergy analysis, it is also revealed that the exergy destruction and its corresponding cost for trans-critical CO_2 booster refrigeration system is 3 times more than the NH_3 based refrigeration system during super-critical operation.
- Incorporation of a heat recovery system that is only 50% efficient and a work recovery expander that is 65% efficient in place of expansion valve can leads to potential COP improvement by about 30%.

- By employing the proposed scheme of a trans-critical CO_2 heat pump system, ~37 kWh of hear can be recovered coupled with saving of ~5000 liters/week of ground compare to the present consumption rate.
- While overall coal consumption is reduced 48% but total electricity consumption is increased by ~ 50%.
- Reduction in CO_2 emission achieved is about 45.7% and Net expenditure on energy is reduced by about 33.8%. These lead to PBP of 40 months for installation of the proposed trans-critical CO_2 heat pump system.

7.1.3. Experimental investigation of a scroll expander

Performance analysis of a scroll expander was carried out in an open loop test set-up for a limited range of pressure (sub-critical) available from a CO_2 cylinder. A back propagation Artificial Neural Network (ANN) was trained and validated using the experimental data which can be further used for prediction of work recovery for a wider range of sub-critical temperature with CO_2 as working fluid. Major conclusions from the study are as following:

- The maximum deviation between experimental and simulation based outcome for rotational speed and shaft work are within ±7.5%, and ±11.1% respectively.
- Work recovery system is found to be most effective in the pressure ratio range of 2 to 3.3. Maximum shaft work generation recorded is 2800 W at pressure ratio 3.2.
- An ANN model trained and validated for the whole range of sub-critical operation of CO_2 scroll expander is found to have good agreement with experimental data. This should be useful to easily assess the actual work recovery from a scroll expander while operating in a CO_2 refrigeration system during sub-critical operation.

7.2. Future Scope of Work

India has signed the Montreal Protocol along with various other developing countries in October 2016. A further drop in the intake and production of synthetic refrigerants on global perspectives is offered by the recent (December 2016) adoption of the Kigali Amendment to the Montreal Protocol. Under these Protocol, India is bond to phase out synthetic refrigerants by 2030. The success of sub-critical CO_2 system in European countries having cold ambient cannot be transplanted in India having climate pre-dominantly in the super-critical zone for CO_2. This require tremendous in-house research focus. The commercial success of CO_2 system in India is a precursor to its wide spread use as a replacement for the synthetic refrigerants. So, maximum focus ought to be on improving the COP of trans-critical CO_2 system in future research.

- New and innovative multi-stage system configurations to be explored and evaluated for various applications at warm weather conditions using work recovery expander, ejector or new technology.
- Two-phase flow characteristic inside scroll expander can be studied during trans-critical operation, in order to improve the design of the same that is more suitable for high pressure (warm climate) operation.
- Pilot implementation of CO_2 heat pump system in dairy industry for waste heat recovery to be explored.