ANALYSIS AND OPTIMIZATION OF PROCESS PARAMETERS OF HEAT EXCHANGERS AND TURBINES FOR HELIUM REFRIGERATOR/LIQUEFIER

A THESIS SUBMITTED IN PARTIAL FULFILLMENT OF THE
REQUIREMENTS FOR THE DEGREE OF

Master of Technology
In
Mechanical Engineering

By

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2015

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CERTIFICATE

This is to certify that the thesis entitled, "**Analysis and Optimization of Process Parameters of Heat Exchangers and Turbines for Helium Refrigerator/Liquefier**" submitted to **National Institute of Technology, Rourkela** by **Ashutosh Mishra,** Roll No**. 213ME5449** f**or** the award of the Degree of **Master of Technology** in **Mechanical Engineering** with specialization in**" Cryogenic and Vacuum Technology"** is a record of bonafide research work carried out by him under my supervision and guidance. The results presented in this thesis have not been, to the best of my knowledge, submitted to any other University/ Institute for the award of any degree or diploma.

The thesis, in my opinion, has reached the standards fulfilling the requirement for the award of degree of **Master of Technology** in accordance with regulations of the Institute.

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This is to certify that the dissertation, entitled

"Analysis and Optimization of Process Parameters of Heat Exchangers and Turbines for Helium Refrigerator/Liquefier"

Is a bonafide work done by

Ashutosh Mishra

Under my close guidance and supervision in the Large Cryogenic Plant and Cryosystem Group

Of

Institute for Plasma Research, Gandhinagar, Gujarat

For the partial fulfillment of the award of the Degree of **Master of Technology** in **Mechanical Engineering** with specialization in "**Cryogenic and vacuum Technology**"

at

National Institute of Technology, Rourkela.

The work presented here, to the best of my knowledge, has not been submitted to any university

For the award of similar degree.

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ACKNOWLEDGEMENT

I avail this opportunity to express my indebtedness to my guide **Mr. A.K. Sahu**, Divisional Head of large cryogenic plant and Cryo system, Institute for Plasma Research, Bhat, Gandhinagar, Gujarat, and co-guide **Prof. R.K. Sahoo**, Mechanical Engineering Department, National Institute of Technology, Rourkela, for their valuable guidance, constant encouragement and kind help at various stages for the execution of this dissertation work. I consider myself fortunate to have worked under their supervision. I also express my sincere gratitude to **Prof. S.K. Sarangi**, Director of NIT Rourkela and **Prof. S.S. Mohapatra**, Head of The Department of Mechanical Engineering at NIT Rourkela for providing valuable institute and departmental facilities. I would like to express my reverent thankfulness to my colleagues and classmates as well as all the staff members from both the reputed institutes for their help and support throughout the project.

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ABSTRACT

A Helium liquefier and refrigerator is very vital component of many superconducting magnets, fusion devices, Tokamaks etc. So it is very important to optimize helium liquefier/refrigerator. This project work involves analysis and optimization of the process parameters (helium flow rate, pressure and temperature) for main components (8 different heat exchangers and 3 different turbo-expanders) of helium plant of refrigeration capacity 1 kW at 4.5 K. This is a part of the indigenous helium plant development work going on at IPR, Bhat, Gandhinagar, Gujrat. Nevertheless, this plant can be operated in mixed mode also as helium refrigerator-cum-liquefier (HRL), although it is optimized for refrigeration load. To optimize process of any helium refrigeration/liquefaction cycle, it is very important to consider one independent variable at a time and under valid assumptions, study and analyze its effect on the process. From the analysis, the optimized value of the concerned and considered process variable is selected. The main components of an HRL that affect process parameters are compressor, heat exchangers ,expansion engines and expansion valves .The present analysis is basically concerned with the parameters of the heat exchangers vis –a-vis expansion engines. In the present analysis mainly total compressor mass flow rate, fraction of total compressor mass flow diverted towards expansion engines (turbo expanders), inlet temperature to various expansion engine and heat exchangers are analyzed and optimized using steady state approach. The present study, analysis and optimization of the important process parameters is done taking logical assumptions and fulfilling important practical constraints that are explained in this report. The work involves:

- Study different thermodynamic configurations of HRL.
- Study the HRL, existing at IPR.
- Study different component working principle and design aspects also.
- Study and analyze different practical factors and inefficiencies of main components that can affect the performance of HRL.
- Find out different possible methods to analyze the given thermodynamic configuration to find liquefaction and refrigeration capacity.

- Choose the best method from point-4 and make a computer code to analyze and get the HRL performance.
- Generate different graphical trends from analysis and optimization for variation of different process parameters of components.
- Find the optimum process parameters of main components.

Contents

List of Figures

List of Tables

CHAPTER -1

1. INTRODUCTION

1.1. General Preamble

If cryogenics is to be defined in terms of liquefaction systems, it is science of liquefying the presumed permanent gases mainly air and its major components like nitrogen, oxygen and hydrocarbon gases and other important phenomenon related to same temperature range. Generally preferred temperature point that differentiates cryogenics from normal low temperature phenomena is 123 K.

Helium liquefaction is considered tougher than liquefaction of other gases. It is mainly due to its rare availability as compared to other gases and its much lower boiling point temperature. Even though, helium liquefaction is very important .Some of the main aspects and basic need of helium liquefaction systems is explained in the coming sections.

1.2. Why helium?

The boiling point of liquid helium is approximately lowest amongst all the earlier called permanent gases. It is about 4.2 K at 1 bar pressure .With the increase in demand of fusion reactors and superconducting magnets, the requirement of high cooling rate at lower and lower temperature also increased. There are two ways of cooling a system. The first one is by utilizing the sensible cooling capacity of the refrigerant while the second one is by utilizing the latent cooling capacity of the saturated refrigerant. Generally in helium liquefier and refrigerator systems, major portion of the sensible cooling capacity of the refrigerant return stream is utilized in precooling the hotter refrigerant stream and the latent cooling is utilized to absorb the transient and steady heat loads in the superconducting magnets, fusion reactors or TOKAMAK. To utilize the latent cooling capacity of any refrigerant firstly it has to be liquefied and thus comes the prime importance of a Helium Liquefier and Refrigerator.

1.3. About Helium

Generally when helium is referred it is implicitly He^4, the dominant one (in quantity) between the two stable isotopes of helium. The other one is He^3, only found in ppm levels in normal helium gas. It is about 1.3 ppm in natural helium [1].The normal boiling point of liquid helium is about 4.216(at 1 bar pressure).It does not freeze under its own vapor pressure at all. It is odorless and colorless. The entire liquid-vapor coexistence of the normal fluid takes place between 2.1768 and 5.1953 Kelvin, or within 3 degrees only. For other fluids the liquid-vapor coexistence spans a much larger range of temperature. As a consequence of helium having such a short two-phase boundary, any temperature change for a liquid-vapor helium system is amplified in importance by at least an order of magnitude as compared to other fluids [2] .Normally the liquid helium is found in two distinct liquid phases. The first one is termed as type 1 whereas the other being more important in heat transfer business is termed as type 2 or *superfluid helium.* The temperature point (pressure vs temperature plot) at which the normal type 1 helium transits to type 2 superfluid helium is known as lambda point. It is primarily due to the resemblance of the specific heat vs temperature plot of liquid helium with the Greek letter.

- **CRITICAL POINT** [2]

TEMPERATURE = 5.1953 KELVIN
PRESSURE= 0.2275 MEGA PASCAL
DENSITY = 17.399 MOLES/LITER

LAMBDA POINT

TEMPERATURE =2.1768 KELVIN
PRESSURE= 0.50418E-2 MEGA PASCAL
LIQUID DENSITY =36.514 MOLES/LITER

MOLECULAR WEIGHT= 4.0026

1.4. General liquefaction cycles

1.4.1 Thermodynamically ideal system

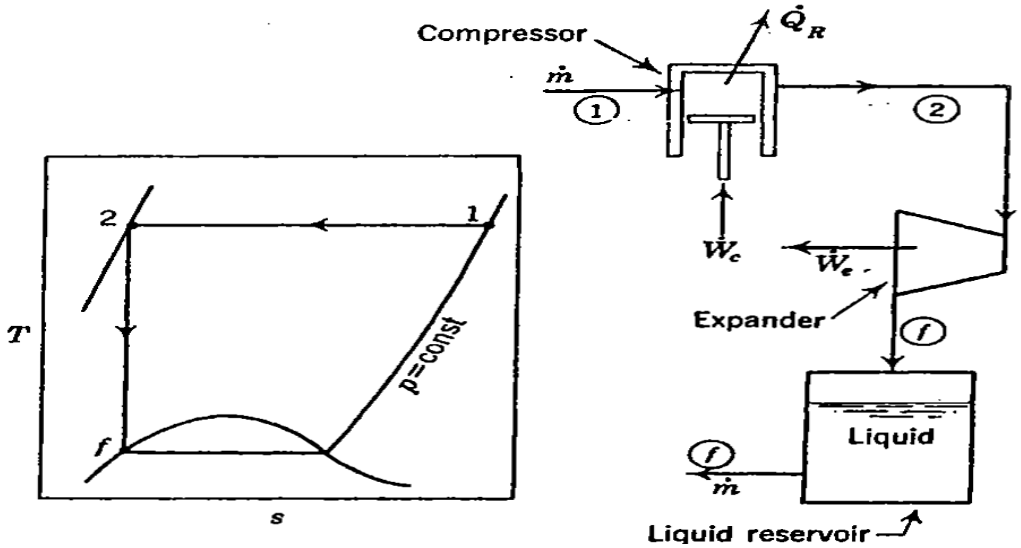

Figure1.1 T-S diagram and schematic diagram of Thermodynamically Ideal Cycle [1]

The system shown in the above figure is considered as thermodynamically ideal liquefaction cycle.

In this, isothermal compression is followed by isentropic expansion to saturated state. Thereafter liquid is taken out for application purpose.

But it is practically not feasible.

Reason:

The gas is required to be isothermally compressed to a pressure from where it can be taken up to saturated state by isentropic expansion. Generally such a pressure is too high to attain using available compressors.

This is thermodynamically ideal liquefaction cycle as the first two processes are same as Carnot cycle. This Cycle is generally used for comparison purpose.

1.4.2 Cascade system

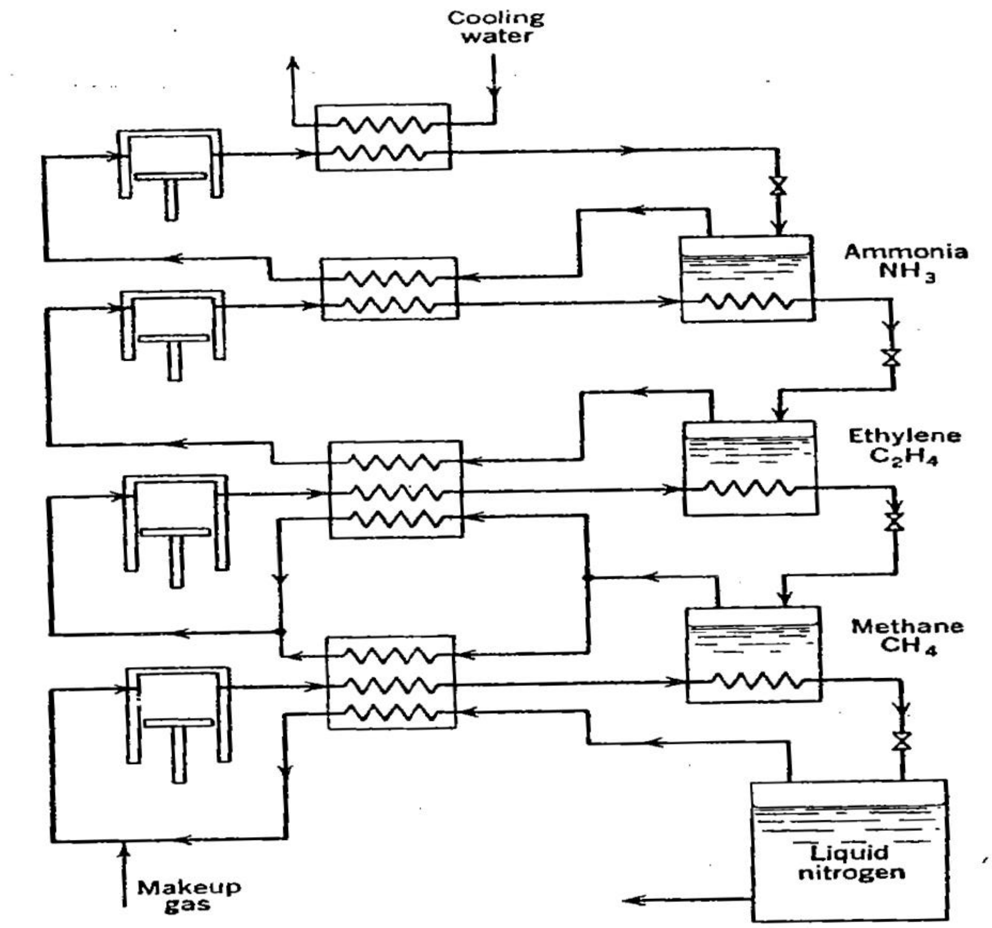

Figure1.2 Schematic Diagram of Cascade System [1]

This liquefaction system is thermodynamically close to it the ideal cycle. In this, a secondary refrigerant combination brings the primary refrigerant to a temperature from where it can be liquefied using isenthalpic expansion valve.

To elaborate further, in the above figure:

Ammonia is liquefied →liquefied ammonia bath precools ethylene → liquid ethylene is produced using isenthalpic expansion valve → liquefied ethylene bath precools methane →liquid methane is produced using isenthalpic expansion valve → liquefied methane bath precools nitrogen →liquid nitrogen is produced using isenthalpic expansion valve.

In the above example, the primary concern is liquid nitrogen for which this whole cascade precooling system is utilized.

Main Advantages:

1. Low pressure requirement (almost 20-30 atm for nitrogen).

2. Thermodynamically close to ideal system, so efficient.

Main disadvantages:

1. Complex precooling loops.

2. Maintenance issues.

Importance:

The precooling concept is highly useful as it makes the system more efficient and is used in many liquefaction processes.

1.4.3 Linde cycle:

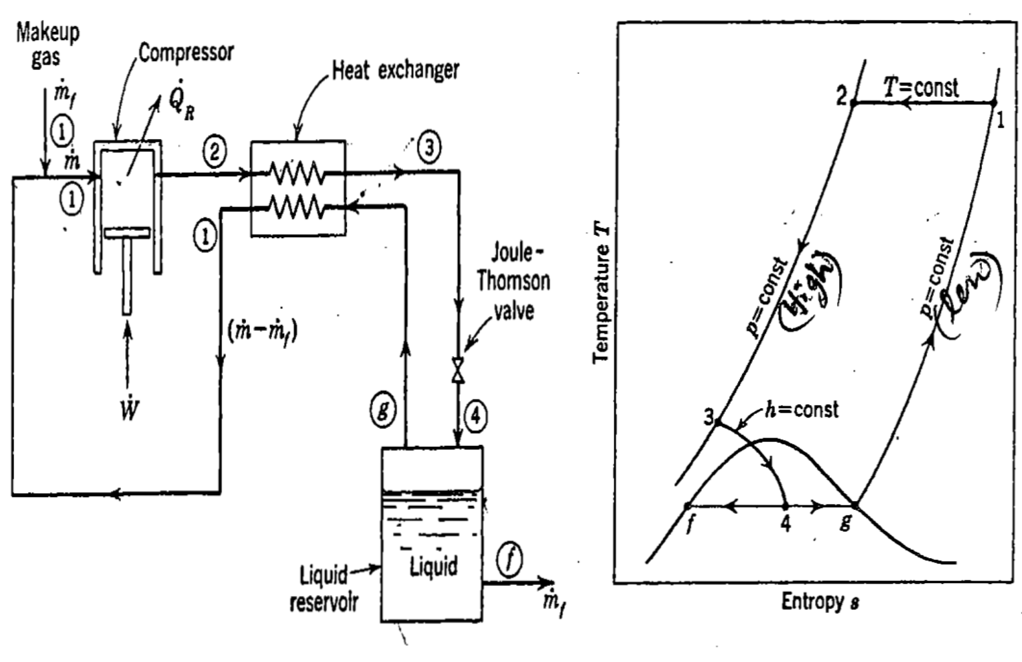

Figure1.3 Schematic diagram and T-S diagram of Linde Cycle [1]

This system or cycle includes following process:

1-2: isothermal compression (after cooling adiabatic compression)

2-3: and g-1 is the mutual heat exchange process that is ideally carried out at constant pressure P_2 and P_1 respectively.

3-4: isenthalpic expansion in the expansion valve.

At 4 liquid and gas are separated .liquid is taken out whereas the gas is used for precooling the gas from 2-1.

This is a very simple system generally used to liquefy air or nitrogen having inversion temperature above ambient temperature. It cannot be directly used to liquefy helium.

Reason:

Helium gas has maximum inversion temperature much below ambient temperature (about 40 K). So the isenthalpic expansion valve will heat the gas instead cooling it.

So, if this system is to be used for helium liquefaction, hydrogen or neon precooling has to be provided.

1.4.4 Claude cycle:

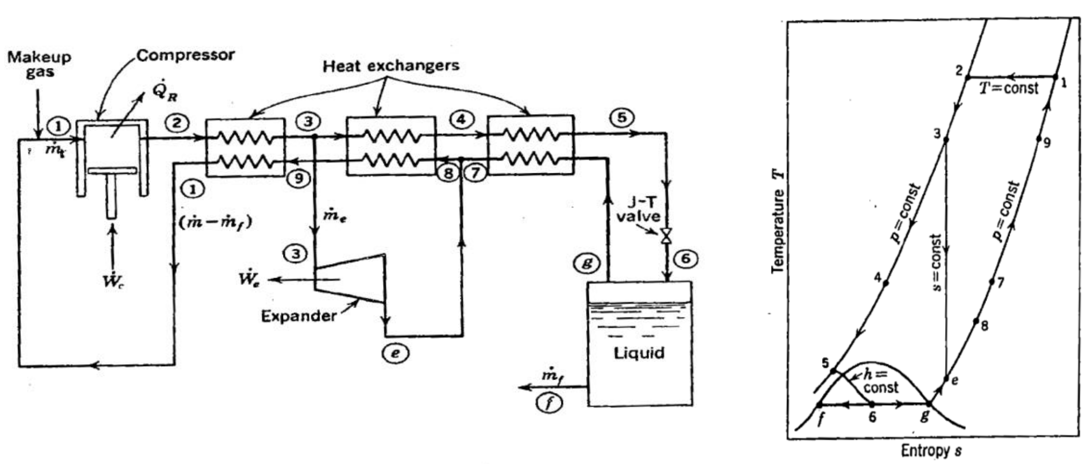

Figure 1.4 Schematic diagram and T-S diagram of Claude Cycle [1]

This cycle has introduced a very important and decent concept to liquefaction system by adding expansion engine. The hot gas stream is allowed to do some work in the expansion engine. So in this way it approaches the thermodynamically ideal system. Expansion through engine is thermodynamically better comparative to expansion through valve. Expansion through valve is more irreversible than through engines.

Expansion valve:

Main advantage:

1. Lower cost and design complexity.
2. Can easily handle two phase flow or phase change during expansion.

Main disadvantage:

1. Irreversible expansion, so thermodynamically less preferable.
2. To use it for cooling purpose ,the gas has to be below its inversion temperature at that pressure .This imposes an extra liability of first precooling the gas up to the required inversion temperature.

Expansion engine:

Main advantage:

1. External work is done which may or may not be used in compression.
2. Polytrophic expansion is thermodynamically better than the earlier one.

Main disadvantages:

1. Complex design and comparative higher cost.
2. Phase change during expansion poses a serious problem on blades, so phase change during expansion is not preferable.

Conclusion:

To use expansion engine optimally for expansion and to use expansion valve as close as possible to the saturation region.

It consists of following processes:

1-2: isothermal compression (polytrophic expansion, thereafter cooling)

2-3 &9-1: mutual heat exchange in the first heat exchanger.

At point 3: an optimum fraction of the initial mass flow is diverted to the expansion engine.

3-e: the gas is expanded in the expansion engine.

3-4 & 8-9: ideally mutual heat exchange process at constant pressure.

4-5 &g-7: ideally mutual heat exchange process at constant pressure.

The ‘e’ stream i.e. outlet of expansion engine is adiabatically mixed with the cold outlet of third heat exchanger i.e. ‘7’.

5-g- : the gas is expanded in the expansion valve. This takes it two phase region, from where the two phases ‘f’& ‘g’ are separated. ’g’ is returned to precool the incoming hot gas whereas ‘f’ is taken out or application.

1.4.5 Collins cycle:

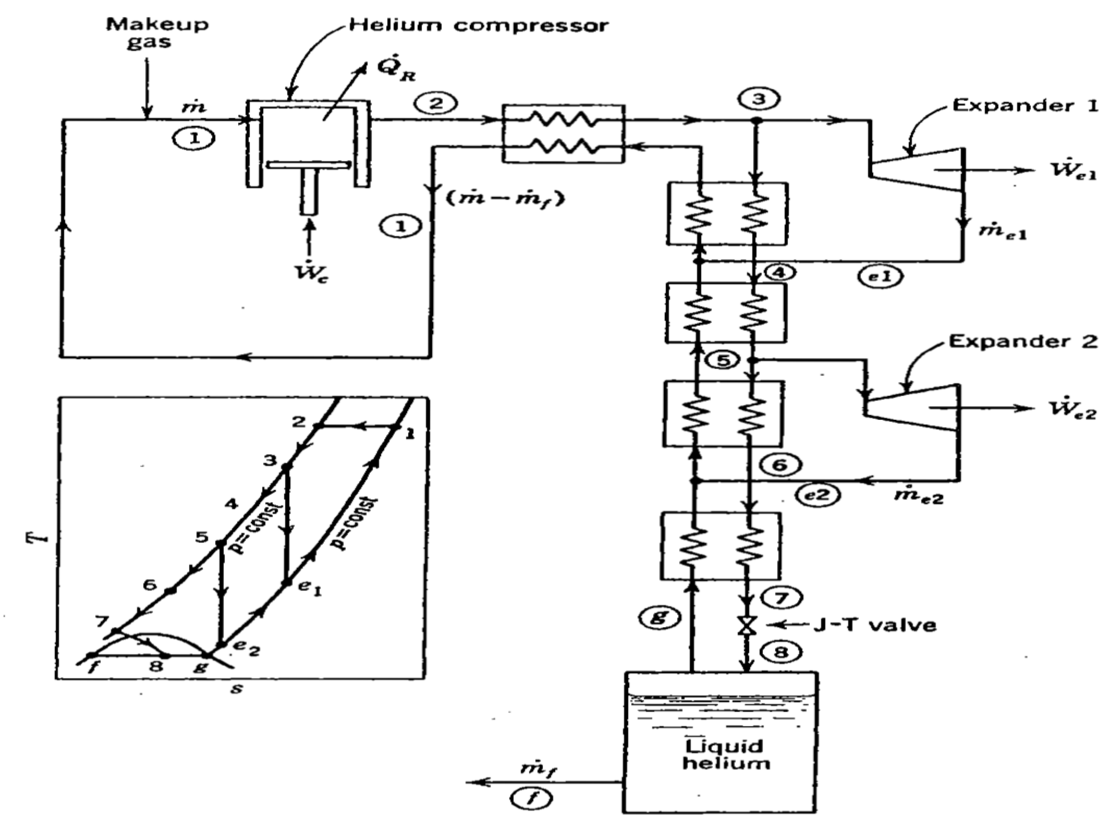

Figure 1.5 Schematic diagram and T-S diagram of Collins Cycle[1]

This system is an extension or modification of Claude system. It uses heat exchangers and multiple expansion engines along with JT valve in the configuration as shown in the above figure.

Working is similar to Claude system .In addition it consists of more number of expansion engines and to retain the cooling capacity of return stream more number of heat exchangers.

In the above configuration ,two expansion engines are operating in parallel and their outlet streams are mixed with the return stream while the main stream is expanded through JT valve and is precooled using other expansion engines and heat exchangers.

1.4.6 Present configuration:

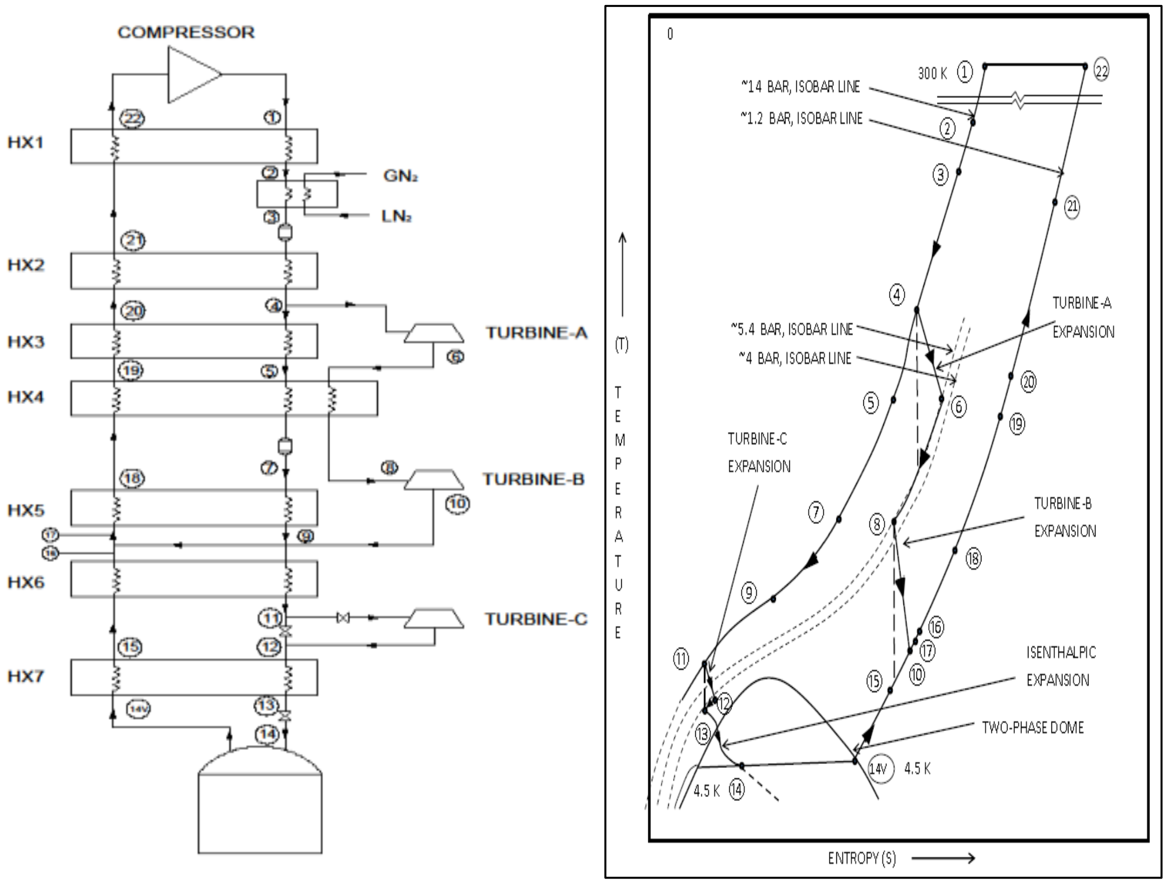

Figure 1.6 Schematic and general T-S diagram of the present configuration

The present configuration is also an extension or modification of the Claude system. In the above configuration three expansion engines are used. In which the first two operate in series to expand the diverted stream whereas the third expansion engine provides precooling to the main stream before expansion through JT valve. Also seven heat exchangers are used at judicial pinch points to retain the cooling capacity of the return stream efficiently. In addition there is a provision of liquid nitrogen precooling after the first heat exchanger.

The detailed process is as follows:

COMPRESSOR

22-1: isothermal compression (polytrophic screw compression and then after cooling).

HX1

1-2 & 21-22 : mutual heat excahnge process between hot helium gas coming from compressor and cold helium gas return stream from HX 2.

LN_2

2-3: Liquid Nitrogen precooling.

HX 2

3-4 & 20-21: mutual heat excahnge process between hot helium gas coming from LN_2 and cold helium gas return stream from HX 3 .

At point ‘4’ : helium gas from HX 2 outlet is divide into streams.the main stream and the diverted stream to first expansion engine.

TURBINE A

4-6: The diverted helium gas is expanded in the first expansion engine upto some optimized intermediate pressure.

HX 3

4-5 & 19-20: mutual heat excahnge process between hot helium gas coming from LN_2 and cold helium gas return stream from HX 4.

HX 4

5-7, 6-8 & 18-19: Three stream heat exchange. Can be subdivided among two heat exchange processes; first between cold return stream from HX 5 and high pressure hot stream from HX 3.while second between cold return stream from HX 5 and medium pressure hot stream from turbine A outlet.

TURBINE B

8-10: the medium pressure stream is expanded in expansion engine up to return stream pressure.

The outlet from second expansion engine i.e. ‘10’ is adiabatically mixed with the HX 6 cold outlet ‘16’ to final state ‘17’.

HX 5

7-9 & 17-18 : :mutual heat excahnge process between hot helium gas coming from HX 4 and cold helium gas return stream at ‘17’.

HX 6

9-11 & 15-16 : mutual heat excahnge process between hot helium gas coming from HX 5 and cold helium gas return stream from HX 7.

TURBINE C

11-12 : thehot oulet from HX 6 is expanded in third expansion engineupto some optimised intermediate pressure.

HX 7

12-13 & 14v-15 : mutual heat excahnge process between hot helium gas coming from third expansion engine and cold helium gas return stream at at almost saturated vapour condition.

JT VALVE

13-14 : Finally the helium gas is expanded in the isenthaphic valve to the return stram pressure.

At point 14 the vapour helium and the liquid helium is separated.

CHAPTER -2

2. LITERATURE REVIEW

Thomas et al. [3] have analyzed the role of heat exchangers parameters on Collins cycle using Aspen HYSYS® V7.0, assuming, steady state operation, efficiency being independent of pressure ,temperature and mass flow, neglecting pressure drop in pipe lines and heat exchangers and accounting heat leak /loss and other important factors in the definition of UA itself. It is concluded that the heat exchangers at the inlet of the expanders have comparatively greater impact on liquid production; rest exchanger effectiveness is linearly related to the liquid production. When effectiveness of the concerned heat exchanger is varied keeping effectiveness of other heat exchangers at .95 ,the refrigeration or the liquid production changes drastically only below .5 effectiveness. Also it is found that there is a limiting UA for every heat exchanger and increasing their effectiveness has no effect on deciding the optimum mass flow through the expander Vis –a –Vis heat exchangers.

Thomas et al. [4] have performed parametric studies on Collins cycle using aspen HYSYS, taking expander as primary component for analysis. Under nearly the same as above assumptions, one considered parameter is changed or varied while the other important ones are fixed or kept constant at the mentioned base values. When compressor suction and discharge pressure is 14 bar and 1.01 bar respectively, compressor efficiency is 60%, efficiency of turbine 1 and turbine 2 is 70%, while the heat exchanger effectiveness is .97, it has been found that maximum liquid production is obtained when 80% of the compressor flow is equally distributed between both expanders. Also it depends linearly on expander efficiency.

Atrey [5] has analyzed Collins helium liquefaction cycle. It has been found that there is an optimum mass flow through expansion engine for which it shall provide maximum yield per unit work input. First to find nodal point temperatures, a computer program has been developed that uses steady state energy balance and bisection method for liquid yield as convergence criteria. It has been found for 35% and 45 % of the total compressor mass flow through expander 1 and expander 2 respectively, the cycle produces maximum yield per unit work input. Efficiency of the expansion engines or expanders has linear effect on liquid yield. Whereas effectiveness of the heat exchangers does not affect the optimum mass flow through the expansion engines but do affect the minimum mass flow for liquefaction. Also increasing effectiveness of the heat exchangers just before the expansion engines and in between them is found more advantageous than others.

Lilong et al. [6] have analyzed the Claude Cycle of the EAST Cryogenic system. A computer program has been developed to analyze the same. Also the analysis and results have been validated through experiments.

For mode 1000 W/4.4 K+ 3.5 g/s LHe, it has been concluded that there is sufficient margin in heat exchangers UA value. Up to 1.5 times of the designed value the plant capacity increases significantly. But after 1.5 times, the capacity curve gets smoother. Whereas for first heat exchanger HX I, eighth heat exchanger HX VIII (liquid nitrogen precooling) increase in designed UA does not affect the plant capacity. Also since the analysis is concerned about up gradation of an existing system, increase in heat transfer area is out of consideration .So, effect of mass flow on exchanger's efficiency is analyzed. It is validated that with increase in mass flow rate, UA increases but the NTU decreases .Due to the same, the efficiency decreases. It is concluded that if mix mode operation is converted to refrigeration mode, the equivalent refrigeration decreases due to the constraints of the heat exchange process.

Also the turbine efficiency has direct effect on plant capacity, almost linear. For operational mode of 760 W/4.5 K+ 560 W /4.3 K =16 g/s LHe, the optimum mass flow is 224.5 g/s with a mass diversion ratio of .5 at the first turbine.

Jadhav[7] has analyzed the effect of compressor outlet pressure on Helium refrigerator/Liquefier and optimized the same. It has been found that there is an optimum outlet pressure at which the plant gives maximum refrigeration. In the analysis, for 140 g/s compressor flow and 210 g/s compressor flow 17.1 bar and 16 bar have been concluded optimum.

CHAPTER -3

3. METHODOLOGY

General Assumptions

- Most of the time the first heat exchanger HX1,which exchanges heat between hot helium gas (coming from compressor and oil removal system CORS) at nearly 310 K and the cold helium gas coming out of the second exchanger, has not been included in the analysis.

Reason: it is observed and thus presumed that the liquid nitrogen precooling heat exchanger brings down the hot outlet temperature of the helium gas from HX 1 to 80 K. Therefore irrespective of the hot stream outlet temperature from HX 1, the hot stream inlet temperature to HX 2 is 80 K. Thus the analysis to optimize various process parameters begins with HX 2.

- The system is considered to be at steady-state.
- GASPAK v3.35 is used to calculate helium properties.
- Pressure drop in most of the heat exchangers is neglected.
- Judicial heat leak is assumed in every heat exchanger.

Abbreviations Used

P_{hot}= hot stream pressure

P_{cold}= cold stream pressure

m_h= mass flow in hot stream

m_c= mass flow in cold stream

T_{hin} = hot stream inlet temperature

T_{hout} =hot stream outlet temperature

T_{cin} = cold stream inlet temperature

T_{cout} = cold stream outlet temperature

h_{in}= hot stream inlet enthalpy

h_{out} =hot stream outlet enthalpy

h_{cin} = cold stream inlet enthalpy

h_{cout} = cold stream outlet enthalpy

η= Efficiency

$hout\ s$ = enthalpy obtained through isentropic expansion at the outlet of turbine

s_{in}=inlet stream entropy

s_{out}= outlet stream entropy

lmtd= Log mean Temperature Difference

Formulae Used

I. $$\sum_{i=1}^{n}(m_c*(h_{cout}-h_{cin}))=\sum_{i=1}^{n}(m_h*(h_{in}-h_{out}))+heatleak$$

II. $$\eta=\frac{h_{in}-h_{out}}{h_{in}-h_{out\,ideal}}$$

III. $liquid\ fraction = 1 - vapor\ fraction$

IV. *Total liquid produced=liquid produced*mass low at JT inlet*

V. *refrigeration =latent heat *mass of helium gas used in application*

VI. *Q=UA*lmtd*

3.1. Method opted to optimize mass flow through turbine A vis-a-vis HX 3

For given compressor pressure and mass flow:

HX 2

Hot stream inlet temperature =80.01

Reason: outlet from liquid nitrogen precooling heat exchanger and 80 K absorber bed.

Hot stream outlet temperature = inlet temperature of turbine A

Cold stream inlet temperature = calculated from HX 3

Cold stream outlet temperature = f (P_{cold}, h_{cout})

using formulae I,

$$h_{cout} = h_{cin} + \frac{(m_h * (h_{in} - h_{out})) + heatleak}{m_c}$$

Turbine A

Inlet temperature = user defined and fixed while changing mass flow rate to find the optimum mass flow through turbine A vis-à-vis HX 3 and JT valve.

Pressure drop =judicial pressure drop considering the manufacturing or designing factors in view. A separate analysis may be done to optimize the pressure drop.

η=assume certain reasonable efficiency according to manufacturing capability or requirement point of view.

Outlet temperature can be calculated using formulae II as under:

$T_{in} \rightarrow h_{in}$= f ($P_{in}$, T_{in}) → s_{in}=f (P_{in}, T_{in}) → h_{out} ideal=f (p_{out}, s_{in}) → h_{out} using formulae II →t_{out}

HX 3

Hot stream inlet temperature =turbine A inlet temperature.

Hot stream outlet temperature =turbine A outlet temperature.

Reason: to keep the temperatures of both high pressure hot stream and medium pressure hot stream inlet temperatures same for efficient heat transfer between the cold stream and hot streams only rather between the two hot streams due to temperature difference.

Cold stream inlet temperature = calculated from HX 4

Cold stream outlet temperature = f (P_{cold} , h_{cout})

Using formulae I,

$$h_{cout} = h_{cin} + \frac{(m_h * (h_{in} - h_{out})) + heatleak}{m_c}$$

HX 4

High pressure (from HX 3) Hot stream inlet temperature =turbine A outlet temperature= Medium pressure (from Turbine A) Hot stream inlet temperature

Reason: as explained earlier

High pressure Hot stream outlet temperature =turbine B inlet temperature=medium pressure hot stream outlet temperature

Reason: same as above i.e. to keep the temperature of both high pressure hot stream and medium pressure hot stream inlet temperature same for efficient heat transfer between the cold stream and hot streams only or to check heat transfer between the two hot streams due to temperature difference.

Cold stream inlet temperature = calculated from HX 5(user input)

Cold stream outlet temperature = f (P_{cold}, h_{cout})

Using formulae I

$$h_{cout} = h_{cin} + \frac{(mh*(h_{in} - h_{out}))_{HP} + (mh*(h_{in} - h_{out}))_{MP} + heatleak}{m_c}$$

Turbine B

Inlet temperature = user defined and fixed while changing mass flow rate to find the optimum mass flow through turbine A vis-à-vis HX 3 and JT valve.

NOTE: Turbine B inlet temperature is also user defined based on one of the many important user objectives like keeping same volumetric flow rate through both turbines etc. Or else a separate analysis may be done to find out an optimum inlet temperature for turbine B at some optimized turbine A inlet temperature or for an optimum combination of temperature.

Pressure drop =judicial pressure drop considering the manufacturing or designing factors in view. A separate analysis may be done to optimize the pressure drop.

η =assume certain reasonable efficiency according to manufacturing capability or requirement point of view.

Outlet temperature can be calculated using formulae II as under:

$T_{in} \rightarrow h_{in}= f (P_{in}, T_{in}) \rightarrow s_{in}=f (P_{in}, T_{in}) \rightarrow h_{out}$ ideal=f $(p_{out}, s_{in}) \rightarrow h_{out}$ using formulae II $\rightarrow t_{out}$

HX 5

Hot stream inlet temperature =hot stream outlet temperature of HX4

Cold stream inlet temperature = turbine B outlet temperature

Reason: for efficient adiabatic mixing of the two streams, one from turbine B outlet and the other from HX 6 cold outlet, both the temperatures have been taken same or as close as possible.

Cold stream outlet temperature =user defined , it is varied along with the mass flow variation through turbine A & B in order to satisfy practical constraints mostly

1. To keep UA of all the heat exchangers under practical user desired values.(here some base values have been decided for each exchanger) .While varying the mass flow through turbine A ,HX 5 cold outlet temperature is manipulated or adjusted in order to keep the UA of all the exchangers below desired value(generally some multiple of base values.)
2. Sometimes to satisfy efficient heat transfer(avoiding temperature cross)

Hot stream outlet temperature = f (P_{hot}, h_{out})

Using formulae I,

$$h_{out} = h_{in} - \frac{(m_c * (h_{cout} - h_{cin})) - heatleak}{m_h}$$

HX 6

Hot stream inlet temperature =hot stream outlet temperature from HX 5

Hot stream outlet temperature = turbine C inlet temperature

Cold stream outlet temperature =f (pressure, h_{cout})

Cold stream inlet temperature = f (P_{cold}, h_{cin})

Using formulae I and energy balance equation,

$$h_{cout} = \frac{(mc * h_{cin})_{HX5} - (m * h_{out})_{TurbineB}}{m_{cHX6}}$$

$$h_{cin} = h_{cout} - \frac{(m_h * (h_{in} - h_{out})) + heatleak}{m_c}$$

Turbine C

Inlet temperature = user defined and fixed while changing mass flow rate to find the optimum mass flow through turbine A vis-à-vis HX 3 and JT valve.

Pressure drop =judicial pressure drop considering the manufacturing or designing factors in view. A separate analysis may be done to optimize the pressure drop.

NOTE: Turbine C inlet temperature is also user defined based on one of the many important user objectives like practical designing and operating problems at that low temperature etc. Or else a separate analysis may be done to find out an optimum inlet temperature for turbine C at some optimized turbine A inlet temperature or for an optimum combination of temperature.

η =assume certain reasonable efficiency according to manufacturing capability or requirement point of view.

Outlet temperature can be calculated using formulae II as under:

$T_{in} \rightarrow h_{in}$= f ($P_{in}$, T_{in}) $\rightarrow s_{in}$=f (P_{in}, T_{in}) $\rightarrow h_{out}$ ideal=f (p_{out}, s_{in}) $\rightarrow h_{out}$ using formulae II $\rightarrow t_{out}$

HX 7

Hot stream inlet pressure = turbine C outlet pressure

Hot stream inlet temperature = turbine C outlet temperature

Cold stream outlet temperature =cold stream inlet temperature from HX 6

Cold stream inlet temperature = 4.408 K

Reason:

Pcold = 1.2 bar ,saturation temperature of liquid helium at 1.2 bar is 4.407, so Tcin may be taken as 4.407 or a general superheat of .001 K after the application.

Hot stream outlet temperature = f (P_{hot}, h_{out})

Using formulae I

$$h_{out} = h_{in} - \frac{(m_c * (h_{cout} - h_{cin})) - heatleak}{m_h}$$

JT Valve

Inlet temperature and pressure = hot stream outlet temperature and pressure from HX 7.

Outlet temperature can be calculated as:

T_{in} →h_{in} = h_{out} ideal →t_{out}=f(P_{cold}, h_{out}) →vapor fraction & latent heat of liquid helium →liquid fraction → total liquid produced → refrigeration

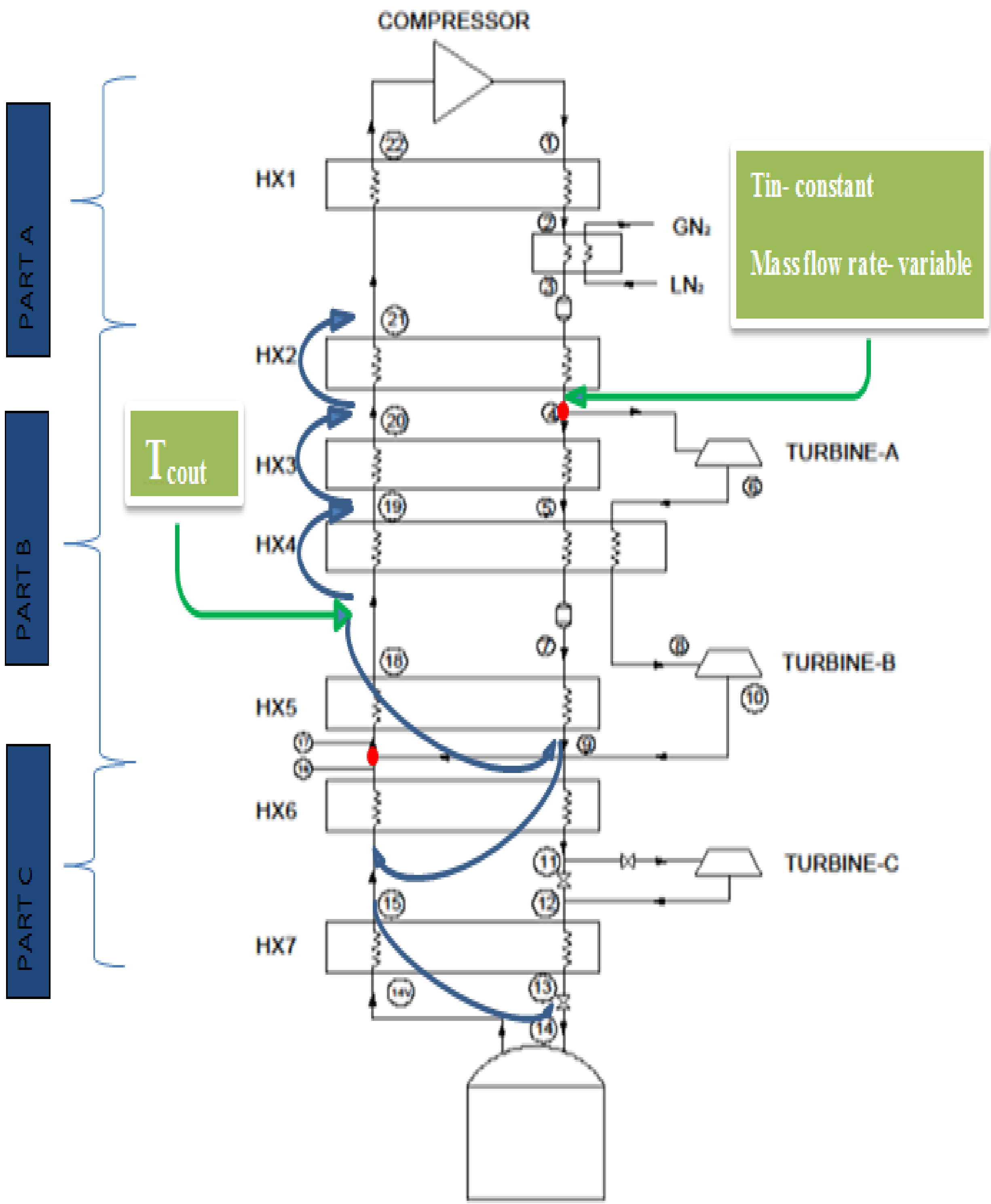

Figure 3.1 Schematic diagram for methodology to optimise mass flow through turbine A

3.2. Method opted to optimize inlet temperature of turbine A vis-a-vis HX 3

For a given compressor pressure, mass flow and fixed liquid helium requirement.

HX 2

Hot stream inlet temperature =80.01

Reason: outlet from liquid nitrogen precooling heat exchanger and 80 K absorber bed.

Hot stream outlet temperature = inlet temperature of turbine A

Cold stream inlet temperature = calculated from HX 3

Cold stream outlet temperature = f (P_{cold}, h_{cout})

using formulae I,

$$h_{cout} = h_{cin} + \frac{(m_h * (h_{in} - h_{out})) + heatleak}{m_c}$$

Turbine A

Inlet temperature = user defined and variable while mass flow rate is fixed at the optimum low fraction obtained from above analysis to find the optimum inlet temperature of turbine A vis-à-vis HX 3.

Pressure drop =judicial pressure drop considering the manufacturing or designing factors in view. A separate analysis may be done to optimize the pressure drop.

η =assume certain reasonable efficiency according to manufacturing capability or requirement point of view.

Outlet temperature can be calculated using formulae II as under:

T_{in} → h_{in}= f (P_{in}, T_{in}) → s_{in}=f (P_{in}, T_{in}) → h_{out} ideal=f (p_{out}, s_{in}) → h_{out} using formulae II →t_{out}

HX 3

Hot stream inlet temperature =turbine A inlet temperature.

Hot stream outlet temperature =turbine A outlet temperature.

Reason: to keep the temperatures of both high pressure hot stream and medium pressure hot stream inlet temperatures same for efficient heat transfer between the cold stream and hot streams only rather between the two hot streams due to temperature difference.

Cold stream inlet temperature = calculated from HX 4

Cold stream outlet temperature = f (P_{cold}, h_{cout})

Using formulae I,

$$h_{cout} = h_{cin} + \frac{(m_h * (h_{in} - h_{out})) + heatleak}{m_c}$$

HX 4

High pressure (from HX 3) Hot stream inlet temperature =turbine A outlet temperature= Medium pressure Hot stream (from Turbine A) inlet temperature

Reason: as explained earlier

High pressure Hot stream outlet temperature =turbine B inlet temperature=medium pressure hot stream outlet temperature

Reason: same as above i.e. to keep the temperature of both high pressure hot stream and medium pressure hot stream inlet temperature same for efficient heat transfer between the cold stream and hot streams only or to check heat transfer between the two hot streams due to temperature difference.

Cold stream inlet temperature = calculated from HX 5(user input)

Cold stream outlet temperature = f (P_{cold} , h_{cout})

Using formulae I

$$h_{cout} = h_{cin} + \frac{(mh*(h_{in} - h_{out}))_{HP} + (mh*(h_{in} - h_{out}))_{MP} + heatleak}{m_c}$$

Turbine B

Inlet temperature = user defined and variable while changing the inlet temperature of Turbine A accordingly.

NOTE: Turbine B inlet temperature is also user defined based on one of the many important user objectives like keeping same volumetric flow rate through both turbines etc. Or else a separate analysis may be done to find out an optimum inlet temperature for turbine B at some optimized turbine A inlet temperature or for an optimum combination of temperature.

Pressure drop =judicial pressure drop considering the manufacturing or designing factors in view. A separate analysis may be done to optimize the pressure drop.

η =assume certain reasonable efficiency according to manufacturing capability or requirement point of view.

Outlet temperature can be calculated using formulae II as under:

$T_{in} \rightarrow h_{in}= f\ (P_{in}, T_{in}) \rightarrow s_{in}=f\ (P_{in}, T_{in}) \rightarrow h_{out}$ ideal=f $(p_{out}, s_{in}) \rightarrow h_{out}$ using formulae II $\rightarrow t_{out}$

HX 5

Hot stream inlet temperature =hot stream outlet temperature of HX4

Cold stream inlet temperature = turbine B outlet temperature

Reason: for efficient adiabatic mixing of the two streams, one from turbine B outlet and the other from HX 6 cold outlet, both the temperatures have been taken same or as close as possible.

Cold stream outlet temperature =user defined , it is varied along with the mass flow variation through turbine A & B in order to satisfy practical constraints mostly

3. To keep UA of all the heat exchangers under practical user desired values.(here some base values have been decided for each exchanger .While varying the inlet temperature of turbine A , HX 5 cold outlet temperature is manipulated or adjusted in order to keep UA of all the exchangers below desired value(generally some multiple of base values.)
4. Sometimes to satisfy efficient heat transfer(avoiding temperature cross)

Hot stream outlet temperature = f (P_{hot}, h_{out})

Using formulae I,

$$h_{out} = h_{in} - \frac{(m_c * (h_{cout} - h_{cin})) - heatleak}{m_h}$$

HX 6

Hot stream inlet temperature =hot stream outlet temperature from HX 5

Hot stream outlet temperature = turbine C inlet temperature

Cold stream outlet temperature =f (pressure, h_{cout})

Cold stream inlet temperature = f (P_{cold}, h_{cin})

Using formulae I and energy balance equation,

$$h_{cout} = \frac{(mc * h_{cin})_{HX5} - (m * h_{out})_{TurbineB}}{m_{cHX6}}$$

$$h_{cin} = h_{cout} - \frac{(m_h * (h_{in} - h_{out})) + heatleak}{m_c}$$

Turbine C

Inlet temperature = user defined and variable while changing inlet temperature of turbine A.

Pressure drop =judicial pressure drop considering the manufacturing or designing factors in view. A separate analysis may be done to optimize the pressure drop.

NOTE: Turbine C inlet temperature is also user defined based on one of the many important user objectives like practical designing and operating problems at that low temperature etc. in present methodology it is primarily varied along with HX 5 cold outlet temperature at any turbine A inlet temperature to maximize refrigeration while maintain a reasonable minimum approach in HX 7 and UA of other exchanger under the desired value. Or else a separate analysis may be done to find out an optimum inlet temperature for turbine C at some optimized turbine A inlet temperature or for an optimum combination of temperature.

η =assume certain reasonable efficiency according to manufacturing capability or requirement point of view.

Outlet temperature can be calculated using formulae II as under:

$T_{in} \rightarrow h_{in}= f\ (P_{in}, T_{in}) \rightarrow s_{in}=f\ (P_{in}, T_{in}) \rightarrow h_{out}$ ideal=f $(p_{out}, s_{in}) \rightarrow h_{out}$ using formulae II $\rightarrow t_{out}$

HX 7

Hot stream inlet pressure = turbine C outlet pressure

Hot stream inlet temperature = turbine C outlet temperature

Cold stream outlet temperature =cold stream inlet temperature from HX 6

Cold stream inlet temperature = 4.408 K

Reason:

Pcold = 1.2 bar ,saturation temperature of liquid helium at 1.2 bar is 4.407 ,so T_{cin} may be taken as 4.407 or a general superheat of .001 K after the application.

Hot stream outlet temperature = f (P_{hot}, h_{out})

Using formulae I

$$h_{out} = h_{in} - \frac{(m_c * (h_{cout} - h_{cin})) - heatleak}{m_h}$$

JT Valve

Inlet temperature and pressure = hot stream outlet temperature and pressure from HX 7.

Outlet temperature can be calculated as:

T_{in} →h_{in} = h_{out} ideal →t_{out}=f(P_{cold}, h_{out}) →vapor fraction & latent heat of liquid helium →liquid fraction → total liquid produced → refrigeration

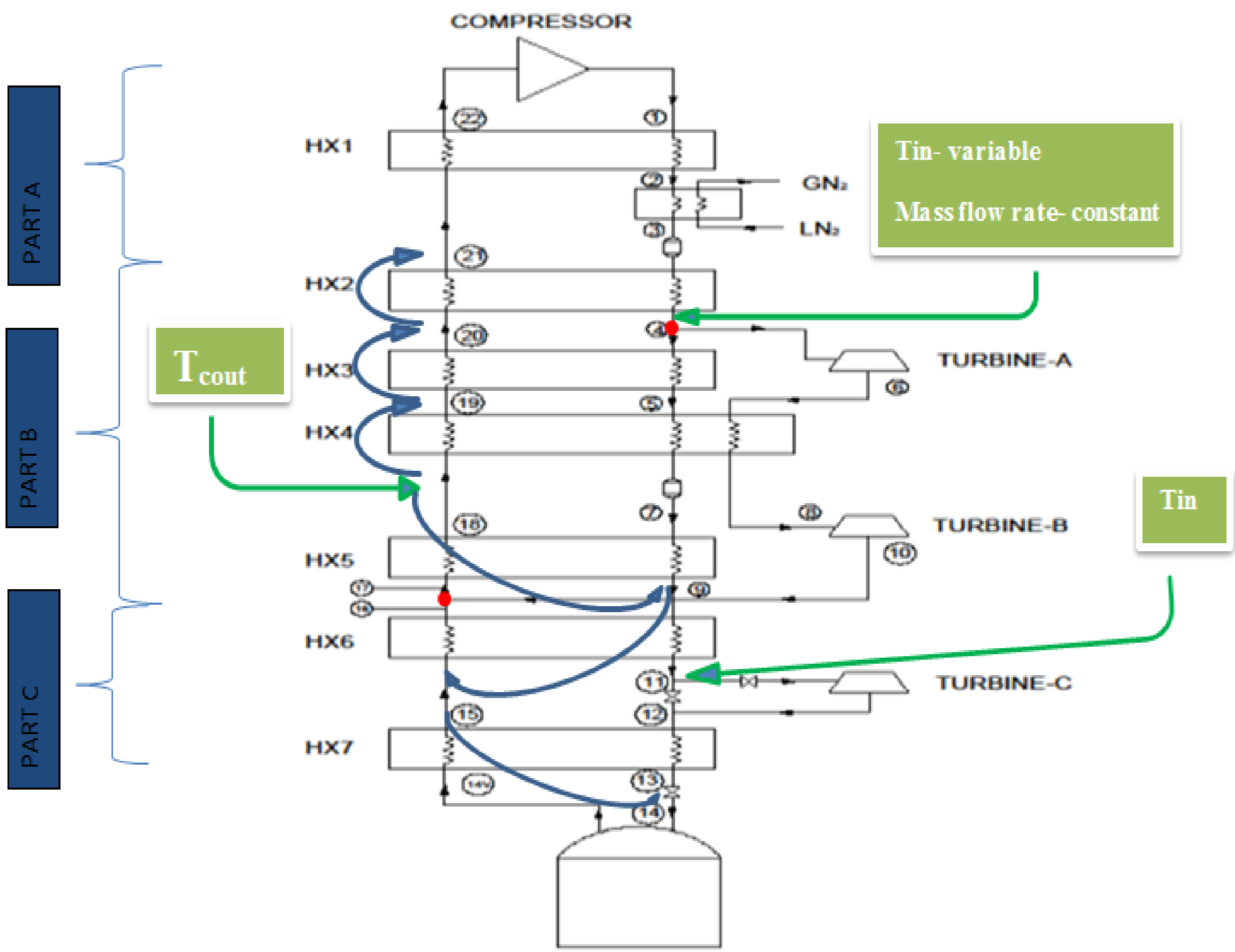

Figure 3.2 Schematic diagram regarding methodology to optimize inlet temperature to turbine A

CHAPTER -4

4. RESULTS AND DISCUSSION

4.1 Selection of Method

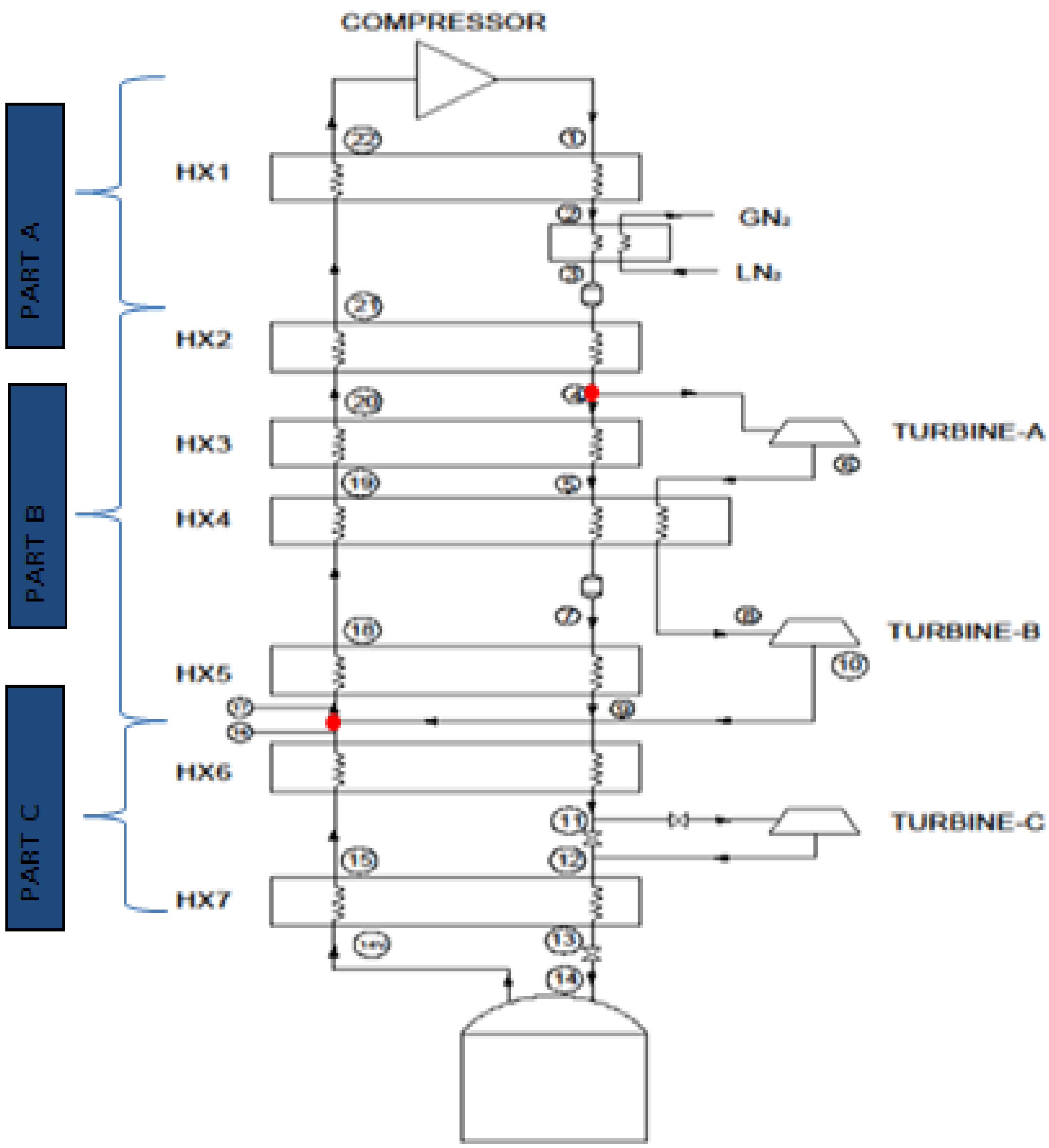

Figure 4.1 different analysis parts of the present cycle configuration.

Here, as shown in the above figure

The complete plant configuration can be divided into three parts in accordance with their temperature zones and their thermodynamic importance for the analysis.

Part-A: From compressor outlet temperature of approximately 310K to LN_2 pre cooler outlet temperature of about 80K.

Part-B: From LN_2 pre cooler outlet temperature of about 80Kto Turbine B outlet temperature of almost 10.5 K.

Part-C: From Turbine B outlet temperature of almost 10.5 K to JT valve outlet temperature.

As it has been already explained in the preceding section that it is almost cinch to have 80 K at the outlet of LN_2 pre cooler. Therefore, the present analysis started with part- B.

Starting form part B, three different approaches has been tried.

1. *Top to bottom temperature approximation*: In this method certain logical temperature is assumed in the first heat exchanger. Thereafter rest of the temperatures points are calculated under logical assumptions fulfilling all the important constraints.
2. *Bottom to top temperature approximation*: In this method certain logical temperature is assumed in the last heat exchanger. Thereafter rest of the temperatures are calculated under logical assumptions fulfilling all the important constraints
3. *Centre temperature approximation*: in this method certain logical temperature is assumed at the outlet of in the fifth heat exchanger. Thereafter rest of the temperatures are calculated under logical assumptions fulfilling all the important constraints
   - ✓ Centre temperature approximation method simulates whole mass flow spectrum through Turbine A with least errors encountered. So it is preferred and explained in the preceding section
   - ❖ In the first and second method viz. Top to bottom temperature approximation and Bottom to top temperature approximation the main problem encountered was of inefficient heat transfer in some heat exchangers when different mass flow variation or temperature variation is tried. The inefficient heat transfer or the problem of temperature cross in heat exchangers is least in the center temperature approximation method.

4.2 Optimum Mass Flow Fraction Analysis

4.2.1. 140g/s (2×70) & 7 g/s Liquefaction

*Here the **mass flow** through turbine A=turbine B (vis-à-vis JT inlet) is varied.*

Constant user input

Compressor mass flow= (2*70) = 140 g/s

Cold box inlet mass flow=132.7 g/s= (compressor mass flow - mass flow to bearings)

Turbine A Inlet temperature =35.3 K; efficiency=.76

Turbine A Inlet temperature =15.62 K; efficiency=.72

Turbine A Inlet temperature =7.5 K; efficiency=.64

Logical assumptions

HX4

High pressure Hot inlet temperature = medium pressure Hot inlet temperature.

High pressure Hot outlet temperature = High pressure Hot outlet temperature.

HX 5

Temperature in all the three streams adiabatically mixing before the cold inlet is nearly equal.

Variable

Cold stream outlet temperature is varied to adjust the UA value of all the heat exchangers under judicial value with varying mass flow through turbines.

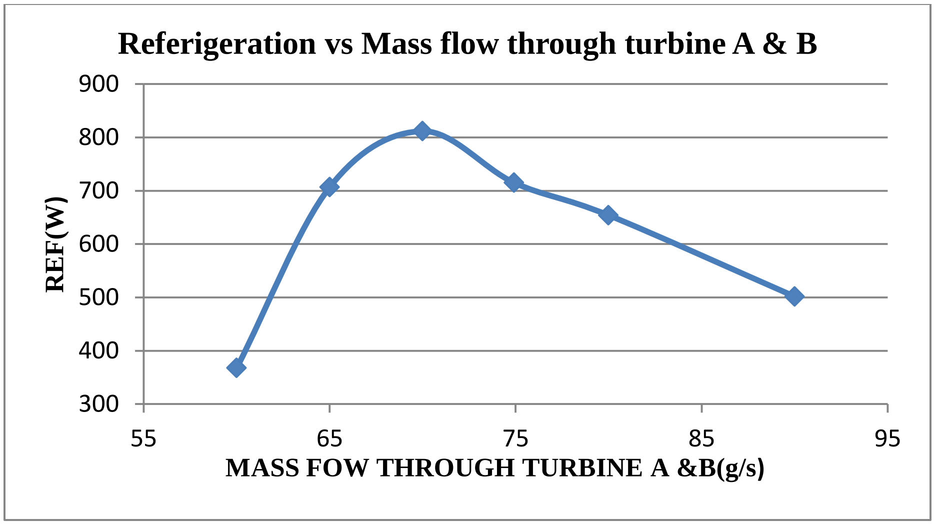

Figure 4.2 Plot of Refrigeration Vs Mass flow through turbine A for 140 g/s compressor flow and 7 g/s liquefaction

As shown in graph,

- ✓ Refrigeration is almost 800 W when 70 g/s of helium gas out of 132.7 g/s coming from heat exchanger HX 1 is sent through turbine A .the above mentioned mass flow is almost 53% of the total cold box flow. So, it can be concluded that for the above mentioned configuration and conditions the optimum value of mass flow through turbine A is 70 g/s.
- ❖ So, it is quite evident that there is an optimum mass flow through turbine A for which the cycle will deliver highest refrigeration satisfying the constraints, mostly the UA criteria. Mass flow or mass flow fraction less or more than the optimum flow will deliver less refrigeration. Also ,while changing the mass flow through turbine A ,it is noted that ,mostly for the lower mass flow fractions than the optimum one through turbine A ,the exchangers above HX 5 viz. HX 4,HX 3,HX 2 become critical whereas for the higher values HX 5,HX 6, HX7 become critical (one or more than one at a time). Here, critical means that the UA value for that heat exchanger has reached the constraint limit defined or decided by the user at the beginning of the analysis on practical ,space availability and other basis. When any heat exchanger reaches its critical value, any further manipulation in the cycle pinch point temperatures or the

inlet and outlet temperatures of the heat exchangers is not done and the refrigeration value is noted to be plotted.

4.2.2. 140g/s and 0 g/s Liquefaction

Procedure same as above except that

liquid taken out=0 g/s

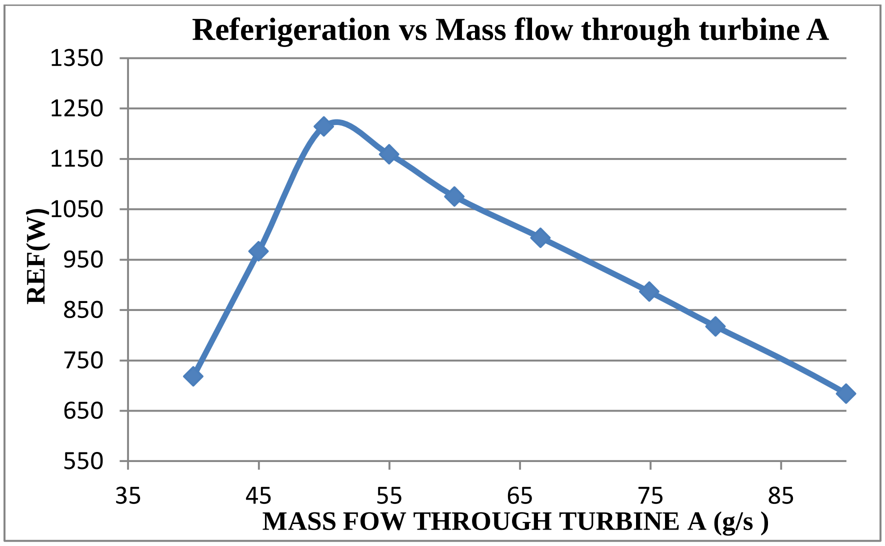

Figure 4.3 Plot of Refrigeration Vs Mass flow through turbine A for 140 g/s compressor flow and 0 g/s liquefaction

as shown in above graph,

- ✓ Refrigeration is almost 1200 W when 50 g/s of helium gas out of 132.7 g/s coming from heat exchanger HX 1 is sent through turbine A .the above mentioned mass flow is almost 38% of the total cold box flow. So, it can be concluded that for the above mentioned configuration and conditions the optimum value of mass flow through turbine A is 50 g/s.
- ❖ So, it is quite evident that there is an optimum mass flow through turbine A for which the cycle will deliver highest refrigeration satisfying the constraints, mostly the UA criteria. Mass flow or mass flow fraction less or more than the optimum flow will deliver less refrigeration. Also ,while changing the mass flow through turbine A ,it is noted that ,mostly for the lower mass flow fractions than the optimum one through turbine A ,the exchangers above HX 5 viz. HX 4,HX 3,HX 2 become critical whereas for the higher values HX 5,HX 6, HX7 become critical(one or more than one at a

time). Here, critical means that the UA value for that heat exchanger has reached the constraint limit defined or decided by the user at the beginning of the analysis on practical ,space availability and/or other basis. When any heat exchanger reaches its critical value, any further manipulation in the cycle pinch point temperatures or the inlet and outlet temperatures of the heat exchangers is not done and the refrigeration value is noted to be plotted.

Also as the liquid helium taken out from the total liquid produced is decreased or plant is operated in refrigeration mode rather than the mixed mode, the optimum flow fraction through turbine A also decreases. It might be because of the increased cooling capacity of the return stream.

4.2.3. 160g/s (2×80) & 7 g/s Liquefaction

compressor mass flow=160g/s

liquid taken out =7 g/s

Cold box inlet mass flow= 151.65 g/s

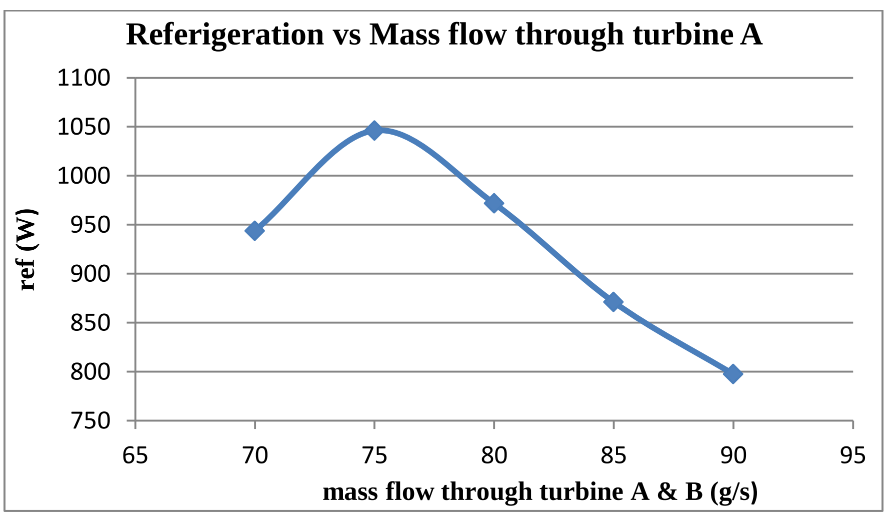

Figure 4.4 Plot of Refrigeration Vs Mass flow through turbine A for 160 g/s compressor flow and 7 g/s liquefaction

As shown in above graph,

- ✓ Refrigeration is almost 1050 W when 75 g/s of helium gas out of 151.65 g/s coming from heat exchanger HX 1 is sent through turbine A .the above mentioned mass flow is almost 50% of the total cold box flow. So, it can be concluded that for the above mentioned configuration and conditions the optimum value of mass flow through turbine A is 75 g/s.

4.2.4. 240 g/s (3×80) & 12 g/s Liquefaction

Procedure same as above.except that

compressor mass flow=240g/s

Cold box inlet mass flow= 227.48 g/s= (compressor mass flow - mass flow to bearings bearings)

liquid taken out =12 g/s=(7/140)g/s

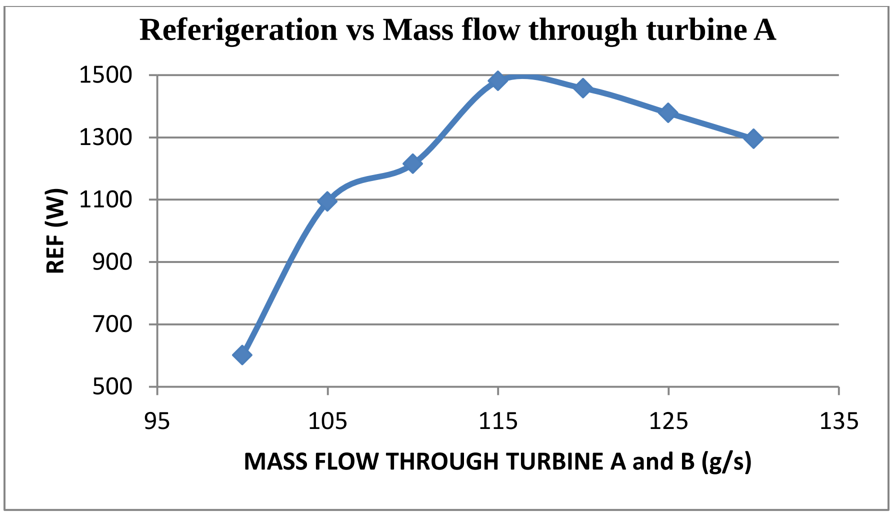

Figure 4.5 Plot of Refrigeration Vs Mass flow through turbine A for 240 g/s compressor flow and 12 g/s liquefaction

As shown in above graph,

- ✓ Refrigeration is almost 1500 W when 120 g/s of helium gas out of 227.48 g/s coming from heat exchanger HX 1 is sent through turbine A .the above mentioned mass flow is almost 53% of the total cold box flow. So, it can be concluded that for the above mentioned configuration and conditions the optimum value of mass flow through turbine A is 120 g/s.

4.2.5. 120 g/s (2×60) &0 g/s Liquefaction

*Here the **mass flow** through turbine A=turbine B (vis-à-vis JT inlet) is varied.*

<u>Constant user input</u>

Compressor mass flow= (2*60) = 120 g/s

Cold box inlet mass flow=112.7 g/s= (compressor mass flow-mass flow to bearings bearings)

	Inlet pressure	Inlet temperature	Efficiency
Turbine A	13.5 bar	35.3 K	.65
Turbine B	5.9 bar	16.31 K	.65
Turbine C	13.10 bar	7.5 K	.6

Note: Inlet temperature of turbine B is selected in a way to keep same volumetric flow rate in both turbine A and turbine B.

<u>Logical assumptions and Variable</u>

Same as above

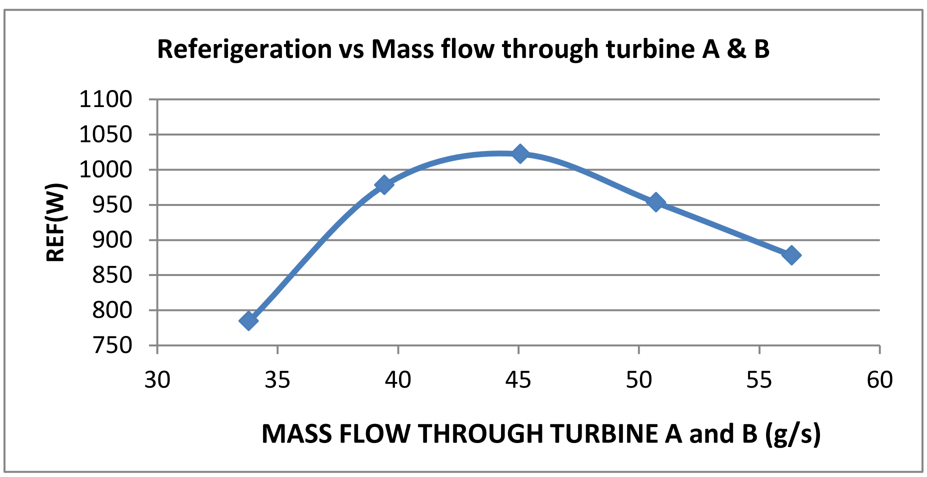

Figure 4.6 Plot of Refrigeration Vs Mass flow through turbine A for 120 g/s compressor flow and 0 g/s liquefaction

As shown in above graph,

- ✓ Refrigeration is almost 1025 W when 45 g/s of helium gas out of 112.7 g/s coming from heat exchanger HX 1 is sent through turbine A .the above mentioned mass flow is almost 40% of the total cold box flow. So, it can be concluded that for the above mentioned configuration and conditions the optimum value of mass flow through turbine A is 45 g/s.

4.2.6. Effect of Limiting value of UA on Optimum Mass Flow Through Turbine A

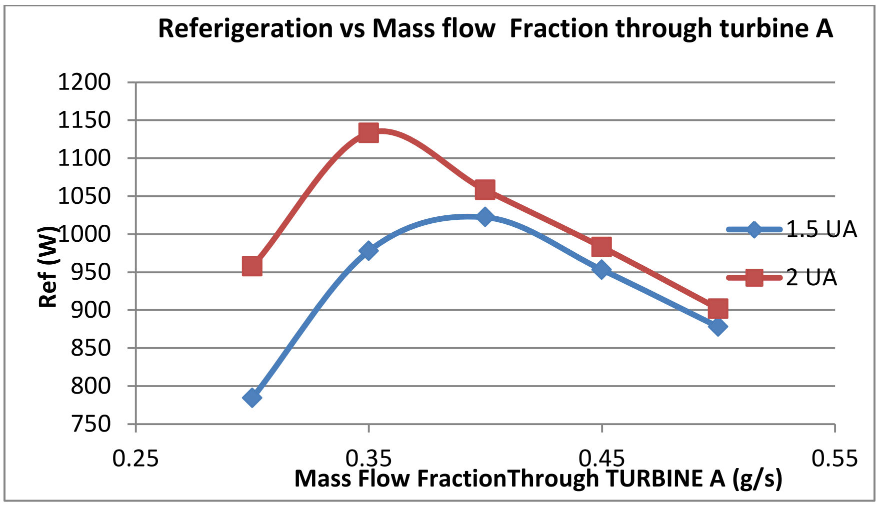

Figure 4.7 Effect of limiting value of UA on optimum mass flow through turbine A

As shown in above graph,

- ✓ When the limiting UA value for each heat exchanger is taken as 1.5 times the respective base value, the refrigeration is almost 1025 W when 45 g/s of helium gas i.e. almost 40% of the total cold box flow of 112.7 g/s .Whereas
- ❖ When the limiting UA value for each heat exchanger is taken as 2 times the respective base value, refrigeration is almost 1125 W when 40 g/s of helium gas i.e. almost 35% of the total cold box flow of 112.7 g/s.
- ❖ It is evident that if the limiting UA value for the heat exchangers is increased less mass flow fraction is optimum .It might be due to the effect that with the increase in UA value for heat exchanger there cold retaining efficiency increases, so may be with less mass flow fraction more refrigeration can be obtained.

4.3 ANALYSIS OF OPTIMUM INLET TEMPERATURE FOR TURBINE A

4.3.1. 120 g/s &0 g/s liquefaction

*Here the **turbine A inlet temperature** is varied.*

Constant user input

Compressor mass flow= (2*60) = 120 g/s

Cold box inlet mass flow=112 g/s= (compressor mass flow-mass flow to bearings bearings)

	Inlet pressure	Inlet temperature	Efficiency
Turbine A	13.5 bar	variable	.65
Turbine B	5.9 bar	variable	.65
Turbine C	13.10 bar	variable	.6

Note: Inlet temperature of Turbine A is varied. In accordance with inlet temperature of turbine A, inlet temperature of turbine B is varied to keep same volumetric flow rate in both Turbine A and turbine B. Whereas inlet temperature to Turbine C is varied to increase the plant capacity and keep minimum approach in HX 7 and UA of other exchangers under decided values.

Logical assumptions

Same as above

Variable

Primarily Cold stream outlet temperature of HX 5 is varied while inlet temperature to turbine C is also varied to adjust the UA value of all the heat exchangers under judicial value with varying temperatures through turbines. Also minimum approach in Heat exchanger 7 has been kept constant at the below mentioned value.

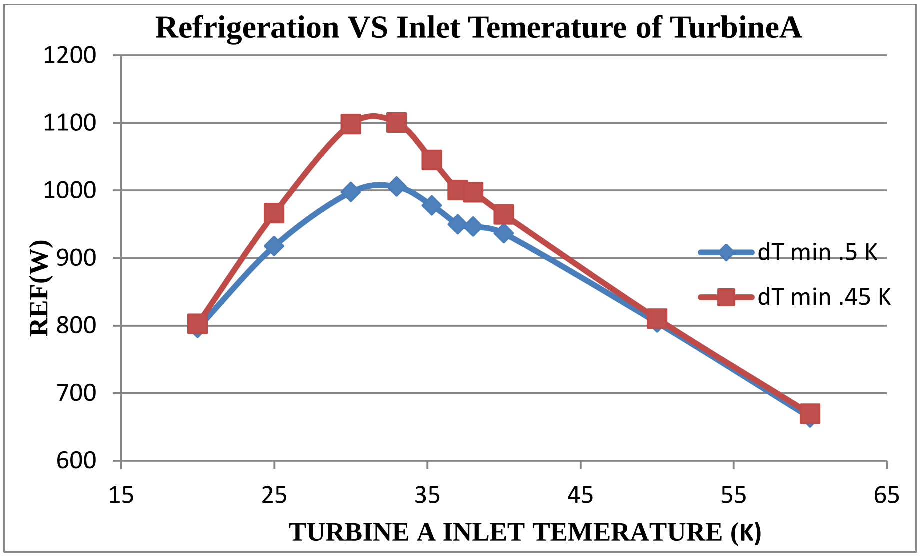

Figure 4.8 Plot of Refrigeration Vs Turbine A inlet temperature comparing effect of minimum approach in last heat exchanger HX 7

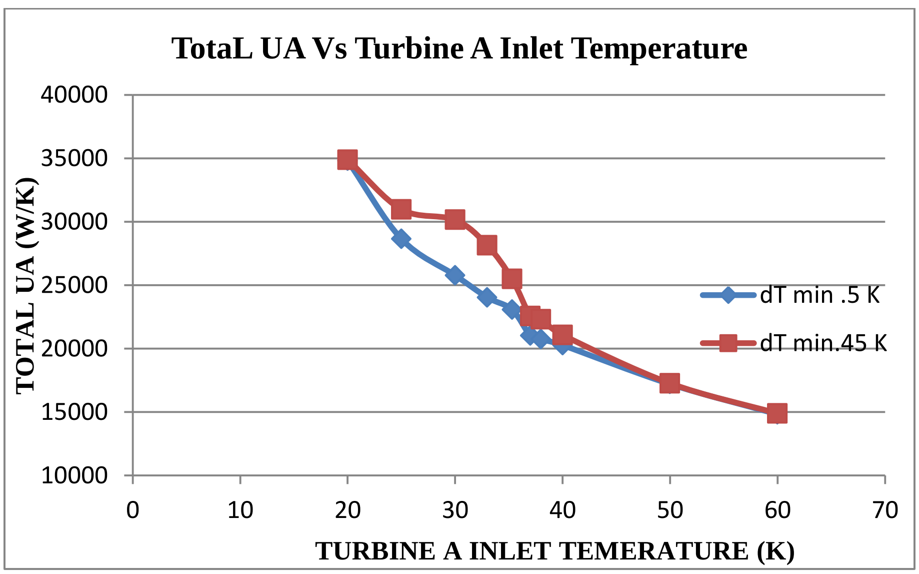

Figure 4.9 Plot of Total UA Vs Turbine A inlet temperature comparing effect of minimum approach in last heat exchanger HX 7

As shown in above graph,

- ✓ Refrigeration is almost 1000 W when turbine A inlet temperature is 33 K and 30 K while the minimum approach temperature in HX 7 is kept constant at .5 K Whereas, Refrigeration is almost 1100 W when turbine A inlet temperature is 33 K and 30 K while the minimum approach temperature in HX 7 is kept constant at .45 K.

- In the above analysis ,throughout the temperature variation ,the minimum approach in the heat exchanger HX 7 is kept constant.to control the same along with the individual UA values of heat exchangers both turbine C inlet temperature and HX 5 cold out temperature are changed in synchronization intuitively.
- It is evident that between 30 K and 33 K the refrigeration is almost constant. But as total UA increases as turbine A inlet temperature goes down due to higher increase in HX 2 UA. It is justifiable to choose 33 K as the optimum one.
- When the above two curves are compared on minimum approach basis, it is evident that at lower and higher temperatures than the optimum zone and its close vicinity the change in minimum approach temperature for HX 7 from .5 K to .45 K does not affect the refrigeration or the total UA much. One of the many possible reasons could be that at lower and higher temperatures than the optimum zone and its close vicinity HX 7 does not have critical affect rather other heat exchanger has.

4.4 OTHER IMPORTANT ANALYSIS

4.4.1. Refrigeration with only two turbines

Here rest of the process is same except that third turbine has been eliminated and the hot outlet stream of HX 6 is fed directly to HX 7.

Table 4.1 Comparison between operation with and without third turbine C regarding refrigeration value and UA values of different heat exchangers

	Tin	UA							
	TA, TB	hx 1	hx2	hx3	hx4	hx5	hx6	hx7	ref
3 T	33 , 15.37	36131	13636	1740	7933	1872	1907	1043	1100
2 T	same	36131	13636	1740	7933	1872	1907	433	779

- ❖ Without third turbine, all the pinch points and UA of the respective heat exchangers would not change except for the seventh heat exchanger HX 7.

Also Refrigeration without third turbine =

- ➢ Refrigeration with third turbine – (multiplication factor *work output from third turbine)

As here,

Table 4.2 Correlation between results with and without third turbine C

	TA,TB	Ref	3rd turbine work	diff	factor
3 T	33 , 15.37	1100	322.1	321.3	0.998
2 T	same	779	0		

4.4.2. Effectiveness of different heat exchangers

Table 4.3 effectiveness values for different heat exchangers

for 120 compressor flow ,45 g/s through turbine A ,.45K minimum approach in HX 7									
Tin		effectiveness							
TA, TB, TC		HX 1	HX2	HX3	HX4	HX5	HX6	HX7	Ref
33 , 15.37, 7	excel	0.98	0.96	0.92	0.96	0.85	0.94	0.73	1100.32

4.4.3. Effect of JT valve inlet temperature and pressure on Liquid fraction

Variable - JT inlet Temperature

Constant - JT inlet pressure

Formulae used - Isenthalpic expansion

Here at any particular JT inlet pressure, JT inlet temperature is varied. At any JT inlet temperature, its inlet enthalpy can be calculated. Now using ideal isenthalpic relation, outlet temperature and thus vapor fraction at JT outlet can be calculated. From vapor fraction liquid fraction can be calculated using formulae V. This plot can be used to decide optimum JT inlet pressure vis-à-vis Turbine C outlet pressure if it is active in process circuit.

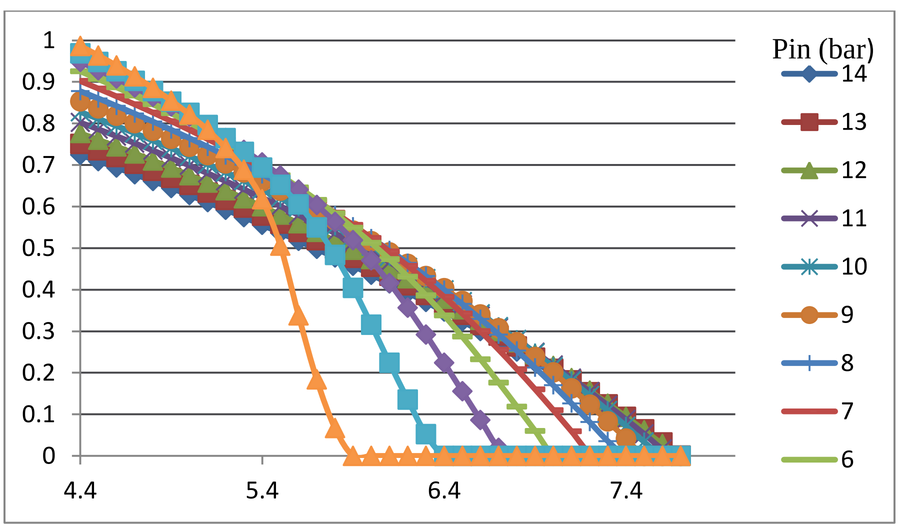

Figure 4.10 Plot of Liquid fraction vs JT Inlet temperature at different JT inlet pressure

- To select the inlet pressure of JT valve is same as to select the outlet pressure of third Turbine C. It can be concluded using the below two premises:

1. The trend in graph shows that if the inlet temperature to JT valve is lower than approximately 5.4 K than the isenthalpic expansion at 3 bar and 4 bar gives more liquid fraction. It is synonymous to the logic of greater the pressure drop in third turbine, more the work output and thus more refrigeration.

2. But the pressure at the outlet of third turbine should not be lower than 4 bar so as to avoid the non-linear property variation and other effects, when the supercritical helium is used in refrigeration purpose of the superconducting system.

- Therefore, 4 bar pressure at the outlet of third turbine C and JT inlet is justified.

CHAPTER -5

5. Validation and Analysis using aspen HYSYS

5.1. Process Flow Diagram

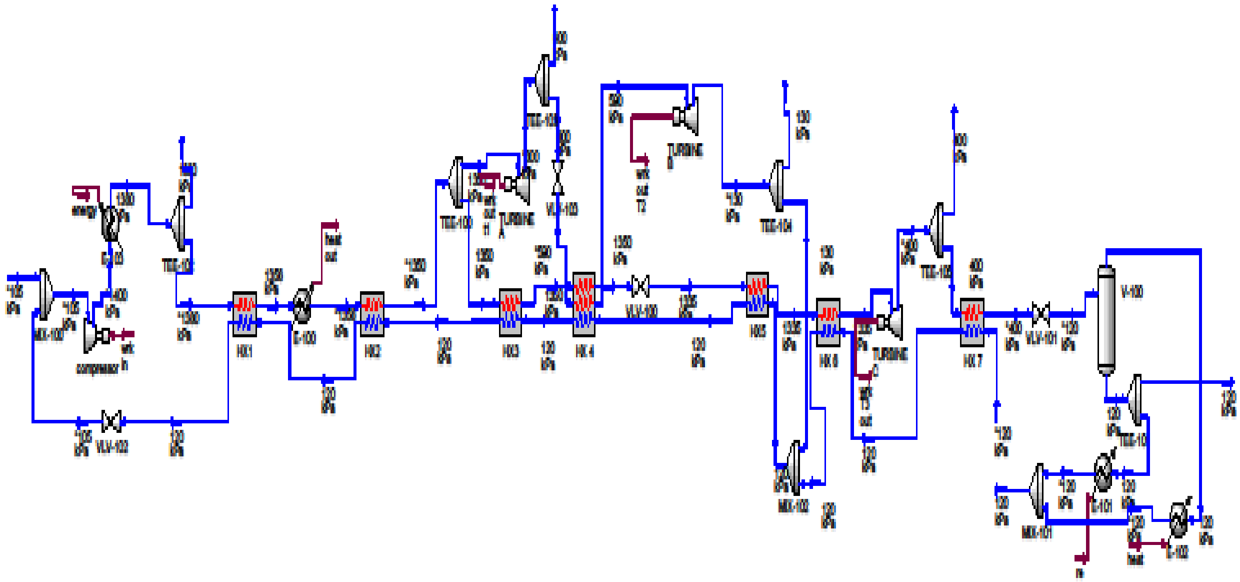

Figure 5.1 process flow diagram

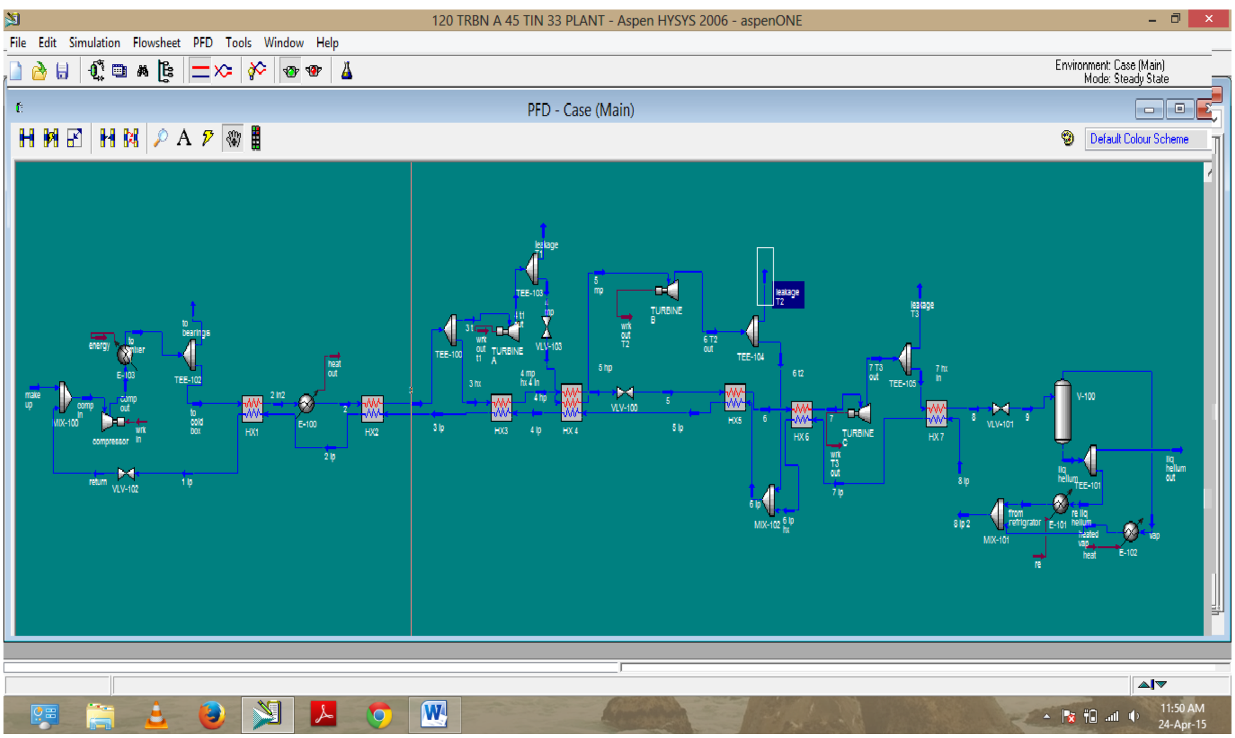

Figure 5.2 screen shot of the process flow diagram

5.2. Optimum mass flow fraction through Turbine A

Table 5.1 worksheet page 1

aspentech

LEGENDS
Calgary, Alberta
CANADA

Case Name:	C:\Program Files (x86)\AspenTech\Aspen HYSYS 2006\Cases\120 TRBI
Unit Set:	NewUser1
Date/Time:	Sun May 03 08:43:54 2015

Workbook: Case (Main)

Material Streams Fluid Pkg: All

Name		make up	return	comp in	1	2 in2
Vapour Fraction		1.0000	1.0000	1.0000	1.0000 *	1.0000 *
Temperature	(K)	306.9	307.0	307.0 *	310.0 *	83.85
Pressure	(kPa)	105.0 *	105.0 *	105.0 *	1330 *	1330 *
Molar Flow	(kgmole/h)	7.824	100.1	107.9	119.3	100.7
Mass Flow	(g/s)	8.700	111.3	120.0 *	132.7 *	112.0
Liquid Volume Flow	(m3/h)	0.2525	3.230	3.482	3.851	3.250
Heat Flow	(kW)	13.87	177.5	191.3	214.2	49.15
Name		**2**	**3**	**3 t**	**3 hx**	**4 mp**
Vapour Fraction		1.0000 *	1.0000 *	1.0000	1.0000	1.0000
Temperature	(K)	80.01 *	35.30 *	35.30	35.30	28.91
Pressure	(kPa)	1356 *	1350 *	1350	1350	600.0
Molar Flow	(kgmole/h)	100.7	100.7	40.47	60.26	40.29
Mass Flow	(g/s)	112.0	112.0	45.00 *	67.00	44.80
Liquid Volume Flow	(m3/h)	3.250	3.250	1.306	1.944	1.300
Heat Flow	(kW)	46.91	20.47	8.225	12.25	6.676
Name		**4 hp**	**5 hp**	**5 mp**	**6 t2**	**6**
Vapour Fraction		1.0000	1.0000	1.0000	1.0000	1.0000 *
Temperature	(K)	28.90 *	16.31 *	16.31 *	10.99	11.52
Pressure	(kPa)	1350	1350	590.0	120.0 *	1284
Molar Flow	(kgmole/h)	60.26	60.26	40.29	40.11	60.26
Mass Flow	(g/s)	67.00	67.00	44.80	44.60	67.00
Liquid Volume Flow	(m3/h)	1.944	1.944	1.300	1.294	1.944
Heat Flow	(kW)	9.907	5.033	3.590	2.465	2.924
Name		**5**	**7**	**7 T3 out**	**8**	**9**
Vapour Fraction		1.0000	1.0000	1.0000	0.0000	0.1893
Temperature	(K)	16.25	7.500 *	6.000	5.051	4.408
Pressure	(kPa)	1284 *	1284	400.0 *	400.0 *	120.0 *
Molar Flow	(kgmole/h)	60.26	60.26	60.26	59.99	59.99
Mass Flow	(g/s)	67.00	67.00	67.00	66.70	66.70
Liquid Volume Flow	(m3/h)	1.944	1.944	1.944	1.935	1.935
Heat Flow	(kW)	5.033	0.9442	0.6147	-3.354e-002	-3.354e-002
Name		**vap**	**liq helium**	**re liq helium**	**liq helium out**	**from refrigrator**
Vapour Fraction		1.0000	0.0000	0.0000 *	0.0000	1.0000 *
Temperature	(K)	4.408	4.408	4.408	4.408	4.408 *
Pressure	(kPa)	120.0	120.0	120.0	120.0	120.0 *
Molar Flow	(kgmole/h)	11.35	48.63	48.63	0.0000	48.63
Mass Flow	(g/s)	12.63	54.07	54.07	0.0000 *	54.07
Liquid Volume Flow	(m3/h)	0.3664	1.569	1.569	0.0000	1.569
Heat Flow	(kW)	0.1937	-0.2273	-0.2273	0.0000	0.8298
Name		**8 lp**	**7 lp**	**6 lp hx**	**6 lp**	**5 lp**
Vapour Fraction		1.0000	1.0000	1.0000	1.0000	1.0000
Temperature	(K)	4.408 *	5.721	10.99	10.99 *	14.56 *
Pressure	(kPa)	120.0 *	120.0	120.0	120.0	120.0
Molar Flow	(kgmole/h)	59.99	59.99	59.99	100.1	100.1
Mass Flow	(g/s)	66.70 *	66.70	66.70	111.3	111.3
Liquid Volume Flow	(m3/h)	1.935	1.935	1.935	3.230	3.230
Heat Flow	(kW)	1.024	1.684	3.688	6.154	8.289

Hyprotech Ltd. Aspen HYSYS Version 2006 (20.0.0.6728) Page 1 of 2

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Table 5.2 worksheet page 2

LEGENDS Calgary, Alberta CANADA	Case Name:	C:\Program Files (x86)\AspenTech\Aspen HYSYS 2006\Cases\120 TRB
	Unit Set:	NewUser1
	Date/Time:	Sun May 03 08:43:54 2015

Workbook: Case (Main) (continued)

Material Streams (continued) Fluid Pkg: All

Name		4 lp	3 lp	2 lp	jt out	jt vap
Vapour Fraction		1.0000	1.0000	1.0000	—	1.0000 *
Temperature	(K)	28.23	32.28	78.03	4.407	4.407 *
Pressure	(kPa)	120.0	120.0	120.0	—	—
Molar Flow	(kgmole/h)	100.1	100.1	100.1	—	—
Mass Flow	(g/s)	111.3	111.3	111.3	—	—
Liquid Volume Flow	(m3/h)	3.230	3.230	3.230	—	—
Heat Flow	(kW)	16.29	18.64	45.13	—	—
Name		jt liq	1 lp	comp out	to bearings	4 t1 out
Vapour Fraction		0.0000 *	1.0000	1.0000	1.0000	1.0000
Temperature	(K)	4.407 *	307.0	1051	310.0	28.91
Pressure	(kPa)	—	120.0	1400 *	1380	600.0 *
Molar Flow	(kgmole/h)	—	100.1	107.9	7.195	40.47
Mass Flow	(g/s)	—	111.3	120.0	8.000 *	45.00
Liquid Volume Flow	(m3/h)	—	3.230	3.482	0.2321	1.306
Heat Flow	(kW)	—	177.5	655.6	12.91	6.705
Name		leakage T1	6 T2 out	leakage T2	7 hx in	leakage T3
Vapour Fraction		1.0000	1.0000	1.0000	1.0000	1.0000
Temperature	(K)	28.91	10.99	10.99	6.000	6.000
Pressure	(kPa)	600.0	120.0	120.0	400.0	400.0
Molar Flow	(kgmole/h)	0.1799	40.29	0.1799	59.99	0.2698
Mass Flow	(g/s)	0.2000 *	44.80	0.2000 *	66.70	0.3000 *
Liquid Volume Flow	(m3/h)	5.803e-003	1.300	5.803e-003	1.935	8.705e-003
Heat Flow	(kW)	2.980e-002	2.476	1.106e-002	0.6119	2.752e-003
Name		heated vap	8 lp 2	4 mp hx 4 in	to cold box	to mixer 1
Vapour Fraction		1.0000	1.0000	1.0000	1.0000	1.0000
Temperature	(K)	4.408 *	4.408	28.90 *	310.0	310.0 *
Pressure	(kPa)	120.0 *	120.0	590.0 *	1380 *	1380
Molar Flow	(kgmole/h)	11.35	59.99	40.29	100.7	107.9
Mass Flow	(g/s)	12.63	66.70	44.80	112.0	120.0
Liquid Volume Flow	(m3/h)	0.3664	1.935	1.300	3.250	3.482
Heat Flow	(kW)	0.1938	1.024	6.676	180.8	193.7

Energy Streams Fluid Pkg: All

Name		wrk in	heat out	wrk out t1	wrk out T2	wrk T3 out
Heat Flow	(kW)	464.3	2.236	1.520	1.114	0.3295
Name		re	heat	energy		
Heat Flow	(kW)	1.057	4.098e-005	461.9		

As shown in the above tables ,the procedure followed for the plant analysis regarding optimizing the mass flow through turbine A vis-à-vis HX3 is same as used in Excel sheet and explained in the methodology section. The parameters mentioned with an asterisk mark '*' are to be specified by the user while simulating the plant in aspen HYSYS

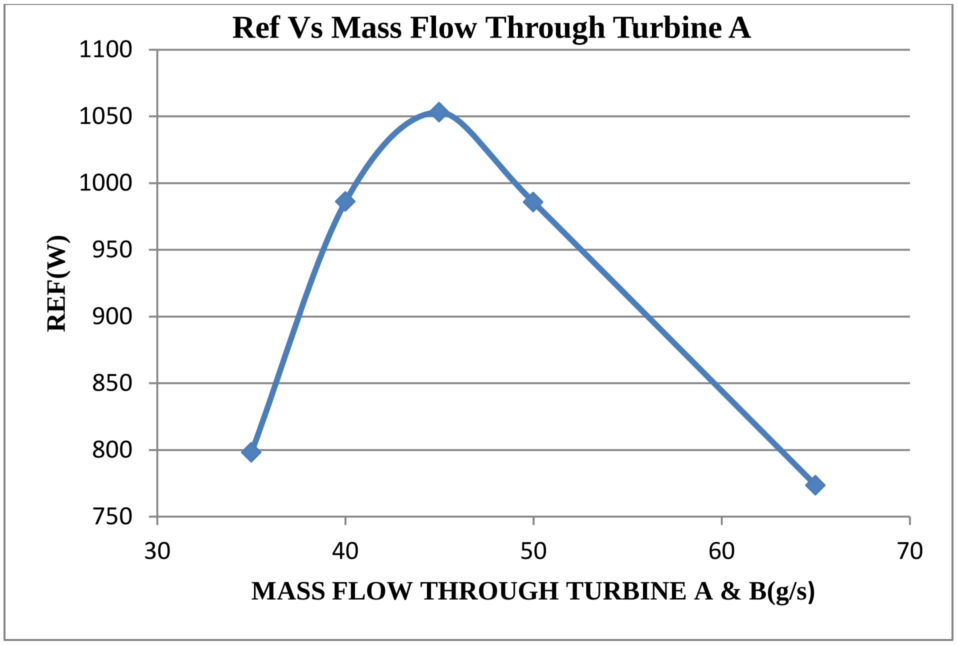

Figure 5.3 Plot of Refrigeration Vs Mass flow through turbine using aspen HYSYS

As shown in above graph,

- ✓ Refrigeration is almost 1050 W when 45 g/s of helium gas out of 112.7 g/s coming from heat exchanger HX 1 is sent through turbine A .the above mentioned mass flow is almost 40% of the total cold box flow. So, it can be concluded that for the above mentioned configuration and conditions the optimum value of mass flow through turbine A is 45 g/s.

Table 5.3 UA values of different heat exchangers at different mass flow through turbine A for 120 g/s compressor flow and 0 g/s liquefaction

Massflow(g/s)	UA							
	hx1	hx2	hx3	hx4	hx5	hx6	hx7	ref
35	36060	15580	1925	7946	1190	1215	271	798
40	35760	14550	1956	9653	1647	1364	650	986
45	35800	12320	1643	9329	1963	1502	1146	1053
50	36060	9746	1163	7145	1731	1421	1136	985.6
65	36180	5990	521.2	4237	1203	1179	1138	773.5

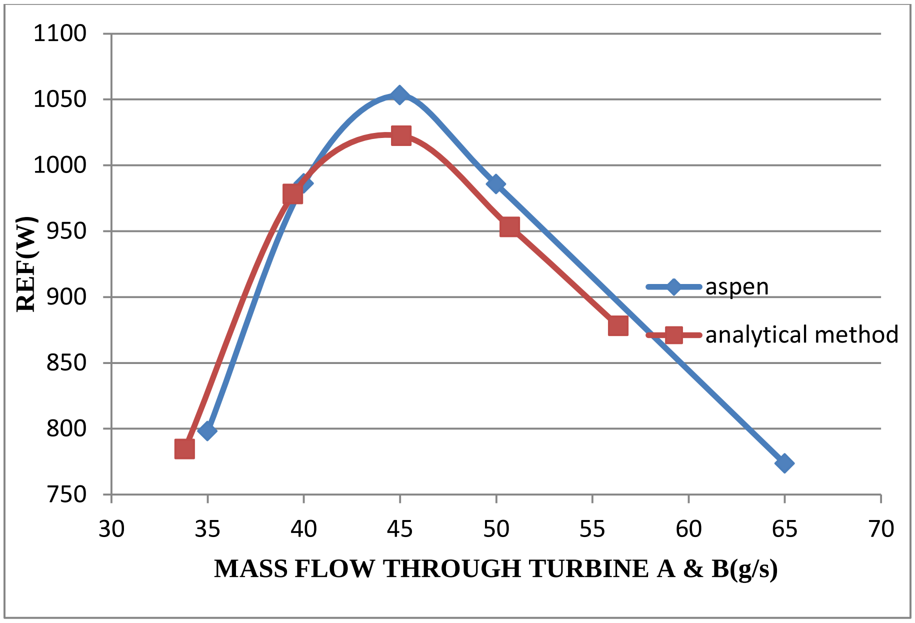

Figure 5.4 Plot of Refrigeration Vs Mass flow through turbine A comparing aspen HYSYS and analytical results

- Other than the above discussed points .There is a slight increase in refrigeration when calculated using aspen HYSYS simulation and analysis. A comparison of the calculated UA values of heat exchangers between aspen HYSYS and excel is studied in the following sections.

5.3. Comparison between aspen HYSYS and analytically calculated parameter Values

Table 5.4 Comparison of excel analysis results with Aspen HYSYS results regarding UA values of different heat exchangers

Tin	UA									
TA, TB, TC		hx 1	hx2	hx3	hx4	hx5	hx6	hx7	ref	total UA
33 , 15.37, 7	excel	36131	13635	1739	7933	1871	1907	1043	1100	64263
33 , 15.37, 7	aspen	35710	14710	1713	8850	1988	1683	868	1094	65522
	%change	1.167	-7.879	1.54	-11.5	-6.2	11.77	16.81	0.57	-2

Table 5.5 Comparison of excel analysis results with Aspen HYSYS results regarding minimum approach values of different heat exchangers

Tin									
TA,TB,TC		hx 1	hx2	hx3	hx4	hx5	hx6	hx7	ref
33 , 15.37, 7	excel	3	1.556	0.492	0.492	0.652	0.633	0.45	1100
33 , 15.37, 7	aspen	3*	1.52	0.463	0.489	0.6	0.619	0.44	1094

❖ As shown in the above comparison that the refrigeration value calculated from both methods is almost equal, though there is slight difference between the UA values calculated from both methods. Amongst all the heat exchangers, the deviation is highest in last heat exchanger HX 7. One of the possible reasons for it is that while calculating through excel programs UA value has been calculated using formulae VI. But in lower temperature range where there is non-linear property variation or the temperature approach curves are not exactly logarithmic, this relation may not hold good.

5.4. EFFECT OF UA VALUE OF HEAT EXCAHNGERS ON REFRIGERATION

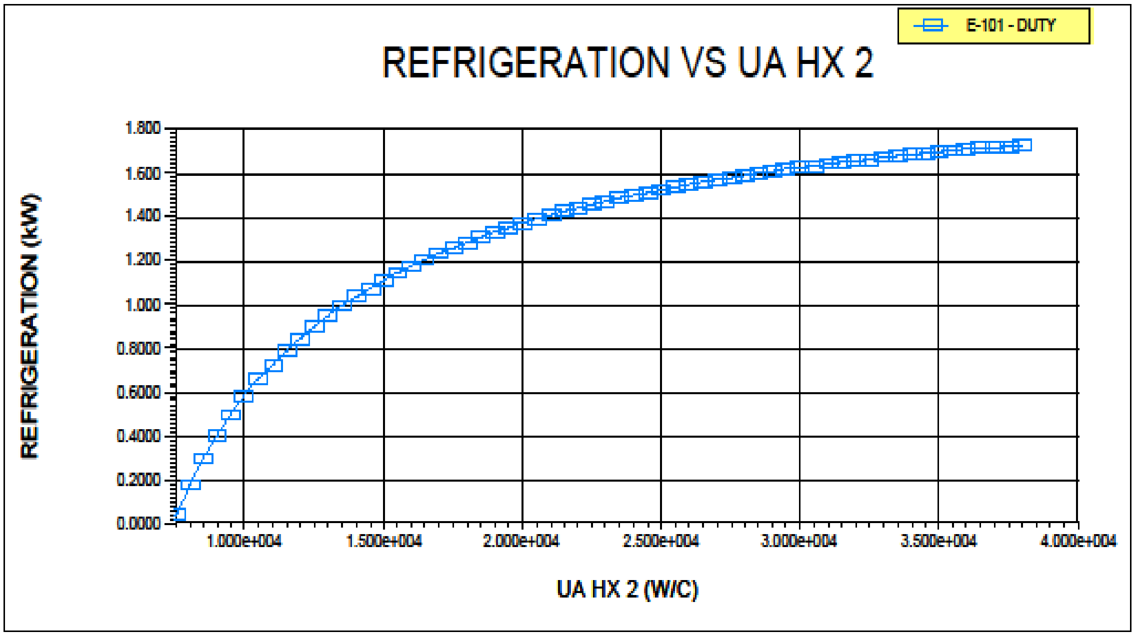

Figure 5.5 Plot of Refrigeration Vs UA of second heat exchanger HX 2

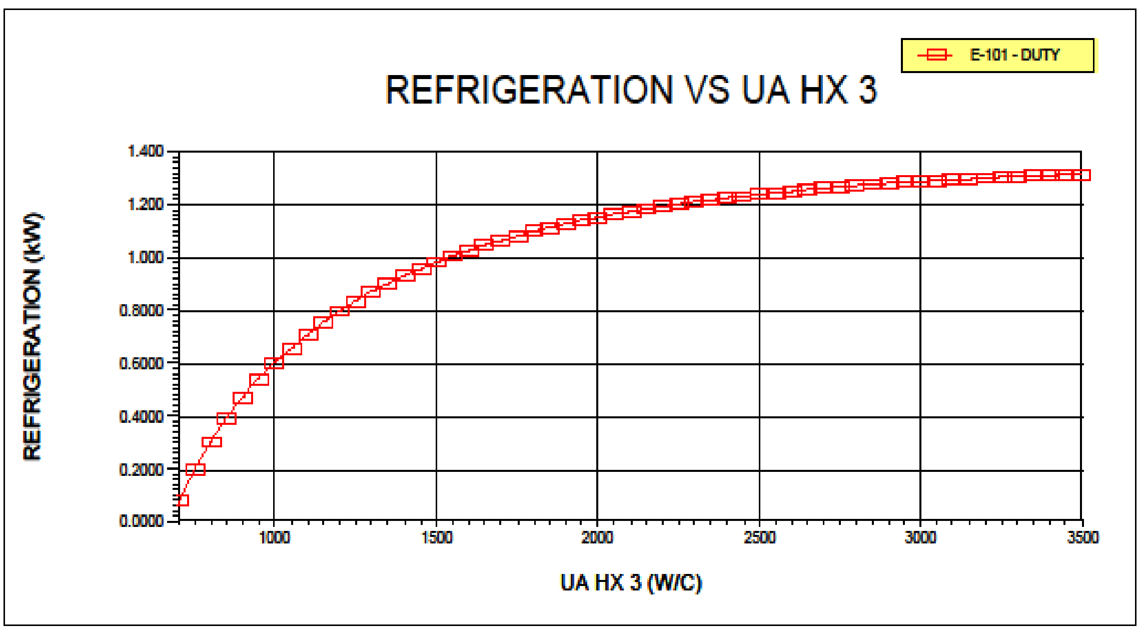

Figure 5.6 Plot of Refrigeration Vs UA of third heat exchanger HX 2

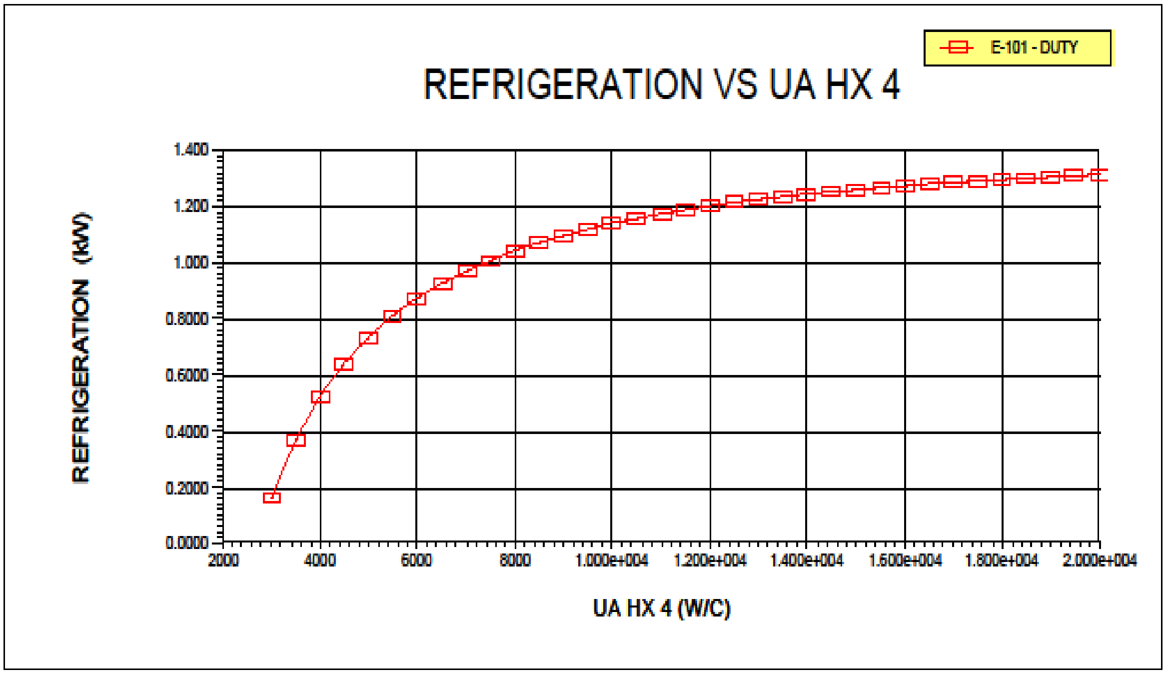

Figure 5.7 Plot of Refrigeration Vs UA of fourth heat exchanger HX 4

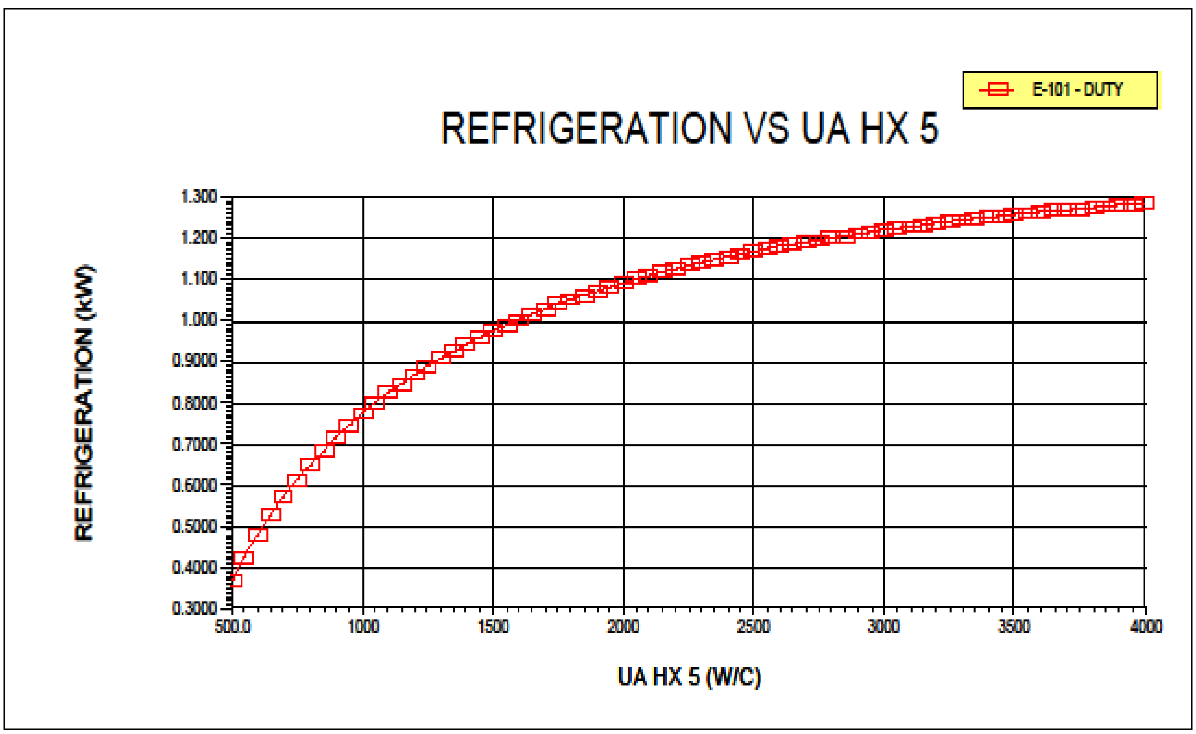

Figure 5.8 Plot of Refrigeration Vs UA of fifth heat exchanger HX 5

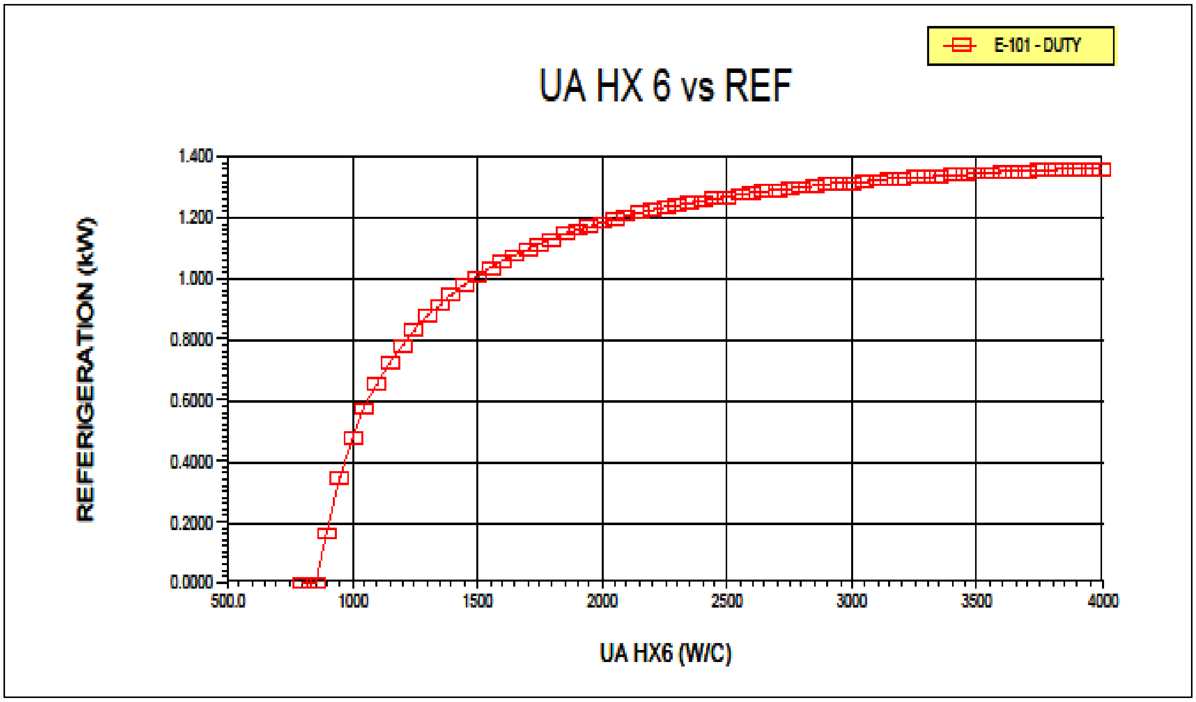

Figure 5.9 Plot of Refrigeration Vs UA of sixth heat exchanger HX 6

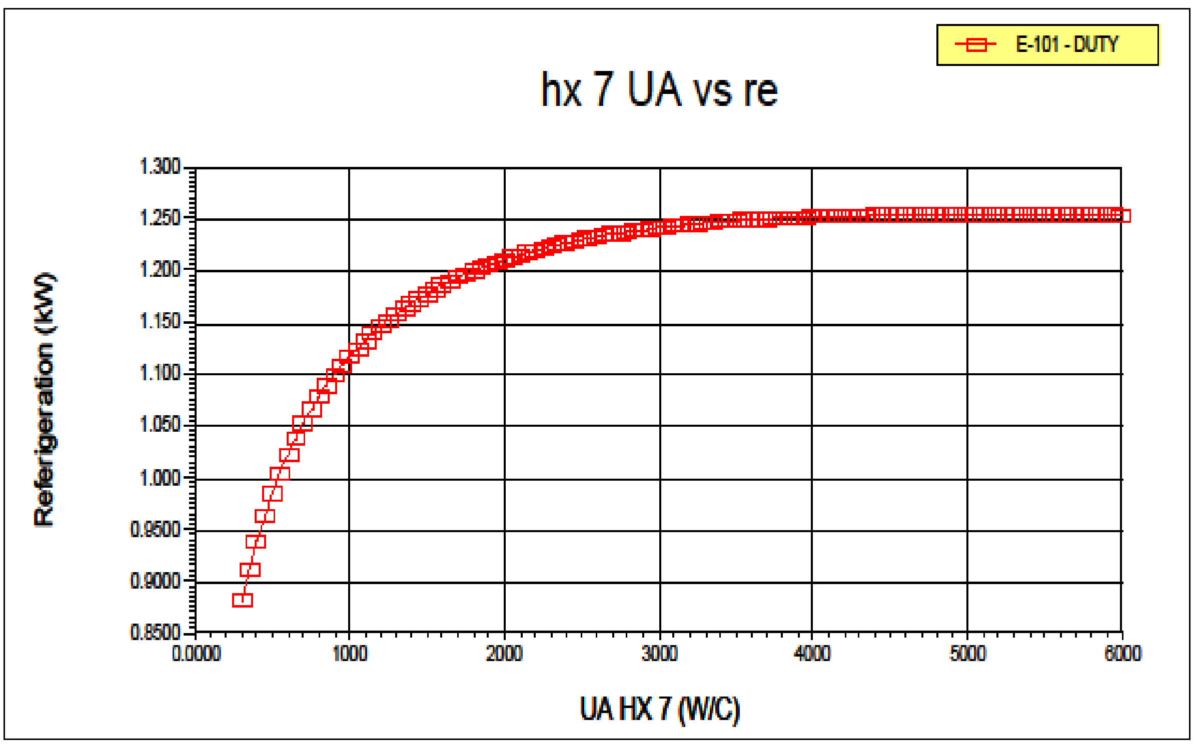

Figure 5.10 Plot of Refrigeration Vs UA of seventh heat exchanger HX 7

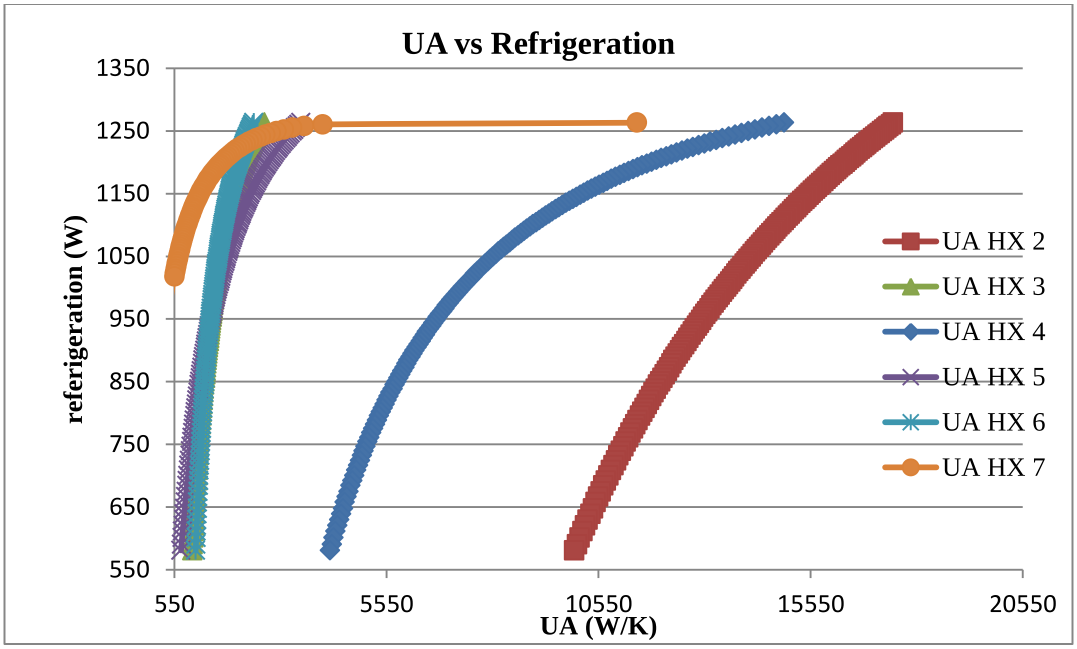

Figure 5.11 Plot of Refrigeration Vs UA of all heat exchangers

- The above graphs can be used to find out the respective theoretical UA values of all the heat exchangers for a particular refrigeration load.

- Also it is clear from the above graphs that there is a saturation value of UA for each exchanger after which it does not affect the refrigeration much i.e. even after increasing UA refrigeration cannot be increased much.

5.5. Temperature approach in heat exchangers

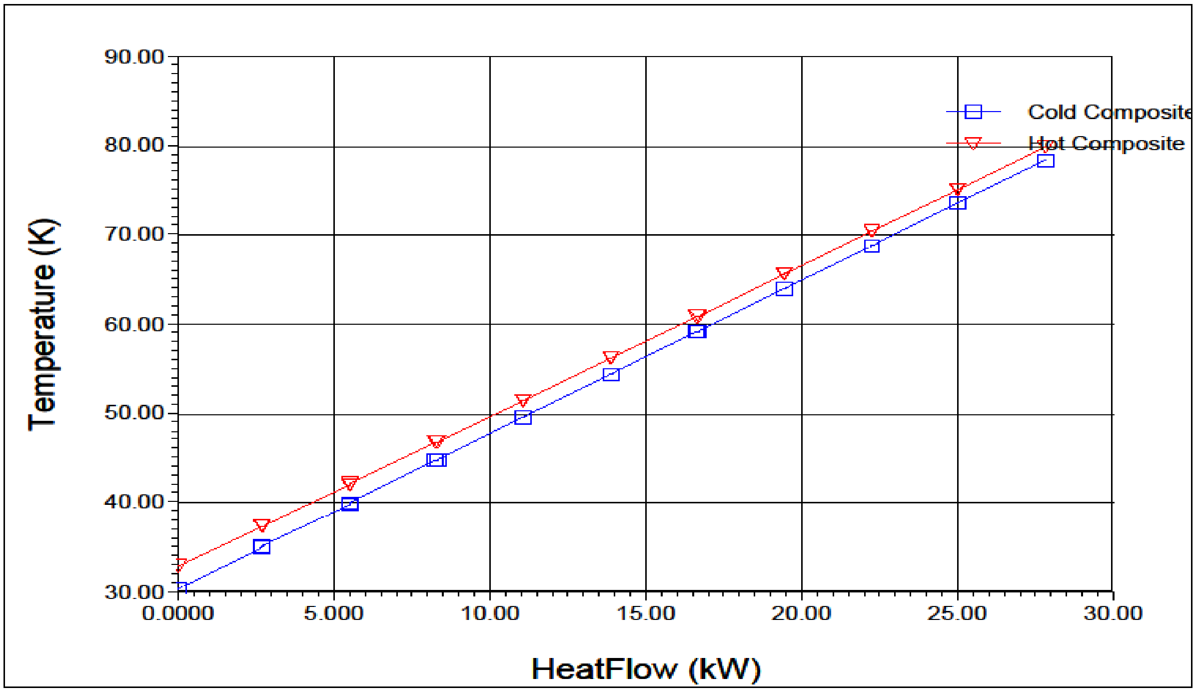

Figure 5.12 Temperature approach in second heat exchanger HX 2

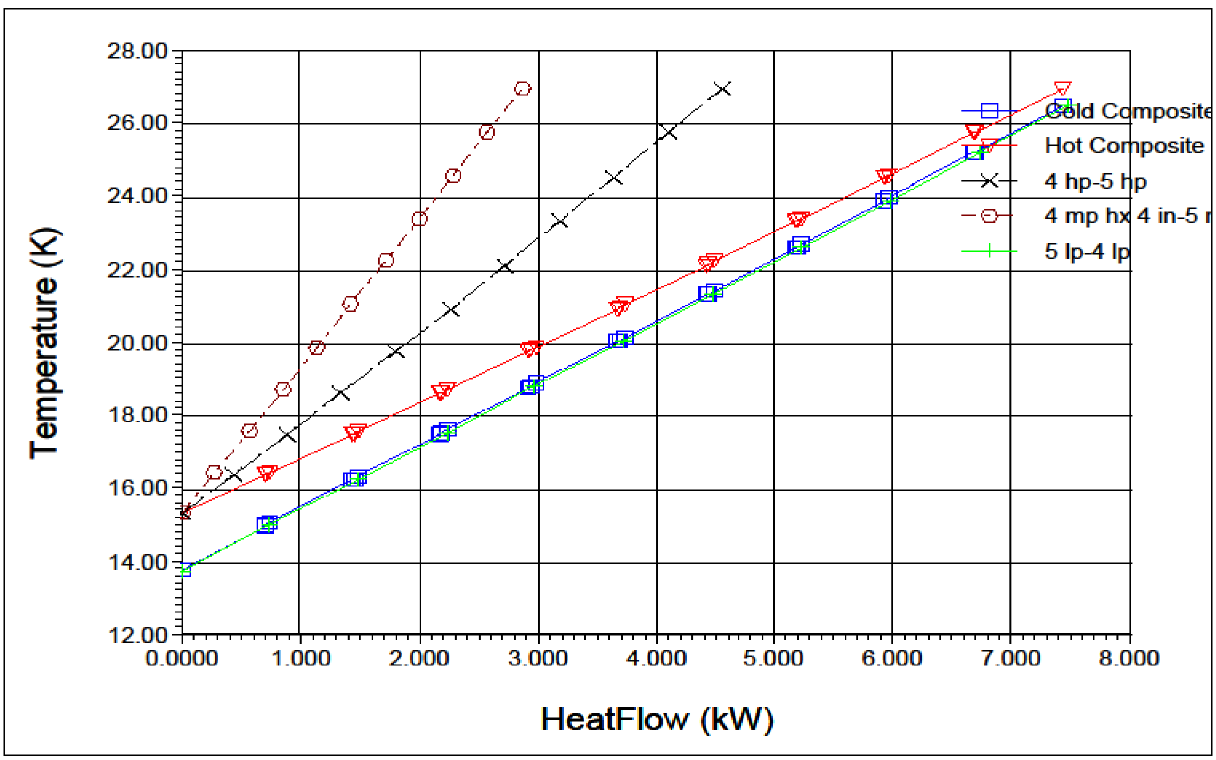

Figure 5.13 Temperature approach in three stream heat exchanger HX 4

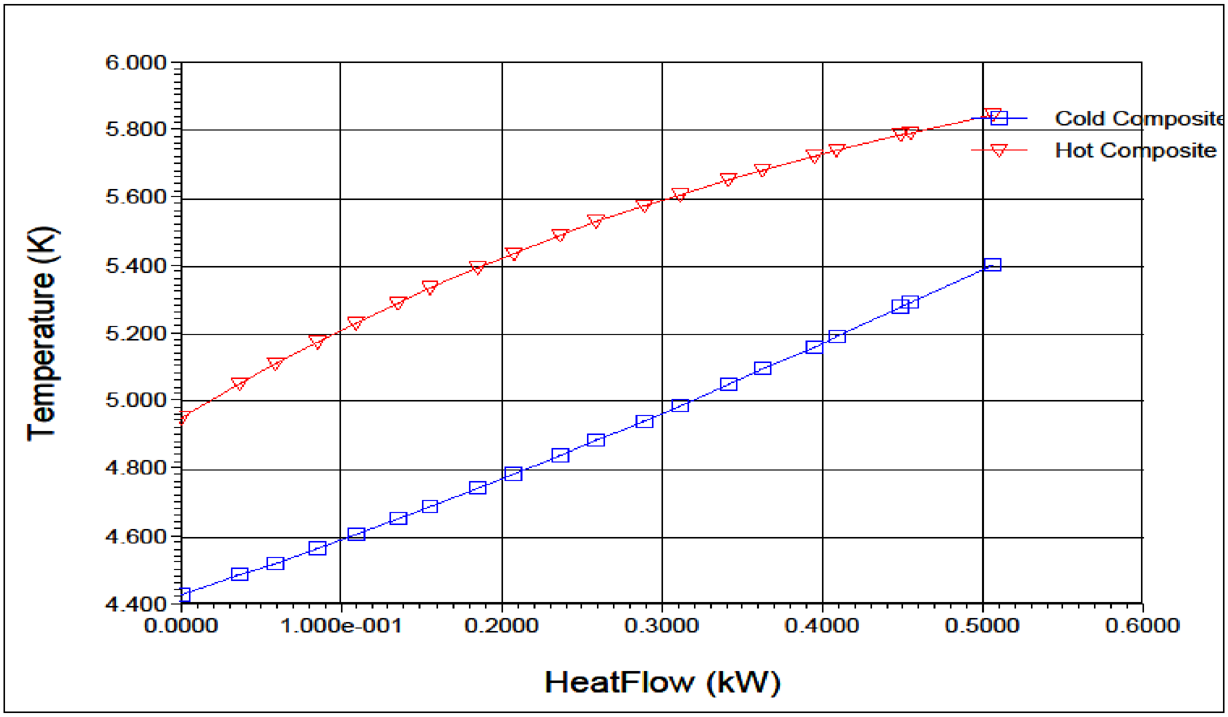

Figure 5.14 Temperature approach in HX 7

The above graphs from aspen HYSYS simulation results, shows how the two streams approaches each other's temperature and the amount of heat flow between the streams for attaining the required temperature.

CHAPTER -6

6. CONCLUSION

1. For the present target capacity of 1 KW, the optimum mass flow under the mentioned constraints is 45 g/s which is 40 % of the total mass flow through cold box.
2. The optimum inlet temperature of turbine A vis- a-vis HX 3 is 33 K.
3. The optimum pressure at JT inlet vis-à-vis Turbine C outlet under the estimated JT inlet temperature range is 4 bar.
4. Refrigeration with third turbine is almost summation of refrigeration without third turbine and refrigeration due to expansion in third turbine.
5. The result obtained from analytical programs match to the results from aspen HYSYS with acceptable deviation.
6. There is a saturation UA value for each heat exchanger after which increase in UA value does not affect the refrigeration capacity of the plant significantly.

CHAPTER -7

7. FUTURE WORK

1. Pressure ratio or the pressure drop in turbine A and turbine C can be optimized.
2. A more detailed method to optimize turbine A inlet temperature can be used. The method used in the present analysis is more concerned with the limiting UA values and the minimum approach in the last heat exchanger. For the same turbine C inlet temperature is varied intuitively in combination with cold outlet temperature of HX 5.
3. Optimization can be done by modifying the present configuration also.

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