DESIGN OF SMALL SCALE LIQUEFACTION CYCLE FOR NATURAL GAS

A thesis submitted in partial fulfillment of the requirements for the award of degree
of

Master of Technology in Marine Engineering and Management

by

Mr. NISHIT GUPTA
(Reg. No.1701215008)

under the guidance of

Mr. ARUN KISHORE ESWARA
(Associate Professor, IMU Kolkata Campus)

Department of Marine Engineering
Indian Maritime University Kolkata Campus

Kolkata - 700088.

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INDIAN MARITIME UNIVERSITY KOLKATA CAMPUS

Department of Marine Engineering

CERTIFICATE

This is to certify that the thesis entitled **"DESIGN OF SMALL SCALE LIQUEFACTION CYCLE FOR NATURAL GAS"** submitted by Nishit Gupta to Indian Maritime University Kolkata Campus for the award of the degree in Master of Technology in Marine Engineering and Management, is a bonafide record of the project work carried out by him under our supervision. The contents of this thesis, in full or in parts have not been submitted to any other institute or University for the award of any degree or diploma.

The Project has been carried out at Indian Maritime University Kolkata Campus.

Mr. A.K.Eswara
Project Guide
Indian Maritime University
Kolkata Campus

Dr. K.Mitra
Academic Co-ordinator
Indian Maritime University.
Kolkata Campus.

External Examiner
(H. R. KHARE)

EVALUATION SHEET

Name of the candidate	Nishit Gupta
Title of the project	DESIGN OF SMALL SCALE LIQUEFACTION CYCLE FOR NATURAL GAS
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This dissertation is approved by the Board of Examiners

External Examiner : (H. R. KHARE)

Internal Examiner :

INDIAN MARITIME UNIVERSITY
KOLKATA CAMPUS

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Nishit Gupta
Nishit Gupta

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ABSTRACT

Liquefied natural gas is found to be the most economical mode of transportation for distances covering more than 3500 miles. The boiling point of natural gas is 111.7 K at atmospheric pressure and falls under the category of cryogenics. The process has components such as compressor, heat exchanger, expansion valve, insulating material, storage tank and pipes. The liquefaction of natural gas is achieved by processing natural gas in the liquefaction cycle. There are many parameters affecting the cycle such as compressor efficiency, heat exchanger effectiveness, ambient temperature, friction losses in pipes and insulating materials.

In this thesis, some of the above mentioned parameters are considered while some parameters are neglected or assumed appropriately. It is observed that out of the existing cycles such as simple Linde-Hampson cycle, Pre-cooled cycle, Claude cycle and Kapitza cycle, each cycle has its own benefits and drawbacks. The fraction of liquefaction is found maximum for simple Claude cycle while the work required also reduces due to expansion of the high pressure gas through reciprocating expansion engine. The iterative procedure to find the configuration of each cycle is explained in the thesis and can be used with minimum bare inputs. An experimental setup can be built through these configurations and better study can be performed. To obtain the results, residue in the iterations is taken to 0.1 for temperature, pressure and mass flow rate both. In the Claude cycle, for mass flow rate of 1.02 kg/sec and the pressure ratio of 40, the fraction of liquefaction is found 0.0646 while in the kapitza cycle, for mass flow rate of 1.35kg/sec and the pressure ratio of 40, the fraction of liquefaction is found as 0.040. It is found that, the pressure required to liquefy the gas is not practical in the case of Linde-Hampson cycle. Further, it is found that Claude and Kapitza cycle can be used for experimental purpose.

List of Symbols and Abbreviation

Tcf-trillion cubic feet
MMBTU-million BTU
MTPA- million tonne per annum
W_c- Work done by the compressor
m_1- Inlet mass to the compressor
FOM- Figure of Merit
y - Fraction of total gas liquefied
m_f - Mass of gas liquefied
h_f - Enthalpy of the liquid methane
FOM - Figure of Merit
W_i- Ideal work required to liquefy gas
W_l - Actual work for liquefaction process
LNG- Liquefied Natural Gas
J-T –Joule Thomson
PR- Pressure Ratio
HX- Heat Exchanger

CHAPTER 1

INTRODUCTION

1.1 Cryogenic

The word cryogenics has its origin in the Greek language where "kryos" means frost or cold and "gen" is a common root for the English verb to generate. It is also defined as the science and technology of temperatures below 123 K. Major applications of cryogenics technology are liquefaction of gases, storage and transport of gases, separation of gases, biological and medical applications, alteration in material properties by reduced temperature etc.

Natural gas has remained the fastest growing energy resource in most regions of the world for more than two decades, driven by the low greenhouse gas emissions as well as high conversion efficiency in power generation. For almost a century, natural gas has been transported safely, reliably, and economically via pipelines. Pipelines proved to be ideally suited to the supply and market conditions of the twentieth century, when large reservoirs of conventional natural gas are found in accessible locations. Pipelines provided the stability and security of supply and continue to do so where large accessible gas reserves remain. However, over the past decades it has become clear that significant quantities of new gas reserves are not so conveniently located. Attention has shifted to more isolated large gas reservoirs that were previously thought to be too remote, or technically too difficult and costly to develop. A number of solutions for exploiting stranded gas reserves are currently being developed and considered for commercialization. On the other hand, over the past three decades, only the liquefied natural gas (LNG) industry has successfully brought many large remote gas fields to the gas markets that are unreachable by pipeline (e.g., Japan, South Korea). Today, the LNG supply chains have diversified and introduced competition into markets previously "captured" by pipeline gas suppliers, and have improved the security of energy supply of many consuming nations and reduced the geopolitical and political constraints on global gas supply.

1.2 Modes of transportation

Natural gas can be transported through 5 following modes:

1. Pipeline

2. LNG
3. GTL
4. Gas to Wire-HVDC
5. CNG
6. Gas to solid hydrates

Pipeline, CNG and LNG are fully commercial and mature technologies. GTL and gas to wire-HVDC are commercial methods but require technology developments to improve efficiency. Gas to solid hydrates is under R&D stage only.

Selection of existing technologies is dependent on the distance to consumer markets and gas field production rates. Currently stranded gas is transported via two established technologies: pipeline and LNG.

The volume reduction is 1/600th (approx.) of the volume of natural gas at the burner tip. LNG becomes competitive to pipelines in long distance transport. For long distance transport, the gas pipeline option would require large diameter pipes and gas recompression facilities to overcome the transmission pressure drop, which are very costly. It is expensive to construct pipelines in the ocean and hence LNG is used to transport natural gas. The cost of transporting gas via pipelines and LNG are compared in the Figure 1.

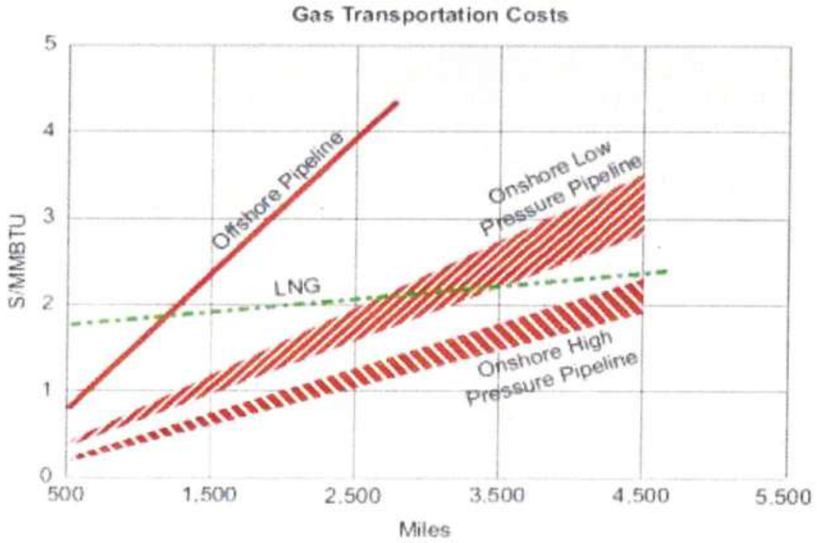


Figure 1: Comparison of the cost of transporting gas via pipeline and LNG; for 1 tcf/yr and including re-gasification costs [32].

1.3 Basic properties

The basic properties of LNG begin with information of chemical and physical properties which is pre-requisite for assessing potential LNG safety hazards and risks. The properties of LNG varies with composition, which depends on reservoir source and its fractional distillation process. While LNG is predominately methane (about 87 mole % to 99 mole %), its composition also includes other higher hydrocarbons, typically, the C2 to C4 and heavier, nitrogen and trace amounts of sulfur (less than 4 ppmv), and CO2 (50 ppmv; see Table 1).

Table I: Typical composition of LNG from various Liquefaction Plants

Component (Mole %)	Nigeria LNG	Arun LNG	Brunei LNG	Oman LNG	Atlantic LNG	Kenai LNG
Methane	87.9	88.48	89.4	90	95	99.8
Ethane	5.5	8.36	6.3	6.35	4.6	0.1
Propane	4	1.56	2.8	0.15	0.38	0
Butane	2.5	1.56	1.3	2.5	0	0
Nitrogen	0.1	0.04	0.2	1	0.02	0.1

(ILEX Energy Consulting, 2003)

The byproducts of the combustion do not contain any Sulphur dioxides and very less nitrogen oxides, which makes it a clean source of energy. LNG is an odorless, colorless, and noncorrosive cryogenic liquid at normal atmospheric pressure.

LNG is nontoxic. However, as with any gaseous materials, natural gas release from LNG can cause asphyxiation due to lack of oxygen in an unventilated, confined area, and can be ignited if mixed with the right concentrations of air.

The boiling point of LNG varies with the composition but is typically said to be -162°C. The density of LNG is 430 kg/m^3 to 470 kg/m^3, which is less than half of the density of water. LNG being much lighter than water, if spilled floats on water. LNG being heavier than air stays near the ground. As the surrounding heat is transferred to the gas it warms up and at the temperature of -166°F the vapour becomes buoyant since the vapours are lighter than air. A release of LNG in an enclosed space will displace the air making breathing more difficult.

Vapours released from LNG, mix with air and create vapour cloud. The cloud may be flammable and explosive. The flammability limit is 5 percent to 15 percent by volume in air. Outside this limit methane air mixture is not flammable.

When fuel concentration exceeds upper limit, it cannot burn due to little oxygen present. When the fuel concentration is below lower limit, it cannot burn due to very rich air fuel ratio. So the LNG handling should be done in well-ventilated area [33].

1.4 Units and conversion factors

LNG production is commonly expressed in million metric tons per year. The units often used are MMTPA, MTPA, tpy, or a more SI conforming expression Mt/a, which stands for mega but can also be read as million tons per annum.

1.5 Traditional LNG supply chain

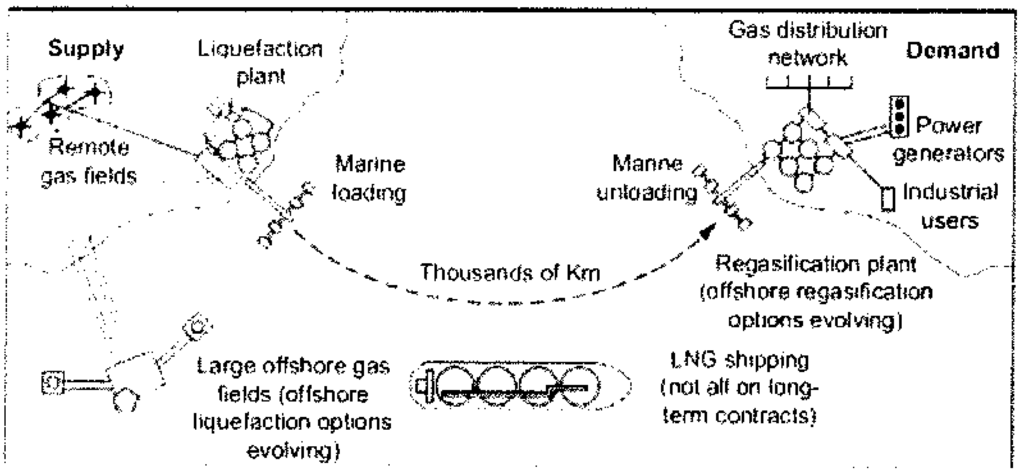


Figure 2: key elements of traditional LNG supply chain[34].

The major components of the traditional LNG supply chain, including pipeline connections between the stages, are shown in Figure 2.

1.6 Odourization

Since natural gas is colourless and odourless, a leak is difficult to detect without a proper instrument. So odourization of regassified natural gas is a must. It is must before distributing gas to consumers. The properties of odourant depend on the requirement of the local regulations.

1.7 LNG quality

WOOBE INDEX-Woobe Index is used in the gas turbine industry for fuel gas specification and gas turbine performance guarantee. It is also one of the specifications for gas interchangeability required for pipeline distribution. The index is calculated by dividing the higher heating value of the gas by the square root of the gas density or MW (molecular weight) relative to air.

In the United States and Western Europe, most of the LPG components are extracted for sales as liquids, and the residual gas is sold to consumer pipeline; consequently, the gas compositions and heating values do not vary very much. However, as shown in Figure 3, the LNG compositions and heating values vary significantly among different export terminals, and there are concerns regarding the gas interchangeability with the local gas contents (see Figure 3).

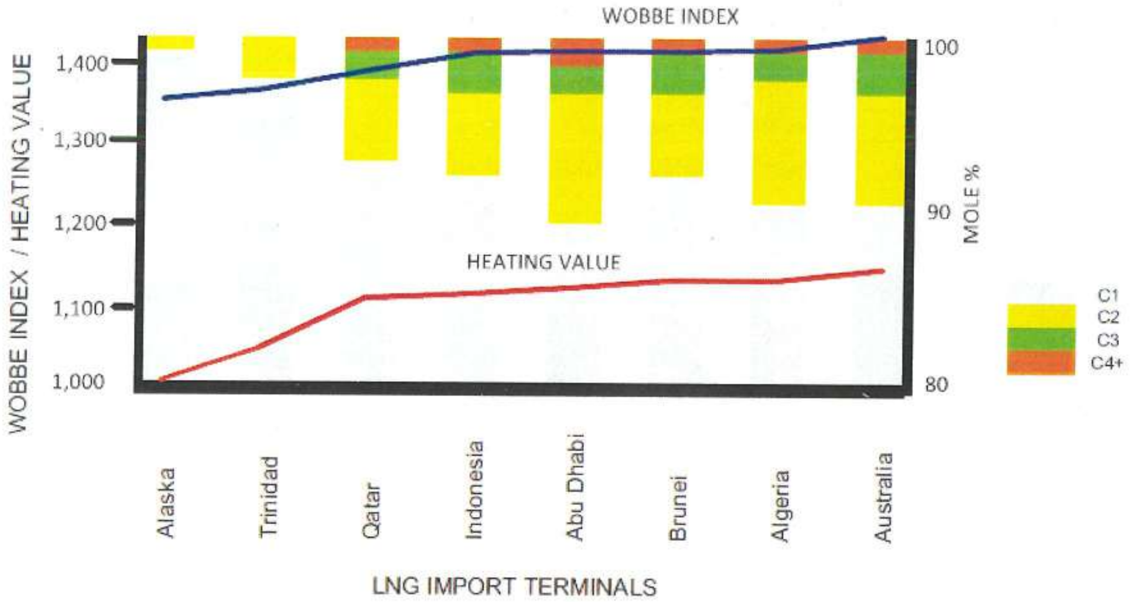


Figure 3: Composition and higher heating values of various LNG sources [35].

Woobe index control- The common approach to reduce the Btu value and Wobbe Index of a gas is by diluting the vaporized LNG with nitrogen, up to the pipeline limit for inert content, usually 2 to 3%. Nitrogen is an effective medium for lowering the heating value of the pipeline gas. The addition of nitrogen also increases the molecular weight of the gas mixture that further lowers the Wobbe Index value. For example, for California pipeline gas, the gas specification must meet the 1360 Wobbe Index specification, which is the specification of existing combustion equipment and power generation stations. To meet the California Woobe Index specification, Alaska LNG is the only gas that meets the low Wobbe Index specification without nitrogen dilution. For the Trinidad LNG, about 1 mole % nitrogen is required. For other LNGs, because of the maximum 3 mole %

nitrogen limit, less than 40% of the LNG sources would meet the California specification, as shown in Figure 4.

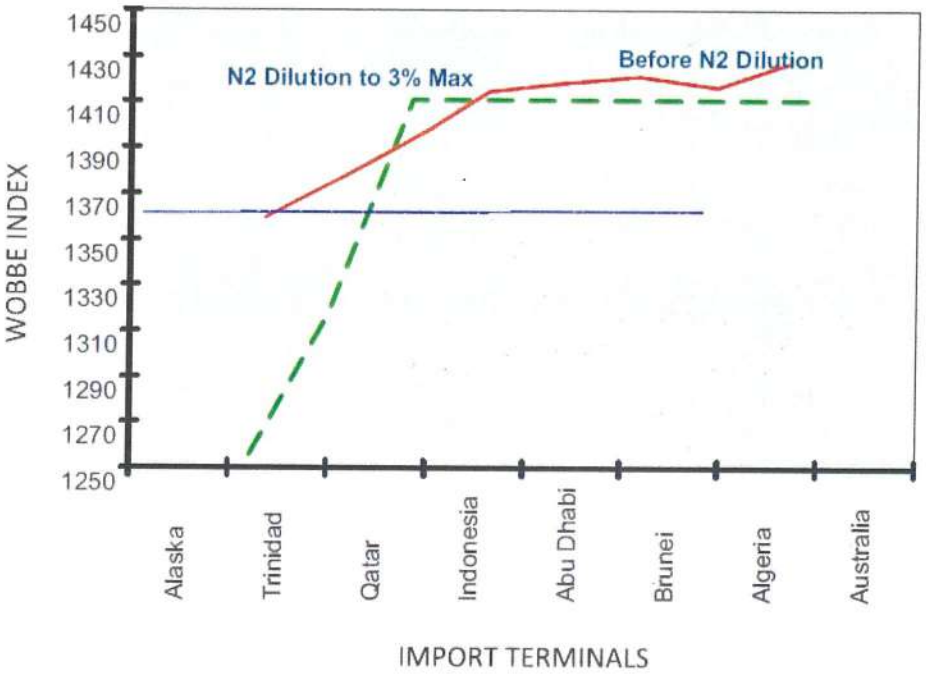


Figure 4: Nitrogen dilution for different LNG sources [35].

1.8 History of LNG

Although the LNG industry is perceived to have developed rapidly, it is based on quite old foundations. Some key development milestones include:

- Michael Faraday liquefied methane in 1820
- Karl von Linde developed compressor refrigeration machine in 1873
- First LNG plant built in 1912 in West Virginia, began operation in 1917
- Patent filed for LNG in 1914
- First commercial liquefaction plant built in 1941 in Cleveland, Ohio
- First LNG disaster in 1944 in Cleveland, Ohio
- First LNG tanker, the Methane Pioneer, carried first cargo of 7,000 bbl. LNG in 1959 from Louisiana to UK
- First export / import trade in 1964 between Algeria and UK

- One of the longest continuously operated LNG plants at Kenai, Alaska, started in 1969 and shut down in 2013
- LNG is a demand driven industry and demand is determined by competing energy sources, mainly oil and coal
- From the start of the international trade in the 1960s, LNG demand reached 50 MTPA in 1990, then 100 MTPA in 2000, and 240 MTPA in 2012
- Production is expected to approximately double between 2012 and 2030

1.9 Objective

There are various methods for transportation of natural gas as discussed in the previous sections. The selection of the modes depends on the transportation distances and location of natural gas reserves. The boiling point of natural gas is well below room temperature which makes the process of liquefaction quite work consuming. There are many existing industrial plants available producing millions of metric tonnes per year of natural gas. Before constructing an experimental setup, design calculations should be done to support the construction. Design undergoes a spiral procedure and the needs to be iterated to remove any errors took in the assumptions. The process is initially considered to be reversible and devices/ machines to have hundred percent efficiency. As the design process moves forward these assumptions are reduced and better assumptions are done.

The primary objective of this thesis is to design a small scale liquefaction plant to liquefy natural gas.

1.10 Motivation

Reference number [34] shows that the difficulty in transporting the stranded gas from a remote gas reserves. The cost of the transportation increases drastically for offshore pipelines compared to onshore pipelines. The increased cost can be handled by transporting the gases in liquefied form through LNG tanker. Since we know that natural gas is liquefied at a temperature close to 110K, I was intimidated and decided to study the process, equipments and liquefaction cycles. Hence I decided to design the liquefaction plant for 1 tonne of liquefaction per day.

1.11 Project description and goal

The transportation of natural gas through pipelines from offshore reserves is costly as compared to any other existing mode of transportation. So the gas is liquefied which reduces the volume by 1/600th time and makes the transportation easy and cost effective. To study the different aspects of this cryogenic plant, the design was done. The existing design process or cycles are studied. The cycles majorly consist of compressor, heat exchanger, expansion devices and pipe related work. The basic concept for design of liquefaction cycle is reaching to its liquefaction point. The most of the work is done on the T-s diagram and different processes are shown on it. To liquefy the gas initially the existing cryogenic liquefaction cycle are used such as Linde-Hampson cycle, Pre-cooled Linde-Hampson cycle and Simple Claude cycle. Obtaining the compressor output pressure, temperature achieved after cooling the gas in the heat exchangers and liquefaction fraction of gas along with the work requirement for the process are done in this thesis.

CHAPTER 2

LITERATURE REVIEW

[1] **Sarangi** ***et al.*** (1992) presented an analysis of the thermodynamic cycles commonly used for the liquefaction of hydrogen in order to predict their performance under a given set of operating conditions and component efficiencies. The cycles considered are: (i) precooled Linde-Hampson cycle, (ii) Claude cycle and (iii) Helium- Hydrogen condensing cycle. Extensive calculations of liquid yield and specific work requirement have led to the optimization of operating conditions such as pressure levels, expander mass fraction, etc. The optimum operating conditions also depend on the heat exchanger effectiveness, temperature of pre-coolant baths and number of ortho-para converters.

[2] **(Atrey, 1998)** gave a thermodynamic analysis of the Collins helium liquefaction cycle with two reciprocating expanders. The result from this paper makes it clear that there exists an optimum mass flow fraction to expander for given efficiency of expander and effectiveness of heat exchanger. The analysis quantitatively studies the effect of expander efficiency and heat exchanger effectiveness on the performance of the liquefier. It gives final steady state temperature distribution across the cycle, which is essential data for carrying out the preliminary design of various components in the cycle.

[3] **Veziroglu** ***et al.*** **(2000)** carried out a parametric analysis and optimal decision rules pertaining to system component selection and design were reached. In this work it was found that helium refrigerated hydrogen liquefier is economically more feasible than hydrogen refrigerated hydrogen liquifer.

[4] **Foerg (2001)** has carried out a review on the history of cryogenics: the epoch of the pioneers from the beginning to the year 1911. In this paper history of liquefaction of many gases is described and major contribution by various scientists in this field. Modern science has reached temperatures only one-millionth of a degree above absolute zero, but absolute zero itself cannot be attained.

[5] **Marc *et al.* (2001)** discussed the relations between the thermodynamic property exergy and energy, environmental impact and sustainable development. Historical and modern examples are utilized to demonstrate these concepts. Three main relations between exergy and environmental impact are extensively discussed in terms of order destruction and chaos creation, resource degradation, and waste exergy emissions. It is felt that the potential usefulness of exergy analysis in addressing and solving energy-related sustainable development and environmental problems is substantial, and that exergy is a confluence of energy, environment, and sustainable development.

[6] **Sheffield *et al.* (2001)** reported on second law analyses of a hydrogen liquefier operating on the modified Collins cycle. Two different modifications employing the cycle in question were attempted:

- A helium-refrigerated hydrogen liquefaction system.
- A hydrogen-refrigerated hydrogen liquefaction system.

Analyses were carried out in order to identify potential areas of development and efficiency improvement. A computer code capable of computing system and component efficiencies; exergy losses; and optimum number and operating conditions of compressors, expanders, after coolers, intercoolers, and Joule–Thomson valves was developed. Evaluation of the thermodynamic and transport properties of hydrogen at different temperature levels was achieved by employing a hydrogen property code developed by researchers at the National Bureau of Standards (currently NIST). A parametric analysis was carried out and optimal decision rules pertaining to system component selection and design were reached. Economic analyses were also reported for both systems and indicated that the helium-refrigerated hydrogen liquefier is more economically feasible than the hydrogen-refrigerated hydrogen liquefier.

[7] **Mafi M. *et al.* (2007)** performed an exergy analysis of the multistage cascade low temperature refrigeration system used in olefin plant. The equations of exergy efficiency and exergy destruction for each component of refrigeration system are developed. The relations for the total exergy destruction and overall exergetic efficiency of cascade refrigeration system

are obtained. Also, an expression for minimum work requirement for cascade low temperature refrigeration used in olefin plants is developed. The exergy analysis results on the refrigeration system indicate that the major irreversibility is due to losses within the compression system, driving forces across the heat exchangers, and losses due to refrigerant letdown. The overall exergetic efficiency of the cascade refrigeration system is determined to be 30.88% indicating a great potential for improvements. In this paper, two practical ways including pinch and exergy analysis based method and mathematical programming based method have been used to improve the overall exegetic efficiency of the refrigeration system through decreasing the temperature difference between the process and refrigerant streams in heat exchangers. A mixed refrigerant cycle was developed for olefin plant analyzed in this work utilizing a mixture of methane, ethane, propane and nitrogen as cycle working fluid to replace the pure ethylene refrigeration cycle which is used in conjunction with propylene refrigeration cycle. The results show that mixed refrigerant cycle can improve thermodynamic performance of refrigeration system in the case of using optimal mixture composition and optimal operating pressures. Each mixed refrigerant cycle configuration has own optimal mixture composition and optimal compressor power consumption. We suggest that there is an opportunity for future work to extend a methodology to find the optimal cycle configuration for mixed refrigerant to replace the pure ethylene refrigeration cycle for minimizing shaft work of cascade refrigeration system.

[8] **Berstad *et al.* (2009)** explained comparison criteria for large-scale hydrogen liquefaction processes. Direct comparison of liquefiers based on overall exergy efficiency and specific power consumption will favors those with a higher portion of pre-compression. A methodology for comparing hydrogen liquefaction processes that compensates for non-uniformity in feed specifications has been developed and applied to three different hydrogen liquefiers.

[9] **Sarangi *et al.* (2009)** Simulation of cryogenic cycles was done using the commercial software ASPEN HYSYS. In the present work Cryogenic cycles like Linde cycle was used for liquefaction of air which yields only 5 % of liquid air at the end of the process. To increase the productivity one

more modified cycle was used know as Claude cycle where the liquefaction rate increases to 27 %. Also the work was extended to study Claude with different adiabatic efficiency condition of the expander and optimum condition for the maximum liquid output was found out. Detailed graphical representations of various cases were also done. This project work simulates a Linde cycle, Claude cycle using the simulation tool ASPEN HYSYS. The results obtained from simulation will helps to carry out experiments in lab at optimum condition and in later stages comparison can be made. Much complex cycles like the Kaptiza cycle, Heylandt cycle , Collins liquefaction cycle for liquefaction of Nitrogen can also be simulated using the same ASPEN HYSYS.

[10] **Prasad (2009)** presented a cycle simulation for the nitrogen liquefaction cycle with a compressor, heat exchanger and a J-T valve. It gives us the design data in terms of nodal temperature across the heat exchanger, compressor, chiller etc. and mass flow rates through all the equipment. The above simulation work for Claude liquefaction cycle for nitrogen eliminate time and cost expenditure by successfully proven that for low working pressure for Claude system it is useless to use last or low temperature heat exchanger as well as for high working pressure of Claude cycle first heat exchanger is worth. The simulation can be adapted to bring about any changes in the configuration of the liquefaction cycle and can be successfully applied for other complicated cycle.

[11] **Chowdhury (2009)** carried out the process design using the standard calculation procedure and is validated by using process simulation software, ASPEN HYSYS. The preliminary data required in terms of mass flow rate, pressure and temperatures across heat exchanger, turbo expander and other components of the nitrogen liquefier are found out. Parametric study is carried out to study the role of the different component efficiencies in deciding overall system efficiency. It is found that the liquid yield is directly proportional to the effectiveness of heat exchanger, efficiency of turbo expander and mass fraction diverted through turbo expander. However, on the limit, no yield condition prevails if effectiveness of heat exchangers is less than 0.88 and mass fraction through the turbo expander exceeds 0.95. The effect of increase in pinch point deteriorates the yield. Though higher turbine efficiency favors the yield, the turbine

available for the purpose is limited to 50% efficiency and it limits the yield to only about 4%. The simulation done and the analysis carried out can serve as guide lines for the development of nitrogen liquefier.

[12] **Chowdhury *et al.* (2011)** with the help of a commercial process simulator Aspen HYSYS, the exergy analysis of a helium liquefaction system based on modified Claude cycle is performed. The major conclusions arising out of the study which may be used for the design of large-scale helium liquefaction cycles are given below:

- If, during the off-design conditions, the cycle is operated within a range of cycle pressure ratio, the performance would not deteriorate. For the selected cycle configuration and operating conditions, the range is PR = 10–18, (the compressor suction pressure is atmospheric).
- In the JT-heat exchanger, when pressure increases, the imbalance due to the mismatch in mass flow between the forward and return streams gets compensated by their specific heat imbalance. Exergetic efficiency of the heat exchanger at the lowest temperature of a helium liquefier can be improved by increasing the pressure ratio because the mass imbalance gets compensated by the specific heat imbalance.
- While designing the cycle, owing to their lower exergetic efficiencies, additional care should be taken for ensuring superior heat transfer performance by the high temperature heat exchanger HX1 and the lowest temperature heat exchanger HX5.
- As the first expander flow increases, there is a sharp increase in exergy losses in the high temperature stage. Adding more stages to the cycle may decrease the overall exergy loss.
- There is an optimum flow fraction that has to be passed through each expander to get maximum performance for the cycle. It is also found that the cycle performance is more sensitive to variations in pressure ratio when the expander flows (both the expanders) exceeds their optimum values. For the selected configuration and operating conditions the total optimum expander flow is found to be 80% of the total flow through the cycle.

[13] **Chowdhury *et al.* (2011)** has chosen expander as the subject matter of analyses in the present study. As the sensible cold of helium vapor is lost

in liquefiers, the expanders in liquefaction cycles have to provide more refrigeration than those in refrigeration cycles.

The expander parameters such as rate of mass flow, operating pressure, inlet temperature, etc. are interdependent, and hence, it is difficult to predict the system behavior with variation of a particular parameter. This necessitates the use of process simulators. Parametric studies have been performed on Collins helium liquefaction cycle using ASPEN HYSYS. Collins cycle has all the basic characteristics of a large scale helium liquefier and the results of this study may be extrapolated to understand the behavior of large scale helium liquefiers. The study shows that the maximum liquid production is obtained when 80% of the compressor flow is diverted through the expanders and it is equally distributed between the two expanders. The relationships between the liquid production and the isentropic efficiency of expanders are almost linear and both the higher and lower temperature expanders exhibit similar trends.

[14] **Thomas *et al.* (2011)** with the help of a commercial process simulator Aspen HYSYS V7.0, a parametric study on the effects of variation of heat exchanger performance on Collins helium liquefaction cycle has been performed. The major conclusions are:

- Heat exchanger effectiveness has a linear relationship with liquid production.
- The performances of those heat exchangers that determine the inlet temperature to expanders have more influence on the liquid production.
- Maximum benefit in liquefaction is obtained when the available additional heat transfer surface area is distributed among the heat exchangers in proportion to the base UA. This leads to equal effectiveness for all heat exchangers, though the value increases or decreases from the base value depending on whether the heat transfer area is increases or decreased.
- Though a Collins cycle with heat exchangers having limiting or saturation UAs generate maximum liquid, slight degradation of their UAs do not affect liquid production drastically. In fact only with degradation beyond 50% of limiting UAs (all the heat exchangers

- have effectiveness values above 0.95 at this level), the losses in the heat exchangers sharply increase and decrease in liquid production becomes significant.
- There is no impact of the variation of effectiveness of the heat exchangers on the optimum flow that has to be diverted through expanders.

[15] **Ghosh *et al.* (2011)** the following conclusions may be drawn from the author's exergy analysis:

- Compressor discharge pressure should be kept between 18 and 22 bar for helium liquefaction cycles.
- 80% of the compressor flow should be diverted through the expanders for highest performance of helium liquefiers.
- Total expander flow should be equally divided among the expanders to obtain maximum liquefaction.
- Higher efficiencies should be sought for all the expanders irrespective of their operating temperatures.

[16] **Pacio *et al.* (2011)** presented a review on heat exchanger thermal hydraulic models for cryogenic applications, different types of heat exchangers used in cryogenics, their application and advantages. In this work a systematic review of the state of art and challenges in modeling cryogenic heat exchangers is presented. They include lumped parameters, distributed parameters and stream-evolution models.

2.1 Summary of Literature Review

There are many literatures available in the field of cryogenics, which are used to conduct further research and development. The prime person in the field of cryo-cooling is Carl Paul Gottfried Linde, a German scientist, who discovered refrigeration cycle and invented the first industrial scale air separation and gas liquefaction processes. His patents lead to the field of cryogenics. The further work was carried by Nobel Prize winner in physics, "Heike Kamerlingh Onnes".

Through the literature I found various existing liquefaction cycle such Linde-Hampson cycle, Pre-cooled Linde- Hampson cycle, Claude cycle, Kapitza cycle and others. The design configurations of these cycles are not openly available and needs to be configured from scratch. However, there is lot of work been to find the optimum conditions in the cycle. The cycle's effectiveness depends on the mass flow rate through the expander in case of Claude cycle and Kapitza cycle. The optimum range is 60-70% of the total mass flow rate.

The cryogenic liquefaction such as Linde cycle yields only 5% of the liquid air at the end of the process, while advanced cycle such as Claude cycle yields up-to 27% of the liquid air. Also the most important device in the cryogenic cycle is the Heat-Exchanger. The effective should be as high as 0.88 and the turbo-expander mass flow can be as high as 0.95.

CHAPTER 3

METHODOLOGY

3.1 Project Execution Stage

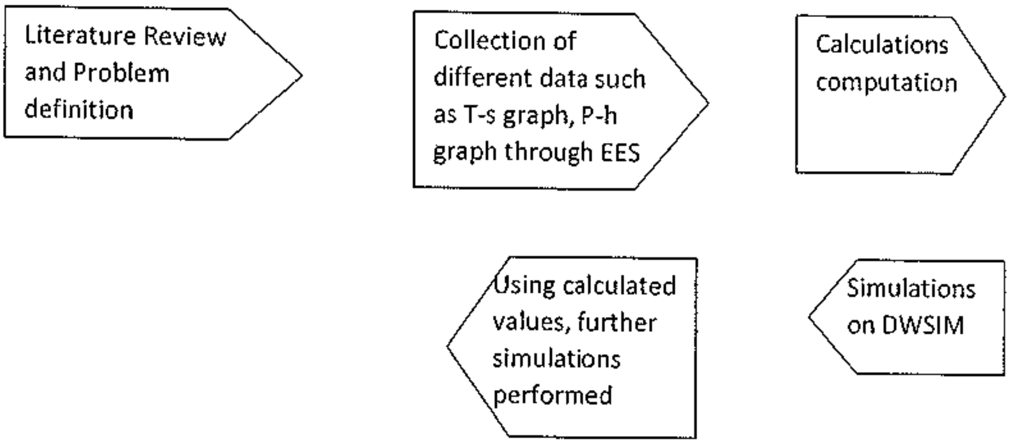


Figure 5: Stages of project

3.2 Designing process

Designing is a spiral process, which makes it repetitive.

INPUT TO DESIGN

- Previous LNG reference of similar type
- Previous work loops and shortcomings
- R&D related work

BASIC DESIGN SPIRAL

- Define the problem
- Gathering of data to work out the problem
- Assuming the data intelligently
- Solving the problem by using the available data
- Using the result remove the assumed data and solve the problem again
- Performing iterations until desired residue is reached

3.3 Technical Specification.

3.3.1 Cryogenics

The production of liquefied natural gas comes under cryogenics. The definition of cryogenics may vary from regions to regions and organizations to organizations. The temperature is not well defined; but scientist assume a gas to be cryogenic if it can be liquefied at or below -150°C.

The U.S. National Institute of Standards and Technology has chosen to consider the field of cryogenics as that involving temperatures below -180°C or 93 K. This is logical dividing line, since the normal boiling point of the permanent gases such as helium, hydrogen, neon, nitrogen, oxygen and normal air lie below -180°C while the Freon refrigerants have the boiling points above -180°C.

3.3.2 Cryogenics Fluids

Some of the cryogenic fluids with their boiling point in Kelvin are listed below.

Table II: Cryogenic liquid and their boiling point

Fluid	Boiling Point (K)
Helium -3	3.19
Helium-4	4.214
Hydrogen	20.27
Neon	27.09
Nitrogen	77.09
Air	78.8
Fluorine	85.24
Argon	87.24
Oxygen	90.18
Methane	111.7

3.3.3 Application of Cryogenic Heat Transfer

The importance of heat transfer (or the reduction of heat transfer) in cryogenic systems is illustrated by the variety of its applications. Some representative applications are discussed as follows:

1. *Cryogenic fluid storage vessels* (*Dewars*). In Dewars designed for land-based service, the rate of heat transfer through the insulation must be as small as practical. Cryogenic insulations are usually evacuated to minimize gaseous conduction. Floating radiation shields (i.e., MLI) are used to minimize radiation heat transfer and low-thermal-conductivity spacers are used to minimize conduction through the solid. The suspension system used to support the inner vessel within the outer vessel must minimize the heat conducted through the support, while carrying the load associated with the inner vessel and its contents. High-strength stainless steel rods are often used for the vessel suspension system because stainless steel has a low thermal conductivity.

In Dewars designed for space applications, the inner vessel may be shielded by both actively cooled radiation shields and vapor-cooled radiation shields. Radiant heat transfer is intercepted by the shields and transferred out of the system to minimize the evaporation rate of the cryogenic liquid. Graphite–epoxy and glass–epoxy composite straps are used to support the inner vessel within the outer vessel.

2. *Superconducting magnet energy storage systems.* Heat transfer along the current leads into superconducting magnet systems may be detrimental to the operation of the magnet refrigeration system because of the associated thermal load. Special attention has, therefore, been directed toward the design of low-heat-transfer electrical feed-throughs or thermal intercept devices for these systems (Niemann et al. 1994).

3. *Superconducting power transmission lines.* Commercial development of superconducting electrical cables began in the late 1960s after the Nb–Ti and Nb3Sn alloys became available in large-scale lengths (Rogalla and Kes 2012). Cables constructed from these so-called *low-temperature superconductors* (*LTS*) were cooled to temperatures on the order of 4 K (−452°F) using liquid helium. Heat transfer to the liquid helium coolant caused excessive evaporation of the LHe; this problem was solved by shielding the helium circuits with liquid-nitrogen cooled thermal shields. At present, the LTS cable is not economically competitive with conventional ambient-temperature resistive power cables except possibly in applications where a large quantity of electrical power is to be delivered to a

relatively small area. Since the discovery of the so-called *high-temperature superconductors* (*HTS*), renewed interest has developed in superconducting energy transmission systems. The HTS cables must be cooled to cryogenic temperatures; however, these materials operate at temperatures around 80 K or −315°F and can therefore use relatively inexpensive liquid nitrogen as the coolant. The HTS cables are usually covered (clad) with a high-thermal-conductivity material, such as copper, in order to reduce the severity of "fault current surges" where a portion of the cable momentarily changes from superconducting to normal behavior causing the electrical current to suddenly increase from 5 kA to as high as 100 kA for a short period of time.

4. *Air separation systems.* High-performance heat exchangers are essential to the air separation systems that are used to produce liquid nitrogen, liquid oxygen, and liquid argon. The effectiveness of these heat exchangers must be approximately 95% or higher for the system to operate in an economical manner. The recuperative heat exchangers allow the incoming warm air stream to be precooled by the outgoing cold gas stream, reducing the need for external refrigeration. The cold heat exchangers must be insulated to reduce heat transfer from ambient temperature surroundings. The distillation column in an air separation system involves simultaneous heat and mass transfer between liquid and vapor phases (Grenier and Petit 1986).

5. *Natural gas liquefaction systems.* Since the Hope Natural Gas Company built the first liquefied natural gas (LNG) peak shaving plant in 1941, there have been several projects designed and constructed for this purpose (Culbertson and Horn 1970). Some municipalities have also converted their public transportation vehicles so that they are fueled by LNG, increasing the demand and dependence. Because of the high mass flow rates and high effectiveness requirements, the LNG heat exchangers in large-scale LNG liquefaction plants tend to be quite large. For example, heat exchangers built for an LNG plant in Tobruk, Libya, used 160 km (660 miles) of heat exchanger tubing wrapped around a vertical mandrel having a height of 58 m (190 ft), which is approximately the height of a 16-floor building.

6. *MEMS (microelectromechanical systems) coolers.* The reliability and electronic noise characteristics of many miniature semiconductor electronic devices are

improved when the device is cooled to cryogenic temperatures. Micro-miniature Joule– Thomson refrigerators have been developed to achieve cooling over very small areas (spot cooling) for these devices (Little 1990; Maytal and Pfotenhauer 2013). These refrigerators involve heat exchangers with flow channels that are approximately 200 μm (0.008 in.) wide × 30 μm (0.001 in.) deep. The thermal design of these heat exchangers presents a unique challenge because the governing heat transfer relationships must be modified when the channel dimensions are on the order of the mean free path for the flowing fluid (Tzou 1996).

7. *Aerospace systems.* The duration of aerospace missions may be as long as 5 years. Therefore, any refrigerant liquid, such as liquid helium, that is carried onboard for cooling instrumentation and other equipment must be thermally protected. The liquid may be surrounded by thermal shields that are actively cooled by Stirling cycle refrigerators or thermoelectric coolers in order to maintain an extremely low heat transfer rate to the coolant and ensure adequate mission duration (Lee et al. 1992; Nieczkoski et al. 1992).

8. *Cryosurgery systems.* One surgical technique that is used in practically all surgical specialties (e.g., neurosurgery, ophthalmological surgery, and gynecological surgery) involves freezing a small region of the defective tissue in order to cause the destruction of the offending material (Cooper 1971; Rubinsky 2000). The devices that are used to affect the cooling process involve either cryogens or small cryo-coolers and the thermal analysis of the tissue freezing process is critical in order to predict and control the extent of the frozen lesion (Skye et al. 2012).

3.3.4 Properties of some cryogens

Table III: Selected properties of some cryogenic liquids at their NBP

Liquid	Normal Boiling Point K	Density kg/m3	Latent Heat kJ/kg	Specific Heat kJ/kg-K	Viscosity mPa-s	Thermal Conductivity W/m-K
Helium-3	3.19	58.9	8.49	4.61	0.00162	0.0171
Helium-4	4.222	124.9	20.70	5.263	0.00317	0.0186
Hydrogen	20.37	70.85	448.7	9.772	0.0126	0.1005
Neon	27.11	1205	85.77	1.867	0.127	0.123

Air	78.90	875.1	204.9	1.933	0.270	0.140
Argon	87.3	1395	169.3	1.237	0.260	0.129
Oxygen	90.19	1142	213.2	1.697	0.195	0.151
Methane	111.7	422.3	510.8	3.481	0.117	0.189

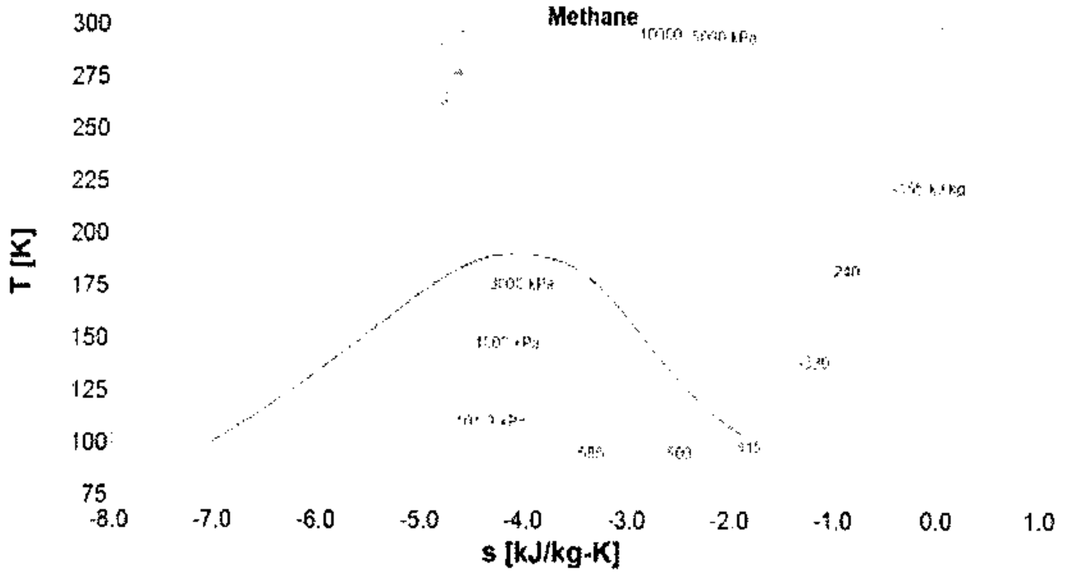


Figure 6: T-s diagram of Methane showing constant pressure and constant enthalpy lines

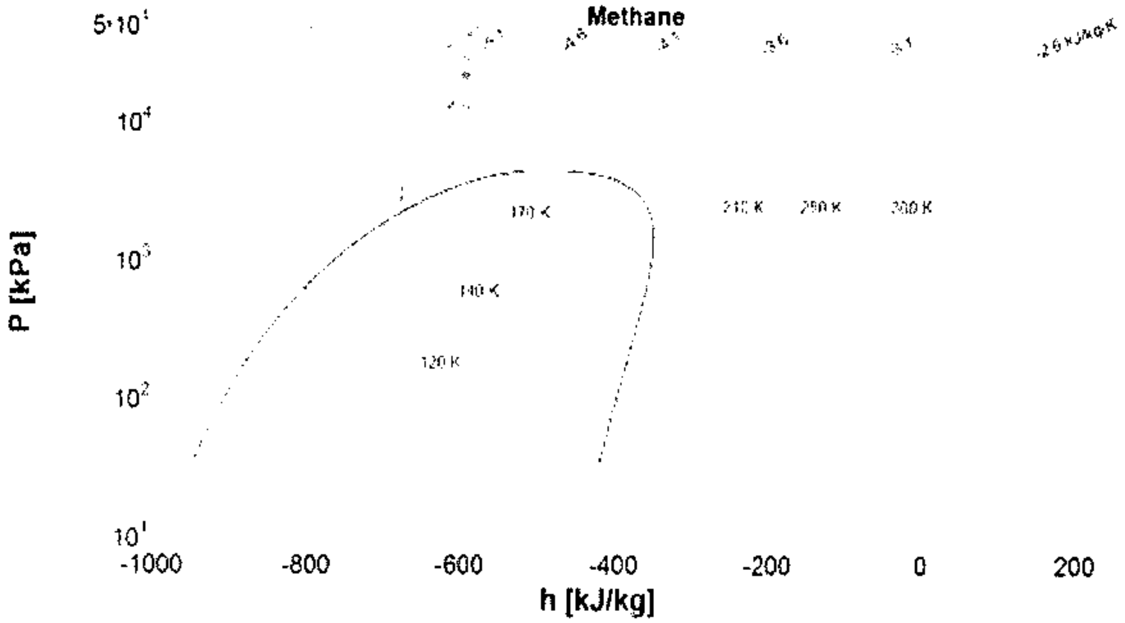


Figure 7: P-h diagram of methane showing constant enthalpy and isothermal lines

3.3.4 Basics of Refrigeration

3.3.4.1 Refrigeration and its unit

Refrigeration is the process of cooling a space, substance, or system to lower and/or maintain its temperature below the ambient one. The unit of refrigeration is expressed in "Tonne of refrigeration". A tonne of refrigeration is defined as the amount of refrigeration effect produced by uniform melting of 1 tonne of ice from and at 0 ℃ in 24 hours.

Latent heat of ice is 335 kJ/kg

1 Tonne =1000*335 kJ in 24 hours

= 232.6 kJ/min

In practice, 1 Tonne of refrigeration is taken equivalent to 210 kJ/min or 3.5 kW.

COP- Coefficient of performance is ratio of heat extracted in the refrigerator to the work done on the refrigerant. It is known as theoretical COP.

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

Where

- Q is the useful heat supplied or removed by the considered system.
- W is the work required by the considered system.

3.3.4.2 Air cycle refrigeration

Air cycle refrigeration systems belong to the general class of gas cycle refrigeration systems, in which a gas is used as the working fluid. The gas does not undergo any phase change during the cycle, consequently, all the internal heat transfer processes are sensible heat transfer processes. Gas cycle refrigeration systems find applications in air craft cabin cooling and also in the liquefaction of various gases.

3.3.4.2.1 Air standard cycle assumption

- The working fluid is a fixed mass of air that behaves as an ideal gas

- The cycle is assumed to be a closed loop cycle with all inlet and exhaust processes of open loop cycles being replaced by heat transfer processes to or from the environment
- All the processes within the cycle are reversible, i.e., the cycle is internally reversible
- The specific heat of air remains constant throughout the cycle

An analysis with the above assumptions is called as cold Air Standard Cycle (ASC) analysis. This analysis yields reasonably accurate results for most of the cycles and processes encountered in air cycle refrigeration systems. The analysis fails when one considers a cycle consisting of a throttling process, as the temperature drop during throttling is zero for an ideal gas, whereas the actual cycles depend exclusively on the real gas behavior to produce refrigeration during throttling.

The temperature of the ideal gas can be reduced in two ways:

- Performing some work by the gas in an isentropic process.
- Sensible heat exchange in the cooler environment

When the gas perform work in isentropic process, its internal energy reduces and the as we know that the internal energy depends on the temperature, the temperature of the gas reduces.

3.3.4.2.2 Reversed Carnot cycle

Reversed Carnot cycle is an ideal refrigeration cycle for constant temperature external heat source and heat sinks. Figure 6 shows the schematic of a reversed Carnot refrigeration system using a gas as the working fluid along with the cycle diagram on T-s and P-v coordinates. As shown, the cycle consists of the following four processes:

Process 1-2: Reversible, adiabatic compression in a compressor

Process 2-3: Reversible, isothermal heat rejection in a compressor

Process 3-4: Reversible, adiabatic expansion in a turbine

Process 4-1: Reversible, isothermal heat absorption in a turbine

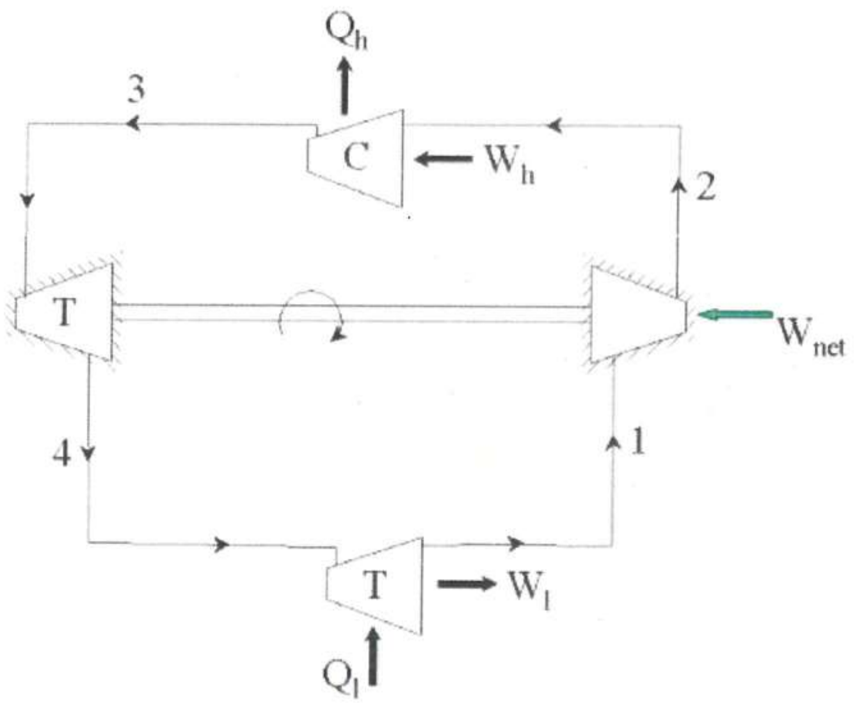


Figure 8: Reversed Carnot cycle

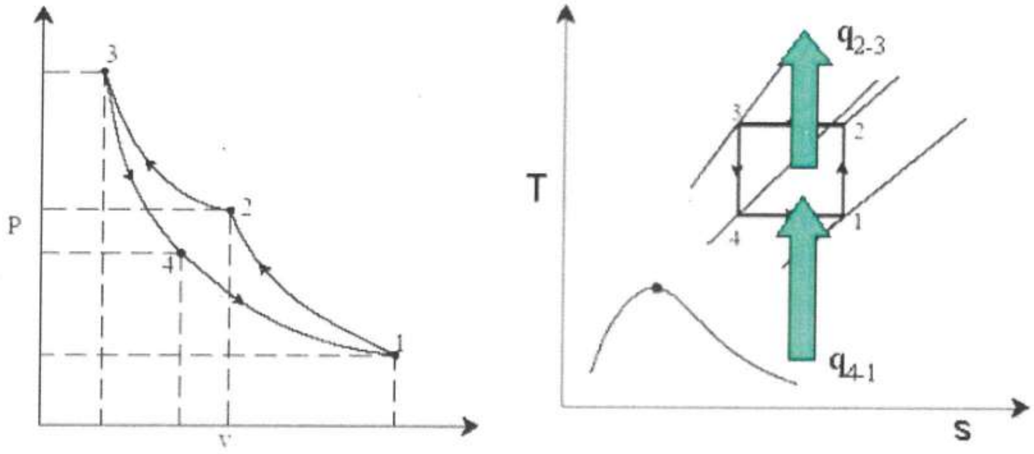


Figure 9: Reversed carnot refrigeration in P-v and T-s coordinates

$$\mathrm{COP}_{\mathrm{Carnot}} = \left|\frac{q_{4-1}}{w_{\mathrm{net}}}\right| = \left(\frac{T_l}{T_h - T_l}\right)$$

3.3.4.2.3 Limitations of Carnot cycle:

Carnot cycle is an idealization and it suffers from several practical limitations. One of the main difficulties with Carnot cycle employing a gas is the difficulty of achieving isothermal heat transfer during processes 2-3 and 4-1. For a gas to have heat transfer isothermally, it is essential to carry out work transfer from or to the system when heat is transferred to the system (process 4-1) or from the system (process 2-3). This is difficult to achieve in practice. In addition, the volumetric refrigeration capacity of the Carnot system is very small leading to large compressor displacement, which gives rise to large frictional effects. All actual processes are irreversible; hence completely reversible cycles are idealizations only.

3.3.4.3 Ideal Reverse Brayton cycle

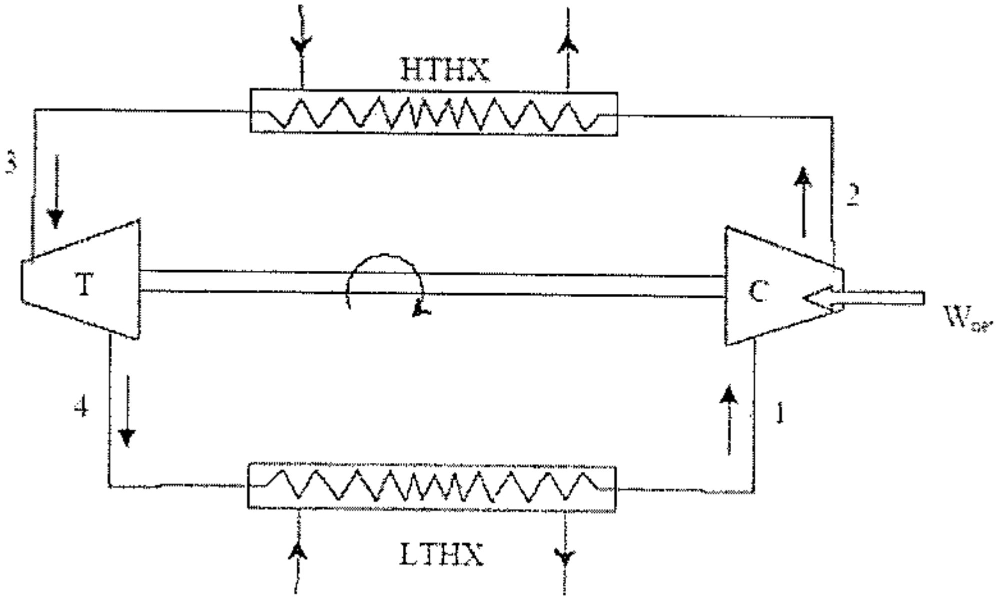


Figure 10: Brayton Cycle

This is an important cycle frequently employed in gas cycle refrigeration systems. This cycle is also called as Joule or Bell-Coleman cycle. Figure 8 shows the brayton cycle.

Process 1-2: Reversible, adiabatic compression in a compressor

Process 2-3: Reversible, isobaric heat rejection in a heat exchanger

Process 3-4: Reversible, adiabatic expansion in a turbine

Process 4-1: Reversible, isobaric heat absorption in a heat exchanger

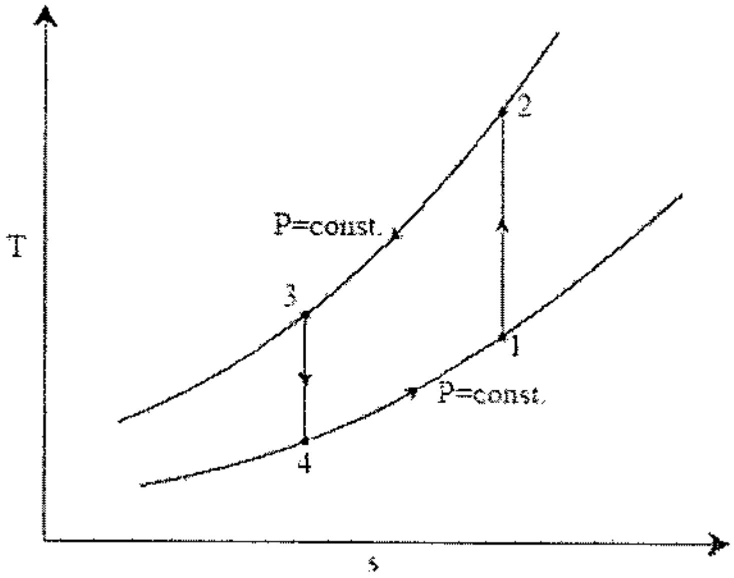


Figure 11: T-s diagram for brayton cycle

$$\text{COP}= \frac{1}{r_p^{\frac{\gamma-1}{\gamma}}-1}$$

$$r_p=\frac{p_3}{p_4}$$

$P_2=P_3$ & $P_4=P_1$

From the above expression for COP, the following observations can be made:

a) For fixed heat rejection temperature (T3) and fixed refrigeration temperature (T1), the COP of reverse Brayton cycle is always lower than the COP of reverse Carnot cycle that is

$$\text{COP}_{\text{Brayton}}=\left(\frac{T_4}{T_3-T_4}\right)<\text{COP}_{\text{Carnot}}=\left(\frac{T_1}{T_3-T_1}\right)$$

b) COP of Brayton cycle approaches COP of Carnot cycle as T1 approaches T4 (thin cycle), however, the specific refrigeration effect [cp(T1-T4)] also reduces simultaneously

c) COP of reverse Brayton cycle decreases as the pressure ratio r_p increases

The actual reverse Brayton cycle differs from the ideal cycle due to:

i. Non-isentropic compression and expansion processes
ii. Pressure drops in cold and hot heat exchangers

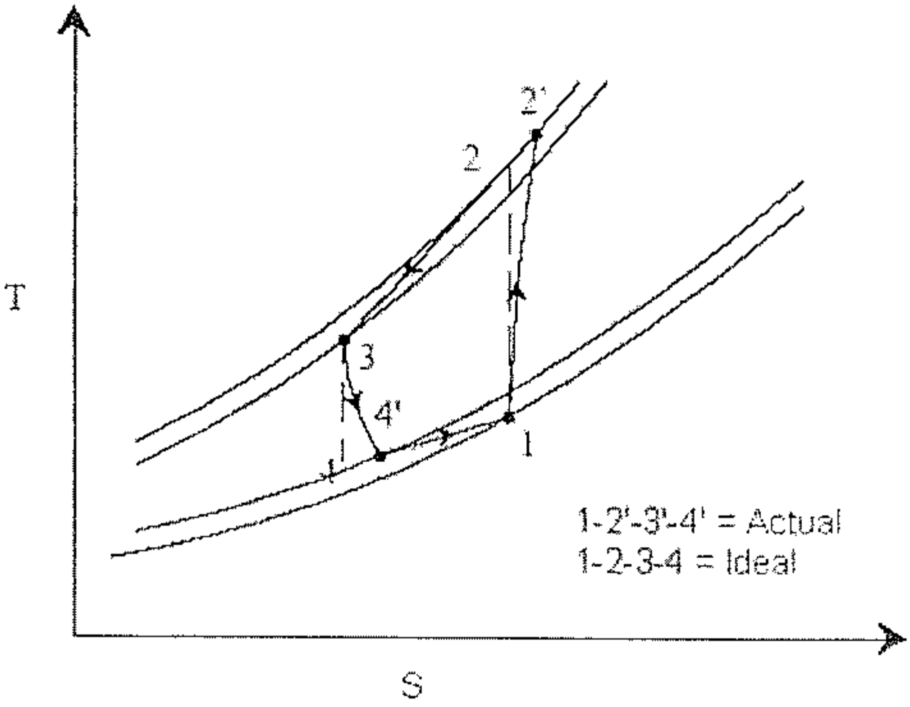


Figure 12: Comparison of ideal and actual Brayton cycles T-s plane

Figure 10 shows the ideal and actual cycles on T-s diagram. Due to these irreversibilities, the compressor work input increases and turbine work output reduces. The actual work transfer rates of compressor and turbine are then given by:

$$W_{1-2,act} = \frac{W_{1-2,isen}}{\eta_{c,isen}}$$

$$W_{3-4,act} = \eta_{t,isen} W_{3-4,isen}$$

Where, $\eta_{c,isen}$ and $\eta_{t,isen}$ are the isentropic efficiencies of compressor and turbine, respectively. In the absence of pressure drops, these are defined as:

$$\eta_{c,isen} = \frac{(h_2 - h_1)}{(h_{2'} - h_1)} = \frac{(T_2 - T_1)}{(T_{2'} - T_1)}$$

$$\eta_{t,isen} = \frac{(h_{3'} - h_{4'})}{(h_3 - h_4)} = \frac{(T_{3'} - T_{4'})}{(T_3 - T_4)}$$

The actual net work input, $w_{net,act}$ is given by:

$$W_{net,act} = W_{1-2,act} - W_{3-4,act}$$

Thus, the net work input increases due to increase in compressor work input and reduction in turbine work output. The refrigeration effect also reduces due to the irreversibility. As a result, the COP of actual reverse Brayton cycles will be considerably lower than the ideal cycles. Design of efficient compressors and turbines plays a major role in improving the COP of the system.

In practice, reverse Brayton cycles can be open or closed. In open systems, cold air at the exit of the turbine flows into a room or cabin (cold space), and air to the compressor is taken from the cold space. In such a case, the low side pressure will be atmospheric. In closed systems, the same gas (air) flows through the cycle in a closed manner. In such cases it is possible to have low side pressures greater than atmospheric. These systems are known as dense air systems. Dense air systems are advantageous as it is possible to reduce the volume of air handled by the compressor and turbine at high pressures. Efficiency will also be high due to smaller pressure ratios. It is also possible to use gases other than air (e.g. helium) in closed systems.

3.3.5 Components in Liquefied Natural Gas

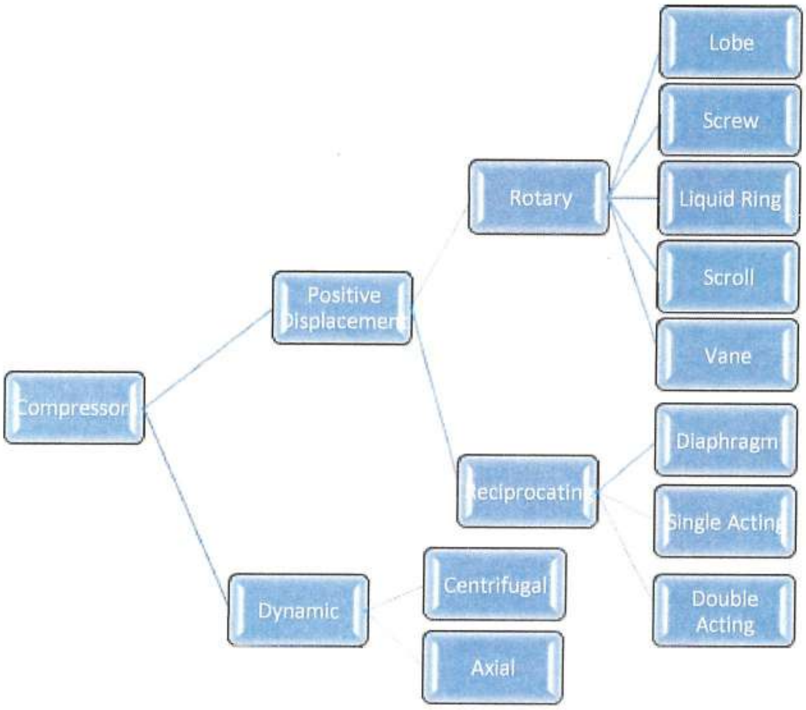
Different Components in Liquefaction Plant

- Compressors
- Heat Exchangers
- Throttling device
- Storage tank

3.3.5.1 Compressors

Compressors are mechanical device used to pressurize the gas and reduce its volume. There are mainly two types of compressor: Positive displacement and dynamic compressor

I. *Type of compressor*



a) *Positive displacement compressor*

In positive displacement compression, the air is drawn into one or more compression chambers, which are then closed from the inlet. Gradually the volume of each chamber decreases and the air is compressed internally. When the pressure has reached the designed build-in pressure ratio, a port or valve is opened and the air is discharged into the outlet system due to continued reduction of the compression chamber's volume.

b) Dynamic Compressor

In dynamic compression, air is drawn between the blades on a rapidly rotating compression impeller and accelerates to a high velocity. The gas is then discharged through a diffuser, where the kinetic energy is transformed into static pressure. Most dynamic compressors are turbo compressors with an axial or radial flow pattern.

In a dynamic compressor, the pressure increase takes place while the gas flows. The flowing gas accelerates to a high velocity by means of the rotating blades on an impeller. The velocity of the gas is subsequently transformed into static pressure when it is forced to decelerate under expansion in a diffuser. Depending on the main direction of the gas flow used, these compressors are called radial or axial compressors. As compared to displacement compressors, dynamic compressors have a characteristic whereby a small change in the working pressure results in a large change in the flow rate.

Each impeller speed has an upper and lower flow rate limit. The upper limit means that the gas flow velocity reaches sonic velocity. The lower limit means that the counter pressure becomes greater than the compressor's pressure build-up, which means return flow inside the compressor. This in turn results in pulsation, noise and the risk for mechanical damage.

II. Compression in several stages

In theory, air or gas may be compressed isentropically (at constant entropy) or isothermally (at constant temperature). Either process may be part of a theoretically reversible cycle. If the compressed gas could be used immediately at its final temperature after compression, the isentropic compression process would have certain advantages. In reality, the air or gas is rarely used directly after compression, and is usually cooled to ambient temperature before use. Consequently, the isothermal compression process is preferred, as it requires less work. A common, practical approach to executing this isothermal compression process involves cooling the gas during compression. At an effective working pressure of 7 bar, isentropic compression theoretically requires 37% higher energy than isothermal compression.

A practical method to reduce the heating of the gas is to divide the compression into several stages. The gas is cooled after each stage before being compressed further to the final pressure. This also increases the energy efficiency, with the best result being obtained when each compression stage has the same pressure ratio. By increasing the number of compression stages, the entire process approaches isothermal compression. However, there is an economic limit for the number of stages the design of a real installation can use.

III. *Type of compressor selection*

a) *Piston compressor*

It is least expensive compressor. They are easy to maintain. However they are some drawbacks too.

Piston or reciprocating compressors are meant for intermittent use. They can only work for 50 to 60% of its duty cycle. They require proper cooling time; in order them not to fail. Also they can be noisy.

b) *Rotary screw compressor*

Rotary Compressor provides continuous supply of the fluid. They are meant for 100% continuous duty cycle. Furthermore, smaller screw compressors can be tank mounted allowing for additional storage capacity which, coupled with an integrated dryer, allows for plug and play installation. However they are costly.

IV. *Size of compressor*

Size of compressor depends on the pressure required and flow rate required for the operation. Pressure can be expressed in bar, kPa and other units depending on the geographical location. Flow rate is expressed in Litre/sec, cubic meter per hour and others.

Considering the above mentioned criterion we would choose the dynamic compressor whose specifications will be mentioned in the later section.

Few specifications of the compressors available in the market are listed below.

Table IV: Specifications of H and GT series compressor

Specification	H series compressor (Atlas copco)	GT series compressor (Atlas copco)
Suction pressure (max.)	1.4 bar(a) (8 bar)*	80 bar(a)
Discharge pressure (max.)	30 bar(a)	200 bar(a)
Suction temperature	-40 to 50°C	-200 to +200°C
Effective inlet-flow range	3,500 – 42,500 (85,000**) m3/h	250 – 400,000 m3/h
Maximum No. of stages	1 – 4	1 – 8
Gases handled	Air, nitrogen	All gases
Impeller types	Open	Open / closed
Shaft / impeller connection	Microspline	Hirth serration
Seals		
Labyrinth	•	•
Carbon ring	•	•
Dry-gas seal		•
Capacity / pressure control		
Variable inlet-guide vanes (IGV)	•	•
Variable diffuser-guide vanes (DGV)	•	•
Variable speed		•
Inlet throttle	•	•
API	672	672/617
Axial thrust compensation	High-speed axial bearings	High-speed axial bearings or thrust collar
Oil system	Manufacturer's	Manufacturer's

	standard or API 614	standard or API 614
Coupling	Dry	Dry

Multistage compressor: When the pressure is high the temperature also increases and can cause harm to the compressor. Hence the fluid is cooled before achieving final pressure in intercooler. The inter-cooling reduces the volume of fluid which used to increase due to heat. The compressed fluid from the first stage enters the intercooler where it is cooled. This fluid is given as input to the second stage where it is compressed again. The multistage compressor can develop a pressure of around 50 bar and above.

Open/Closed impeller: The open impeller has a series of vanes attached to a central unit. This is done for the mounting on the shaft. However, this design is more sensitive to wear and tear of the blades. The closed impeller has a side wall on the either sides (top and bottom) of the vanes. Closed impellers are the most commonly used impellers in the industry since they can deal with volatile and explosive fluids. The closed impeller is initially really efficient, but with time, loses its efficiency as the clearance of the wear ring increases, whereas the efficiency of an open impeller can be maintained through clearance adjustment.

Open Impeller Closed Impeller

Figure 13: Types of impeller

3.3.5.2 Heat Exchanger

A heat exchanger is a device that is used to transfer thermal energy (enthalpy) between two or more fluids, between a solid surface and a fluid, or between solid particulates and a fluid, at different temperatures and in thermal contact.

3.3.5.3 Throttling device

Throttling device is a generic term that simply dissipates pressure energy by irreversibly converting it to pressure energy thermal energy. Throttling device do not give useful work. A throttle need not have same inlet and outlet flow velocities, and, therefore, it may have a significant specific kinetic energy changes across it. A throttling device is commonly taken to be adiabatic regardless of whether it is actually insulated or not.

The throttling phenomenon, also known as Joule – Kelvin /Joule –Thompson effect, may be demonstrated by considering a steady-state, steady flow (SSSF) process across a restriction, with a resulting drop in pressure. The restriction can be a porous plug, an orifice plate, a butterfly valve, any type of flow or pressure control valve. Even geometry like sudden contraction or sudden expansion may give the desired effect.

3.3.5.3.1 Porous Plug experiment

The porous plug experiment was designed to measure temperature changes when a fluid flows steadily through a porous plug which is inserted in a thermally insulated, horizontal pipe. The apparatus used by Joule and Thomson is shown in Figure 14

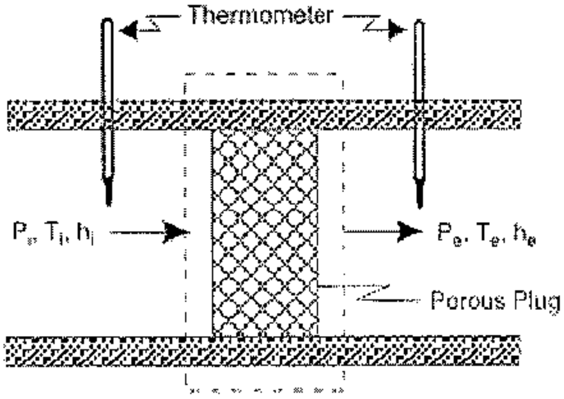


Figure 14: Porous plug setup

A gas at pressure P_i and temperature T_i flows continuously through a porous plug in a tube and emerges into a space which is maintained at a constant pressure P_e. The device is thermally insulated and kept horizontal. Consider the dotted portion as control volume.

Q=0, W=0. This result in $h_i=h_e$.

Therefore, whenever a fluid expands from a region of high pressure to a region of low pressure through a porous plug, partially opened valve or some obstruction, without exchanging any energy as heat and work with the surrounding (neglecting, the changes in PE and KE), the enthalpy of the fluid remains constant, and the fluid is said to have undergone a throttling process. If the downstream pressure is held at several different values successively and at each of these T_e is measured, we shall obtain a situation explained by Figure 15

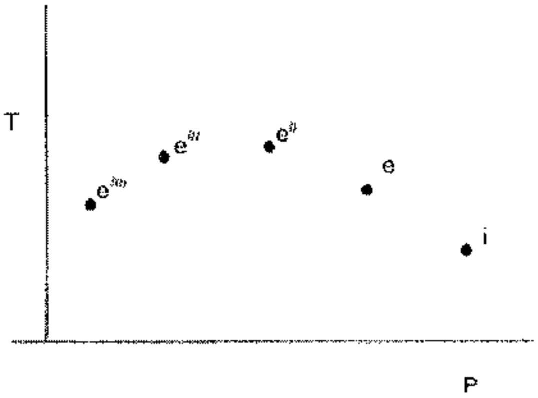


Figure 15: T-P diagram for inversion curve

Each plotted point represents a state for which the enthalpy is equal to h_i. If we join all these points we shall be able to obtain a constant enthalpy line

Such a plot of *P* versus T yields may also be called **isenthalpic curve.** The slope of the isenthalpic curve is called the Joule-Thomson coefficient and given by

$$\mu_{JT} = \left(\frac{\partial T}{\partial P}\right)_h$$

The experiments are conducted with different P_i and T_i in order to find out the isenthalpic curves. A family of isenthalpic curves is shown in Figure 14 which is typical of all real gases.

The point at which $\mu_{JT} = 0$ called the **inversion point**. The locus of all inversion points is the inversion curve.

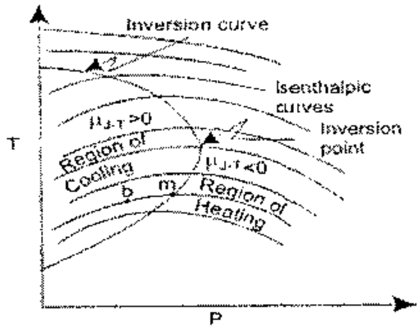


Figure 16: Inversion curve

In the region left of the inversion curve, $\mu_{JT} > 0$. In the throttling process the downstream pressure P_e is always less than the upstream pressure P_i. Therefore, whenever a real gas is subjected to throttling, the temperature of the gas decreases if the initial state lies in the region to the left of the isenthalpic curve. This is explained by a process from m to b in Figure 16

To the right of the inversion curve, $\mu_{JT} < 0$. If the initial state of gas lies in the region to right of the inversion curve, the temperature of the gas increases upon throttling. For almost all the gases, at ordinary range of pressures and temperature, $\mu_{JT} < 0$ and the maximum inversion temperature is above the room temperature. The exceptions are hydrogen, helium and neon. For hydrogen, the maximum inversion temperature is 200 K and for helium the maximum inversion temperature is 24 K. If hydrogen is throttled at room temperature, the temperature of the gas increases. To produce low temperature by throttling, the initial temperature of hydrogen should be below 200 K. This is usually accomplished by cooling with liquid nitrogen. Similarly, in the production of liquid helium by throttling, the initial temperature of helium should be below 24 K. Hence it is cooled by liquid hydrogen prior to throttling.

Suppose an ideal gas is throttled. Since throttling process is isenthalpic, and for an ideal gas enthalpy is a function of temperature only, the temperature of an ideal gas does not change during a throttling process. Hence, $\mu_{JT} < 0$ for an ideal gas.

3.3.5.3.2 Types of Throttling Device

- Capillary Tube,
- Hand operated expansion valve,

- Automatic or constant pressure expansion valve,
- Thermostatic expansion valve,
- Low side float valve, and
- High side float valve

3.3.5.3.2.1 Capillary Tube

The capillary tube is used as an expansion device in small capacity hermetic sealed refrigeration units such as in water coolers, domestic refrigerators, room air-conditioners and freezers. It is a copper tube of small internal diameter and of varying length depending upon the application. The inside diameter of the tube used in refrigeration work is generally about 0.5 mm to 2.25mm and the length varies from 0.5m to 5m. It is installed in the liquid line between the condensor and the evaporator. A fine mesh screen(filter) is used at the inlet of the ttube to protect contaminants.

The liquid refrigerants in the ondensor enters the capillary tube. Due to the frictional resistance offered by a small diameter tube, the pressure drops. The frictional resistance is directly proportional to the length and inversely proportional to the diameter; the longer the capillary tube and smaller its inside diameter, greater is its pressure drop created in the refrigerant flow. The greater the pressure drop, greater is the temperature drop. The diameter and the length selected for a certain load cannot be used for other loads.

Advantages of capillary tube:

- The cost of capillary tube used is less than any other expansion device.
- When the compressor stops, refrigerant continues to flow into the evaporator and equalises the pressure between high side and low side of the system. This reduces the starting torque which is big advantage.
- No receiver is necessary.

3.3.5.3.2.2 Hand operated expansion valve

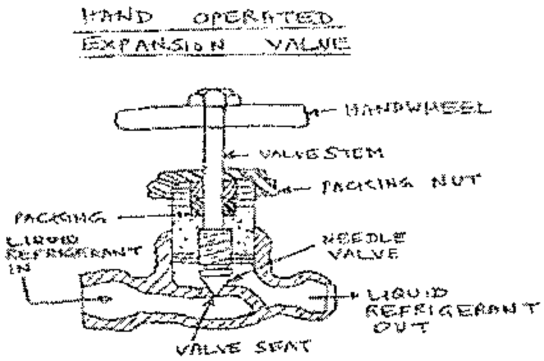


Figure 17: Hand operated expansion valve

A hand operated expansion valve is shown in the figure 17. It requires an operator to control the flow of refrigerant. The conical shaped needle valve extends down into the valve port and restricts the flow area through the port. When closed the valve rests on its conical seat. The use of hand-operated valve is limited to systems operating under constant load for a long period of time such as ice making plant and cold storages. It is not suitable for installations where load varies and compressor is used intermittently to maintain constant temperature.

3.3.5.3.2.3 Automatic or Constant Pressure Expansion Valve

The automatic expansion valve is also known as constant pressure expansion valve because it maintains constant evaporator pressure regardless of the load on the evaporator. Its main moving force is the evaporator pressure. An automatic expansion valve is shown in figure 18.

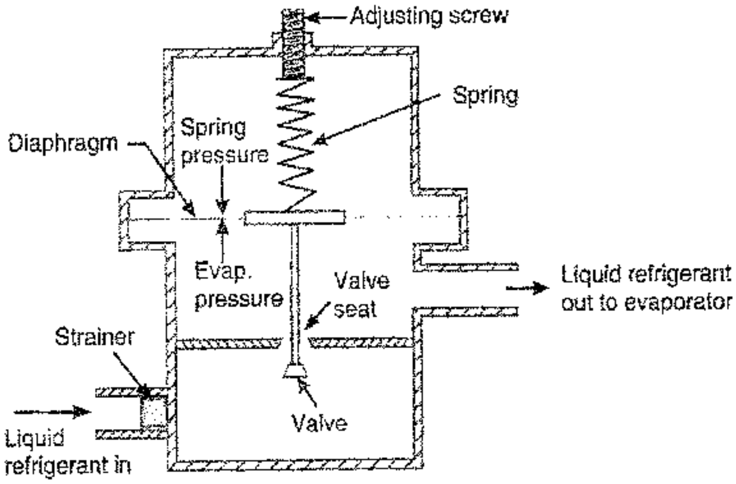


Figure 18: Automatic Expansion valve [31]

3.3.5.3.2.4 *Thermostatic or Constant Superheat Expansion Valve*

It is also called constant expansion valve because it maintains a constant superheat of the vapour refrigeration at the end of the evaporator coil, by controlling the flow of liquid refrigerant through the evaporator.

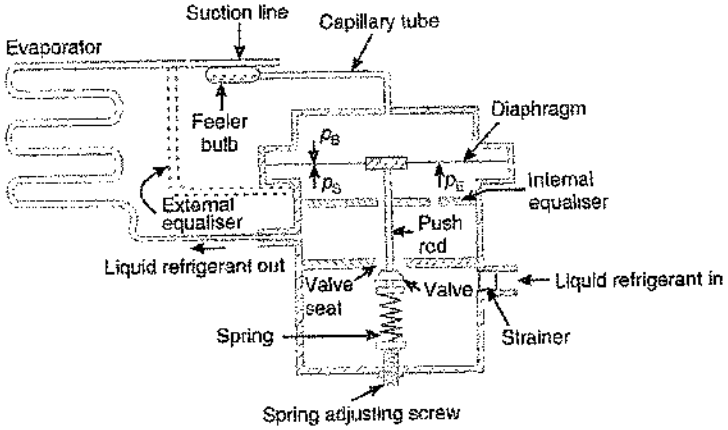


Figure 19: Thermostatic valve [31]

Under normal condition the feeler bulb pressure acting at the top of the diaphragm is balanced by the spring pressure and the evaporator pressure acting at the bottom of the diaphragm. The force tending to close the valve is dependent upon the spring pressure and the evaporator pressure which, in turn depends upon the saturation temperature of the refrigerant in the evaporator coil. The force tending to open the valve depends upon the feeler bulb pressure which, in turn depends upon the temperature of the refrigerant in the bulb. Thus the operation of the valve is controlled by the difference between the two temperaature which is superheat. The degree of the superheat of the vapour refrigerant leaving the evaporator depends upon the initial setting of the spring tension, which can be changed with the help of spring adjusting screw. When the valve is set for a certain superheat, then it maintains that setting under all load conditions on the evaporator.

If the load on the evaporator increases, it causes the liquid refrigerant to boil faster in the evaporator coil. The temperature of the feeler bulb increases due to early vapouriztion of the liquid refrigerant. Thus the feeler bulb pressure increases and this pressure is transmitted through the capillary tube to the diaphragm. The diaphragm moves downwards and opens the valve to admit more quantity of liquid refrigerant to the evaporator. This contiues till the pressure eqilibrium on the diaphragm is reached. On the other hand, when the load on the evaporator decreases, less liquid refrigerant evaporates in the evaporator coil. The excess liquid refrigerant flows towards the evaporator outlet which cools the feeler bulb with the result the feeler bulb pressure decreases due to decrease in its temperature. The low feeler bulb pressure is transmitted through the cpaillary tube to diaphragm and moves it upward. This reduces the opening of the valve and tus the liquid refrigerant to the evaporator. The evaporator presssure decreases due to reduced quantity of liquid refrigerant flowing to the evaporator. This continues till the evaporator pressure and the spring pressure maintain equilibrium with the feeler bulb pressure.

3.3.6 Gas Liquefaction Parameters

Table V: Performance Parameters

Work per unit mass of gas compressed by compressor	$\frac{W_c}{m_i}$
Fraction of total gas liquefied (y)	$\frac{m_f}{m_1}$
Figure of Merit	$\frac{W_{i,c}}{W_c}$

3.3.7 Different types of Gas Liqufaction Processes

Liquefaction of cryogenic gases can be accomplished by cooling and condensation with proper energy interaction between the streams at different conditions of the process without and with external refrigeration[31]. Liquefaction cycles are basically open system while the vapours remaining are circulated back to the system at the inlet of the compressor.

There are many gas liquefaction systems. Some of them are

- Thermodynamic Ideal System
- Linde-Hampson System
- Pre-cooled Linde Hampson System
- Claude system
- Kapitza system

Designing is a task which requires lot of data as input. The input can be derived experimentally or through rigorous iterative process satisfying the universal laws like conservation of mass and energy along with laws of thermodynamics when considered a problem having thermal considerations or heat transfer.

In this thesis, I had very less raw data and uses iterative converging processes. The process is explained for simple Linde-Hampson cycle and pre-cooled Linde-Hampson cycle.

3.3.7.1 *Thermodynamically Ideal system*

In an ideal cycle, there are two processes

- Isothermal compression
- Reversible isentropic expansion

The feed gas should be compressed to a supercritical pressure such that it reaches an entropy value that equals to the saturated liquid. The compressed fluid should be isentropically expanded to obtain saturated liquid state. A schematic flow diagram is shown in figure 17. This cycle is not practical in nature as the pressure ratio is too high. The ideal liquefaction process is the standard for improving the performance of actual liquefaction process.

Note: The letter depicts the usual meaning while the subscripts, depicts the point in the cycle or process.

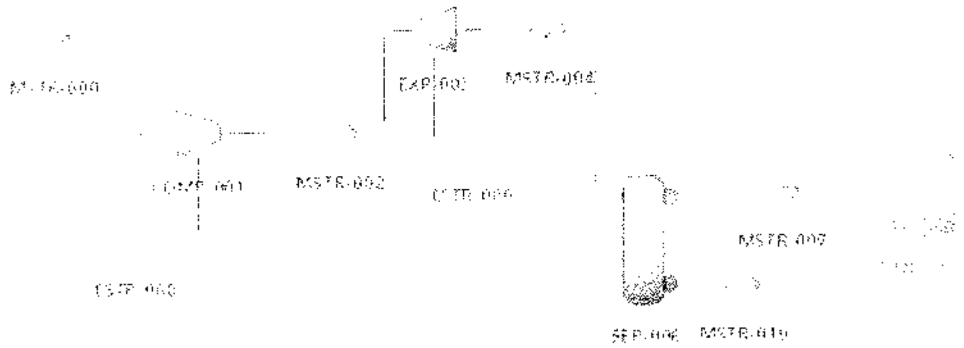


Figure 20: Schematic representation of Thermodynamically Ideal cycle

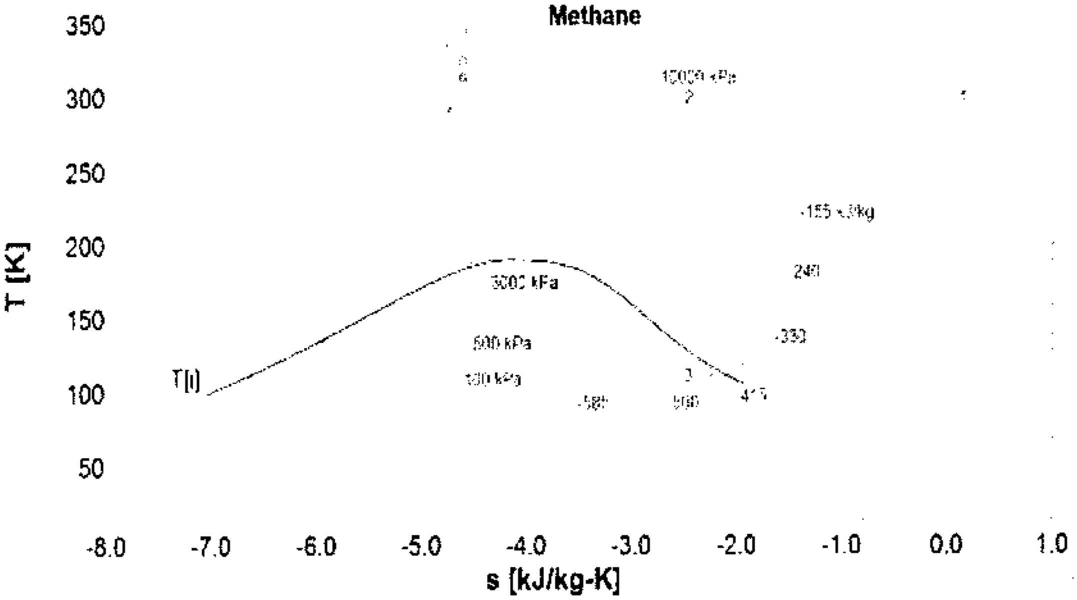


Figure 21: Corresponding T-s graph of thermodynamically ideal cycle

Theoretical Minimum Work for Liquefaction

In this cycle the 100% of the gas is considered to be liquefied. It can also be observed that the cycle requires minimum work input compared to isobaric liquefaction process. The process neglects isobaric cooling external cooling, isenthalpic expansion, latent heat and sensible heat.

Combining first and second law of thermodynamics for an open system steady flow of gas and neglecting kinetic and potential energy changes, equation is written as

$$\dot{Q} - W_l = \sum \dot{m}_o h_o - \sum \dot{m}_l h_l$$

$$mT_1(s_2 - s_1) - \dot{W}_l = \dot{m}\,(h_f - h_1)$$

$$-\frac{\dot{W}_l}{\dot{m}} = T_1(s_1 - s_f) - (h_1 - h_f)$$

Figure of Merit (FOM)

Figure of merit characterizes the performance of the liquefaction process with respect to the ideal liquefaction process [31].

FOM= $\frac{-W_i}{-W_l}$

If the temperature at the entry to the compressor and sink are different than ideal work for the process can be written as:

$$-W_i = T_o(s_1 - s_f) - (h_1 - h_f)$$

3.3.7.2 Simple Linde-Hampson Cycle

The first cryogenic liquefaction cycle ever invented is cascade cycle, but due to inherent irreversiblities and some limitations, the cycles were not success. Linde and Hampson independently developed continuously working cycle for air, nitrogen, methane, oxygen and argon. The cycle is mainly based on throttling expansion for lowering temperature as shown in figure 19.

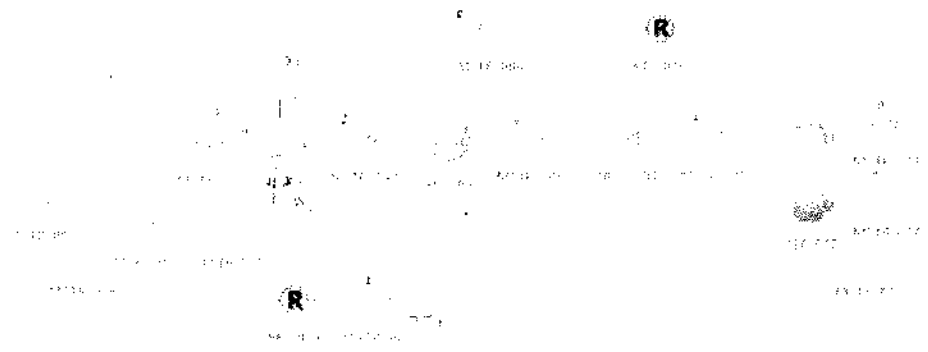


Figure 22: Schematic representation of Simple Linde-Hampson cycle

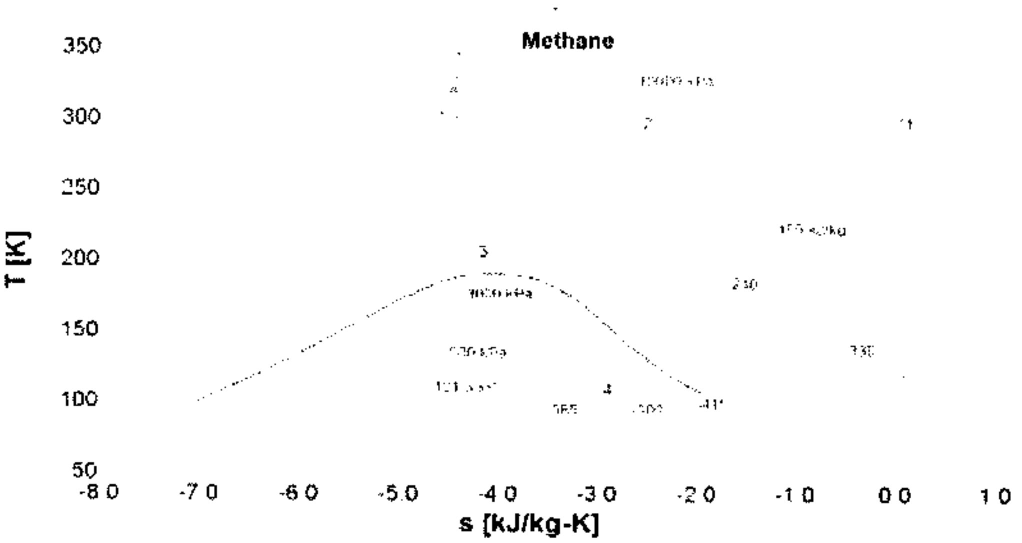


Figure 23: Corresponding T-s diagram of Simple Linde-Hampson cycle

The gas at the inlet is isothermally compressed to a high pressure. If isothermal compression is not possible than the compression is polytropic. The gas is cooled to ambient temperature in the compressor , inter-stage cooler and after coolers. The high pressure compressed gas is cooled in the heat exchanger which is further expanded isenthalpically by throttling to through a valve to ambient pressure. The expanded stream is a mixture of saturated vapour and saturated liquid which needs to be separated. The return gas at atmospheric pressure is used for cooling in the heat exchanger and is mixed with the make-up gas. The fraction of gas liquified is dependent on the temperature and pressure before throttling. [31]

1-2: Isothermal Compression

2-3: Pre-cooling by outgoing stream

3-4: Throttling expansion

4-g: Separation of saturated vapour

4-f: Separation of saturated liquid

g-1: Return of saturated vapour for recycling at the ambient temperature

Applying first law of thermodynamics for steady state. Mass flow rate $\dot{m}$ is considered and assuming ideal heat exchanger efficiency and effectiveness. No temperature of approach at the warm end of heat exchanger and no pressure drops and no heat in leaks,

$$\dot{m}h_2 = (\dot{m} - m_{f)}h_1 \dot{+} m_f h_f$$

Fraction of liquefaction,y, is defined as

$$y = \frac{m_f}{\dot{m}} = \frac{h_1 - h_2}{h_1 - h_f}$$

if the heat exchanger with minimum temperature of approach is considered at the warm end, than the return gas at point 7, is slightly below than the inlet temperature of hot stream, then by enthalpy balance fraaction of liquefaction is

$$y = \frac{h_7 - h_2}{h_7 - h_f}$$

Energy requirement for reversible compression is given as follows:

$$-\frac{\dot{W}}{\dot{m}} = T_1(s_1 - s_2) - (h_1 - h_2)$$

Energy per unit mass liquefied,

$$W_l = \frac{\dot{W}}{\dot{m}_f} = \frac{\dot{W}}{my}$$

The thesis aims to find the configuration of the cycle which is done by accessing the knowns inputs and the final output. The temperature, pressure, enthalpy and entropy are found through EES software by appropriate assumptions.

From figure 23, one can notice that the for an easy liquefaction process, we have to compress to a high pressure or extract lot of heat in the heat exchanger. To obtain liquefaction of natural gas, a pressure of 10,000 kPa is assumed so that after a certain quantity of heat exchanged we reach near to saturation liquid curve. After isenthalpic expansion of gas, we enter in the dome of the natural gas. The known parameters are listed below.

Inlet to compressor, Point 1: Temperature = 27°C or 300.15 K that is ambient temperautre. Pressure =101.325 kPa at the atmospheric pressure. The outlet temperature is same as that of inlet temperature that is 300.15°C and the pressure is assumed to be 10,000kPa. This becomes the inlet tempearture to the heat exchanger. The outlet temperature of the hot stream need to be found, while the data for the cold stream is unkown too. So we go to the required output. We know that the stream final pressure is atmospheric pressure in order to prevent the entry of atmospheric air or other gases inside the storage tank. Hence the constant pressure line at 101.325 kPa is drawn in T-s diagram inside the dome. The dryness fraction is assumed to be 0.8 to locate a point in the graph. The temperautre of liquefaction is 111. So the final point is known to us by using three parameters, temperature, pressure and dryness fraction. To locate point 3 we have to move up the constant enthalpy line on the graph for an enthaply of -500kJ/kg. The corresponding temperature is 209.1 K. The fraction of gas which is not liquefied returns back to the heat exchanger for cooling of the in coming hot stream. In this way the expanded gas which is not liquified but have a temperature of 111.7 K is used for sensible cooling. The enthalpy drop is obviously not same as that of the liquid stream.

$$y = \frac{h_1 - h_2}{h_1 - h_f} = \frac{3.143 + 95.84}{3.143 + 940.5} = 0.104$$

$$-\frac{\dot{W}}{\dot{m}} = T_1(s_1 - s_2) - (h_1 - h_2) = 300(0.01139 + 2.617) - (3.143 + 95.84)$$

$$= 689.534$$

$$W_l = \frac{W}{\dot{m}y} = 6630.134\ kJ/kg$$

Fluid properties are found in Engineering Equation solver by writing a simple code. Refer to appendix 1 for the code and its result.

3.3.7.3 Pre-cooled Linde Hampson cycle

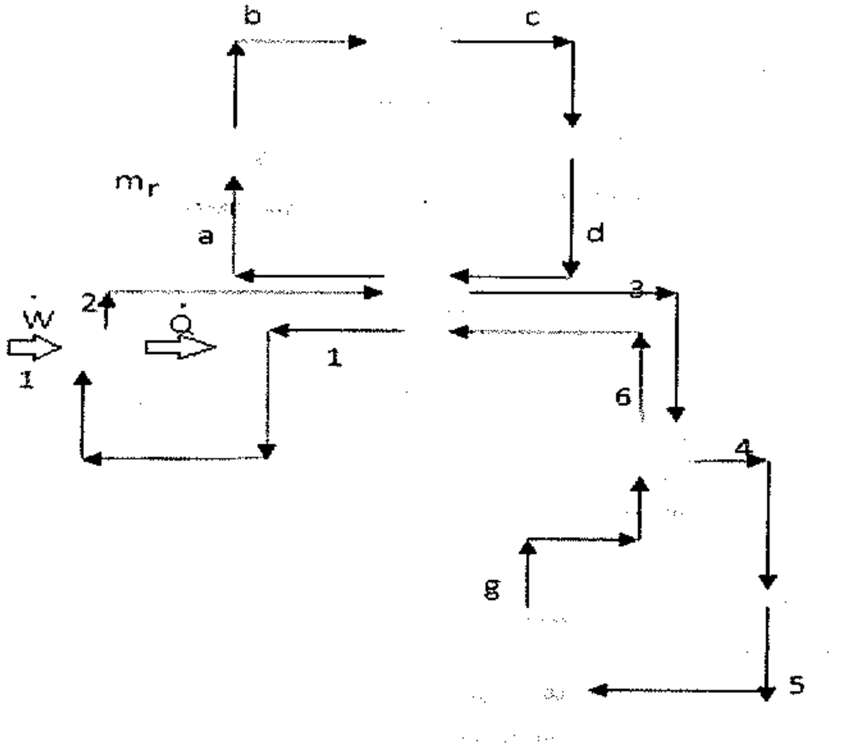


Figure 24: Schematics representation of Precooled Linde-Hampson cycle

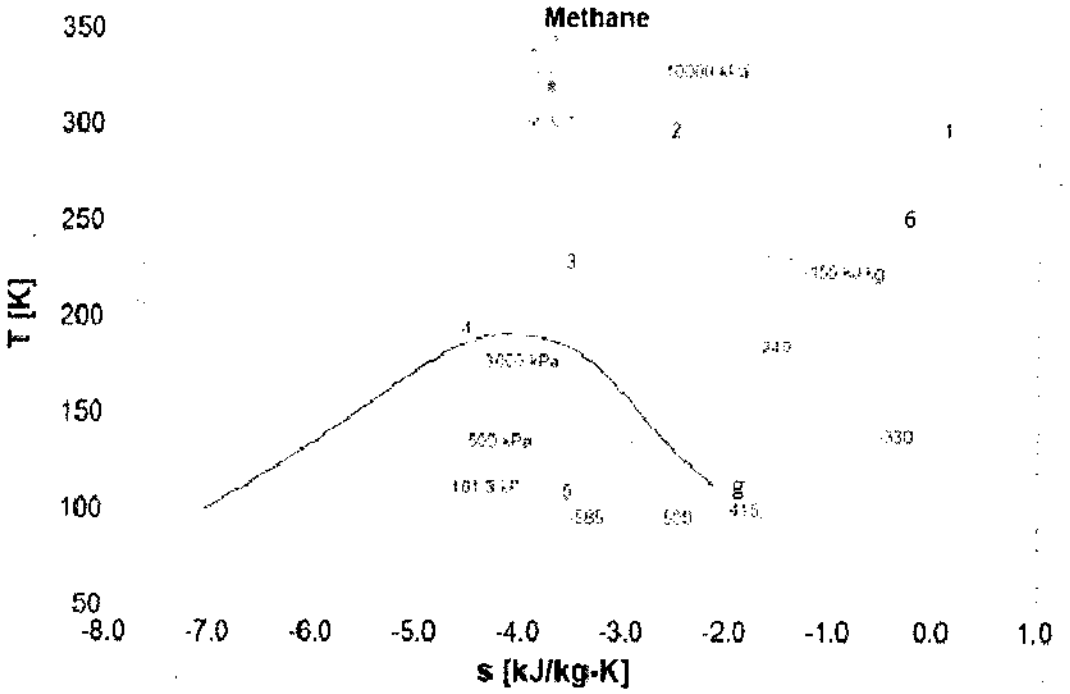


Figure 25: T-s diagram of precooled Linde-Hampson cycle

The pre-cooled Linde-hampson cycle employs external cooling to reduce the temperature of heat exchanger at the warm end which reduces the enthalpy of the stream and better fraction of liquefaction can be obtained. So we can reduce the pressure and perform external cooling of gas to obtain better performance.

Steps of the pre-cooled Linde-Hampson cycle are mentioned as follows:

1-2: Isothermal Compression

2-3: Pre-cooling by a refrigerant and outgoing stream

3-4: Cooling by outgoing saturated vapour stream

4-5: Throttling expansion

5-g: Separation of saturated vapour

5-f: Separation of saturated liquid

g-6: Return of saturated vapour for providing cooling in 3-4

6-1: Return of outgoing stream for providing cooling in 2-3

d-a: Evaporation of the refrigerant to provide cooling in 2-3

a-b: Compression of saturated vapour of the refrigerant

b-c: Cooling and condensation of high pressure refrigerant

c-d: J-T expansion of saturated liquid refrigerant

The work requirement of the two compressors is

$$-\frac{\dot{W}}{\dot{m}} = T_1(s_1 - s_2) - (h_1 - h_2) + r(h_b - h_a)$$

and the fraction of liquefaction is

$$y = \frac{h_1 - h_2}{h_1 - h_f} + r\frac{h_a - h_d}{h_1 - h_f}$$

where r is the refrigerant mass flow ,

$$r = \frac{\dot{m}_f}{\dot{m}}$$

In pre-cooled Linde-Hampson cycle, the refrigerant chosen is propane as the boiling point of the refrigerant is 231.1 K. The temperature is achievable as propane need not be compressed to a high pressure nor the normal boiling point is too high, not to give proper pre-cooling.

Normal boiling point of few refrigerants are mentioned as follws:

Table VI: Refrigerant and their normal boiling point

Refrigerant	Normal Boiling Point (K)
Carbon-dioxide	216.6
Ammonia	240
R134a	247

As described in the simple Linde-Hampson cycle, the procedure for pre-cooled cycle is also followed.

The compressor inlet temperature and pressure are 300.15 K and 101.325kPa respectively. However for the heat exchanger having three stream flow, the temperature for both cold stream is 231.1K. The outlet temperature of hot stream is also same as that of boiling point of the refrigerant, since zero temperature of approach is assumed. The 2nd heat exchanger before expansion valve has inlet temperature of 231.1K of hot stream and 111.7 K is the temperatue of inlet of cold stream. The outlet temperature of the cold stream is 231.1 as assumed earlier. Hence using the energy balance we can calculate inlet temperatre of the hot stream as shown below.

$$mC_p(T_3 - T_4) = m_g C_p(T_6 - T_g)$$

$$mC_{p3}(T_3 - T_4) = 0.8mC_{p6}\left(T_6 - T_g\right)$$

$$C_{p3} = 5.739\frac{kJ}{kg - k}, T_3 = 231.1\ K, C_{p6} = 2.106\frac{kJ}{kg - k}, T_6 = 231.1K, T_g = 111.7$$

Substituting the above values to the above equation we can find $T_4 = 196.048\ kJ$

The next step is isenthalpic expansion. The outlet pressure is 101325 Pa and the dryness fraction reached is 0.6628 for the constant enthalpy of -572.4$\frac{kJ}{kg}$

$$C_{p4} = 5.001\ {}^{kJ}/_{kg - K}$$

$$C_{p\ avg(3,4)} = {}^{5.739 + 5.001}/_{2} = 5.37$$

After calculating temperature of point 4 and average specific heat at constant pressure along with the dryness fraction, we can calculate temperature at point 6 which was assumed previously as follows:

$$mC_{p,avg(3,4)}(T_3 - T_4) = m_g C_{p6}(T_6 - T_g)$$

Substituting the knowns

We obtain $T_7 = 253.6008\,K$

I chose propane to be secondary fluid and we have to obtain the normal boiling point another refrigerating cycle. This cycle is quite simple and is similar to reversed bryaton cycle.

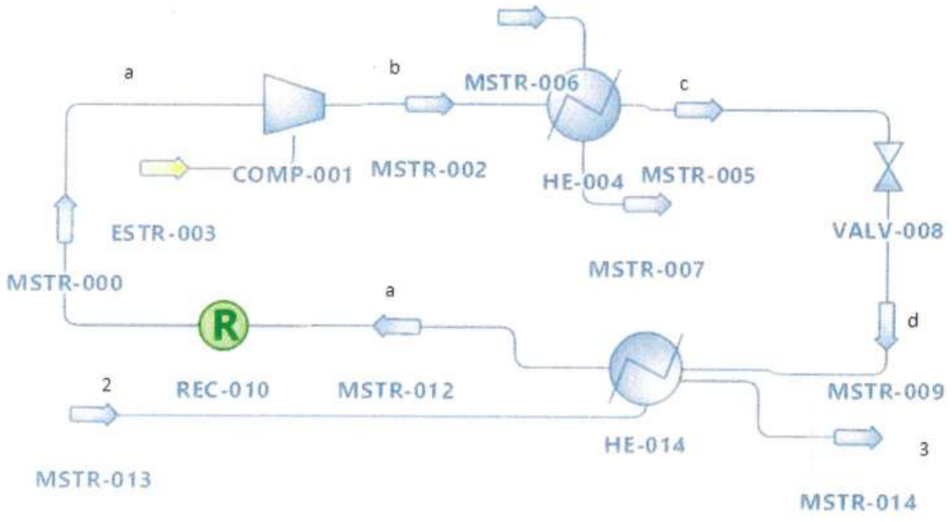


Figure 26: Schematic representation of cycle for cooling of propane

The inlet to the compressor is at 300.15K and 101.325kPa . The compression is adiabatic in nature. Propane is cooled by water at ambient temperature. This is followed by isenthalpic expansion. After the expansion, the liquid propane is utilized to pre-cool incoming methane. The processes are mentioned in short in the following points:

a-b: Adiabatic compression of saturatued vapour of the refrigerant.

b-c: Cooling and condensation of high pressure refrigerant

c-d: Joule Thomson expansion of saturated liquid refrigerant

d-a: Evaporation of the refrigerant to provide pre-cooling in 2-3

the configuration of the cycle is as follows:

Point a:

T=231.1 K, P=101.325 kPa, h=525.9 kJ/kg

Point b:

T=330 K, P=1410 kPa

Point c:

T=314.4 K, P=1410

Point d:

T=231.1, P=101.325, x=0.5, h=311.5kJ/kg

To obtain the mass flow rate of the propane we will use energy balance for three flow heat exchanger. The mass flow rate of natural gas for 1 tonne of liquified natural gas at dryness fraction 0.6628 is 0.2385kg/sec

$$m_p(h_a - h_d) + mC_p(T_1 - T_6) = mC_p(T_2 - T_3)$$

Substituting the values $m_p = 0.301 kg/sec$

For heat exchanger of propane cooling process b-c:

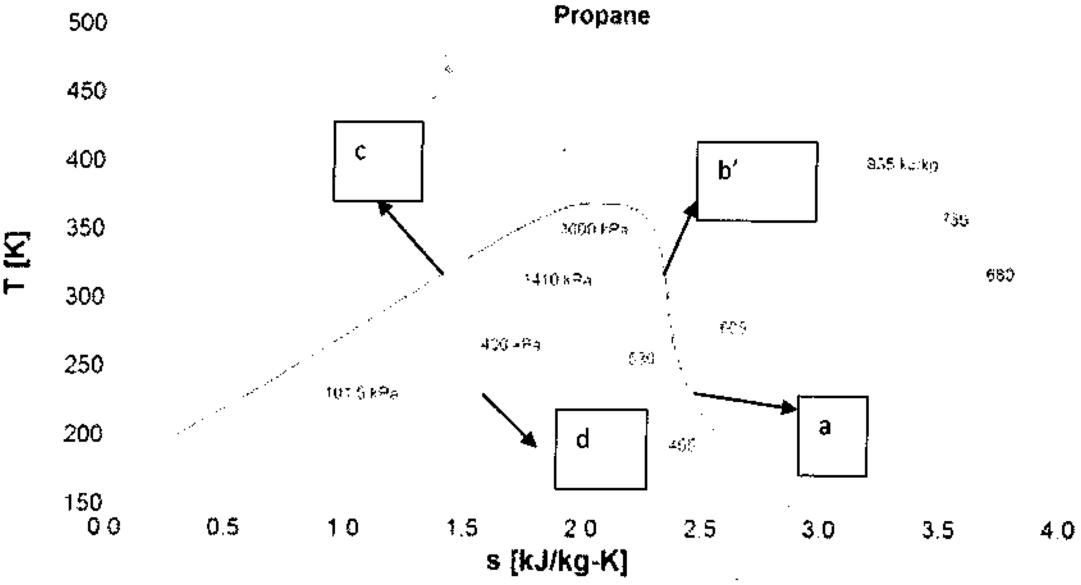


Figure 27: T-s diagram of propane cooling cycle

For the sensible cooling b-b'

$$Q_{b-b'} = m_p C_p dT$$

Substituting the values of $m_p = 0.3kg/sec$, $C_p = 2.194\frac{kJ}{kg-K}$, $T_b = 330\ K$, $T_{b'} = 314.4\ K$

$$Q_{b-b'}\text{=10.2679 kJ/sec}$$

For the process b'-c

Latent heat is removed

$$Q_{b'-c} = (h_{b'} - h_c)$$

Substituting the value $h_{b'} = 615.4\frac{kJ}{kg}$, $h_c = 311.5\ kJ/kg$

$$Q_{b'-c} = 91.17\ kJ/sec$$

Total heat transfer =$Q_{b-b'} +\ Q_{b'-c} = 101.438\ kJ/sec$

This heat is extracted by water. The inlet temperature of water is 300 K and is heated to a temperature of 330 K specific heat of water at constant pressure=4.182 kJ/kg-K for heat exchange of 101.438 kJ/sec

$$Q = \ m_w C_p dt$$

$$101.438 = m_w * 4.18 * (330 - 300)$$

$$m_w = 0.81\ kg/sec$$

$$r = \frac{m_p}{m} = \frac{0.3}{0.2385} = 1.258$$

$$y = \frac{h_1 - h_2}{h_1 - h_f} + r\frac{h_a - h_d}{h_1 - h_f}$$

Substituting the values as $h_1 = 3.143$ kJ/kg , $h_2 = 95.84\frac{kJ}{kg}$, $h_f = 910.9\frac{kJ}{kg}$

$$y = 0.403$$

The work done by the compressor is

$$-\frac{\dot{W}}{\dot{m}} = T_1(s_1 - s_2) - (h_1 - h_2) + r(h_b - h_a)$$

Substituting the value of $s_1 = 0.01139 \frac{kJ}{kg-K}$, $s_2 = 2.617 \frac{kJ}{kg-K}$ and known value of h_1, h_2, r, h_b, h_a

$$-\frac{\dot{W}}{\dot{m}} = 846.2808\ kJ/kg$$

$$W_l = \frac{W}{\dot{m}y} = 2099.95\ kJ/kg$$

Fluid properties are found in Engineering Equation solver by writing a simple code. Refer to appendix 2 for the code and its result.

3.3.7.4 Claude Cycle

The performance of the Linde-Hampson cycle is improved significantly by partially overcoming the inherent irreversibilty in the J-T expansion process for lowering of temperature. In this cycle, the temperature is lowered by using an expansion engine in which not only the temperature can be lowered, but also some work can be generated due to expansion. The expansion is reversible adiabatic process that is isentropic process is utilized for cooling the high pressure gas. About 60 to 80% of the gas is diverted to the expansion engine and is reunited with the return stream after the second heat exchanger. The high pressure gas passes through the second and third heat exchanger and is finally throttled to atmospheric pressure. The vapour from the seprator flows through the heat exchanger in order to cool the incoming gas and is mixed with the expander outlet stream after the second heat exchanger. The return gas at the atmospheric pressure and near-ambient temperature is recycled with the make-up gas. The expansion valve is still needed at the last stage of liquefaction, because the expander cannot efficiently operate with much liquid at the down stream condition. The steps of Claude Liquefaction process are as follows.

Figure 28: Schematic representation of Claude Cycle

1-2: Isothermal compression

2-3: Pre-cooling by the return stream in the first heat exchanger

3-4: Pre-cooling by the return stream in the second heat exchanger

3-e: A part of high pressure gas is isentropically expanded by an expansion engine

4-5: Pre-cooling by the return stream in the third heat exchanger

5-6: Throttling expansion to ambient pressure

6-g: Separation of saturated vapour from the separator

6-f: Separation of saturated liquid from the separator

g-7: Return of saturated vapour at the ambient pressure to provide cooling in4-5

e-8: Mixing of cold exit stream from expansion engine with the return stream

8-9: Return of the mixed stream at the ambient pressure to provide cooling in 3-4

9-1: Return of the mixed stream at the ambient pressure to provide cooling in2-3

Applying enthalpy balance across the outlet of compressor and separator, fraction of liquefaction, is

$$y = \frac{h_1 - h_2}{h_1 - h_f} + x\frac{h_3 - h_e}{h_1 - h_f}$$

Net work requirement is

$$-\frac{\dot{W}}{\dot{m}} = T_1(s_1 - s_2) - (h_1 - h_2) + x(h_3 - h_e)$$

Where,

$$x = \frac{\dot{m}_e}{\dot{m}}$$

The iterations for Claude cycle is done in the software DWSIM. It is a software for process simulation, modeling and optimizing chemical processes.

3.3.7.5 Kapitza Cycle

In the Kapitza cycle, compared to claude cycle having three heat exchangers, first heat exchanger is replaced by two paralle regenerators –one stream is for purifying and cooling the incoming stream and other for preserving the cold low pressure gas. Third low temperature heat exchanger is also eliminated. The reciprocating expansion engine is changed with rotary expansion engine. Rotary expanders are more efficient than reciprocating expansion engines. The kapitza process operates at relatively low pressure, due to the requiurement of regenerators which are designed for approximately 100% effectiveness. The enthalpy balance equations for fraction of liquefaction, work of compression and work of liquefaction are similar to simple Claude Cycle.

Figure 29: Schematic representation of Kapitza Cycle

1-2: Isothermal compression

2-3: Pre-cooling by the return stream in two regenerators by switching alternately

3-4: Pre-cooling by the return stream in the heat exchanger

3-e: A part of high pressure gas is isentropically expanded by an expansion engine

4-5: Throttling expansion to ambient pressure

5-g: Separation of saturated vapour from the separator

5-f: Separation of saturated liquid from the separator

e-g: Return of expander outlet to the saturated vapour from the separator

g-6: Mixing of cold exit stream from expansion engine with the return stream

6-7: Return of the mixed stream through the heat exchanger for cooling in 3-4

7-1: Return of the mixed stream through the regenerator for cooling in 2-3

$$y = \frac{h_1 - h_2}{h_1 - h_f} + x\frac{h_3 - h_e}{h_1 - h_f}$$

$$-\frac{\dot{W}}{\dot{m}} = T_1(s_1 - s_2) - (h_1 - h_2) + x(h_3 - h_e)$$

Where,

$$x = \frac{\dot{m}_e}{\dot{m}}$$

The iterations for Kapitza cycle is done in the software DWSIM. It is a software for process simulation, modeling and optimizing chemical processes.

CHAPTER 4

RESULTS AND DISCUSSIONS

4.1 Simple Linde-Hampson cycle

The points of the cycle can be referred to figure 22

- Point 1:

  P=101.325 kPa, T=300 K, h=3.143 kJ/kg

- Point 2:

  P=10,000 kPa, T=300 K, h=-95.84kJ/kg

- Point 3:

  P=10,000 kPa, T=209.1 K, h=-500kJ/kg

- Point 4:

  P=101.325 kPa, T=111.7 K, h=-500kJ/kg

$$y = \frac{h_1 - h_2}{h_1 - h_f} = \frac{3.143 + 95.84}{3.143 + 940.5} = 0.104$$

$$-\frac{\dot{W}}{\dot{m}} = T_1(s_1 - s_2) - (h_1 - h_2) = 300(0.01139 + 2.617) - (3.143 + 95.84)$$
$$= 689.534$$

$$W_l = \frac{W}{\dot{m}y} = 6630.134\ kJ/kg$$

4.2 Pre-cooled Linde-Hampson cycle

The points of the cycle can be referred to figure 24

- Point 1:

  P=101.325 kPa, T=300 K, h=3.143 kJ/kg

- Point 2:

  P=10,000 kPa, T=300 K, h= - 95.84kJ/kg

- Point 3:

  P=10,000 kPa, T=231.1 K, h= - 358.3kJ/kg

- Point 4:

  P=10,000 kPa, T=196.048 K, h= - 572.4kJ/kg

- Point 5:

  P=101.325 kPa, T=111.7 K, h=-572.4kJ/kg, x=0.6628

- Point g:

  P=101.325 kPa, T=111.7 K, h= - 400.1kJ/kg

- Point 6:

  P=101.325 kPa, T=253.6008 K, h= -98.94 kJ/kg

- Point a:

  T=231.1 K, P=101.325 kPa, h=525.9 kJ/kg

- Point b:

  T=330 K, P=1410 kPa

- Point c:

  T=314.4 K, P=1410

- Point d:

  T=231.1, P=101.325, x=0.5, h=311.5kJ/kg

$$y = \frac{h_1 - h_2}{h_1 - h_f} + r\frac{h_a - h_d}{h_1 - h_f}$$

$$y = 0.403$$

$$-\frac{\dot{W}}{\dot{m}} = T_1(s_1 - s_2) - (h_1 - h_2) + r(h_b - h_a)$$

$$-\frac{\dot{W}}{\dot{m}} = 846.2808\ kJ/kg$$

4.3 Claude Cycle

Figure 30: Schematic representation of Claude

1-2: Adiabatic compression

2-3: Cooler cooling the gas to make the adiabatic compression isothermal

3-4: 1st heat exchanger

4-5: Separator for extracting a part of stream to expand isentropically

6-7: 2nd heat exchanger

7-8: 3rd heat exchanger

8-9: Throttling expansion to ambient pressure

9-13: Separation of saturated vapour from two phases

9-12: Separation of saturated liquid from two phases

4-5: A part of saturated vapour sent to expand isentropically

5-22: A part of high pressure gas sent to isentropically expand through an expansion engine

22-11: Streams from expansion engine and cold stream of heat exchanger to be mixed in mixer

14-10: Return of vapour stream at ambient pressure to provide cooling to 3rd heat exchanger that is cold stream

15-16: Return of vapour stream at ambient pressure to provide cooling to 2nd heat exchanger

17-18: Return of vapour stream at ambient pressure to provide cooling to 1st heat exchanger

The configuration of the cycle are as follows:

Point 1:

T=215.774 K, P=101325 Pa, m=1.02692kg/s, h=-177.84 kJ/kg

Pont 2:

T=557.531 K, P=4000kPa, m=1.02692kg/s

Point 3:

T=300 K, P=4000kPa, m=1.02692kg/s, h=-4.13572 kJ/kg

Point 4:

T=257.048 K, P=4000 kPa, m=1.02692kg/s, h=-90.0855kJ/kg

Point 5(Stream before entry to the expansion engine):

T=257.048 K, P=4000 kPa, m=0.616149kg/s

Point 6:

T=257.048 K, P=4000kPa, m=0.410766kg/s

Point 7:

T=186.068 K, P=4000kPa, m=0.410766kg/s

Point 8:

T=181.118 K, P=4000kPa, m=0.410766kg/s

Point 9:

T=111.611 K. P=101325Pa, m=0.410766kg/s

Point 12 (Vapour Stream):

T=111.611 K, P=101325Pa, m=0.344432kg/s

Point 13 (liquid stream):

T= 111.611 K, P=101325, m=0.0663338kg/s ,h=-903.298kJ/kg

Point 22: Stream after expansion engine

T- 111.611 K, P-101325Pa, m-0.616149kg/s, W= 198.734kW,h=-911.1kJ/kg

Point 14:

T=111.611 K, P=101325Pa, m=0.344432kg/s

Point 10:

T=111.611 K, P=101325 Pa, m=0.327186 kg/s

Point 11:

T=111.611 K, P=101325 Pa, m=0.943336 kg/s

Point 15:

T=111.611 K, P=101325 Pa, m=0.943336 kg/s

Point 16:

T=156.835 K, P=101325 Pa, m=0.927219 kg/s

Point 18:

T= 206.801 K, P=101325 Pa, m=0.927219 kg/s

Point 20:

T=298.15 K, P=101325 Pa, m=0.1 kg/s

$$y = \frac{h_1-h_3}{h_1-h_f} + r\frac{h_4-h_{22}}{h_1-h_f} = \frac{m_f}{m_1} = \frac{0.0664}{1.0269} = 0.0646$$

$$\frac{W}{\dot{m}} = [T_1(s_1 - s_3) - (h_1 - h_3) - x(h_3 - h_{22}) = 884.774 - 198.734 = 686.04\,kW$$

At 75% efficiency of compressor.

4.4 Kapitza Cycle

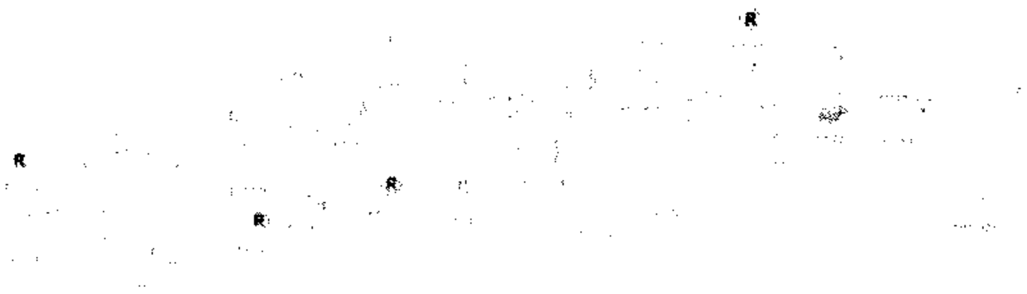


Figure 31: Schematic representation of Kapitza cycle

1-2: Adiabatic compression

2-3: Cooler cooling the gas to make the adiabatic compression isothermal compression

3-4: 1st heat exchanger

4-5: Separator for extracting a part of stream to expand isentropically

5-6: 2nd heat exchanger

6-7: Throttling expansion to ambient pressure

7-8: Separation of saturated vapour from the two phases

7-9: Separation of saturated liquid from the two phases

4-10: A part of high pressure gas sent to expand isentropically

10-11: A part of high pressure gas expanded isentropically in rotary engine

11-12: Return of the expander outlet to the saturated vapour from the separator

8-12: Mixing of cold stream from the expansion engine to the return stream

14-15: Return of mixed stream through the heat exchanger for cooling in 5-6

15-16: Return of mixed stream through the regenerator to provide cooling in 3-4

16-18: Mixing of make-up gas and return streams

The configuration of the cycle are as follows:

Point 1:

T=291.835 K, P=101325 Pa, m=1.35307 kg/s

Pont 2:

T=682.419 K, P=4000 kPa, m=1.39925 kg/s

Point 3:

T=300 K, P=4000 kPa, m=1.39925 kg/s

Point 4:

T=186.068 K, P=4000 kPa, m=1.39925 kg/s

Point 5:

T=186.068 K, P=4000 kPa, m=0.16791 kg/s

Point 6:

T=172.653 K, P=4000 kPa, m=0.16791 kg/s

Point 7:

T=111.611 K, P=101325 Pa, m=0.16791 kg/s

Point 8 (Vapour Stream):

T=111.611 K, P=101325 Pa, m=0.113235 kg/s

Point 9 (liquid stream):

T=111.611 K. P=101325 Pa, m=0.0546754 kg/s

Point 10:

T=186.068 K. P=4000 kPa, m=1.23134 kg/s

Point 11: Stream after expansion engine

T= 111.611 K, P=101325 kPa, m=1.23134 kg/s, h=-495.66 kJ/kg

Point 12:

T=111.611 K, P=101325 Pa, m=1.34458 kg/s,

Point 13:

T= 111.611 K, P=101325 Pa, m=1.34862 kg/s

Point 14:

T=291.802 K, P=101325 Pa, m=1.35276 kg/s

Point 15:

T=298.15 K, P=101325 Pa, m=0.001 kg/s

Point 16:

T=291.835 K, P=101325 Pa, m=1.35307 kg/s

$$y = \frac{h_1 - h_3}{h_1 - h_f} + r\frac{h_{10} - h_{11}}{h_1 - h_f} = \frac{m_f}{m_1} = \frac{0.0546754}{1.35307} = 0.04040$$

$$\frac{W}{m} = [T_1(s_1 - s_3) - (h_1 - h_3) - x(h_3 - h_{22})] = 1558.17 - 193.216 = 1364.954$$

at 75% efficiency of compressor and expander.

CHAPTER 5

CONCLUSIONS

With the increasing energy requirement in the world, a cheap and clean fuel is required to fulfill the appetite. Natural gas is extracted from oil wells. The reservoirs of natural gas are generally found far from the populous locations. The energy requirements need to be fulfilled. This requirement is partly fulfilled by utilizing natural gas. From the study it is found that for long distance beyond, 3500 miles, the cheapest way to transport natural gas is by liquefying the gas. This is the only region for the usage of liquefied natural gas.

In an isenthalpic process the temperature drop is due to non-ideality of the gas; while if the gas is expanded in the expansion engine than the temperature drop is inevitable.

We also observe that the fraction of liquefaction for the same pressure ratio is more in precooled Linde-Hampson cycle than Simple Linde Hampson cycle.

When a comparison is made between Claude cycle and Linde cycle, it is found that the Claude cycle has certain advantages. The pressure ratio is high in Linde Hampson cycle than Claude cycle. Further, there is isentropic expansion of gas which is work producing process.

If the pressure ratio is same for Linde –Hampson cycle and Claude cycle, fraction of liquefaction is more than in Claude cycle.

Also, it can be noted that the specific work for Claude is less than that of simple Linde-Hampson cycle.

Clearly for the same pressure ratio, the fraction of liquefaction is more for the Claude cycle than Pre-cooled Linde-Hampson cycle. The fraction of liquefaction for precooled Linde-Hampson cycle is greater than simple Linde-Hampson cycle.

In the JT-heat exchanger, when pressure increases, the imbalance due to the mismatch in mass flow between the forward and return streams gets compensated by their specific heat imbalance.

CHAPTER 6

FUTURE SCOPE OF WORK

The work done was started from the scratch and required lot of calculations to do. This report surely acts as the raw data for future work. Also Claude cycle and Kapitza cycle are designed in such a way that they can be implemented in real life to liquefy the gas. The experimental setup can be done using these data. The setup should be according to the specification of the thesis. The values can be validated with the values of experimental setup.

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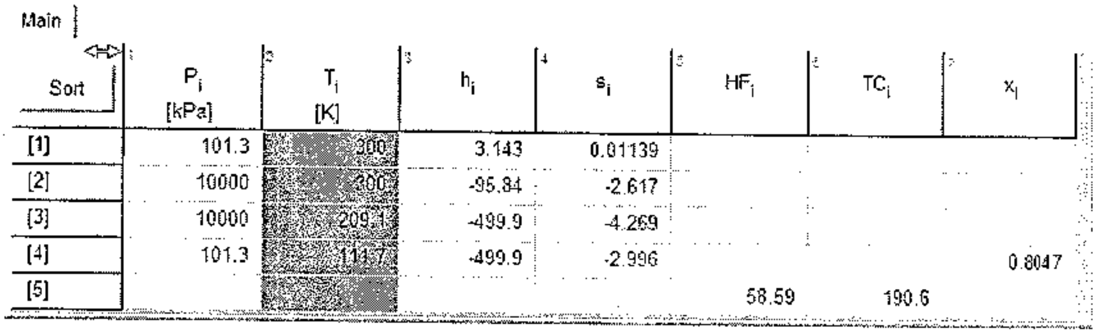
APPENDIX 1

EES SCRIPT FOR CALCULATION OF PROPERTIES FOR SIMPLE LINDE-HAMPSON CYCLE

```
"Point 1"
P[1]=101.325[kPa]
T[1]=300[K]
h[1]=enthalpy(Methane,T=T[1],P=P[1])
s[1]=entropy(Methane,T=T[1],P=P[1])

"point2"
P[2]=10000[kPa]
T[2]=T[1]
h[2]=enthalpy(Methane,T=T[2],P=P[2])
s[2]=entropy(Methane,T=T[2],P=P[2])

"Point3"
P[3]=10000[kPa]
T[3]=209.1[K]
s[3]=entropy(Methane,T=T[3],P=P[3])
h[3]=enthalpy(Methane,T=T[3],P=P[3])

"Point4"
P[4]=101.325[kPa]
h[4]=h[3]
T[4]=temperature(Methane, P=P[4], h=h[4])
s[4]=entropy(Methane,T=T[4],h=h[4])
x[4]=quality(Methane,T=T[4],h=h[4])

"for saturated point"
HF[5]=enthalpy_fusion(Methane)
TC[5]=t_crit(Methane)
s=entropy(Methane,x=0,P=101.325)
h=enthalpy(Methane,x=0,P=101.325)
h6=enthalpy(Methane,T=180,P=10000)
```

```
h7=enthalpy(Methane,T=200,P=10000)
h8=enthalpy(Methane,T=220,P=10000)
v6=volume(Methane,T=180,P=10000)
v7=volume(Methane,T=200,P=10000)
v8=volume(Methane,T=220,P=10000)
cp=cp(Methane,T=200,P=10000)
```

Table VII: Result achieved after performing the calculations in EES



Sort	P_i [kPa]	T_i [K]	h_i	s_i	HF_i	TC_i	x_i
[1]	101.3	300	3.143	0.01139			
[2]	10000	300	-95.84	-2.617			
[3]	10000	209.1	-499.9	-4.269			
[4]	101.3	111.7	-499.9	-2.996			0.8047
[5]					58.59	190.6	

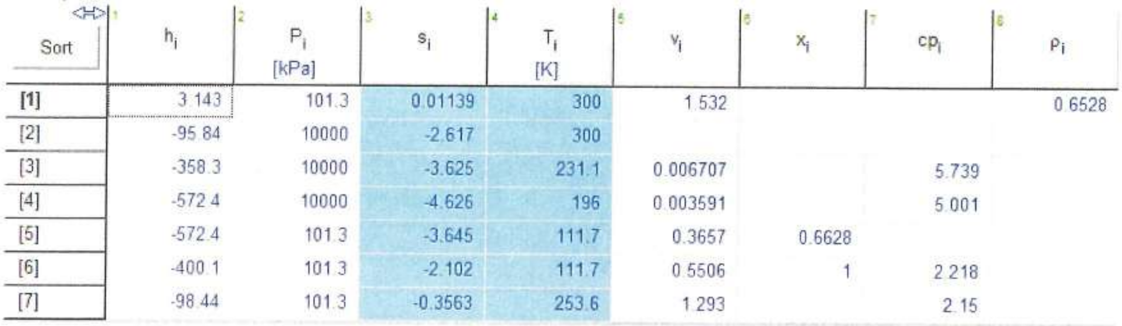
APPENDIX 2

EES SCRIPT FOR CALCULATION OF PROPERTIES FOR PRE-COOLED LINDE-HAMPSON CYCLE AND PROPANE CYCLE

```
DELTAh_vap=enthalpy_vaporization(Methane,P=101.325)
P[1]=101.325[kPa]
T[1]=300[K]
h[1]=enthalpy(Methane,T=T[1],P=P[1])
s[1]=entropy(Methane,T=T[1],P=P[1])
v[1]=volume(Methane,T=T[1],P=P[1])
rho[1]=density(Methane,T=T[1],P=P[1])

P[2]=10000[kPa]
T[2]=T[1]
h[2]=enthalpy(Methane,T=T[2],P=P[2])
s[2]=entropy(Methane,T=T[2],P=P[2])

P[3]=P[2]
T[3]=231.1[K]
h[3]=enthalpy(Methane,T=T[3],P=P[3])
s[3]=entropy(Methane,T=T[3],P=P[3])
v[3]=volume(Methane,T=T[3],P=P[3])
cp[3]=cp(Methane,T=T[3],P=P[3])

"
P[7]=101.325[kPa]
x[7]=1
T[7]=temperature(Methane,P=P[7],x=x[7])
h[7]=enthalpy(Methane,x=x[7],P=P[7])
s[7]=entropy(Methane,P=P[7],x=x[7])
v[7]=volume(Methane,P=P[7],x=x[7])
cp[7]=cp(Methane,T=T[7],P=P[7])
"
P[4]=10000[kPa]
T[4]=196.048[K]
```

```
s[4]=entropy(Methane,T=T[4],P=P[4])
h[4]=enthalpy(Methane,T=T[4],P=P[4])
cp[4]=cp(Methane,T=T[4],P=P[4])
v[4]=volume(Methane,T=T[4],P=P[4])

P[5]=101.325[kPa]
h[5]=h[4]
T[5]=temperature(Methane, P=P[5], h=h[5])
s[5]=entropy(Methane,T=T[5],h=h[5])
x[5]=quality(Methane,T=T[5],h=h[5])
v[5]=volume(Methane,T=T[5],x=x[5])

P[7]=101.325
T[7]=253.6008
cp[7]=cp(Methane,T=T[7],P=P[7])
h[7]=enthalpy(Methane,T=T[7],P=P[7])
s[7]=entropy(Methane,T=T[7],P=P[7])
v[7]=volume(Methane,T=T[7],P=P[7])

P[6]=101.325[kPa]
x[6]=1
T[6]=temperature(Methane,P=P[6],x=x[6])
h[6]=enthalpy(Methane,x=x[6],P=P[6])
s[6]=entropy(Methane,P=P[6],x=x[6])
v[6]=volume(Methane,P=P[6],x=x[6])
cp[6]=cp(Methane,T=T[6],x=x[6])

hf1=enthalpy(Methane,x=0,P=101.325)
HF=enthalpy_fusion(Methane)

h=enthalpy(Methane,T=220,P=4000)
T=temperature(Methane,P=101325,h=h)
```

Table VIII: Result achieved after performing the calculations in EES



Sort	h_i	P_i [kPa]	s_i	T_i [K]	v_i	x_i	cp_i	ρ_i
[1]	3.143	101.3	0.01139	300	1.532			0.6528
[2]	-95.84	10000	-2.617	300				
[3]	-358.3	10000	-3.625	231.1	0.006707		5.739	
[4]	-572.4	10000	-4.626	196	0.003591		5.001	
[5]	-572.4	101.3	-3.645	111.7	0.3657	0.6628		
[6]	-400.1	101.3	-2.102	111.7	0.5506	1	2.218	
[7]	-98.44	101.3	-0.3563	253.6	1.293		2.15	

```
T[1]=231.1[K]
x[1]=1
h[1]=enthalpy(Propane,T=T[1],x=1)
s[1]=entropy(Propane,T=T[1], x=1)
v[1]=volume(Propane,T=T[1],x=x[1])
P[1]=pressure(Propane,T=T[1],v=v[1])

s[1]=s[2]
T[2]=330[K]
P[2]=pressure(Propane,T=T[2],s=s[2])
h[2]=enthalpy(Propane,T=T[2],P=P[2])
v[2]=volume(Propane,T=T[2],P=P[2])
cp[2]=cp(Propane,T=T[2],P=P[2])


P[3]=P[2]
x[3]=0
T[3]=temperature(Propane,P=P[3],x=x[3])
h[3]=enthalpy(Propane,T=T[3],x=x[3])
v[3]=volume(Propane,T=T[3],x=x[3])
s[3]=entropy(Propane,T=T[3],x=x[3])
"cp[3]=cp(Propane,T=T[3],P=P[3])"
h6=enthalpy(Propane,T=T[3],x=x[1])

T[4]=231.1[K]
h[4]=h[3]
```

```
P[4]=pressure(Propane,T=T[4],h=h[4])
s[4]=entropy(Propane,T=T[4], h=h[4])
v[4]=volume(Propane,T=T[4],h=h[4])
"cp[4]=cp(Propane,T=T[4],h=h[4])"
x[4]=quality(Propane,T=T[4],h=h[4])

 DELTAh_vap=enthalpy_vaporization(Propane,T=T[4])
cp1=cp(Water,T=330[K],P=101.325[kPa])
cp2=cp(Water,T=300[K],P=101.325)
```

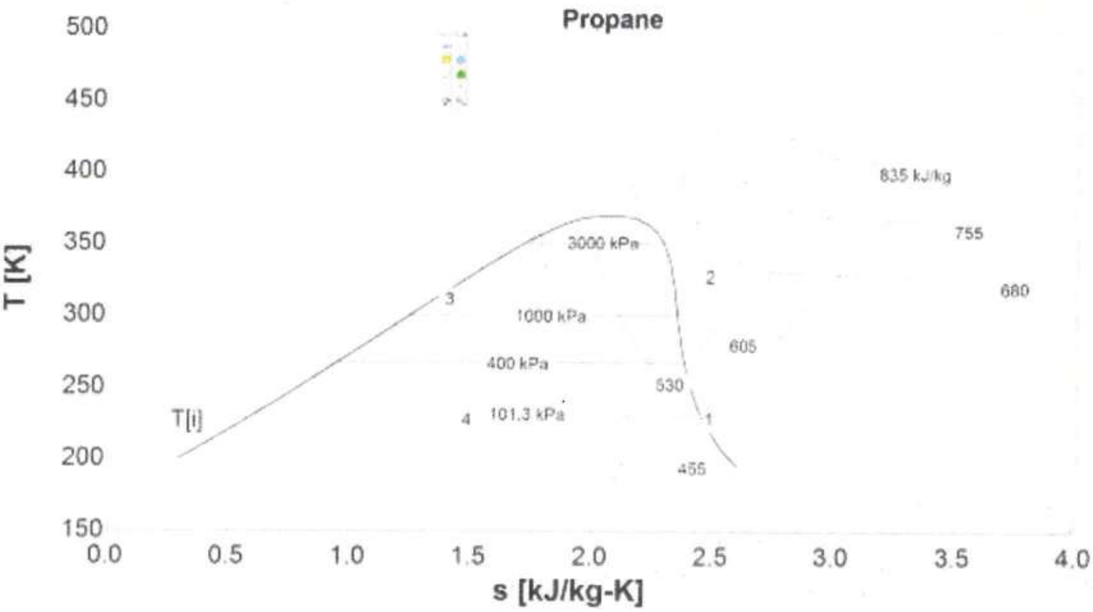


Figure 32: T-s graph of propane cycle